B. Tech. (VII sem) Exam, 2013 I.P. Engg.

Sub: – I.C.E.

Section - A

Q1. (i) a (vi) b (xi) a (xvi) b

(ii) a (vii) b (xii) d (xvii) d

(iii) c (viii) d (xiii) a (xviii) d

(iv) b (ix) a (xiv) b (xix) d

(v) d (x) a (xv) b (xx) c

Q2. (a)

Four-Stroke Engine	Two-Stroke Engine
The thermodynamic 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 thermodynamic 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.
Because of the above, turning moment is not so uniform and hence a heavier flywheel is needed.	Because of the above, turning moment is more uniform and hence a lighter flywheel can be used.
Again, because of one power stroke for two revolutions, power produced for same size of engine is less, or for the same power the engine is heavier and bulkier.	Because of one power stroke for every revolution, power produced for same size of engine is twice, or for the same power the engine is lighter and more compact.
Because of one power stroke in two revolutions lesser cooling and lubrication requirements. Lower rate of wear and tear.	Because of one power stroke in one revolution greater cooling and lubrication requirements. Higher rate of wear and tear.
Four-stroke engines have valves and valve actuating mechanisms for opening and closing of the intake and exhaust valves.	Two-stroke engines have no valves but only ports (some two-stroke engines are fitted with conventional exhaust valve or reed valve).
Because of comparatively higher weight and complicated valve mechanism, the initial cost of the engine is more.	Because of light weight and simplicity due to the absence of valve actuating mechanism, initial cost of the engine is less.
Volumetric efficiency is more due to more time for induction.	Volumetric efficiency is low due to lesser time for induction.
Thermal efficiency is higher; part load efficiency is better.	Thermal efficiency is lower; part load efficiency is poor.
Used where efficiency is important, viz., in cars, buses, trucks, tractors, industrial engines, aeroplanes, power generation etc.	Used where low cost, compactness and light weight are important, viz., in mopeds, scooters, motorcycles, hand sprayers etc.

2(b)

FOUR-STROKE COMPRESSION IGNITION (CI) ENGINES

The four-stroke CI engine is similar to four-stroke SI engine except that a high compression ratio is used in the former, and during the suction stroke, air alone, instead of a fuel-air mixture, is inducted. Due to high compression ratio, the temperature at the end of compression stroke is sufficient to ignite the fuel which is injected into the combustion chamber. In the CI engine a high pressure fuel pump and an injector is provided to inject fuel into combustion chamber. The carburettor and ignition system, necessary in the SI engine, are not required in the CI engine.

The ideal sequence of operation for the four-stroke CI engine is as follows:

1. *Suction stroke*. Only air is inducted during the suction stroke. During this stroke intake valve is open and exhaust valve is closed.

2. *Compression stroke*. Both valves remain closed during compression stroke.

3. *Expansion or power stroke*. Fuel is injected in the beginning of the expansion stroke. The rate of injection is such that the combustion maintains the pressure constant. After the injection of fuel is over (*i.e.* after fuel cut off) the products of combustion expand. Both valves remain closed during expansion stroke.

4. *Exhaust stroke*. The exhaust valve is open and the intake valve remains closed in the exhaust stroke.

Due to higher pressures the CI engine is heavier than SI engine but has a higher thermal efficiency because of greater expansion. CI engines are mainly used for heavy transport vehicles, power generation, and industrial and marine applications.

The typical valve timing diagram for a four-stroke CI engine is as follows:

IVO up to 30° before TDC
IVO up to 50° after BDC
EVO about 45° before BDC
EVO about 30° after TDC
Injection about 15° before TDC

2 (c) Given $D = 250 mm, L = 375 mm$

$V_c = 0.00263 m^3$

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

$= \frac{\pi}{4} \times 0.25^2 \times 0.375$

$V_s = 0.0183 m^3$

$r_k = \frac{V_s + V_c}{V_c} = \frac{0.0183 + 0.00263}{0.00263}$

$= 7.99 \approx 8$

$r_k = 8$

$\eta = 1 - \frac{1}{r_k^{\gamma - 1}} = 1 - \frac{1}{8^{0.4}}$

$\eta = 56.472\%$

$V_1 = V_c + V_s = 0.02093$

$mep = p_m = \frac{\text{Work done}}{V_s}$

$= \frac{P(V_1 - V_2)}{V_s} =$

$= 695.82 \; kN/m^2$

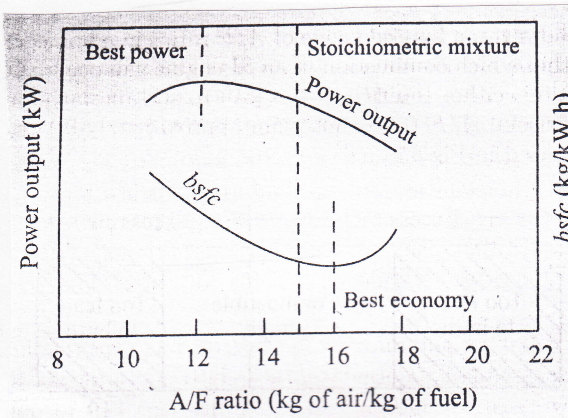
$= 695.82 \; kPa$ Ans

(a).

MIXTURE REQUIREMENTS AT DIFFERENT LOADS AND SPEEDS

The air-fuel ratio at which an engine operates has a considerable influence on its performance. Consider an engine operating at full throttle and constant speed with varying A/F ratio. Under these conditions, the A/F ratio will affect both the power output and the brake specific fuel consumption, as indicated by the typical curves shown in Fig.8.2. The mixture corresponding to the maximum output on the curve is called the *best power mixture* with an A/F ratio of approximately 12:1. The mixture corresponding to the minimum point on the $bsfc$ curve is called the *best economy mixture*. The A/F ratio is approximately 16:1. It may be noted that the best power mixture is much richer than the chemically correct mixture and the best economy mixture is slightly leaner than the chemically correct.

Figure 8.2 is based on full throttle operation. The A/F ratios for the best power and best economy at part throttle are not strictly the same as at full throttle. If the A/F ratios for best power and best economy are constant over the full range of throttle operation and if the influence of other factors is disregarded, the ideal fuel metering device would be merely a two position carburetor. Such a carburetor could be set for the best power mixture when maximum performance is desired and for ~~the best economy~~ mixture when the primary consideration is the fuel economy. These two settings are indicated in Fig.8.3 by the solid horizontal lines X–X′ and Z–Z′, respectively. Actual engine requirements, however, again preclude the use of such a simple and convenient arrangement. These requirements are discussed in the succeeding section.



Variation of Power Output and bsfc with A/F Ratio for an SI Engine

Under normal conditions it is desirable to run the engine on the maximum economy mixture, viz., around 16:1 air-fuel ratio. For quick acceleration and for maximum power, rich mixture, viz., 12:1 air-fuel ratio is required.

spark ignition engine a sufficiently homogeneous mixture of vapourized fuel, air and residual gases is ignited by a single intense and high temperature spark between the spark plug electrodes (at the moment of discharge the temperature of electrodes exceeds 10,000°C), leaving behind a thin thread of flame. From this thin thread combustion spreads to the envelop of mixture immediately surrounding it at a rate which depends primarily upon the temperature of the flame front itself and to a secondary degree, upon both the temperature and the density of the surrounding envelope.

In this manner there grows up, gradually at first, a small hollow nucleus of flame, much in the manner of a soap bubble. If the contents of the cylinder were at rest, this flame bubble would expand with steadily increasing speed until extended throughout the whole mass. In the actual engine cylinder, however, the mixture is not at rest. It is, in fact, in a highly turbulent condition. The turbulence breaks the filament of flame into a ragged front, thus presenting a far greater surface area from which heat is radiated; hence its advance is speeded up enormously. The rate at which the flame front travels is dependent primarily on the degree of turbulence, but its general direction of movement, that of radiating outward from the ignition point, is not much affected.

The theoretical diagram of combustion is shown in Fig. 5.2 (a) but the actual process is different. According to Ricardo the combustion can be imagined as if developing in two stages, one – the growth and development of a semipropagating nucleus of flame called *ignition lag or preparation phase*, and the other, the spread of the flame throughout the combustion

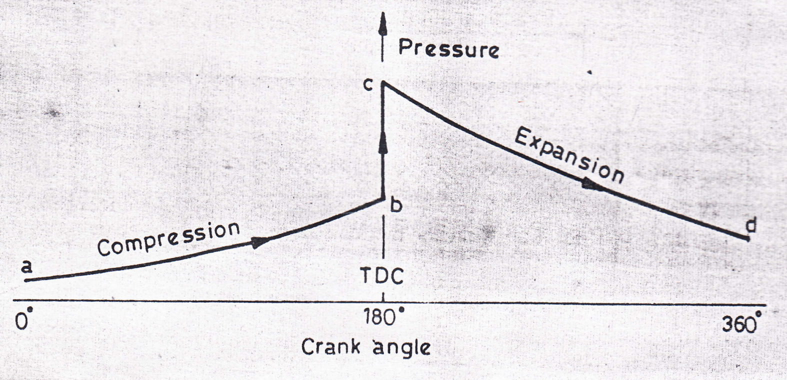


Fig. 5.2 (*a*). Theoretical p-θ diagram.

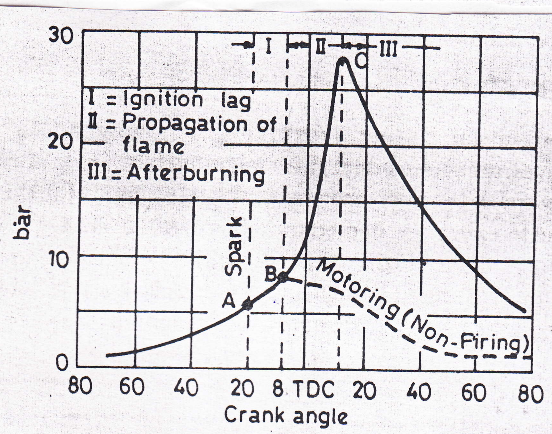


Fig. 5.2 (*b*). Stages of combustion in SI engine.

chamber [see Fig. 5.2 (b)] ... upon the nature of the fuel, upon both temperature and pressure, the proportion of the exhaust gas, and also upon the temperature coefficient of the fuel, that is, the relationship between temperature and rate of acceleration of oxidation or burning. The second stage is a mechanical one, pure and simple. The two stages are not entirely distinct, since the nature and velocity of combustion change gradually. The starting point of the second stage is where first measurable rise of pressure can be seen on the indicator diagram, *i.e.*, the point where the line of combustion departs from the compression line. In Fig. 5.2 (*b*), *A* shows the point of passage of spark (say 28° before *TDC*), *B* the point at which the first rise of pressure can be detected (say, 8° before *TDC*) and *C* the attainment of peak pressure. Thus *AB* represents the first stage (about 20° crank angle rotation) and *BC* the second stage. Although the point *C* makes the completion of the flame travel, it does not follow that at this point the whole of the heat of the fuel has been liberated, for even after the passage of the flame, some further chemical adjustments due to reassociation, etc., and what is generally referred to as *afterburning*, will to a greater or less degree continue throughout the expansion stroke.

The first stage *AB*, by analogy with diesel engines is called ignition lag, which label is wrong in principle. In spark ignition there is practically no ignition lag and a nucleus of combustion arises instantaneously near the spark plug electrodes. But during the initial period flame front spreads very slowly and the fraction of burnt mixture is small so that an increase of pressure cannot be detected on the indicator diagram. The increase of pressure may be just one per cent of maximum combustion pressure corresponding to burning of about 1.5 per cent of the working mixture, and the volume occupied by the combustion products may be about 5 per cent of the combustion chamber space.

Fig. 5.3 shows the change in the pressure p, mean temperature of the gases T, internal energy of the working body ΔU, external work of the gases W and active heat evolution x during combustion in a Russian engine GAZ-21 (r.p.m. = 2000, $\eta_t = 0.337, \gamma = 1.02, p_m = 10\ \text{kg/cm}^2$). At point 1 corresponding to p_{max} the amount of active heat evolved is 73 per cent. During the afterburning phase this amount is 86 per cent of the available heat of combustion. Point 2 corresponding to t_{max} is 11° from the point of p_{max}. The heat losses to the walls and due to incomplete combustion are about 15 per cent when an engine is throttled or sucs a lean mixture x at the point p_m, does not usually exceed 50 per cent and the process of afterburning is much longer.

3(c).

1. *Raising the compression ratio.* Increasing the compression ratio increases both the temperature and pressure (density of the unburned mixture). Increase in temperature reduces the delay period of the end gas. Increase in temperature as well as increase in pressure both lead to greater collisions of molecules resulting in greater formation of chemical species responsible for knocking. Hence the tendency to knock increases.

For a given engine setting and fuel there will be a critical compression ratio above which knock occurs. This compression ratio is called the *highest useful compression ratio.*

Materials with high heat conductivity coefficients such as aluminium alloys are desirable for high compression cylinder heads since a cool combustion chamber wall is essential for high compression without knock. However hot spots may develop because of poor circulation of the coolant or improper distribution of the metal.

2. *Supercharging.* It also increases both temperature and density.

3. *Raising the inlet temperature.* Delay period decreases, velocity of flame travel increases.

4. *Raising the coolant temperature.* Delay period decreases.

5. *Increasing the load (opening the throttle).* An increase in the load increases the temperature of cylinder and combustion chamber walls thereby raising mixture and end gas temperatures. Also the pressure of the charge is increased. Hence the tendency to knock increases.

6. *Raising the temperature of the cylinder and combustion chamber walls.* The temperature of the end gas depends on the design of combustion chamber. Sparking plug and exhaust valve are two hottest parts in the combustion chamber and the end gas should not be compressed against these.

pressure (densities) are increased. Thus tendency to knock increases with advanced spark timings and decreases with retarded spark timings. By retarding the spark timings the peak pressures are reached farther down on the power stroke and are thus of lower magnitude.

If in a given engine the fuel quality is changed and knock takes place, *retarding* the ignition may eliminate the knock; but it will also reduce the engine power.

(B) Density factors. Increasing the density of the unburned mixture by any of the following methods will increase the possibility of knock in the engine.

1. Increasing the compression ratio.
2. Opening the throttle (increasing the load).
3. Supercharging the engine.
4. Increasing the inlet pressure. An increase in the inlet pressure increases the overall pressure during the cycle. The high pressure in the end gas decreases the delay period which increases the tendency of the charge to detonate. However, an increase in the inlet pressure increases the flame velocity, which would reduce the tendency to detonate. But the first effect always predominates. Therefore, with increased pressure tendency to detonate always increases.
5. Advancing the spark timings.

(C) Time factors. Increasing the time of exposure of the unburned mixture to auto-ignition conditions by any of the following factors will increase the possibility of knock in SI engines.

1. *Increasing flame travel distance (combustion chamber design, spark plug position, engine size).* The possibility of knock is increased by increasing the distance the flame has to travel in order to traverse the combustion chamber.

(i) *Combustion chamber shape.* In general, the more compact the combustion chamber, the better will be its anti-knock characteristics, since the flame travel and combustion time will be shorter. Further, if the combustion chamber is highly turbulent, the combustion rate is high and consequently combustion time is further reduced ; this further reduces the tendency to knock.

(ii) *Location of spark plug.* A spark plug which is centrally located in the combustion chamber has minimum tendency to knock as the flame travel is minimum. The flame travel can be reduced by using two or more spark plugs.

(iii) *Location of exhaust valve.* The exhaust valve should be located close to the spark plug so that it is not in the end gas region, otherwise there will be a tendency to knock.

(iv) *Engine size.* The delay period is not very much affected by the size of the cylinder. However, flame requires a longer time to travel across the combustion chamber of a large engine. The large engines, therefore, have a greater knocking tendency than smaller engines. The SI engine is, therefore, generally limited to 100 mm bore.

2. *Decreasing the turbulence of mixture.* Decreasing the turbulence of the mixture decreases the flame speed and hence increases the tendency to knock. Turbulence depends on the design of combustion chamber and one engine speed.

3. *Decreasing the speed of the engine.* A decrease in the engine speed decreases the turbulence of the mixture resulting in reduced flame speed. Also lower the engine speed, longer is the absolute time for the flame to traverse the cylinder which increases the time available for perflame reactions. Hence the tendency to knock is increased at lower speeds.

Note. *High speed knock in SI engines.* Arrigoni *et al.* have reported that most severe knocking conditions are those met by small displacement engines at a sufficiently high constant speed (4000-5000 rpm) and wide open throttle. Motor octane number is by far the most important 'fuel engine' factor in controlling high speed knock at all engine speeds. High knock intensities (that is, 3 V) much greater than trace levels (about 0.2 V) will cause engine damage.

4 (a)

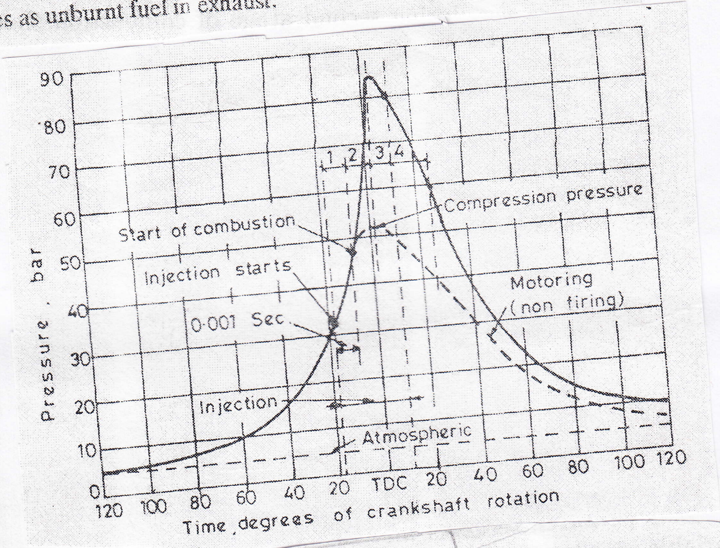
1. First Stage. Ignition delay period during which some fuel has been admitted but has not yet been ignited. The ignition delay is counted from the start of injection to the point where the p-θ curve separates from the pure air compression curve. The delay period is a sort of preparatory phase. The ignition delay is discussed in detail later.

2. Second Stage. Rapid or uncontrolled combustion (probable premixed flame) following ignition. In this second stage the pressure rise is rapid because during the delay period the fuel droplet have had time to spread themselves over a wide area and they have fresh air all around them. The period of rapid or uncontrolled combustion is counted from the end of delay period to the point of maximum pressure on the indicator diagram. About one-third of the heat is evolved during this period.

3. Third Stage. *Controlled combustion* (probable diffusion flame). The second stage of rapid or uncontrolled combustion is followed by the third stage—the controlled combustion. At the end of second stage the temperature and pressure are so high that the fuel droplets injected during the last 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 percent of the total heat of the fuel supplied during the cycle.

To these three stages of combustion, first proposed by Ricardo, a fourth stage can be added—late burning or after-burning. This stage may not be present in all cases.

4. Fourth Stage. *After burning.* Theoretically it is expected that combustion process shall end after the third stage. However, because of poor distribution of the fuel particles, combustion continues during part of the remainder of the expansion stroke. This after-burning can be called the fourth stage of combustion. The duration of the after-burning phase may correspond to 70-80 degrees of crank travel from tdc and the total heat evolved by the end of entire combustion process is 95 to 97% and 3 to 5% of heat goes as unburnt fuel in exhaust.



DELAY PERIOD OR IGNITION LAG

The first stage of combustion in the CI engine, *i.e.* the delay period, exerts a very great influence on both engine design and performance and, therefore, needs a detailed study. In Fig. 6.4 the delay period is shown on

delay in the SI engine as the charge consists of homogeneous mixture of vaporised fuel and air.

The delay period in CI combustion affects rate of pressure rise and, hence, knocking. It also affects engines startability.

It is clear that the pressure reached during the second stage will depend on the duration of the delay period (the longer the delay period the more rapid and higher is 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'. (The diesel knock is discussed in detail later). *Therefore the diesel engine designer aims to keep the delay period as short as possible,* both for smooth running and 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. This will result in high smoke and high fuel consumption. In practice, however, the delay period is more than required and the designer's efforts are always devoted towards shortening it as much as possible.

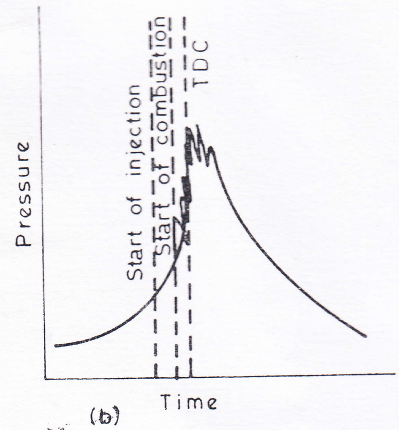
4(b)

~~We have already discussed that~~ if the delay period 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 in SI engines.

It is very instructive to compare the phenomenon of detonation in SI engines with that of knocking in CI engines. There is no doubt that these two phenomena are fundamentally similar. Both are processes of auto-ignition subject to the ignition time-lag characteristics of the fuel-air mixture. However, differences in the knocking phenomena of the SI engine and the CI engine should also be carefully noted :

(1) In the SI engine, the detonation occurs near the end of combustion whereas in the CI engine detonation occurs near the beginning of combustion as shown in Fig. 6.10.

(2) The detonation in the SI engine is of a homogeneous charge causing very high rate of pressure rise and very high maximum pressure. In the CI 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 SI engine.



(b) CI engine

(3) Since in the CI engine the fuel is injected into the cylinder only at the end of the compression stroke there is no question of 'pre-ignition' or 'premature ignition', as in the SI engine.

(4) In the SI engine it is relatively easy to distinguish between knocking and non-knocking operation as the human ear easily finds the distinction. However, in the case of the CI engine the normal ignition is itself by auto-ignition and hence most CI engines have a sufficiently high rate of pressure rise per degree of crank angle to cause audible noise. When such noise becomes excessive or there is excessive vibration in engine structure, in the opinion of the observer, the engine is said to knock. It is clear that personal judgment is involved here. Thus in the CI engine there is no definite distinction between normal and 'knocking' combustion. The maximum rate of pressure rise in the CI engine may reach as high as 10 bar per crank degree angle.

It is most important to note that factors that tend to reduce detonation in the SI engine increase knocking in the CI engine and *vice versa* because of the following reason. The detonation or knocking in the SI engine is due to simultaneous auto-ignition of the last part of the charge. To eliminate detonation in the SI engine we want to prevent altogether the auto-ignition of the last part of the charge and therefore desire a long delay period and high self-ignition temperature of the fuel. To eliminate knock in the CI engine we want to achieve auto-ignition as early as possible and therefore desire a short delay period and low self-ignition temperature of the fuel.

4C

4. **Compression ratio.** Increase in compression ratio reduces the delay period as it raises both temperature and density. Fig. 6.8 shows that with increase in compression ratio the temperature of the air increases (curve *a*). At the same time the minimum auto-ignition temperature decreases due to increased density of the compressed air, resulting in closer contact of the molecules which, thereby, reduces the time of reaction when fuel is injected. As the difference between compressed temperature and

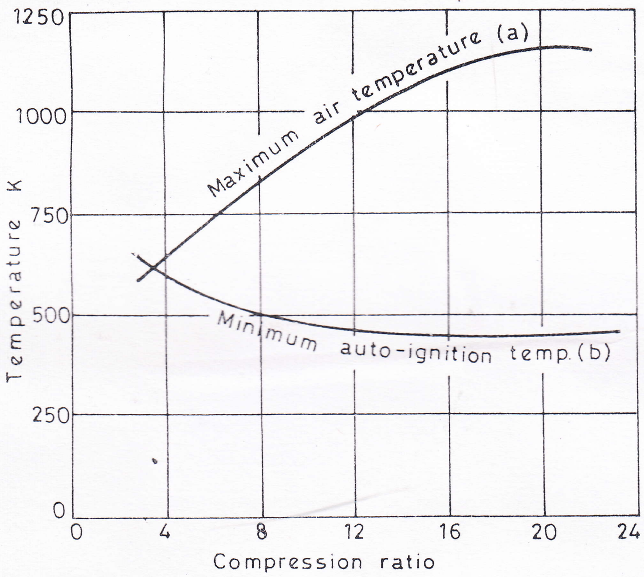


Fig. 6.8. Effect of compression ratio on maximum air temperature and minimum auto-ignition temperature.

the minimum auto-ignition temperature increases, the delay period decreases.

The above theoretical analysis may suggest that in diesel engines the highest possible compression ratio should be used to achieve the lowest delay period. However, there are practical disadvantages of using very high compression ratios. In CI engines the compression space is already very small and the necessity of providing working clearance between the piston and the cylinder head and around the valves compels us to leave thin layers or pockets of air to which the fuel cannot reach. *With a compression ratio of 16 the unused air is already about 20 per cent.* With increase of compression ratio the unused air would be much more decreasing the volumetric efficiency and power. Another disadvantage of high compression ratio is lower mechanical efficiency due to increases in weight of reciprocating parts. *Therefore, in practice the engine designer uses the lowest compression ratio which would satisfy the needs of easy cold starting and light load running at high speeds*. Note that this practice is opposite of the SI engine design practice where the endevour is to use highest possible compression ratio, only limited by detonation.

The maximum peak pressure is only marginally affected by the compression ratio, because with higher compression ratio delay period is shorter and therefore the rise of pressure on ignition is lower.

Injection Timing the effect

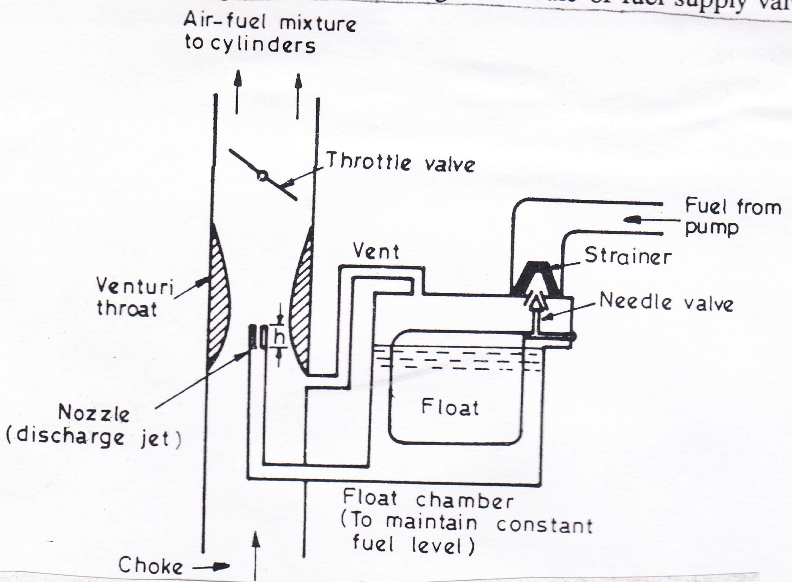
...ffect of injection advance on the pressure variation is shown in Fig.12.16 for three injection advance timings of 9°, 18°, and 27° before *TDC*. The injected quantity of fuel per cycle is constant. As the pressure and temper...ure at the beginning of injection are lower for higher ignition advance, the delay period increases with increase in injection advance. The optimum ...e of injection advance depends on many factors but generally it is about 20° bTDC.

5 (a)

A SIMPLE OR ELEMENTARY CARBURETTOR

To understand a modern carburettor which is a complicated piece of equipment, it is helpful to first study a simple or elementary carburettor which provides *an air-fuel mixture for cruising or normal range at a single speed and then to add it other mechanisms* to provide for other duties like starting, idling, variable load and speed operation and acceleration.

Fig. 11.9 shows a simple carburettor. It consists of a float chamber, nozzle with metering orifice, venturi and throttle valve. The float and a needle valve system maintains a constant height of petrol in the float chamber. If the amount of fuel in the float chamber falls below the designed level, the float lowers, thereby opening the needle of fuel supply valve.



When the designed level has been reached, the float closes the needle valve, thus stopping additional fuel flow from the supply system. Float chamber is vented to the atmosphere.

During suction stroke air is drawn through the venturi. Venturi is a tube of decreasing cross-section which reaches a minimum at the throat (Venturi tube is also known as choke tube and is so shaped that it gives minimum resistance to air flow). The air passing through the venturi increases in velocity and the pressure in the venturi throat decreases. From the float chamber, the fuel is fed to a discharge jet, the tip of which is located in the throat of the venturi. Now because the pressure in the float chamber is atmospheric and that at the discharge jet below atmospheric a pressure differential, called *carburettor depression*, exists between them. This causes discharge of fuel into the air stream and the rate of flow is controlled or metered by the size of the smallest section in the fuel passage. This is provided by the discharge jet and the size of this jet is chosen empirically to give the required engine performance. The pressure at the throat at the fully open throttle condition lies between 4 and 5 cm. Hg below atmospheric, and seldom exceeds 8 cm Hg below atmospheric. To avoid wastage of fuel, the level of the liquid in the jet is adjusted by the float chamber needle valve to maintain the level a short distance below the tip of the discharge jet.

The petrol engine is quantity governed which means that when less power is required at a particular speed the amount of charge delivered to the cylinders is reduced. This is achieved by means of a throttle valve of the butterfly type which is situated after venturi tube. As the throttle is closed less air flows through the venturi tube and less is the quantity of air-fuel mixture delivered to the cylinders and hence less is the power developed. As the throttle is opened, more air flows through the choke tube, and the power of the engine increases.

A simple carburettor of the type described above suffers from a fundamental fault in that it provides increasing richness as the engine speed and air flow increases with full throttle because the density of the air tends to decrease as the rate of air flow increases. Also, it provides too lean mixtures at low speeds at part open throttle. This phenomenon can be explained as follows: Since the throttle regulates the amount of air flowing up the venturi tube, it also checks the quantity of fuel issuing from the nozzle by regulating the vacuum at the throat. At low engine speeds with part open throttle for (at low air flow rates) the vacuum at the throat is small and hence we get too lean a mixture. At high engine speeds the

(b) Qualities OF AN IGNITION SYSTEM

A smooth and reliable functioning of an ignition system is essential for reliable working of an engine. The requirements of such an ignition system are:

(i) It should provide a good spark between the electrodes of the plugs at the correct timing.

(ii) It should function efficiently over the entire range of engine speed.

(iii) It should be light, effective and reliable in service.

(iv) It should be compact and easy to maintain.

(v) It should be cheap and convenient to handle.

(vi) The interference from the high voltage source should not affect the functioning of the radio and television receivers inside an automobile.

5 (c)

FUNCTIONAL REQUIREMENTS OF AN INJECTION SYSTEM

For a proper running and good performance from the engine, the following requirements must be met ...

(i) Accurate metering of the fuel injected per cycle. This is very critical due to the fact that very small quantities of fuel being handled. Metering errors may cause drastic variation from the desired output. The quantity of the fuel metered should vary to meet changing speed and load requirements of the engine.

(ii) Timing the injection of the fuel correctly in the cycle so that maximum power is obtained ensuring fuel economy and clean burning.

(iii) Proper control of rate of injection so that the desired heat-release pattern is achieved during combustion.

(iv) Proper atomization of fuel into very fine droplets.

(v) Proper spray pattern to ensure rapid mixing of fuel and air.

(vi) Uniform distribution of fuel droplets throughout the combustion chamber.

(vii) To supply equal quantities of metered fuel to all cylinders in case of multi cylinder engines.

(viii) No lag during beginning and end of injection i.e., to eliminate dribbling of fuel droplets into the cylinder.

6 (a).

SUPERCHARGING

Supercharging of internal combustion engines is in practice for a long time as a method for improving engine power output. Entering the millennium, a new trend is appearing. The trend points to small displacement engines in order to meet emission legislation on fuel consumption and emission control. The consumers, however, still demands the same performance they are used to.

A good way to meet these needs is to have supercharging which may be called forced induction. As already stated, the purpose of supercharging an engine is to raise the density of the air charge, before it enters the cylinders. Thus, the increased mass of air will be inducted which will then be compressed in each cylinder. This makes more oxygen available for combustion than the conventional method of drawing the fresh air charge into the cylinder (naturally aspirated). Consequently, more air and fuel per cycle will be forced into the cylinder, and this can be effectively burnt during the combustion process to raise the engine power output to a higher value than would otherwise be possible.

The points to be noted in supercharging an engine summarized as:

(i) Supercharging increases the power output of the engine. It does not increase the fuel consumption, per brake kW hour.

(ii) Certain percentage of power is consumed in compressing the air. This power has to be taken from the engine itself. This will lead to some power loss. However, it is seen that the net power output will be more than the power output of an engine of the same capacity, without supercharging.

(iii) The engine should be designed to withstand the higher forces due to supercharging.

(iv) The increased pressure and temperature as a result of supercharging, may lead to detonation, Therefore the fuel used must have better anti-knock characteristics.

In practice, racing car engines use supercharging. The most important areas where supercharging is of vital importance are :

(i) Marine and automotive engines where weight and space are important.

(ii) Engines working at high altitudes. The power loss due to altitude can

6 (b)

TYPES OF SUPERCHARGERS

Supercharger is a pressure-boosting device which supplies air (or mixture) at a higher pressure. A centrifugal or axial flow or displacement type compressor is normally used. If the supercharger is driven by the engine crankshaft, then it is called mechanically driven supercharger. Some superchargers are driven by a gas turbine, which derives its power from the engine exhaust gases. Such a supercharger is called turbocharger. There are three types of superchargers

(i) Centrifugal type

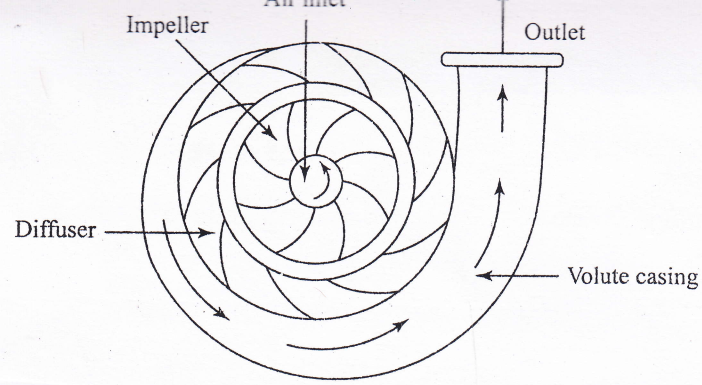
(ii) Root's type

(iii) Vane type

Centrifugal Type Supercharger

The centrifugal type supercharger is commonly used in automotive engines (Fig.19.1). A V-belt from the engine pulley runs the supercharger. First, the air-fuel mixture enters the impeller at the centre. It then passes through the impeller and the diffuser vanes. Finally, air or mixture enters the volute casing and then goes to the engine from the casing. The mixture will come out at higher pressure and this condition is called supercharged condition.

Because of higher pressure more air-fuel mixture is forced into the cylinder. About 30% more air-fuel mixture can be forced into the combustion chamber. The impeller runs at very high speeds, about 80,000 revolutions per minute. Therefore the impeller should be able to withstand the high stresses produced at this speed. Impellers are usually made of duralumin,



Centrifugal Type of Supercharger

Root's Supercharger

~~The details of Root's supercharger is shown in Fig.19.2.~~ The Root's supercharger has two rotors of epicycloids shape, with each rotor keyed to its shaft. One rotor is connected with the other one by means of gears. The gears are of equal size and therefore both the rotors rotate at the same speed. The Root's supercharger operates like a gear pump. The mixture at the outlet of this supercharger will be at much higher pressure than the inlet.

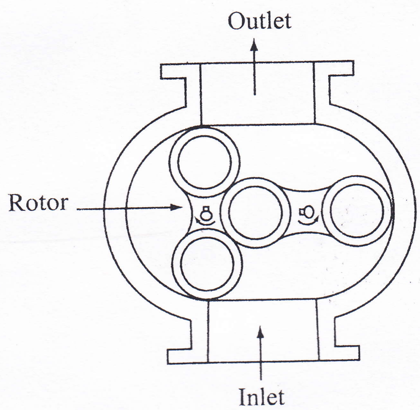


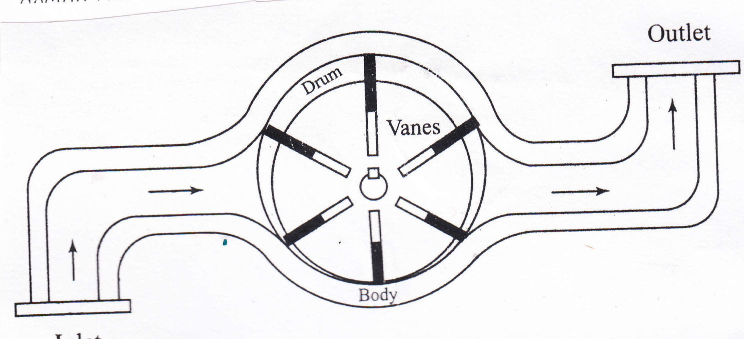
Fig. 19.2 Root's Supercharger

Vane Type Supercharger

Details of a typical vane type supercharger is shown in Fig.19.3. A number of vanes are mounted on the drum which is inside the body of the supercharger. The vanes can slide in or out, against the force of the spring. Because of this arrangement, the vanes are always in contact with the inner surface of the body. The space between the inner surface of the body and the drum decreases from the inlet to the outer side. In this way, the quantity of the mixture which enters at the inlet, decreases in volume, because of which the pressure of the mixture will increase as it reaches the exit.

Comparison between the Three Superchargers

The required characteristics of a centrifugal type supercharger is poor and suitable for only low speeds. The root's supercharger is simple in construction, requires only minimum maintenance and has longer life. The vane type supercharger has a special problem of wear of tips of the vanes with time. Therefore, one has to take into account the application and then



(i) To Provide sufficient qt. of cool, filtered to give positive & adequate lubrication to all moving parts of engine.

The various lubrication systems used for internal combustion engines may be classified as

(i) mist lubrication system

(ii) wet sump lubrication system

(iii) dry sump lubrication system

13.9.1 Mist Lubrication System

This system is used where crankcase lubrication is not suitable. In two-stroke engine, as the charge is compressed in the crankcase, it is not possible to have the lubricating oil in the sump. Hence, mist lubrication is adopted in practice. In such engines, the lubricating oil is mixed with the fuel, the usual ratio being 3% to 6%. The oil and the fuel mixture is inducted through the carburettor. The fuel is vaporized and the oil in the form of mist goes via the crankcase into the cylinder. The oil which strikes the crankcase walls lubricates the main and connecting rod bearings, and the rest of the oil lubricates the piston, piston rings and the cylinder.

The advantage of this system is its simplicity and low cost as it does not require an oil pump, filter, etc. However, there are certain disadvantages which are enumerated below.

(i) It causes heavy exhaust smoke due to burning of lubricating oil partially or fully and also forms deposits on piston crown and exhaust ports which affect engine efficiency.

(ii) Since the oil comes in close contact with acidic vapours produced during the combustion process gets contaminated and may result in the corrosion of bearing surface.

(iii) This system calls for a thorough mixing for effective lubrication. This requires either separate mixing prior to use or use of some additive to give the oil good mixing characteristics.

(iv) During closed throttle operation as in the case of the vehicle moving down the hill, the engine will suffer from insufficient lubrication as the supply of fuel is less. This is an important limitation of this system.

In some of the modern engines, the lubricating oil is directly injected into the carburettor and the quantity of oil is regulated. Thus the problem of oil deficiency is eliminated to a very great extent. In this system the main bearings also receive oil from a separate pump. For this purpose, they will be located outside the crankcase. With this system, formation of

13.9.2 Wet Sump Lubrication System

In the wet sump system, the bottom of the crankcase contains an oil pan or sump from which the lubricating oil is pumped to various engine components by a pump. After lubricating these parts, the oil flows back to the sump by gravity. Again it is picked up by a pump and recirculated through the engine lubricating system. There are three varieties in the wet sump lubrication system. They are

(i) the splash system

(ii) the splash and pressure system

(iii) the pressure feed system

Splash System: This type of lubrication system is used in light duty engines. A schematic diagram of this system is shown in Fig.13.9.

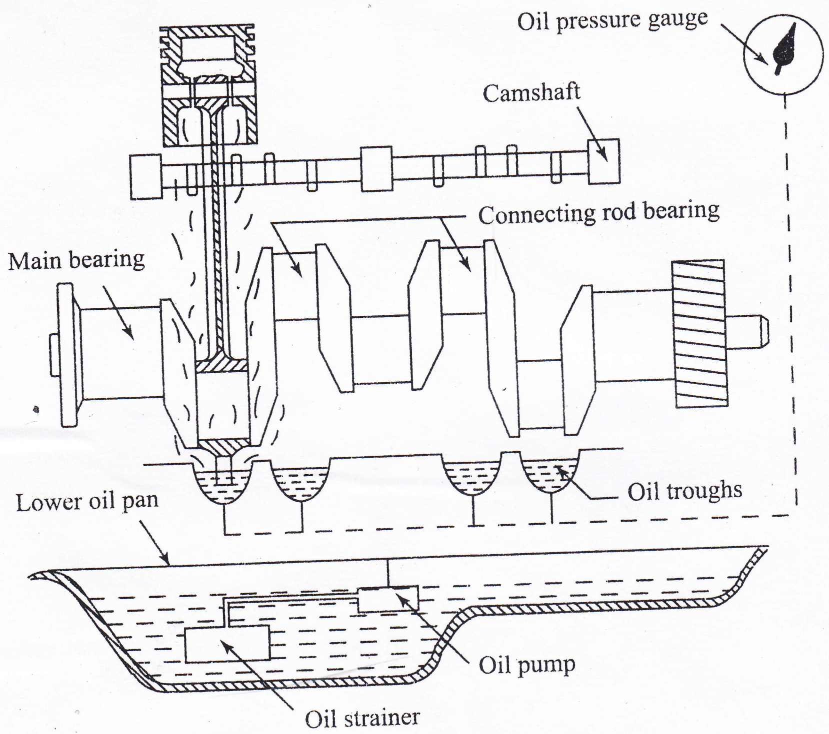
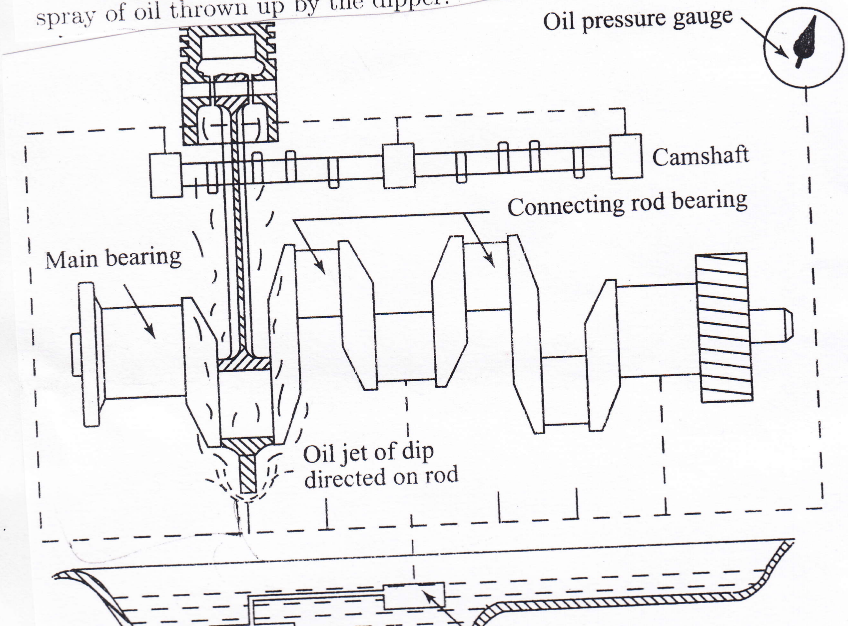


Fig. 13.9 Splash Lubrication System

The lubricating oil is charged into in the bottom of the engine crankcase and maintained at a predetermine level. The oil is drawn by a pump and delivered through a distributing pipe extending the length of the crankcase into splash troughs located under the big end of all the connecting rods. These troughs were provided with overflows and the oil in the troughs are therefore kept at a constant level. A splasher or dipper is provided under each connecting rod cap which dips into the oil in the trough at every revolution of the crankshaft and the oil is splashed all over the interior of the crankcase, into the pistons and onto the exposed portions of the cylinder walls. A hole is drilled through the connecting rod cap through which oil will pass to the bearing surface. Oil pockets are also provided to catch the splashing oil over all the main bearings and also over the camshaft bearings. From the pockets the oil will reach the bearings surface through a drilled hole. The oil dripping from the cylinders is collected in the sump where it is cooled by the air flowing around. The cooled oil is then recirculated.

The Splash and Pressure Lubrication System: This system is shown in Fig.13.10, where the lubricating oil is supplied under pressure to main and camshaft bearings. Oil is also supplied under pressure to pipes which direct a stream of oil against the dippers on the big end of connecting rod bearing cup and thus the crankpin bearings are lubricated by the splash or spray of oil thrown up by the dipper.



main bearings of the crankshaft through distributing channels. A pressure relief valve will also be fitted near the delivery point of the pump which opens when the pressure in the system attains a predetermined value. An oil hole is drilled in the crankshaft from the centre of each crankpin to the centre of an adjacent main journal, through which oil can pass from the main bearings to the crankpin bearing. *From the crankpin it reaches piston pin bearing through a hole drilled in the connecting rod.* The cylinder walls, tappet rollers, piston and piston rings are lubricated by oil spray from around the piston pins and the main and connecting rod bearings. The basic components of the wet sump lubrication systems are (i) pump (ii) strainer (iii) pressure regulator (iv) filter (v) breather.

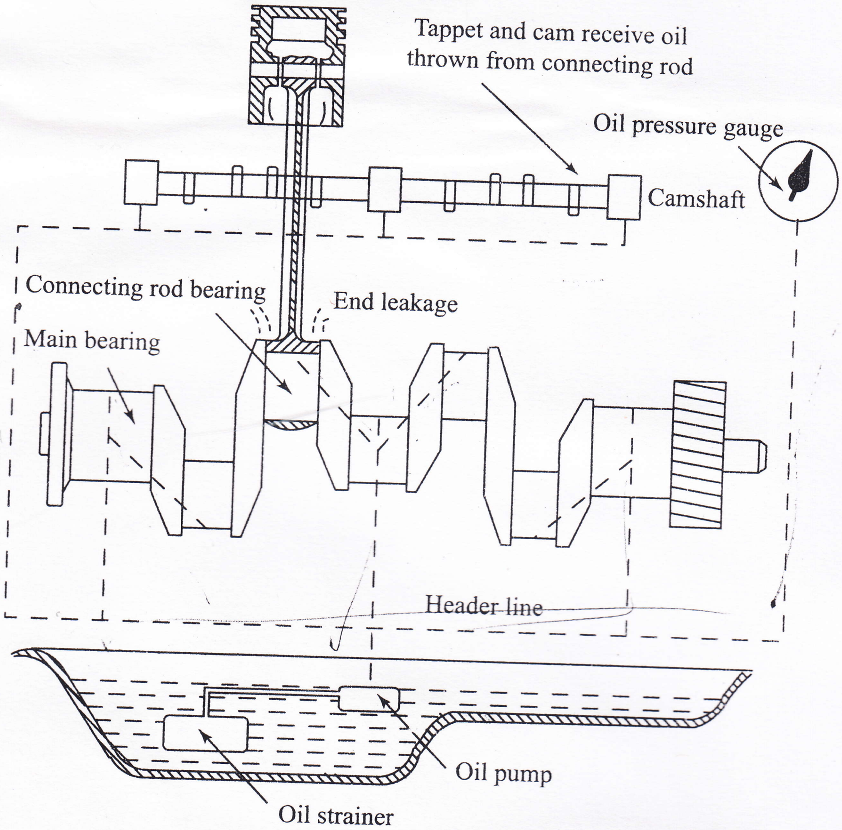


Fig. 13.11 Pressure Feed Lubrication System

Wet Sump.

A typical wet sump and its components are shown in Fig.13.12. Oil is drawn from the sump by a gear or rotor type of oil pump through an oil strainer. The strainer is a fine mesh screen which prevents foreign particles from entering the oil circulating systems. A pressure relief valve is provided which automatically keeps the delivery pressure constant and can be set to any value. When the oil pressure exceeds that for which the valve is set, the valve opens and allows some of the oil to return to the sump thereby relieving the oil pressure in the systems. Fig.13.13 shows a typical gear pump, pressure relief valve and by-pass. Most of the oil from the pump goes directly to the engine bearings and a portion of the oil passes through a cartridge filter which removes the solid particles from the oil. This reduces the amount of contamination from carbon dust and other impurities present in the oil. Since all the oil coming from the pump does not pass directly through the filter, this filtering system is called by-pass filtering system. All the oil will pass through the filter over a period of operation. The advantage of this system is that a clogged filter will not restrict the flow of oil to the engine.

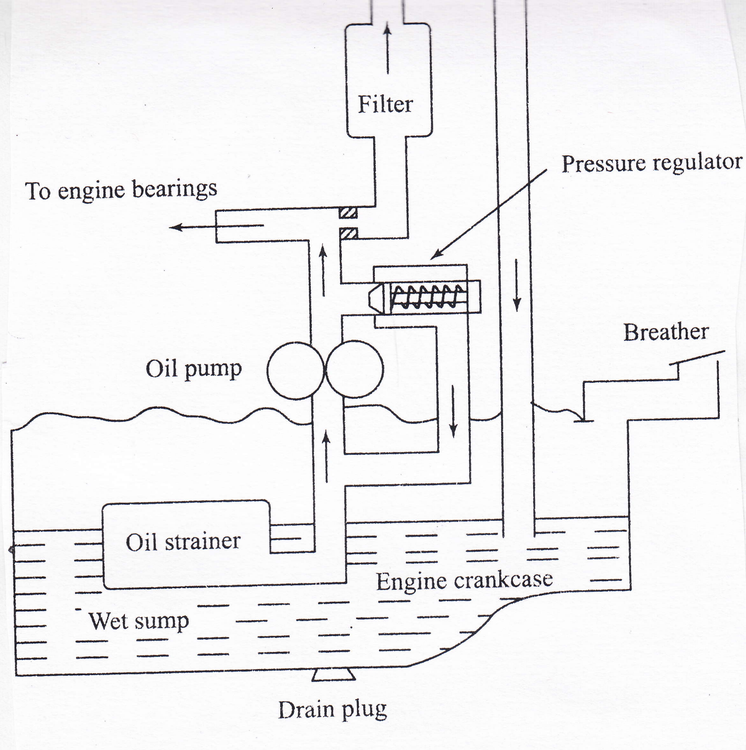


Fig. 13.12 Basic Components of Wet Sump Lubrication System

Dry Sump Lubrication System

A dry sump lubricating system is illustrated in Fig.13.14. In this, the supply of oil is carried in an external tank. An oil pump draws oil from the supply tank and circulates it under pressure to the various bearings of the engine. Oil dripping from the cylinders and bearings into the sump is removed by a scavenging pump which in turn the oil is passed through a filter, and is fed back to the supply tank. Thus, oil is prevented from accumulating in the base of the engine. The capacity of the scavenging pump is always greater than the oil pump. In this system a filter with a bypass valve is placed in between the scavenge pump and the supply tank. If the filter is clogged, the pressure relief valve opens permitting oil to by-pass the filter and reaches the supply tank. A separate oil cooler with either water or air as the cooling medium, is usually provided in the dry sump system to remove heat from the oil.

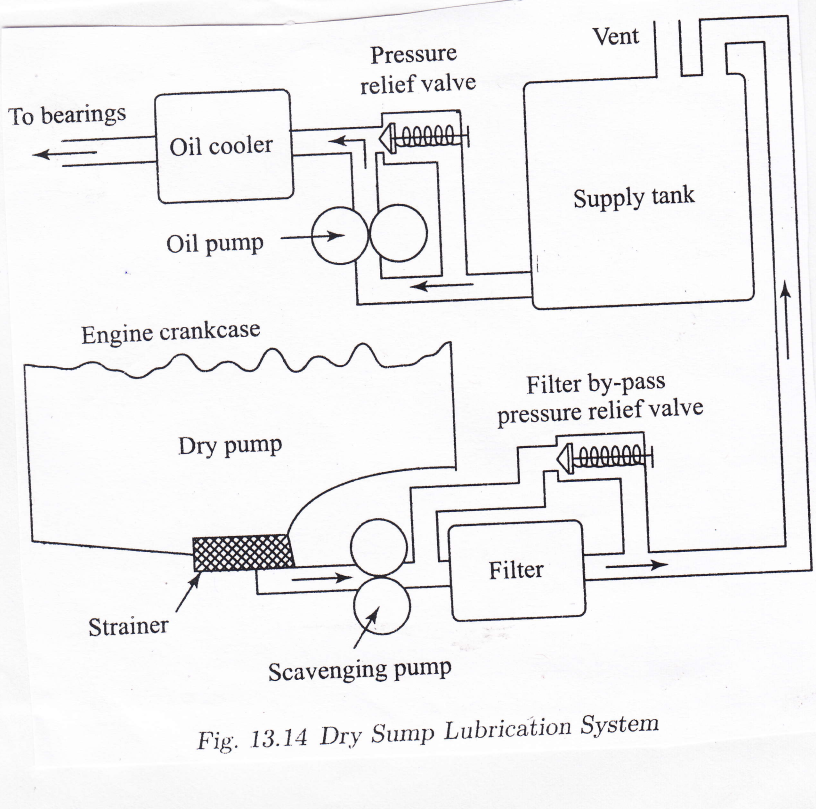


Fig. 13.14 Dry Sump Lubrication System