CHAPTER 2

FOUR-STROKE CYCLE S-I ENGINE - PRINCIPLE OF OPERATION

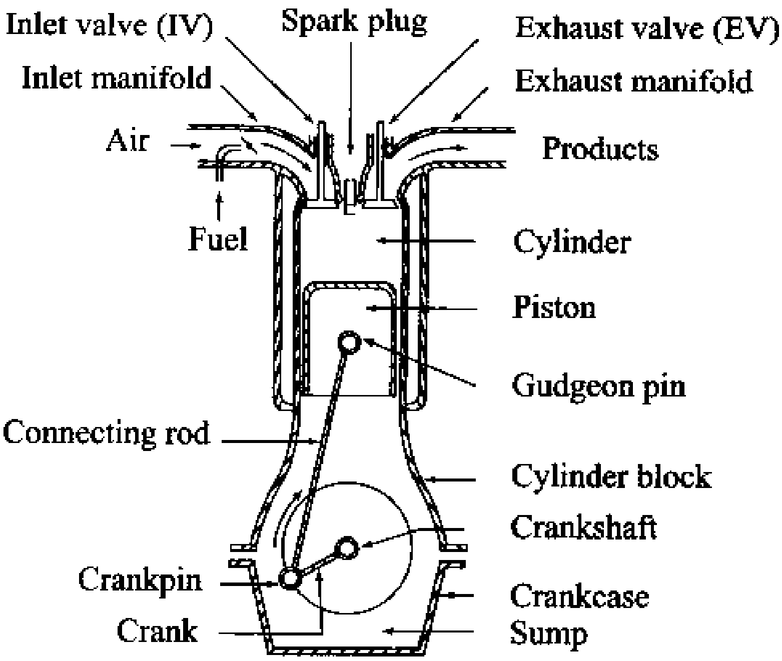


Fig: cross section of a SI Engine

In Four-stroke cycle engine, the cycle of operation is completed in four-strokes of the piston or two revolutions of the crankshaft. Each stroke consists of 180°, of crankshaft rotation and hence a cycle consists of 720°of crankshaft rotation. The series of operations of an ideal four-stroke. SI engine are as follows (see Fig.2.1 & 2.2)

1. Suction stroke

Suction stroke 0-1 starts when the piston is at top dead centre and about to move downwards. The inlet valve is open at this time and the exhaust valve is closed. Due to the suction created by the motion of the piston towards bottom dead centre, the charge consisting of fresh air mixed with the fuel is drawn into the cylinder. At the end of the suction stroke the inlet valve closes.

2. Compression stroke.

The fresh charge taken into the cylinder during suction stroke is compressed by the return stroke of the piston 1-2. During this stroke both inlet and exhaust valves remain closed. The air which occupied the whole cylinder volume is now compressed into clearance volume. Just before the end of the compression stroke the mixture is ignited with the help of an electric spark between the electrodes of the spark plug located in combustion chamber wall. Burning takes place when the piston is almost at top dead centre. During the burning process the chemical energy of the fuel is converted into sensible energy, producing a temperature rise of about 2000°C, and the pressure is also considerably increased.

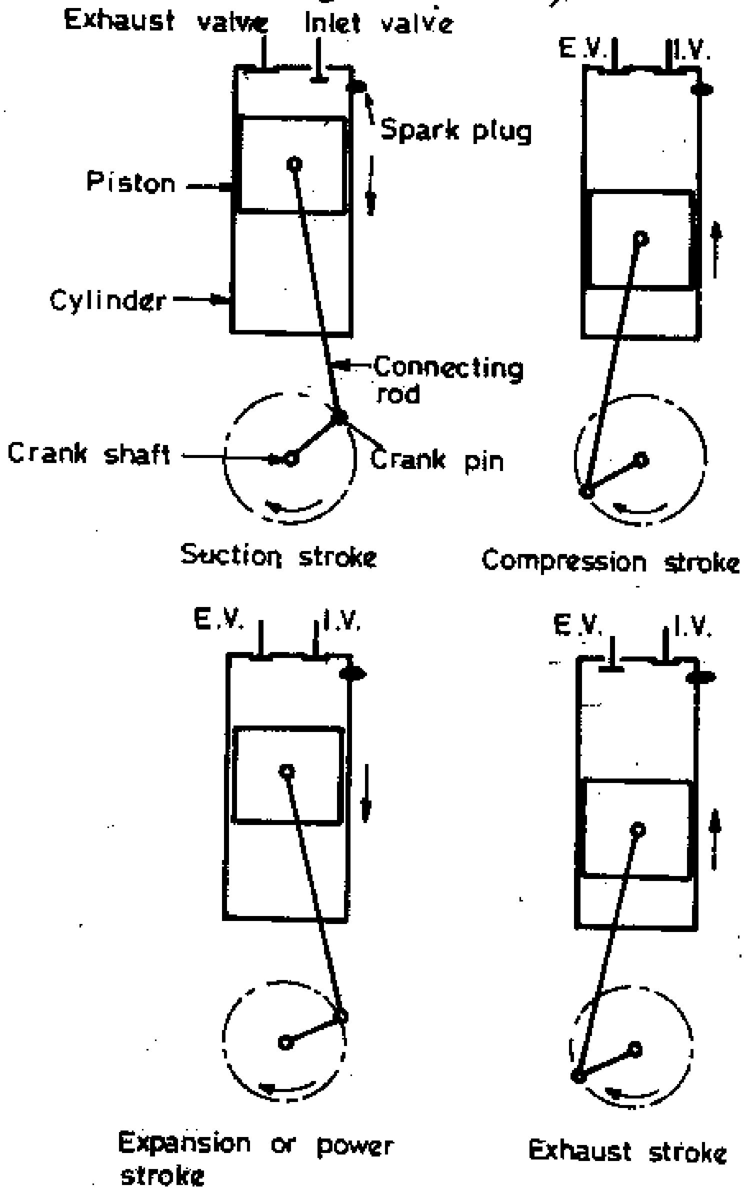
3. Expansion or power stroke.

Due to high pressure the burnt gases force the piston towards bottom dead centre, stroke 3-4, and both the inlet and exhaust valves remaining closed. Thus power is obtained during this stroke. Both pressure and temperature decrease during expansion.

4. Exhaust stroke.

At the end of the expansion stroke the exhaust valve opens, the inlet valve remaining closed, and the piston is moving from bottom dead centre to top dead centre sweeps out the burnt gases from the cylinder, stroke 4-0. The exhaust valve closes at the end of the exhaust stroke and some 'residual' gases remain in the cylinder.

Each cylinder of a four-stroke engine completes the above four operations in two engine revolutions. One revolution of the crankshaft occurs during the suction and compression strokes, and second revolution during the power and exhaust strokes. Thus for one complete cycle, there is only one power stroke while the crankshaft turns by two revolutions. Most of the spark-ignition internal combustion engines are of the four-stroke type. They are most popular for passenger cars and small aircraft applications.



Stroke	*Valve position*
Suction stroke.	Suction valve open. Exhaust valve closed.
Compression stroke.	Both valves closed.
Expansion stroke.	Both valves closed.
Exhaust stroke.	Exhaust valve open. Suction valve closed.

Fig.2.1-The four-stroke spark-ignition (SI) engine cycle (Otto cycle or constant volume cycle)

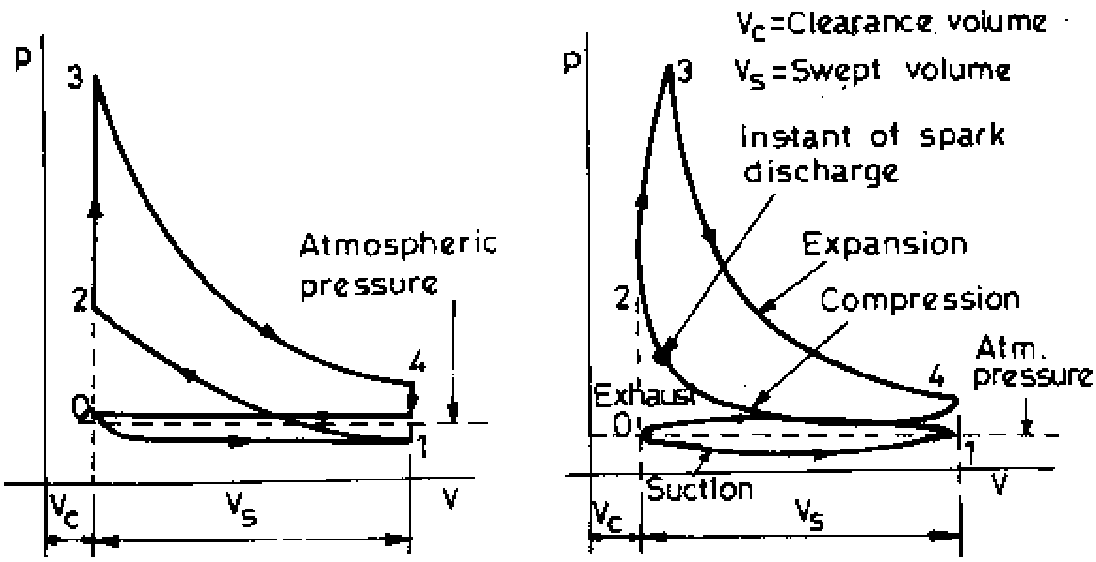


Fig.2.2-Ideal and actual indicator diagrams for four-stroke SI engine

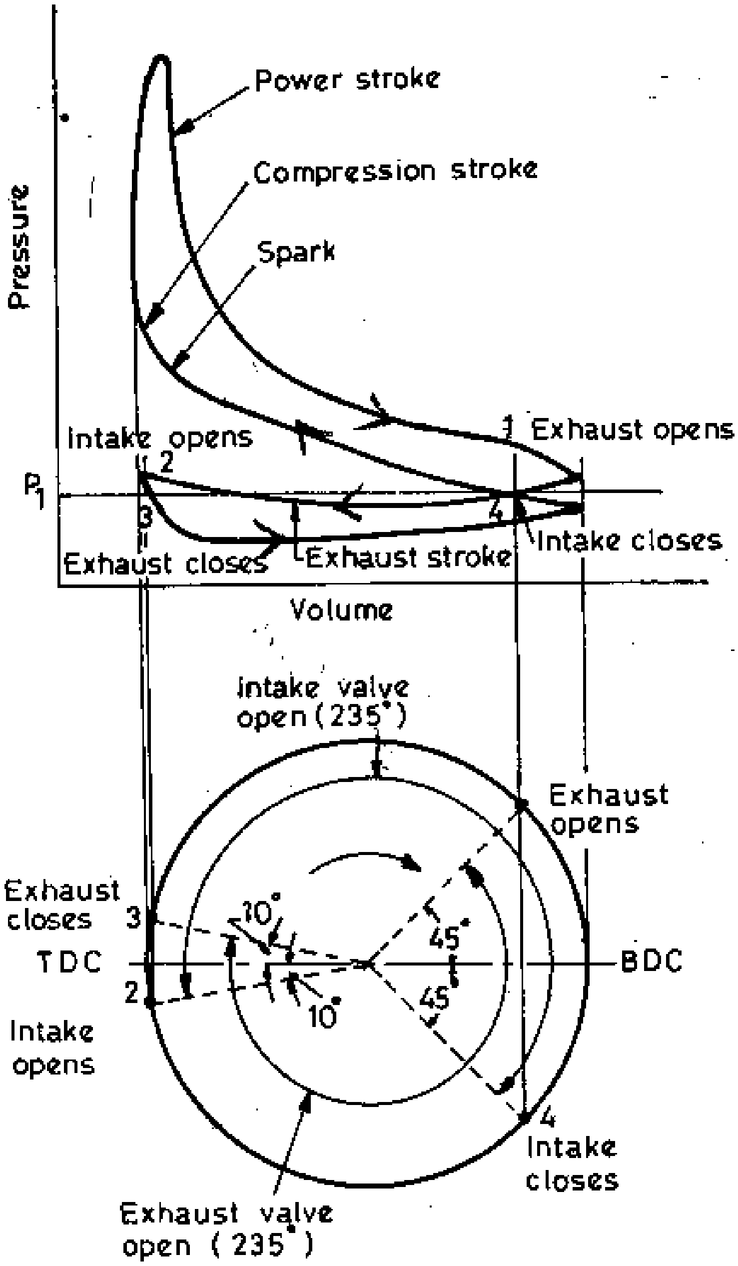


Fig. 2.3 Four-stroke petrol engine valve timing diagram in relation to the pressure volume diagram

Actual Valve Timing Of Four-Stroke Petrol Engine.

Valve timing is the regulation of the points in the cycle at which the valves are set to open and close. As described above in the ideal cycle inlet and exhaust valves open and close at dead centres, but in actual cycles they open or close before or after dead centres as explained below. There are two factors, one mechanical and other dynamic, for the actual valve timing to be different from the theoretical valve timing.

(a) Mechanical factor.

The poppet valves of the reciprocating engines are opened and closed by cam mechanisms. The clearance between cam, tappet and valve must be slowly taken up and valve slowly lifted, at first, if noise and wear is to be avoided. For the same reasons the valve cannot be closed abruptly, else it will 'bounce' on its seat. (Also the cam contours should be so designed as to produce gradual and smooth changes in directional acceleration). Thus the valve opening and closing periods are spread over a considerable number of crankshaft degrees. As a result, the opening of the valve must commence ahead of the time at which it is fully opened (i.e., before dead centres). The same reasoning applies for the closing time and the valves must close after the dead centres. Fig.2.3 shows the actual valve timing diagram of a four-stroke engine in relation to its pressure-volume diagram.

b) Dynamic factor;

Besides mechanical factor of opening and closing of valves, the actual valve timing is set taking into consideration the dynamic effects of gas flow.

Intake valve timing.

Intake valve timing has a bearing on the actual quantity of air sucked during the suction stroke *i.e.* it affects the volumetric efficiency. Fig.2.4 shows the intake valve timing diagram for both low speed & high speed SI engines.

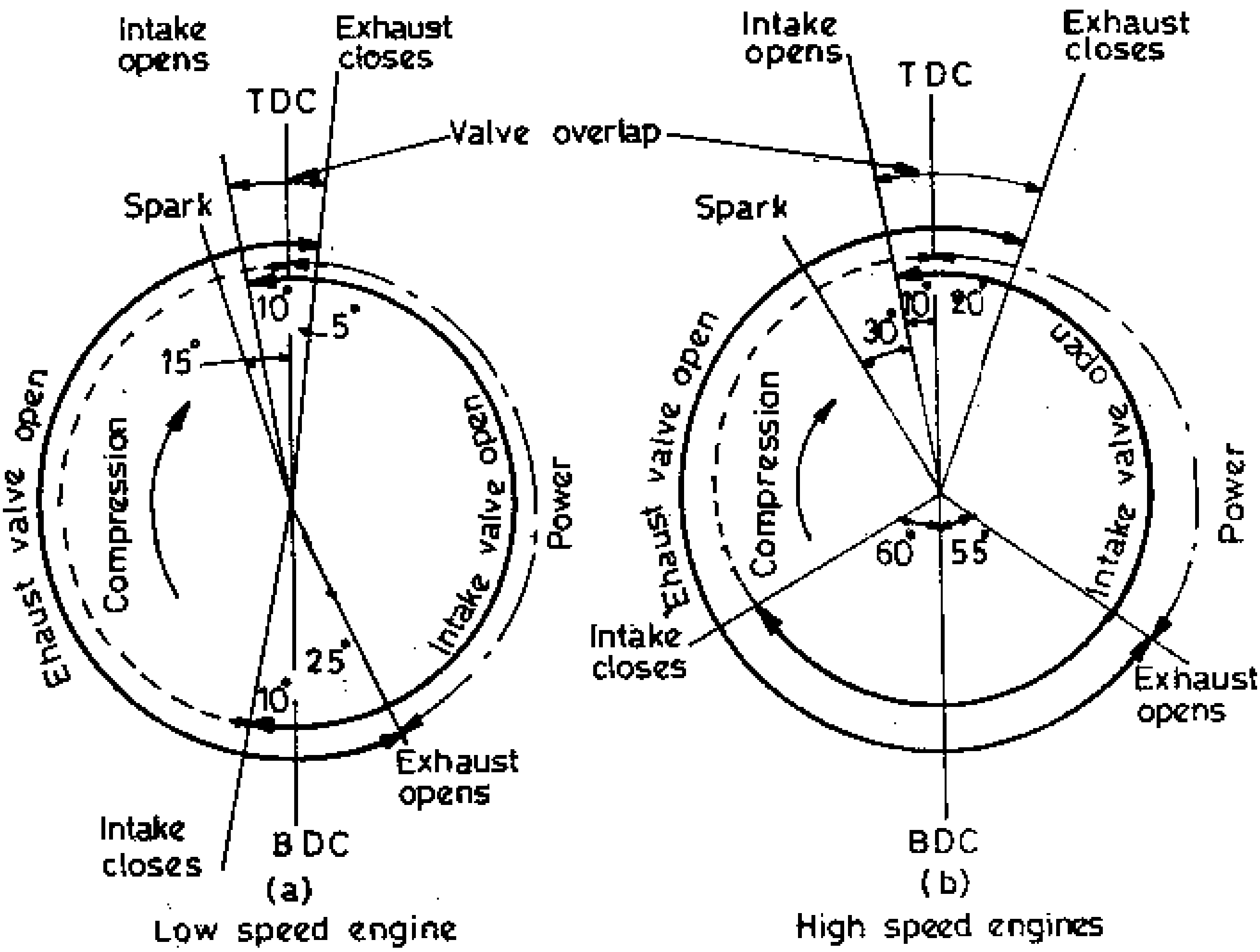


Fig:2.4 Valve timing for low and high speed four-stroke SI engine

It is seen that for both low speed and high speed engine the intake valve opens 10^0 before the arrival of the piston at TDC on the exhaust stroke. This is to insure that the valve will be fully open and the fresh charge starting to flow into the cylinder as soon as possible after TDC. As the piston moves out in the suction stroke, the fresh charge is drawn in through the intake port and valve. When the piston reaches the BDC and starts to move in the compression stroke, the inertia of the entering fresh charge tends to cause it to continue to move into the cylinder. To take advantage of this, the intake valve is closed after BDC so that maximum air is taken in. This is called ram effect. However, if the intake valve is to remain open for too long a time beyond BDC, the up-moving piston on the compression stroke would tend to force some of the charge, already in the cylinder, back into the intake manifold. The time the intake valve should remain open after BDC is decided by the speed of the engine.

At low engine speed, the charge speed is low and so the air inertia is low, and hence the intake valve should close relatively early after BDC for a slow speed engine (say about 10^0 after BDC).

In high speed engines the charge speed is high and consequently the inertia is high and hence to induct maximum quantity of charge due to ram effect the intake valve should close relatively late after BDC (up to 60^0 after BDC).

For a variable speed engine the chosen intake valve setting is a compromise between the best setting for low and high speeds.

There is a limit to the high speed for advantage of ram effect. At very high speeds the effect of fluid friction may be more than offset the advantage of ram effect and the charge for cylinder per cycle falls off.

Exhaust valve timing

The exhaust valve is set to open before BDC (say about 25^0 before BDC in low speed engines and 55^0 before BDC in high speed engines). If the exhaust valve did not start to open until BDC, the pressures in the cylinder would be considerably above atmospheric pressure during the first portion of the exhaust stroke, increasing the work required to expel the exhaust gases. But opening the exhaust valve earlier reduces the pressure near the end of the power stroke and thus causes some loss of useful work on this stroke. However, the overall effect of opening the valve prior to the time the piston reaches BDC results in overall gain in output.

The closing time of exhaust valve effects the volumetric efficiency, By closing the exhaust valve a few degrees after TDC (about 15^0 in case of low speed engines and 20^0 in case of high speed engines) the inertia of the exhaust gases tends to scavenge the cylinder by carrying out a greater mass of the gas left in the clearance volume. This results in increased volumetric efficiency.

Note that there may be a period when both the intake and exhaust valves are open at the same time. This is called valve over-lap (say about 15^0 in low speed engine and 30^0 in high speed engines). This overlap should not be excessive otherwise it will allow the burned gases to be sucked into the intake manifold, or the fresh charge to escape through the exhaust valve.

Table2.1–Typical valve timings for four-stroke SI engines

Position	*Theoretical*	*Actual*	
		Low speed engine	*High speed engine*
Inlet valve opens (IVO)	TDC	10° b TDC	10° b TDC
Inlet valve closes (IVC)	BDC	10° a BDC	60° a BDC
Inlet valve is open for	180°	200°	250°
Exhaust valve opens	BDC	25° b BDC	55° b BDC
Exhaust valve closes	TDC	5° a TDC	20 ° a TDC
Exhaust valve is open for	180°	210°	255°
Valve overlap	Nil	15°	30°
Spark	TDC	15° b TDC	30° b TDC

Note. Valve timing is different for different makes of engines.

b-before, a-after TDC-Top dead centre, BDC-Bottom dead centre.

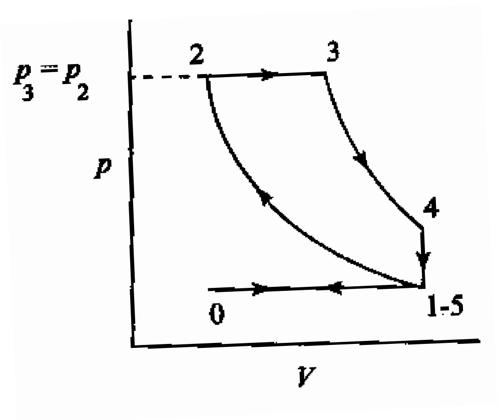
FOUR-STROKE CI ENGINES- PRINCIPLE OF OPERATION

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:



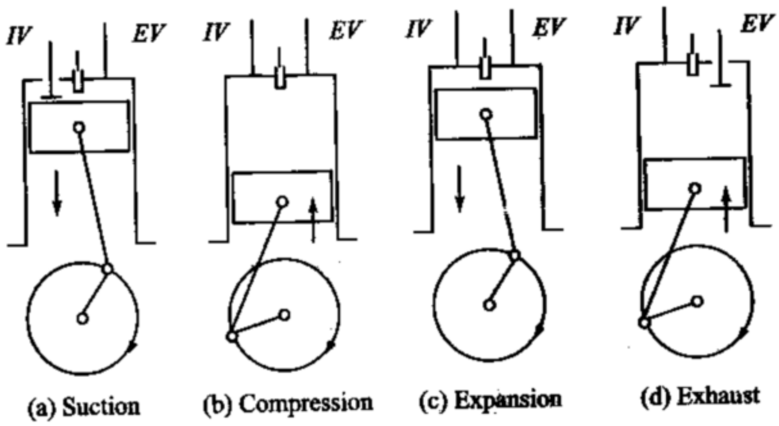


Fig.2.5 Ideal P-V Diagram **Fig.2.6 Cycle of Operation**

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 <u>higher thermal efficiency because of greater expansion.</u> 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

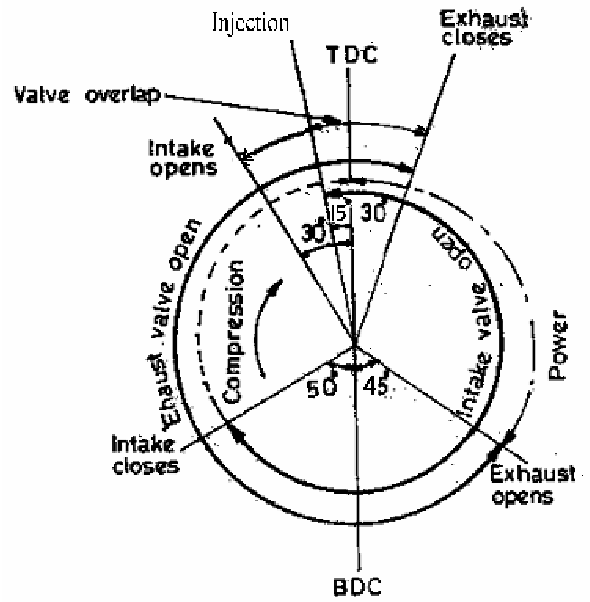
IVO about 30^0 before TDC

IVO up to 50^0 after BDC

EVO about 45^0 before BDC

EVO up to 30^0 after TDC

Injection about 15^0 before TDC



TWO-STROKE CYCLE ENGINE-PRINCIPLE OF OPERATION

In two-stroke engines the cycle is completed in two strokes, *i.e.,* one revolution of the crankshaft as against two revolutions of four-stroke cycle. The difference between two-stroke and four-stroke engines is in the method of filling the cylinder with the fresh charge and removing the burned gases from the cylinder. In a four-stroke engine the operations are performed by the engine piston during the suction and exhaust strokes, respectively. In a two stroke engine suction is accomplished by air compressed in crankcase or by a blower. The induction of compressed air removes the products of combustion, through exhaust ports. Therefore no piston strokes are required for suction and exhaust operations. Only two piston strokes are required to complete the cycle, one for compressing the fresh charge and the other for expansion or power stroke.

Types of two stroke engines

- **Based on scavenging method**
  i)Crankcase & ii) Separately scavenged engine
- **Based on scavenging process (air flow)**
  i)Cross flow scavenging,
  ii)Loop scavenging (MAN, Schnuerle, Curtis type)
  iii)Uni-flow scavenging (opposed piston, poppet valve, sleeve valve)
- **Based on overall port-timing**
  i) Symmetrical & ii) Unsymmetrical scavenging

Crankcase-scavenged two-stroke engine

Figure 2.7 shows the simplest type of two-stroke engine – the crankcase scavenged engine. Fig.2.8 shows its ideal and actual indicator diagrams. Fig.2.9 shows the typical valve timing diagram of a two-stroke engine. The air or charge is sucked through spring-loaded inlet valve when the pressure in the crankcase reduces due to upward motion of the piston during compression stroke. After the compression, ignition and expansion takes place in the usual way: During the expansion stroke the air in the crankcase is compressed. Near the end of expansion stroke piston uncovers the exhaust port, and the cylinder pressure drops to atmospheric as the combustion products leave the cylinder. Further motion of the piston uncovers transfer ports, permitting the slightly compressed air or mixture in the crankcase to enter the engine cylinder. The top of the piston sometimes has a projection to deflect the fresh air to sweep up to the top of the cylinder before flowing to the exhaust ports. This serves the double purpose of scavenging the upper part of the cylinder of combustion products and preventing the fresh charge from .flowing directly to the exhaust ports. The same objective can be achieved without piston deflector by proper shaping of the transfer port. During the upward motion of the piston from bottom dead centre, the transfer ports and then the exhaust port close and compression of the charge begins and the cycle is repeated.

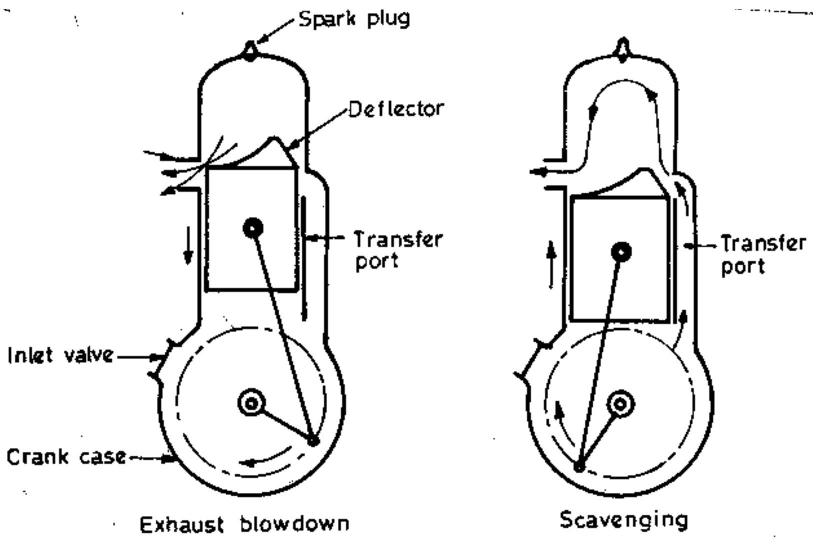


Fig.2.7-Crankcase-scavenged two-stroke engine

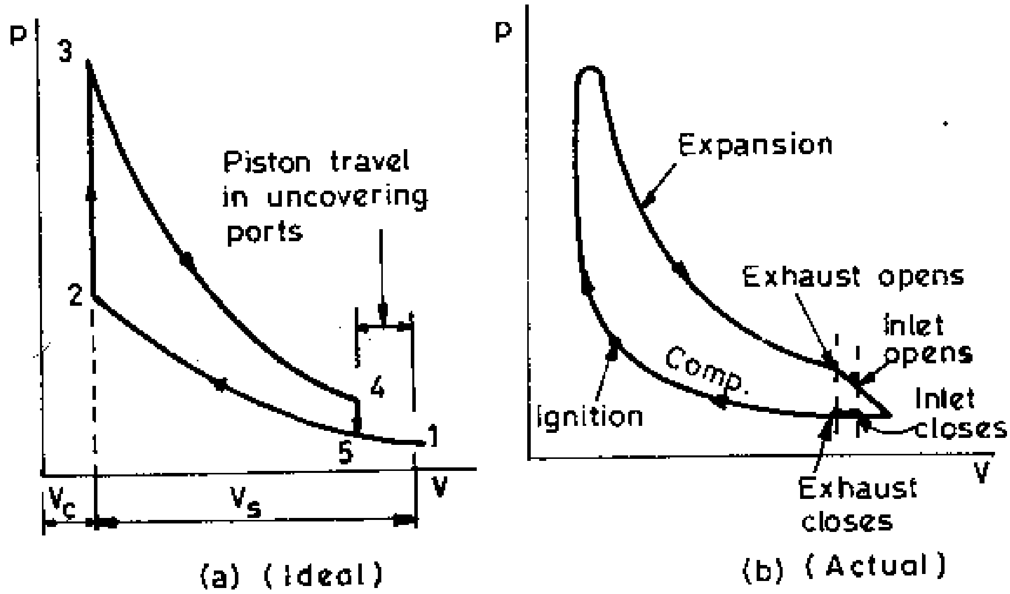


Fig. 2.8 Ideal and actual indicator diagrams for a two-stroke SI engine

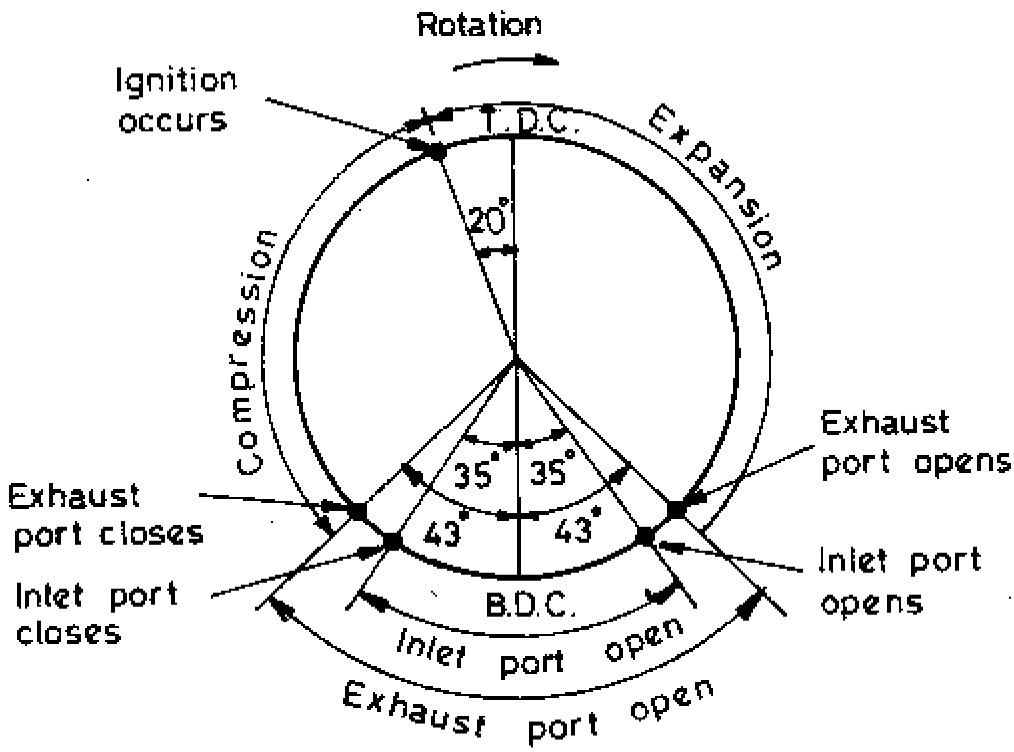


Fig.2.9. Typical valve timing diagram of a two-stroke engine

Separately scavenged engine

In the loop-scavenged engine (Fig. 2.10) an external blower is used to supply the charge, under some pressure, at the inlet manifold. During the downward stroke of the piston exhaust ports are uncovered at about 65° before bottom dead centre. At about 10^0 later the inlet ports open and the scavenging process takes place.

The inlet ports are shaped so that most of the air flows to the top of the cylinder for proper scavenging of the upper part of the cylinder. Piston deflectors are not used as they are heavy and tend to become overheated at high output. The scavenging process is moreefficient in properly designed loop-scavenged engine than in the usual crank-case compression engine with deflector piston.

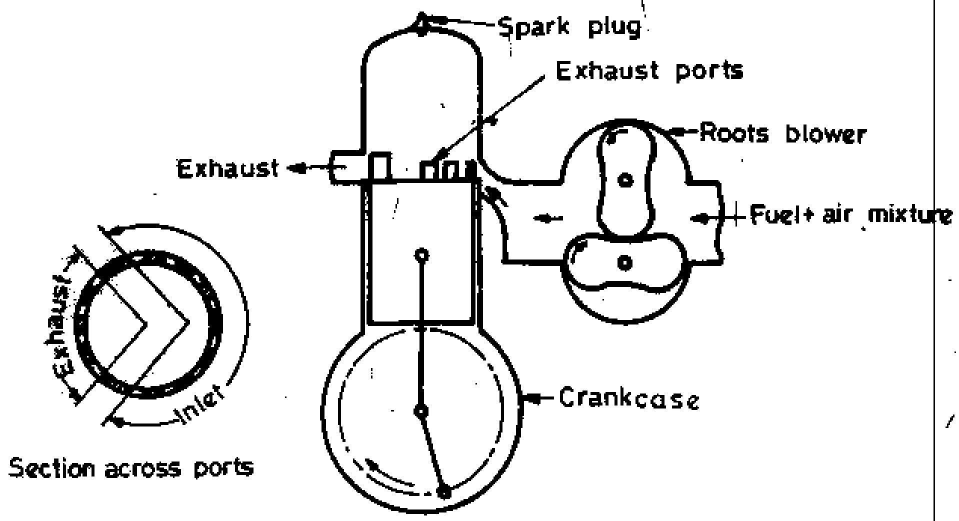


Fig.2.10. Loop-scavenged two-stroke engine (separately scavenged engine)

Opposed piston or end to end scavenged engine (*uniflow scavenged)* two stroke engine.

In this type of engine the exhaust ports or exhaust valves are opened first. The inlet ports give swirl to incoming air which prevents mixing of fresh charge and combustion products during the scavenging process. Early on the compression stroke the exhaust ports close. In loop scavenged engine the port timing is symmetrical, so the exhaust port must close *after* the inlet port closes. These timings prevent this type of engine from filling its cylinder at full inlet pressure. In the end-to-end scavenged engines counter flow within the cylinder is eliminated, and there is less opportunity for mixing of fresh charge and burnt gases. The scavenging should therefore be more efficient.

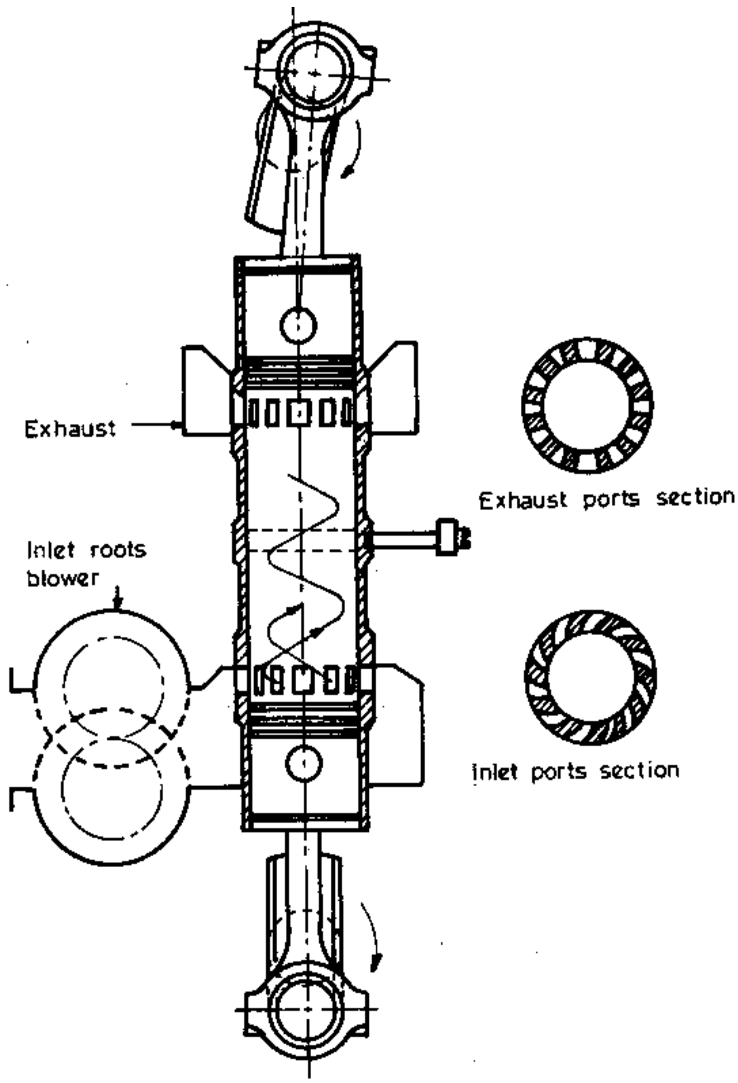


Fig. 2.11. 'End to end' scavenged or uniflow two-stroke engine

Valvetiming for two-stroke engines

Fig. 2.12(a), (b) and (c) show typical valve timing diagram for a crankcase-scavenged two-stroke engine and supercharged two-stroke engine and a four-stroke engine, respectively.

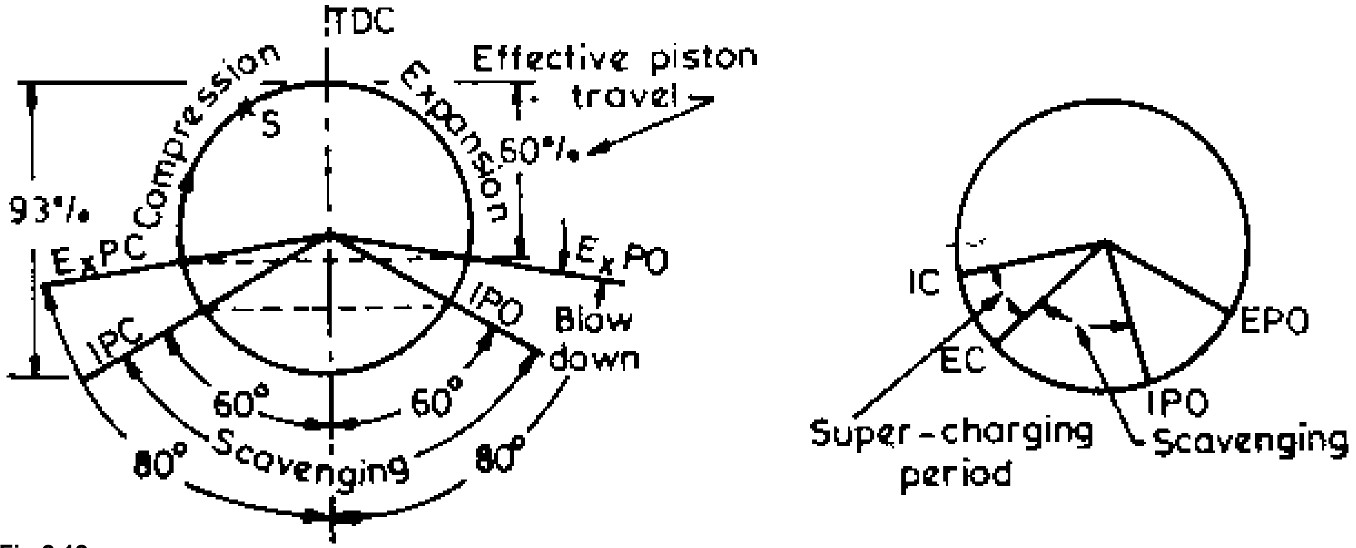
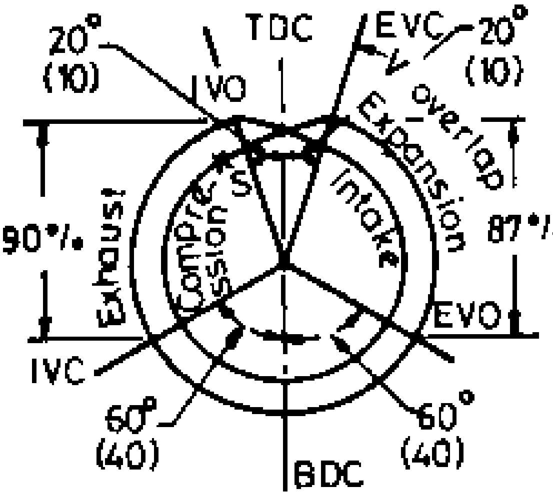


Fig 2.12

(a) Crankcase-scavenged two-stroke engine. (b) Supercharged two-stroke engine.



(c) Four-stroke cycle SI engine (Bracketed values for naturally aspirated diesel engine).

In case of two-stroke engine the exhaust port is opened near the end of the expansion stroke. With piston-controlled exhaust and inlet port arrangement the lower part of the piston stroke is always wasted so as far as the useful power output is concerned; about 15% to 40% of the expansion stroke is ineffective. The actual percentage varies with different designs. This early opening of the exhaust ports during the last part of the expansion stroke is necessary to permit blow down of the exhaust gases and, also to reduce the cylinder pressure so that when the inlet port opens at the end of the blow down process, fresh charge can enter the cylinder. The fresh charge, which comes from the crankcase for scavenging pump, enters the cylinder at a pressure slightly higher than the atmospheric pressure. Some of the fresh charge is lost due to short-circuiting. For petrol engine this means a loss of fuel and high unburnt hydrocarbons in the exhaust.

By comparing the valve timing of two stroke and four-stroke engines, (Fig. 2.12), it is clear that the time available for scavenging and charging of the cylinder of a two stroke engine is almost one-third that available for the .four-stroke engine. For a crankcase-scavenged engine the inlet port closes before the exhaust port whilst for a supercharged engine the inlet port closes after the exhaust port [Fig. 2.12 (b)]. Such timing allows more time for filling the cylinder.

Scavenging process

At the end of the expansion stroke, the combustion chambers of a two-stroke engine is left full of products of combustion. This is because, unlike four-stroke engines, there is no exhaust stroke available to clear the cylinder of burnt gases. The process of clearing the cylinder of burned gases and filling it with fresh mixture (or air}-the combined intake and exhaust process is called scavenging process. This must be completed in a very short duration available between the end of the expansion stroke and start of the charging process.

The efficiency of a two-stroke engine depends to a great degree on the effectiveness of the scavenging process, since bad scavenging gives a low mean indicated pressure and hence, results in a high weight and high cost per bhp for the engine. With insufficient scavenging the amount of oxygen available is low so that the consequent incomplete combustion results in higher specific fuel consumption. Not only that, the lubricating oil becomes more contaminated, so that its lubricating qualities are reduced and results in increased wear of piston and cylinder liners. Poor scavenging also leads to higher mean temperatures and greater heat stresses on the cylinder walls.

Thus it goes without saying that every improvement in the scavenging leads to improvement in engine and its efficiency in several directions and hence, a detailed study of scavenging process and different scavenging systems is worthwhile.

The scavenging process is the replacement of the products of combustion in the cylinder from the previous power stroke with fresh-air charge to be burned in the next cycle. In the absence of an exhaust stroke in every revolution of the crankshaft, this gas exchange process for a two-stroke engine must take place in its entirety at the lower portion of the piston travel. Obviously, it cannot occur instantaneously at bottom dead centre. Therefore, a portion of both the expansion stroke and the compression stroke is utilized for cylinder blow-down and recharging.

The scavenging process can be divided into four distinct periods

Fig. 2.13 show the pressure recordings inside the cylinder for a Flat 782 S engine. When the inlet port opens the gases expanding in the main cylinder tend to escape from it and to pre-discharge into the scavenge air manifold. This process, called pre-blowdown, ends when the exhaust port opens. As soon as the exhaust ports are open, the gases existing in the cylinder at the end of expansion stroke discharge spontaneously into the exhaust manifold and the pressure of the main cylinder drops to a value lower than that existing in the scavenge air manifold. This process, called blowdown, terminates at the moment the gas pressure inside the cylinder attains a value slightly lower than the air-pressure inside the scavenge manifold. During the third phase, called scavenging, which starts at the moment the spontaneous exhaust gases from the cylinder terminates and ends at the moment the exhaust ports are closed; the scavenge air sweeps out all residual gases remaining in the main cylinder at the end of the spontaneous exhaust and replaces them as completely as possible with fresh charge. After scavenging is complete the fresh charge continues to flow till the scavenge ports are open and the pressure in the cylinder rises. This results in better filling of the cylinder. This last part of the scavenging process is called additional-charging.

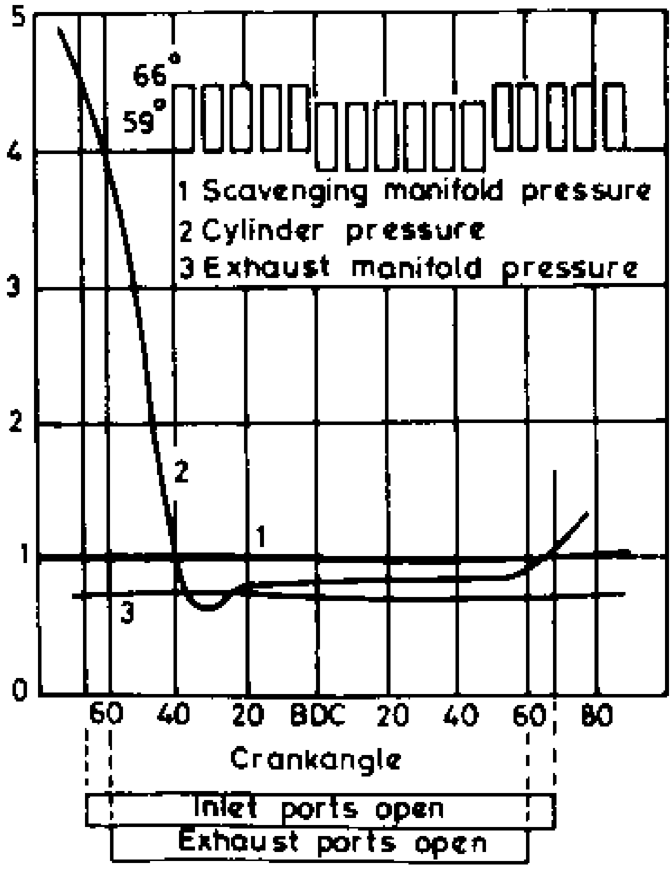


Fig. 2.13 Fiat 782 S engine standard scavenging & typical valve timing diagram of a two-stroke engine

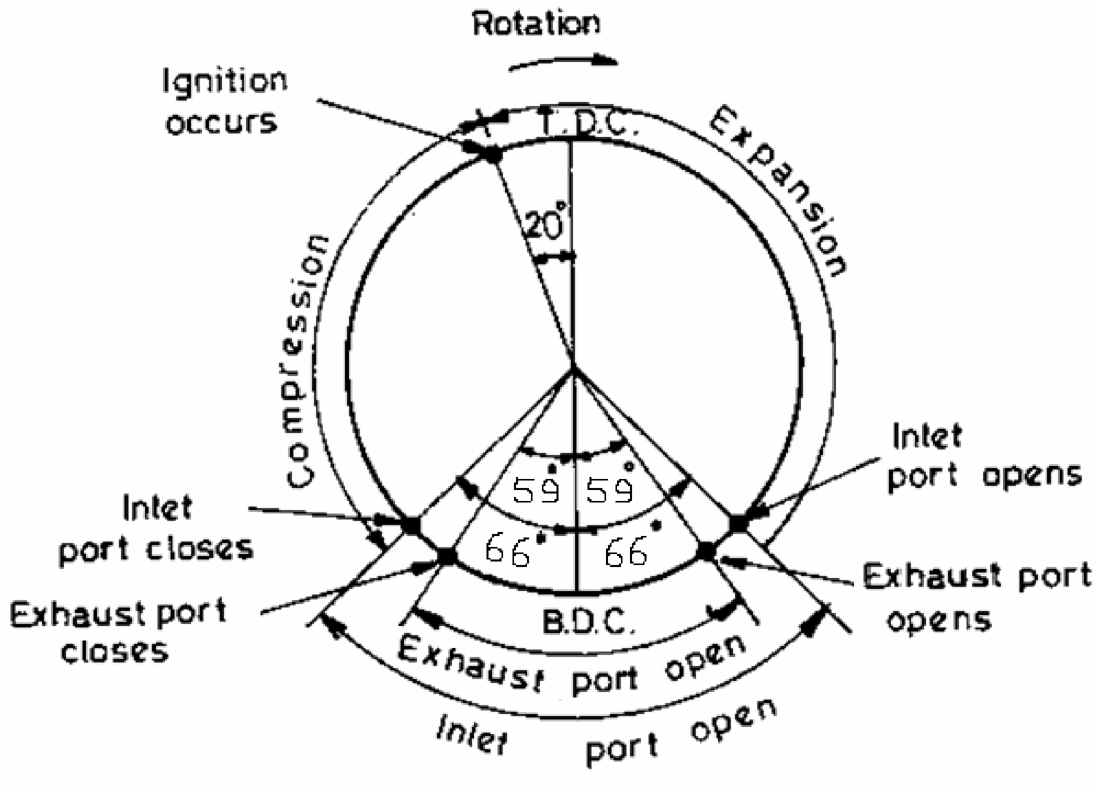


Fig.2.14shows, a typical pressure-volume diagram for a two-stroke engine. In this diagram the total piston stroke has been divided into power stroke and scavenging stroke (This division is arbitrary). The area of the p-v diagram for the power stroke depends very much on the scavenging efficiency. With proper scavenging efficiency the pressure rise due to combustion is lower and hence this area is smaller and lower thermal efficiency is obtained.

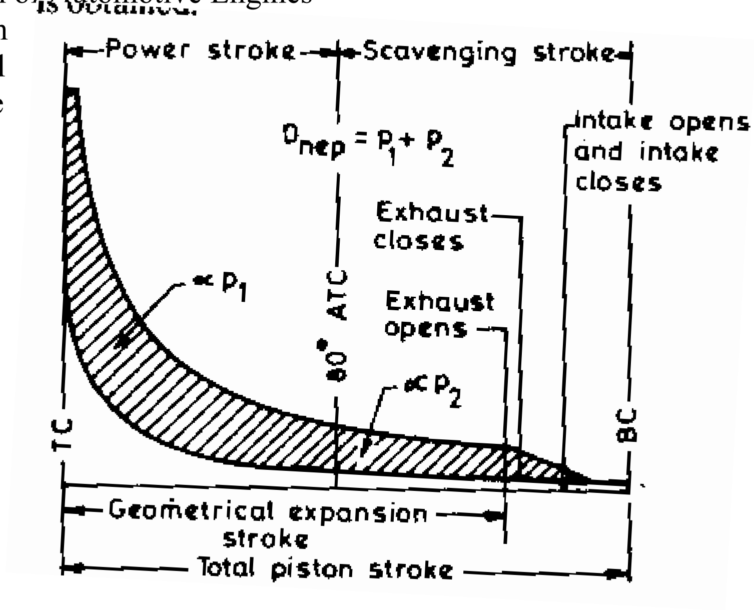


Fig. 2.14 Typical pressure-volume for a two-stroke engine.

Theoretical scavenging processes

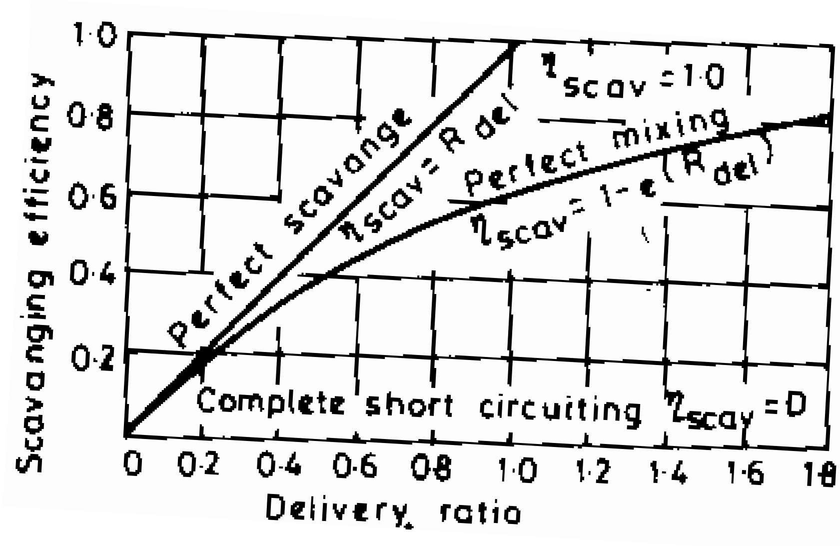


Fig. 2.15 Three theoretical scavenging processes.

Fig.2.15 illustrates three theoretical scavenging processes. They are

- **Perfect scavenging,**
- **Perfect mixing and**
- **Complete shortcircuiting**.

{ **The delivery ratio** $R_{del} = \frac{\text{mass of delivered air (or mixture) per cycle}}{\text{reference mass}}$, compares the actual scavenging air mass (or mixture mass) to that required in an ideal charging process.

(If scavenging is done with fuel-air mixture, as in spark-ignition engines, then mixture mass is used instead of air mass.)

The reference mass is defined as displaced volume × ambient air (or mixture) density.

Ambient air (or mixture) density is determined at atmospheric conditions or at intake conditions. This definition is useful for experimental purposes. For analytical work, it is often convenient to use the trapped cylinder mass m_{tr} as the reference mass. OR in other words the delivery ratio is a measure to the air (mixture) supplied to the cylinder relative to the cylinder content.

If $R_{del} = 1$, it means that the volume of the scavenging air supplied to the cylinder is equal to the cylinder volume (or displacement volume whichever is taken as reference).

Delivery ratio usually varies between 1.2 to 1.5, except for closed crankcase-scavenged, where it is less than unity.

The scavenging efficiency $\eta_{sc} = \frac{\text{mass of delivered air (or mixture) retained}}{\text{mass of trapped cylinder charge}}$, indicates to what extent the residual gases in the cylinder have been replaced with fresh air.

If $\eta_{sc} = 1$, it means that all gases existing in the cylinder at the beginning of scavenging have been swept out completely.}

(I)Perfect scavenging.

Ideally, the fresh fuel-air mixture should remain separated from the residual combustion products with respect to both mass and heat transfer during the scavenging process. Fresh air pumped into the cylinder by the blower through the inlet ports at the lower end of the cylinder pushes the products of combustion ahead of itself and of the cylinder through the exhaust valve at the other end. There is no mixing of air and products. As long as any products remain in the cylinder the flow through the exhaust valves consists of products only. However, as soon as sufficient fresh .air has entered to fill the entire cylinder volume (displacement plus clearance volume) the flow abruptly changes from one of products to one of air. This ideal process would represent perfect scavenging with no short -circuiting loss.

(ii) Perfect mixing.

The second theoretical scavenging process is perfect mixing, in which the incoming fresh charge mixes completely and instantaneously with the cylinder contents, and a portion of this mixture passes out of the exhaust ports at a rate equal to that entering the cylinder. This homogeneous mixture consists initially of products of combustion only and then gradually changes to pure air. This mixture flowing through the exhaust ports is identical with that momentarily existing in the cylinder and changes with it. For the case of perfect mixing the scavenging efficiency can be represented by the following equation:

$\eta_{sc} = 1 - e^{-R_{del}}$, where η_{sc} and R_{del} are scavenging efficiency and delivery ratio respectively.

This is plotted in Fig. 2.15. The result of this theoretical process closely approximates the results of many actual scavenging processes, and is thus often used as a basis of comparison.

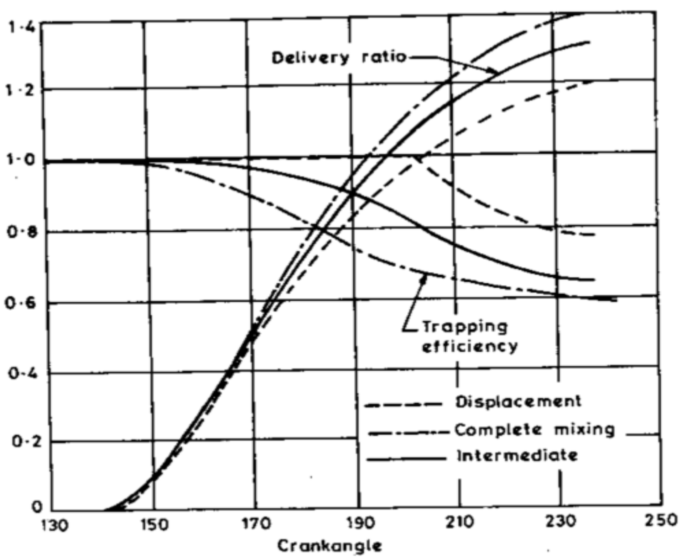
(iii)Short-circuiting.

The third type of scavenging process is that of short-circuiting in which the fresh charge coming from the scavenge manifold directly goes out of the exhaust ports without removing any residual gas. This is a dead loss and its occurrence must be avoided.

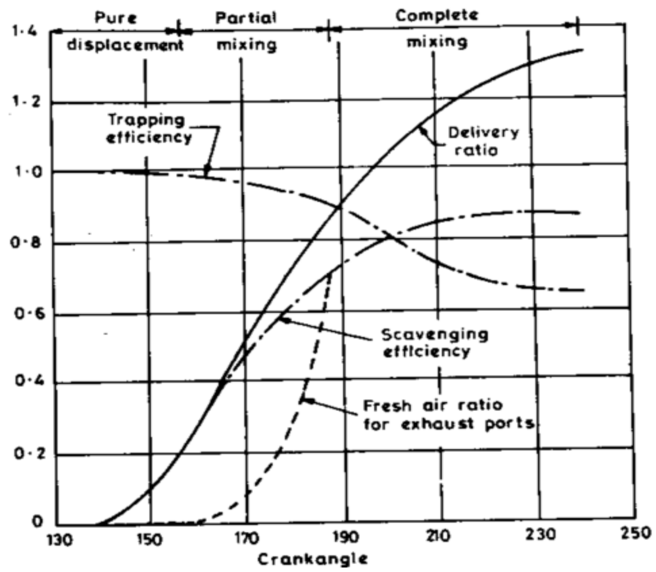
The actual scavenging process is neither one of perfect scavenging nor perfect mixing. It probably consists partially of perfect scavenging, mixing and short-circuiting.

Fig. 2.16shows the delivery ratio and trapping efficiency variation with crankangle for three different scavenging modes., *i.e.,*perfect scavenging (displacement), perfect mixing and intermediate scavenging.

Fig. 2.17shows the scavenging parameters for the intermediate scavenging. This represents the actual scavenging process. It can be seen from this Fig. that a certain amount of combustion products is initially pushed out of the cylinder without being diluted by fresh air. Gradually, mixing and short circuiting causes the out flowing products to be diluted by more and more fresh air until ultimately the situation is the same as for perfect mixing, *i.e.,* the first phase of the scavenging process is a perfect scavenging process which then gradually changes into a complete mixing process.



Fig,2.16 Delivery ratio and efficiency variation with crankcase for three different scavenging modes.



` **Fig. 2.17 Scavenging parameters for intermediate scavenging**

Scavenging parameters ..

The delivery ratio - The delivery ratio represents the ratio of the air volume, under the ambient conditions of the scavenge manifold, introduced per cycle and a reference volume. This reference volume has been variously chosen to be displacement volume, effective displacement volume, total cylinder volume or total effective cylinder volume. Since it is only the quantity or charge in the remaining total cylinder volume at exhaust port closure that enters into the combustion, the total effective cylinder volume should be preferred. The delivery ratio is mass of fresh air delivered to the cylinder divided by a reference mass,

i.e., $R_{del} = \dfrac{\text{mass of delivered air (or mixture) per cycle}}{\text{reference mass}}$,

The delivery ratio compares the actual scavenging air mass (or mixture mass) to that required in an ideal charging process. OR The delivery ratio is a measure to the air (mixture) supplied to the cylinder relative to the cylinder content.

If $R_{del} = 1$, it means that the volume of the scavenging air supplied to the cylinder is equal to the cylinder volume (or displacement volume whichever is taken as reference).

Delivery ratio usually varies between 1.2 to 1.5, except for closed crankcase-scavenged, where it is less than unity.

(If scavenging is done with fuel-air mixture, as in spark-ignition engines, then mixture mass is used instead of air mass.) The reference mass is defined as displaced volume × ambient air (or mixture) density.

Ambient air (or mixture) density is determined at atmospheric conditions or at intake conditions. This definition is useful for experimental purposes. For analytical work, it is often convenient to use the trapped cylinder mass m_{tr} as the reference mass.

The trapping efficiency - The amount of fresh charge retained in the cylinder is not same as that supplied to the cylinder because some fresh charge is always lost due to short-circuiting. Therefore, an additional term, trapping efficiency, is used to indicate the ability of the cylinder to retain the fresh charge. It is defined as the ratio of the amount of charge retained in the cylinder to the total charge delivered to the engine, i.e., $\eta_{tr} = \dfrac{\text{mass of delivered air (or mixture) retained}}{\text{mass of delivered air (mixture)}}$

Trapping efficiency indicates what fraction of the air (or mixture) supplied to the cylinder is retained in the cylinder.. This is mainly controlled by the geometry of the ports and the overlap time.

The scavenging efficiency Scavenging efficiency is the ratio of the mass of scavenge air which remains in the cylinder at the end of the scavenging to the mass of the cylinder itself at the moment when the scavenge and exhaust ports of valves are fully closed. It is given by

$\eta_{sc} = \dfrac{\text{mass of delivered air (or mixture) retained}}{\text{mass of trapped cylinder charge}}$,

indicates to what extent the residual gases in the cylinder have been replaced with fresh air.

If $\eta_{sc} = 1$, it means that all gases existing in the cylinder at the beginning of scavenging have been swept out completely.

The purity of the charge: purity $= \dfrac{\text{mass of air in trapped cylinder charge}}{\text{mass of trapped cylinder charge}}$, indicates the degree of dilution, with burned gases, of the unburned mixture in the cylinder.

The charging efficiency $\eta_{sc} = \dfrac{\text{mass of delivered air (or mixture) retained}}{\text{displaced volume x ambient density}}$, indicates how effectively the cylinder volume has been filled with fresh air (or mixture)

Relative cylinder charge.- The air or mixture retained, together with the residual gas, remaining in the cylinder after flushing out the products of combustion constitutes the cylinder charge. Relative cylinder charge is a measure of the success of filling cylinder irrespective of the composition of charge. The relative cylinder charge may be either more or less than unity depending upon the scavenging pressure and port heights.

Excess air factor, λ - The value (R_{del}-1) is called the excess air factor. If the delivery ratio is 1.4, the excess air factor is 0.4.

Classification based on scavenging process

The simplest method of introducing the charge into the cylinder is to employ crankcase compression as shown in Fig.2.7. This type of engine is classified as the crankcase scavenged engine.

In another type, a separate blower or a pump (Fig.2.8) may be used to introduce the charge through the inlet port. They are classified as the separately scavenged engines.

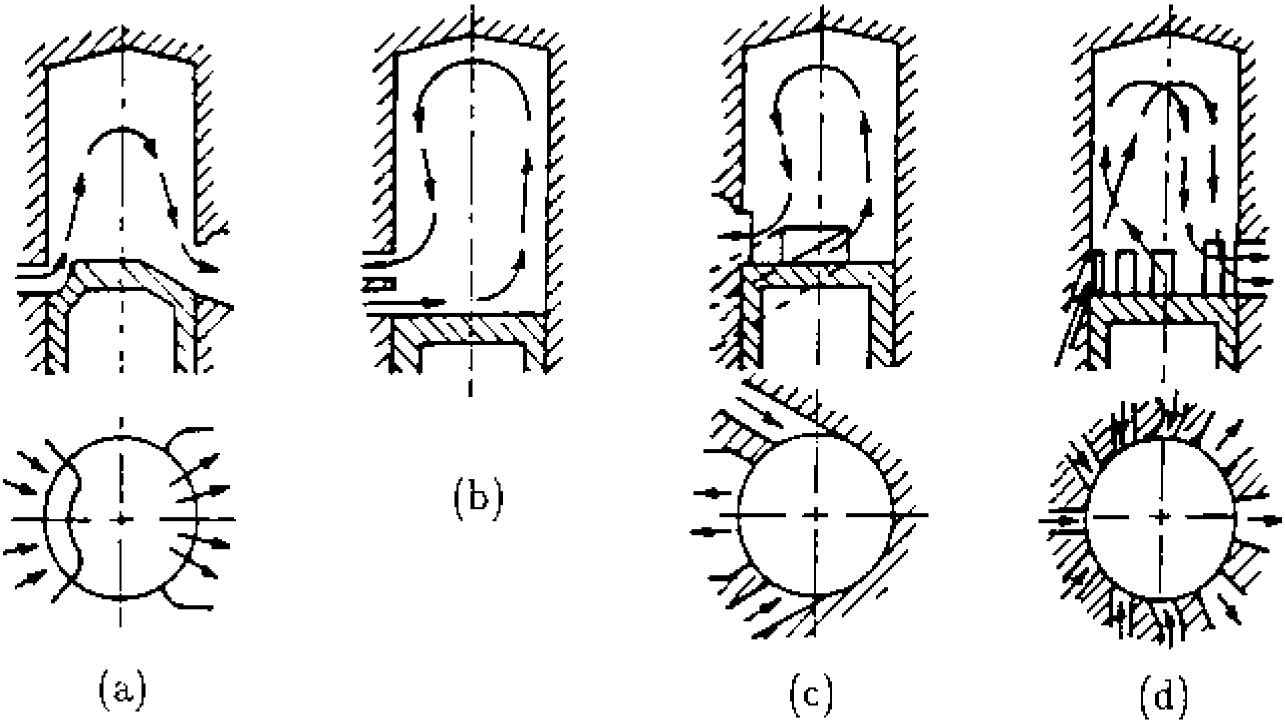


Fig.2.16 Methods of Scavenging (a)Cross Scavenging (b) Loop Scavenging, M.A.N. Type (c)Loop Scavenging Schnuerle Type, (d) Loop Scavenging, Curtis Type

Another classification of two-stroke cycle engines is based on the air flow.

Based on a transversal air stream, the most common arrangement is cross scavenging, illustrated in Fig.2.16 (a). Most small engines are cross-scavenged. The cross scavenging system employs inlet and exhaust ports placed in opposite sides of the cylinder wall. The incoming air is directed upward, to combustion chamber on one side of the cylinder and then down on the other side to force out the exhaust gases through the oppositely located exhaust ports. This requires that the air should be guided by use of either a suitably shaped deflector formed on piston top or by use of inclined ports. With this arrangement the engine is structurally simpler than that with the uniflow scavenging, due to absence of valves, distributors, and relative drive devices. The inlet and exhaust of gases is exclusively controlled by the .opening and closure of ports by piston motion. The main disadvantage of this system is that the scavenging air is not able to get rid of the layer of exhaust gas near the wall resulting in poor scavenging. Some of the fresh charge also goes directly into the exhaust port. The result of these factors is poor bmep of cross-scavenged engines.

Based on a transversal air stream, with loop or reverse scavenging, the fresh air first sweeps across the piston top, moves up and then down and finally out through the exhaust. Loop or reverse scavenging avoids the short -circuiting of the cross-scavenged engine and thus improves upon its scavenging efficiency. The inlet and exhaust ports are placed on the same side of the cylinder wall.

In the M.A.N. type of loop scavenge, Fig.2.16(b), the exhaust and inlet ports are on the same side, the exhaust above the inlet.

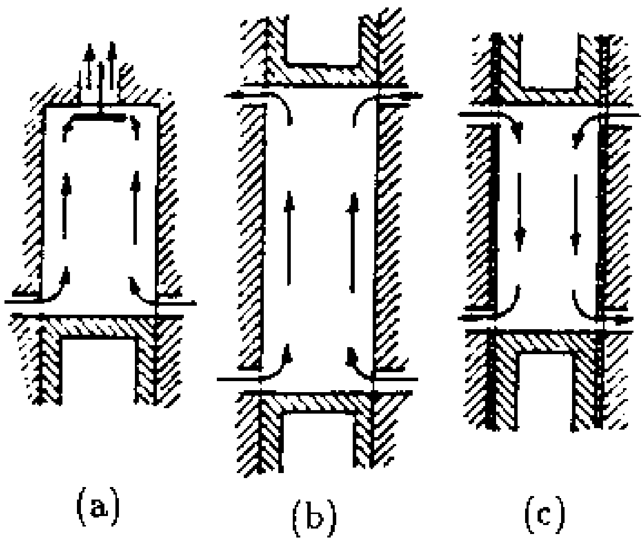
In the Schnuerle type, Fig.2.16(c), the ports are side by side. the inlet ports are placed on both sides of the exhaust ports so that the incoming air enters in two streams uniting on the cylinder wall opposite the exhaust ports, flows upwards, turns under the cylinder head, then flows downwards the other side to the exhaust ports. Such a system of air deflection reduces the possibilities of short-circuiting to minimum. With this system flat-top pistons without deflectors are used. The speed of loop or reversed scavenged engine is not restricted by mechanical limitations because valves are not used, the charging process being controlled by the piston only. The speed can thus, exceed that of valve controlled two-stroke engines. Owing to the absence of cams, valves and valve gear, engines are simple and sturdy. They have a high resistance to thermal stresses and are, thus, well suited to higher supercharge. The major mechanical problem with a loop scavenged two-stroke engine is that of obtaining an adequate oil supply to the cylinder wall consistent with reasonable lubricating oil consumption and cylinder wear. This difficulty arises because when the piston is at top dead centre there is only a very narrow sealing belt available to prevent leakage of oil from crankcase into the exhaust ports. Since for loop scavenging greater cylinder distance is necessary to accommodate scavenge-air passage between the cylinder, a strong connecting rod and crankshaft need for supercharged engine can be used.

The Curtis type of scavenging, Fig.2.16(d), is similar to the Schnuerle type, except that upwardly directed inlet ports are placed also opposite the exhaust ports.

The most perfect method of scavenging is the uniflow method, based on a unidirectional air stream. The fresh air charge is admitted at one end of the cylinder and the exhaust escapes at the other end flowing through according to parallel flow lines normally having a slight rotation to stabilize the vertical motion. Air acts like an ideal piston and pushed on the residual gas in the cylinder after the blowdown period and replaces it at least in principle, throughout the cylinder. The air flow is from end to end, and little short-circuiting between the intake and exhaust openings is possible. Due to absence, at least in theory, of any eddies or turbulence it is easier in a uniflow scavenging system to push the products of combustion out of the cylinder without mixing with it and short circuiting. Thus, the uniflow system has highest scavenging efficiency. Construction simplicity is, however, sacrificed because this system requires either opposed pistons, poppet valves or sleeve valve all of which increases the complication.

The three available arrangements for uniflow scavenging are shown in Fig.2.17 A poppet valve is used in (a) to admit the inlet air or for the exhaust, as the Case may be. In (b) the inlet and exhaust ports are both controlled by separate pistons that move in opposite directions. In (c) the inlet and exhaust ports are controlled by the combined motion of piston and sleeve. In an alternative arrangement one set of ports is controlled by the piston and the other set by a sleeve or slide valve. All uniflow systems permit unsymmetrical scavenging and supercharging.

Fig.2.17 Uniflow Scavenging
(a) Poppet Valve
(b) Opposed Piston
(c) Sleeve Valve



Reverse flow scavenging is shown in Fig.2.17 In this type the inclined ports are used and the scavenging air is forced on to the opposite wall of the cylinder where it is reversed to the outlet ports. One obvious disadvantage of this type is the limitation on the port area. For long stroke engines operating at low piston speeds, this arrangement has proved satisfactory.

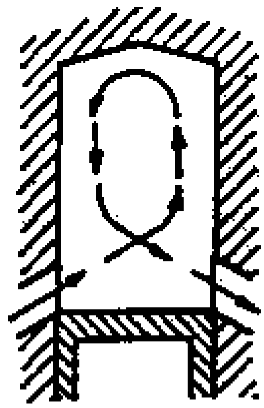
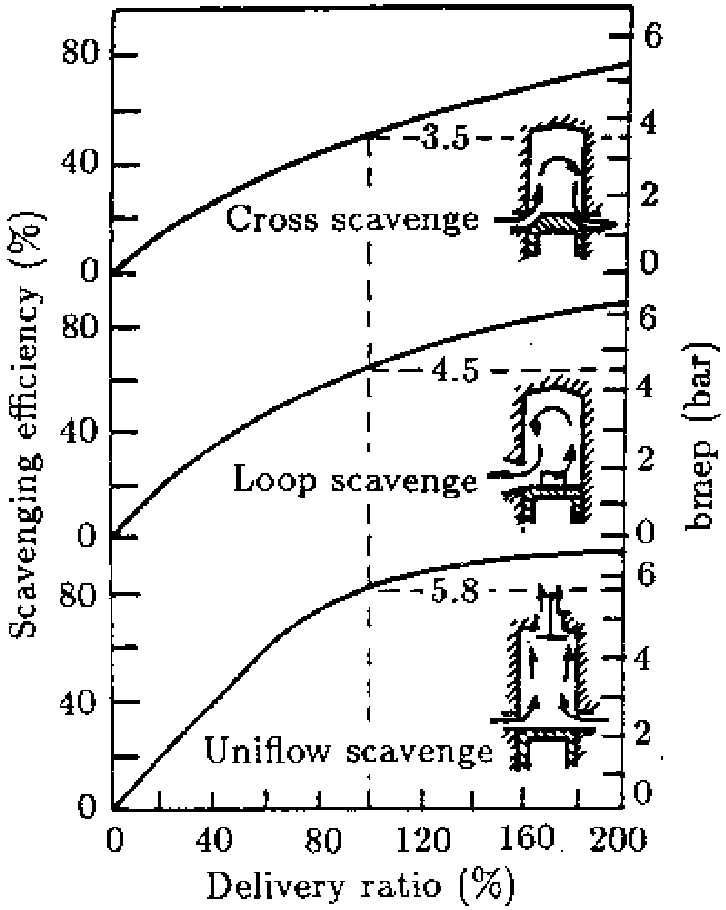


Fig2.17 Reverse Flow Scavenging



An interesting comparison of the merits of two cycle engine air scavenging methods is illustrated in Fig.2.18. In fact, specific output of the engine is largely determined by the efficiency of the scavenging system-and is directly related to the brake mean effective pressure. As shown in Fig.2.18 scavenging efficiency varies with the delivery ratio and the type of scavenging. In this respect cross scavenging is least efficient and gives the lowest brake mean effective pressure. The main reason for this is that the scavenging air flows through the cylinder but does not expel the exhaust residual gases effectively. Loop scavenging method is better than the cross scavenging method. Even with a delivery ratio of 1.0 in all cases the scavenging efficiencies are about 53, 67 and 80 per cent for cross scavenging, loop scavenging and uniflow scavenging systems with corresponding values of *bmep* as 3.5,4.5 and 5.8 bar.

Fig.2.18 Scavenging Efficiency

Comparison of different scavengingsystems

Fig.2.19 compares the scavenging efficiencies of three different types of scavenging system.
The cross-scavenging system employs inlet and exhaust ports placed in opposite sides of the cylinder wall. In the loop scavenging system, inlet and exhaust ports are in the same side of the cylinder wall and in uniflow scavenging system, the inlet and exhaust port are at opposite ends of the cylinder.
It can be seen that uniflow scavenging gives by far the best scavenging, that loop scavenging is good, and that in .general, cross-scavenging is the worst.

The scavenging curve for the uniflow scavenging is very near to that of perfect scavenging that for loop scavenging is near the perfect mixing. With good loop scavenging the scavenging curve is generally above the perfect mixing curve and that of cross-scavenging engines it is, generally, below the perfect mixing curve.

Table 2.2 compares the port areas available for different scavenging systems. Largest flow areas are available with uniflow system. In such a case the whole circumference of cylinder wall is available and the inlet port area can be as high as 35 per cent of the piston area. Due to the use of exhaust valve the exhaust flow area is small - about 18 per cent. In cross-scavenging the size of the inlet and exhaust ports is limited to about 25 and 18 per cent of piston area respectively because the ports are located on the opposite sides of cylinder wall. Schurnle type of loop scavenging requires that both the ports must be located within about three-quarters of the cylinder circumference. This limits the size of inlet and exhaust ports to about 18 and 14 per cent of piston area only. The data for a typical four-stroke engine are also given for comparison. However, while comparing with the four-stroke engine it must be kept in mind that though the flow area is small, the time available for flow is almost three times more than that available for the two-stroke engine.

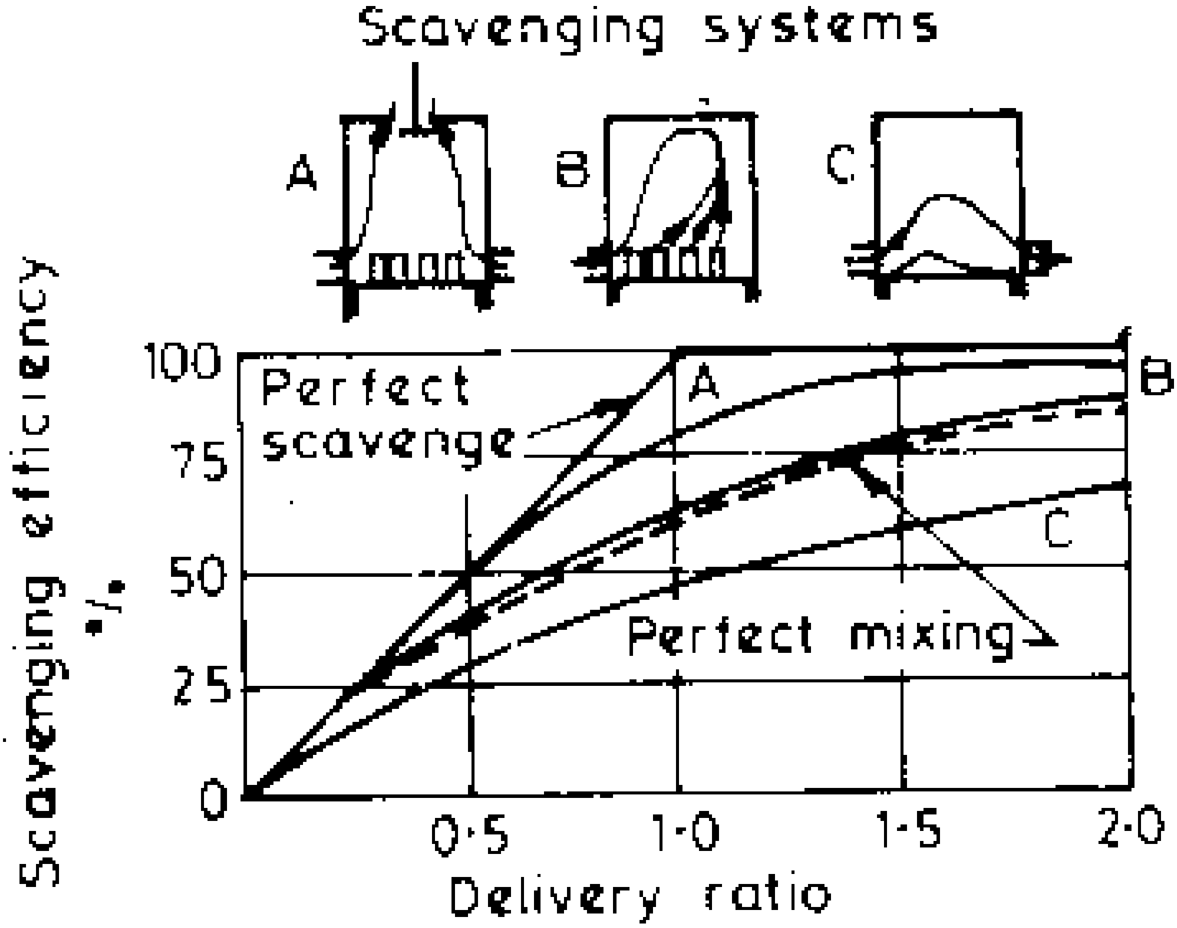


Fig. 2.19 scavenging efficiency, versus delivery ratio of different scavenging system.

Table 2.2 Typical values for areas for different scavenging systems

S.No.	*Scavenging Type*	*Port area, % of Piston area*	
		Scavenging port area	*Exhaust port area*
1.	Parallel flow scavenging with poppet or exhaust port	35	18
2.	Cross scavenging	25	21
3.	Schüürle scavenging	18	14
4.	Four-stroke engine	11	10

Loop or cross-scavenged engines with their inlet ports limited half of the cylinder circumference fall in low speed category. Uniflow scavenged engines with adequate air inlet port are and limited exhaust port areas fall in medium speed category, whilst the opposed piston engine takes on to high speeds because of its high rate of exhaust port opening, freedom from valve gear speed limits, good scavenging and perfect balancing. Un-supercharged uniflow engine has a considerable higher mean effective pressure than the loop-scavenged engine. There is more freedom in design of combustion chamber for loop scavenging. This results in low fuel consumption and the engine is simple to make and easy to produce. Table 2.3 compares the typical bmep values obtainable with different types of scavenging systems. The output of both uniflow and loop scavenged engines is limited 'by the thermal stresses imposed. But the loop scavenged engine due to its simple cylinder head can better withstand the thermal stresses.

Table 2.3 Typicalvaluesof bmep for the C.I. two-stroke oil engines

Types of scavenging	*Range bar*	*Average bar*
Uniflow engine opposed piston	5-8.5	6.3
Uniflow engine with one piston	4.2-7	6
Return flow scavenging, separate pump	3.7-5	4.4
Step piston engine	2.8-4	3.2
Cross-head engine, straight piston	2.4-3.4	2.8
Crankcase scavenged engine	2-2.8	2.5

Table 2.4compares the representative port timings for different types of two-stroke engines.

Table 2.4. Port timings for different two-stroke engines

Engine type	*Exhaust*		*Lead angle deg.*	*Scavenge*		*Super charge deg.*
	Opens	*Closes*		*Open*	*Closes*	
1. Crankcase scavenging	60-76	60-67	10-20	45-60	45-60	0
2. Loop scavenging	60-76	60-76	10-20	45-60	45-60	0
3. Uniflow scavenging	80-90	38-59	20-40	38-55	38-55	0-70
4. Opposed piston	60-67	50-67	10-20	50-65	50-65	10-20

Port design

The Design of the inlet and exhaust ports for two stroke engines depends on various parameters. Some of the important basic parameters are;

a) Scavenging method
b) Shape, inclination & width of ports
c) Amount of air/charge delivered
d) Scavenging pressure
e) Mean inlet velocity –fn. Of pr. Ratio, temp. of scavenging & scavenging factor
f) Duration(crank angle) of port opening & average port height uncovered by piston
   Blowdown time area (for exhaust)–[which is a fn. of temperature of exhaust Gas, expansion end volume(fn. of displacement volume), exhaust Gas pr., scavenging pr., & indicated mean effective pressure]
g) Inlet duration, exhaust lead* & hence exhaust duration
h) Number of ports & height of ports

*during exhaust Lead, only exhaust port is kept open, & during super charging only inlet port is kept open.

- **THE DIFFERENT SCAVENGING METHODS ARE AS FOLLOWS**

➢ *BASED ONSCAVENGING PROCESS(AIR FLOW)*

I. CROSS FLOW -for low power o/p engines eg. Two wheelers,
Simple, but more short circuiting, hence more charge loss, super charging is not possible. It is found that port position is limited with in 50% of circumference.

II. LOOP FLOW -for medium o/p engines.
Air takes loop, less short circuiting, hence less charge loss

A. MAN type -intake & exh. ports positioned one below the other. -Good
B. SCHNURLE type-intake & exh. ports positioned side by side. -Better
C. CURTIS type -intake on one side & exhaust on the other side. -Best

A comparison

III. UNIFLOW (BEST) –for very High o/p engines
Ex. large power marine engines, locomotive engines etc
As intake port is on one side & exhaust port on the other side. & the flow is uni-directional, ports can be wider. Residual gases are low. Ports can be located all around the circumference. Opposed piston engines also use this type. Ports with poppet valves & Sleeve valves have been used.

➢ *BASED ONSCAVENGING METHOD*

I.CRANKCASE SCAVENGED ENGINE (crank case compression)
-petroil lubrication is adopted. Hence lubricating oil is also burnt. So pollution is more. Compression is bad, more petrol consumption, and more residual gases. Generally used along with symmetrically scavenged engine, but lower delivery ratio (generally 0.7), Simple and suitable for small engines. Suitable for low o/p engines (5-20bhp)

II.SEPARATE BLOWER / PUMP SCAVENGED ENGINE
-higher scavenging pressure & delivery ratio is possible. Residual gases are low. Used in bulky arrangements i.e. above 100 hp engines

- *BASED ON OVERALL PORT TIMING*

I. SYMMETRICAL PORT TIMING - *EPO-IPO-IPC-EPC*

-Opening and closing of the ports by the piston is symmetrical.
Advantage-arrangement of the mechanism is very simple.
Disadvantage- more short circuiting, hence more charge loss, super charging is not possible. Suitable for low power o/p engines up to 5bhp i.e. scooters / moped engines.

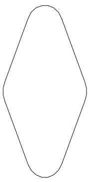
II. UN-SYMMETRICAL PORT TIMING - *EPO-IPO-EPC-IPC*

-Opening and closing of the ports by the piston is un-symmetrical. Mechanism is complex.
Advantages- super charging is possible - by the following ways
Supercharging valve-rotary valves,
Poppet valves by suitably designing the cam mechanism,
Using sleeve /slide valve, but it is mechanically complicated,
& using opposed piston

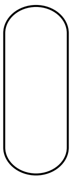
- **The common different Shapes of ports are as follows**

Rectangular -BEST

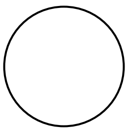
With rounded corners, which gives maximum flow area & smooth edges reduce friction



&



Rhomboidal & Oblong -good w.r.to ring entrance avoidance



Circular-only some applications (only for intake)

Inclination -is given for better mixing, scavenging, turbulence, swirl and combustion.

Width -for Uniflow scavenging -0.6πD (entire circumference available for porting
-for lLoop scavenging -0.2πD (both ports are on same side of the wall)
-for Crossflow scavenging -0.3πD (50% of circumference is available for porting)

Ports should be sufficiently wider for max. flow area, But should not create problem of piston ring entrance into it.

- Amount of air/charge delivered

The delivery ratio is a measure of the air (mixture) supplied to the cylinder relative to the cylinder content.

The delivery ratio $R_{del} = \frac{\text{mass of delivered air (or mixture) per cycle}}{\text{reference mass}}$,

If $R_{del} = 1$, it means that the volume of the scavenging air supplied to the cylinder is equal to the cylinder volume (or displacement volume whichever is taken as reference).
Delivery ratio usually varies between 1.2 to 1.5, except for closed crankcase-scavenged, where it is less than unity.
$R_{del} = 0.7$ to 0.8 – for crank case scavenging

$$\left.\begin{array}{l} R_{del} = 1.4 \text{ –normal value} \\ R_{del} = 1.3 \text{ –for fuel economy} \\ R_{del} = 1.5 \text{ –for high o/p} \end{array}\right\} \text{For separately scavenged engines}$$

The scavenging efficiency $\eta_{sc} = \dfrac{\text{mass of delivered air (or mixture) retained}}{\text{mass of trapped cylinder charge}}$,

Indicates to what extent the residual gases in the cylinder have been replaced with fresh air.
If $\eta_{sc} = 1$, i.e. all gases existing in the cylinder at the beginning of scavenging have been swept out completely}

- Scavenging pressure
  Proper scavenging pressures to be adopted for the respective scavenging method

- Mean inlet velocity

Mean inlet velocity to be calculated, which is a function of pressure ratio, temp. of scavenging & scavenging factor.

- Duration(crank angle) of port opening & average port height uncovered by piston

With Duration (crank angle) of port opening, average port height & port timing can be calculated.

- Number of ports & height of ports.

No. of ports are selected to ensure enough (max.) width, with sufficient bridge to sustain mechanical and thermal load & to avoid piston ring failure i.e. entering in port area. After selecting no. of ports, width of the ports may be calculated and adopted. The height of ports is a major factor in timing of ports.

The flow of gases through a two-stroke cycle engine is diagrammatically represented in fig. The hatched areas represent fresh air or mixture and the cross hatched areas represent combustion gases. The width of the channels represents the quantity of the gases expressed by volume at NTP condition.

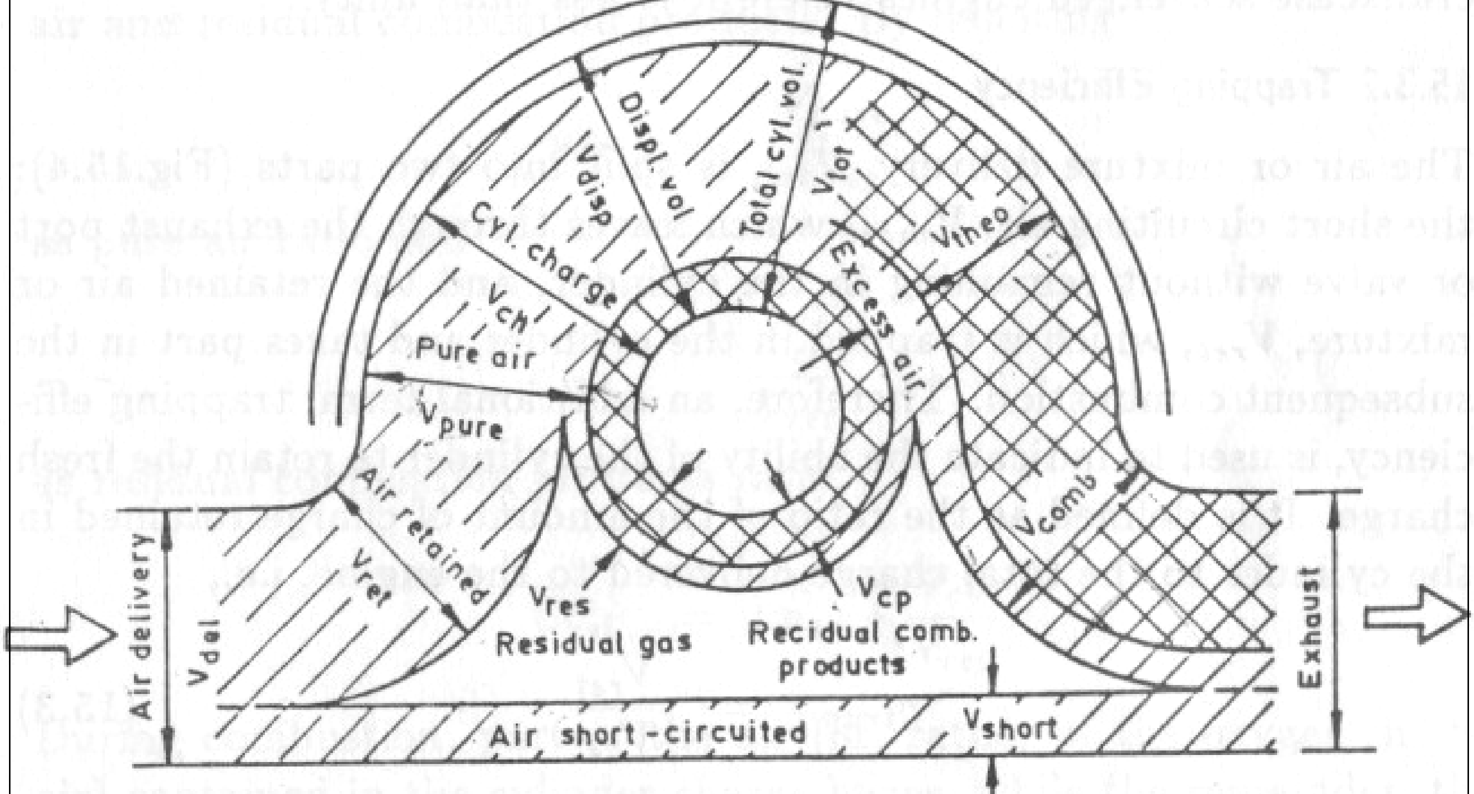


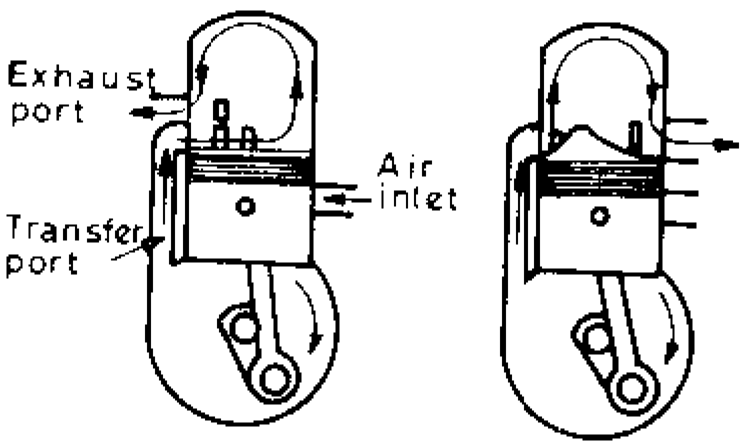
Fig. Scavenging Diagram for Two-stroke Cycle SI Engine

Scavenging pumps

Since the pumping action is not carried out by the piston of a two-stroke engine, a separate pumping mechanism, called the *scavenging pump,* is required to supply scavenging air to the cylinder. Different types of scavenging pumps used range from crankcase compression, piston type blowers to roots blower. The design of a two-stroke engine is significantly affected by the type of scavenging pump used; hence a careful selection of the scavenging pump is a pre-condition to good performance.

Crankcase Scavenging. The most obvious and cheapest in initial cost is the use of crankcase for compressing the incoming air and then transferring it to the cylinder through a transfer port. Fig.2.20 shows such a system. This system is, however, very uneconomical and inefficient in operation. This is because the amount of air which can be used for scavenging is less than the swept volume of the cylinder due to low volumetric efficiency of the crankcase which contains a large dead space. Thus, the delivery ratio of a crankcase scavenged engine is always less than unity.

Since the delivery ratio is less than unity it is not possible to scavenge the cylinder completely of the products of combustion and some residual gases always remain in the cylinder. This results in low mean effective pressure for the crankcase scavenged engine. Typical values are 3 to 4 bar. The output of the engine is strictly limited because the amount of the charge transferred through the transfer port is only 40-50% of the cylinder volume.



(a) two ports (b) three ports

Fig. 2.20 Two-stoke crankcase scavenged engines

A further disadvantage is that the oil vapors from the crankcase mixes with the scavenging air. This results in high oil consumption. Because of these disadvantages the crankcase scavenging is not preferred and for high output two-stroke engines a scavenging pump is a must.

Piston, Roots, and Centrifugal blowers

Piston type blowers as shown in Fig.2.21(a) are used only for low speed and single or two cylinder engines. For all other type of engines either roots or centrifugal blowers are used. The roots blower is preferred for small and medium output engines. While the centrifugal blower, is preferred for large and high output engines. From Fig. 2.22 it is clear that the centrifugal blower has a relatively flat characteristic curve compared to the steep characteristic curve of the 'roots blower. An increase in the flow-resistance due to deposits, etc., thus, has a much greater effect on the scavenging air; output of a centrifugal blower than on that of a roots blower. If deposits accumulate, an engine having a centrifugal blower will start smoking earlier than that having a roots blower. Therefore, roots blower is preferred due to its lower sensitivity to flow resistance changes for systems where space for exhaust ports is limited.

The control of air delivery of centrifugal blowers can be done by throttling the air on the intake side. This, however, would not reduce the scavenging power required by the centrifugal blower. In the roots blower the air delivery is controlled by a throttle-actuated by-pass valve between blower inlet and outlet. Such a control divides the air-flow into two parts and only half the flow passes through the engine. This saves a substantial amount of scavenging power and hence results in lower specific fuel consumption.

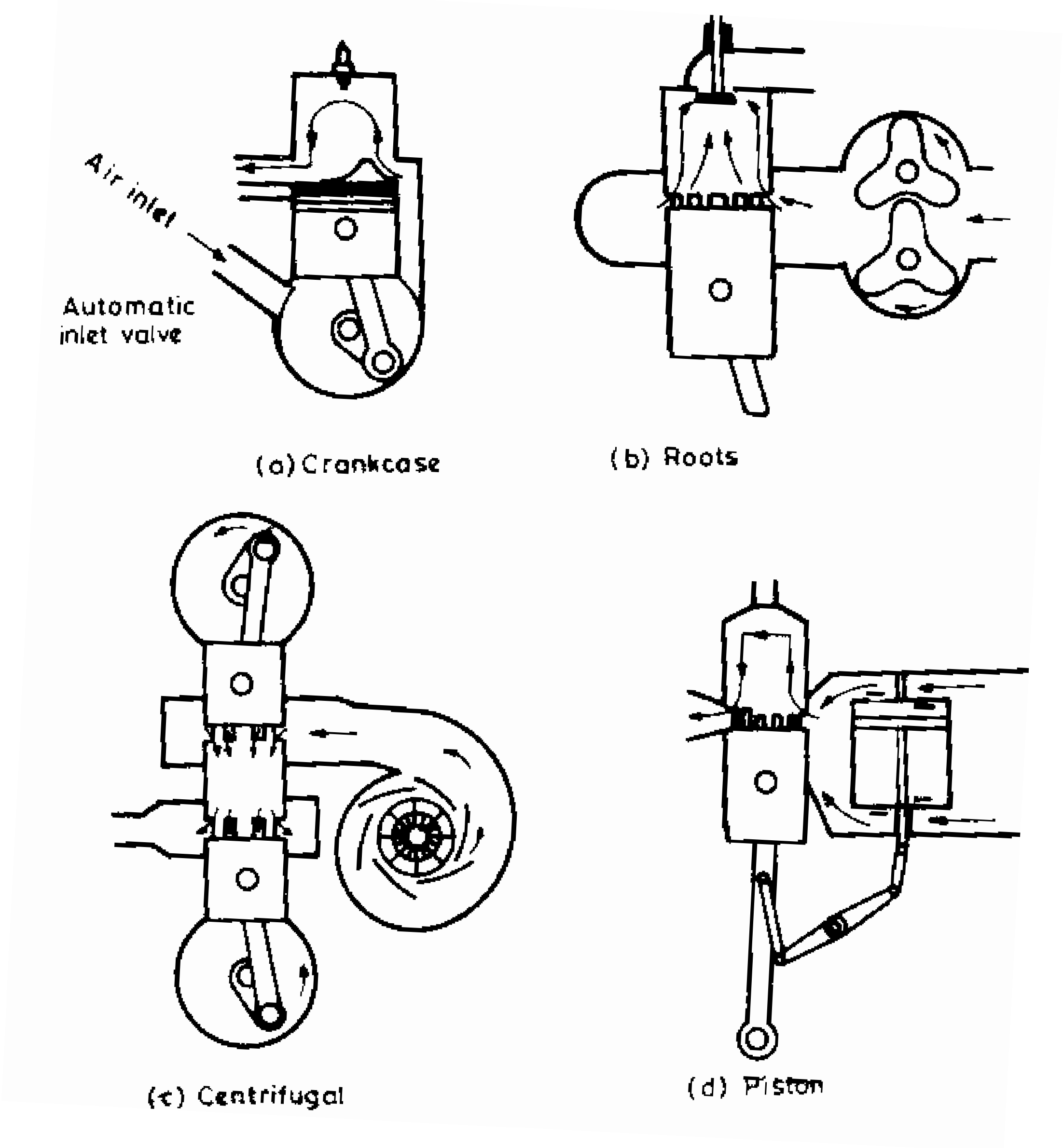


Fig. 2.21 Scavenging-pump types.

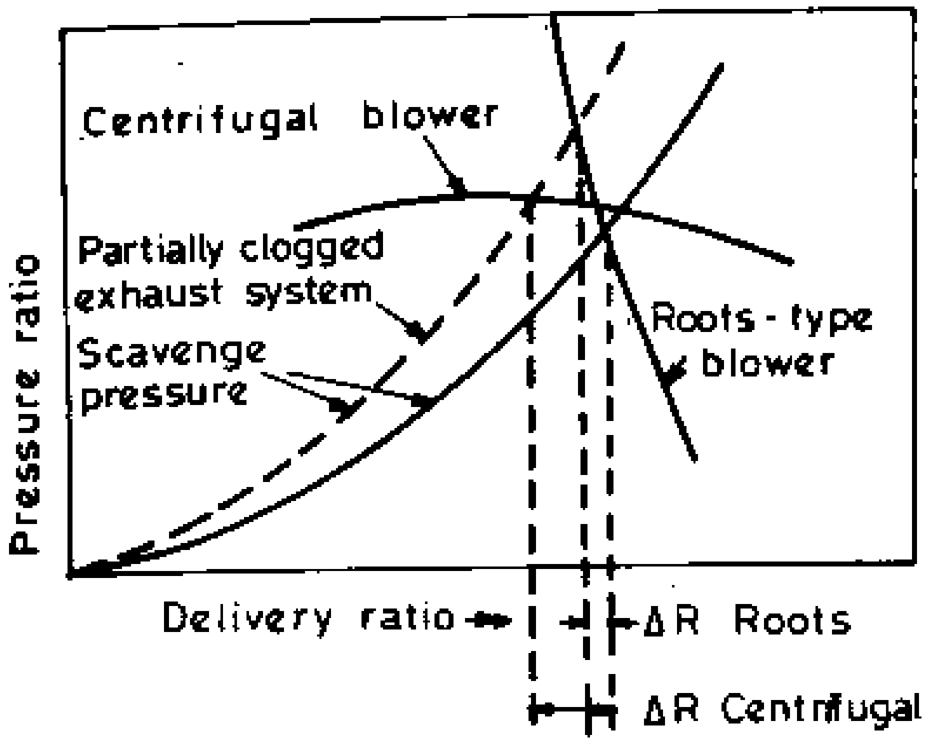


Fig. 2.22 Pressure characteristics of centrifugal and roots blower.

Comparison of two-stroke SI and CI engines

The two-stroke SI engine suffers from two big disadvantages-fuel loss and idling difficulty. The two-stroke CI engine does not suffer from these disadvantages and hence CI engine is more suitable for two-stroke operation.

If the fuel is supplied to the cylinders after the exhaust ports are closed, there will be no loss of fuel and the indicated thermal efficiency of the two-stroke engine will be as good as that of four-stroke engine. However, in an SI engine using carburettor, the scavenging is done with fuel-air mixture and only the fuel mixed with the retained air is used for combustion. To avoid the fuel loss instead of carburettor fuel injection just before the exhaust port closure may be used.

The two-stroke SI engine runs irregularly and may even stop at low speeds when mean effect pressure is reduced to about 2bar. This is because large amount of residual gas (more than in four-stroke engine) mixing with small amount of charge. At low speeds there may be back firing due to slow burning rate. Fuel injection improves idling and also eliminates backfiring as there is no fuel present in the inlet system.

In CI engines there is no loss of fuel as the charge is only air and there is no difficulty at idling because the fresh charge (air) is not reduced.

Advantages and disadvantages of two-stroke engines

Two-stroke engines have certain advantages as well as disadvantages compared to four-stroke engines. In the following sections the main advantages and disadvantages are discussed briefly.

Advantages of Two-stroke Engines

(i) As there is a working stroke for each revolution, the power developed will be nearly twice that of a four-stroke engine of the same dimensions and operating at the same speed.
(ii) The work required to overcome the friction of the exhaust and suction strokes is saved.
(iii) As there is a working stroke in every revolution, a more uniform turning moment is obtained on the crankshaft and therefore, a lighter flywheel is required.
(iv) Two-stroke engines are lighter than four-stroke engines for the same power output and speed.
(v) For the same output, two-stroke engines occupy lesser space.
(vi) The construction of a two-stroke cycle engine is simple because it has ports instead of valves. This reduces the maintenance problems considerably.
(vii) In case of two-stroke engines because of scavenging, burnt gases do not remain in the clearance space as in case of four-stroke engines.

Disadvantages of Two-Stroke Engines

(i) High speed two-stroke engines are less efficient owing to the reduced volumetric efficiency.
(ii) With engines working on Otto cycle, a part of the fresh mixture is lost as it escapes through the exhaust port during scavenging. This increases the fuel consumption and reduces the thermal efficiency.
(iii) Part of the piston stroke is lost with the provision of the ports thus the effective compression is less in case of two-stroke engines.
(iv) Two-stroke engines are liable to cause a heavier consumption of lubricating oil.
(v) With heavy loads, two-stroke engines get heated due to excessive heat produced. Also at light loads, the running of engine is not very smooth because of the increased dilution of charge.

SI and CI Engine application

We have seen that both SI and CI engines have certain advantages and disadvantages. The selection of a type of engine for particular application needs consideration of various factors.

The SI engine offers the following advantages:

(1) Low initial cost.
(2) Low weight for a given power output.
(3) Smaller size for a given power output.
(4) Easy starting.
(5) Less noise.
(6) Less objectionable exhaust gas odor and less smoke.

The SI engine finds wide application in automobiles because passenger comfort and in small airplanes because of low weight. Two stroke petrol engines finds extensive use in motor cycles, scooters, mopeds, pleasure motor boats, etc., because of simplicity and low cost. The SI engine is also used for light mobile duty like lawn movers, mobile generating sets, water pumps, air compressors, etc...

The CI engine offers the following advantages.

(1) Low specific fuel consumption at both full load and part load conditions.
(2) Utilizes less expensive fuels.
(3) Reduced fire hazard,
(4) Long operating life.
(5) Better suited for supercharging.
(6) Better suited for two-stroke cycle operating, as there is no loss of fuel in scavenging.

Because of fuel economy the CI engine finds wide usage in buses, trucks, locomotives, stationary generating plants, heavy duty equipment such as bulldozers, tractors and earthmoving machinery. Because of the reduced fire hazard the CI engine is also used for confined installations and marine use. The great advantage of the CI engine is lower fuel consumption which counteracts the disadvantage of higher initial cost, if the engine is used for long duties. (*Table 2.6a gives complete comparison of the two types of engines*.)

Comparison of two-stroke and four-stroke- engines (table 2.5)

The two-stroke engine was developed to obtain valve simplification and a greater output from the same size of engine. Two-stroke engines have no valves but only ports (some two-stroke engines are fitted with conventional exhaust valve). This simplicity of the two-stroke engine makes it cheaper to produce.

Theoretically a two-stroke engine will develop twice the power of a comparable four-stroke engine because of one power stroke every revolution (compared to one power stroke every two revolutions of four-stroke engine). This makes the two-stroke engine cheaper and more compact than a comparable four-stroke engine.

In actual practice power is not exactly doubled but is only about 30% extra because of (a) reduced effective stroke, and (b) due to increased heating caused by increased power strokes. The maximum speed is kept less than 4-stroke engine. The other advantages of the two-stroke engine are more uniform torque on crankshaft and complete exhaust of products of combustion.

However, when applied to spark-ignition engine the two-stroke cycle has certain disadvantages which have restricted its use to only small engines suitable for motor cycles, scooters, mopeds, lawn mowers, out-board engines, etc. In spark-ignition engine (petrol engine) the charge consists of a mixture of air and fuel. During scavenging, as both inlet and exhaust ports are open simultaneously for some time, some part of the fresh charge containing fuel escapes with exhaust. This results in high fuel consumption and hence lower thermal efficiency. The other drawback of two-stroke SI engine is the lack of flexibility- the capacity to run with equal efficiency at any speed. If the throttle is closed below the best point, the amount of fresh mixture entering the cylinder is not enough to clear out all the exhaust, some of which remains to contaminate the fresh charge. This results in irregular running of the engine.

The two-stroke diesel engine does not suffer from these defects. There is no loss of fuel with exhaust gases as the intake charge in diesel engine is air only. The two-stroke diesel engine is therefore used quite widely. Many of the biggest diesel engines work on this cycle. They are generally bigger than 60cm bore and are used in marine propulsion.

A disadvantage common to all two-stroke engines, petrol as well as diesel, is greater cooling and lubrication requirements due to one power stroke in each revolution of crankshaft. Consumption of lubricating oil is also high in the two-stroke engine due to higher temperatures.

Table 2.5 Comparison of four-stroke and two-stroke cycle engines

	Four-stroke cycle	*Two-stroke cycle*
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 is more (thoretically twice, actually about 1.3 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. Greater rate of wear and tear.
5.	The four-stroke engine contains valves and valve mechanism.	Two-stroke engines have no valves but only ports (some two-stroke engines are fitted with conventional exhaust valve or reed valve).
6.	Because of the heavy weight and complication of valve mechanism, higher in initial cost.	Because of light weight and simplicity due to the absence of valve mechanism, cheaper in initial cost.
7.	Volumetric efficiency more due to greater 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. In two-stroke petrol engines some fuel is exhausted during scavenging.
9.	Used where efficiency is important, in cars, buses, trucks, tractors, industrial engines, aeroplanes, power generation, etc.	Used were (a) low cost, and (b) compactness and light weight important. Two-stroke (air-cooled) petrol engines used in very small sizes only : lawn mowers, scooters, motor cycles, mopeds etc. (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.

Fundamental differences between SI and CI engines

Both SI and CI engines are internal combustion engines and have much in common. However, there are also certain fundamental differences that cause their operation to vary considerably. These are given in Table 2.6

Table 2.6 Comparison of SI and CI engines

	Description	*SI Engine*	*CI Engine*
1.	Basic cycle	Based on Otto cycle.	Based on Diesel cycle.
2.	Fuel	Petrol (Gasoline). High self-ignition temperature desirable.	Diesel oil. Low self-ignition temperature desirable.
3.	Introduction of fuel	Fuel and air introduced as a gaseous mixture in the suction stroke. Carburettor necessary to provide the mixture (except in not so common petrol injection engines). Throttle controls the quantity of mixture introduced.	Fuel is injected directly into combustion chamber at high pressure at the end of compression stroke. Carburettor is eliminated but a high pressure fuel pump and injector necessary. Quantity of fuel regulated in pump.
4.	Ignition	Requires an ignition system with spark plug in the combustion chamber.	Self ignition due to high temperature, caused by high compression of air, when fuel is injected. Ignition system and spark plug is eliminated.
5.	Compression ratio range	6 to 10.5. Upper limit of C.R. fixed by antiknock quality of fuel.	14 to 22. Upper limit of C.R. is limited by the rapidly increasing weight of the engine structure as the compression ratio is further increased.
6.	Speed	Higher maximum revolution per minute due to lighter weight.	Maximum r.p.m. lower.
7.	Efficiency	Maximum efficiency lower due to low compression ratio.	Higher maximum efficiency due to higher compression ratio.
8.	Weight	Lighter.	Heavier due to higher pressures.

table 2.6a detailed comparison of SI & CI engines

S.No.	Variable factor or characteristics	SI Engine	CI Engine	Remarks
1.	Thermodynamic cycle	Otto cycle	Diesel and Duel Combustion cycle	Otto cycle more efficient at a given compression ratio and heat input
2.	Combustion	Spark-ignition	Compression-ignition	
3.	Governing (speed & load control)	Quantity governing by throttling; almost constant A/F ratio	By rack, quality governing, air constant, fuel charge variable, no throttling, variable A/F ratio	Quality govrerning more efficient
4.	Compression ratio	6-11, restricted by detonation (average 7-9).	13-22 (average 15-18) Higher CR reduces knocking. Resrticted by mechanical and thermal stresses.	CI engine has higher thermal efficiency due to higher CR
5.	Operating pressures			CI engine heavily built
	(a) Compression pressure	7 to 15 bar	30 to 50 bar	
	(b) Max. pressure	45 to 50 bar	60 to 70 bar	
	(c) Operating speed	Max. 10 bar	Max 20 bar	
6.	(a) Operating speed	High speed (2000-6000 rpm)	Low sped (4000 rpm) medium speed (400-1200), high speed (1200-3500) up to 11 m/s	SI engine small in size for the same horse power
	(b) Piston speed	High 16 m/s		
7.	Distribution of fuel	A/F ratio is not optimum in multicylinder engines	Excellent distribution of fuel in multicylinder engines	Better efficiency and balance in the CI engine
8.	Supercharging	Limited by detonation, used only in aircraft engine	Inherently suitable, widely used. Limited by blower power and mechanical and thermal stresses.	
9.	Exhaust gas temperature	High, due to low thermal efficiency	Low, due to high thermal efficiency	SI exhaust valves subjected to intense heat
10.	Starting	Easy, low cranking effort	Difficult, high cranking effort	CI engine cold weather strating difficult
11.	Weight per unit power	Low (0.5 to 4.5 kg/kW)	High (3.3 to 13.5 kg/kW)	SI engine lower weight engine for the same power
12.	Power per unit displacement	High (30 kW/litre)	Low (15 kW/litre)	SI engine, hence used in small aircrafts, small volume
13.	Acceleration	Not so good, but compensated by acceleration pump	Good	
14.	Reliability	Good, normal troubles in carburettor and ignition system	Good, greater reserve of power, rated by smoke, not max. power, normal trouble in injection and generating systems	
15.	(a) Specific fuel consumption	Full load low, worse at part load and idling	Full load better, part load much better than SI as no throttling	Most important advantage of the CI engine and hence wide application
	(b) Fuel economy	Costly fuel, density low, calorific value slightly higher, less calories per litre	Cheaper fuel, density high, calorific value slightly low, more calories per litre	
	Full load	Medium	Good	
	Part load	Poor	Good	

S.No.	Variable factor or characteristics	SI Engine	CI Engine	Remarks
16.	Fuel safety (fire hazard)	Volatile fuel, more fire hazard	Less volatile, less fire hazard	CI engine safe in marine and other confined installations
17.	(a) Initial, capital cost	Low	High due to heavy weight and sturdy construction, costly construction, 1.25 - 1.5 times	
	(b) Running cost	High	Low	
18.	Operating life	Less	More due to sturdy construction. Rating lower than maximum power	
19.	Maintenance cost	Minor maintenance similar to CI	Major overall required less frequently	
20.	Noise and vibration	Less	More idle noise major problem	For comfort, passenger cars mostly use SI engines
21.	Odour and smoke	Less objectionable	More objectionable	
22.	Two-stroke operation	Less suitable, fuel loss in scavenging	More suitable, no fuel loss in scavenging	But small 2-stroke SI engines widely used in motor cycles, scooters, mopeds due to simplicity and cheapness.
23.	Applications	Passenger cars, small mobile applications aircrafts. Two stroke engines scooters, motor cycle mopeds due to cheaper and simpler engine.	Buses, trucks, locomotives, stationary generating plants. Heavy duty equipment like tractors, earth moving machinery; ships	Advantages and disadvantages in both types. For long, continuous running CI always preferred due to low running costs.
24.	Combustion problem	Kneck in unburnt mixture.	Diesel knock caused by long delay. High cetane number, low self-ignition temperature	
25.	Fuel	Petrol high octane number, high self-ignition temperature		
26.	Air-fuel ratio	10 to 17	18 to 100	
27.	Fuel supply method	Cheap; carburettor	Expensive; fuel pump and injector	
28.	Very high power	No	Yes	

References-

1. I C Engines By M L Mathur & RP Sharma
2. I C Engines By Ganesan