4

Diesel Engine Power Plant

4.1. INTRODUCTION

Diesel engine power plants are installed *where supply of coal and water is not available in sufficient quantity* or *where power is to be generated in small quantity* or *where standby sets are required for continuity of supply such as in hospitals, telephone exchanges, radio stations and cinemas.* These plants in the range of 2 to 50 MW capacity are used as *central stations* for supply authorities and works and they are universally adopted to supplement hydro-electric or thermal stations where stand-by generating plants are essential for starting from cold and under emergency conditions.

In several countries, the demand for diesel power plants is increased for electric power generation because of difficulties experienced in construction of new hydraulic plants and enlargement of old hydro-plants. A long term planning is required for the development of thermo and hydro-plants which cannot keep the pace with many times the increased demand by the people and industries.

The diesel units used for electric generation are *more reliable* and *long-lived piece of equipment* compared with other types of plants.

4.2. ADVANTAGES AND DISADVANTAGES OF DIESEL POWER PLANTS

The *advantages* and *disadvantages* of diesel power plants are listed below :

Advantages :

1. Design and installation are very simple.
2. Can respond to varying loads without any difficulty.
3. The standby losses are less.
4. Occupy less space.
5. Can be started and put on load quickly.

6. Require less quantity of water for cooling purposes.
7. Overall capital cost is lesser than that for steam plants.
8. Require less operating and supervising staff as compared to that for steam plants.
9. The efficiency of such plants at part loads does not fall so much as that of a steam plant.
10. The cost of building and civil engineering works is low.
11. Can burn fairly wide range of fuels.
12. These plants can be located very near to the load centres, many times in the heart of the town.
13. No problem of ash handling.
14. The lubrication system is more economical as compared with that of a steam power plant.
15. The diesel power plants are *more efficient than steam power plants* in the range of 150 MW capacity.

Disadvantages :

1. High operating cost.
2. High maintenance and lubrication cost.
3. Diesel units capacity is limited. These cannot be constructed in large size.
4. In a diesel power plant noise is a serious problem.
5. Diesel plants cannot supply overloads continuously whereas steam power plant can work under 25% overload continuously.
6. The diesel power plants are not economical where fuel has to be imported.
7. The life of a diesel power plant is quite small (2 to 5 years or less) as compared to that of a steam power plant (25 to 30 years).

4.3. APPLICATIONS OF DIESEL POWER PLANT

The diesel power plants find wide application in the following fields :

1. Peak load plant
2. Mobile plants
3. Stand-by units
4. Emergency plant
5. Nursery station
6. Starting stations
7. Central stations—where capacity required is small (5 to 10 MW)
8. Industrial concerns where power requirement is small say of the order of 500 kW, diesel power plants become more economical due to their higher overall efficiency.

4.4. SITE SELECTION

The following *factors* should be considered while *selecting the site for a diesel power plant* :

1. Foundation sub-soil condition. The conditions of sub-soil should be such that a foundation at a reasonable depth should be capable of providing a strong support to the engine.

2. Access to the site. The site should be so selected that it is accessible through rail and road.

3. Distance from the load centre. The location of the plant should be near the load centre. This reduces the cost of transmission lines and maintenance cost. The power loss is also minimised.

4. Availability of water. Sufficient quantity of water should be available at the site selected.

5. Fuel transportation. The site selected should be near to the source of fuel supply so that transportation charges are low.

4.5. HEAT ENGINES

Any type of engine or machine which derives heat energy from the combustion of fuel or any other sources and converts this energy into mechanical work is termed as a ***heat engine.***

Heat engines may be classified into two main classes as follows :

1. External Combustion Engines.
2. Internal Combustion Engines.

1. External combustion engine (E.C. engines). In this case, *combustion of fuel takes place outside the cylinder* as in case of *steam engines* where the heat of combustion is employed to generate steam which is used to move a piston in a cylinder. Other examples of external combustion engines are *hot air engines, steam turbine* and *closed cycle gas turbine.* These engines are generally used for driving locomotives, ships, generation of electric power etc.

2. Internal combustion engines (I.C. engines). In this case, *combustion of the fuel with oxygen of the air occurs within the cylinder of the engines.* The internal combustion engines group includes engines employing mixtures of combustible gases and air, known as *gas engines,* those using lighter liquid fuel or spirit known as *petrol engines* and those using heavier liquid fuels, known as *oil compression ignition* or *diesel engines.*

4.6. CLASSIFICATION OF I.C. ENGINES

Internal combustion engines may be *classified* as given below :

1. According to cycle of operation

(*i*) Two stroke cycle engines

(*ii*) Four stroke cycle engines.

2. According to cycle of combustion

(*i*) Otto cycle engine (combustion at constant volume)

(*ii*) Diesel cycle engine (combustion at constant pressure)

(*iii*) Dual-combustion or Semi-Diesel cycle engine (combustion partly at constant volume and partly at constant pressure).

3. According to arrangement of cylinder

(*i*) Horizontal engine (*ii*) Vertical engine

(*iii*) V-type engine (*iv*) Radial engine etc.

4. According to their uses :

(*i*) Stationary engine (*ii*) Portable engine

(*iii*) Marine engine (*iv*) Automobile engine

(*v*) Aero engine etc.

5. According to the fuel employed and the method of fuel supply to the engine cylinder

(*i*) Oil engine (*ii*) Petrol engine

(*iii*) Gas engine (*iv*) Kerosene engine etc.

(*v*) Carburettor, hot bulb, solid injection and air injection engine.

6. According to the speed of the engine

(*i*) Low speed engine (*ii*) Medium speed engine

(*iii*) High speed engine.

7. **According to method of ignition**

(*i*) Spark ignition S.I. engine (*ii*) Compression ignition C.I. engine.

8. **According to method of cooling the cylinder**

(*i*) Air-cooled engine (*ii*) Water-cooled engine.

9. **According to method of governing**

(*i*) Hit and miss governed engine (*ii*) Quality governed engine

(*iii*) Quantity governed engine.

10. **According to valve arrangement**

(*i*) Over head valve engine (*ii*) L-head type engine

(*iii*) T-head type engine (*iv*) F-head type engine.

11. **According to number of cylinders**

(*i*) Single cylinder engine (*ii*) Multi-cylinder engine.

4.7. DIFFERENT PARTS OF I.C. ENGINES

Fig. 4.1 shows the cross-section of an *air-cooled I.C. engine* with principle parts.

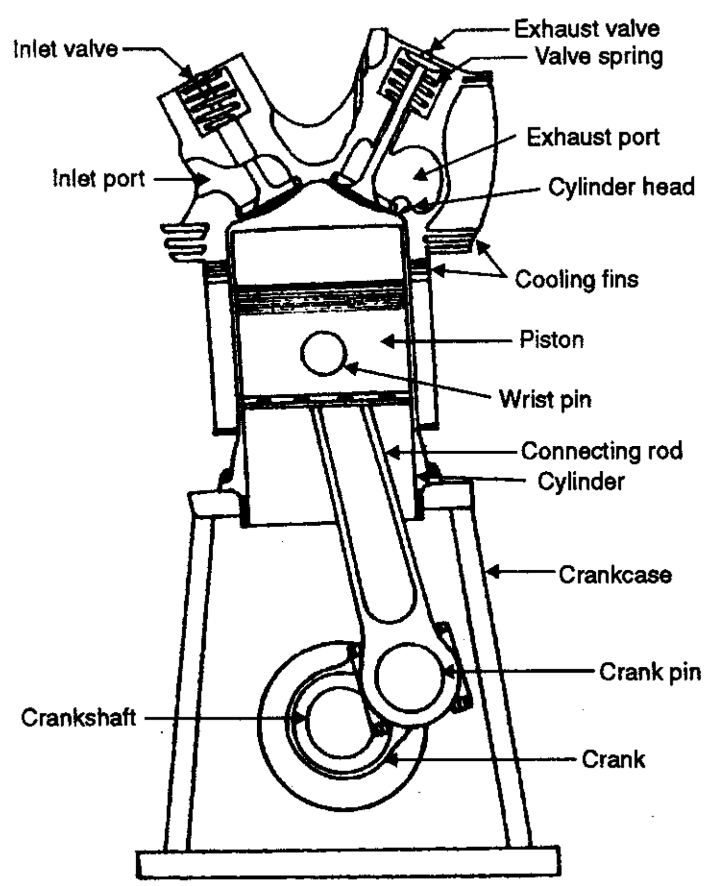


Fig. 4.1. Air-cooled I.C. engine.

A. Parts common to both petrol and diesel engines

1. Cylinder
2. Cylinder head
3. Piston
4. Piston rings
5. Gudgeon pin
6. Connecting rod
7. Crankshaft
8. Crank
9. Engine bearing
10. Crankcase
11. Fly wheel
12. Governor
13. Valves and valve operating mechanisms.

B. Main parts for petrol engines only

1. Spark plugs
2. Carburettor
3. Fuel pump.

C. Main parts of diesel engines only

1. Fuel pump
2. Fuel injector or atomiser.

4.8. TERMS CONNECTED WITH I.C. ENGINES

Refer Fig. 4.2.

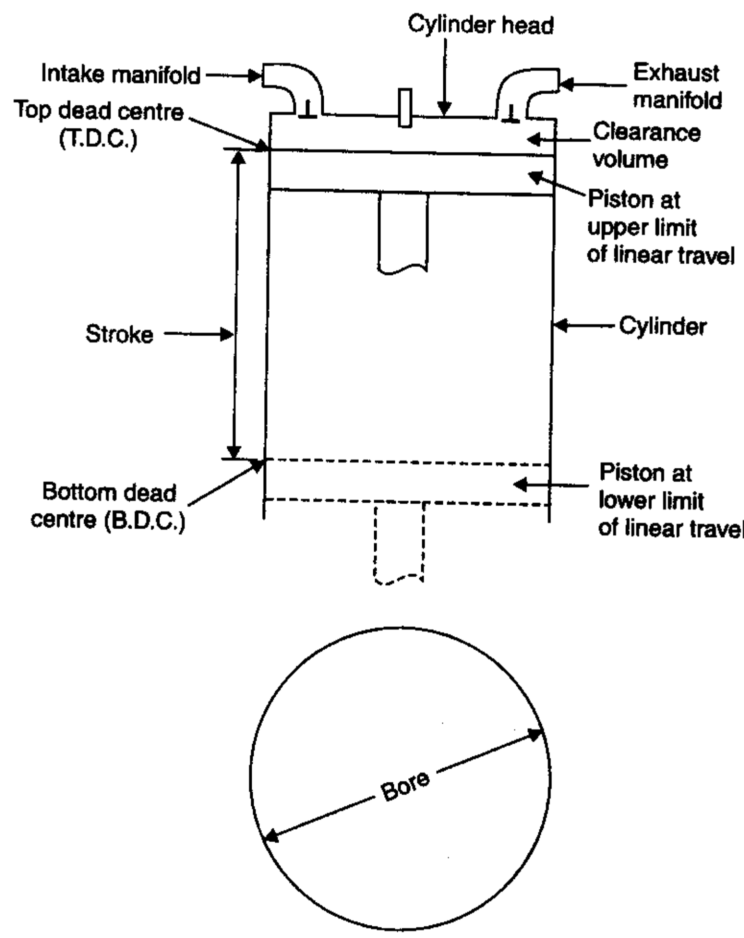


Fig. 4.2. Terms relating I.C. engines.

Bore. *The inside diameter of the cylinder is called* ***bore.***

Stroke. As the piston reciprocates inside the engine cylinder, it has got limiting upper and lower positions beyond which it cannot move and reversal of motion takes place at these limiting positions.

The linear distance along the cylinder axis between two limiting positions, is called ***stroke.***

Top Dead Centre (T.D.C.). *The top most position of the piston towards cover end side of the cylinder is called* ***top dead centre.*** In case of horizontal engines, this is known as *"inner dead centre."*

Bottom Dead Centre (B.D.C.). *The lowest position of the piston towards the crank end side of the cylinder is called* ***bottom dead centre.*** In case of horizontal engines it is called *"outer dead centre."*

Clearance volume. *The volume contained in the cylinder above the top of the piston, when the piston is at top dead centre, is called the* ***clearance volume.***

Swept volume. *The volume swept through by the piston in moving between top dead centre and bottom dead centre, is, called* ***swept volume*** *or* ***piston displacement.*** Thus, when piston is at bottom dead centre, total volume = swept volume + clearance volume.

Compression ratio. *It is the ratio of total cylinder volume to clearance volume.*

Refer Fig. 4.2. Compression ratio (r) is given by

$$r = \frac{V_s + V_c}{V_c}$$

where V_s = Swept volume, and

V_c = Clearance volume.

The compression ratio varies from 5 : 1 to 9 : 1 in *petrol engines* and from 14 : 1 to 22 : 1 in *diesel engines.*

Piston speed. *The average speed of the piston is called* ***piston speed.***

Piston speed = $2\,LN$

where, L = Length of the stroke, and

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

4.9. FOUR STROKE CYCLE DIESEL ENGINES

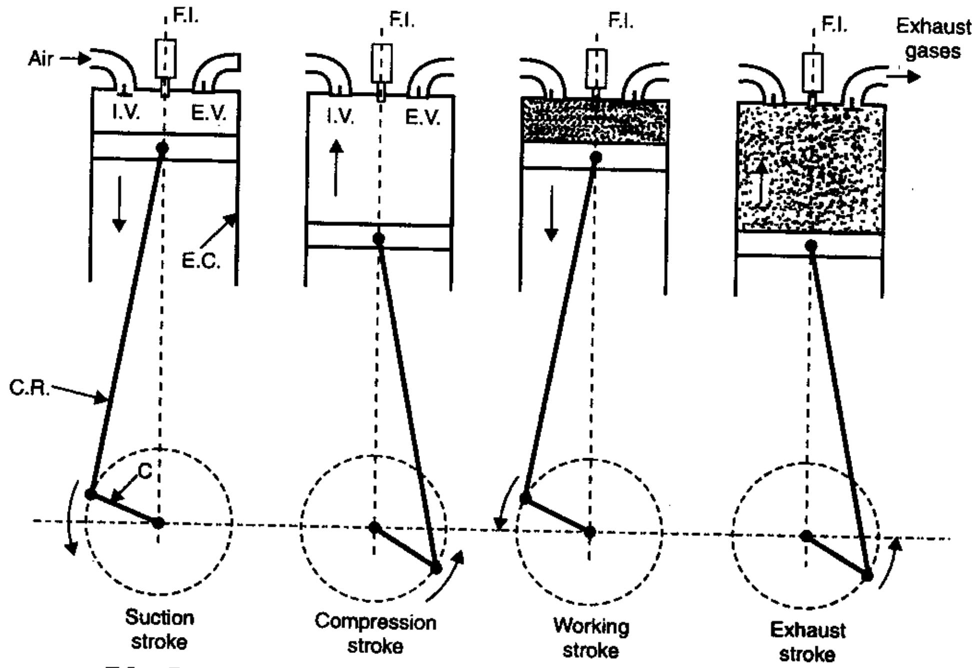
This types of engine comprises of the following *four strokes* :

1. Suction stroke. With the movement of the piston from T.D.C. to B.D.C. during this stroke, the inlet valve opens and the air at atmospheric pressure is drawn inside the engine cylinder ; the exhaust valve however remains closed. This operation is represented by the line 5-1 (Fig. 4.4).

2. Compression stroke. The air drawn at atmospheric pressure during the suction stroke is compressed to high pressure and temperature (to the value of 35 bar and 600°C respectively) as the piston moves from B.D.C. to T.D.C. This operation is represented by 1-2 (Fig. 4.4). Both the inlet and exhaust valves do not open during any part of this stroke.

3. Expansion or working stroke. As the piston starts moving from T.D.C. a metred quantity of fuel is injected into the hot compressed air in fine sprays by the fuel injector and it (fuel) starts burning at constant pressure shown by the line 2-3. At the point 3 *fuel supply is cut off.* The fuel is injected at the end of compression stroke but in actual practice the ignition of the fuel starts before the end of the compression stroke. The hot gases of the cylinder expand adiabatically to point 4, thus doing work on the piston. The expansion is shown by 3-4 (Fig. 4.4).

4. Exhaust stroke. The piston moves from the B.D.C. to T.D.C. and the exhaust gases escape to the atmosphere through the exhaust valve. When the piston reaches the T.D.C. the exhaust valve closes and the cycle is completed. This stroke is represented by the line 1-5 (Fig. 4.4).



F.I. = Fuel injector | I.V. = Inlet valve | E.V. = Exhaust valve
E.C. = Engine cylinder | C.R. = Connecting rod | C = Crank

Fig. 4.3. Four stroke diesel cycle engine.

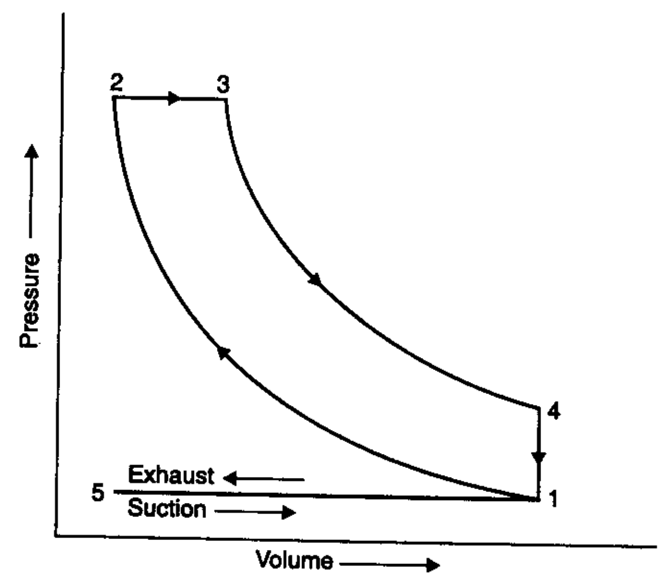


Fig. 4.4. Theoretical p-V diagram of a four stroke diesel cycle.

Fig. 4.5 shows the actual indicator diagram for a four-stroke Diesel cycle engine. It may be noted that line 5-1 is below the atmospheric pressure line. This is due to the fact that owing to the restricted area of the inlet passages the entering air can't cope with the speed of the piston. The exhaust line 4-5 is slightly above the atmospheric line. This is because of the restricted exhaust passages which do not allow the exhaust gases to leave the engine cylinder quickly.

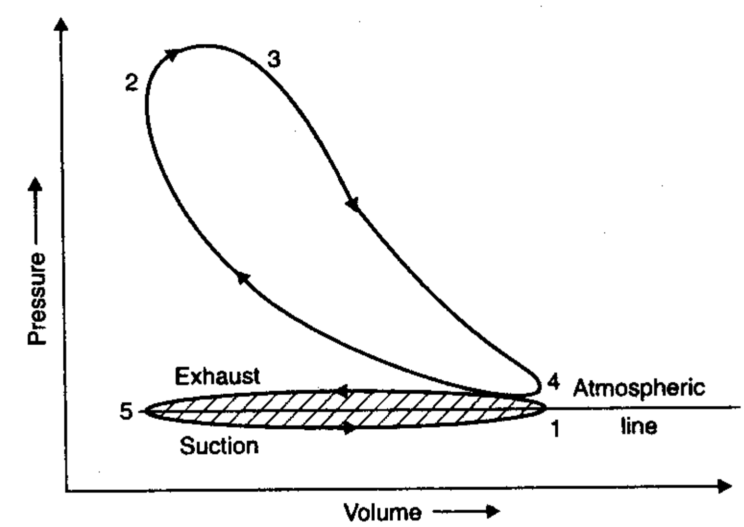
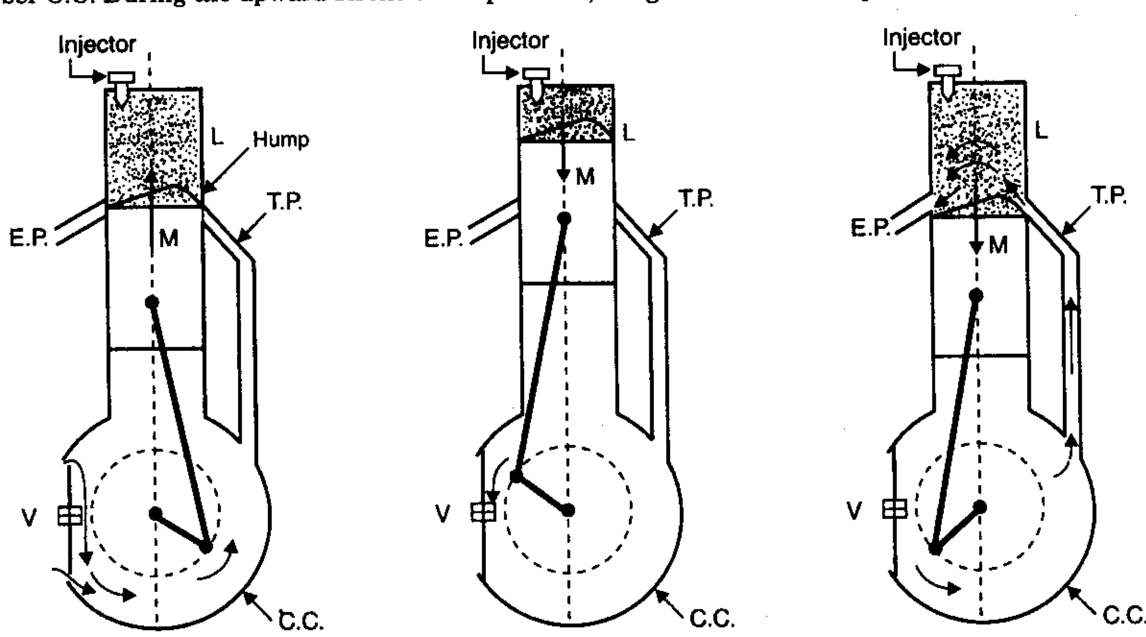


Fig. 4.5. Actual p-V diagram of four stroke diesel cycle.

The loop of area 4-5-1 is called *negative loop* ; it gives the *pumping loss* due to admission of air and removal of exhaust gases. The area 1234 is the total or gross work obtained from the piston and net work can be obtained by subtracting area 451 from area 1234.

4.10. TWO STROKE CYCLE DIESEL ENGINES

Fig. 4.6 shows two stroke diesel engine. The cylinder L is connected to the closed crank chamber C.C. During the upward stroke of the piston M, the gases in L are compressed and at the same



L = Cylinder ; M = Piston ; E.P. = Exhaust port ; T.P. = Transfer port ; C.C. = Crank chamber ; V = Valve

Fig. 4.6. Two stroke cycle engine.

time *fresh air* enters the crankchamber through the valve V. When the piston moves downward, V closes and air in the crank chamber is compressed. Refer Fig. 4.6 *(i)*. The piston is moving upwards and is compressing the air which has previously been supplied to L and before it (piston) reaches the T.D.C. (top dead centre) the fuel injector supplies fuel to the engine cylinder. Ignition of fuel takes place due to high temperature of air and gases are produced. These gases expand and the piston then travels downward [Fig. 4.6 *(ii)*] and near the end of this stroke the piston uncovers the exhaust port (E.P.) and the burnt exhaust gases escape through this port [Fig. 4.6 *(iii)*]. The transfer port (T.P.) then is uncovered immediately, and the compressed air from the crank chamber flows into the cylinder and is deflected upwards by the hump provided on the head of the piston. It may be noted that the incoming air helps the removal of gases from the engine-cylinder. The piston then again starts moving from B.D.C. to T.D.C. and the air gets compressed when exhaust port (E.P.) and transfer port (T.P.) are covered by the piston ; thus the cycle is repeated.

4.11. COMPARISON OF FOUR STROKE AND TWO STROKE CYCLE ENGINES

	Four stroke cycle engines	*Two stroke cycle engines*
1.	The cycle is completed in four strokes of the piston or in two revolutions of the crankshaft. Thus one power stroke is obtained in every two revolutions of the crankshaft.	The cycle is completed in two strokes of the piston or in one revolution of the crankshaft. Thus one power stroke is obtained in each revolution of the crankshaft.
2.	Because of the above turning-movement is not so uniform and hence heavier flywheel is needed.	More uniform turning movement and hence lighter flywheel is needed.
3.	Again because of one power stroke for two revolutions, power produced for same size of engine is small or for the same power the engine is heavy and bulky.	Because of one power stroke for one revolution, power produced for same size of engine in more (theoretically twice, actually about 1.8 times) or for the same power the engine is light and compact.
4.	Because of one power stroke in two revolutions lesser cooling and lubrication requirements. Lesser rate of wear and tear.	Because of one power stroke in one revolution greater cooling and lubrication requirement. Great rate of wear and tear.
5.	The four stroke engine contains valve and valve mechanism.	Two stroke engines have no valves but only ports (some two stroke engines are fitted with conventional exhaust valves).
6.	Because of the heavy weight and complication of valve mechanism, higher is the initial cost.	Because of light weight and simplicity due to absence of valve mechanism, cheaper in initial cost.
7.	Volumetric efficiency more due to more time of induction.	Volumetric efficiency less due to lesser time for induction.
8.	Thermal efficiency higher, part load efficiency better than two stroke cycle engine.	Thermal efficiency lower, part load efficiency lesser than four stroke cycle engine.
9.	Used where efficiency is important ; in *cars, buses, trucks, tractors, industrial engines, aeroplane, power generators etc.*	In two stroke petrol engine some fuel is exhausted during scavenging. Used where (a) low cost, and (b) compactness and light weight important. Two stroke (air cooled) petrol engines used in very small sizes only, *lawn movers, scooters, motor cycles* (lubricating oil mixed with petrol). Two stroke diesel engines used in very large sizes, more than 60 cm bore, for *ship propulsion* because of low weight and compactness.

4.12. COMPARISON BETWEEN A PETROL ENGINE AND A DIESEL ENGINE

	Petrol engine	*Diesel engine*
1.	Air petrol mixture is sucked in the engine cylinder during suction stroke.	Only air is sucked during suction stroke.
2.	Spark plug is used.	Employs an injector.
3.	Power is produced by spark ignition.	Power is produced by compression ignition.
4.	Thermal efficiency up to 25%.	Thermal efficiency up to 40%.
5.	Occupies less space.	Occopies more space.
6.	More running cost.	Less running cost.
7.	Light in weight.	Heavy in weight.
8.	Fuel (Petrol) costlier.	Fuel (Diesel) cheaper.
9.	Petrol being volatile is dangerous.	Diesel is not dangerous as it is non-volatile.
10.	Pre-ignition possible.	Pre-ignition not possible.
11.	Works on Otto cycle.	Works on diesel cycle.
12.	Less dependable.	More dependable.
13.	Used in *cars* and *motor cycles*.	Used in heavy duty vehicles like *trucks*, *buses* and *heavy machinery*.

4.13. ESSENTIAL COMPONENTS OF A DIESEL POWER PLANT

Refer Fig. 4.7. The essential components of a diesel power plant are listed and discussed below :

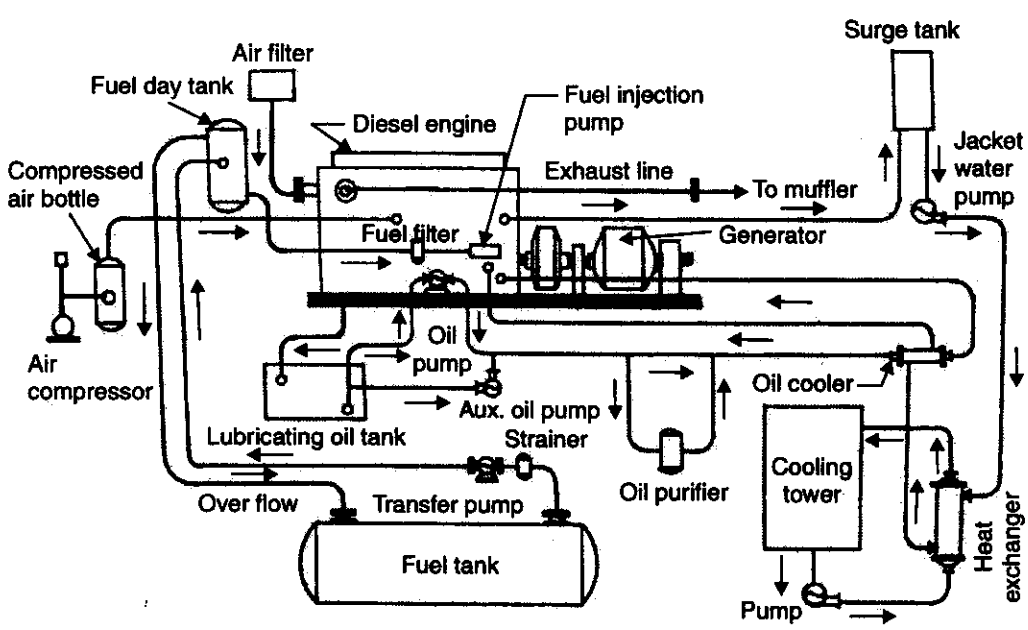


Fig. 4.7. Schematic arrangement of a diesel power plant.

1. Engine
2. Air intake system
3. Exhaust system
4. Fuel system

5. Cooling system
6. Lubrication system
7. Engine starting system
8. Governing system.

4.13.1. Engine

This is the main component of the plant which develops the required power. It is generally directly coupled to the generator as shown in Fig. 4.7.

4.13.2. Air Intake system

The air intake system conveys fresh air through pipes or ducts to : *(i)* Air intake manifold of four stroke engines *(ii)* The scavenging pump inlet of a two stroke engine and *(iii)* The supercharger inlet of a supercharged engine.

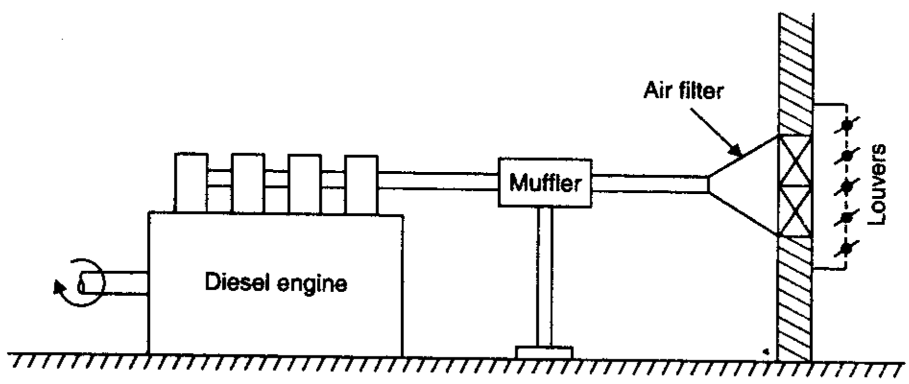


Fig. 4.8. Air intake system.

The air system begins with an intake located outside the building provided with a *filter* to catch dirt which would otherwise cause excessive wear in the engine. The filters may be of *dry or oil bath*. Electrostatic precipitator filters can also be used. *Oil impingement type of filter* consists of a frame filled with metal shavings which are coated with a special oil so that the air in passing through the frame and being broken up into a number of small filaments comes into contact with the oil whose property is to sieze and hold any dust particles being carried by the air. The *dry type* of filter is made of cloth, felt, glass wool etc. In case of *oil bath type of filter* the air is swept over or through a pool of oil so that the particles of dust become coated. Light weight steel pipe is the material for intake ducts. In some cases, the engine noise may be transmitted back through the air intake system to the outside air. In such cases a silencer is provided between the engine and the intake.

Following *precautions* should be taken while constructing a suitable air intake system :

1. Air intakes may not be located inside the engine room.
2. Air should not be taken from a confined space otherwise air pulsations can cause serious vibration problems.
3. The air-intake line used should neither have too small a diameter nor should be too long, otherwise there may crop up engine starvation problem.
4. Air intake filters may not be located close to the roof of the engine room otherwise pulsating air flow through the filters can cause serious vibrations of the roof.
5. Air intake filters should not be located in an inaccessible location.

4.13.3. Exhaust System

Refer Fig. 4.9. The purpose of the exhaust system is to discharge the engine exhaust to the atmosphere outside the building. The exhaust manifold connects the engine cylinder exhausts outlets to the exhaust pipe which is provided with a muffler to reduce pressure in the exhaust line and eliminate most of the noise which may result if gases are discharged directly into the atmosphere.

The exhaust pipe leading out of the building should be short in length with minimum number of bends and should have one or two flexible tubing sections which take up the effects of expansion, and isolate the system from the engine vibration. Every engine should be provided with its independent exhaust system.

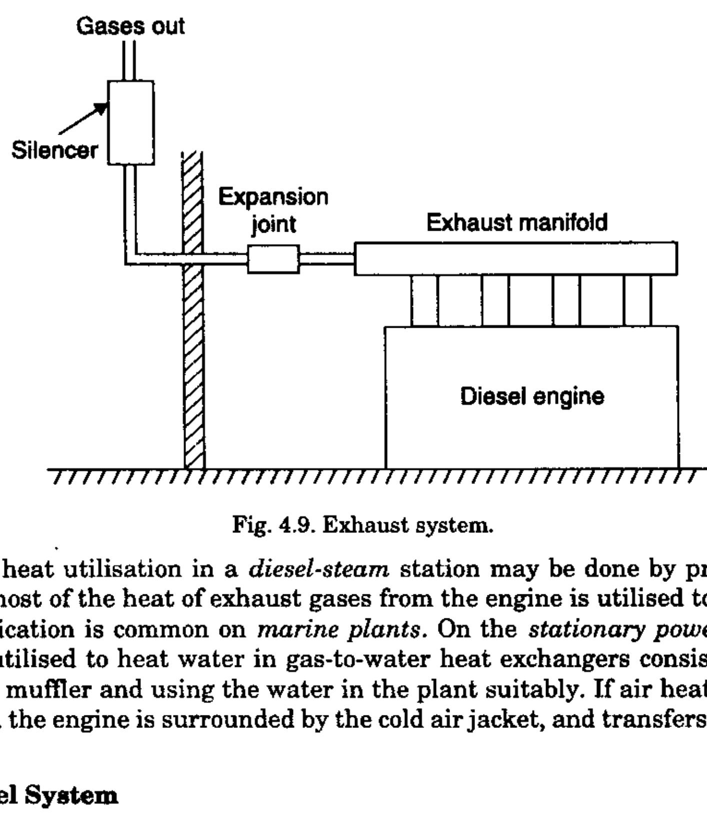


Fig. 4.9. Exhaust system.

The waste heat utilisation in a *diesel-steam* station may be done by providing waste-heat boilers in which most of the heat of exhaust gases from the engine is utilised to raise low pressure steam. Such application is common on *marine plants*. On the *stationary power plant* the heat of exhaust may be utilised to heat water in gas-to-water heat exchangers consisting of a water coil placed in exhaust muffler and using the water in the plant suitably. If air heating is required, the exhaust pipe from the engine is surrounded by the cold air jacket, and transfers the heat of exhaust gases to the air.

4.13.4. Fuel System

Refer Fig. 4.10.

The fuel oil may be delivered at the plant site by trucks, railroad tank cars or barge and tankers. From tank car or truck the delivery is through the unloading facility to main storage tanks and then by transfer pumps to small service storage tanks known as *engine day tanks*. Large storage capacity allows purchasing fuel when prices are low. The main flow is made workable and practical by arranging the piping equipment with the necessary heaters, by passes, shut-offs, drain lines, relief valves, strainers and filters, flow meters and temperature indicators. The actual flow plans depend on type of fuel, engine equipment, size of the plant etc. The tanks should contain manholes for internal access and repair, fill lines to receive oil, vent lines to discharge vapours, overflow return lines for controlling oil flow and a suction line to withdraw oil. *Coils heated by hot water or steam reduce oil viscosity to lower pumping power needs.*

The minimum storage capacity of at least a month's requirement of oil should be kept in bulk, but where advantage of seasonal fluctuations in cost of oil is to be availed, it may be necessary to provide storage for a few month's requirements. *Day tanks* supply the daily fuel need of engines and may contain a minimum of about 8 hours of oil requirement of the engines. These tanks are usually placed high so that oil may flow to engines under gravity.

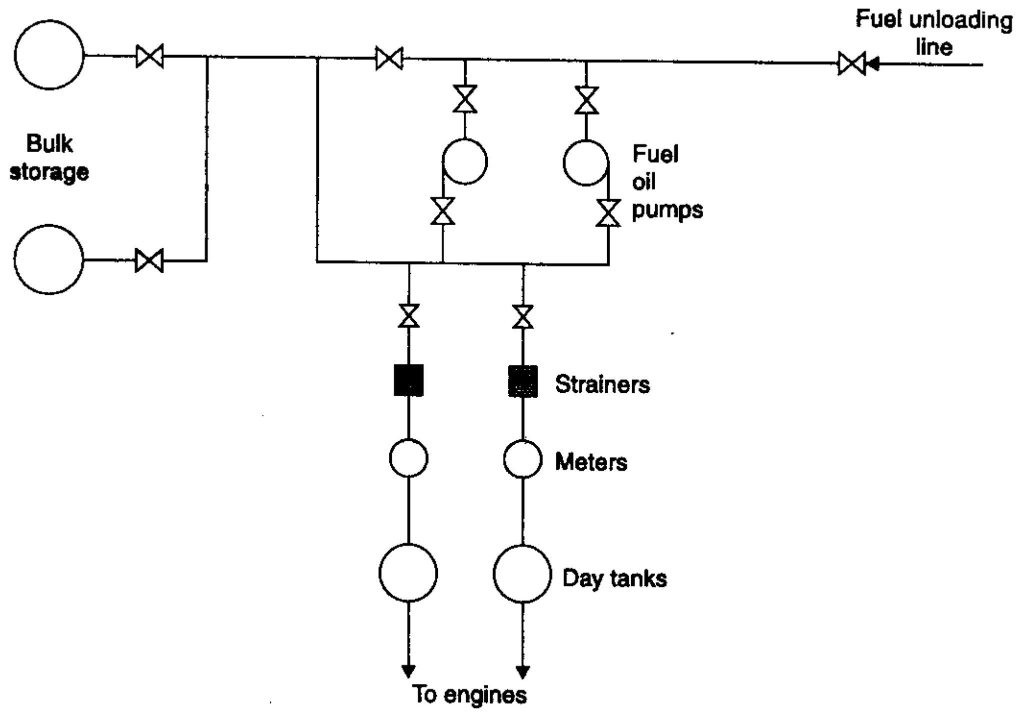


Fig. 4.10. System of fuel storage for a diesel power plant.

For satisfactory operation of a fuel oil supply system the following points should be taken care of :

1. There should be provisions for cleanliness and for changing over of lines during emergencies.
2. In all suction lines the pipe joints should be made tight.
3. Before being covered, all oil lines should be put under air pressure and the joints tested with soap solution. Small air leaks into the line can be the source of exasperating operating difficulties and are hard to remedy once the plant is in operation.
4. The piping between filter and the engine should be thoroughly oil flushed before being first placed in service.
5. Considerable importance should be given for cleanliness in handling bulk fuel oil. Dirt particles will ruin the fine lap of injection pumps or plug the injection nozzle orifices. So *high-grade filters* are of paramount importance to the diesel oil supply system.

Fuel Injection system

The mechanical heart of the Diesel engine is the *fuel injection system*. The engine can perform no better than its fuel injection system. A very small quantity of fuel must be measured out, injected, atomised, and mixed with combustion air. The mixing problem becomes more difficult—the larger the cylinder and faster the rotative speed. Fortunately the high-speed engines are the small-bore automotive types ; however, special combustion arrangements such as precombustion chambers, air cells, etc. are necessary to secure good mixing. *Engines driving electrical generators have lower speeds and simple combustion chambers.*

Functions of a fuel injection system :

1. Filter the fuel.
2. Meter or measure the correct-quantity of fuel to be injected.
3. Time the fuel injection.

4. Control the rate of fuel injection.
5. Automise or break up the fuel to fine particles.
6. Properly distribute the fuel in the combustion chamber.

The injection systems are manufactured with great accuracy, especially the parts that actually meter and inject the fuel. Some of the tolerances between the moving parts are very small of the order of *one micron*. Such closely fitting parts require special attention during manufacture and hence the injection systems are *costly*.

Types of fuel injection systems

The following fuel injection systems are commonly used in diesel power station :

1. Common-rail injection system
2. Individual pump injection system
3. Distributor

Atomisation of fuel oil has been secured by (*i*) *air blast* and (*ii*) *pressure spray*. Early diesel engines used air fuel injection at about 70 bar. This is sufficient not only to inject the oil, but also to atomise it for a rapid and thorough combustion. The expense of providing an air compressor and tank led to the development of "*solid*" injection, using a liquid pressure of between 100 and 200 bar which is sufficiently high to atomise the oil it forces through spray nozzles. Great advances have been made in the field of solid injection of the fuel through research and progress in fuel pump, spray nozzles, and combustion chamber design.

1. *Common-rail injection system :*

Two types of common-rail injection systems are shown in Figs. 4.11 and 4.12 respectively.

Refer Fig. 4.11. A single pump supplies high-pressure fuel to header, a relief valve holds pressure constant. The control wedge adjusts the lift of mechanical operated valve to set amount and time of injection.

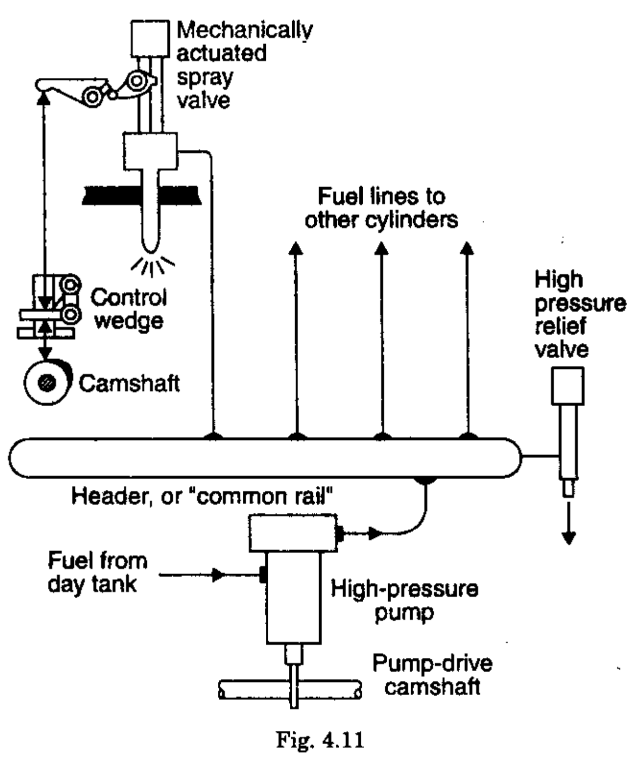


Fig. 4.11

Refer Fig. 4.12. Controlled-pressure system has pump which maintains set head pressure. Pressure relief and timing valves regulate injection time and amount. Spring loaded spray valve acs merely as a check.

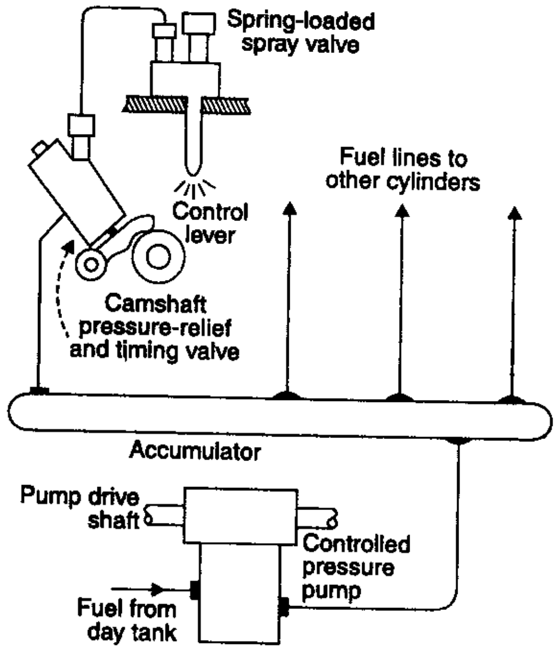


Fig. 4.12

2. *Individual pump injection system :*

Refer Fig. 4.13. In this system an individual pump or pump cylinder connects directly to each fuel nozzle. Pump meters charge and control injection timing. Nozzles contain a delivery valve actuated by the fuel-oil pressure.

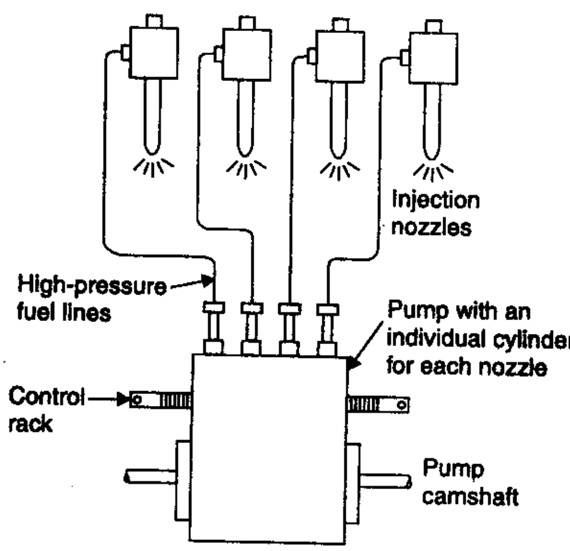


Fig. 4.13. Individual pump injection system.

3. *Distributor system :*

Refer Fig. 4.14. In this system, the fuel is metered at a central point ; a pump pressurises, meters the fuel and times the injection. From here, the fuel is distributed to cylinders in correct firing order by cam operated poppet valves which open to admit fuel to the nozzles.

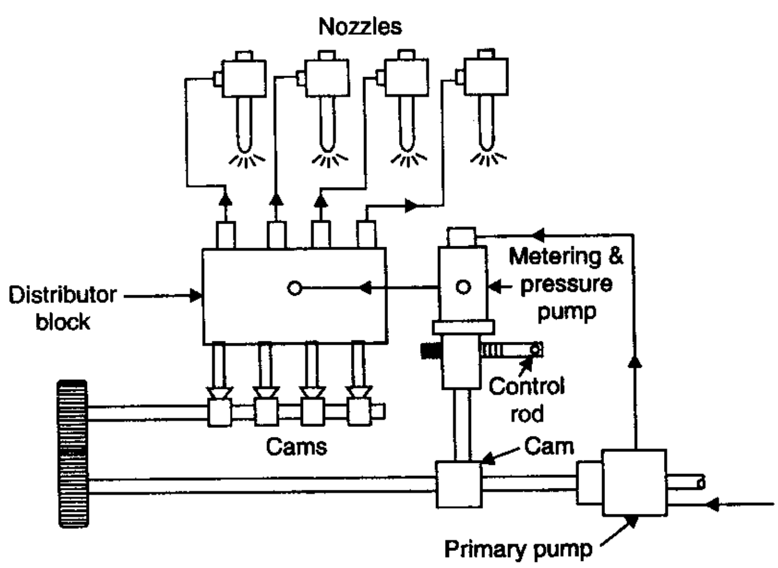


Fig. 4.14. Distributor system.

Fuel Pump

Refer Fig. 4.15. L is the plunger which is driven by a cam and tappet mechanism at the bottom (not shown in Fig. 4.15), B is the barrel in which the plunger reciprocates. There is the rectangular vertical groove in the plunger which extends from top to another helical groove. V is the delivery valve which lifts off its seat under the liquid fuel pressure and against the spring force (S). The fuel pump is connected to fuel atomiser through the passage P. SP and Y are the spill and supply ports respectively. When the plunger is at its bottom stroke the ports SP and Y are uncovered (as shown in Fig. 4.15) oil from low pressure pump (not shown) after being filtered is forced into the barrel. When the plunger moves up due to cam and tappet mechanism, a stage reaches when both the ports SP and Y are closed and with the further upward movement of the plunger the fuel gets compressed. The high pressure thus developed lifts the delivery valve off its seats and fuel flows to atomiser through the passage P. With further rise of the plunger, at a certain moment, the port SP is connected to the fuel in the upper part of the plunger through the rectangular vertical groove by the helical groove, as a result of which a sudden drop in pressure occurs and the delivery valve falls back and occupies its seat against the spring force. The plunger is rotated by the rack R which is moved in or out by the governor. By changing the angular position of the helical groove (by rotating the plunger) of the plunger relative to the supply port, the length of stroke during which the oil is delivered can be varied and thereby quantity of fuel delivered to the engine is also varied accordingly.

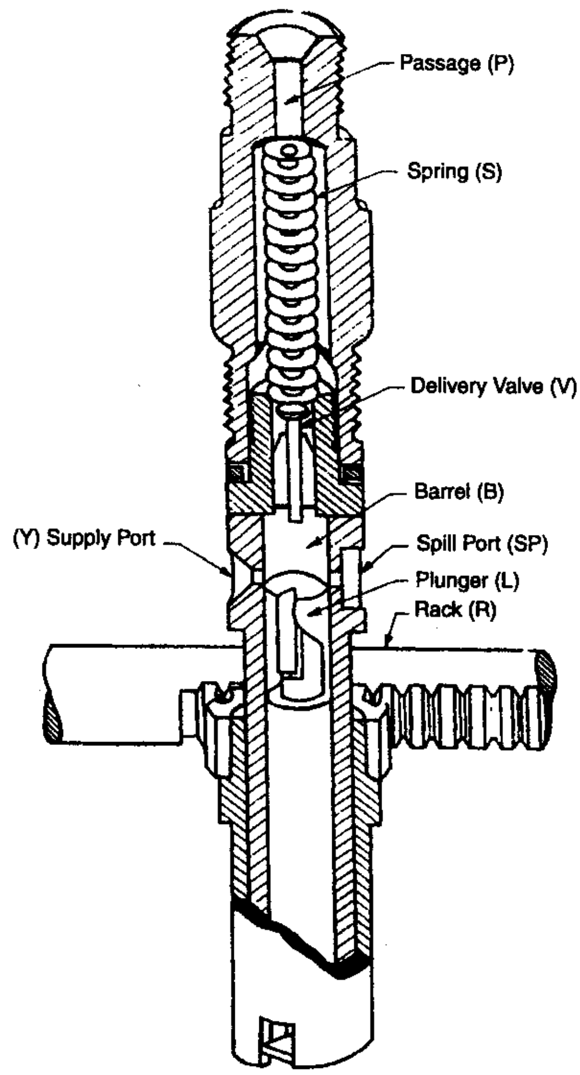


Fig. 4.15. Fuel pump.

Fuel atomiser or injector

Refer Fig. 4.16. It consists of a nozzle valve (*NV*) fitted in the nozzle body (*NB*). The nozzle valve is held on its seat by a spring '*S*' which exerts pressure through the spindle *E*. '*AS*' is the adjusting screw by which the nozzle valve lift can be adjusted. Usually the nozzle valve is set to lift at 135 to 170 bar pressure. *FP* is the feeling pin which indicates whether valve is working properly or not. The fuel under pressure from the fuel pump enters the injector through the passages *B* and *C* and lifts the nozzle valve. The fuel travels down nozzle *N* and injected into the engine cylinder in the form of fine spray. Then the pressure of the oil falls, the nozzle valve occupies its seat under the spring force and fuel supply is cut off. Any leakage of fuel accumulated above the valve is led to the fuel tank through the passage *A*. The leakage occurs when the nozzle valve is *worn out*.

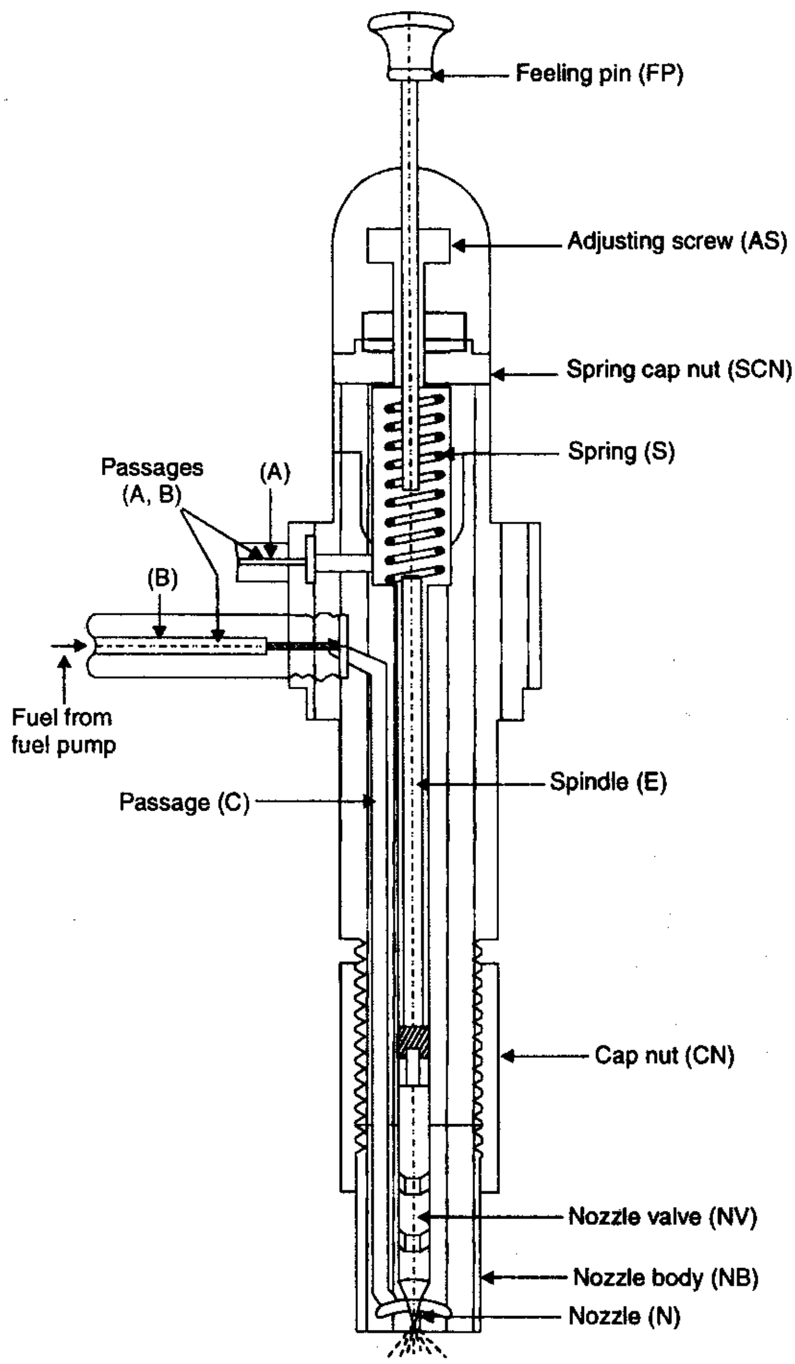


Fig. 4.16. Fuel atomizer or injector.

4.13.5. Cooling Systems

In an I.C. engine, the temperature of the gases inside the engine cylinder may very from 35°C or less to as high as 2750°C during the cycle. If an engine is allowed to run without external cooling, the cylinder walls, cylinder and piston will tend to assume the average temperature of the gases to which they are exposed, which may be of the order of 1000 to 1500°C. Obviously at such high temperature, the metals will loose their characteristics and piston will expand considerably and sieze the liner. Of course theoretically thermal efficiency of the engine will improve without cooling but actually the engine will sieze to run. If the cylinder wall temperature is allowed to rise above a certain limit, about 65°C, the lubricating oil will begin to evaporate rapidly and both cylinder and piston may be damaged. Also high temperature may cause excessive stress in some parts rendering them useless for further operation. In view of this, part of the heat generated inside the engine cylinder is allowed to be carried away by the cooling system. *Thus cooling system is provided on an engine for the following reasons* :

1. The even expansion of piston in the cylinder may result in seizure of the piston.
2. High temperatures reduce strength of piston and cylinder liner.
3. Overheated cylinder may lead to pre-ignition of the charge, in case of spark ignition engine.
4. Physical and chemical changes may occur in lubricating oil which may cause sticking of piston rings and excessive wear of cylinder.

Almost 25 to 35 per cent of total heat supplied in the fuel is removed by the cooling medium. Heat carried away by lubricating oil and heat lost by radiation amounts 3 to 5 per cent of total heat supplied.

There are mainly two methods of cooling I.C. engine :

1. Air cooling 2. Liquid cooling.

1. Air cooling

In this method, heat is carried away by the air flowing over and around the engine cylinder. It is used in scooters, motorcycles etc. Here fins are cast on the cylinder head and cylinder barrel which provide additional conductive and radiating surface (Fig. 4.17). The fins are arranged in such a way that they are at right angles to the cylinder axis.

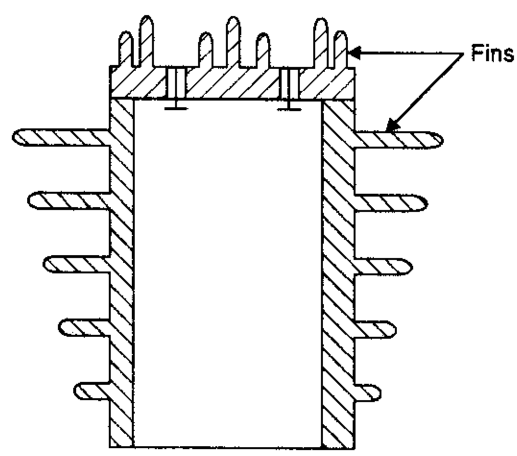


Fig. 4.17. Air cooling.

Advantages :

1. The design of the engine becomes simpler as no water jackets are required. The cylinder can be of identical dimensions and individually detachable and therefore cheaper to renew in case of accident etc.

2. Absence of cooling pipes, radiator etc. makes the cooling system simpler.
3. No danger of coolant leakage etc.
4. The engine is not subjected to freezing troubles etc. usually encountered in case of water cooled engine.
5. The weight per B.P. of the air-cooled engine is less than that of water-cooled engine.
6. In this case engine is rather a self contained unit as it requires no external components *e.g.* radiator, headers, tank etc.
7. Installation of air-cooled engines is easier.

Disadvantages :

1. Their movement is noisy.
2. Non-uniform cooling.
3. The output of air-cooled engine is less than that of a liquid-cooled engine.
4. Maintenance is not easy.
5. Smaller useful compression ratio.

2. Liquid cooling

In this method of cooling engines, the cylinder walls and heads are provided with jackets through which the cooling liquid can circulate. The heat is transferred from cylinder walls to the liquid by convection and conduction. The liquid becomes heated in its passage through the jackets and is itself cooled by means of an air-cooled radiator system. The heat from liquid in turn is transferred to air.

Various methods are used for circulating the water around the cylinder and cylinder head. These are :

1. Thermo-syphon cooling
2. Forced or pump cooling
3. Cooling with thermostatic regulator
4. Pressurised water cooling
5. Evaporative cooling.

1. *Thermo-syphon cooling :*

The basis of this type of cooling is the fact that water becomes light on heating. Fig. 4.18 shows the thermo-syphon cooling arrangement. The top of radiator is connected to the top of water

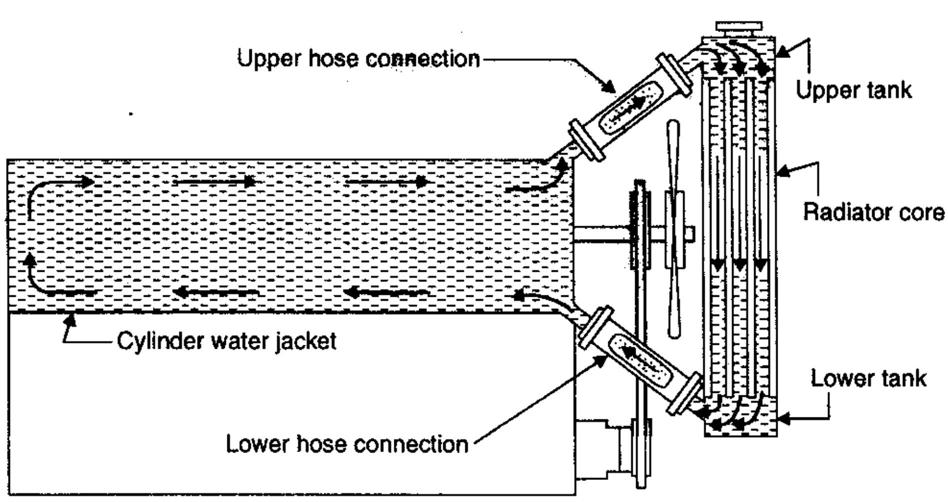


Fig. 4.18. Thermo-syphon cooling.

jacket by a pipe and bottom of the radiator to the bottom of the water jacket. Water travels down the radiator across which air is passed to cool it. The air flow can take place due to vehicle motion or a fan can be provided for the purpose.

This system has the advantage that it is quite *simple and automatic* and is without any water pump unless there is leak, there is nothing to get out of order.

The major short coming of this system is that cooling depends only on the temperature and is independent of the engine speed. The rate of circulation is *slow and insufficient*. The circulation of water starts only after the engine has become hot enough to cause thermo-syphon action. This system requires that the radiator be above the engine for gravity flow of water to engine.

Thermo-syphon system is *not widely used at present.*

2. *Forced or pump system :*

Refer Fig. 4.19. In this system, a pump is used to cause positive circulation of water in the water jacket. Usually the pump is belt driven from the engine.

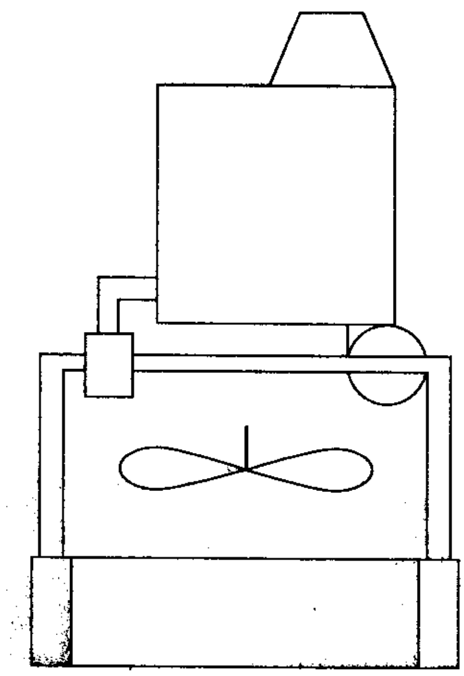


Fig. 4.19. Forced or pump system.

The advantage of forced system is that *cooling is ensured* under all conditions of operation.

This system entails the following *demerits* :

(*i*) The cooling is independent of temperature. This may, under certain circumstances, result in overcooling the engine. (*ii*) While moving uphill the cooling requirement is increased because more fuel is burned. However, the coolant circulation is reduced which may result in over-heating the engine. (*iii*) As soon as the engine is stopped the cooling also ceases. This is undesirable because cooling must continue till the temperatures are reduced to normal values.

3. *Thermostat cooling :*

Too lower cylinder barrel temperature, may result in severe corrosion damage due to condensation of acids on the barrel wall. To avoid such a situation it is customary to use a thermostat (a temperature controlling device) to stop flow of coolant below a pre-set cylinder barrel temperature.

Most modern cooling system employ a thermo-static device which prevents the water in the engine jacket from circulating through the radiator for cooling until its temperature has reached to a value suitable for efficient engine operation.

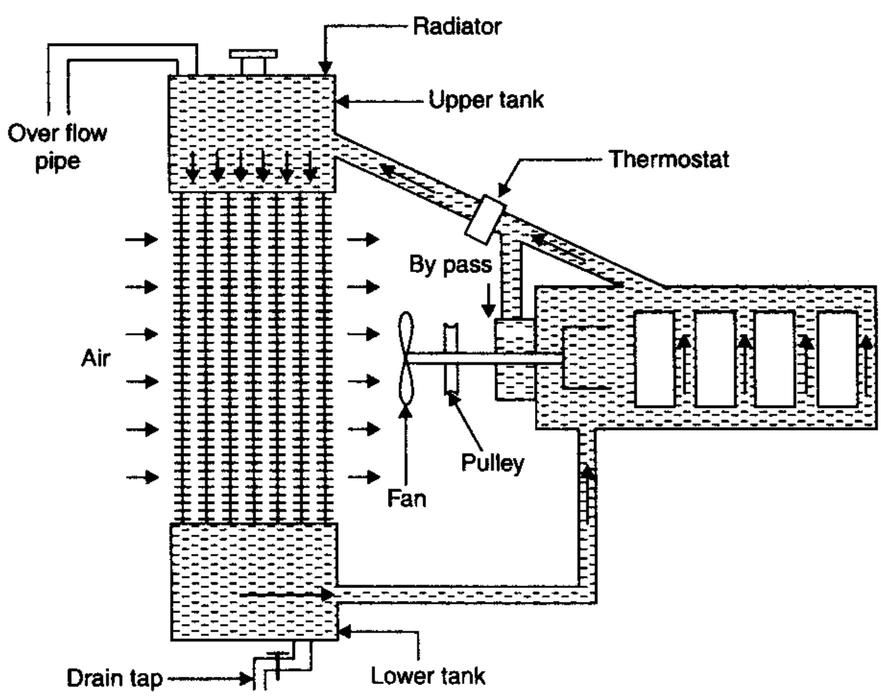


Fig. 4.20. Thermostatically controlled cooling system.

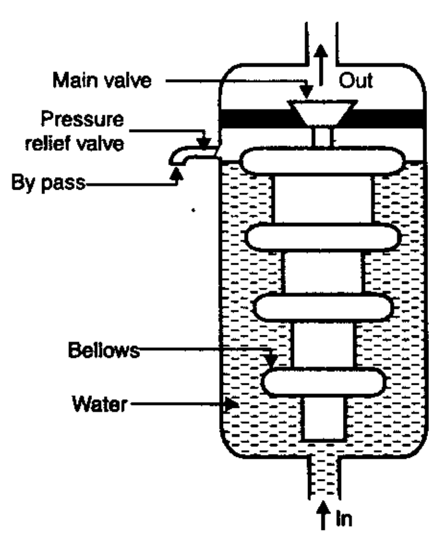


Fig. 4.21. Typical thermostat.

Fig. 4.20 shows a systematic diagram of a thermostatically controlled cooling system. Also shown is a typical thermostat (Fig. 4.21). It consists of bellows which are made of thin copper tubes, partially filled with a volatile liquid like ether or methyl alcohol. The volatile liquid changes into vapour at the correct working temperature, thus creating enough pressure to expand the bellows. The temperature at which the thermostat operates is set by the manufacturers and cannot be altered. The movement of the bellows opens the main valve in the ratio of temperature rise, increasing or restricting the flow of water from engine to the radiator. Hence when the normal temperature of the engine has been reached the valve opens and circulation of water commences. When the unit is closed the gas condenses and so the pressure falls. The bellows collapse and the thermostat seats on its seat and circulation around thermostat stops. When the thermostat valve is not open and the engine is running the water being pumped rises in pressure and causes the pressure relief valve to open. Thus the water completes its circulation through the by-pass as shown in Figs. 4.20 and 4.21. Now when the temperature of water around the engine-cylinder rises upto a certain limit, it causes the thermostat valve to open. The pressure of water being pumped falls and pressure relief valve closes. So the flow of cooling water in the normal circuit commences through the radiator. This accelerates the rise of temperature of the cylinder walls and water and more power is developed in a few moments of the starting of the engine.

Another method of warming up the radiator water upto the normal temperature is by utilising the shutter in the radiator in order to restrict the incoming air through the radiator till the engine warms up. Thereafter, the shutter is opened gradually so that the desired rate of cooling is achieved.

4. *Pressurised water cooling*

The boiling point of the coolant can be increased by increasing its pressure. This allows a greater heat transfer to occur in the radiator due to a larger temperature differential. Usually the water pressure is kept between 1.5 bar to 2.0 bar. Use of pressurised water cooling requires an additional valve, called vacuum valve, to avoid formation of vacuum when the water is cooled after engine has stopped. A safety valve in the form of pressure relief valve is provided so that whenever cap is opened the pressure is immediately relieved.

5. *Evaporative cooling*

In this system, also called steam or vapour cooling, the temperature of the cooling water is allowed to reach a temperature of 100°C. This method of cooling utilises the high latent heat of vapourisation of water to obtain cooling with minimum of water. Fig. 4.22 shows such a system. The

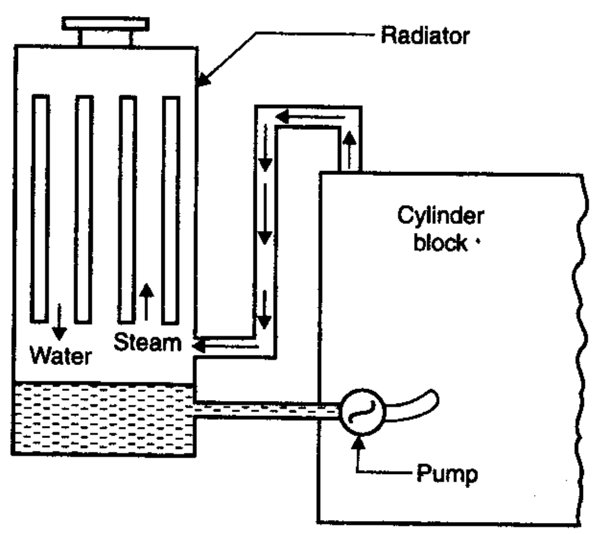


Fig. 4.22. Evaporating cooling.

cooling circuit is such that coolant is always liquid but the steam formed is flashed off in the separate vessel. The make up water so formed is sent back for cooling. This system is used for cooling of many types of *industrial engines.*

Advantages of liquid cooling :

1. Compact design of engine with appreciable smaller frontal area is possible.
2. The fuel consumption of a high compression liquid-cooled engine is rather lower than for air-cooled one.
3. More even cooling of cylinder barrels and heads due to jacketing makes it easier to reduce the cylinder head and valve seating temperature.
4. In case of water-cooled engine installation is not necessary at the front of mobile vehicles, aircrafts etc. as the cooling system can be conveniently located wherever required. This is not possible in case of air-cooled engines.
5. The size of the engine does not involve serious problem as far as design of cooling system is concerned. In case of air-cooled engines particularly in high horse power range difficulty is encountered in circulation of required quantity of air for cooling purposes.

Disadvantages :

1. This is dependent system in which supply of water/liquid for circulation in the jacket is required.
2. Power absorbed by the pump for water circulation is considerably higher than that for cooling fans.
3. In the event of failure of cooling system serious damage may be caused to the engine.
4. Cost of system is considerably high.
5. System requires considerable attention for the maintenance of various parts of system.

4.13.6. Lubrication Systems

Lubrication is the admittance of oil between two surface having relative motion. The purpose of lubrication may be one or more of the following :

1. To reduce friction and wear between the parts having relative motion.
2. To cool the surfaces by carrying away heat generated due to friction.
3. To seal a space adjoining the surfaces such as piston rings and cylinder liner.
4. To clean the surface by carrying away the carbon and metal particles caused by wear.
5. To absorb shock between bearings and other parts and consequently reduce noise.

The main parts of an engine which need lubrication are as given below :

(*i*) Main crankshaft bearings. (*ii*) Big-end bearings.
(*iii*) Small end or gudgeon pin bearings. (*iv*) Piston rings and cylinder walls.
(*v*) Timing gears. (*vi*) Camshaft and cam shaft bearings.
(*vii*) Valve mechanism.
(*viii*) Valve guides, valve tappets and rocker arms.

Various lubrication systems used for I.C. engines may be classified as :

1. Wet sump lubrication system.
2. Dry sump lubrication system.
3. Mist lubrication system.

Wet sump lubrication system

These systems employ a large capacity oil sump at the base of crankchamber, from which the oil is drawn by a low pressure oil pump and delivered to various parts. Oil then gradually returns back to the sump after serving the purpose.

(a) ***Splash system.*** Refer Fig. 4.23. This system is used on some *small four stroke stationary engines*. In this case the caps on the big end bearings of connecting rods are provided with scoops which, when the connecting rod is in the lowest position, just dip into oil troughs and thus directs the oil through holes in the caps to the big end bearings. Due to splash of oil it reaches the lower portion of the cylinder walls, crankshaft and other parts requiring lubrication. Surplus oil eventually flows back to the oil sump. Oil level in the troughs is maintained by means of a oil pump which takes oil from sump, through a filter.

Splash system is suitable for low and medium speed engines having moderate bearing load pressures. For high performance engines, which normally operate at high bearing pressures and rubbing speeds this system does not serve the purpose.

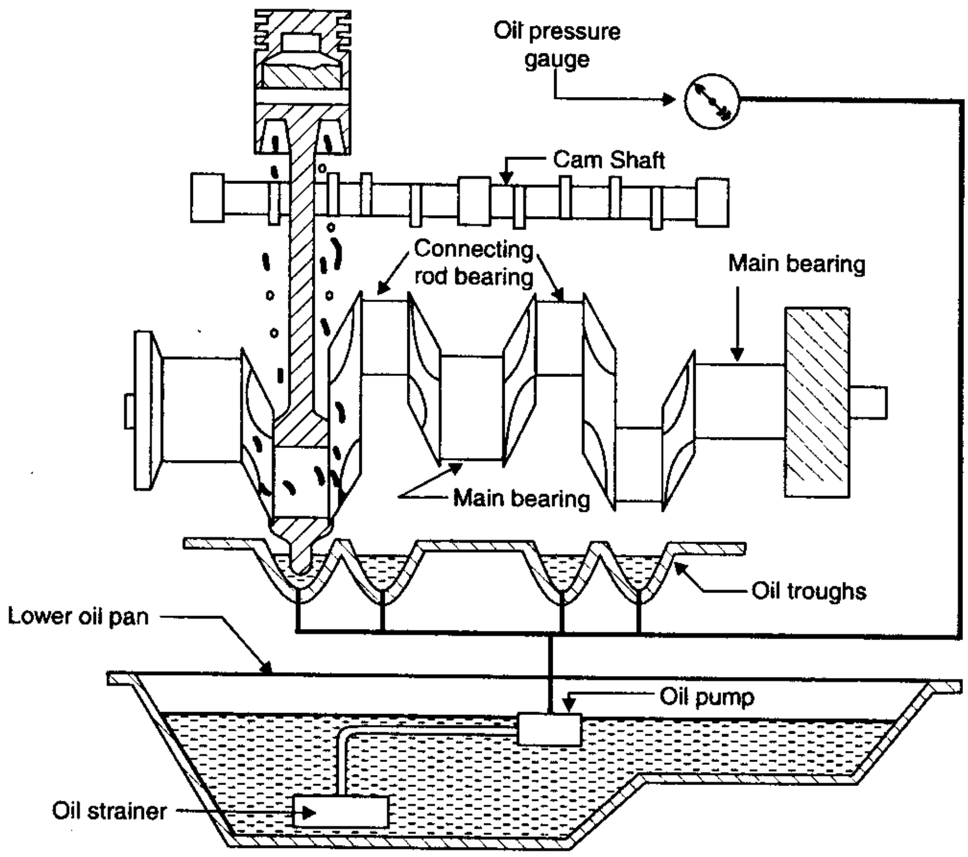


Fig. 4.23. Splash system.

(b) ***Semi-pressure system.*** This method is a combination of splash and pressure systems. It incorporates the advantages of both. In this case main supply of oil is located in the base of crank-chamber. Oil is drawn from the lower portion of the sump through a filter and is delivered by means of a gear pump at pressure of about 1 bar to the main bearings. The big end bearings are lubricated by means of a spray through nozzles. Thus oil also lubricates the cams, crankshaft bearings, cylinder walls and timing gears. An oil pressure gauge is provided to indicate satisfactory oil supply.

The system is less costly to install as compared to pressure system. It enables higher bearing loads and engine speeds to be employed as compared to splash system.

(c) ***Full pressure system.*** In this system, oil from oil sump is pumped under pressure to the various parts requiring lubrication. Refer Fig. 4.24. The oil is drawn from the sump through filter and pumped by means of a gear pump. Oil is delivered by the pressure pump at pressure ranging from 1.5 to 4 bar. The oil under pressure is supplied to main bearings of crankshaft and camshaft. Holes drilled through the main crankshafts bearing journals, communicate oil to the big end bearings and also small end bearings through holes drilled in connecting rods. A pressure gauge is provided to confirm the circulation of oil to the various parts. A *pressure regulating valve* is also provided on the delivery side of this pump to prevent excessive pressure.

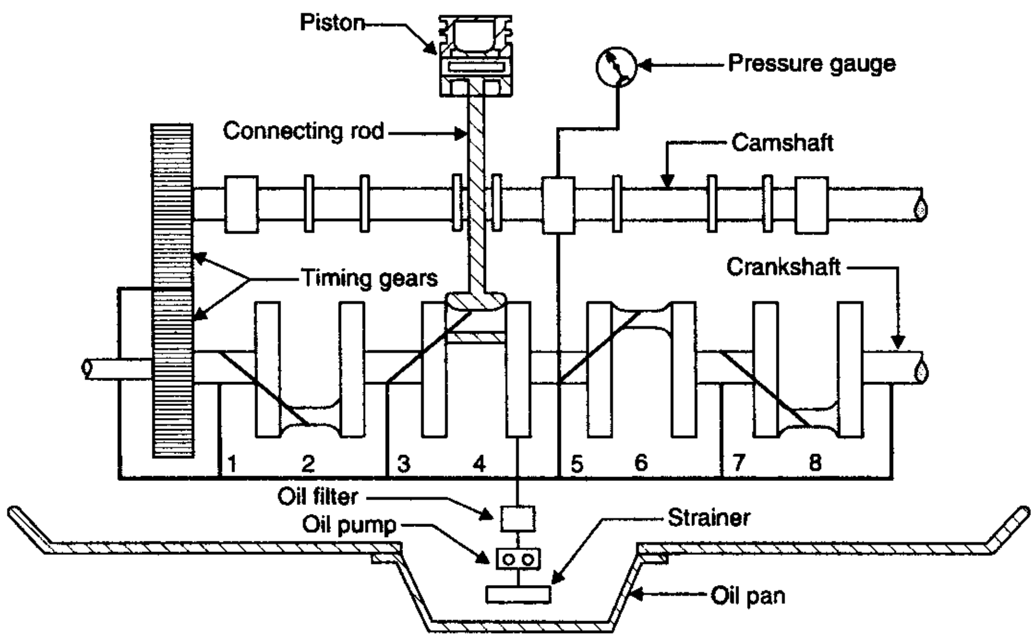


Fig. 4.24. Full pressure system.

This system finds favour from most of the engine manufacturers as it allows high bearing pressure and rubbing speeds.

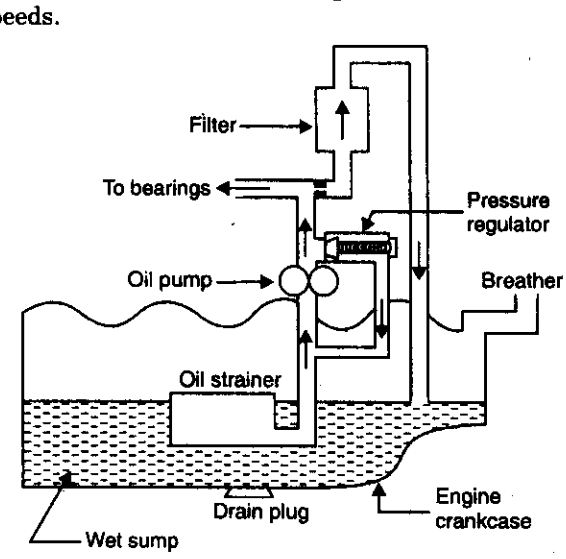


Fig. 4.25. Wet sump lubrication system.

The general arrangement of *wet sump lubrication system* is shown in Fig. 4.25. In this case oil is always contained in the sump which is drawn by the pump through a strainer.

2. Dry sump lubrication system

Refer Fig. 4.26. In this system, the oil from the sump is carried to a separate storage tank out side the engine cylinder block. The oil from sump is pumped by means of a sump pump through filters to the storage tank. Oil from storage tank is pumped *to the engine cylinder* through oil cooler. Oil pressure may varies from 3 to 8 kgf/cm^2. *Dry sump lubrication system is generally adopted for high capacity engines.*

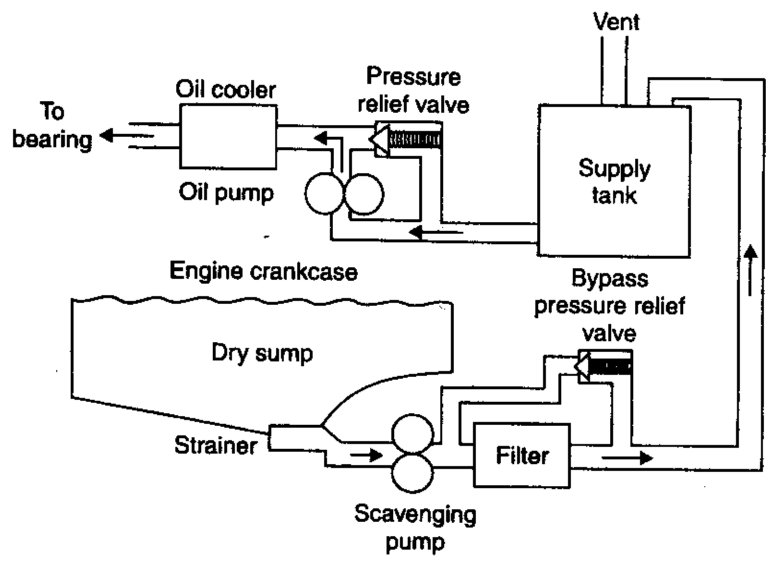


Fig. 4.26. Dry sump librication system.

3. Mist lubrication system

This system is used for two stroke cycle engines. Most of these engines are crankcharged, *i.e.* they employ crankcase compression and thus, are not suitable for cran case lubrication. These engines are lubricated by adding 2 to 3 per cent lubricating oil in the fuel tank. The oil and fuel mixture is induced through the carburettor. The gasoline is vaporised ; and the oil in the form of mist, goes via crankcase into the cylinder. The oil which impinges on the crankcase walls lubricates the main and connecting rod bearings, and rest of the oil which passes on the cylinder during charging and scavanging periods, lubricates the piston, piston rings and the cylinder.

Advantages

1. System is simple.
2. Low cost (because no oil pump filter etc. are required).

Disadvantages

1. A portion of the lubricating oil invariably burns in combustion chamber. This bearing oil when burned, and still worse, when partially burned in combustion chamber leads to heavy exhaust emissions and formation of heavy deposit on piston crown, ring grooves and exhaust port which interferes with the efficient engine operation.
2. One of the main functions of lubricating oil is the protection of anti-friction bearings etc. against corrosion. Since the oil comes in close contact with acidic vapours produced during the combustion process, it rapidly loses its anti-corrosion properties resulting in corrosion damage of bearings.

3. For effective lubrication oil and fuel must be thoroughly mixed. This requires either separate mixing prior to use or use of some additive to give the oil good mixing characteristics.
4. Due to higher exhaust temperature and less efficient scavening the crank case oil is diluted. In addition some lubricating oil burns in combustion chamber. This results in 5 to 15 per cent higher lubricant consumption for two stroke engine of similar size.
5. Since there is no control over the lubrication oil, once introduced with fuel, most of the two stroke engines are *over-oiled* most of the time.

4.13.7. Engine Starting System

The following three are the *commonly used starting systems* in large and medium size engines :

1. Starting by an auxiliary engine.
2. Use of electric motors or self starters.
3. Compressed air system.

1. Starting by an auxiliary engine (generally petrol driven) :

In this system an auxiliary engine is mounted close to the main engine and drives the latter through a clutch and gears. The clutch is first disengaged and the auxiliary engine started by hand or by a self starter motor. When it has warmed up and runs normally the drive gear is engaged through the clutch, and the main engine is cranked for starting. To avoid the danger of damage to drive gear it is desirable to have an over-running clutch or starter type drive.

2. Use of electric motors or self starters :

These are employed for small diesel and gasoline engines. A storage battery of 12 to 36 volts is used to supply power to an electric motor which is geared to the flywheel with arrangement for automatic disengagement after the engine has started. The motor draws a heavy current and is designed to be engaged continuously for about 30 seconds only, after which it is required to cool off for a minute or so, and then re-engaged. This is done till the engine starts up. When the engine is running a small d.c. generator on the engine serves to charge the battery.

3. Compressed air system :

The compressed air system is commonly used for starting large diesel engines employed for stationary power plant service. Compressed air at about 17 bar supplied from an air tank or bottle is admitted to a few of the engine cylinders making them work like reciprocating air motors to run the engine shaft. Fuel is admitted to the remaining cylinders and ignites in the normal way causing the engine to start. The air bottle or tank is charged by a motor or gasoline engine driven compressor. The system includes the following :

(*i*) Storage tank/vessel (*ii*) A safety valve
(*iii*) Interconnecting pipe work.

Methods of Starting and Stopping Engines

Although starting procedure may differ from engine to engine but some common steps are listed below :

Starting of engines :

1. In case of *electric motor* starting check the condition of storage battery. If *compressed air system* is used, then air pressure may be checked first and the air system inspected for possible leakage.

2. As prescribed by the manufactures, all necessary checks for fuel, lubricating oil and cooling water should be made.

3. Crank the engine after ensuring that all load is put off and decompression device is in use, and then let it start.

4. For a few minutes run the engine at slow speed and observe the working of fuel pumps, lubricating systems, fuel and oil pressures etc. The lubrication of generator and excited bearings should also be checked.

5. Increase the engine speed gradually till it synchronizes with the station bus-bar.

6. Connect the generator to the bus-bar when it is in synchronism and increase the engine speed till it begins to share the desired load.

Stopping of engines :

1. Reduce the speed of the engine gradually until practically no power is delivered by the alternator.

2. Disconnect the unit from the bus and allow the engine to idle for a few minute, and then stop it in confirmity with the instructions detailed by the manufacturers.

4.13.8. Governing System

The function of the governing system is to maintain the speed of the engine constant irrespective of load on the plant. This is done generally by varying the fuel supply to the engine according to the load.

4.14. COMBUSTION PHENOMENON IN C.I. ENGINES

The process of combustion in the compression ignition (C.I.) engine is fundamentally different from that in a spark-ignition engine. In C.I. engine combustion occurs by the high temperature produced by the compression of the air, *i.e.* it is an *auto-ignition*. For this a minimum compression ratio of 12 is required. The efficiency of the cycle increases with higher values of compression ratio but the maximum pressure reached in the cylinder also increases. This requires heavier construction. The upper limit of compression ratio in a C.I. engine is due to mechanical factor and is a compromise between high efficiency and low weight and cost. The normal compression ratios are in the range of 14 to 17, but may be upto 23. The air-fuel ratios used in the C.I. engine lie between 18 and 25 as against about 14 in the S.I. engine, and hence C.I. engines are bigger and heavier for the same power than S.I. engines.

In the C.I. engine, the intake is air alone and the fuel is injected at high pressure in the form of fine droplets near the end of compression. This leads to delay period in the C.I. engine, is greater than that in the S.I. engine. The *exact phenomenon of combustion in the C.I. engine* is explained below.

Each minute droplet of fuel as it enters the highly heated air of engine cylinder is quickly surrounded by an envelope of its own vapour and this, in turn, and an appreciable interval is inflamed at the surface of the envelope. To evaporate the liquid, latent heat is abstracted from the surrounding air which reduces the temperature of the thin layer of air surrounding the droplet, and some time must elapse before this temperature can be raised again by abstracting heat from the main bulk of air in this vicinity. As soon as this vapour and the air in actual contact with it reach a certain temperature, ignition will take place. Once ignition has been started and a flame established the heat required for further evaporation will be supplied from that released by combustion. The vapour would be burning as fast as it can find fresh oxygen, *i.e.*, it will depend upon the rate at which it is moving through the air or the air is moving past it.

In the C.I. engine, the fuel is not fed in at once but is spread over a definite period. The first arrivals meet air whose temperature is only little above their self-ignition temperature and the delay is more or less prolonged. The later arrivals find air already heated to a far higher temperature by the burning of their predecessors and therefore light up much more quickly, almost as they issue from the injector nozzle, but their subsequent progress is handicapped for there is less oxygen to find.

If the air within the cylinder were motionless, only a small proportion of the fuel would find sufficient oxygen, for it is impossible to distribute the droplets uniformly throughout the combustion space. Therefore some air movement is absolutely essential, as in the S.I. engine. But there is a fundamental difference between the air movements in the two types of engines. In the S.I. engine we call it *turbulence* and mean a confusion of whirls and eddies with no general direction of flow, (to break up the surface of the flame front, and to distribute the shreds of flame throughout an externally prepared combustible mixture). In the C.I. engine we call it *air swirl* and mean an orderly movement of the whole body of the air, with or without some eddying or turbulence, so as to bring a continuous supply of fresh air to each burning droplet and sweep away the products of combustion which otherwise tend to suffocate it.

Three phases of C.I. engine combustion

In the C.I. engine, combustion may be considered in three distinct stages as shows in Fig. 4.27.

1. Ignition delay period.
2. Period of rapid or uncontrolled combustion.
3. Period of controlled combustion.

The *third phase is followed by 'after burning'* (or burning on the expansion stroke), *which may be called the fourth phase of combustion.*

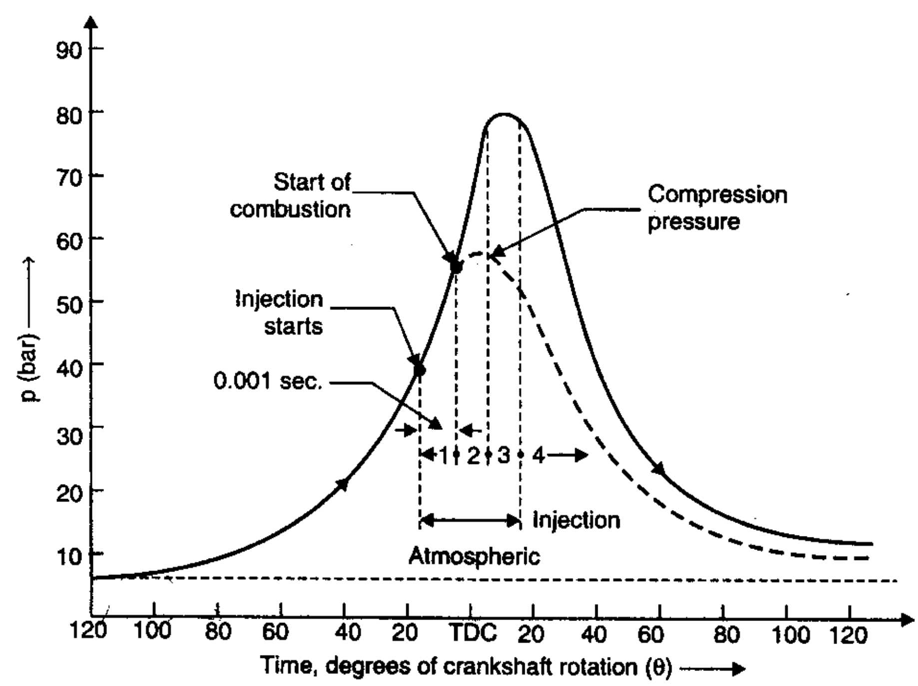


Fig. 4.27. Combustion phenomenon of C.I. engine.

1. ***Ignition delay period.*** The delay period is counted from the start of injection to the point where the p-θ combustion curve departs from air compression (or no ignition or motoring) curve. The delay period can be roughly sub-divided into *physical delay* and *chemical delay*. The *period of physical delay* is the time between the beginning of injection and the attainment of chemical reaction

conditions. In the *physical delay period,* the fuel is atomized, vaporized, mixed with air, and raised in temperature. In the *chemical delay period* reaction starts slowly and then accelerates until inflammation or ignition takes place (it may be noted that the ignition delay in the S.I. engine is essentially equivalent to the chemical delay in the C.I. engine).

The delay period exerts a great influence in the C.I. engine combustion phenomenon. It is clear that the pressure reached during the second stage will depend upon the duration of the delay period ; the longer the delay, the more rapid and higher the pressure rise, since more fuel will be present in the cylinder before the rate of burning comes under control. This causes rough running and may cause *diesel knock*. Therefore we must *aim to keep the delay period as short as possible, both for the sake of smooth running and in order to maintain control over the pressure changes. But some delay period is necessary otherwise the droplets would not be dispersed in the air for complete combustion.* However, the delay period imposed upon is greater than what is needed and the designer's efforts are to shorten it as much as possible.

2. *Period of rapid or uncontrolled combustion.* The second stage of combustion in C.I. engines, after the delay period, is the period of rapid or uncontrolled combustion. This period is counted from the end of the delay period to the point of maximum pressure on the indicator diagram. In this second stage of combustion, the rise of pressure is rapid because during the delay period the droplets of fuel have had time to spread themselves out over a wide area and they have fresh air all around them. About one-third of heat is evolved during this process.

The rate of pressure rise depends on the amount of fuel present at the end of delay period, degree of turbulence, fineness of atomization and spray pattern.

3. *Period of controlled combustion.* At the end of second stage of combustion, the temperature and pressure, are so high that the fuel droplets injected in the third stage burn almost as they enter and any further pressure rise can be controlled by purely mechanical means, *i.e.* by the injection rate. The period of controlled combustion is assumed to end at maximum cycle temperature. The heat evolved by the end of controlled combustion is about 70 to 80 per cent.

4. *After burning.* The combustion continues even after the fuel injection is over, because of poor distribution of fuel particles. This burning may continue in the expansion stroke upto 70° to 80° of crank travel from T.D.C. This continued burning, called the *after burning*, may be considered as the *fourth stage of the combustion.* The total heat evolved by the end of entire combustion process is 95 to 97% ; 3 to 5% of heat goes as unburned fuel in exhaust.

In the p-V diagram, the stages of combustion are not seen because of little movement of piston with crank angle at the end and reversal of stroke. So for studying the combustion stages, therefore, a pressure-crank angle or time, p-θ or p-t diagram is invariable used. In the actual diagram, the various stages of combustion look merged, yet the individual stage is distinguishable.

4.15. DELAY PERIOD (OR IGNITION LAG) IN C.I. ENGINES

—In C.I. (compression ignition) engine, the fuel which is in atomised form is considerably colder than the hot compressed air in the cylinder. Although the actual ignition is almost instantaneous, an appreciable time elapses before the combustion is in full progress. This time occupied is called the *delay period* or *ignition lag. It is time immediately following injection of the fuel during which the ignition process is being initiated and the pressure does not rise beyond the value it would have due to compression of air* :

—The delay period extends for about 13°, movement of the crank. The time for which it occurs decreases with increase in engine speed.

The delay period depends upon the following :

(*i*) Temperature and pressure in the cylinder at the time of injection.

(*ii*) Nature of the fuel mixture strength.

(*iii*) Relative velocity between the fuel injection and air turbulence.

(*iv*) Presence of residual gases.

(*v*) Rate of fuel injection.

(*vi*) To small extent the finess of the fuel spray.

The delay period increases with load but is not much affected by injection pressure.

—*The delay period should be as short as possible since a long delay period gives a more rapid rise in pressure rise and thus causes knocking.*

4.16. DIESEL KNOCK

If the delay period in C.I. engines is long a large amount of fuel will be injected and accumulated in the chamber. The auto-ignition of this large amount of fuel may cause high rate of pressure rise and high maximum pressure which may cause *knocking* in diesel engines. A long delay period not only increases the amount of fuel injected by the moment of ignition but also improves the homogeneity of the fuel-air mixture and its chemical preparedness for explosion type self-ignition similar to detonation is S.I. engines.

The following are the differences in the knocking phenomena of the S.I. and C.I. engines :

1. In the *S.I. engine*, the detonation occurs near the *end of combustion* whereas in the *C.I. engine* detonation occurs near the *beginning of combustion.*
2. The detonation in the S.I. engine is of a *homogeneous charge* causing *very high rate of pressure rise* and very high maximum pressure. In the C.I. engine, the fuel and air are *imperfectly mixed* and hence the *rate of pressure rise is normally lower than that in the detonating part of the charge in the S.I. engine.*
3. In the C.I. engine the fuel is injected into the cylinder only at the end of the compression stroke, there is no question of pre-ignition as in S.I. engine.
4. In the S.I. engine, it is relatively easy to distinguish between knocking and non-knocking operation as the human ear easily finds the distinction.

4.17. CETANE NUMBER

—The cetane rating of a diesel fuel is a *measure of its ability to autoignite quickly* when it is injected into the compressed and heated air in the engine. Though ignition delay is affected by several engine design parameters such as compression ratio, injection rate, injection time, inlet air temperature etc., it is also dependent on hydrocarbon composition of the fuel and to some extent on its volatility characteristic. *The cetane number is a numerical measure of the influence the diesel fuel has in determining the ignition delay. Higher the cetane rating of the fuel lesser is the propensity for diesel knock.*

—The procedure for obtaining cetane number is similar to that for obtaining the octane number of petrols. Reference mixtures of *cetane* ($C_{16}H_{34}$) (high ignitability), and *α-methyl-naphthalene* ($C_{11}H_{10}$) (low ignitability), are used. *The mixture is made by volume and the ignitability of the test fuel is quoted as the percentage of cetane in the reference mixture which has the same ignitability.*

—For higher speed engines, the cetane number required is about 50, for medium speed engines about 40, and for slow speed engines about 30.

—Cetane number is the most important single fuel property which affects the exhaust emissions, noise and startability of a diesel engine. In general, *lower the cetane number higher are the hydrocarbon emissions and noise level.* Low centane fuels increase ignition delay so that start of combustion is near to top dead centre. This is similar to retarding of injection timing which is also known to result in higher hydrocarbon levels.

—In general, *a high octane value implies a low cetane value.*

4.18. BASIC DESIGNS OF C.I. ENGINE COMBUSTION CHAMBERS

In C.I. engines several types of combustion chambers are used. Each of these has its own peculiarities, and desirable, as well as undesirable features. Any one of these combustion chambers may produce good results in one field of application, but less desirable, or even poor results in another. No one combustion chamber design has yet been developed which will produce the best result is all types of engines. The particular design chosen, then, must be that which accomplishes the best performance for the application desired.

Four specific design which find wide use in C.I. engines are discussed below :

1. The non-turbulent type
   (*i*) Open combustion chamber.
2. The turbulent type
   (*i*) Turbulent chamber (*ii*) Precombustion chamber
   (*iii*) Energy cell.

1. Non-turbulent type

Fig. 4.28 (*a*) illustrates the usual design of *open combustion chamber*, which is representative of non-turbulent type. The fuel is injected directly into the upper portion of the cylinder, which acts as the combustion chamber. This type depends little on turbulence to perform the mixing. Consequently, the heat loss to the chamber walls is relatively low, and easier starting results. In order to obtain proper penetration and dispersal of the fuel necessary for mixing with the air, however, high injection pressures and multi-orifice nozzles are required. This necessitates small nozzle openings and result in more frequent clogging or diversion of the fuel spray by accumulated carbon particles, with consequent higher maintenance costs.

This type of chamber is ordinarily used on *low speed engines*, where injection is spread through a greater period of time and thus ignition delay is a relatively less important factor. Consequently, less costly fuels with longer ignition delay may be used.

2. Turbulent type

The turbulent chamber, precombustion chamber, and energy cell are variation of the turbulent type of chamber, and are illustrated in Fig. 4.28 (*b*), (*c*) and (*d*).

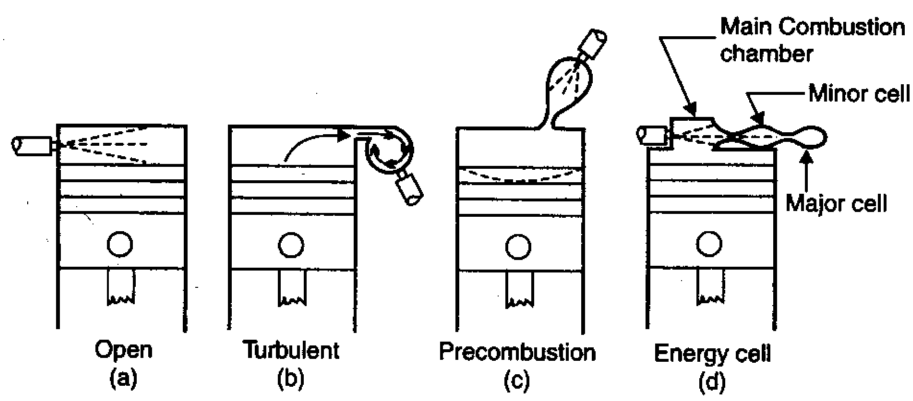


Fig. 4.28. Some commonly used C.I. engine combustion chambers.

—In the *'turbulent chamber'*, Fig. 4.28 *(b)*, the upward moving piston forces a flow of air into a small antechamber, thus imparting a rotary motion to the air passing the pintle type nozzle. As the fuel is injected into the rotating air, it is partially mixed with this air, and commences to burn. The pressure built up in the antechamber by the expanding burning gases force the burning and unburned fuel and air mixtures back into the main chamber, again imparting high turbulence and further assisting combustion.

—In the *'precombustion chamber'*, Fig. 4.28 *(c)*, the upward moving piston forces part of the air into a side chamber, called the precombustion chamber. Fuel is injected into the air in the precombustion chamber by a pintl type nozzle. The combustion of the fuel and air produces high pressures in the precombustion chamber, thus creating high turbulence and producing good mixing and combustion.

—The *'energy cell'* is more complex than the precombustion chamber. It is illustrated in Fig. 4.28 *(d)*. As the piston moves up on the compression stroke, some of the air is forced into the major and minor chambers of the energy cell. When the fuel is injected through the pintle type nozzle, part of the fuel passes across the main combustion chamber and enters the minor cell, where it is mixed with the entering air. Combustion first commences in the main combustion chamber where the temperature is higher, but the rate of burning is slower in this location, due to insufficient mixing of the fuel and air. The burning in the minor cell is slower at the start, but due to better mixing, progresses at a more rapid rate. The pressures built up in the minor cell, therefore, force the burning gases out into the main combustion chamber, thereby creating added turbulence and producing better combustion in this chamber. In the mean time, pressure is built up in the major cell, which then prolongs the action of the jet stream entering the main chamber, thus continuing to induce turbulence in the main chamber.

Summarily it may be said that a particular combustion chamber design must be chosen to perform a given job. No one combustion chamber can produce an ultimate of performance in all tasks. The design of the chamber must be based on a compromise, after full considerations of the following factors : *(i)* Heat lost to combustion chamber walls, *(ii)* Injection pressure, *(iii)* Nozzle design, *(iv)* Maintenance, *(v)* Ease of starting, *(vi)* Fuel requirement, *(vii)* Utilisation of air, *(viii)* Weight relation of engine to power output, *(ix)* Capacity for variable speed operation.

4.19. SUPERCHARGING

—*The purpose of supercharging is to raise the volumetric efficiency above that value which can be obtained by normal aspiration.*

The engine is an air pump. Increasing the air consumption permits greater quantities of fuel to be added, and results in a greater potential output. The indicated power produced is almost directly proportional to the engine air consumption. While brake power is not so closely related to air consumption, it is nevertheless, dependent upon the mass of air consumed. It is desirable, then, that the engine takes in greatest possible mass of air.

Three possible methods which might be utilized to increase the air consumption of an engine are :

1. *Increasing the piston displacement*, but this increases the size and weight of the engine, and *introduces additional cooling problems*.
2. *Running the engine at higher speeds*, which results in *increased fluid and mechanical friction losses, and imposes greater inertia stresses on engine parts.*
3. *Increasing the density of the charge*, such that a greater mass of charge is introduced into the same volume or same total piston displacement.

The last method of increasing the air capacity of an engine is widely used, and is termed *supercharging.*

The apparatus used to increase the air density is known as a *supercharger.* It is merely a *compressor* which provides a denser charge to the engine, thereby enabling the consumption of a greater mass of charge with the same total piston displacement. During the process of compressing the charge, the supercharger produces the following effects :

(*i*) *Provides better mixing of the air-fuel mixture.* The turbulent effect created by the supercharger assists in additional mixing of the fuel and air particles. The arrangement of certain types of superchargers, particularly the *centrifugal type*, also encourages more even distribution of the charge to the cylinders.

(*ii*) *The temperature of the charge is raised as it is compressed, resulting in a higher temperature within the cylinders.* This is partially beneficial in that it helps to produce better vapourisation of fuel (in case of S.I. engines) but deterimental in that it tends to lessen the density of the charge. The increase in temperature of the charge also effects the detonation of the fuel.

Supercharging tends to increase the possibility of detonation in a S.I. engine and lessen the possibility in a C.I. engine.

(*iii*) Power is required to drive the supercharger. This is usually taken from the engine and thereby removes, from over-all engine output, some of the gain in power obtained through supercharging.

Compressors used are of the following three types :

(*i*) ***Positive displacement type*** used with many reciprocating engines in stationary plants, vehicles and marine installations.

(*ii*) ***Axial flow type*** seldom used to supercharge reciprocating engines, it is widely used as the compressor unit of the *gas turbines.*

(*iii*) ***Centrifugal type*** *widely used* as the *supercharger for reciprocating engines, as well as compressor for gas turbines.* It is almost exclusively used as the supercharger with reciprocating power plants for *aircraft because it is relatively light and compact, and produces continuous flow rather than pulsating flow as in some positive displacement types.*

Supercharging of S.I. engines

—An ideal p-V diagram for a supercharged constant volume cycle is shown in Fig. 4.29.

Apart from increases in pressures over the un-supercharged cycle the main difference is that *pumping loop* (1-5-6-7-1) is now a positive one. The net indicated power (I.P.) is obtained by adding the contribution from pumping loop to that of the *power loop* (1-2-3-4-1). The gas is exhausted from the cylinder at the pressure of point 4 on the diagram. This value is high compared with atmospheric pressure and a considerable amount of energy is lost by allowing the gas to blow down in this way.

The power required to drive a blower mechanically connected to the engine must be subtracted from the engine output to obtain the net brake power (B.P.) of the supercharged engine.

Then, $$I_{\text{m.e.p.}} = \left(\frac{\text{area } 12341 + \text{area } 15671}{\text{length of diagram}}\right) \times \text{spring number} \quad ...(4.1)$$

and brake power (B.P.) = ($\eta_{\text{mech.}}$ × I.P.) – (power required to drive blower)

(for mechanical driven blowers only) ...(4.2)

(*i*) As far as S.I. engines are concerned, supercharging is employed *only for 'aircraft' and 'racing car engines'*. This is because the *increase in supercharging pressure increases the tendency to detonate.*

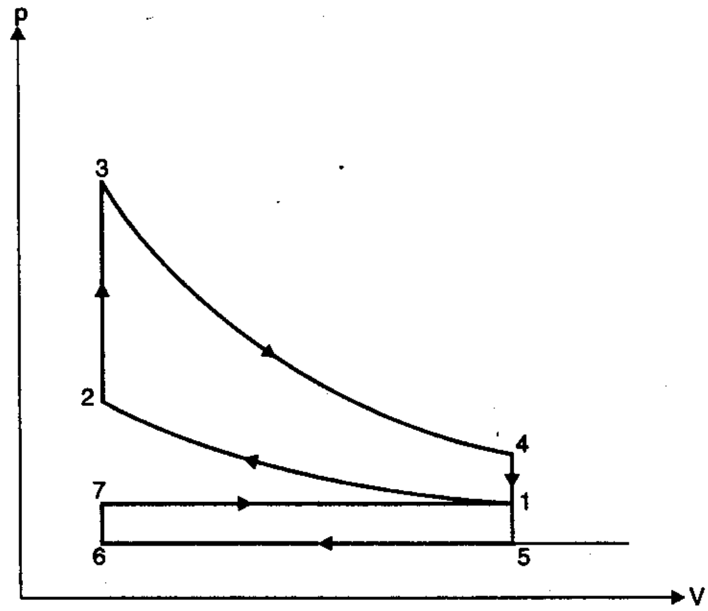


Fig. 4.29. *p-V* diagram for a supercharged constant volume cycle.

(*ii*) Supercharging of petrol engines, because of its poor fuel economy, is not very popular and is used only when a large amount of power is needed or when more power is needed *to compensate altitude loss.*

Supercharging of C.I. engines

Fig. 4.30 shows the *p-V* diagram for a supercharged constant pressure (diesel) cycle.

Unlike S.I. engines supercharging does not result in any combustion problem, *rather it improves combustion in diesel engine.* Increase in pressure and temperature of the intake air reduces significantly delay and hence the rate of pressure rise resulting in a *better, quieter and smoother combustion.* This improvement in combustion allows a poor quality fuel to be used in a diesel engine and it is also not sensitive to the type of fuel used. *The increase in intake temperature reduces volumetric and thermal efficiency but increase in density due to pressure compensates for this and intercooling is not necessary except for highly supercharged engines.*

If an unsupercharged engine is supercharged it will increase the reliability and durability of the engine due to smoother combustion and lower exhaust temperatures. The degree of supercharging is limited by thermal and mechanical load on the engine and strongly depends on the type of supercharger used and design of the engine.

Effects of supercharging on performance of the engine

1. The *'power output'* of a supercharged engine is *higher* than its naturally aspirated counterpart.
2. The *'mechanical efficiencies'* of supercharged engines are *slightly better* than the naturally aspirated engines.
3. Inspite of better mixing and combustion due to reduced delay a mechanically supercharged otto engine almost always have *'specific fuel consumption' higher than a naturally aspirated engine.*

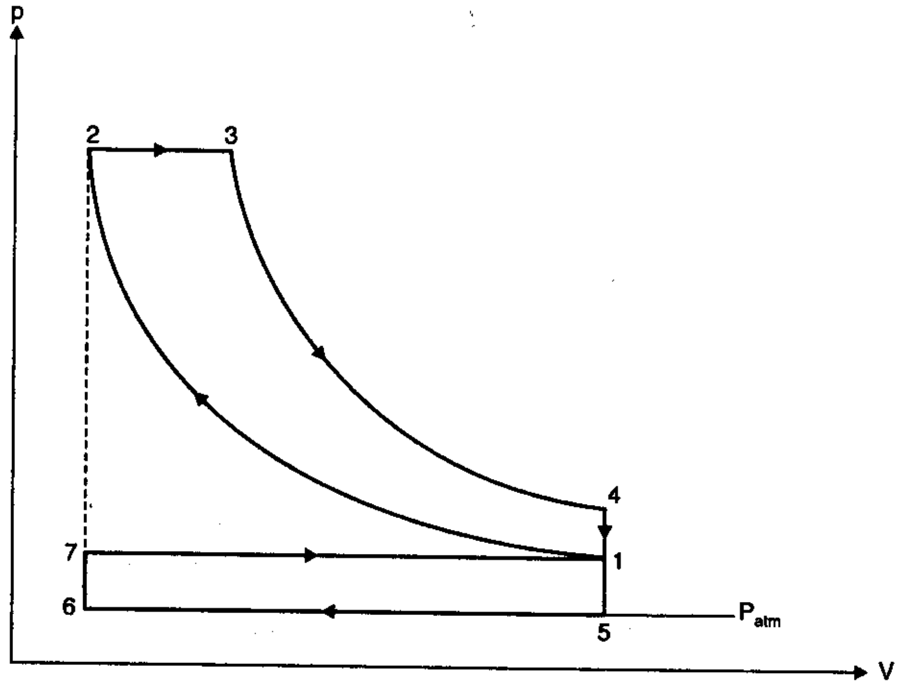


Fig. 4.30. *p-V* diagram for a supercharged constant pressure (diesel) cycle.

Fig. 4.31 shows the effect of supercharging on power and fuel.

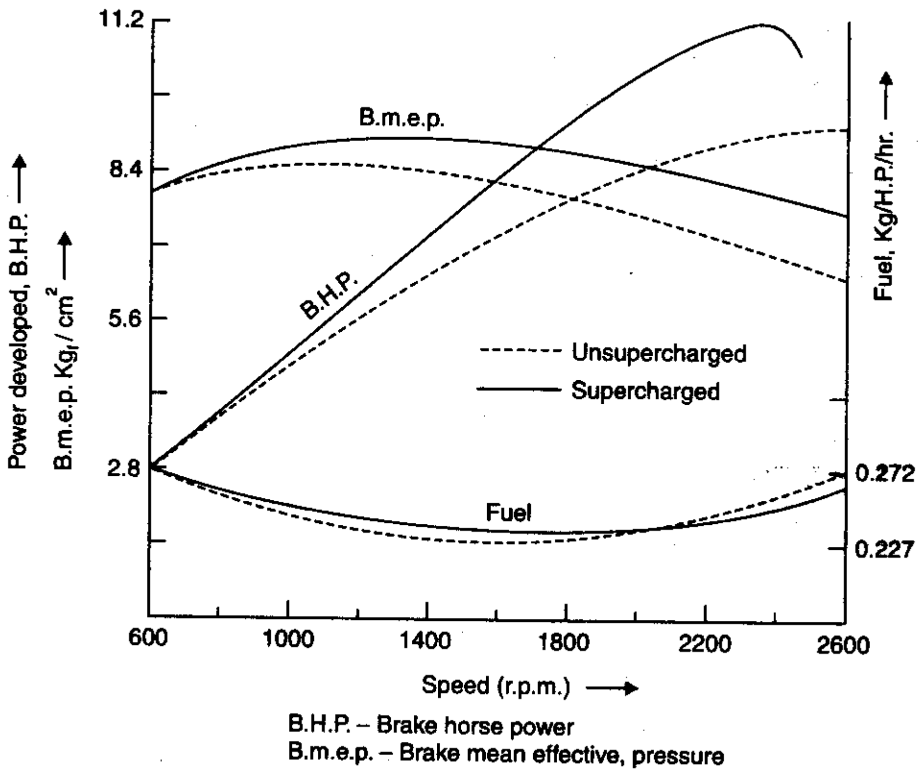


Fig. 4.31. Effect of supercharging on power and fuel.

4.20. OPERATION OF A DIESEL POWER PLANT

When diesel alternator sets are put in parallel, "*hunting*" or "*phase swinging*" may be produced *due to resonance* unless due care is taken in the design and manufacture of the sets. This condition occurs *due to resonance between the periodic disturbing forces of the engine and natural frequency of the system.* The engine forces result from uneven turning moment on the engine crank which are corrected by the flywheel effect. *"Hunting" results from the tendency of each set trying to pull the other into synchronism* and is characterised by flickering of lights.

To ensure *most economical operation of diesel engines* of different sizes when working together and sharing load it is necessary that they should carry the same percentage of their full load capacity at all times as the fuel consumption would be lowest in this condition. For best operation performance the manufacturer's recommendations should be strictly followed.

In order to get good performance of a diesel power plant the following points should be taken care of :

1. It is necessary to maintain the *cooling temperature* within the prescribed range and use of very cold water should be avoided. The cooling water should be free from suspended impurities and suitably treated to be scale and corrosion free. If the ambient temperature approaches freezing point, the cooling water should be drained out of the engine when it is kept idle.

2. During operation the *lubrication system* should work effectively and requisite pressure and temperature maintained. The engine oil should be of the correct specifications and should be in a fit condition to lubricate the different parts. A watch may be kept on the consumption of lubricating oil as this gives an indication of the true internal condition of the engine.

3. *The engine should be periodically run even when not required to be used and should not be allowed to stand idle for more than 7 days.*

4. *Air filter, oil filters and fuel filters* should be periodically serviced or replaced as recommended by the manufacturers or if found in an unsatisfactory condition upon inspection.

5. Periodical checking of engine compression and firing pressures and also exhaust temperatures should be made.

— The engine exhaust usually provides a good indication of satisfactory performance of the engine. *A black smoke in the exhaust is a sign of inadequate combustion or engine overloading.*

— *The loss of compression resulting from wearing out of moving parts lowers the compression ratio causing inadequate combustion.* These defects can be checked by taking *indicator diagrams* of the engine after reasonable intervals.

4.21. TYPES OF DIESEL ENGINE USED FOR DIESEL POWER PLANTS

The diesel engines may be four-stroke or two stroke cycle engines. *The two-stroke cycle engines are favoured for diesel power plants.*

Efforts are being made to use "*duel fuel engines*" in diesel power plants for better economy and proper use of available gaseous fuels in the country. The gas may be a waste product as in the case of sewage treatment installations or oil fuels where the economic advantage is self evident. With the wider availability of natural gas, the duel fuel engines may become an attractive means of utilising gas as fuel at off-peak tariffs for the electric power generation.

Working of Duel Fuel Engines

The various strokes of a duel fuel engine are as follows :

1. Suction stroke. During this stroke *air and gas are drawn* in the engine cylinder.

2. Compression stroke. During this stroke the pressure of the mixture drawn is increased. Near the end of this stroke the *'pilot oil' is injected into the engine cylinder. The compression heat first ignites the pilot oil* and then gas mixture.

3. Working/power stroke. During this stroke the gases (at high temperature) expand and thus power is obtained.

4. Exhaust stroke. The exhaust gases are released to the atmosphere during the stroke.

4.22. LAYOUT OF A DIESEL ENGINE POWER PLANT

Fig. 4.32 shows the layout of a diesel engine power plant.

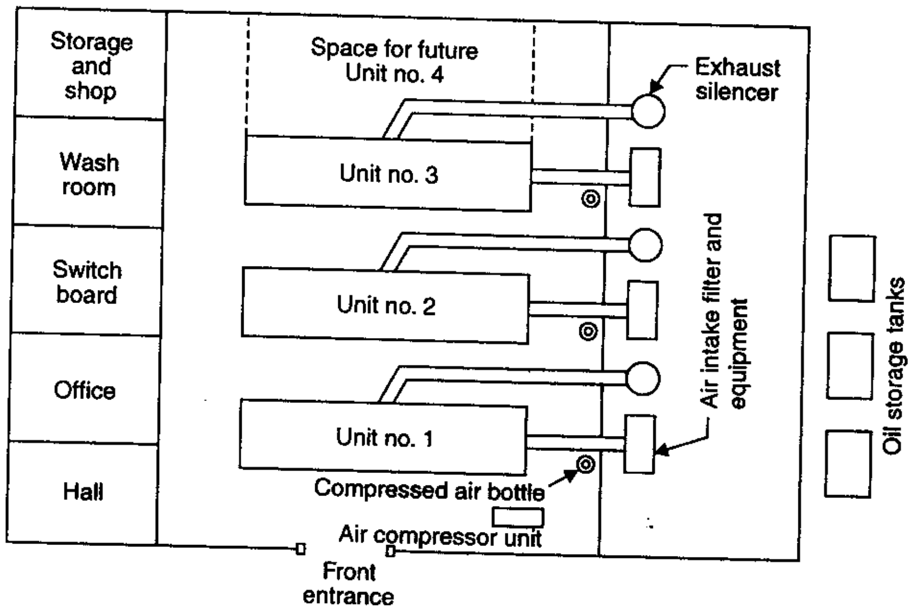


Fig. 4.32. Layout of a diesel engine power plant.

The most common arrangement for diesel engines is with parallel centre lines, with some room left for extension in future. The repairs and usual maintenance works connected with such engines necessitate sufficient space around the units and consideration should be given to the need for dismantling and removal of large components of the engine generator set. The air intakes and filters as well as the exhaust mufflers are located outside the building or may be separated from the main engine room by a partition wall. The latter arrangement is not vibration free. Adequate space for oil storage and repair shop as well as for office should be provided close to the main engine room. Bulk storage of oil may be outdoor. The engine room should be well ventilated.

4.23. PERFORMANCE OF I.C. ENGINES

Engine performance is an 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.

In evaluation of engine performance certain basic parameters are chosen and the effect of various operation conditions, design concept and modifications on these parameters are studied. The *basic performance parameters* are enumerated and discussed below :

1. Power and mechanical efficiency
2. Mean effective pressure and torque
3. Specific output
4. Volumetric efficiency
5. Fuel-air ratio
6. Specific fuel consumption
7. Thermal efficiency and heat balance
8. Exhaust smoke and other emissions
9. Specific weight.

1. Power and mechanical efficiency

(*i*) **Indicated power.** *The total power developed by combustion of fuel in the combustion chamber is called* ***indicated power.***

$$\text{I.P.} = \frac{np_{mi}\, LANk \times 10}{6} \text{ kW} \quad \text{...(4.1)}$$

where, n = Number of cylinders,

p_{mi} = Indicated mean effective pressure, bar,

L = Length of stroke, m,

A = Area of piston, m^2, and

$k = \frac{1}{2}$ for 4-stroke engine

= 1 for 2-stroke engine.

In MKS Units

$$\text{I.H.P.} = \frac{np_{mi}\, LAN\, k}{4500} \quad \text{...[4.1 (a)]}$$

where I.H.P. = Indicated horse power,

n = Number of cylinders,

p_{mi} = Indicated mean effective pressure, kgf/cm^2,

L = Length of stroke, m,

A = Area of piston, cm^2, and

$k = \frac{1}{2}$ for 4-stroke engine

= 1 for 2-stroke engine.

(*ii*) **Brake power (B.P.).** *The power developed by an engine at the output shaft is called the* ***brake power.***

$$\text{B.P.} = \frac{2\pi NT}{60 \times 1000} \text{ kW} \quad \text{...(4.2)}$$

where, N = Speed in r.p.m., and

T = Torque in N-m.

In MKS Units

$$\text{B.H.P.} = \frac{2\pi NT}{4500} \quad \text{...[4.2 (a)]}$$

where B.H.P. = Brake horse power,

N = Speed in r.p.m., and

T = Torque in kgf m.

The difference between I.P. and B.P. is called ***frictional power, F.P.***

i.e., $$\text{F.P.} = \text{I.P.} - \text{B.P.} \quad \text{...(4.3)}$$

The ratio of B.P. to I.P. is called *mechanical efficiency*

i.e., *Mechanical efficiency*, $$\eta_{mech.} = \frac{\text{B.P.}}{\text{I.P.}} \quad \text{...(4.4)}$$

In MKS Units

$$\text{F.H.P.} = \text{I.H.P.} - \text{B.H.P.} \quad \text{...[4.4 (a)]}$$

and $$\eta_{mech.} = \frac{\text{B.H.P.}}{\text{I.H.P.}} \quad \text{...[4.4 (b)]}$$

2. Mean effective pressure and torque

***Mean effective pressure** is defined as hypothetical pressure which is thought to be acting on the piston throughout the power stroke.* If it is based on I.P. it is called *indicated mean effective pressure* ($I_{m.e.p.}$ or p_{mi}) and if based on B.P. it is called *brake mean effective pressure* ($B_{m.e.p.}$ or p_{mi}). Similarly, frictional mean effective pressure ($F_{m.e.p.}$ or p_{mf}) can be defined as :

$$F_{m.e.p.} = I_{m.e.p.} - B_{m.e.p.} \quad \text{...(4.5)}$$

The torque and mean effective pressure are related by the engine size.

Since the power (P) of an engine is dependent on its size and speed, therefore it is not possible to compare engine on the basis of either power or torque. *Mean effective pressure is the true indication of the relative performance of different engines.*

3. Specific output

It is defined as the *brake output per unit of piston displacement* and is given by :

$$\text{Specific output} = \frac{\text{B.P.}}{A \times L}$$

$$= \text{Constant} \times p_{mb} \times \text{r.p.m.} \quad \text{...(4.6)}$$

For the same piston displacement and brake mean effective pressure (p_{mb}) an engine running at higher speed will give more output.

4. Volumetric efficiency

It is defined as the ratio of actual volume (reduced to N.T.P.) of the charge drawn in during the suction stroke to the swept volume of the piston.

The average value of this efficiency is from 70 to 80 per cent but in case of *supercharged engine* it may be *more* than 100 per cent, if air at about atmospheric pressure is forced into the cylinder at a pressure greater than that of air surrounding the engine.

5. Fuel-air ratio

It is the ratio of the mass of fuel to the mass of air in the fuel-air mixture.

"Relative fuel air ratio" is defined as the *ratio of the actual fuel air ratio to that of stoichiometric fuel-air ratio required to burn the fuel supplied.*

6. Specific fuel consumption (s.f.c.)

It is the mass of fuel consumed per kW developed per hour, and is a criterion of economical power production.

i.e., $$\text{s.f.c.} = \frac{m_f}{\text{B.P.}} \text{ kg/kWh.}$$

In MKS Units

s.f.c. = $\frac{m_f}{\text{B.H.P.}}$ kg/B.H.P.-hr where m_f is the fuel consumed in kg/hr

7. Thermal efficiency and heat balance

Thermal efficiency. *It is the ratio of indicated work done to energy supplied by the fuel.*

If, $\dot{m}_f$ = Mass of fuel used in kg/sec., and

C = Calorific value of fuel (lower),

Then *indicated thermal efficiency* (based on I.P.),

$$\eta_{th.(I)} = \frac{\text{I.P.}}{\dot{m}_f \times C} \qquad ...(4.7)$$

and *brake thermal efficiency* (based on B.P.)

$$\eta_{th.(B)} = \frac{\text{B.P.}}{\dot{m}_f \times C} \qquad ...(4.8)$$

In MKS Units

$$\eta_{th.(I)} = \frac{\text{I.H.P.} \times 4500}{J \times m_f \times C} \qquad ...[4.7\ (a)]$$

and $$\eta_{th.(B)} = \frac{\text{B.H.P.} \times 4500}{J \times m_f \times C} \qquad ...[4.8\ (a)]$$

Heat balance sheet

The performance of an engine is generally given by heat balance sheet.

To draw a heat balance sheet for I.C. engine, it is run at constant load. Indicator diagram is obtained with the help of an indicator. The quantity of fuel used in a given time and its calorific value, the amount, inlet and outlet temperature of cooling water and the weight of exhaust gases are recorded. After calculating I.P. (or I.H.P.) and B.P. (or B.H.P.), the heat in different items is found as follows :

Heat supplied by fuel :

For *petrol and oil engines*, heat supplied = $m_f \times C$, where m_f and C are mass used per minute (kg) and lower calorific value (kJ or kcal) of the fuel respectively.

For *gas engines*, heat supplied = $V \times C$, where V and C is volume at N.T.P. (m^3/min.) and lower calorific value of gas respectively.

(*i*) ***Heat absorbed in I.P.***

Heat equivalent of I.P. (per minute) = I.P. × 60 kJ ...(4.9)

In MKS units

Heat equivalent of I.H.P. (per minute) = $\frac{\text{I.H.P.} \times 4500}{J}$ kcal ...[4.9 (*a*)]

(*ii*) ***Heat taken away by cooling water***

If, m_w = Mass of cooling water used per minute,

t_1 = Initial temperature of cooling water,

t_2 = Final temperature of cooling water,

Then, heat taken away by water = $m_w \times c_{pw} \times (t_2 - t_1)$ ...(4.10)

where c_{pw} = Specific heat of water.

(iii) ***Heat taken away by exhaust gases***

If, m_e = Mass of exhaust gases (kg/min),

c_{pg} = Mean specific heat at constant pressure,

t_e = Temperature of exhaust gases,

t_r = Room (or boiler house) temperature,

Then, heat carried away by exhaust gases $= m_e \times c_{pg}(t_e - t_r)$ ...(4.11)

Note. The mass of exhaust gases can be obtained by adding together mass of fuel supplied and mass of air supplied.

The *heat balance sheet* from the above data can be drawn as follows :

Item	*kJ*	*Percent*
Heat supplied by fuel	...	...
(i) Heat absorbed in I.P. (or I.H.P.)	...	...
(ii) Heat taken away by cooling water	...	...
(iii) Heat carried away by exhaust gases	...	...
(iv) Heat unaccounted for (by difference)	...	...
Total	...	...

8. Exhaust smoke and other emissions

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.

9. Specific weight

It is defined as the weight of the engine in kg for each B.H.P. developed. It is an indication of the engine bulk.

10. Basic measurements

To evaluate the performance of an engine following *basic measurements* are usually undertaken :

1. Speed
2. Fuel consumption
3. Air consumption
4. Smoke density
5. Exhaust gas analysis
6. Brake power
7. Indicated power and friction power
8. Heat going to cooling water
9. Heat going to exhaust.

11. Measurement of speed

The speed may be measured by :

(i) Revolution counters

(ii) Mechanical tachometer

(iii) Electrical tachometer.

12. Fuel measurement

The fuel consumed by an engine can be measured by the following methods :

(i) Fuel flow method

(ii) Gravemetric method

(iii) Continuous flow meters.

13. Measurement of air consumption

The air consumption can be measured by the following methods :

(*i*) Air box method (*ii*) Viscous-flow air meter.

(*i*) ***Air box method***

Fig. 4.33 shows the arrangement of the system. It consists of air-tight chamber fitted with a sharp edged orifice of known co-efficient of discharge. The orifice is located away from the suction connection to the engine. Due to the suction of engine, there is a pressure depression in the air box or chamber which causes the flow through the 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. It is assumed that the intermittent suction of the engine will not effect the air pressure in the air box as volume of the box is sufficiently large, and pressure in the box remains same.

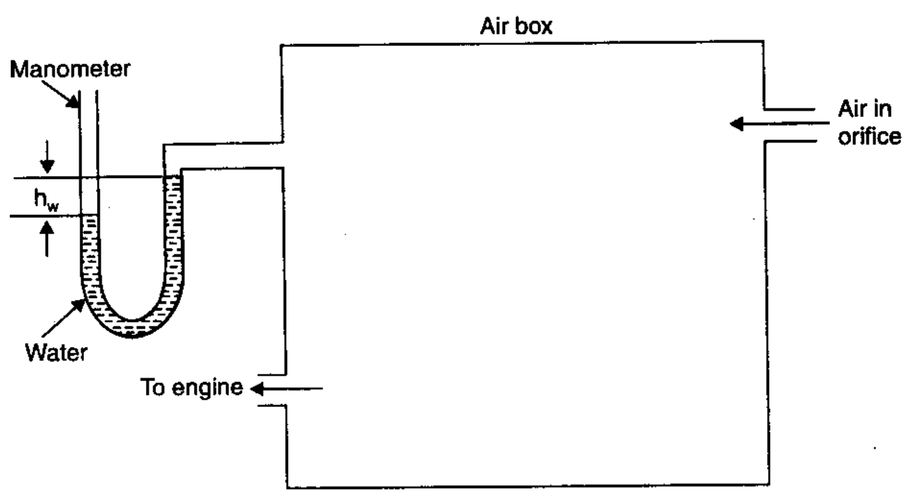


Fig. 4.33. Air-box method for measuring air.

A water manometer is used to measure the pressure difference causing the flow through the orifice. The depression across the orifice should not exceed 100 to 150 mm of water.

Let, A = Area of orifice, m^2,

d = Diameter of orifice, cm,

h_w = Head of water in cm causing the flow,

C_d = Co-efficient of discharge for orifice,

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

ρ_w = Density of water in kg/m^3.

Head in metres of air (H) is given by :

$$H.\rho_a = \frac{h_w}{100}\,\rho_w$$

$$\therefore \qquad H = \frac{h_w}{100} \times \frac{\rho_w}{\rho_a} = \frac{h_w}{100} \times \frac{1000}{\rho_a} = \frac{10h_w}{\rho_a} \text{ m of air}$$

The velocity of air passing through the orifice is given by,

$$C_a = \sqrt{2gH} \text{ m/s} = \sqrt{2g\,\frac{10h_w}{\rho_a}} \text{ m/s}$$

The volume of air passing through the orifice,

$$V_a = C_d \times A \times C_a = C_d A \sqrt{2g \frac{10h_w}{\rho_a}} = 14\, AC_d \sqrt{\frac{h_w}{\rho_a}} \text{ m}^3/\text{s}$$

$$= 840\, AC_d \sqrt{\frac{h_w}{\rho_a}} \text{ m}^3/\text{min.}$$

Mass of air passing through the orifice is given by

$$m_a = V_a \rho_a = 14 \times \frac{\pi d^2}{4 \times 100^2} \times C_d \sqrt{\frac{h_w}{\rho_a}} \times \rho_a$$

$$= 0.0011\, C_d \times d^2 \sqrt{h_w \rho_a} \text{ kg/s}$$

$$= 0.066\, C_a \times d^2 \sqrt{h_w \rho_a} \text{ kg/min.} \quad ...(4.12)$$

(*ii*) **Viscous-flow air meter**

Alcock viscous-flow air meter is another design of air meter. It is not subjected to the errors of the simple types of flow meters. With the air-box the flow is proportional to the square root of the pressure difference across the orifice. With the Alcock meter the air flows through a form of honeycomb so that flow is viscous. The *resistance of the element is directly proportional to the air velocity* and is measured by means of an inclined manometer. Felt pads are fitted in the manometer connections to damp out fluctuations. The meter is shown in Fig. 4.34.

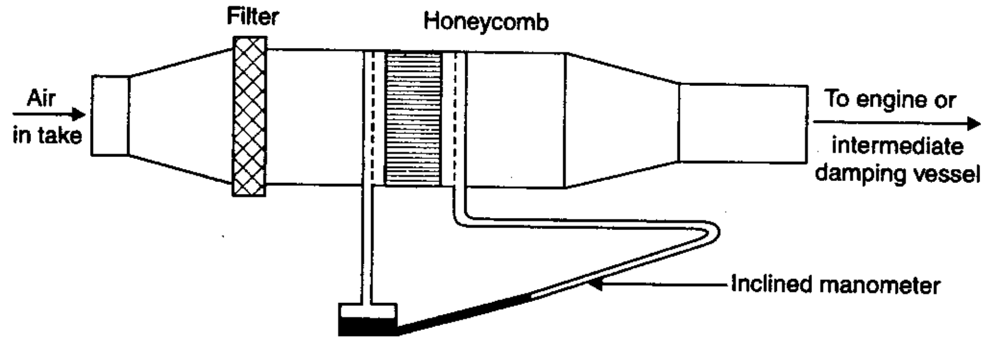


Fig. 4.34. Alcock viscous-flow air meter.

The accuracy is improved by fitting a damping vessel between the meter and the engine to reduce the effect of pulsations.

4. Measurement of exhaust smoke

The following smoke meters are used :

(*i*) Bosch smoke meter (*ii*) Hatridge smoke meter

(*iii*) PHS smoke meter.

5. Measurement of exhaust emission

Substances which are emitted to the atmosphere from any opening down stream of the exhaust part of the engine are termed as ***exhaust emissions.*** Some of the more commonly used instruments for measuring exhaust components are given below :

(*i*) Flame ionisation detector (*ii*) Spectroscopic analysers

(*iii*) Gas chromatography.

6. Measurement of B.P. (or B.H.P.)

The B.P. of an engine can be determined by a brake of some kind applied to the brake pulley of the engine. The arrangement for determination of B.P. of the engine is known as *dynamometer*. The dynamometers are classified into following two classes :

(*i*) Absorption dynamometers (*ii*) Transmission dynamometers.

(*i*) **Absorption dynamometers.** *Absorption dynamometers are those that absorb the power to be measured by friction.* The power absorbed in friction is finally dissipated in the form of heat energy.

Common forms of absorption dynamometers are :

- Prony brake
- Rope brake
- Hydraulic brake
- Fan brake
- Electrical brake dynamometers

(*a*) Eddy current dynamometer

(*b*) Swinging field d.c. dynamometer.

(*ii*) **Transmission dynamometers.** These are also called *torquemeters*. These are very accurate and are used where continuous transmission of load is necessary. These are used mainly in automatic units.

Here we shall discuss *Rope brake dynamometer* only :

Rope brake dynamometer

Refer Fig. 4.35. A rope is wound round the circumference of the brake wheel. To prevent the rope from slipping small wooden blocks (not shown in the Fig. 4.35) are laced to rope. To one end of the rope is attached a spring balance (S) and the other end carries the load (W). The speed of the engine is noted from the tachometer (revolution counter).

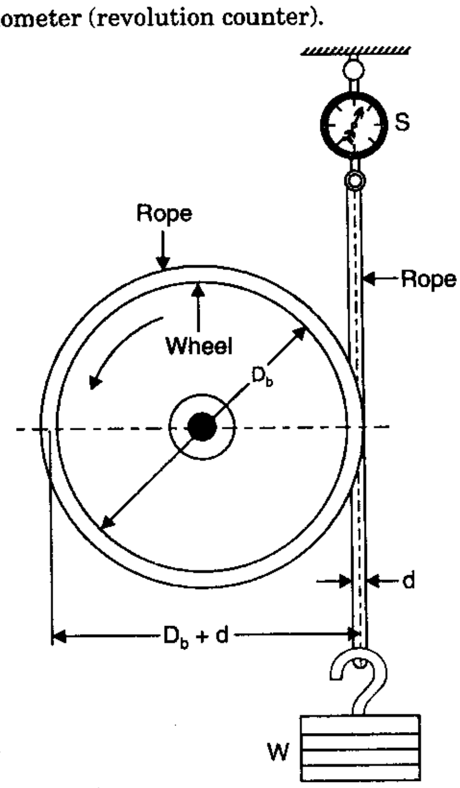


Fig. 4.35. Rope brake dynamometer.

If, W = Weight at the end of the rope N (or kg),
S = Spring balance reading, N (or kg),
N = Engine speed, r.p.m.,
D_b = Diameter of the brake wheel in meters,
d = Diameter of the rope in meters,
$(D_b + d)$ = Effective diameter of the brake wheel,

Then work/revolution = Torque × angle turned per revolution

$$= (W - S) \times \left(\frac{D_b + d}{2}\right) \times 2\pi = (W - S)(D_b + d) \times \pi$$

Work done/min. $= (W - S)\,\pi\,(D_b + d)\,N$

Work done/sec. $= \dfrac{(W - S)\,\pi\,(D_b + d)\,N}{60}$

$$\therefore \quad \text{B.P.} = \frac{(W - S)\,\pi\,(D_b + d)\,N}{60 \times 1000}\ \text{kW} \qquad ...(4.13)$$

$$= \frac{(W - S)\,\pi\,D_b N}{60 \times 1000} \quad \text{...... if } d \text{ is neglected}$$

or
$$\left(= \frac{T \times 2\pi N}{60 \times 1000}\ \text{kW}\right) \qquad ...(4.14)$$

In MKS Units

$$\text{B.H.P.} = \frac{(W - S)\,\pi(D_b + d)\,N}{4500} \qquad ...[4.13\,(a)]$$

$$= \frac{(W - S)\,\pi D_b N}{4500} \quad \text{... if } d \text{ is neglected}$$

or
$$\left(= \frac{T \times 2\pi N}{4500}\right) \qquad ...[4.14\,(a)]$$

Rope brake is cheap and easily constructed but not very accurate because of changes in friction co-efficient of the rope with temperature.

7. Measurement of Indicated power (I.P.)

The power developed in the engine cylinder or at the piston is necessarily greater than that at the crankshaft due to engine losses. Thus,

I.P. = B.P. + engine losses.

Indicated power is usually determined with the help of a p-V diagram taken with the help of an indicator. In case indicated power cannot be measured directly, it is made possible by measuring the brake power and also the engine losses. If the *indicator diagram* is available, the indicated power may be *computed by measuring the area of* diagram, either with a *planimeter* or by *ordinate method*, and dividing by the stroke measurement in order to obtain the mean effective pressure (m.e.p.).

i.e.
$$p_{mi} = \frac{\text{Net area of diagram in mm}^2}{\text{Length of diagram in mm}} \times \text{Spring constant}$$

where p_{mi} is in bar. ...(4.15)

(The spring constant is given in *bar per mm* of vertical movement of the indicator stylus.)

In MKS Units

$$p_{mi} = \frac{\text{Net area of diagram in cm}^2}{\text{Length of diagram in cm}} \times \text{Spring constant (kgf/cm}^2\text{-cm)} \qquad ...[4.15\ (a)]$$

Engine indicators

The main types of engine indicators are :

1. Piston indicator
2. Balanced diaphragm type indicator :

(*i*) The Farnborough balanced engine indicator
(*ii*) Dickinson-Newell indicator
(*iii*) MIT balanced pressure indicator
(*iv*) Capacitance-type balance pressure indicator.

3. Electrical indicators

In addition to this, *optical indicators* are also used.

Calculation of indicated power (I.P.) :

If p_{mi} = indicated mean effective pressure, bar,
A = area of piston, m^2,
L = length of stroke, m,
N = speed of the engine, r.p.m.,
$k = \frac{1}{2}$ for 4-stroke engine,
$= 1$ for 2-stroke engine.

Then, force on the piston $= p_{mi} \times A \times 10^5$ N

Work done per working stroke = Force × Length of stroke

$$= p_{mi} \times A \times 10^5 \times L \text{ N-m}$$

Work done per second = Work done per stroke × Number of working stroke per second

$$= p_{mi} \times L \times A \times 10^5 \times \frac{N}{60} \times k \text{ N-m/s or J/s}$$

$$= \frac{p_{mi} \times LANk \times 10^5}{60 \times 1000} \text{ kW}$$

Indicated power, $\text{I.P.} = \dfrac{p_{mi}\ LANk \times 10}{60 \times 1000}$ kW

If n is the number of cylinders, then

$$\text{I.P.} = \frac{np_{mi}\ LANk \times 10}{6} \text{ kW} \qquad ...(4.16)$$

In MKS Units

Calculation of I.H.P.

If p_{mi} = Indicated mean effective pressure, kgf/cm^2,
A = Area of the piston, cm^2,
L = Length of stroke, m,
N = Engine speed, r.p.m.,
$k = \frac{1}{2}$ for 4-stroke engine,
$= 1$ for 2-stroke engine,

Then, force on the piston $= p_{mi} \times A$ kgf

Work done per working stroke

$$= p_{mi} \times A \times L \text{ kgf m}$$

Work done per minute = Work done per stroke × Number of working stroke per minute

$$= p_{mi} \times A \times L \times N \times k$$

$$\therefore \quad \text{I.H.P.} = \frac{p_{mi} LANk}{4500}$$

If n is the number of cylinder, then

$$\text{I.H.P.} = \frac{np_{mi} LANk}{4500} \qquad \text{...[4.16 (a)]}$$

Morse test

This test is only applicable to *multi-cylinder engines*.

The engine is run at the required speed and the torque is measured. One cylinder is cut out, by shorting the plug if an S.I. engine is under test, or by disconnecting an injector if a C.I. engine is under test. *The speed falls because of the loss of power with one cylinder cut out, but is restored by reducing the load.* The torque is measured again when the speed has reached its original value. If the value of I.P. of the cylinders are denoted by I_1, I_2, I_3 and I_4 (considering a four-cylinder engine), and the power losses in each cylinder are denoted by L_1, L_2, L_3 and L_4, then the value of B.P., B at the test speed with all cylinders firing is given by

$$B = (I_1 - L_1) + (I_2 - L_2) + (I_3 - L_3) + (I_4 - L_4) \qquad ...(i)$$

If *number 1 cylinder is cut out,* then the contribution I_1 is lost ; and if the losses due to that cylinder remain the same as when it is firing, then the B.P., B_1, now obtained at the same speed is

$$B_1 = (0 - L_1) + (I_2 - L_2) + (I_3 - L_3) + (I_4 - L_4) \qquad ...(ii)$$

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

$$B - B_1 = I_1 \qquad ...(4.17)$$

Similarly, $B - B_2 = I_2$ when cylinder number 2 is cut out,

and $B - B_3 = I_3$ when cylinder number 3 is cut out,

and $B - B_4 = I_4$ when cylinder number 4 is cut out.

Then, for the engine,

$$I = I_1 + I_2 + I_3 + I_4 \qquad ...(4.18)$$

Measurement of frictional power (F.P.)

The frictional power of an engine can be determined by the following methods :

1. Willan's line method (used for C.I. engines only)
2. Morse test
3. Motoring test
4. Difference between I.P. and B.P.

1. *Willan's line method*

At a constant engine speed the load is reduced in increments and the corresponding B.P. and gross fuel consumption readings are taken. A graph is then drawn of fuel consumption against B.P. as in Fig. 4.36. The graph drawn in called the *Willan's line* (analogous to Willan's line for a steam engine), and is extrapolated back to cut the B.P. axis at the point L. The reading OL is taken as the power loss of the engine at that speed. The fuel consumption at zero B.P. is given by OM ; and if the relationship between fuel consumption and B.P. is assumed to be linear, then a fuel consumption OM is equivalent to a power loss of OL.

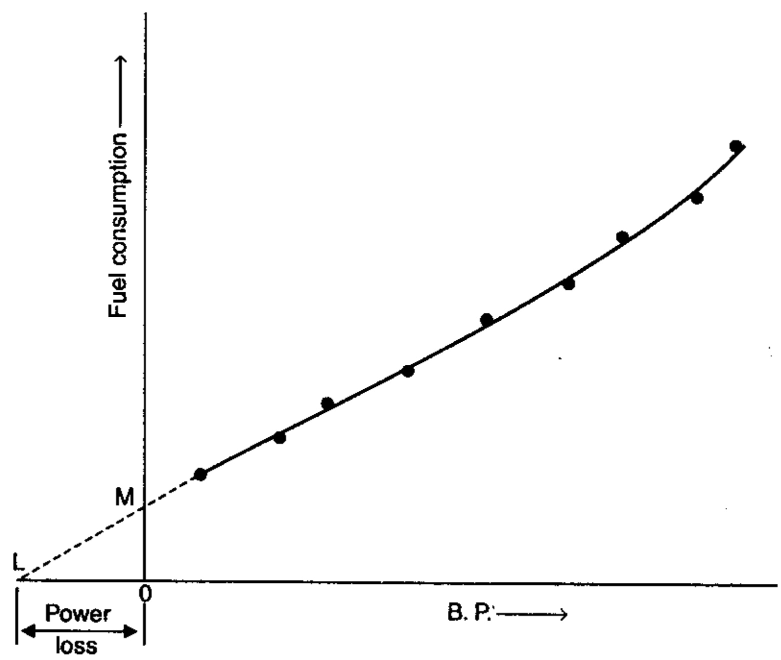


Fig. 4.36. Willian's line method.

2. *Morse test*

In 'Morse test' (already discussed), frictional power can be found by subtracting $(B.P.)_n$ from $(I.P.)_n$

i.e. $$F.P. = (I.P.)_n - (B.P.)_n$$

where n is the number of cylinders.

3. *Motoring test*

In this test the engine is first run upto the desired speed by its own power and allowed to remain under the given speed and load conditions for sometime so that oil, water and engine component temperatures reach stable conditions. The power of the engine during this period is absorbed by a dynamometer (usually of electrical type). The fuel supply is then cut off and by suitable electric switching devices the dynamometer is converted to run as a motor to drive or '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 F.P of the engine.

4. *Difference between I.P. and B.P.*

The method of finding the F.P. by finding the difference between I.P. as obtained from an indicator diagram, and B.P. as obtained by a dynamometer is the *ideal method*. However, due to difficulties in obtaining accurate indicator diagrams, especially at high engine speeds, this method is usually *only used in research laboratories and its use at commercial level is very limited.*

WORKED EXAMPLES

Example 4.1. *Following data refer to a four stroke double acting diesel engine having cylinder diameter 200 mm and piston stroke 350 mm.*

m.e.p. on cover side = *6.5 bar*

m.e.p. on crank side = *7 bar*

Speed	*= 420 r.p.m.*
Diameter of piston rod	*= 20 mm*
Dead load on the brake	*= 1370 N*
Spring balance reading	*= 145 N*
Brake wheel diameter	*= 1.2 m*
Brake rope diameter	*= 20 mm*

Calculate the mechanical efficiency of the engine.

Solution. *Given :* $P_{mi(cover)}$ = 6.5 bar, $p_{mi(crank)}$ = 7 bar, D = 0.2 m, L = 0.35 m,

N = 420 r.p.m., d_{rod} = 20 mm = 0.02 m, W = 1370 N, S = 145 N,

D_b = 1.2 m, d = 0.02 m, $k = \frac{1}{2}$ ... 4-stroke cycle engine

Mechanical efficiency ; $\eta_{mech.}$ **:**

Area of cylinder on cover end side,

$$A_{cover} = \pi/4\, D^2 = (\pi/4) \times (0.2)^2 = 0.03141 \text{ m}^2$$

Effective area of cylinder on crank end side,

$$A_{crank} = \pi/4\,(D^2 - d_{rod}^2) = \pi/4\,(0.2^2 - 0.02^2) = 0.0311 \text{ m}^2$$

Indicated power on cover end side,

$$\text{I.P.}_{(cover)} = \frac{p_{mi\,(cover)} \times LANk \times 10}{6}$$

$$= \frac{6.5 \times 0.35 \times 0.03141 \times 420 \times \frac{1}{2} \times 10}{6} = 25 \text{ kW}$$

Indicated power on crank end side,

$$\text{I.P.}_{(crank)} = \frac{p_{mi\,(crank)} \times LANk \times 10}{6}$$

$$= \frac{7 \times 0.35 \times 0.0311 \times 420 \times \frac{1}{2} \times 10}{6} = 26.67 \text{ kW}$$

Total $\qquad \text{I.P.} = \text{I.P.}_{(cover)} + \text{I.P.}_{(crank)} = 25 + 26.67 = 51.67 \text{ kW}$

Now, brake power, $\qquad \text{B.P.} = \dfrac{(W - S)\,\pi\,(D_b + d)N}{60 \times 1000} = \dfrac{(1370 - 145)\pi(1.2 + 0.02) \times 420}{60 \times 1000} \text{ kW}$

$= 32.86$ kW

Mechanical efficiency, $\eta_{mech.} = \dfrac{\text{B.P.}}{\text{I.P.}} = \dfrac{32.86}{51.67} = 0.6359 = \mathbf{63.59\%}$. **(Ans.)**

Example 4.2. *The following data refer to an oil engine working on Otto four stroke cycle :*

Brake power	*= 14.7 kW*
Suction pressure	*= 0.9 bar*
Mechanical efficiency	*= 80%*
Ratio of compression	*= 5*
Index of compression curve	*= 1.35*
Index of expansion curve	*= 1.3*
Maximum explosion pressure	*= 24 bar*

Engine speed = *1000 r.p.m.*

Ratio of stroke : bore = *1.5*

Find the diameter and stroke of the piston.

Solution. Refer Fig. 4.37.

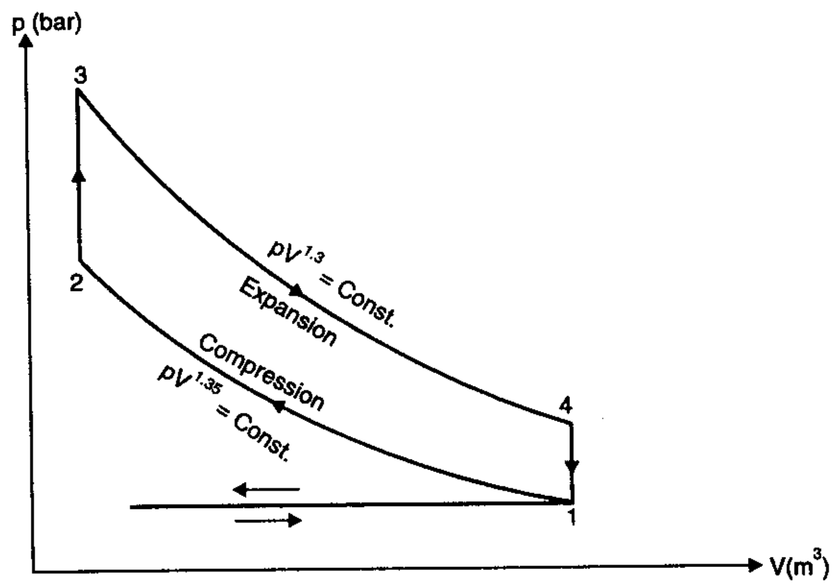


Fig. 4.37

Given : B.P. = 14.7 kW, p_1 = 0.9 bar, $\eta_{mech.}$ = 80%, r = 5, p_3 = 24 bar

$$N = 1000 \text{ r.p.m.}, \quad \frac{L}{D} = 1.5$$

$$D = ?, \quad L = ?$$

Compression ratio, $r = \dfrac{V_1}{V_2} = \dfrac{V_4}{V_3}$

To find p_2, considering *compression process 1-2,* we have

$$p_1V_1^{1.35} = p_2V_2^{1.35}$$

or
$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{1.35} = (5)^{1.35} = 8.78$$

$\therefore$
$$p_2 = p_1 \times 8.78 = 0.9 \times 8.78 = 7.9 \text{ bar}$$

To find p_4, considering *expansion process 3-4,* we have

$$p_3V_3^{1.3} = p_4V_4^{1.3}$$

or
$$\frac{p_3}{p_4} = \left(\frac{V_4}{V_3}\right)^{1.3} = (5)^{1.3} = 8.1$$

$\therefore$
$$p_4 = \frac{p_3}{8.1} = \frac{24}{8.1} = 2.96 \text{ bar}$$

Work done/cycle $= \text{Area 1-2-3-4}$

$= \text{Area under the curve 3-4} - \text{Area under the curve 1-2}$

$$= \frac{p_3V_3 - p_4V_4}{1.3 - 1} - \frac{p_2V_2 - p_1V_1}{1.35 - 1}$$

$$= \frac{p_3V_3 - p_4V_4}{0.3} - \frac{p_2V_3 - p_1V_4}{0.35} \qquad [\because V_1 = V_4 \text{ and } V_2 = V_3]$$

$$= \frac{10^5(24V_3 - 2.96V_4)}{0.3} - \frac{10^5(7.9V_3 - 0.9V_4)}{0.35}$$

$$= 10^5\,[(80V_3 - 9.86V_4) - (22.57V_3 - 2.57V_4)]$$

$$= 10^5(80V_3 - 9.86V_4 - 22.57V_3 + 2.57V_4)$$

$$= 10^5(57.43V_3 - 7.29V_4)$$

$$= 10^5(57.43V_3 - 7.29 \times 5V_3) \qquad \left[\because \frac{V_4}{V_3} = 5\right]$$

$$= 10^5 \times 20.98\, V_3 \text{ N-m.}$$

Mean effective pressure (theoretical),

$$p_m = \frac{\text{Work done / cycle}}{\text{Stroke volume } (V_s)}$$

$$= \frac{10^5 \times 20.98V_3}{(V_4 - V_3)} = \frac{10^5 \times 20.98V_3}{5V_3 - V_3} = 10^5 \times 5.245 \text{ N/m}^2 \text{ or } 5.245 \text{ bar.}$$

Now, $\eta_{mech.} = \dfrac{\text{B.P.}}{\text{I.P.}}$

$\therefore$ $\text{I.P.} = \dfrac{\text{B.P.}}{\eta_{mech.}} = \dfrac{14.7}{0.8} = 18.37 \text{ kW.}$

To find D and L :

$$\text{I.P.} = \frac{p_{mi}LANk \times 10}{6} \text{ kW}$$

$$14.7 = \frac{5.245 \times 1.5D \times \pi/4 \times D^2 \times 1000 \times \frac{1}{2} \times 10}{6}$$

$\therefore$ $$D^3 = \frac{14.7 \times 6 \times 4 \times 2}{5.245 \times 1.5 \times \pi \times 1000 \times 10} = 0.002855$$

or $D = 0.1418 \simeq$ **0.142 m or 142 mm. (Ans.)**

and $L = 1.5D = 1.5 \times 142 =$ **213 mm. (Ans.)**

Example 4.3. *The following readings were taken during the test of a single cylinder four stroke oil engine :*

Cylinder diameter	*= 250 mm*
Stroke length	*= 400 mm*
Gross m.e.p.	*= 7 bar*
Pumping m.e.p.	*= 0.5 bar*

Engine speed = *250 r.p.m.*
Net load on the brake = *1080 N*
Effective diameter of the brake = *1.5 metres*
Fuel used per hour = *10 kg*
Calorific value of fuel = *44300 kJ/kg*
Calculate : (i) Indicated power ; (ii) Brake power ;
(iii) Mechanical efficiency ; (iv) Indicated thermal efficiency.

Solution. *Given :* $D = 250$ mm $= 0.25$ m, $L = 400$ mm $= 0.4$ m, $p_{mg} = 7$ bar,

$p_{mp} = 0.5$ bar, $N = 250$ r.p.m., $D_b = 1.5$ m, $\dot{m}_f = \frac{10}{3600} = 0.00277$ kg/s

$C = 44300$ kJ/kg, $n = 1$, $(W - S) = 1080$ N

Net $p_m = p_{mg} - p_{mp} = 7 - 0.5 = 6.5$ bar.

(*i*) **Indicated power I.P. :**

$$\text{I.P.} = \frac{np_m LANk \times 10}{6} = \frac{1 \times 6.5 \times 0.4 \times \pi/4 \times 0.25^2 \times 250 \times \frac{1}{2} \times 10}{6} \text{ kW} = \mathbf{26.59\ kW.}$$

(Ans.)

(*ii*) **Brake power, B.P. :**

$$\text{B.P.} = \frac{(W - S)\pi D_b N}{60 \times 1000} \text{ kW} = \frac{1080 \times \pi \times 1.5 \times 250}{60 \times 1000} = \mathbf{21.2\ kW.} \quad \textbf{(Ans.)}$$

(*iii*) **Mechanical efficiency, $\eta_{mech.}$:**

$$\eta_{mech.} = \frac{\text{B.P.}}{\text{I.P.}} = \frac{21.2}{26.59} = \mathbf{0.797 \text{ or } 79.7\%.} \quad \textbf{(Ans.)}$$

(*iv*) **Indicated thermal efficiency, $\eta_{th(I)}$:**

$$\eta_{th(I)} = \frac{\text{I.P.}}{\dot{m}_f \times C} = \frac{26.59}{0.00277 \times 44300} = \mathbf{0.216 \text{ or } 21.6\%.} \quad \textbf{(Ans.)}$$

Example 4.4. *The brake thermal efficiency of a diesel engine is 30 per cent. If the air to fuel ratio by weight is 20 and the calorific value of the fuel used is 41800 kJ/kg, what brake mean effective pressure may be expected at S.T.P. conditions ?*

Solution. Brake thermal efficiency, $\eta_{th\,(B)} = 30\%$
Air-fuel ratio by weight = 20
Calorific value of fuel used, $C = 41800$ kJ/kg

Brake mean effective pressure, p_{mb} :

$$\text{Brake thermal efficiency} = \frac{\text{Work produced}}{\text{Heat supplied}}$$

$$0.3 = \frac{\text{Work produced}}{41800}$$

∴ Work produced per kg of fuel $= 0.3 \times 41800 = 12540$ kJ
Mass of air used per kg of fuel = 20 kg
S.T.P. conditions refer to 1.0132 bar and 15°C

$$\text{Volume of air used} = \frac{mRT}{p} = \frac{20 \times 287 \times (273 + 15)}{1.0132 \times 10^5} = 16.31 \text{ m}^3$$

Brake mean effective pressure,

$$p_{mb} = \frac{\text{Work done}}{\text{Cylinder volume}} = \frac{12540 \times 1000}{16.31 \times 10^5} = \mathbf{7.69\ bar.} \quad \textbf{(Ans.)}$$

☞**Example 4.5.** *A 2-cylinder C.I. engine with a compression ratio 13 : 1 and cylinder dimensions of 200 mm × 250 mm works on two stroke cycle and consumes 14 kg/h of fuel while running at 300 r.p.m. The relative and mechanical efficiencies of engine are 65% and 76% respectively. The fuel injection is effected upto 5% of stroke. If the calorific value of the fuel used is given as 41800 kJ/kg, calculate the mean effective pressure developed.*

Solution. Refer Fig. 4.38.

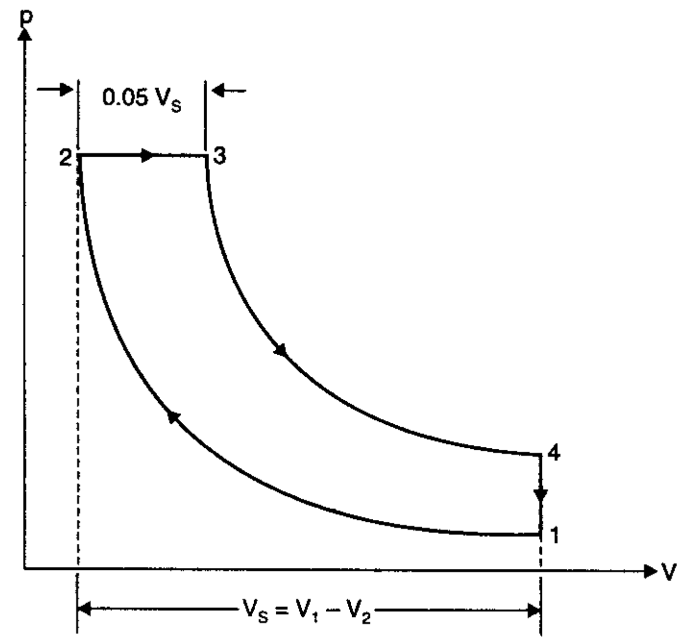


Fig. 4.38

Diameter of cylinder,	$D = 200\ \text{mm} = 0.2\ \text{m}$
Stroke length,	$L = 250\ \text{mm} = 0.25\ \text{m}$
Number of cylinders,	$n = 2$
Compression ratio,	$r = 14$
Fuel consumption	= 14 kg/h
Engine speed,	$N = 300$ r.p.m.
Relative efficiency,	$\eta_{relative} = 65\%$
Mechanical efficiency,	$\eta_{mech.} = 76\%$
Cut-off	= 5% of stroke
Calorific value of fuel,	$C = 41800$ kJ/kg
	$k = 1$ ...... for two-stroke cycle engine
Cut-off ratio,	$\rho = \frac{V_3}{V_2}$

Also $$V_3 - V_2 = 0.05V_s = 0.05(V_1 - V_2)$$

or $$V_3 - V_2 = 0.05\,(13V_2 - V_2) \qquad \left[\because \frac{V_1}{V_2} = r = 13\right]$$

or $$V_3 - V_2 = 0.05 \times 12V_2 = 0.6V_2$$

$$\therefore \quad \frac{V_3}{V_2} = 1.6$$

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

$$= 1 - \frac{1}{1.4(14)^{1.4-1}}\left[\frac{1.6^{1.4}-1}{1.6-1}\right]$$

$$= 1 - 0.248 \times 1.55 = 0.615\% \text{ or } 61.5\%$$

Also, $$\eta_{\text{relative}} = \frac{\eta_{\text{thermal}}}{\eta_{\text{air-standard}}}$$

$$0.65 = \frac{\eta_{\text{thermal}}}{0.615}$$

$$\therefore \quad \eta_{\text{thermal}} = 0.65 \times 0.615 = 0.4$$

But $$\eta_{\text{thermal (I)}} = \frac{\text{I.P.}}{\dot{m}_f \times C}$$

$$0.4 = \frac{\text{I.P.}}{\frac{14}{3600} \times 41800}$$

$$\text{I.P.} = \frac{0.4 \times 14 \times 41800}{3600} = 65 \text{ kW}$$

Now, $$\eta_{\text{mech.}} = \frac{\text{B.P.}}{\text{I.P.}}$$

$$0.76 = \frac{\text{B.P.}}{65}$$

$$\therefore \quad \text{B.P.} = 0.76 \times 65 = 49.4 \text{ kW}$$

Mean effective pressure can be calculated based on I.P. or B.P. of the engine.

$$\text{I.P.} = \frac{np_{mi}LANk \times 10}{6}$$

where p_{mi} = Indicated mean effective pressure

$$65 = \frac{2 \times p_{mi} \times 0.25 \times \pi/4 \times 0.2^2 \times 300 \times 1 \times 10}{6}$$

$$\therefore \quad p_{mi} = \frac{65 \times 6 \times 4}{2 \times 0.25 \times \pi \times 0.2^2 \times 300 \times 10} = \textbf{8.27 bar.} \quad \textbf{(Ans.)}$$

and brake mean effective pressure,

$$p_{mb} = 0.76 \times 8.27 = \textbf{6.28 bar.} \quad \textbf{(Ans.)}$$

Example 4.6. *A single cylinder 4-stroke diesel engine gave the following results while running on full load :*

Area of indicator card	$= 300\ mm^2$
Length of diagram	$= 40\ mm$
Spring constant	$= 1\ bar/mm$
Speed of the engine	$= 400\ r.p.m.$
Load on the brake	$= 370\ N$
Spring balance reading	$= 50\ N$
Diameter of brake drum	$= 1.2\ m$
Fuel consumption	$= 2.8\ kg/h$
Calorific value of fuel	$= 41800\ kJ/kg$
Diameter of the cylinder	$= 160\ mm$
Stroke of the piston	$= 200\ mm$

Calculate : (i) Indicated mean effective pressure.

(ii) Brake power and brake mean effective pressure.

(iii) Brake specific fuel consumption, brake thermal and indicated thermal efficiencies.

Solution. *Given :* $N = 400$ r.p.m., $W = 370\ N$, $S = 50\ N$, $D_b = 1.2$ m, $m_f = 2.8$ kg/h, $C = 41800$ kJ/kg, $D = 0.16$ m, $L = 0.2$ m,

$k = \frac{1}{2}$ ...... for 4 stroke cycle engine.

(*i*) **Indicated mean effective pressure, p_{mi} :**

$$p_{mi} = \frac{\text{Area of indicator diagram or card} \times \text{spring constant}}{\text{Length of diagram}}$$

$$= \frac{300 \times 1}{40} = \mathbf{7.5\ bar.} \quad \textbf{(Ans.)}$$

Indicated power, $$\text{I.P.} = \frac{np_{mi}LANk \times 10}{6} = \frac{1 \times 7.5 \times 0.2 \times \pi/4 \times 0.16^2 \times 400 \times \frac{1}{2} \times 10}{6}$$

$$= 10.05\ \text{kW}.$$

(*ii*) **B.P. :, p_{mb} :**

Brake power, $$\text{B.P.} = \frac{(W - S)\,\pi\, D_b N}{60 \times 1000} = \frac{(370 - 50)\pi \times 1.2 \times 400}{60 \times 1000} = \mathbf{8.04\ kW.} \quad \textbf{(Ans.)}$$

Also, $$\text{B.P.} = \frac{np_{mb} \times LANk \times 10}{6}$$

$$8.04 = \frac{1 \times p_{mb} \times 0.2 \times \pi/4 \times 0.16^2 \times 400 \times \frac{1}{2} \times 10}{6}$$

$$\therefore \quad p_{mb} = \frac{8.04 \times 6 \times 4 \times 2}{0.2 \times \pi \times 0.16^2 \times 400 \times 10} \simeq \mathbf{6\ bar.} \quad \textbf{(Ans.)}$$

(*iii*) **b.s.f.c. :, $\eta_{th(B)}$:, $\eta_{th.(I)}$:**

Brake specific fuel consumption,

$$b.s.f.c = \text{Fuel consumption per B.P. hour}$$

$$= \frac{2.8}{8.04} = \mathbf{0.348\ kg/B.P.\ hour.} \quad \textbf{(Ans.)}$$

Brake thermal efficiency,

$$\eta_{th.(B)} = \frac{\text{B.P.}}{\dot{m}_f \times C} = \frac{8.04}{\frac{2.8}{3600} \times 41800} = \mathbf{0.2473 \text{ or } 24.73\%.} \quad \textbf{(Ans.)}$$

Indicated thermal efficiency,

$$\eta_{th.(I)} = \frac{\text{B.P.}}{\dot{m}_f \times C} = \frac{10.05}{\frac{2.8}{3600} \times 41800} = \mathbf{0.3091 \text{ or } 30.91\%.} \quad \textbf{(Ans.)}$$

☞**Example 4.7.** *The following observations were recorded during the test on a 6-cylinder, 4-stroke Diesel engine :*

Bore	*= 125 mm*
Stroke	*= 125 mm*
Engine speed	*= 2400 r.p.m.*
Load on dynamometer	*= 490 N*
Dynamometer constant	*= 16100*
Air orifice diameter	*= 55 mm*
Co-efficient of discharge	*= 0.66*
Head causing flow through orifice	*= 310 mm of water*
Barometer reading	*= 760 mm Hg*
Ambient temperature	*= 25°C*
Fuel consumption	*= 22.1 kg/h*
Calorific value of fuel	*= 45100 kJ/kg*
Percent carbon in the fuel	*= 85%*
Percent hydrogen in the fuel	*= 15%*
Pressure of air at the end of suction stroke	*= 1.013 bar*
Temperature at the end of suction stroke	*= 25°C*

Calculate : (i) Brake mean effective pressure, (ii) Specific fuel consumption, (iii) Brake thermal efficiency, (iv) Volumetric efficiency, and (v) Percentage of excess air supplied.

Solution. *Given :* $n = 6, D = 0.125$ m, $L = 0.125$ m, $N = 2400$ r.p.m.

$W = 490\ N$, C_D = dynamometer constant = 16100

d_0 = Orifice diameter = 0.055 m, $C_d = 0.66$, $h_w = 310$ mm

$$\dot{m}_f = \frac{22.1}{3600} = 0.00614 \text{ kg/s}, C = 45100 \text{ kJ/kg},$$

$$k = \frac{1}{2} \text{ ..... for 4-stroke cycle engine.}$$

(*i*) **Brake mean effective pressure, p_{mb} :**

Brake power, $\text{B.P.} = \frac{W \times N}{C_D} = \frac{490 \times 2400}{16100} = 73 \text{ kW}$

Also $\text{B.P.} = \frac{n p_{mb} LANk \times 10}{6}$

$$73 = \frac{6 \times p_{mb} \times 0.125 \times \pi/4 \times 0.125^2 \times 2400 \times \frac{1}{2} \times 10}{6}$$

$$\therefore \quad p_{mb} = \frac{76 \times 6 \times 4 \times 2}{6 \times 0.125 \times \pi \times 0.125^2 \times 2400 \times 10} = \textbf{3.96 bar.} \quad \textbf{(Ans.)}$$

(*ii*) **Specific fuel consumption, b.s.f.c. :**

$$\text{b.s.f.c.} = \frac{22.1}{73} = \textbf{0.3027 kg/kWh.} \quad \textbf{(Ans.)}$$

(*iii*) **Brake thermal efficiency, $\eta_{th.(B)}$:**

$$\eta_{th.(B)} = \frac{\text{B.P.}}{\dot{m}_f \times C} = \frac{73}{0.00614 \times 45100} = \textbf{0.2636 or 26.36\%.} \quad \textbf{(Ans.)}$$

(*iv*) **Volumetric efficiency, $\eta_{vol.}$:**

Stroke volume of cylinder $= \pi/4\, D^2 \times L$

$$= \pi/4 \times 0.125^2 \times 0.125 = 0.00153 \text{ m}^3$$

The volume of air passing through the orifice of the air box per minute is given by,

$$V_a = 840\, A_0\, C_d \sqrt{\frac{h_w}{\rho_a}}$$

where, C_d = Discharge co-efficient of orifice = 0.66

A_0 = Area of cross-section of orifice

$= \pi/4\, d_0^2 = \pi/4 \times (0.055)^2 = 0.00237 \text{ m}^2$

h_w = Head causing flow through orifice in cm of water

$$= \frac{310}{10} = 31 \text{ cm}$$

ρ_a = Density of air at 1.013 bar and 25°C

$$= \frac{p}{RT} = \frac{1.013 \times 10^5}{287 \times (25 + 273)} = 1.18 \text{ kg/m}^3.$$

$$\therefore \quad \text{Volume of air, } V_a = 840 \times 0.00237 \times 0.66 \sqrt{\frac{31}{1.18}} = 6.73 \text{ m}^3/\text{min}$$

$\therefore$ Actual volume of air per cylinder

$$= \frac{6.73}{n} = \frac{6.73}{6} = 1.12 \text{ m}^3/\text{min}$$

$\therefore$ Air supplied per stroke per cylinder

$$= \frac{1.12}{(2400/2)} = 0.000933 \text{ m}^3$$

$$\therefore \quad \eta_{vol.} = \frac{\text{Volume of air actually supplied}}{\text{Volume of air theoretically required}}$$

$$= \frac{0.000933}{0.00153} = \textbf{0.609 or 60.9\%.} \quad \textbf{(Ans.)}$$

(*v*) **Percentage of excess air supplied :**

Quantity of air required per kg of fuel for complete combustion

$$= \frac{100}{23}\left[C \times \frac{8}{3} + H_2 \times \frac{8}{1}\right]$$

where C is the fraction of carbon and H_2 is the fraction of hydrogen present in the fuel respectively.

$$= \frac{100}{23}\left[0.85 \times \frac{8}{3} + 0.15 \times 8\right] = 15.07 \text{ kg/kg of fuel}$$

Actual quantity of air supplied per kg of fuel

$$= \frac{V_a \times \rho_a \times 60}{22.1} = \frac{6.73 \times 1.18 \times 60}{22.1} = 21.56 \text{ kg}$$

$$\therefore \text{ Percentage excess air } = \frac{21.56 - 15.07}{15.07} \times 100 = \mathbf{43.06\%.} \quad \textbf{(Ans.)}$$

Example 4.8. *In a trial of a single cylinder oil engine working on dual cycle, the following observations were made :*

Compression ratio	*= 15*
Oil consumption	*= 10.2 kg/h*
Calorific value of fuel	*= 43890 kJ/kg*
Air consumption	*= 3.8 kg/min*
Speed	*= 1900 r.p.m.*
Torque on the brake drum	*= 186 N-m*
Quantity of cooling water used	*= 15.5 kg/min*
Temperature rise	*= 36°C*
Exhaust gas temperature	*= 410°C*
Room temperature	*= 20°C*
c_p for exhaust gases	*= 1.17 kJ/kg K*

Calculate : (i) Brake power,

(ii) Brake specific fuel consumption, and

(iii) Brake thermal efficiency.

Draw heat balance sheet on minute basis.

Solution. *Given :* $n = 1, r = 15, m_f = 10.2$ kg/h, $C = 43890$ kJ/kg, $m_a = 3.8$ kg/min., $N = 1900$ r.p.m., $T = 186$ N-m, $m_w = 15.5$ kg/min,

$t_{w_2} - t_{w_1} = 36°C, t_g = 410°C, t_r = 20°C, c_p = 1.17.$

(*i*) **Brake power, B.P. :**

$$\text{B.P.} = \frac{2\pi NT}{60 \times 1000} = \frac{2\pi \times 1900 \times 186}{60 \times 1000} = \mathbf{37 \text{ kW.}} \quad \textbf{(Ans.)}$$

(*ii*) **Brake specific fuel consumption, b.s.f.c. :**

$$\text{b.s.f.c.} = \frac{10.2}{37} = \mathbf{0.2756 \text{ kg/kWh.}} \textbf{ (Ans.)}$$

(*iii*) **Brake thermal efficiency, $\eta_{th(B)}$:**

$$\eta_{th\,(B)} = \frac{\text{B.P.}}{\dot{m}_f \times C} = \frac{37}{\frac{10.2}{3600} \times 43890} = \mathbf{0.2975 \text{ or } 29.75\%.} \textbf{ (Ans.)}$$

—Heat supplied by the fuel per minute

$$= \frac{10.2}{60} \times 43890 = 7461 \text{ kJ/min}$$

(*i*) Heat equivalent of B.P.

$$= \text{B.P.} \times 60 = 37 \times 60 = 2220 \text{ kJ/min.}$$

(*ii*) Heat carried away by cooling water

$$= m_w \times c_{pw} (t_{w_2} - t_{w_1}) = 15.5 \times 4.18 \times 36 = 2332 \text{ kJ/min.}$$

(*iii*) Heat carried away by exhaust gases

$$= m_g \times c_{pg} \times (t_g - t_r)$$

$$= \left(\frac{10.2}{60} + 3.8\right) \times 1.17 \times (410 - 20) = 1811 \text{ kJ/min.}$$

Heat balance sheet (minute basis)

Item	*kJ*	*Per cent*
Heat supplied by fuel	*7461*	*100*
(*i*) Heat absorbed in B.P.	2220	29.8
(*ii*) Heat taken away by cooling water	2332	31.2
(*iii*) Heat carried away exhaust gases	1811	24.3
(*iv*) Heat unaccounted for (by difference)	1098	14.7
Total	7461	100

☞Example 4.9. *From the data given below, calculate indicated power, brake power and draw a heat balance sheet for a two stroke diesel engine run for 20 minutes at full load :*

r.p.m.	= *350*
m.e.p.	= *3.1 bar*
Net brake load	= *640 N*
Fuel consumption	= *1.52 kg*
Cooling water	= *162 kg*
Water inlet temperature	= *30°C*
Water outlet temperature	= *55°C*
Air used/kg of fuel	= *32 kg*
Room temperature	= *25°C*
Exhaust temperature	= *305°C*
Cylinder bore	= *200 mm*
Cylinder stroke	= *280 mm*
Brake diameter	= *1 metre*
Calorific value of fuel	= *43900 kJ/kg*
Steam formed per kg of fuel in the exhaust	= *1.4 kg*
Specific heat of steam in exhaust	= *2.09 kJ/kg K*
Specific heat of dry exhaust gases	= *1.0 kJ/kg K.*

Solution. *Given :* N = 350 r.p.m., p_{mi} = 3.1 bar, $(W - S)$ = 640 N, m_f = 1.52 kg, m_w = 162 kg, t_{w_1} = 30°C, t_{w_2} = 55°C, m_a = 32 kg/kg of fuel, t_r = 25°C, t_g = 305°C, D = 0.2 m, L = 0.28 m, D_b = 1 m, C = 43900 kJ/kg, c_{ps} = 2.09, c_{pg} = 1.0, k = 1 ..... for two stroke cycle engine.

(*i*) **Indicated power, I.P. :**

$$\text{I.P.} = \frac{n p_{mi} LANk \times 10}{6}$$

$$= \frac{1 \times 3.1 \times 0.28 \times \pi/4 \times 0.2^2 \times 350 \times 1 \times 10}{6} = \mathbf{15.9\ kW.}\ \ \textbf{(Ans.)}$$

(*ii*) **Brake power, B.P. :**

$$\text{B.P.} = \frac{(W - S)\,\pi\, D_b N}{60 \times 1000} = \frac{640 \times \pi \times 1 \times 350}{60 \times 1000} = \mathbf{11.73\ kW.}\ \ \textbf{(Ans.)}$$

Heat supplied in 20 minutes

$= 1.52 \times 43900 = 66728$ kJ

(*i*) Heat equivalent of I.P. in 20 minutes

$=$ I.P. $\times 60 \times 20 = 15.9 \times 60 \times 20 = 19080$ kJ

(*ii*) Heat carried away by cooling water

$= m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$

$= 162 \times 4.18 \times (55 - 30) = 16929$ kJ

Total mass of air $= 32 \times 1.52 = 48.64$ kg

Total mass of exhaust gases

= Mass of fuel + Mass of air

$= 1.52 + 48.64 = 50.16$ kg

Mass of steam formed $= 1.4 \times 1.52 = 2.13$ kg

$\therefore$ Mass of dry exhaust gases $= 50.16 - 2.13 = 48.03$ kg

(*iii*) Heat carried away by dry exhaust gases

$= m_g \times c_{pg} \times (t_g - t_r)$

$= 48.03 \times 1.0 \times (305 - 25) = 13448$ kJ

(*iv*) Heat carried away by steam

$= 2.13[h_f + h_{fg} + c_{ps}(t_{sup} - t_s)]$

$$\left[\begin{array}{l}\text{At 1.013 bar pressure (atmospheric pressure assumed) :} \\ h_f = 417.5 \text{ kJ/kg},\ h_{fg} = 2257.9 \text{ kJ/kg}\end{array}\right]$$

$= 2.13\,[417.5 + 2257.9 + 2.09\,(305 - 99.6)]$

$= 6613$ kJ/kg ..... neglecting sensible heat of water at room temperature

Heat balance sheet (20 minute basis)

Item	*kJ*	*Per cent*
Heat supplied by fuel	*66728*	*100*
(*i*) Heat equivalent of I.P.	19080	28.60
(*ii*) Heat carried away by cooling water	16929	25.40
(*iii*) Heat carried away by dry exhaust gases	13448	20.10
(*iv*) Heat carried away steam in exhaust gases	6613	9.90
(*v*) Heat unaccounted for (by difference)	10658	16.00
Total	66728	100.00

☞**Example 4.10.** *During a test on a two stroke oil engine on full load the following observations were recorded :*

Speed	= 350 r.p.m.
Net brake load	= 590 N
Mean effective pressure	= 2.8 bar
Oil consumption	= 4.3 kg/h
Jacket cooling water	= 500 kg/h
Temperature of jacket water at inlet and outlet	= 25°C and 50°C respectively
Air used per kg of oil	= 33 kg
Temperature of air in test room	= 25°C
Temperature of exhaust gases	= 400°C
Cylinder diameter	= 220 mm
Stroke length	= 280 mm
Effective brake diameter	= 1 metre
Calorific value of oil	= 43900 kJ/kg
Proportion of hydrogen in fuel oil	= 15%
Mean specific heat of dry exhaust gases	= 1.0 kJ/kg K
Specific heat of steam	= 2.09 kJ/kg K

Calculate : (i) Indicated power, and (ii) Brake power.

Also draw up heat balance sheet on minute basis.

Solution. *Given :* $n = 1$, $N = 350$ r.p.m., $(W - S) = 590$ N, $p_{mi} = 2.8$ bar

$m_f = 4.3$ kg/h, $m_w = 500$ kg/h, $t_{w_1} = 25°C$, $t_{w_2} = 50°C$

$m_a = 33$ kg/kg of oil, $t_r = 25°C$, $t_g = 400°C$, $D = 0.22$ m

$L = 0.28$ m, $D_b = 1$ m, $C = 43900$ kJ/kg, $c_{pg} = 1.0$, $c_{ps} = 2.09$

$k = 1$ ..... for two stroke cycle engine.

(*i*) **Indicated power, I.P. :**

$$\text{I.P.} = \frac{np_{mi}LANk \times 10}{6}$$

$$= \frac{1 \times 2.8 \times 0.28 \times \pi/4 \times 0.22^2 \times 350 \times 1 \times 10}{6} = \mathbf{17.38\ kW.} \quad \textbf{(Ans.)}$$

(*ii*) **Brake power, B.P. :**

$$\text{B.P.} = \frac{(W - S)\,\pi\, D_b N}{60 \times 1000} = \frac{590 \times \pi \times 1 \times 350}{60 \times 1000} = \mathbf{10.81\ kW.} \quad \textbf{(Ans.)}$$

Heat supplied per minute $= \dfrac{4.3}{60} \times 43900 = 3146$ kJ/min.

(*i*) Heat equivalent of I.P. $= 17.38 \times 60 = 1042.8$ kJ/min.

(*ii*) Heat lost to cooling water

$$= m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$$

$$= \frac{500}{60} \times 4.18 \times (50 - 25) = 870.8 \text{ kJ/min.}$$

Now, $2H_2 + O_2 = 2H_2O$

$$\begin{matrix} 4 & 32 & 36 \\ 1 & 8 & 9 \end{matrix}$$

i.e., 1 kg of H_2 produces 9 kg of H_2O

$\therefore$ Mass of H_2O produced per kg of fuel burnt

$= 9 \times H_2 \times$ mass of fuel used/min.

$$= 9 \times 0.15 \times \frac{4.3}{60} = 0.0967 \text{ kg/min.}$$

Total mass of exhaust gases (wet)/min.

= Mass of air/min. + mass of fuel/min.

$$= \frac{(33+1) \times 4.3}{60} = 2.436 \text{ kg/min.}$$

Mass of dry exhaust gases/min.

= Mass of wet exhaust gases/min – mass of H_2O produced/min.

$= 2.436 - 0.0967 = 2.339$ kg/min.

(*iii*) Heat lost to dry exhaust gases

$= m_g \times c_{pg} \times (t_g - t_r)$

$= 2.339 \times 1.0 \times (400 - 25) = 887$ kJ/min.

(*iv*) Assuming that steam in exhaust gases exists as superheated steam at atmospheric pressure and exhaust gas temperature, the enthalpy of 1 kg of steam at atmospheric pressure $1.013 \simeq 1$ bar and 400°C

$= h_{sup} - h$ (where h is the sensible heat of water at room temperature)

$= [h_f + h_{fg} + c_{ps}(t_{sup} - t_s)] - 1 \times 4.18 \times (25 - 0)$

$= [417.5 + 2257.9 + 2.09 (400 - 25)] - 104.5$

$= 3355$ kJ/min.

$\therefore$ Heat carried away by steam $= 0.0967 \times 3355 = 320.6$ kJ/min.

Heat balance sheet (minute basis)

Item	*kJ*	*Percent*
Heat supplied by fuel	*3146*	*100*
(*i*) Heat equivalent of I.P.	1042.8	33.15
(*ii*) Heat carried away by cooling water	870.8	27.70
(*iii*) Heat carried away by dry gases	887	28.15
(*iv*) Heat carried away by steam	320.6	10.20
(*v*) Heat unaccounted for (by difference)	24.8	0.80
Total	314.6	100

Example 4.11. *During a test on a Diesel engine the following observations were made :*

The power developed by the engine is used for driving a D.C. generator. The output of the generator was 210 A at 200 V ; the efficiency of generator being 82%. The quantity of fuel supplied to the engine was 11.2 kg/h ; Calorific value of fuel being 42600 kJ/kg. The air-fuel ratio was 18 : 1.

The exhaust gases were passed through a exhaust gas calorimeter for which the observations were as follows : Water circulated through exhaust gas calorimeter = 580 litres/h. Temperature rise of water through calorimeter = 36°C. Temperature of exhaust gases at exit from calorimeter = 98°C. Ambient temperature = 20°C.

Heat lost to jacket cooling water is 32% of the total heat supplied.

If the specific heat of exhaust gases be 1.05 kJ/kg K draw up the heat balance sheet on minute basis.

Solution. Output of generator = 210 A at 200 V

Generator efficiency = 82%

Fuel used = 11.2 kg/h

Calorific value of fuel = 42600 kJ/kg

Air-fuel ratio = 18 : 1

Mass of water circulated through calorimeter, m_c = 580 litres or 580 kg/h

Temperature rise of water, $t_{w_2} - t_{w_1}$ = 36°C

Temperature of exhaust gases at exit from calorimeter = 98°C

Ambient temperature = 20°C

Heat lost to jacket cooling water = 32% of the total heat supplied

Specific heat of exhaust gases = 1.05 kJ/kg K

Total power generated = $VI = 200 \times 210 = 42000$ W = 42 kW

Power available at the brakes of the engine, $\text{B.P.} = \dfrac{42}{0.82} = 51.22$ kW

Total heat supplied to the engine = Fuel supplied per min. × calorific value of fuel

$$= \frac{11.2}{60} \times 42600 = 7952 \text{ kJ/min.}$$

(*i*) Heat equivalent of B.P. $= 51.22 \times 60 = 3073$ kJ/min

Mass of exhaust gases formed per minute

$$= \text{Fuel supplied/min.} \left(\frac{A}{F} \text{ ratio} + 1\right) \qquad \left[\frac{A}{F} \text{ ratio means air-fuel ratio}\right]$$

$$= \frac{11.2}{60}(18 + 1) = 3.55 \text{ kg/min.}$$

(*ii*) Heat carried away by exhaust gases/min.

= Heat gained by water in exhaust gas calorimeter from exhaust gases + heat in exhaust gases at exit from exhaust gas calorimeter above room temperature

$$= m_w \times c_{pw} \times (t_{w_2} - t_{w_1}) + m_g \times c_{pg}(t_g - t_r)$$

$$= \frac{580}{60} \times 4.18 \times 36 + 3.55 \times 1.05\,(98 - 20)$$

$$= 1454.6 + 290.7 = 1745.3 \text{ kg/min.}$$

(*iii*) Heat lost to jacket cooling water

$$= 0.32 \times 7952 = 2544.6 \text{ kJ/min.}$$

Heat balance sheet (minute basis)

Item	*kJ*	*Percent*
Heat supplied	*7952*	*100*
(*i*) Heat equivalent of B.P.	3073	38.7
(*ii*) Heat carried away by exhaust gases	1745.3	21.9
(*iii*) Heat lost to jacket cooling water	2544.6	32.0
(*iv*) Heat unaccounted for (by difference)	589.1	7.4
Total	7952	100

☞**Example 4.12.** *During a trial of a single cylinder, 4-stroke diesel engine the following observations were recorded :*

Bore	= *340 mm*
Stroke	= *440 mm*
r.p.m.	= *400*
Area of indicator diagram	= *465 mm*^{2}
Length of diagram	= *60 mm*
Spring constant	= *0.6 bar/mm*
Load on hydraulic dynamometer	= *950 N*
Dynamometer constant	= *7460*
Fuel used	= *10.6 kg/h*
Calorific value of fuel	= *49500 kJ/kg*
Cooling water circulated	= *25 kg/min*
Rise in temperature of cooling water	= *25°C*
The mass analysis of fuel is :	
Carbon	= *84%*
Hydrogen	= *15%*
Incombustible	= *1%*
The volume analysis of exhaust gases is :	
Carbon dioxide	= *9%*
Oxygen	= *10%*
Nitrogen	= *81%*
Temperature of exhaust gases	= *400°C*
Specific heat of exhaust gases	= *1.05 kJ/kg°C*
Ambient temperature	= *25°C*
Partial pressure of steam in exhaust gases	= *0.030 bar*
Specific heat of superheated steam	= *2.1 kJ/kg°C.*

Draw up heat balance sheet on minute basis.

Solution. *Given :* $n = 1, D = 0.34$ m, $L = 0.44$ m, $N = 400$ r.p.m., $W = 950$ N,

C_d (dynamometer constant) = 7460, $m_f = 10.6$ kg/h,

$C = 49500$ kJ/kg, $m_w = 25$ kg/min., $(t_{w_2} - t_{w_1}) = 25°C$,

$t_g = 400°C$, $c_{pg} = 1.05$ kJ/kg°C, $p_{ps} = 2.1$ kJ/kg°C.

Mean effective pressure,

$$p_{mi} = \frac{\text{Area of indicator diagram} \times \text{Spring constant}}{\text{Length of indicator diagram}}$$

$$= \frac{465 \times 0.6}{60} = 4.65 \text{ bar}$$

Indicated power, $\text{I.P.} = \dfrac{np_{mi}LANk \times 10}{6}$

$$= \frac{1 \times 4.65 \times 0.44 \times \pi/4 \times 0.34^2 \times 400 \times \frac{1}{2} \times 10}{6} = 61.9 \text{ kW}$$

Brake power, $\text{B.P.} = \dfrac{W \times N}{C_d} = \dfrac{950 \times 400}{7460} = 50.9 \text{ kW}$

Friction power, F.P. = I.P. – B.P. = 61.9 – 50.9 = 11 kW

Heat supplied per minute

= Fuel used per min. × Calorific value

$$= \frac{10.6}{60} \times 49500 = 8745 \text{ kJ/min.}$$

(*i*) Heat equivalent of B.P. = B.P. × 60 = 50.9 × 60 = 3054 kJ/min.

(*ii*) Heat lost in friction = F.P. × 60 = 11 × 60 = 660 kJ/min.

(*iii*) Heat carried away by cooling water

$$= m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$$

= 25 × 4.18 × 25 = 2612.5 kJ/min.

Mass of air supplied per kg of fuel

$$= \frac{N \times C}{33(CO + CO_2)} = \frac{81 \times 84}{33(0 + 9)} = 22.9 \text{ kg}$$

Mass of exhaust gases formed per kg of fuel

= 22.9 + 1 = 23.9 kg

Mass of exhaust gases formed/min.

$$= 23.9 \times \frac{10.6}{60} \; 4.22 \text{ kg}$$

Mass of steam formed per kg of fuel

= 9 × 0.15 = 1.35 kg

∴ Mass of steam formed per min.

$$= 1.35 \times \frac{10.6}{60} = 0.238 \text{ kg/min.}$$

Mass of dry exhaust gases formed per min.

= 4.22 – 0.238 = 3.982 kg.

(*iv*) Heat carried away by dry exhaust gases/min.

$$= m_g \times c_{pg} \times (t_g - t_r)$$

= 3.982 × 1.05 × (400 – 25) = 1568 kJ/min.

Steam is carried away by exhaust gases. The temperature of steam is also the same as that of exhaust gases *e.g.* 400°C.

At partial pressure of steam 0.03 bar, the saturation temperature is 24.1°C. Therefore, steam is superheated.

Enthalpy of steam $= h_g + c_{ps}(t_{sup} - t_s)$

$= 2545.5 + 2.1\,(400 - 24.1) = 3334.89$ kJ/kg.

(*v*) $\therefore$ Heat carried by steam in exhaust gases

$= 3334.89 \times 0.238 = 793.7$ kJ/min.

(*vi*) Heat unaccounted for

= Total heat supplied—heat equivalent of B.P.
– heat lost in friction—heat carried away by cooling water
– heat carried away by dry exhaust gases
– heat carried away by steam in exhaust gases

$= 8745 - (3054 + 660 + 2612.5 + 1568 + 793.7) = 56.8$ kJ/min.

Heat balance sheet on minute basis

Item	*kJ*	*Per cent*
Heat supplied	*8745*	*100*
(*i*) Heat equivalent of B.P.	3054	34.92
(*ii*) Heat lost in friction	660	7.55
(*iii*) Heat carried away by cooling water	2612.5	29.87
(*iv*) Heat carried away by dry exhaust gases	1568	17.93
(*v*) Heat carried away by steam in exhaust gases	793.7	9.07
(*vi*) Heat unaccounted for	56.8	0.66
Total	8745	100

SUPERCHARGING

☞**Example 4.13.** *The average indicated power developed in a C.I. engine is 13 kW/m³ of free air induced per minute. The engine is a three-litre four-stroke engine running at 3500 r.p.m., and has a volumetric efficiency of 81%, referred to free air conditions of 1.013 bar and 15°C. It is proposed to fit a blower, driven mechanically from the engine. The blower has an isentropic efficiency of 72% and works through a pressure ratio of 1.72. Assume that at the end of induction the cylinders contain a volume of charge equal to the swept volume, at the pressure and temperature of the delivery from the blower. Calculate the increase in brake power to be expected from the engine.*

Take all mechanical efficiencies as 78%.

Solution. Capacity of the engine $= 3$ litres $= 0.003\ m^3$

Swept volume $= \dfrac{3500}{2} \times 0.003 = 5.25\ m^3/min.$

Unsupercharged induced volume $= 5.25 \times \eta_{vol.}$

$= 5.25 \times 0.81 = 4.25\ m^3$

Blower delivery pressure $= 1.72 \times 1.013 = 1.74$ bar

Temperature after isentropic compression

$$= 288 \times (1.72)^{(1.4-1)/1.4} = 336.3\ \text{K} \qquad \left[\because \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

$\therefore$ Blower delivery temperature $= 288 + \left(\dfrac{336.3 - 288}{0.72}\right) = 355$ K

$$\left[\because \eta_{isen.} = \frac{T_2 - T_1}{T'_2 - T_1} \text{ or } T' = T_1 + \frac{T_2 - T_1}{\eta_{isen.}}\right]$$

The blower delivery is 5.25 m^3/min at 1.74 bar and 355 K.

Equivalent volume at 1.013 bar and 15°C

$$= \frac{5.25 \times 1.74 \times 288}{1.013 \times 355} = 7.31 \text{ m}^3/\text{min.}$$

$\therefore$ Increase in induced volume = 7.31 – 4.25 = 3.06 m^3/min.

$\therefore$ Increase in indicated power from air induced

= 13 × 3.06 = 39.78 kW

Increase in I.P. due to the increased induction pressure

$$= \frac{(1.74 - 1.013) \times 10^5 \times 5.25}{10^3 \times 60} = 6.36 \text{ kW}$$

i.e., Total increase in I.P. = 39.78 + 6.36 = 46.14 kW

$\therefore$ Increase in engine B.P. $= \eta_{mech.} \times 46.14$

= 0.78 × 46.14 = 35.98 kW

From this must be *deducted the power required to drive the blower*

$$\text{Mass of air delivered by blower} = \frac{1.74 \times 10^5 \times 5.25}{60 \times 287 \times 355} = 0.149 \text{ kg/s}$$

Work input to blower $= mc_p\,(355 - 288)$

= 0.149 × 1.005 × 67

$$\therefore \text{ Power required} = \frac{0.149 \times 1.005 \times 67}{0.78} = 12.86 \text{ kW}$$

$\therefore$ *Net increase in B.P.* = 35.98 – 12.86 = **23.12 kW. (Ans.)**

Example 4.14. *A six-cylinder, four stroke CI engine is tested against a water brake dynamometer for which B.P. = WN/17 × 10^3 in kW where W is the brake load in newton and N is the speed of the engine in the r.p.m. The air consumption was measured by means of a sharp edged orifice. During the test following observations were taken :*

Bore	*= 10 cm*
Stroke	*= 14 cm*
Speed	*= 2500 r.p.m.*
Brake load	*= 480 N*
Barometer reading	*= 76 cm of Hg*
Orifice diameter	*= 3.3 cm*
Coefficient of discharge of orifice	*= 0.62*
Pressure drop across orifice	*= 14 cm of Hg*
Room temperature	*= 25°C*
Fuel consumption	*= 0.32 kg/min.*

Calculate the following :

(i) The volumetric efficiency ; (ii) The brake mean effective pressure (bmep) ; (iii) The engine torque ; (iv) The brake specific fuel consumption (bsfc). **(AMIE Summer, 2000)**

Solution. (*i*) **Volumetric efficiency, η_{vol} :**

V_s = Swept volume,

$$= \frac{\pi}{4} D^2 L \times \frac{N}{60 \times 2} \times \text{No. of cylinders, for 4-stroke.} \quad (\text{where } N = \text{r.p.m.})$$

$$= \frac{\pi}{4} (0.1)^2 \times 0.14 \times \frac{2500}{60 \times 2} \times 6 = 0.137 \text{ m}^3/\text{s}.$$

Barometer $= 76 \text{ cm Hg} = \left[\frac{76}{100} \times 13.6 \times 10^3 \times 9.81\right] \times 10^{-3} = 101.4 \text{ kN/m}^2$

$$\rho_a = \frac{p}{R_a T} = \frac{101.3}{0.287\,(273 + 25)} = 1.1844 \text{ kg/m}^3$$

$$\Delta p = 14 \text{ cm of Hg} = \frac{14}{100} \times 13.6 \times 10^3 \times 9.81 = 18.678 \times 10^3 \text{ N/m}^2$$

$$\Delta p = \rho_a \times 9.81 \times h_a,$$

where, h_a = Head, m of air, causing flow

or $$h_a = \frac{18.678 \times 10^3}{1.1844 \times 9.81} = 1607.5 \text{ m of air}$$

V_a = Volume flow rate of air, at free air conditions

$$= C_d \; \frac{\pi}{4} (d_0)^2 \sqrt{2gh_a}$$

$$= 0.62 \times \frac{\pi}{4} \left(\frac{3.3}{100}\right)^2 \sqrt{2 \times 9.81 \times 1607.5} = 0.094 \text{ m}^3/\text{s}.$$

$$\%\eta_{vol} = \frac{V_a}{V_s} \times 100 = \frac{0.094}{0.137} \times 100 = \mathbf{68.6\%.} \quad \textbf{(Ans.)}$$

(*ii*) **The brake mean effective pressure, $p_{mb.}$:**

$$\text{B.P.} = \frac{WN}{17} \times 10^{-3} \text{ kW} = \frac{480 \times 2500}{17} \times 10^{-3} = 70.588 \text{ kW}$$

$$= p_{mb} LA \times \frac{N}{60} \times \frac{1}{2} \times 6, \text{ for six cylinder, four stroke}$$

or $$p_{mb} = \frac{70.588 \times 60 \times 2}{0.14 \times \frac{\pi}{4} (0.1)^2 \times 2500 \times 6} = \mathbf{513.57 \text{ kN/m}^2.} \quad \textbf{(Ans.)}$$

(*iii*) **Engine torque, T :**

$$\text{B.P.} = 2\pi NT$$

or $$\text{Torque, } (T) = \frac{\text{B.P.}}{2\pi N} = \frac{70.588 \times 10^3}{2\pi \times \frac{2500}{60}} = \mathbf{269.63 \text{ N-m.}} \quad \textbf{(Ans.)}$$

(*iv*) **Brake specific fuel consumption, bsfc :**

$$bsfc = \frac{m_f \text{ (kg/h)}}{\text{B.P.}} = \frac{0.32 \times 60}{70.588} = \mathbf{0.272 \text{ kg/kWh.}} \quad \textbf{(Ans.)}$$

Example 4.15. *An engine is required to develop 100 kW, the mechanical efficiency of the engine is 86% and the engine uses 55 kg / h of fuel. Due to improvement in the design and operating conditions, there is reduction in engine friction to the extent of 4.8 kW. If the indicated thermal efficiency remains the same, determine the saving in fuel in kg / h.* **(AMIE Winter, 1999)**

Solution. Required brake power = 100 kW

Indicated power $= \dfrac{100}{0.86} = 116.28$ kW

$$(sfc)_{\text{indicated}} = \frac{55}{116.28} = 0.473 \text{ kg/kWh}$$

Friction power $= 116.28 - 100 = 16.28$ kW.

Given that indicated thermal efficiency remains same after improvement, the $(sfc)_{\text{ind.}}$ also remains the same.

After improvement :

Brake power $= 100$ kW

Friction power $= 16.28 - 4.8 = 11.48$ kW

Indicated power $= 100 + 11.48 = 111.48$ kW

Fuel consumption $= 111.48 \times 0.473 = 52.73$ kg / h

Saving in fuel $= 55 - 52.73 = 2.27$ kg / h

or $\dfrac{2.27}{55} \times 100 =$ **4.127%. (Ans.)**

HIGHLIGHTS

1. Any type of engine or machine which derives heat energy from the combustion of fuel or any other source and converts this energy into mechanical work is termed as a heat engine.
2. Essential components of a diesel power plant are :
   (*i*) Engine (*ii*) Air intake system
   (*iii*) Exhaust system (*iv*) Fuel system
   (*v*) Cooling system (*vi*) Lubrication system
   (*vii*) Engine starting system (*viii*) Governing system.
3. Commonly used fuel injection system in a diesel power station :
   (*i*) Common-rail injection system (*ii*) Individual pump injection system
   (*iii*) Distribution system.
4. In liquid cooling following methods are used for circulating the water around the cylinder and cylinder head :
   (*i*) Thermo-system cooling (*ii*) Forced or pump cooling
   (*iii*) Cooling with thermostatic regulator (*iv*) Pressurised cooling
   (*v*) Evaporative cooling.
5. Various lubrication systems use for I.C. engines are :
   (*i*) Wet sump lubrication system (*ii*) Dry sump lubrication system
   (*iii*) Mist lubrication system.
6. The following three are the commonly used starting systems in large and medium size engines ;
   (*i*) Starting by an auxiliary engine (*ii*) Use of electric motors or self starters
   (*iii*) Compressed air system.
7. The purpose of supercharging is to raise the volumetric efficiency above that value which can be obtained by normal aspiration.

8. **Performance of I.C. engines.** Some important relations :

(*i*) Indicated power (I.P.) $= \dfrac{np_{mi} LANk \times 10}{6}$ kW

$$\left[\begin{array}{l}\textbf{In MKS Units}\\ \text{Indicated horse power, I.H.P.} = \dfrac{np_{mi}\, LANk}{4500}\end{array}\right]$$

(*ii*) Brake (B.P.) $= \dfrac{(W - S)\pi(D_b + d)N}{60 \times 1000}$ kW

or $\left(= \dfrac{2\pi NT}{60 \times 1000}\ \text{kW}\right)$

$$\left[\begin{array}{l}\textbf{In MKS Units}\\ \text{Brake horse power, B.H.P.} = \dfrac{(W - S)\pi(D_b + d)\, N}{4500}\\ \text{or} \qquad \left(= \dfrac{2\pi NT}{4500}\right)\end{array}\right]$$

(*iii*) Mechanical efficiency, $\eta_{\text{mech.}} = \dfrac{\text{B.P. (or B.H.P.)}}{\text{I.P. (or I.H.P.)}}$

(*iv*) Thermal efficiency (indicated),

$$\eta_{\text{th. (I)}} = \frac{\text{I.P.}}{\dot{m}_f \times C}$$

and thermal efficiency (brake), $\eta_{\text{th. (B)}} = \dfrac{\text{B.P.}}{\dot{m}_f \times C}$

where m_f = mass of fuel used in kg/sec.

$$\left[\begin{array}{l}\textbf{In MKS Units}\\ \eta_{\text{th. (I)}} = \dfrac{\text{I.H.P.} \times 4500}{J \times m_f \times C}\\ \eta_{\text{th. (B)}} = \dfrac{\text{B.H.P.} \times 4500}{J \times m_f \times C}\\ \text{where } m_f = \text{Mass of fuel used in kg/min.}\end{array}\right]$$

(*v*) $\eta_{\text{relative}} = \dfrac{\eta_{\text{thermal}}}{\eta_{\text{air-standard}}}$

(*vi*) Measurement of air consumption by air box method :

Volume of air passing through the orifice,

$$V_a = 840\, AC_d \sqrt{\frac{h_w}{\rho_a}}$$

and mass of air passing through the orifice,

$$m_a = 0.066\, C_d \times d^2 \sqrt{h_w\, \rho_a} \text{ kg/min.}$$

where, A = Area of orifice, m^2,

d = Diameter of orifice, cm,

h_w = Head of water in 'cm' causing the flow, and

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

THEORETICAL QUESTIONS

1. What are the advantages and disadvantages of diesel power plants ?
2. State the applications of diesel power plant.

3. What factors should be considered while selecting a site for a diesel power plant ?
4. With the help of neat sketches give the construction and working of a four stroke diesel cycle engine.
5. List the essential components of a diesel power plant and explain them briefly.
6. Name and explain briefly various types of fuel injection systems.
7. Explain with the help of a neat sketch the working of a thermostatically controlled cooling system.
8. Explain briefly the following lubrication systems :
   (*i*) Wet sump lubrication system. (*ii*) Dry sump lubrication systems.
9. Describe briefly the commonly used starting systems in large and medium size engines.
10. Discuss briefly the basic designs of C.I. engine combustion.
11. Write a short note on "Supercharging".
12. Give the types of diesel engines used for diesel power plants.
13. Give the layout of a diesel engine power plant.

UNSOLVED EXAMPLES

1. A single cylinder petrol engine working on two stroke cycle develops indicated power of 5 kW. If the mean effective pressure is 7.0 bar and the piston diameter is 100 mm, calculate the average speed of the piston. **[Ans.** 109.1 m/s]
   [**Hint.** Average piston speed = $2LN$.]
2. A 4-cylinder petrol engine works on a mean effective pressure of 5 bar and engine speed of 1250 r.p.m. Find the indicated power developed by the engine if the bore is 100 mm and stroke 150 mm. **[Ans.** 6.11 kW]
3. A 4-cylinder four stroke S.I. engine is designed to develop 44 kW indicated power at a speed of 3000 r.p.m. The compression ratio used is 6. The law of compression and expansion is $pV^{1.3}$ = constant and heat addition and rejection takes place at constant volume. The pressure and temperature at the beginning of compression stroke are 1 bar and 50°C. The maximum pressure of the cycle is limited to 30 bar. Calculate the diameter and stroke of each cylinder assuming all cylinders have equal dimensions. Assume diagram factor = 0.8 and ratio of stroke/bore = 1.5. **[Ans.** D = 95 mm, L = 142.5 mm]
4. During the trial of a four stroke diesel engine, the following observations were recorded :
   Area of indicator diagram = 475 mm^2, length of indicator diagram = 62 mm, spring number = 1.1 bar/mm, diameter of piston = 100 mm, length of stroke = 150 mm, engine r.p.m. = 375.
   Determine : (*i*) Indicated mean effective pressure ; (*ii*) Indicated power.
   [Ans. (*i*) 8.43 bar ; (*ii*) 3.1 kW]
5. A 4-cylinder, four stroke diesel engine runs at 1000 r.p.m. The bore and stroke of each cylinder are 100 mm and 160 mm respectively. The cut off is 6.62% of the stroke. Assuming that the initial conditions of air inside the cylinder are 1 bar and 20°C, mechanical efficiency of 75%, calculate the air-standard efficiency and brake power developed by the engine.
   Also, calculate the brake specific fuel consumption if the air/fuel ratio is 20 : 1. Take R for air as 0.287 kJ/kg K and clearance volume as 0.000084 m^3. **[Ans.** 61.4%, 21.75 kW, 0.4396 kg/kWh]
6. During a trial of a two stroke diesel engine the following observations were recorded :
   Engine speed = 1500 r.p.m., load on brakes = 120 kg, length of brake arm = 875 mm.
   Determine : (*i*) Brake torque, (*ii*) Brake power. **[Ans.** (*i*) 1030 N-m ; (*ii*) 161.8 kW]
7. A four stroke gas engine develops 4.2 kW at 180 r.p.m. and at full load. Assuming the following data, calculate the relative efficiency based on indicated power and air-fuel ratio used. Volumetric efficiency = 87%, mechanical efficiency = 74%, clearance volume = 2100 cm^3, sweept volume = 9000 cm^3, fuel consumption = 5 m^3/h, calorific value of fuel = 16750 kJ/m^3. **[Ans.** 50.2%, 7.456 : 1]
8. During the trial of a four stroke cycle gas engine the following data were recorded :

Area of indicator diagram	= 565.8 mm^2
Length of indicator diagram	= 74.8 mm
Spring index	= 0.9 bar/mm

Cylinder diameter	= 220 mm
Stroke length	= 430 mm
Number of explosions/min	= 100

Determine : (*i*) Indicated mean effective pressure; (*ii*) Indicated power.

[Ans. (*i*) 6.8 bar ; (*ii*) 18.5 kW]

9. The following observations were recorded during a trial of a four stroke engine with rope brake dynamometer :

Engine speed = 650 r.p.m., diameter of brake drum = 600 mm, diameter of rope = 50 mm, dead load on the brake drum = 32 kg, spring balance reading = 4.75 kg.

Calculate the brake power. **[Ans.** 5.9 kW]

10. The following data refer to a four stroke petrol engine :

Engine speed = 2000 r.p.m., ideal thermal efficiency = 35%, relative efficiency = 80%, mechanical efficiency = 85%, volumetric efficiency = 70%.

If the engine develops 29.42 kW brake power calculate the cylinder swept volume. **[Ans.** 0.00185 m^3]

11. A single cylinder four-stroke gas engine has a bore of 178 mm and a stroke of 330 mm and is governed by hit and miss principle. When running at 400 r.p.m. at full load, indicator cards are taken which give a working loop mean effective pressure of 6.2 bar, and a pumping loop mean effective pressure of 0.35 bar. Diagrams from the dead cycle give a mean effective pressure of 0.62 bar. The engine was run light at the same speed (*i.e.* with no load), and a mechanical counter recorded 47 firing strokes per minute.

Calculate : (*i*) Full load brake power, (*ii*) Mechanical efficiency of the engine.

[Ans. (*i*) 13.54 kW ; (*ii*) 84.7%]

12. During a 60 minutes trial of a single cylinder four stroke engine the following observations were recorded :

Bore = 0.3 m, stroke = 0.45 m, fuel consumption = 11.4 kg, calorific value of fuel = 42000 kJ/kg, brake mean effective pressure = 6.0 bar, net load on brakes = 1500 N, r.p.m. = 300, brake drum diameter = 1.8 m, brake rope diameter = 20 mm, quantity of jacket cooling water = 600 kg, temperature rise of jacket water = 55°C, quantity of air as measured = 250 kg, exhaust gas temperature = 420°C, c_p for exhaust gases = 1 kJ/kg K, ambient temperature = 20°C.

Calculate :

(*i*) Indicated power ; (*ii*) Brake power ;

(*iii*) Mechanical efficiency ; (*iv*) Indicated thermal efficiency.

Draw up a heat balance sheet on minute basis.

[Ans. (*i*) 47.7 kW, (*ii*) 42.9 kW, (*iii*) 89.9%, (*iv*) 35.86%]

13. A quality governed four-stroke, single cylinder gas engine has a bore of 146 mm and a stroke of 280 mm. At 475 r.p.m. and full load the net load on the friction brake is 433 N, and the torque arm is 0.45 m. The indicator diagram gives a net area of 578 mm^2 and a length of 70 mm with a spring rating of 0.815 bar/mm.

Calculate :

(*i*) The indicated power (*ii*) Brake power (*iii*) Mechanical efficiency.

[Ans. (*i*) 12.5 kW (*ii*) 9.69 kW (*iii*) 77.5%]

14. A two-cylinder four stroke gas engine has a bore of 380 mm and a stroke of 585 mm. At 240 r.p.m. the torque developed is 5.16 kN- m.

Calculate :

(*i*) Brake power (*ii*) Mean piston speed in m/s

(*iii*) Brake mean effective pressure. **[Ans.** (*i*) 129.8 kW (*ii*) 4.68 m/s ; (*iii*) ; 4.89 bar]

15. The engine of Problem 14 is supplied with a mixture of coal gas and air in the proportion of 1 to 7 by volume. The estimated volumetric efficiency is 85% and the calorific value of the coal gas is 16800 kJ/m^3. Calculate the brake thermal efficiency of the engine. **[Ans.** 27.4%]

16. A 4-cylinder, four-stroke diesel engine has a bore of 212 mm and a stroke of 292 mm. At full load at 720 r.p.m., the b.m.e.p. is 5.93 bar and the specific fuel consumption is 0.226 kg/kWh. The air/fuel ratio as determined by exhaust gas analysis is 25 : 1. Calculate the brake thermal efficiency and volumetric efficiency of the engine.

Atmospheric conditions are 1.01 bar and 15°C and calorific value for the fuel may be taken as 44200 kJ/kg. [**Ans.** 36% ; 76.5%]

17. A 4-cylinder petrol engine has an output of 52 kW at 2000 r.p.m. A Morse test is carried out and the brake torque readings are 177, 170, 168 and 174 N-m respectively. For normal running at this speed the specific fuel consumption is 0.364 kg/kWh. The calorific value of fuel is 44200 kJ/kg.

Calculate : (*i*) Mechanical efficiency ; (*ii*) Brake thermal efficiency of the engine.

[**Ans.** (*i*) 82% ; (*ii*) 22.4%]

18. A V-8 four stroke petrol engine is required to give 186.5 kW at 440 r.p.m. The brake thermal efficiency can be assumed to be 32% at the compression ratio of 9 : 1. The air/fuel ratio is 12 : 1 and the volumetric efficiency at this speed is 69%. If the stroke to bore ratio is 0.8, determine the engine displacement required and the dimensions of the bore and stroke. The calorific value of the fuel is 44200 kJ/kg, and the free air conditions are 1.013 bar and 15°C. [**Ans.** 5.12 litres ; 100.6 mm ; 80.5 mm]

19. During the trial (60 minutes) on a single cylinder oil engine having cylinder diameter 300 mm, stroke 450 mm and working on the four stroke cycle, the following observations were made :

Total fuel used = 9.6 litres, calorific value of fuel = 45000 kJ/kg, total number of revolutions = 12624, gross indicated mean effective pressure = 7.24 bar, pumping i.m.e.p. = 0.34 bar, net load on the brake = 3150 N, diameter of brake wheel drum = 1.78 m, diameter of the rope = 40 mm, cooling water circulated = 545 litres, cooling water temperature rise = 25°C, specific gravity of oil = 0.8.

Determine : (*i*) Indicated power.

(*ii*) Brake power. (*iii*) Mechanical efficiency.

Draw up the heat balance sheet on minute basis. [**Ans.** (*i*) 77 kW ; (*ii*) 61.77 kW ; (*iii*) 80.22%]

20. The following results were obtained on full load during a trial on a two stroke oil engine :

Engine speed	= 350 r.p.m.
Net brake load	= 600 N
m.e.p.	= 2.75 bar
Oil consumption	= 4.25 kg/h
Temperature rise of jacket cooling water	= 25°C
Air used per kg of oil	= 31.5 kg
Temperature of air in test room	= 20°C
Temperature of exhaust gases	= 390°C
Following data also apply to the above test :	
Cylinder diameter	= 220 mm
Stroke	= 280 mm
Effective brake diameter	= 1 metre
Calorific value of oil	= 45000 kJ/kg
Proportion of hydrogen in fuel oil	= 15%
Partial pressure of steam in exhaust gases	= 0.04 bar
Mean specific heat of exhaust gases	= 1.0 kJ/kg K
Specific heat of superheated steam	= 2.1 kJ/kg K
Specific heat of water	= 4.186 kJ/kg K

Determine : (*i*) Indicated power.

(*ii*) Brake power. (*iii*) Mechanical efficiency.

Draw up heat balance sheet for the test. [**Ans.** (*i*) 17.1 kW ; (*ii*) 11 kW ; (*iii*) 64.33%]

21. A 4-cylinder, four-stroke diesel engine develops 83.5 kW at 1800 r.p.m. with specific fuel consumption of 0.231 kg/kWh, and air/fuel ratio of 23 : 1. The analysis of fuel is 87% carbon and 13% hydrogen, and the calorific value of the fuel is 43500 kJ/kg. The jacket cooling water flows at 0.246 kg/s and its temperature rise is 50 K. The exhaust temperature is 316°C. Draw up an energy balance for the engine. Take R = 0.302 kJ/kg K and c_p = 1.09 kJ/kg K for the dry exhaust gases and c_p = 1.86 kJ/kg K for superheated steam. The temperature in the test house is 17.8°C, and the exhaust gas pressure is 1.013 bar.

[**Ans.** B.P. = 35.8%, cooling water = 22.1%, exhaust = 24%, radiation and unaccounted = 16.7%]

COMPETITIVE EXAMINATIONS QUESTIONS

1. (*a*) State and explain the factors which are required to be considered in the choice of diesel engine for a diesel power plant.

(*b*) With the help of diagrams, show the general shape of (*i*) fuel consumption (kg/kWh) *versus* percentage rated load, and (*ii*) energy produced per unit weight of fuel (kWh/kg) *versus* percentage capacity factor characteristic of diesel engines. How is the rating of a diesel engine effected by the altitude above sea-level ?

(*c*) Define (*i*) the thermal efficiency, in the terms of indicated horse-power, weight of fuel etc., and (*ii*) the brake power in kW of a diesel engine.

2. (*a*) Why is supercharging necessary in diesel power plant ? What methods are used for supercharging the diesel engines ?

(*b*) Draw a neat diagram of a fuel storage and fuel supply system used for diesel power plants. What are the advantages of underground fuel storage ?

3. (*a*) Under what conditions diesel generating plants are preferred ?

(*b*) On what factors is the size of the generating plant selected ?

(*c*) Draw the neat diagram of a cooling system used for diesel power plant showing all the essential components. What are the advantages of double circuit over single circuit system ? What precautions should be taken to ensure that cooling is satisfactory ?

4. For a diesel power station discuss briefly about the following :

(*a*) Cooling system ; (*b*) Lubricating system ; (*c*) Supercharging system.

5. (*a*) Under what conditions diesel power plants are selected in preference to steam power plants ? Explain giving suitable examples.

(*b*) Describe the working of a gas turbine plant with the help of T-ϕ chart. How can the efficiency be improved ?

6. (*a*) Draw line diagram to show the layout of diesel power plant and describe it in brief.

(*b*) State the factors to be considered for selection of prime movers in a medium sized power plant.

7. (*a*) Explain the starting and stopping procedure in a diesel power plant.

(*b*) Make a layout of a modern diesel power plant showing the following systems :

(*i*) Air intake system ; (*ii*) Cooling system ;
(*iii*) Fuel supply system ; (*iv*) Lubrication system ;
(*v*) Exhaust system.

8. (*a*) What are the advantages of supercharging ? Explain the methods used for supercharging diesel engines.

(*b*) Calculate the cooling water required in litres per minute from the following performance data available for a particular diesel engine :

Brake horse-power	= 4500
Jacket loss	= 23%
Water temperature rise	= 35°C
Mechanical efficiency	= 83%
Indicated thermal efficiency	= 34%

9. (*a*) Give a general layout of a diesel engine power plant showing clearly all the essential circuits.

(*b*) A four cylinder 40 cm diameter and 60 cm stroke, four stroke diesel engine has a fuel consumption of 0.20 kg per I.H.P. hour based on 10,000 kcal/kg oil. Engine speed is 275 r.p.m. with an indicated mean effective pressure of 8.5 kgf/cm^2. Jacket water carries away 24% of the heat supplied. Calculate the quantity of water in kg/hr needed to maintain an outlet temperature of jacket water at 50°C. Temperature of atmosphere is 20°C.

Calculate also the indicated thermal efficiency of this engine.

10. The following data is obtained during a test on a 4-stroke, 4-cylinder diesel engine :

Cylinder diameter	= 35 cm
Stroke	= 40 cm

Speed of the engine	= 315 r.p.m.
Indicated m.e.p.	= 7 kgf/cm^2
B.H.P. of the engine	= 330
Fuel consumption	= 80 kg/hour
Calorific value of fuel	= 10500 kcal/kg
Hydrogen content in hydrocarbon fuel	= 13%
Air consumption	= 30 kg/min
Cooling water circulated	= 90 kg/min
Rise in temperature of cooling water	= 38°C
Exhaust gas temperature	= 324°C
Specific heat of air	= 0.24
Specific heat of exhaust gases	= 0.26
Ambient temperature	= 24°C
Specific heat for superheated steam	= 0.5
Partial pressure of steam in exhaust gases	= 0.030 kgf/cm^2

Find :

(*i*) Mechanical efficiency, indicated thermal efficiency and specific fuel consumption on I.H.P. basis.
(*ii*) Cost of energy per kWh assuming cost of fuel as Rs. 2.50 per litre and specific gravity of fuel as 0.93.
(*iii*) Draw up the heat balance on minute and percentage basis.

Find : (*i*) The fixed cost of power generation per kW per annum ; (*ii*) The total cost of power generation per kWh. Cost of primary distribution is chargeable to generation.

11. Write brief notes on the following :
(*a*) Lubrication system in diesel power plant ; (*b*) Intercooling in gas turbine plant ;
(*c*) Earthing of a power system ; (*d*) Flat plate solar collectors.

12. What are the advantages and disadvantages of diesel power plant over steam power plant ? What is the status of diesel power plants in our country ?

13. (*a*) Make a neat layout line diagram of a diesel power plant showing in particular the fuel system and the water cooling system.
(*b*) What are the primary advantages of a diesel power plant ?
(*c*) Cooling water from a 400 kW diesel power plant enters a cooling tower at 60°C and leaves the tower at 37.8°C in an atmosphere with a dry bulb temperature of 32.2°C and a wet bulb temperature of 27.8°C. If thermal efficiency of the plant is 30% and 32% of the energy input is lost to the cooling water, estimate the kg/min of cooling water handled by the tower and the cooling efficiency of the tower.

5

Gas Turbine Power Plants

5.1. GAS TURBINES—GENERAL ASPECTS

Probably a windmill was the first turbine to produce useful work, wherein there is no pre-compression and no combustion. The characteristic features of a gas turbine as we think of the name today include a *compression process* and a *heat-addition* (or combustion) process. The gas turbine represents perhaps the *most satisfactory way of producing very large quantities of power in a self-contained and compact unit.* The gas turbine may have an ample future use in conjunction with the oil engine. For smaller gas turbine units, the inefficiencies in compression and expansion processes become greater and to improve the thermal efficiency it is necessary to use a heat exchanger. In order that a small gas turbine may compete for economy with the small oil engine or petrol engine it is necessary that a compact effective heat exchanger be used in the gas turbine cycle. The thermal efficiency of the gas turbine alone is still quite modest 20 to 30% compared with that of a modern steam plant 38 to 40%. It is possible to construct *combined plants* whose efficiencies are of order of 45% or more. Higher efficiencies might be attained in future.

The following are the major fields of application of gas turbines :

1. Aviation
2. Power generation
3. Oil and gas industry
4. Marine propulsion.

The efficiency of a gas turbine is not the criteria for the choice of this plant. A gas turbine is used in *aviation* and marine fields because *it is self contained, light weight not requiring cooling water and generally fit into the overall shape of the structure.* It is selected for '*power generation*' because of its *simplicity, lack of cooling water, needs quick installation and quick starting*. It is used in *oil and gas industry* because of *cheaper supply* of *fuel and low installation cost.*