THERMAL ENGINEERING

II B.TECH II SEM MECH R15A0308

UNIT-I

Actual Cycles and their Analysis:

Introduction:

The actual cycles for IC engines differ from fuel air cycles and air standard cycles in many respects. The actual cycle efficiency is much lower than air standard efficiency due various losses occurring in the actual engine operation. The major losses are due to

1. Variation of specific heats with temperature.
2. Dissociation of the combustion products.
3. Progressive combustion.
4. Heat transfer into the walls of combustion chamber.
5. Incomplete combustion of fuel.
6. Blow down at the end of exhaust process.
7. Gas exchange process.

Comparison of Air standard and Actual cycles:

The actual cycles for internal combustion engines differ from air standard cycles in many respects. These differences are mainly due to

i) The working substance being a mixture of air fuel vapor or finely atomized liquid fuel in air combined with the products of combustion left from the previous cycle.

ii) The variation of specific heats with temperature.

iii) The change in chemical composition of the working substance.

iv) Progressive combustion rather than the instantaneous combustion.

v) Heat transfer to and from the working medium.

vi) Substantial exhaust blow down loss, i.e. loss of work on the expansion stroke due to early opening of exhaust valve.

vii) Gas leakage, fluid friction etc, in actual engines.

Out of all the above factors, major factor is influenced by

i) Time loss factor i.e. loss due to time required for mixing of fuel and air and also for combustion.

ii) Heat loss factor i.e. loss of heat from gases to cylinder walls.

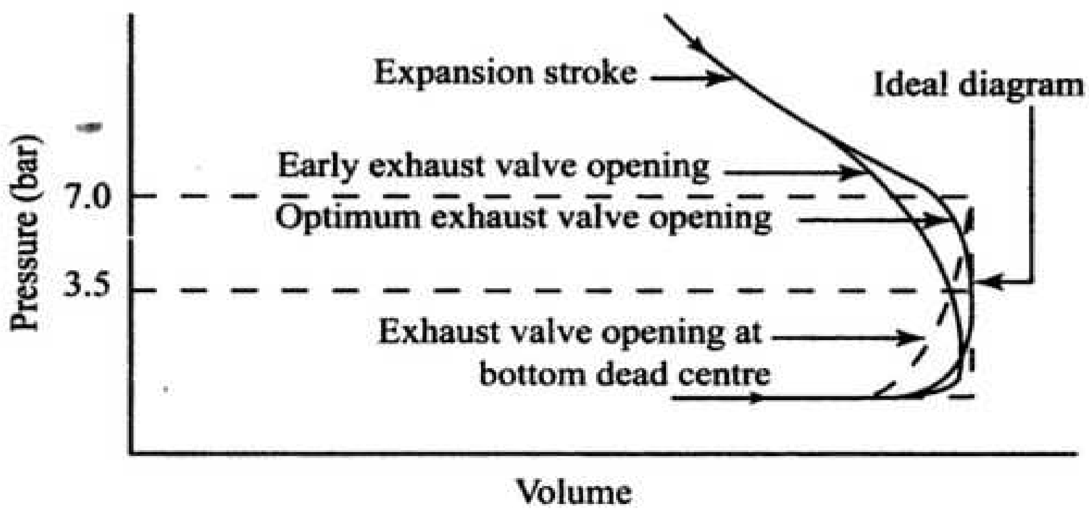
iii) Exhaust blow down factor i.e. loss of work on the expansion stroke due to early opening of exhaust valve.

Time Loss Factor:

In air standard cycles the heat addition is assumed to be an instantaneous process whereas in an actual cycle it is over a definite period of time. The time required for the combustion is such that under all circumstances some change in volume takes place while it is in progress. The crankshaft will usually turn about 30^0 to 40^0 between the initiation of spark and the end of combustion. There will be a time loss during this period and is called time loss factor. Therefore maximum pressure is not at TDC, but just after TDC. Time loss depends upon flame velocity which, in turn depends on type of fuel used, air fuel ratio, and shape of combustion chamber. This is to have P_{max} at TDC. If the spark is initiated at TDC, peak pressure would be low due to expansion of gases. Further if spark is initiated too early, additional work is required to compress the burning gases which are a direct loss. Therefore, the optimum time to start the combustion is 15^0 to 30^0 before TDC.

Exhaust Blow Down:

The cylinder pressure at the end of exhaust stroke is about 7 bar depending upon the compression ratio employed. If the exhaust valve is opened at the bottom dead centre (BDC), the piston has to do work against high pressure during the early part of exhaust stroke. If the exhaust valve is opened too early, a part of the expansion stroke is lost. The best compromise is to open the exhaust valve 40^0 to 70^0 before BDC thereby reducing the cylinder pressure to halfway (say 3.5 bar) before the exhaust stroke begins, which is shown in the figure.



Effect of Exhaust Valve Opening Time on Blowdown

Loss Due to Gas Exchange Processes

The difference of work done in expelling the exhaust gases and the work done by the fresh charge during the suction stroke is called the pumping work. In other words loss due to the gas exchange process (pumping loss) is due to pumping gas from lower inlet pressure p_i to higher exhaust pressure p_e. *The pumping loss increases at part throttle because throttling reduces the suction pressure.* Pumping loss also increases with speed. The gas exchange processes affect the volumetric efficiency of the engine. The performance of the engine, to a great deal, depends on the volumetric efficiency. Hence, it is worthwhile to discuss this parameter in greater detail here.

Volumetric Efficiency

As already stated in section 1.8.4, volumetric efficiency is an indication of the breathing ability of the engine and is defined as the ratio of the volume of air actually inducted at ambient condition to swept volume. However, it may also be defined on mass basis as the ratio of the actual mass of air drawn into the engine during a given period of time to the theoretical mass which should have been drawn in during that same period of time, based upon the total piston displacement of the engine, and the temperature and pressure of the surrounding atmosphere.

The above definition is applicable only to the naturally aspirated engine. In the case of the supercharged engine, however, the theoretical mass of air should be calculated at the conditions of pressure and temperature prevailing in the intake manifold. The volumetric efficiency is affected by many variables, some of the important ones are:

(i) *The density of the fresh charge :* As the fresh charge arrives in the hot cylinder, heat is transferred to it from the hot chamber walls and the hot residual exhaust gases, raising its temperature. This results in a decrease in the mass of fresh charge admitted and a reduction in volumetric efficiency. The volumetric efficiency is increased by low temperatures (provided there are no heat transfer effects) and high pressure of the fresh charge, since density is thereby increased, and more mass of charge can be inducted into a given volume.

(ii) *The exhaust gas in the clearance volume :* As the piston moves from TDC to BDC on the intake stroke, these products tend to expand and occupy a portion of the piston displacement greater than the clearance volume, thus reducing the space available to the incoming charge. In addition, these exhaust products tend to raise the temperature of the fresh charge, thereby decreasing its density and further reducing volumetric efficiency.

(iii) *The design of the intake and exhaust manifolds :* The exhaust manifold should be so designed as to enable the exhaust products to escape readily, while the intake manifold should be designed so as to bring in the maximum possible fresh charge. This implies minimum restriction is offered to the fresh charge flowing into the cylinder, as well as to the exhaust products being forced out.

(iv) *The timing of the intake and exhaust valves :* Valve timing is the regulation of the points in the cycle at which the valves are set to open and close. Since, the valves require a finite period of time to open or close for smooth operation, a slight "lead" time is necessary for proper opening and closing. The design of the valve operating cam provides for the smooth transition from one position to the other, while the cam setting determines the timing of the valve.

LOSS DUE TO RUBBING FRICTION

These losses are due to friction between the piston and the cylinder walls, friction in various bearings and also the energy spent in operating the auxiliary equipment such as cooling water pump, ignition system, fan, etc. The piston ring friction increases rapidly with engine speed. It also increases to a small extent with increase in mean effective pressure. The bearing friction and the auxiliary friction also increase with engine speed. The efficiency of an engine is maximum at full load and decreases at part loads. It is because the percentage of direct heat loss, pumping loss and rubbing friction loss increase at part loads. The approximate losses for a gasoline engine of high compression ratio, say 8:1 using a chemically correct mixture

ACTUAL AND FUEL-AIR CYCLES OF CI ENGINES

In the diesel cycle the losses are less than in the Otto cycle. The main loss is due to incomplete combustion and is the cause of main difference between fuel-air cycle and actual cycle of a diesel engine. This is shown in Fig.5.11. In a fuel-air cycle the combustion is supposed to be completed at the end of the constant pressure burning whereas in actual practice *after burning* continues up to half of the expansion stroke. The ratio between the actual efficiency and the fuel-air cycle efficiency is about 0.85 in the diesel engines.

In fuel-air cycles, when allowance is made for the presence of fuel and combustion products, there is reduction in cycle efficiency. In actual cycles, allowances are also made for the losses due to phenomena such as

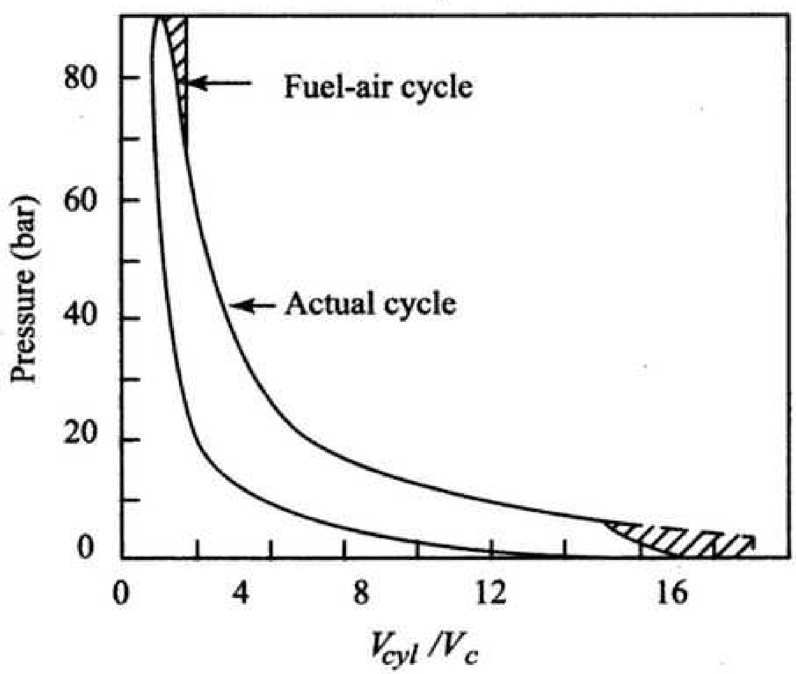


Fig. 5.11 Actual Diesel Cycle Vs Equivalent Fuel Combustion Limited Pressure Cycle for Two-Stroke Diesel Engine

heat transfer and finite combustion time. This reduces the cycle efficiency further. For complete analysis of actual cycles, computer models are being developed nowadays. These models are helpful in understanding the various processes that are taking place in an engine. Models are developed for both not only for CI engines but also for SI engines.

Internal Combustion Engines

An internal combustion engine (ICE) is a heat engine where the combustion of a fuel occurs with an oxidizer (usually air) in a combustion chamber that is an integral part of the working fluid flow circuit. In an internal combustion engine, the expansion of the high-temperature and high-pressure gases produced by combustion applies direct force to some component of the engine. The force is applied typically to pistons, turbine blades, rotor or a nozzle. This force moves the component over a distance, transforming chemical energy into useful mechanical energy.

Types of Internal Combustion Engines:

Internal combustion engines can be classified into a large number of types based on several criteria. The classification of IC engines is given below:

1. Based on the fuel used
    1. Diesel Engine
    2. Petrol Engine (or Gasoline Engine)
2. Based on the type of cycle
    1. Otto Cycle Engine
    2. Diesel Cycle Engine
    3. Dual Cycle Engine
3. Based on the number of strokes per cycle
    1. Two-stroke Engine
    2. Four-stroke Engine
4. Based on the number of cylinders
    1. Single Cylinder Engine
    2. Multi cylinder Engine
        - 2.i. Twin Cylinder Engine
        - 2.ii. Three Cylinder Engine
        - 2.iii. Four Cylinder Engine
        - 2.iv. Six Cylinder Engine
        - 2.v. Eight Cylinder Engine
        - 2.vi. Twelve Cylinder Engine
        - 2.vii. Sixteen Cylinder Engine
5. Based on the type of ignition
    1. Spark Ignition Engine (S.I. Engine)
    2. Compression Ignition Engine (C.I. Engine)
6. Based on the lubrication system used
    1. Dry sump lubricated engine
    2. Wet sump lubricated Engine
7. Based on the cooling system used
    1. Air-cooled Engine
    2. Water-cooled Engine
8. Based on the arrangement of valves
    1. L-head Engine
    2. I-head Engine
    3. T-head Engine
    4. F-head Engine

9. Based on the position of cylinders
    1. Horizontal Engine
    2. Vertical Engine
    3. Radial Engine
    4. Opposed Piston Engine
    5. Opposed Cylinder Engine
    6. V Engine
    7. W Engine
    8. Inline Engine
10. Based on the pressure boost given to the inlet air or air-fuel mixture
    1. Naturally aspired Engine
    2. Supercharged Engine
    3. Turbocharged Engine
    4. Crankcase compressed Engine
11. Based on application
    1. Automobile Engine
    2. Aircraft Engine
    3. Locomotive Engine
    4. Marine Engine
    5. Stationary Engine

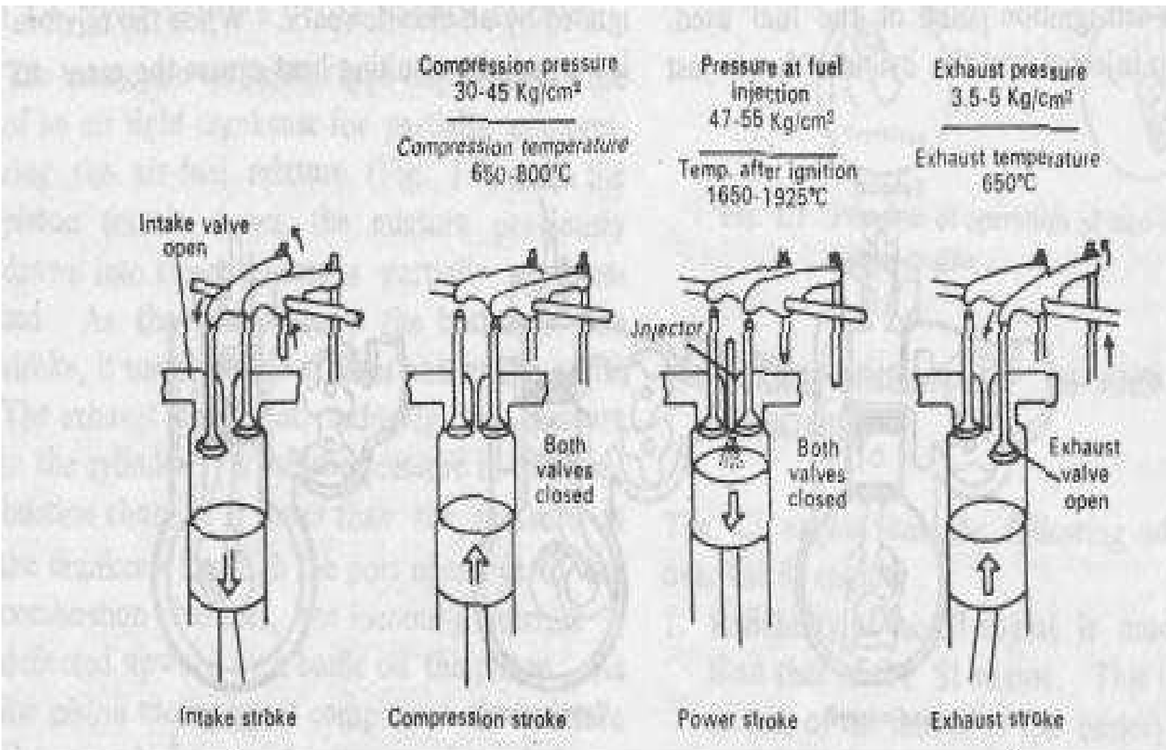
WORKING PRINCIPLE OF FOUR-STROKE CI ENGINE

In four-stroke cycle engines there are four strokes completing two revolutions of the crankshaft. These are respectively, the suction, compression, power and exhaust strokes. In Fig. 3, the piston is shown descending on its suction stroke. Only pure air is drawn into the cylinder during this stroke through the inlet valve, whereas, the exhaust valve is closed. These valves can be operated by the cam, push rod and rocker arm. The next stroke is the compression stroke in which the piston moves up with both the valves remaining closed. The air, which has been drawn into the cylinder during the suction stroke, is progressively com-pressed as the piston ascends. The compression ratio usually varies from 14:1 to 22:1. The pressure at the end of the compression stroke ranges from 30 to 45 kg/cm^2. As the air is progressively compressed in the cylinder, its temperature increases, until when near the end of the compression stroke, it becomes sufficiently high (650-80O ^oC) to instantly ignite any fuel that is injected into the cylinder. When the piston is near the top of its compression stroke, a liquid hydrocarbon fuel, such as diesel oil, is sprayed into the combustion chamber **under high pressure (140-160 kg/cm^2)**, higher than that existing in the cylinder itself. This fuel then ignites, being burnt with the oxygen of the highly compressed air.

During the fuel injection period, the piston reaches the end of its compression stroke and commences to return on its third consecutive stroke, viz., **power stroke**. During this stroke the hot products of combustion consisting chiefly of carbon dioxide, together with the nitrogen left from the compressed air expand, thus forcing the piston downward. This is only the working stroke of the cylinder.

During the power stroke the pressure falls from its maximum combustion value **(47-55 kg/cm^2),** which is usually higher than the greater value of the compression pressure (45 kg/cm^2), to about **3.5-5 kg/cm^2** near the end of the stroke. The exhaust valve then opens, usually a little earlier than when the piston reaches its lowest point of travel. The exhaust gases are swept out on the following upward stroke of the piston. The exhaust valve remains open throughout the whole stroke and closes at the top of the stroke.

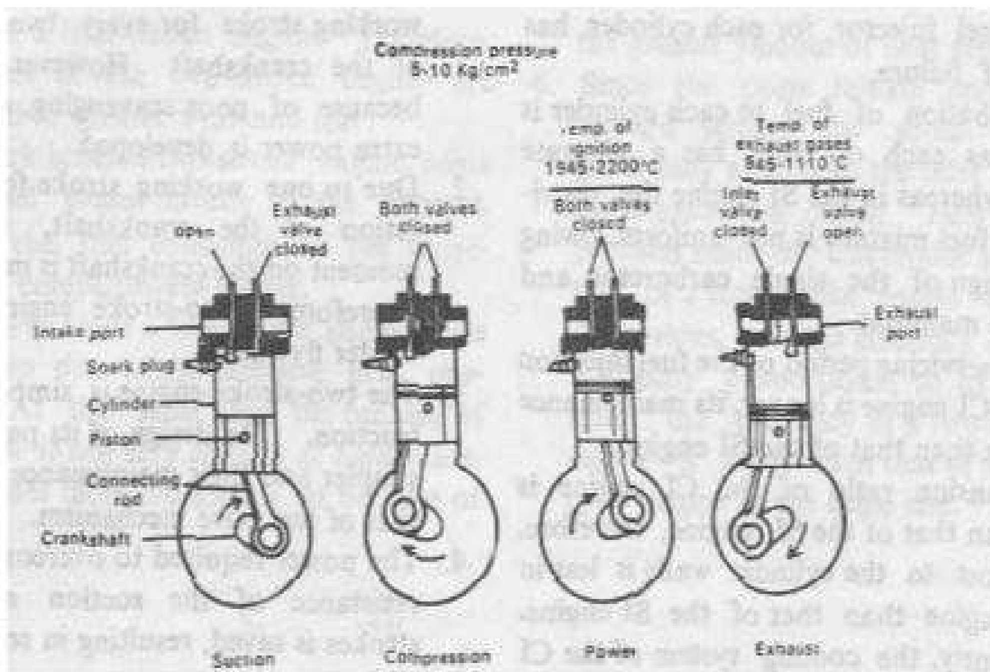
The reciprocating motion of the piston is converted into the rotary motion of the crankshaft by means of a connecting rod and crankshaft. The crankshaft rotates in the main bearings, which are set in the crankcase. The flywheel is fitted on the crankshaft in order to smoothen out the uneven torque that is generated in the reciprocating engine.



Principle of four-stroke CI engine

WORKING PRINCIPLE OF FOUR-STROKE SPARK IGNITION ENGINE

In this gasoline is mixed with air, broken up into a mist and partially vaporized in a carburetor (Fig. 5). The mixture is then sucked into the cylinder. There it is compressed by the upward movement of the piston and is ignited by an electric spark. When the mixture is burned, the resulting heat causes the gases to expand. The expanding gases exert a pressure on the piston (power stroke). The exhaust gases escape in the next upward movement of the piston. The strokes are similar to those discussed under four-stroke diesel engines. The various temperatures and pressures are shown in Fig. 6. The compression ratio varies from 4:1 to 8:1 and the air-fuel mixture from 10:1 to 20:1.

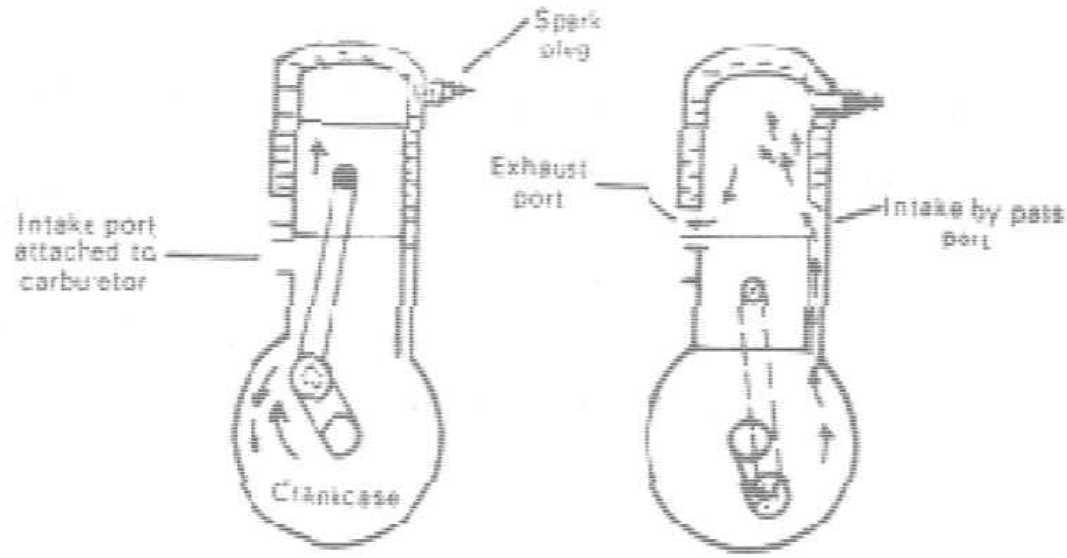


Four-stroke petrol engine

WORKING PRINCIPLE OF TWO-STROKE CYCLE PETROL ENGINE

The two-cycle carburettor type engine makes use of an airtight crankcase for partially compressing the air-fuel mixture (Fig. 6). As the piston travels down, the mixture previously drawn into the crankcase is partially compressed. As the piston nears the bottom of the stroke, it uncovers the exhaust and intake ports. The exhaust flows out, reducing the pressure in the cylinder. When the pressure in the combustion chamber is lower than the pressure in the crankcase through the port openings to the combustion chamber, the incoming mixture is deflected upward by a baffle on the piston. As the piston moves up, it compresses the mixture above and draws into the crankcase below a new air-fuel mixture.

The, two-stroke cycle engine can be easily identified by the air-fuel mixture valve attached to the crankcase and the exhaust Port located at the bottom of the cylinder.



Two stroke petrol engine

COMPARISON OF CI AND SI ENGINES

The CI engine has the following advantages over the SI engine.

1. Reliability of the CI engine is much higher than that of the SI engine. This is because in case of the failure of the battery, ignition or carburettor system, the SI engine cannot operate, whereas the CI engine, with a separate fuel injector for each cylinder, has less risk of failure.

2. The distribution of fuel to each cylinder is uniform as each of them has a separate injector, whereas in the SI engine the distribution of fuel mixture is not uniform, owing to the design of the single carburettor and the intake manifold.

3. Since the servicing period of the fuel injection system of CI engine is longer, its maintenance cost is less than that of the SI engine.

4. The expansion ratio of the CI engine is higher than that of the SI engine; therefore, the heat

loss to the cylinder walls is less in the CI engine than that of the SI engine. Consequently, the cooling system of the CI engine can be of smaller dimensions.

5. The torque characteristics of the CI engine are more uniform which results in better top gear performance.

6. The CI engine can be switched over from part load to full load soon after starting from cold, whereas the SI engine requires warming up.

7. The fuel (diesel) for the CI engine is cheaper than the fuel (petrol) for SI engine.

8. The fire risk in the CI engine is minimised due to the absence of the ignition system.

9. On part load, the specific fuel consumption of the CI engine is low.

ADVANTAGES AND DISADVANTAGES OF TWO-STROKE CYCLE OVER FOUR-STROKE CYCLE ENGINES

Advantages:

1) The two-stroke cycle engine gives one working stroke for each revolution of the crankshaft. Hence theoretically the power developed for the same engine speed and cylinder volume is twice that of the four-stroke cycle engine, which gives only one working stroke for every two revolutions of the crankshaft. However, in practice, because of poor scavenging, only 50-60% extra power is developed.
2) Due to one working stroke for each revolution of the crankshaft, the turning moment on the crankshaft is more uniform. Therefore, a two-stroke engine requires a lighter flywheel.
3) The two-stroke engine is simpler in construction. The design of its ports is much simpler and their maintenance easier than that of the valve mechanism.
4) The power required to overcome frictional resistance of the suction and exhaust strokes is saved, resulting in some economy of fuel.
5) Owing to the absence of the cam, camshaft, rockers, etc. of the valve mechanism, the mechanical efficiency is higher.
6) The two-stroke engine gives fewer oscillations.
7) For the same power, a two-stroke engine is more compact and requires less space than a four-stroke cycle engine. This makes it more suitable for use in small machines and motorcycles.
8) A two-stroke engine is lighter in weight for the same power and speed especially when the crankcase compression is used.
9) Due to its simpler design, it requires fewer spare parts.
10) A two-stroke cycle engine can be easily reversed if it is of the valve less type.

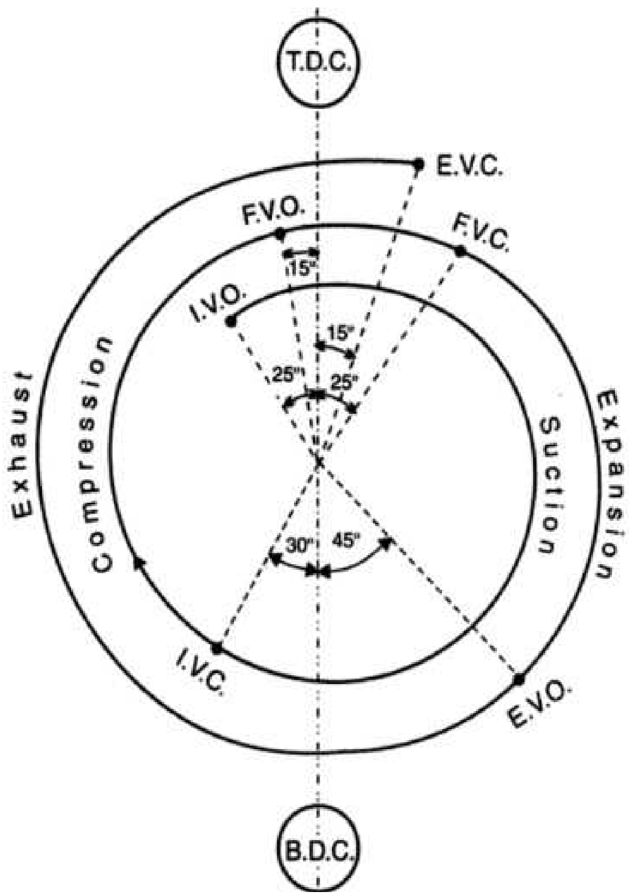
Disadvantages:

1. The scavenging being not very efficient in a two-stroke engine, the dilution of the charges takes place which results in poor thermal efficiency.
2. The two-stroke spark ignition engines do not have a separate lubrication system and normally, lubricating oil is mixed with the fuel. This is not as effective as the lubrication of a four-stroke engine. Therefore, the parts of the two-stroke engine are subjected to greater wear and tear.

3. In a spark ignition two-stroke engine, some of the fuel passes directly to the exhaust. Hence, the fuel consumption per horsepower is comparatively higher.
4. With heavy loads a two-stroke engine gets heated up due to the excessive heat produced. At the same time the running of the engine is riot very smooth at light loads.
5. It consumes more lubricating oil because of the greater amount of heat generated.

VALVE TIMING DIAGRAM OF 4 STROKE ENGINE

Diesel engines. Fig. shows the actual valve timing diagram of a *four stroke "Diesel cycle" engine* (theoretical valve timing diagram, is however the same as Fig. Inlet valve opens 10° to 25° in advance of T.D.C. position and closes 25° to 50° after the B.D.C. position. Exhaust valve opens 30° to 50° in advance of B.D.C. position and closes 10° to 15° after the T.D.C. position. The fuel injection takes place 5° to 10° before T.D.C. position and continues up to 15° to 25° near T.D.C. position.



PORT TIMING DIAGRAM OF TWO STROKE ENGINE

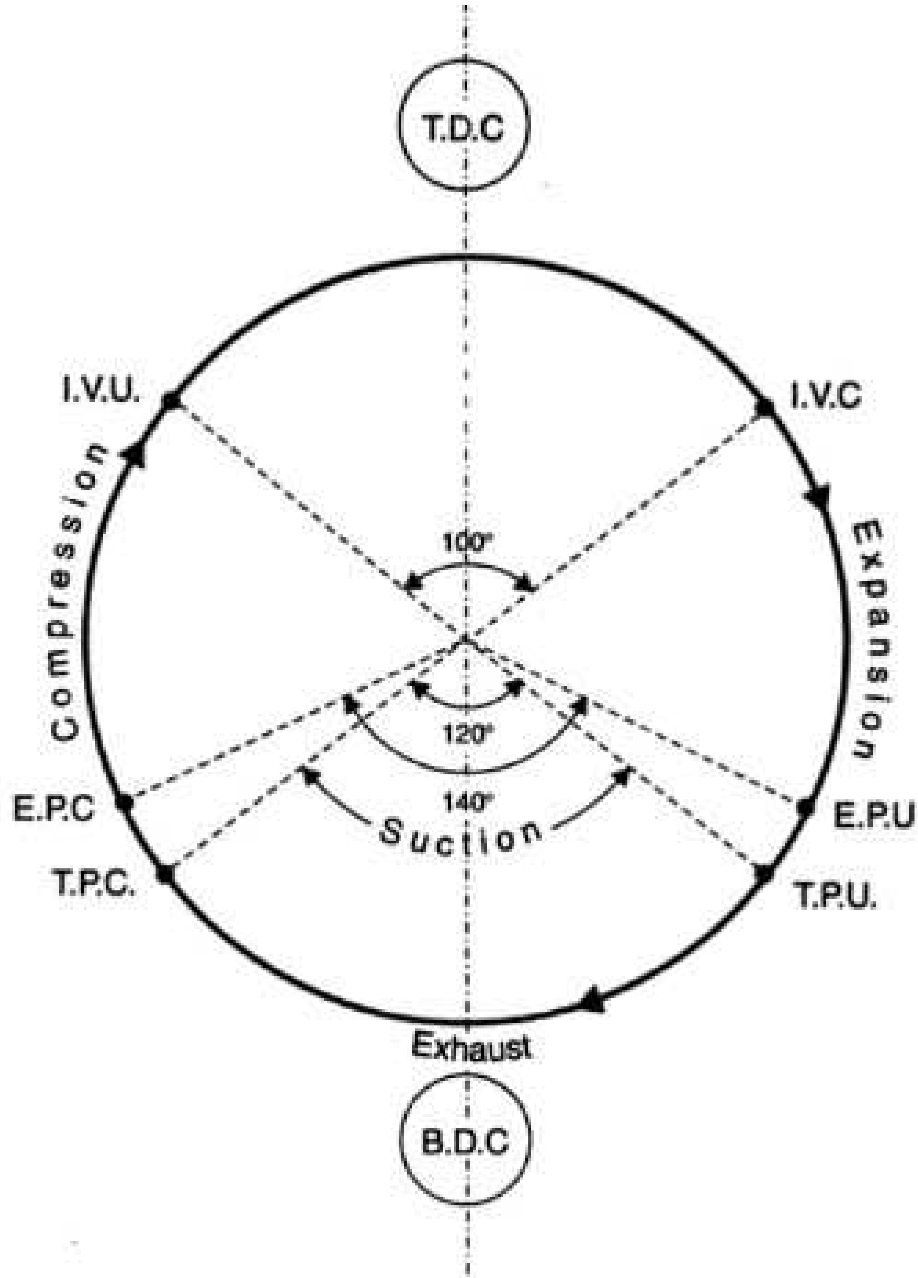


Fig. 2.45. Port timing diagram.

COOLING SYSTEM

There are mainly two types of cooling systems:

(a) Air cooled system, and

(b) b) Water cooled system.

Air Cooled System

Air cooled system is generally used in small engines say up to 15-20 kW and in aero plane engines.

In this system fins or extended surfaces are provided on the cylinder walls, cylinder head, etc. Heat generated due to combustion in the engine cylinder will be conducted to the fins and when the air flows over the fins, heat will be dissipated to air.

The amount of heat dissipated to air depends upon:

(b.a) Amount of air flowing through the fins.

(b.b) Fin surface area.

(b.c) Thermal conductivity of metal used for fins.

Cylinder with Fins

Advantages of Air Cooled System

Following are the advantages of air cooled system:

a.a.1. Radiator/pump is absent hence the system is light.

a.a.2. In case of water cooling system there are leakages, but in this case there are no leakages.

a.a.3. Coolant and antifreeze solutions are not required.

a.a.4. This system can be used in cold climates, where if water is used it may freeze.

Disadvantages of Air Cooled System

a) Comparatively it is less efficient.

b) It is used in aero planes and motorcycle engines where the engines are exposed to air directly.

WATER COOLING SYSTEM

In this method, cooling water jackets are provided around the cylinder, cylinder head, valve seats etc. The water when circulated through the jackets, it absorbs heat of combustion. This hot water will then be cooling in the radiator partially by a fan and partially by the flow developed by the forward motion of the vehicle. The cooled water is again re-circulated through the water jackets.

Types of Water Cooling System

There are two types of water cooling system:

a) **Thermo Siphon System**

In this system the circulation of water is due to difference in temperature (i.e. difference in densities) of water. So in this system pump is not required but water is circulated because of density difference only.

Thermo Siphon System of Cooling

Pump Circulation System

In this system circulation of water is obtained by a pump. This pump is driven by means of engine output shaft through V-belts.

Pump Circulation System

Components of Water Cooling System:

Water Cooling System using Thermostat Valve

Water Cooling System of a 4-cylinder Engine

Water cooling system mainly consists of:

(a) Radiator,

(b) Thermostat valve,

(c) Water pump,

(d) Fan,

(e) Water Jackets, and

(f) Antifreeze mixtures.

Radiator

It mainly consists of an upper tank and lower tank and between them is a core. The upper tank is connected to the water outlets from the engines jackets by a hose pipe and the lover tank is connect to the jacket inlet through water pump by means of hose pipes.

There are 2-types of cores:

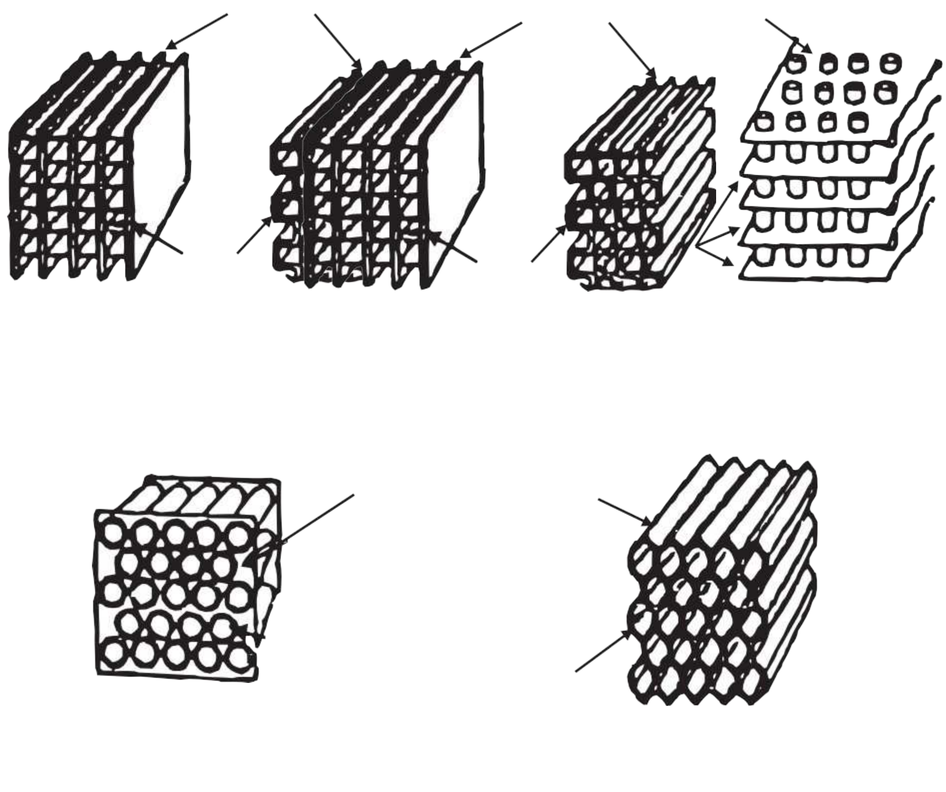
(f.a) Tubular

(f.b) Cellular as shown.

When the water is flowing down through the radiator core, it is cooled partially by the fan which blows air and partially by the air flow developed by the forward motion of the vehicle.

As shown through water passages and air passages, wafer and air will be flowing for cooling purpose.

It is to be noted that radiators are generally made out of copper and brass and their joints are made by soldering.



Types of Cores (a) Tabullar Radiator Sections and (b) Circullar Radiator Sections

Modern ignition systems

The development of high speed, high compression internal combustion engine requires a reliable high-speed ignition system. This is met by a high-tension ignition system that uses a spark plug as the source of ignition. The electrical energy to the spark plug is supplied by one of the following systems and is termed accordingly.

1. Battery ignition system
2. Magneto ignition system

Main Parts of battery ignition system:

Battery

A battery is used to provide energy for ignition. It is work as storage of energy and charged by dynamo, which is driven by engine. It converts chemical energy to electric energy. Two types of battery used in spark ignition system, lead acid battery and alkaline battery. The first one is used in light duty commercial vehicle and the other one is used in heavy duty commercial vehicle. It is housed in primary side of ignition coil.

Ignition switch:

It is used to turn on and off the ignition system. Battery is connected to the primary winding of ignition coil by ignition switch and ballast resistor.

Ballast resistor:

It is connected in series with primary winding to regulate current in primary winding. It is used to prevent injury due to overheating of ignition coil. It controls the current passes through primary winding. It is made by iron. Iron has property of increase electrical resistance rapidly by increase in temperature at certain limit. This additional resistance resists flowing current which control the temperature of ignition coil.

Ignition coil:

Ignition coil is the main body of battery ignition system. The purpose of ignition coil is to step up the battery voltage (6 or 12) to a high voltage, which is sufficient to produce spark at spark plug. It consist a magnetic core or soft wire or sheet, and two electrical winding called primary winding and secondary winding. The primary winding has generally 200-300 turn and the end are connected to exterior terminal. The secondary has almost 21000 turns of copper wire which is insulated to withstand with high voltage. It is located inside the primary winding and its one end connected to secondary winding and other end is grounded either to primary winding or to the metal case. This entire unit is enclosed in a metal container which makes it a compact unit.

Contact breaker:

This is a mechanical device making and braking the primary circuit to ignition coil. When the points are closed current flow in ignition coil and when it open, flow of current stopped.

Capacitor:

It is simple electrical capacitors in which two metal plate are separated by an insulating material with a distance. Commonly air is used as insulating material but for particular technical requirement some high quality insulating material is used.

Distributor:

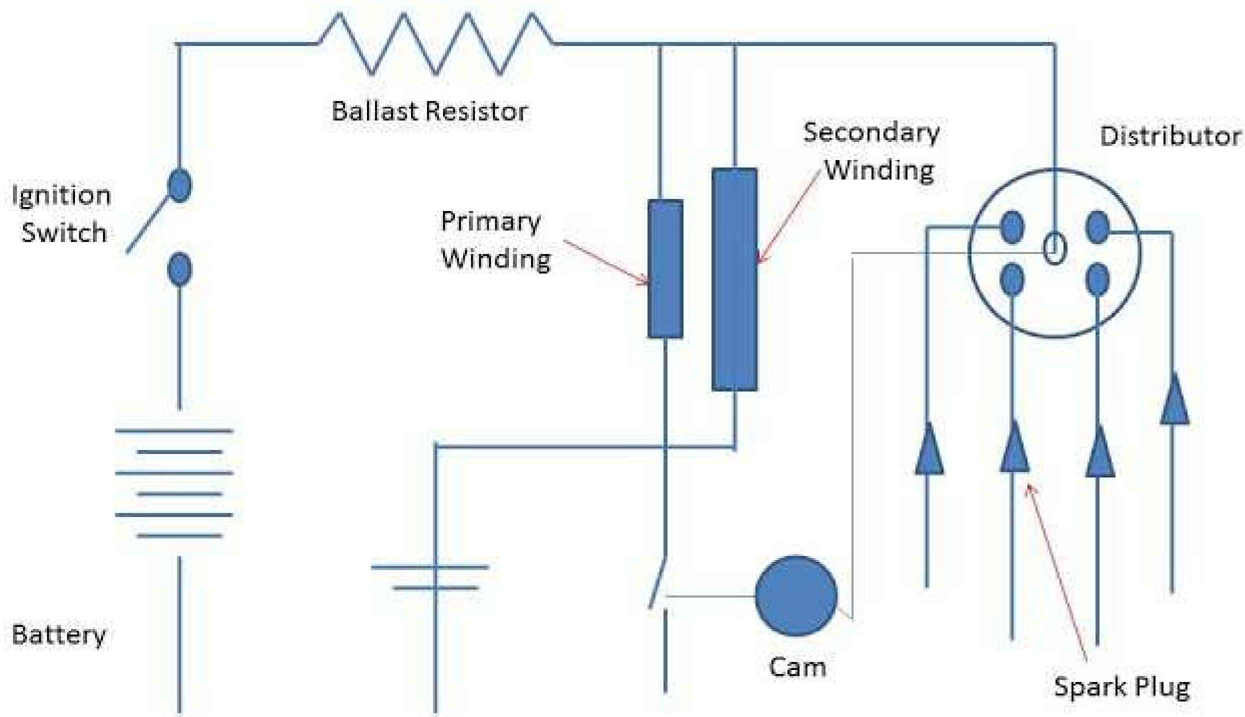
Distributor is used in multi cylinder engine to regulate spark in each spark plug at correct sequence. It distribute ignition surge in individual spark plug in correct sequence. There are two types of distributor. One is known as carbon brush type and the other one is gap type. In carbon brush type carbon brush carried by the rotor arm sliding over the metallic segment embedded into the distributor cap or molded insulating material. This makes electric connection or secondary winding with spark plug. In gap type distributor electrode of rotor arm pass close to but does not make contact with the distributor cap. So there is no wear of electrode.

Spark Plug:

A spark plug generally has two electrodes which are separated with each other. A high potential discharge flows through it which generate spark and ignite the combustion mixture in cylinder. It mainly consist two electrodes a steel shell and an insulator. The central electrode connected with the supply of ignition coil. It is well insulated with the outer steel shell which is grounded. There is a small air gap between steel shell and central electrode, between which spark is generated. The electrode usually made by high nickel alloy so it can withstand with high temperature and corrosion resistance.

Working of Battery Ignition System:

In the battery ignition system ignition coil stores the energy in form of magnetic field and deliver it at the instant of ignition, in form of high voltage current with high tension wire to correct spark plug. The diagram of four cylinder battery ignition system is as follow.



Battery Ignition System for Four Cylinder SI Engine

First low voltage current flow form battery to the primary coil through ignition switch and ballast resistor.

Ballast resistor regulates the temperature of ignition coil by regulating current passing form it.

The ignition capacitor connected in parallel with contact breaker. One end of secondary winding is also grounded through contact breaker.

When the ignition switch is closed, the primary winding of the coil is connected to the positive terminal, and current flow through it known as primary current.

The current flows form primary coil produces a magnetic field which induces an EMF in secondary coil.

The cam regulates the contact breaker. Wherever the breaker open, current flows into condenser, which charged the condenser.

As the condenser become charger the primary current falls and the magnetic field collapses. This will induces a much higher voltage in condenser.

Now the condenser discharge into the battery which reverses the direction of both primary current and magnetic field. This will induce a very high EMF in secondary winding.

Now this high voltage EMF produce spark at correct spark plug through distributor.

Advantages and Disadvantages:

Advantages:

1. At the time of starting or at low speed good spark is available.
2. The battery which is used to generate spark can be used to light other auxiliary like headlight, tell light etc.
3. Initial expenditure is less and it has low maintenance cost.
4. Ignition system is not affected by adjusting spark timing in battery ignition system.

Disadvantages:

1. Time available of built up the current and stored energy is decrease as speed of engine increases.
2. Contact breaker subjected to both electrical and mechanical wear which results short maintenance interval.
3. The primary voltage decreases as the engine speed increase. So it is not fully reliable of high speed engine.

Magneto Ignition System:

Parts of Magneto Ignition System:

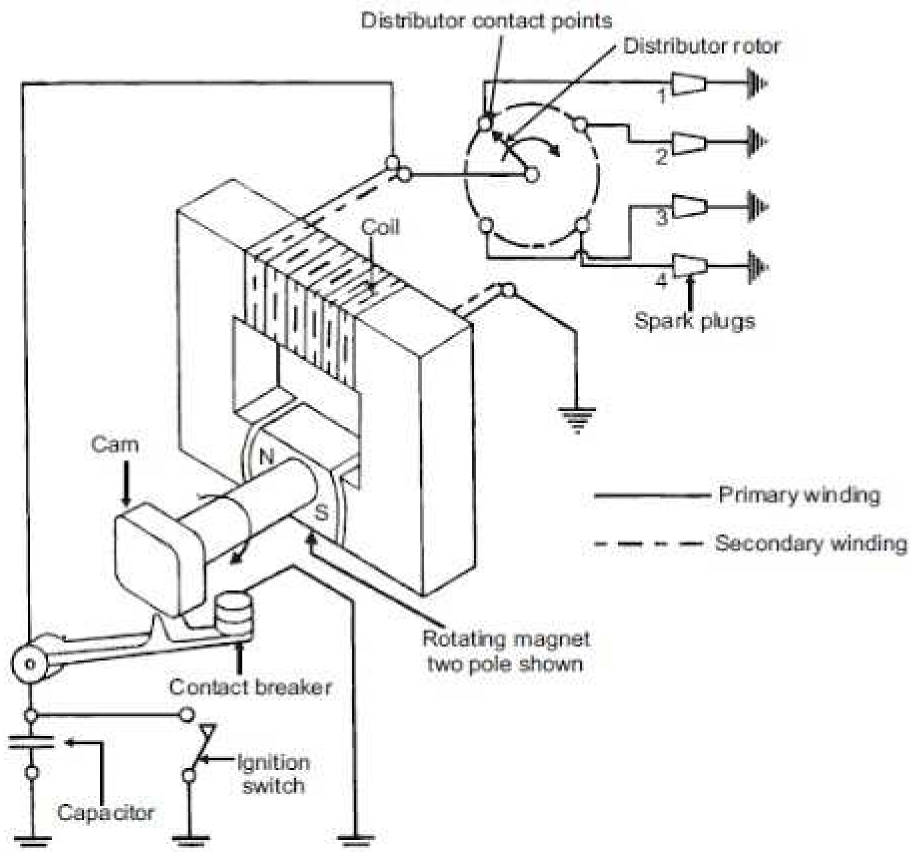
Magneto: It is the major part of this type of ignition system because it is source of energy. A magneto is a small electric generator which is rotate by the engine and is capable of produce a very high voltage and does not need battery as a source of external energy. The magneto contains both primary and secondary winding thus it does not require a separate coil to boost up the voltage required to operate the spark plug. There are two types of magneto. First one is known as armature rotating type and other one is known as magnet rotating type. In the first type, the armature rotates between the stationary magnets. On the other hand in second type armature is stationary and the magnates are rotating around armature.

Distributor: Distributor is used in multi cylinder engine to regulate spark in each spark plug at correct sequence. It distribute ignition surge in individual spark plug in correct sequence. There are two types of distributor. One is known as carbon brush type and the other one is gap type. In carbon brush type carbon brush carried by the rotor arm sliding over the metallic segment embedded into the distributor cap or molded insulating material. This makes electric connection or secondary winding with spark plug. In gap type distributor electrode of rotor arm pass close to but does not make contact with the distributor cap. So there is no wear of electrode.

Spark Plug: A spark plug generally has two electrodes which are separated with each other. A high potential discharge flow through it which generate spark and ignite the combustion mixture in cylinder. It mainly consist two electrodes a steel shell and an insulator. The central electrode connected with the supply of ignition coil. It is well insulated with the outer steel shell which is grounded. There is a small air gap between steel shell and central electrode, between which spark is generated. The electrode usually made by high nickel alloy so it can withstand with high temperature and corrosion resistance.

Capacitor: It is simple electrical capacitors in which two metal plate are separated by an insulating material with a distance. Commonly air is used as insulating material but for particular technical requirement some high quality insulating material is used.

Working of Magneto Ignition System: The working principle of magneto ignition system is same as battery ignition system except in the magneto ignition system Magneto is used to produce energy except battery. The diagram of four cylinder magneto ignition system is as follows.



MAGNETO IGNITION SYSTEM

First when the engine starts or during cranking magneto rotate which generates a very high voltage.

The ignition capacitor connected in parallel with contact breaker. One end of magneto winding is also grounded through contact breaker.

The cam regulates the contact breaker. Wherever the breaker open, current flows into condenser, which charged the condenser.

As the condenser become charger the primary current falls and the magnetic field collapses. This will induces a much higher voltage in condenser.

Now this high voltage EMF produce spark at correct spark plug through distributor.

As the engine speed is low at starting, the current generated by the magneto is quite small. As the engine speed increases the flow of current also increases. Thus with magneto ignition system there is always starting problem and sometimes a separate battery needed for stating. This ignition system is best suited at high speed so it is used in racing cars, aircraft engines etc.

Advantages and Disadvantages:

Advantages:

1. This system is more reliable at medium and high speed.

2. It is more reliable because no battery is used.

3. It requires less frequently maintenance.

Disadvantages:

1. It has starting problem due to low cranking speed at starting.

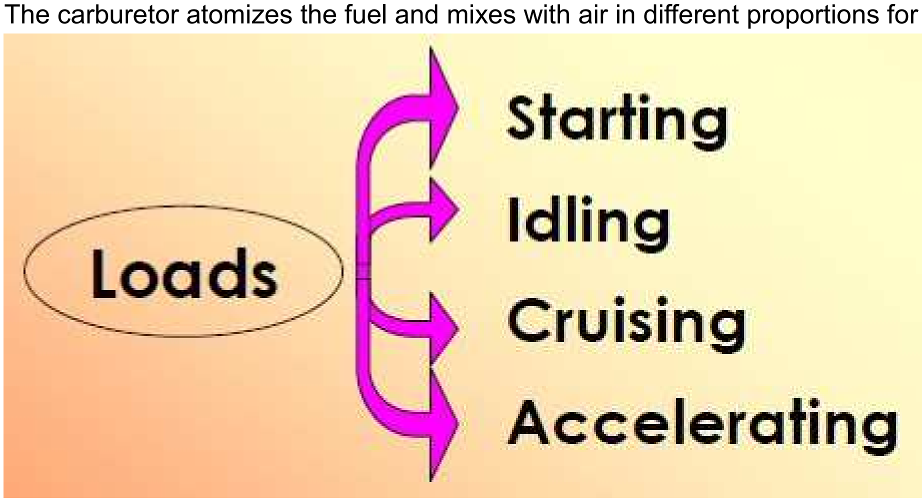
2. It is more expensive compare to battery ignition system.

3. There is possibility of misfire due to leakage because wiring carry very high voltage.

Carburetion

The process of mixture preparation in an SI engine is called carburetion. This air-fuel mixture is prepared outside the cylinder in a device called CARBURETOR.

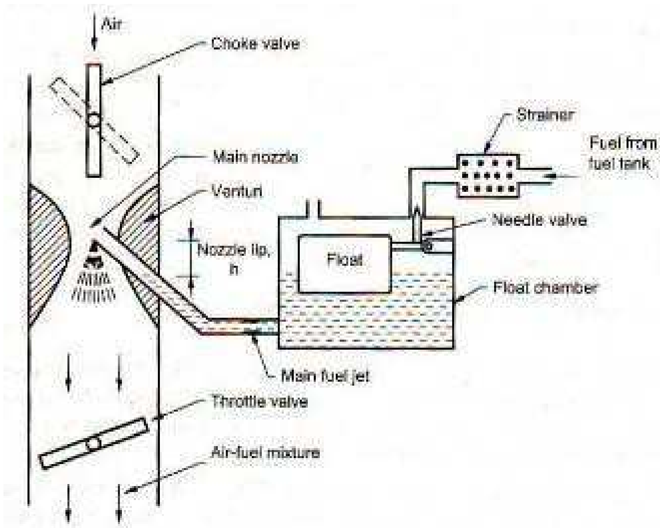
The carburetor atomizes the fuel and mixes with air in different proportions for various LOAD conditions.



Simple carburetor

The function of a carburetor is to vaporize the petrol (gasoline) by means of engine suction and to supply the required air and fuel (petrol) mixture to the engine cylinder. During the suction stroke, air flows from atmosphere into the cylinder. As the air passes through the venturi, velocity of air increases and its pressure falls below the atmosphere. The pressure at the nozzle tip is also below the atmospheric pressure. The pressure on the fuel surface of the fuel tank is atmospheric. Due to which a pressure difference is created, which causes the flow of fuel through the fuel jet into the air stream. As the fuel and air pass ahead of the venturi, the fuel gets vaporized and required uniform mixture is supplied to the engine.

The quantity of fuel supplied to the engine depends upon the opening of throttle valve which is governed by the governor.



The main parts of a simple carburetor are:

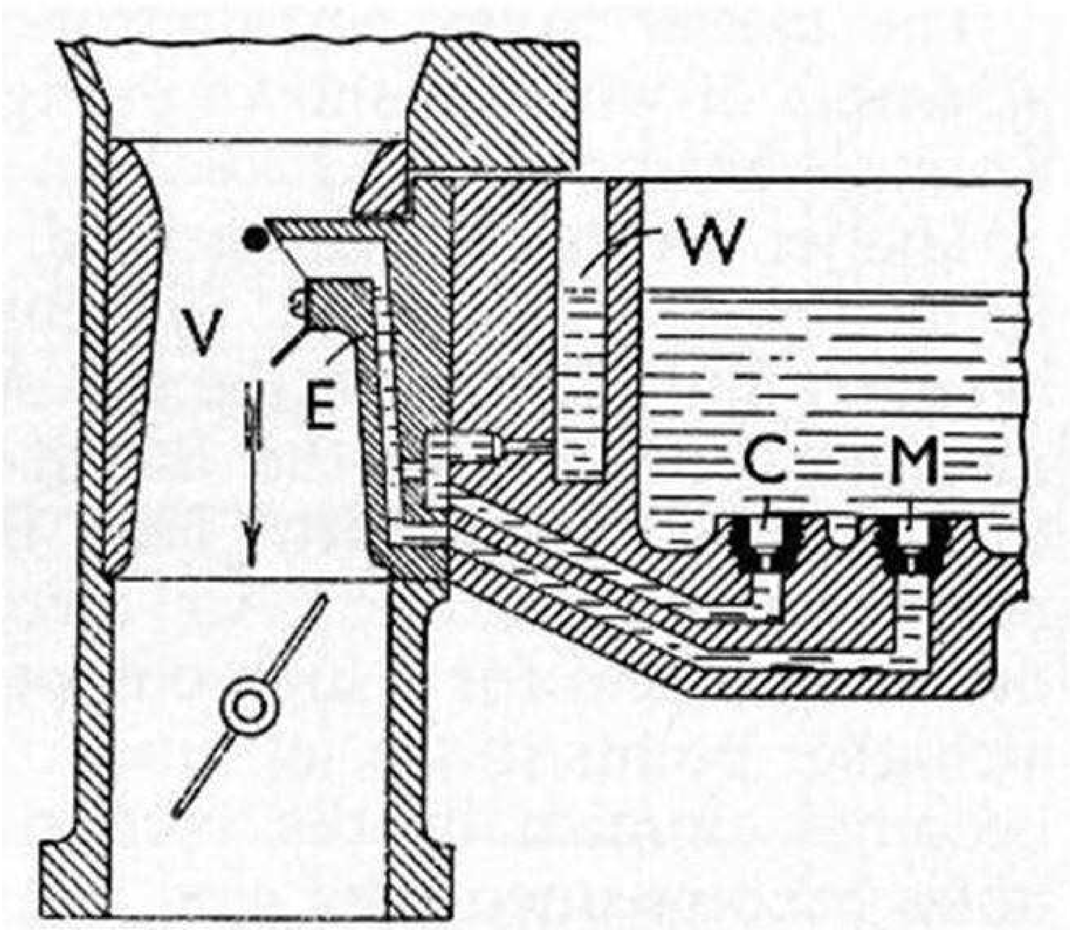
Float chamber: The level of fuel in the float chamber is maintained slightly below the tip of the nozzle. If the level of petrol is above then the petrol will run from the nozzle and drip from the carburetor. If the petrol level is kept low than the tip of the nozzle then part of pressure head is lost in lifting the petrol up to the tip of nozzle. Generally it is kept at 5mm from the level of petrol in the float chamber. The level of the fuel is kept constant with the help of float and needle valve. The needle valve closes the inlet supply from main tank if the level rises above the required level. If the level of fuel decreases then the needle valve opens the supply. Generally the fuel level is kept 5mm below the nozzle tip.

Venturi: When the mixture passes through the narrowest section its velocity increases and pressure falls below the atmospheric. As it passes through the divergent section, pressure increases again.

Throttle valve: It controls the quantity of air and fuel mixture supplied to the engine through intake manifold and also the head under which the fuel flows.

Choke: It provides an extra rich mixture during to the engine starting and in cold weather to warm up the engine. The choke valve is nearly closed during clod starting and warming. It creates a high vacuum near the fuel jet which causes flow of more fuel from the jet.

Zenith carburetor



The main object of the zenith carburetor is to supply the required quantity of fuel and air mixture of the correct strength as dedicated to the load condition of the engine.

In this, float chamber is supplied with fuel from the fuel tank through a pipe. Whenever the float chamber falls short of fuel, the fuel from the fuel tank flows into the chamber at the fastest speed. The speed of fuel will match the requirement of an engine. Hence the float rises up, till it reaches a certain level. At this time, a needle valve moves down and rest against the seat. So, it resulting the stoppage of fuel supply from the fuel tank. The main jet is directly connected to the float chamber. While the auxiliary jet which is also called as compensating jet draws fuel from an auxiliary chamber (Reservoir). This auxiliary chamber is connected to the float chamber through an orifice. Both, main and auxiliary jet is opened up in the venturi. The air to the carburetor is supplied through the passage. The throttle valve is located at the end of the carburetor and connected to the engine suction pipe. The opening and the closing of the throttle valve controls the quantity of air-fuel mixture supplied to the engine suction manifold. An auxiliary nozzle from an auxiliary chamber (Reservoir) is located at one end of the by-pass. The other end of this nozzle opens up near the throttle valve.

<u>Working of Zenith Carburetor at Starting and Low-Speed Running</u>

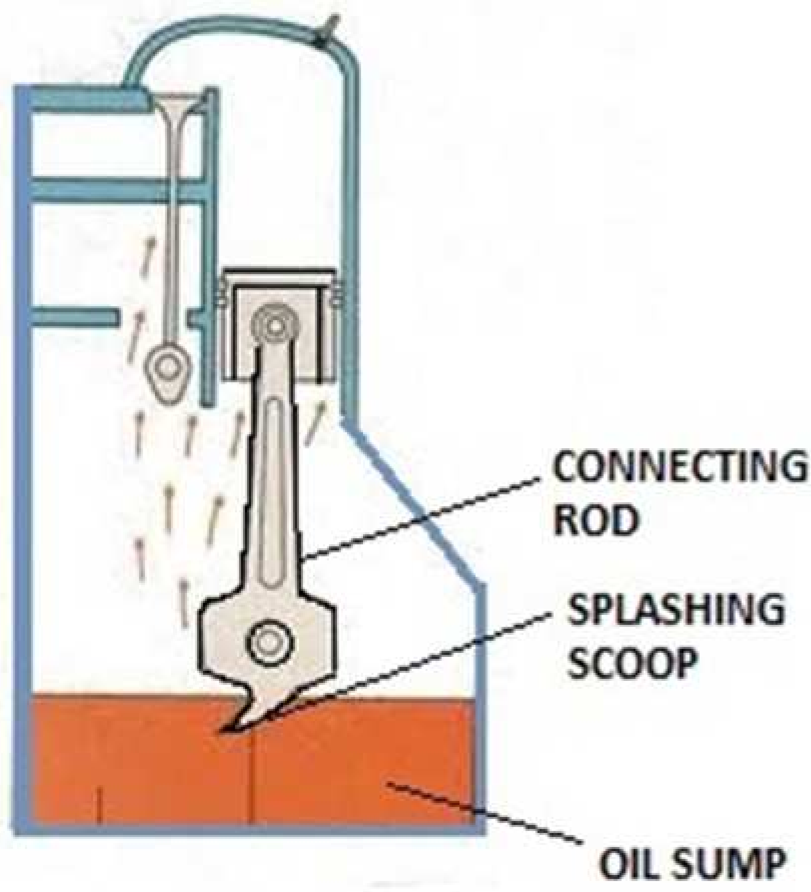
Because of lower velocity of air at the time of starting or slow speed of the engine, the suction produced at the venturi is quite insufficient to operate the main and the auxiliary jet in a nozzle. To improve the velocity of air, the throttle valve is closed to such an extent that there is only a small contracted passage is provided near the end of by-pass. By this, the velocity of air, passing through the region increases, producing the high suction. This operates the nozzle at the auxiliary chamber and the air-fuel mixture supplied through the holes. There is starting and slow running device is fitted in the reservoir (Auxiliary Chamber). To vary the supply of air to the nozzle, the set screw given is slackened and the whole assembly is taken out. By the suitable number of rotation of screw joint, the position of an auxiliary nozzle is set.

Lubrication System:

Splash type lubrication

This type of lubrication is generally used in some small four-stroke engines. In its construction a cap is present on the big end of the connecting rod which consists of a scoop. When the connecting rod is at the lowest position, the scoop gets dipped into the oil, thus it directs the oil into the holes present in the bearing. Due to the splash of the scoop oil reaches the lower position of the cylinder walls, crankshaft and other parts which requires lubrication. Oil level inside the pump is maintained a pump which takes oil from the sump through a filter.

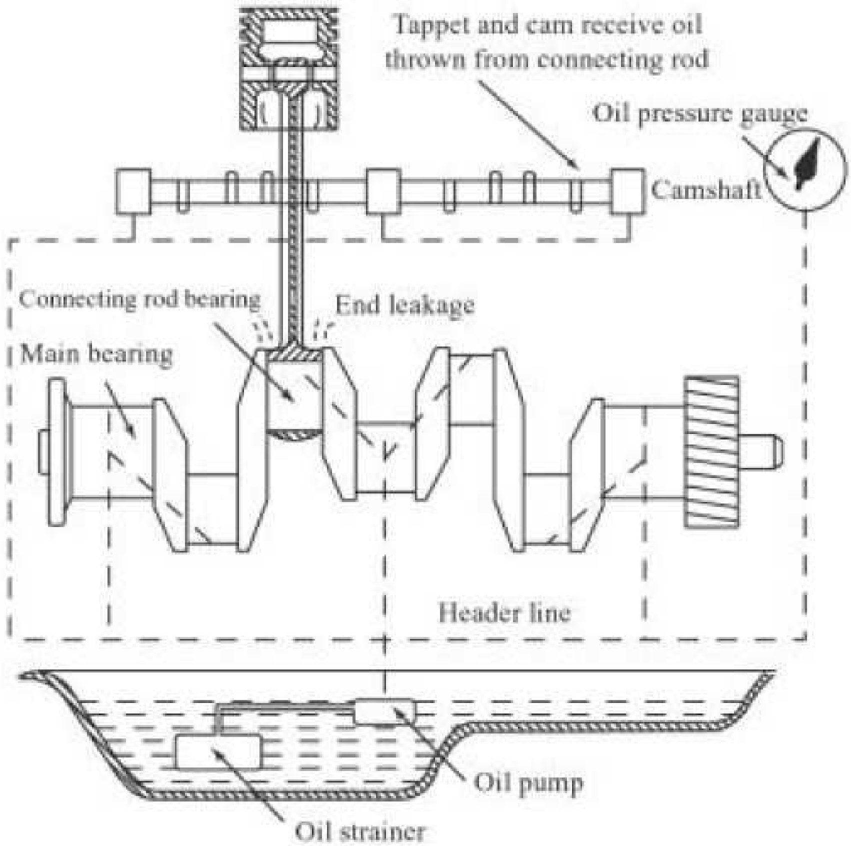
It is suitable for low and medium speed engines, which is generally having moderate bearing load pressure. This system does not serve properly for high speed engines, which normally operates at high bearing pressure.



SPLASH SYSTEM LUBRICATION

FULL PRESSURE SYSTEM

In this system oil is pumped from oil sump and it is distributed to various parts requiring lubrication. The oil is drawn from the oil sump through a filter and it is pumped by means of gear pump. Here oil is delivered at a pressure of 1.5 bar to 4 bar.



The oil with pressure is supplied to the main bearing in the crankshaft and camshaft. Holes drilled through the main crankshafts bearing journals, communicate oil to big end bearing and also small end bearings through holes drilled in the connecting rods. Generally a pressure gauge will be provided to confirm the circulation of oil to various parts. On the delivery side a pressure regulation valve is provided, to prevent the excessive pressure.

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

COMBUSTION PROCESS IN SI ENGINES

Combustion may be defined as a relatively rapid chemical combination of hydrogen and carbon in fuel with oxygen in air resulting in liberation of energy in the form of heat.

Following conditions are necessary for combustion to take place

1. **The presence of combustible mixture**
2. **Some means to initiate mixture**
3. **Stabilization and propagation of flame in Combustion Chamber**

In S I Engines, carburetor supplies a combustible mixture of petrol and air and spark plug initiates combustion

IGNITION LIMITS

Ignition of charge is only possible within certain limits of fuel-air ratio. Ignition limits correspond approximately to those mixture ratios, at lean and rich ends of scale, where heat released by spark is no longer sufficient to initiate combustion in neighbouring unburnt mixture. For hydrocarbons fuel the stoichiometric fuel air ratio is 1:15 and hence the fuel air ratio must be about 1:30 and 1:7

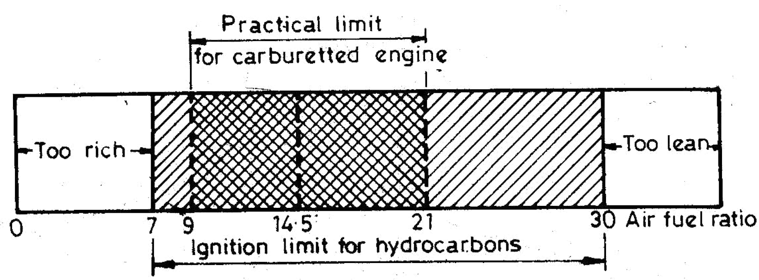


Fig. 5.1. Ignition limits for hydrocarbons.

THEORIES OF COMBUSTION IN SI ENGINE

Combustion in SI engine may roughly divided into two general types: Normal and Abnormal (knock free or Knocking). Theoretical diagram of pressure crank angle diagram is

shown. (a→b) is compression process, (b→c) is

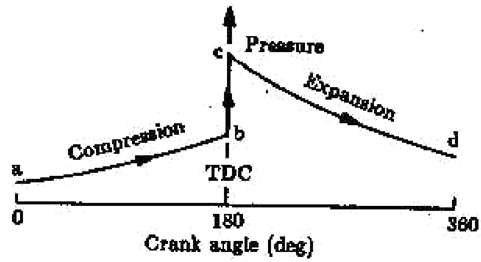


Fig.10.1 Theoretical p-θ Diagram

combustion process and (c→d) is an expansion process. In an ideal cycle it can be seen from the diagram, the entire pressure rise during combustion takes place at constant volume i.e., at TDC. However, in actual cycle this does not happen.

RICHARD'S THEORY OF COMBUSTION.

Sir Ricardo, known as father of engine research describes the combustion process can be imagined as if it is developing in two stages:

1. Growth and development of a self propagating nucleus flame. (Ignition lag)
2. Spread of flame through the combustion chamber

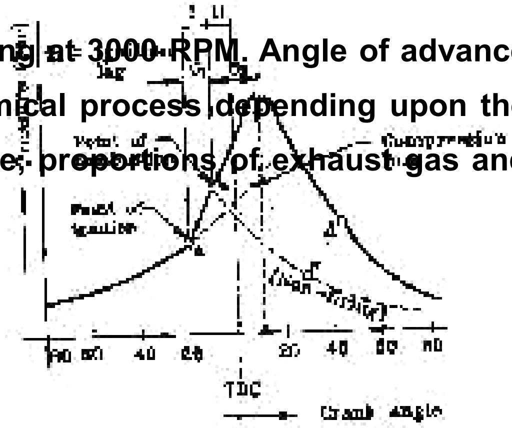
THREE STAGE OF COMBUSTION (VTU July/Aug 05/Feb 06/July 06)

According to Ricardo, There are three stages of combustion in SI Engine as shown

1. Ignition lag stage
2. Flame propagation stage
3. After burning stage

1. Ignition lag stage: There is a certain time interval between instant of spark and instant where there is a noticeable rise in pressure due to combustion. This time lag is called IGNITION LAG.

   Ignition lag is the time interval in the process of chemical reaction during which molecules get heated up to self ignition temperature , get ignited and produce a self propagating nucleus of flame. The ignition lag is generally expressed in terms of crank angle ($\theta1$). The period of ignition lag is shown by path ab. Ignition lag is very small and lies between 0.00015 to 0.0002 seconds. An ignition lag of 0.002 seconds corresponds to 35 deg crank rotation when the engine is running at 3000 RPM. Angle of advance increase with the speed. This is a chemical process depending upon the nature of fuel, temperature and pressure, proportions of exhaust gas and rate of oxidation or burning.



2. Flame propagation stage:

Once the flame is formed at “b”, it should be self sustained and must be able to propagate through the mixture. This is possible when the rate of heat generation by burning is greater than heat lost by flame to surrounding.

After the point “b”, the flame propagation is abnormally low at the beginning as heat lost is more than heat generated. Therefore pressure rise is also slow as mass of mixture burned is small. Therefore it is necessary to provide angle of advance 30 to 35 deg, if the peak pressure to be attained 5-10 deg after TDC. The time required for crank to rotate through an angle θ2 is known as combustion period during which propagation of flame takes place.

3.After burning:

Combustion will not stop at point “c” but continue after attaining peak pressure and this combustion is known as after burning. This generally happens when the rich mixture is supplied to engine.

FACTORS AFFCTING THE FLAME PROPAGATION (VTU Aug 06/July 07/Jan 07)

Rate of flame propagation affects the combustion process in SI engines. Higher combustion efficiency and fuel economy can be achieved by higher flame propagation velocities. Unfortunately flame velocities for most of fuel range between 10 to 30 m/second.

The factors which affect the flame propagations are

1. **Air fuel ratio**
2. **Compression ratio**
3. **Load on engine**
4. **Turbulence and engine speed**
5. **Other factors**

1. A : F ratio. The mixture strength influences the rate of combustion and amount of heat generated. The maximum flame speed for all hydrocarbon fuels occurs at nearly 10% rich mixture. Flame speed is reduced both for lean and as well as for very rich mixture. Lean mixture releases less heat resulting

lower flame temperature

and lower flame speed. Very rich mixture results incomplete combustion (C CO instead of C0 and also results in production of less heat and flame speed remains low. The effects of A: F ratio on p-v diagram and p-0 diagram are shown below :

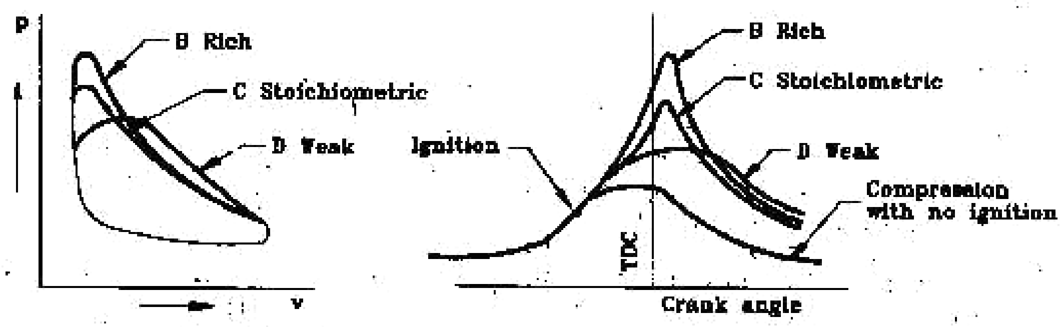


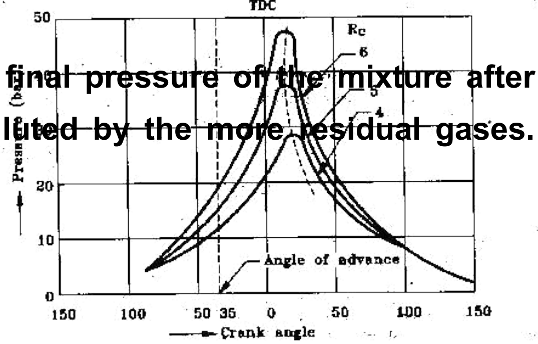
Fig. 17.7. Indicator diagrams for stoichiometric and weak mixtures.

2. **Compression ratio:** The higher compression ratio increases the pressure and temperature of the mixture and also decreases the concentration of residual gases. All these factors reduce the ignition lag and help to speed up the second phase of combustion. The maximum pressure of the cycle as well as mean effective pressure of the cycle with increase in compression ratio. Figure above shows the effect of compression ratio on pressure (indirectly on the speed of combustion) with respect to crank angle for same A: F ratio and same angle of advance. Higher compression ratio increases the surface to volume ratio and thereby increases the part of the mixture which after-burns in the third phase.

3. **Load on Engine.** With increase in load, the cycle pressures increase and the flame speed also increases.

In S.I. engine, the power developed by an engine is controlled by throttling. At

lower <u>load</u> and higher throttle, the initial and final pressure of the mixture after compression decrease and mixture is also diluted by the more residual gases. This reduces the flame



propagation and prolongs the ignition lag. This is the reason, the advance mechanism is also provided with change in load on the engine. This difficulty can be partly overcome by providing rich mixture at part loads but this definitely increases the chances of after-burning. The after burning is prolonged with richer mixture. In fact, poor combustion at part loads and necessity of providing richer mixture are the main disadvantages of S,I. engines which causes wastage of fuel and discharge of large amount of CO with exhaust gases.

4. Turbulence : Turbulence plays very important role in combustion of fuel as the flame speed is directly proportional to the turbulence of the mixture. This is because, the turbulence increases the mixing and heat transfer coefficient or heat transfer rate between the burned and unburned mixture. The turbulence of the mixture can be increased at the end of compression by suitable design of the combustion chamber (geometry of cylinder head and piston crown).

Insufficient turbulence provides low flame velocity and incomplete combustion and reduces the power output. But excessive turbulence is also not desirable as it increases the combustion rapidly and leads to detonation. Excessive turbulence causes to cool the flame generated and flame propagation is reduced.

Moderate turbulence is always desirable as it accelerates the chemical reaction, reduces ignition lag, increases flame propagation and even allows weak mixture to burn efficiently.

Engine Speed

The turbulence of the mixture increases with an increase in engine speed. For this reason the flame speed almost increases linearly with engine speed. If the engine speed is doubled, flame to traverse the combustion chamber is halved. Double the original speed and half the original time give the same number of crank degrees for flame propagation. The crank angle required for the flame propagation , which is main phase of combustion will remain almost constant at all speeds. This is an important characteristics of all petrol engines.

Engine Size

Engines of similar design generally run at the same piston speed. This is achieved by using small engines having larger RPM and larger engines having smaller RPM. Due to same piston speed, the inlet velocity, degree of turbulence and flame speed are nearly same in similar engines regardless of the size. However, in small engines the flame travel is small and in large engines large. Therefore, if the engine size is doubled the time required for propagation of flame through combustion space is also doubled. But with lower RPM of large engines the time for flame propagation in terms of crank would be nearly same as in small engines. In other words, the number of crank degrees required for flame travel will be about the same irrespective of engine size provided the engines are similar.

5. Other Factors. Among the other factors, the factors which increase the flame speed are supercharging of the engine, spark timing and residual gases left in the engine at the end of exhaust stroke. The air humidity also affects the flame velocity but its exact effect is not known. Anyhow, its effect is not large compared with A :F ratio and turbulence.

PHENOMENON OF KNOCKING IN SI ENGINE (VTU July06/Jan 07)

Knocking is due to auto ignition of end portion of unburned charge in combustion chamber. As the normal flame proceeds across the chamber, pressure and temperature of unburned charge increase due to compression by burned portion of charge. This unburned compressed charge may auto ignite under certain temperature condition and release

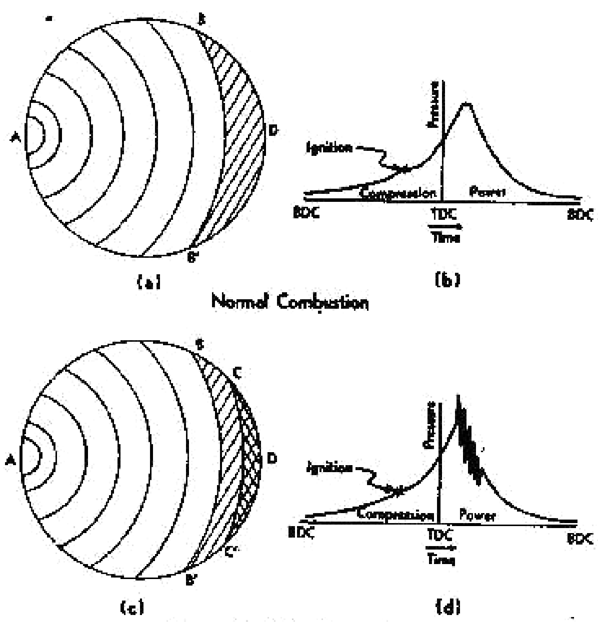
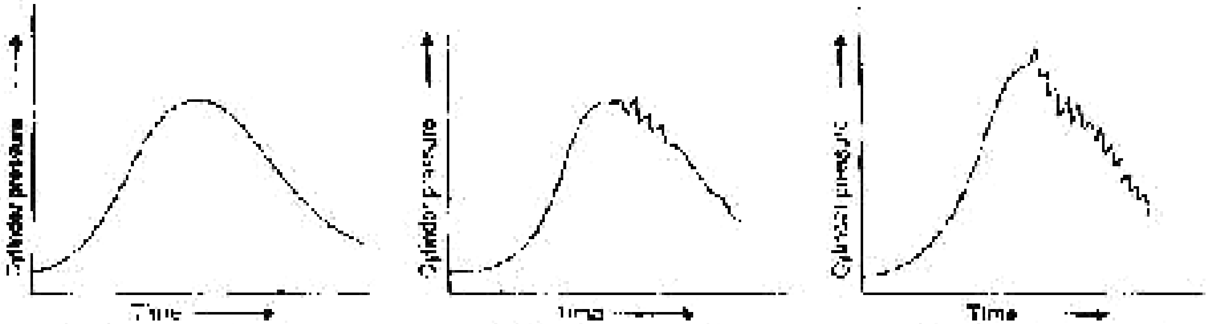


FIG. 8-4. Schematic presentation of the principles of normal and detonating combustion processes.

the energy at a very rapid rate compared to normal combustion

process in cylinder. This rapid release of energy during auto ignition causes a high pressure differential in combustion chamber and a high pressure wave is released from auto ignition region. The motion of high pressure compression waves inside the cylinder causes vibration of engine parts and pinging noise and it is known as knocking or detonation. This pressure frequency or vibration frequency in SI engine can be up to 5000 Cycles per second.



(i) Normal combustion without knock (ii) Combustion with slight knock (iii) Combustion with heavy knock

Denotation is undesirable as it affects the engine performance and life, as it abruptly increases sudden large amount of heat energy. It also put a limit on compression ratio at which engine can be operated which directly affects the engine efficiency and output.

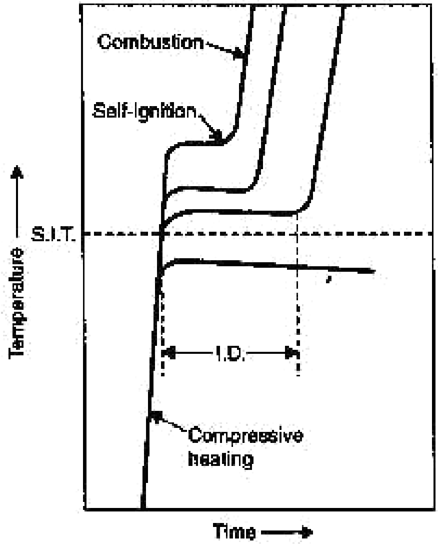
AUTO IGINITION (VTU July 2007)

A mixture of fuel and air can react spontaneously and produce heat by chemical reaction in the absence of flame to initiate the combustion or self-ignition. This type of self-ignition in the absence of flame is known as Auto-Ignition. The temperature at which the self-ignition takes place is known as self-igniting temperature. The pressure and temperature abruptly increase due to auto-ignition because of sudden release of chemical energy.

This auto-ignition leads to abnormal combustion known as detonation which is undesirable because its bad effect on the engine performance and life as it abruptly increases sudden large amount of heat energy. In addition to this knocking puts a limit on the compression ratio at which an engine can be

operated which directly affects the engine efficiency and output.

Auto-ignition of the mixture does not occur instantaneously as soon as its temperature rises above the self-ignition temperature. Auto-ignition occurs only when the mixture stays at a temperature equal to or higher than the self-ignition temperature for a "finite time". This time is known as delay period or reaction time for auto-ignition. This delay time as a function of compression ratio is shown in adjacent figure.



S.I.T. → Self-ignition temperature
I.D. → Ignition delay
Fig. 9.17. Self-ignition charactristics of fuels.

As the compression ratio increases, the delay period decreases and this is because of increase in initial (before combustion) pressure and temperature of the charge. The self-ignition temperature is a characteristic of fuel air mixture and it varies from fuel to fuel and mixture strength to mixture - strength of the same fuel.

PRE -IGINITION (VTU July 2007)

Pre-ignition is the ignition of the homogeneous mixture of charge as it comes in contact with hot surfaces, in the absence of spark .

Auto ignition may overheat the spark plug and exhaust valve and it remains so hot that its temperature is sufficient to ignite the charge in next cycle during the compression stroke before spark occurs and this causes the pre-ignition of the charge.

Pre-ignition is initiated by some overheated projecting part such as the sparking plug electrodes, exhaust valve head, metal corners in the combustion chamber, carbon deposits or protruding cylinder head gasket rim etc.

pre-ignition is also caused by persistent detonating pressure shockwaves scoring away the stagnant gases which normally protect the combustion

chamber walls. The resulting increased heat flow through the walls, raises the surface temperature of any protruding poorly cooled part of the chamber, and this there fore provides a focal point for pre-ignition.

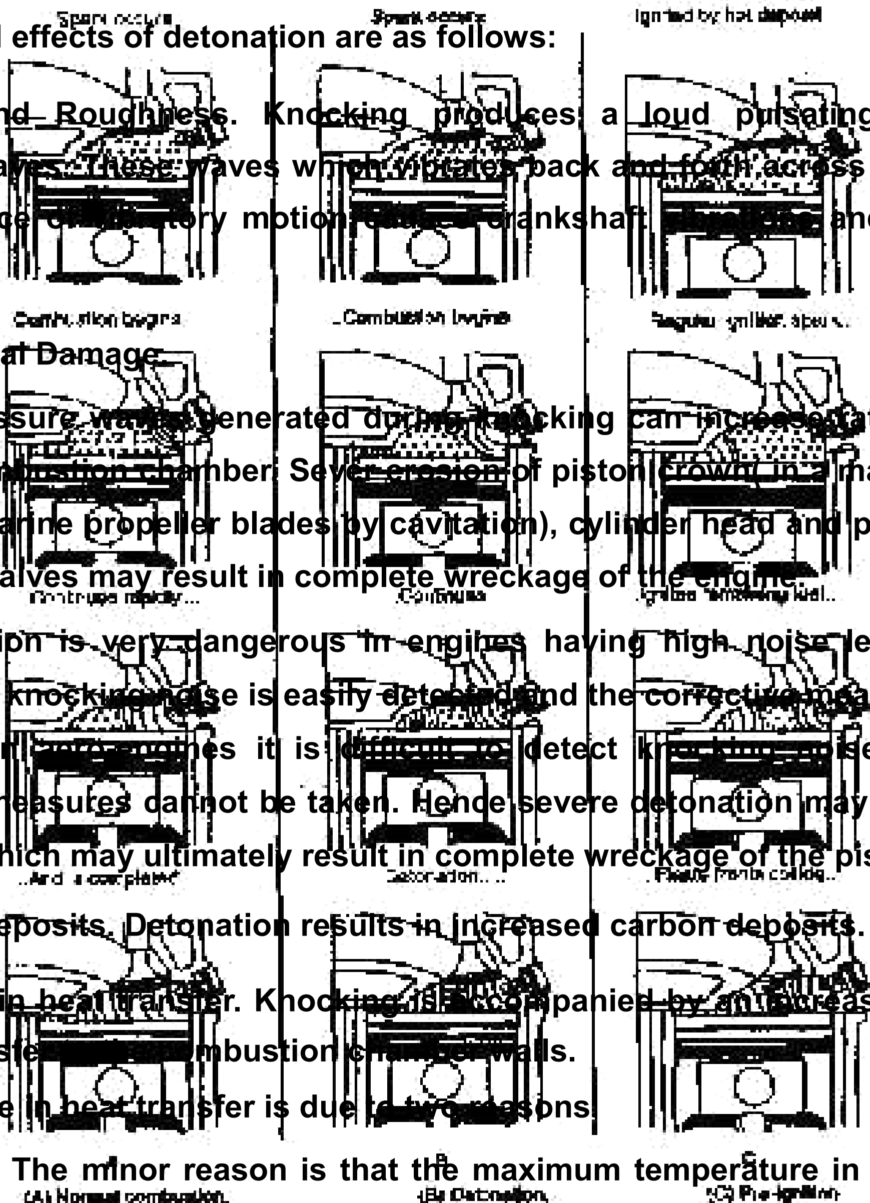
Effects of Pre-ignition

- **It increase the tendency of denotation in the engine**
- **It increases heat transfer to cylinder walls because high temperature gas remains in contact with for a longer time**
- **Pre-ignition in a single cylinder will reduce the speed and power output**
- **Pre-ignition may cause seizer in the multi-cylinder engines, only if only cylinders have pre-ignition**

DIFFERENCE BETWEEN NORMAL/ABNORMAL COMBUSTION AND PRE- INGINITON

EFFECT OF DETONATION (VTU Jan 2006)

The harmful effects of detonation are as follows:



1. **Noise and Roughness.** **Knocking produces a loud pulsating noise and pressure waves. These waves which vibrates back and forth across the cylinder. The presence of vibratory motion causes crankshaft vibrations and the engine runs rough.**

2. **Mechanical Damage.**

(a)High pressure waves generated during knocking can increase rate of wear of parts of combustion chamber. Sever erosion of piston crown(in a manner similar to that of marine propeller blades by cavitation), cylinder head and pitting of inlet and outlet valves may result in complete wreckage of the engine.

(b) Detonation is very dangerous in engines having high noise level. In small engines the knocking noise is easily detected and the corrective measures can be taken but in aero-engines it is difficult to detect knocking noise and hence corrective measures cannot be taken. Hence severe detonation may persist for a long time which may ultimately result in complete wreckage of the piston.

3. **Carbon deposits. Detonation results in increased carbon deposits.**

4. **Increase in heat transfer. Knocking is accompanied by an increase in the rate of heat transfer to the combustion chamber walls.**

The increase in heat transfer is due to two reasons.

- **The minor reason is that the maximum temperature in a detonating engine is about 150°C higher than in a non-detonating engine, due to rapid completion of combustion**

- **The major reason for increased heat transfer is the scouring away of protective layer of inactive stagnant gas on the cylinder walls due to pressure waves. The inactive layer of gas normally reduces the heat transfer by protecting the combustion and piston crown from direct contact with flame.**

5. **Decrease in power output and efficiency. Due to increase in the rate of heat transfer the power output as well as efficiency of a detonating engine decreases.**

6 Pre-ignition. The increase in the rate of heat transfer to the walls has yet another effect. It may cause local overheating, especially of the sparking plug, which may reach a temperature high enough to ignite the charge before the passage of spark, thus causing pre-ignition. An engine detonating for a long period would most probably lead to pre-ignition and this is the real danger of detonation.

EFFECT OF ENGINE OPERATING VARIABLES ON THE ENGINE KNOCKING DETONATION (VTU July 2005)

The various engine variable affecting knocking can be classified as :

- **Temperature factors**
- **Density factors**
- **Time factors**
- **Composition factors**

(A) TEMPERATURE FACTORS.

Increasing the temperature of the unburned mixture increase the possibility of knock in the SI engine We shall now discuss the effect of following engine parameters on the temperature of the unburned mixture:

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 which in turn increases the tendency to knock.

SUPERCHARGING. It also increases both temperature and density, which increase the knocking tendency of engine

COOLANT TEMPERATURE Delay period decreases with increase of coolant temperature , decreased delay period increase the tendency to knock

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 uneven temperature leads to pre-ignition and hence the knocking.

(B) DENSITY FACTORS.

Increasing the density of unburnt mixture will increase the possibility of knock in the engine. The engine parameters which affect the density are as follows:

Increased compression ratio increase the density

Increasing the load opens the throttle valve more and thus the density Supercharging increase the density of the mixture

Increasing the inlet pressure increases the overall pressure during the cycle. The high pressure end gas decreases the delay period which increase the tendency of knocking.

Advanced spark timing : quantity of fuel burnt per cycle before and after TDC position depends on spark timing. The temperature of charge increases by increasing the spark advance and it increases with rate of burning and does not allow sufficient time to the end mixture to dissipate the heat and increase the knocking tendency

(C) TIME FACTORS.

Increasing the time of exposure of the unburned mixture to auto-ignition conditions increase the possibility of knock in SI engines.

Flame travel distance: If the distance of flame travel is more, then possibility of knocking is also more. This problem can be solved by combustion chamber design, spark plug location and engine size. Compact combustion chamber will have better 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.

Location of sparkplug. 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.

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.

Engine size. Large engines have a greater knocking tendency because flame requires a longer time to travel across the combustion chamber. In SI engine therefore , generally limited to 100mm

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.

(D) COMPOSITION.

(Influence of chemical structure on knocking – VTU August 2005)

The properties of fuel and A/F ratio are primary means to control knock :

(a) **Molecular Structure.** The knocking tendency is markedly affected by the type of the fuel used. Petroleum fuels usually consist of many hydro-carbons of different molecular structure. The structure of the fuel molecule has enormous effect on knocking tendency. Increasing the carbon-chain increases the knocking tendency and centralizing the carbon atoms decreases the knocking tendency.

Unsaturated hydrocarbons have less knocking tendency than saturated hydro-carbons.

Paraffins

Increasing the length of carbon chain increases the knocking tendency. Centralising the carbon atoms decreases the knocking tendency.

Adding methyl group (CH to the side of the carbon chain in the centre position decreases the knocking tendency.

Olefins

Introduction of one double bond has little effect on anti-knock quality but two or three double bond results less knocking tendency except C and C

Napthenes and Aromatics

Napthenes have greater knocking tendency than corresponding aromatics. With increasing double-bonds, the knocking tendency is

reduced.

Lengthening the side chains increases the knocking tendency whereas branching of the side chain decreases the knocking tendency.

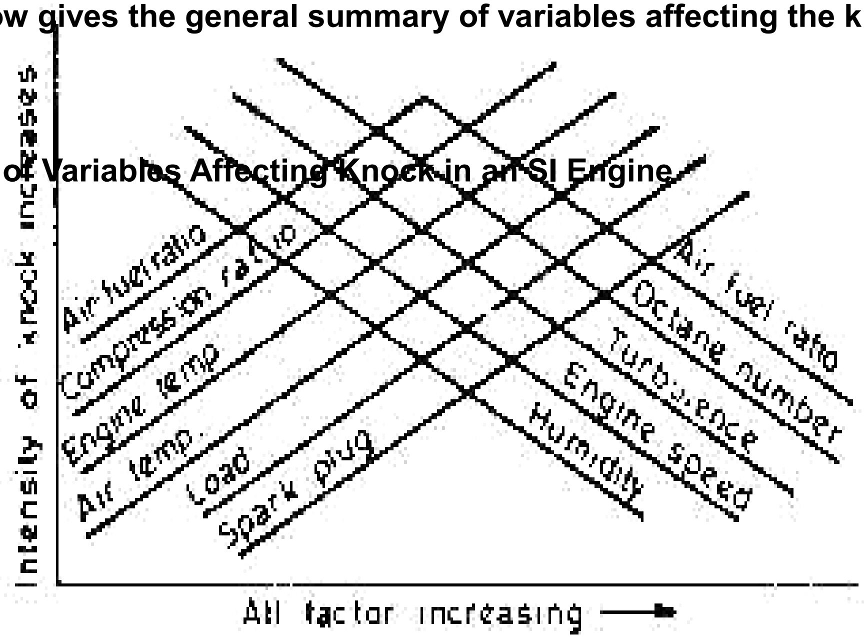
(b) Fuel-air ratio. The most important effect of fuel-aft ratio is on the reaction time or ignition delay. When the mixture is nearly 10% richer than stoichiomiric (fuel-air ratio = 0.08) ignition lag of the end gas is minimum and the velocity of flame propagation is maximum. By making the mixture leaner or richer (than F/A 0.08) the tendency to knock is decreased. A too rich mixture is especially effective in decreasing or eliminating the knock due to longer delay and lower temperature of compression.

(c)Humidity of air. Increasing atmospheric humidity decreases the tendency to knock by decreasing the reaction time of the fuel

The trends of the most of the above factors on knocking tendency of the engine is given below:

Table below gives the general summary of variables affecting the knock in an SI engine

Summary of Variables Affecting Knock in an SI Engine



Effect of engine variables on Knocking in SI engine (VTU Jan 2007)

Increase in variable	Major effect on unburned charge	Action to be taken to reduce knocking	Can operator usually control?
Compression ratio	Increases temperature & pressure	Reduce	No
Mass of charge inducted	Increases pressure	Reduce	Yes
Inlet temperature	Increases temperature	Reduce	In some cases
Chamber wall temperature	Increases temperature	Reduce	Not ordinarily
Spark advance	Increases temperature & pressure	Retard	In some cases
A/F ratio	Increases temperature & pressure	Make very rich	In some cases
Turbulence	Decreases time factor	Increase	Somewhat (through engine speed)
Engine speed	Decreases time factor	Increase	Yes
Distance of Flame travel	Increases time factor	Reduce	No

1. Compression ratio: The pressure and temperature at the end of compression increases with increase in compression ratio. This in turn increase the maximum pressure during the combustion and creates a tendency to knock
2. Supercharging : increase the temperature and density of mixture and thus the tendency to knock is increased.
3. Turbulence : decreasing the turbulence of mixture decreases the flame speed and hence increase the tendency to knock
4. Octane rating of fuel : higher the octane number, less the tendency to knock. Paraffins have maximum tendency to knock and aromatic series have minimum tendency to knock . (also see Influence of chemical structure on knocking)

Factors that limits the compression ratio in petrol engine (VTU July 2007)

In petrol engine we use the mixture of air and petrol and thermal efficiency of petrol engine increase with increase in compression ratio. But the value of compression ration is limited by phenomenon of knocking. The pressure and temperature at the end of compression increases with increase in compression ratio. This in turn increase the maximum pressure during the combustion and creates a tendency to knock. Thus, Higher compression ratio, higher is the tendency to knock, therefore the value of compression ratio in petrol engine is limited to 6 to 10.

Compression ratio can be marginally improved by using fuel with Tetra-ethyl lead. TEL delays the auto ignition and allows it to occur at higher temperature and thus reduces knocking. The use of TEL is now in disfavor because of atmospheric pollution (lead is toxic and has serious environmental and health hazards).

KNOCK RATING OF SI ENGINE FUELS (OCTANE NUMBER) (VTU Jan 2006)

The tendency to detonate depends on composition of fuel. Fuel differ widely in their ability to resist knock. The property of fuel which describes how fuel will or will nor self ignite is called the OCTANE NUMBER. It is defined as the percentage of Iso-octane by volume in a mixture of Iso-octane and n-heptane which exactly matches the knocking

tendency of a given fuel, in a standard fuel under given standard operating conditions. The rating of a particular SI fuel is done by comparing its antiknock performance with that of standard reference fuel which is usually combination of Iso-octane and n-heptane. Iso-octane (C_8H_{18}) which has a very high resistance to knock and therefore it is arbitrarily assigned a rating of 100 octane number. N-heptane (C_7H_{16}) which is very prone to knock and therefore given a zero value.

For example: Octane number 80 means that the fuel has same knocking tendency as mixture of 80% iso-octane and 20% n-heptane (by volume basis).

A fuel having an octane number of 110 means fuel has the same tendency to resist as a mixture of 10 cc of Tetra ethyl lead (TEL) in one U.S gallon of Iso-octane.

HIGHEST USEFUL COMPRESSION RATIO (HUCR) (VTU July 2005)

The thermal efficiency of IC engine increase with increase in Compression Ratio. The maximum compression ratio of any SI engine is limited by its tendency to knock. HUCR is the highest compression ratio employed at which a fuel can be used in a specified engine under specified set of operating conditions, at which detonation first becomes audible with both ignition and mixture strength adjusted to give highest efficiency.

HUCR of different fuel

Iso-octane	10.96
n-heptane	3.75
Toulene	15
Cyclo hexane	8.20

Anti Knock Agents

The knock resistance tendency of a fuel can be increased by adding anti-knock agents. The anti knock agents are substances which decreases the rate of

2 5 . 4

preflame reaction by delaying the auto ignition of the end mixture in engine until flame generated by spark

plug.

TEL b is most powerful anti knock agents. TEL increase the efficiency of

engine and increase the specific output of SI engine. Its use will not improve the performance of engine which is not knocking unless the spark advanced. CR is increased or higher inlet pressure is used to take advantage of an increase in octane number. The use of leaded gasoline. However is not perfect solution to problem. It leads to emission of lead into atmosphere which is known to be very hazardous.

The following table shows some anti knock agents and effectiveness.

Compound	Chemical Symbol	Weight for given effect (gm)	Relative weight
Tetraethyl lead	$Pb(C_2H_5)_4$	0.0295	1
Aniline	$C_6H_5NH_2$	1	34
Ethyl Iodide	C_2H_5 I	1.55	53
Ethyl alcohol	C_2H_5 OH	4.75	161
Xylene	$[C_6H_4CH_3]_2$	8.00	271
Toluene	$C_6H_5CH_3$	8.8	298
Benzene	C_6H_6	9.8	332

The following table gives the relative effectiveness of anti knocks

Compound	Relative effectiveness
Tetra ethyl lead (TEL)	100
Methyl cyclo pentadienyl	65
Manganese tricarbonyl Ironcarbonyl	43
Copper methyl arnino methyle necetate	40
Nickel carbonyl	30
Tri ethyl bismuth	20
Tetra ethyl tin	3
N-Methyl aniline-Ethyl iodide	11

The effect of anti knock agents on HUCR is shown below

S I engines are generally not supercharged.” Justify this statement.

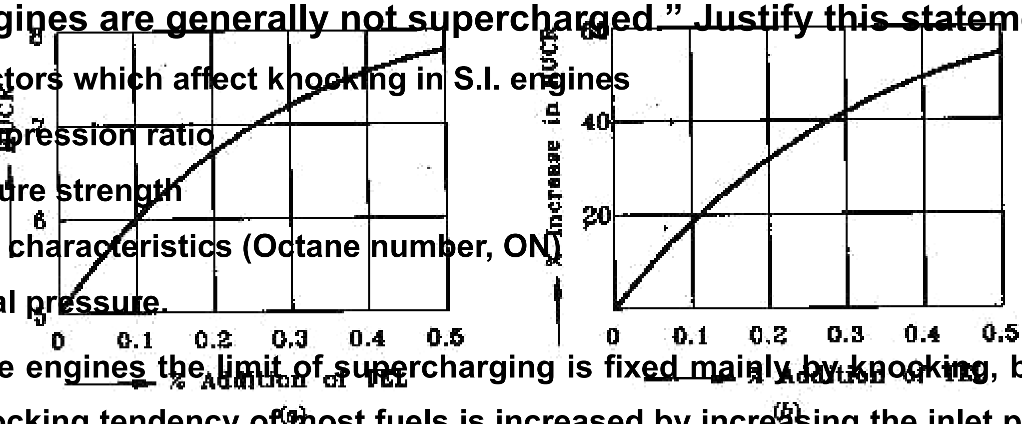
The factors which affect knocking in S.I. engines

— Compression ratio

— Mixture strength

— Fuel characteristics (Octane number, ON)

— Initial pressure.



In these engines the limit of supercharging is fixed mainly by knocking, because the knocking tendency of most fuels is increased by increasing the inlet pressure and temperature, or both. At the same ON requirement, if the charge density is increased the compression ratio has to be decreased considering the knock limits. Thus the power by the supercharged engine is increased but at reduced thermal efficiency. Further, supercharged S.I. engines are usually to run on rich mixture, for maximum power. This also results in a higher S F C. Therefore, S.I. engines are not generally supercharged, except to compensate for loss of power at high altitudes.

COMBUSTION PROCESS IN CI ENGINES

In SI engine, uniform A: F mixture is supplied, but in CI engine A: F mixture is not homogeneous and fuel remains in liquid particles, therefore quantity of air supplied is 50% to 70% more than stiochiometric mixture.

The combustion in SI engine starts at one point and generated flame at the point of ignition propagates through the mixture for burning of the mixture, where as in CI engine, the combustion takes place at number of points simultaneously and number of flames generated are also many. To burn the liquid fuel is more difficult as it is to be evaporated; it is to be elevated to ignition temperature and then burn.

STAGES OF COMBUSTION IN CI ENGINE (JAN 2007/JULY2006)

The combustion in CI engine is considered to be taking place in four phases:

- Ignition Delay period /Pre-flame combustion
- Uncontrolled combustion
- Controlled combustion
- After burning

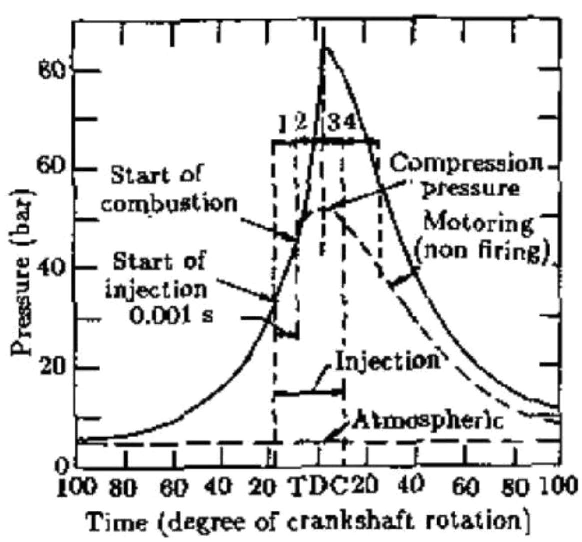


Fig1.Stages of combustion

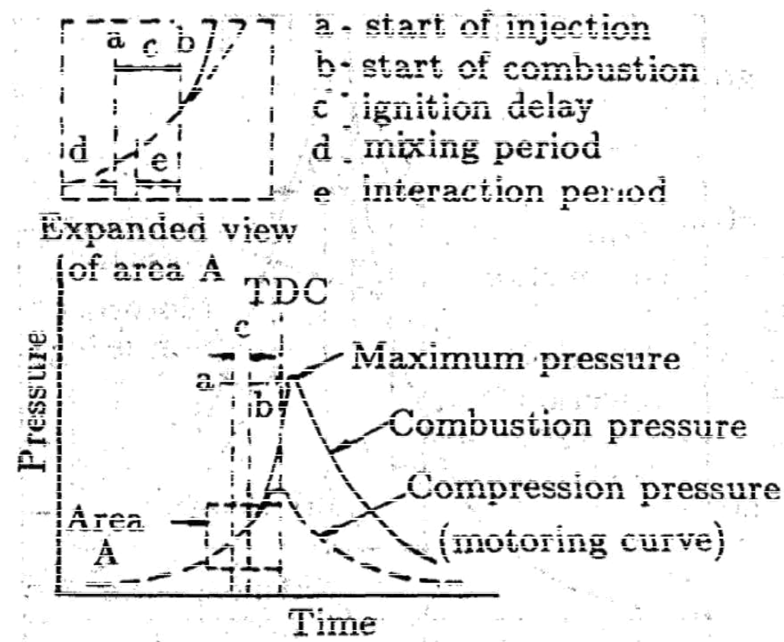


Fig 2. Pressure Time diagram illustrating Ignition delay

Ignition Delay period /Pre-flame combustion

The fuel does not ignite immediately upon injection into the combustion chamber. There is a definite period of inactivity between the time of injection and the actual burning this period is known as the ignition delay period.

In Figure 2. the delay period is shown on pressure crank angle (or time) diagram between points a and b. Point “a” represents the time of injection and point “b” represents the time of combustion. The ignition delay period can be divided into two parts, the physical delay and the chemical delay.

The delay period in the CI engine exerts a very great influence on both engine design and performance. It is of extreme importance because of its effect on both the combustion rate and knocking and also its influence on engine starting ability and the presence of smoke in the exhaust.

2 Period of Rapid Combustion

The period of rapid combustion also called the uncontrolled combustion, is that phase in which the pressure rise is rapid. During the delay period, a considerable amount of fuel is accumulated in combustion chamber, these accumulated fuel droplets burns very rapidly causing a steep rise in pressure. The period of rapid combustion is counted from end of delay period or the beginning of the combustion to the point of maximum pressure on the indicator diagram. The rate of heat-release is maximum during this period. This is also known as uncontrolled combustion phase, because it is difficult to control the amount of burning / injection during the process of burning.

It may be noted that the pressure reached during the period of rapid combustion will depend on the duration of the delay period (the longer the delay the more rapid and higher is the pressure rise since more fuel would have been present in the cylinder before the rate of burning comes under control).

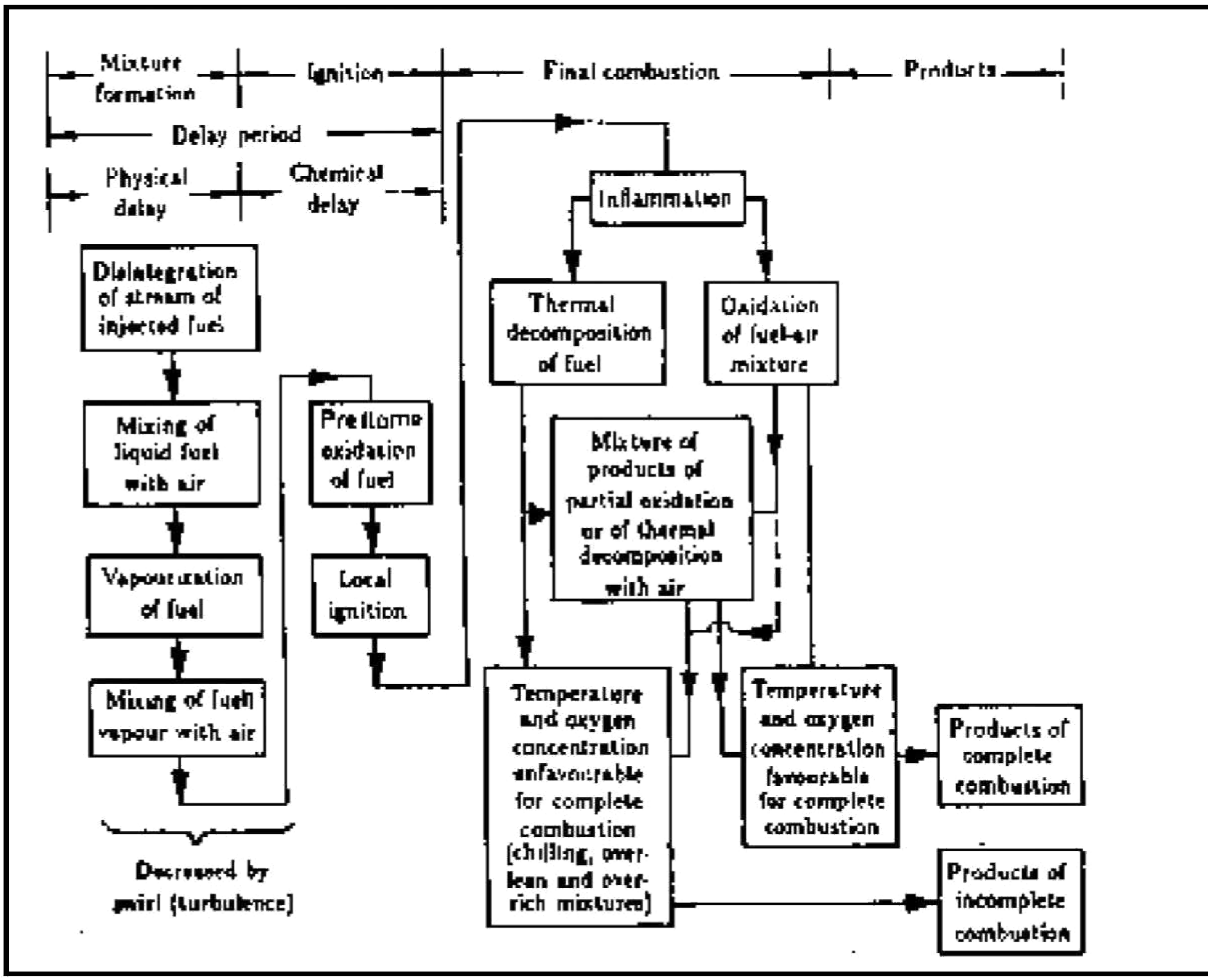
3 Period of Controlled Combustion

The rapid combustion period is followed by the third stage, the controlled combustion. The temperature and pressure in the second stage are so high that fuel droplets injected burn almost as they enter and find the necessary oxygen and any further pressure rise can be controlled by injection rate. The period of controlled combustion is assumed to end at maximum cycle temperature.

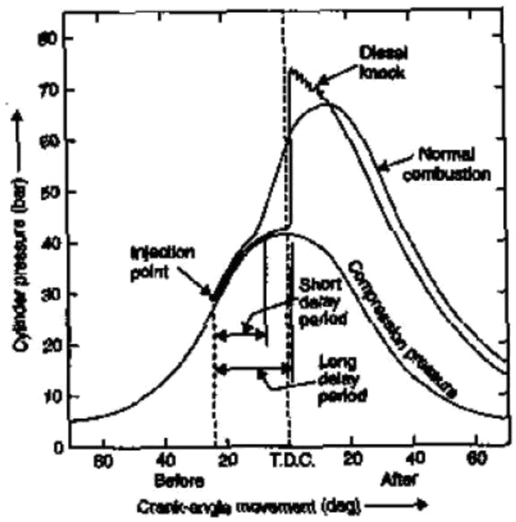
4 Period of After-Burning

Combustion does not stop with the completion of the injection process. The unburnt and partially burnt fuel particles left in the combustion chamber start burning as soon as they come into contact with the oxygen. This process continues for a certain duration called the after-burning period. This burning may continue in expansion stroke up to 70 to 80% of crank travel from TDC.

The sequence of the events in the entire combustion process in a CI engine including the delay period is shown in Figure 3 by means of a block diagram.



Ignition Delay or Ignition Lag (VTU Feb 2006)



The delay period is the time between the start of injection and start of combustion. The delay period extends for about 13 deg movement of crank. This delay time decreases with increase in speed. If there is no delay, the fuel would burn at injector and there would be oxygen deficiency around the injector, which results in incomplete combustion. If the delay period is too long, amount of fuel availability for simultaneous explosion , is too great , which results in rapid pressure rise. The delay period should be as short as possible since long delay period gives more rapid rise in pressure and thus causes knocking.

Component of Ignition Delay or Ignition Lag (VTU Feb 2006)

Ignition delay can be divided into two parts:

Physical Delay: The physical delay is the time between the beginning of injection and the attainment of chemical reaction conditions. During this period, the fuel is atomized, vaporized, mixed with air and raised to its self-ignition temperature. This physical delay depends on the type of fuel, i.e., for light fuel the physical delay is small while for heavy viscous fuels the physical delay is high. The physical delay is greatly reduced by using high injection pressures and high turbulence to facilitate breakup of the jet and improving evaporation.

Chemical Delay: During the chemical delay reactions start slowly and then accelerate until inflammation or ignition takes place. Generally, the chemical delay is larger than the physical delay. However, it depends on the temperature of the surroundings and at high temperatures, the chemical reactions are faster and the physical delay

Total delay period = Physical delay + Chemical delay

$$t_t = t_p + t_c \ ,$$

In CI engine $t_p >> t_c$,

In SI engine $t_p \approx 0$

Combustion phenomenon in CI engine V/s combustion in SI engine. (VTU July 2006)

SL NO	COMUSTION IN SI ENGINE	COMBUSTION IN CI ENGINE
1	Homogeneous mixture of petrol vapour and air is compressed (CR 6:1 to 11:1) at the end of compression stroke and is ignited at one place by spark plug.	Air alone is compressed through large Compression ratio (12:1 to 22:1)and fuel is injected at high pressure of 110 to 200 bar using fuel injector pump.
2	Single definite flame front progresses through air fuel mixture and entire mixture will be in combustible range	Fuel is not injected at once, but spread over a period of time. Initial droplets meet air whose temperature is above self ignition temperature and ignite after ignition delay.
3	For effective combustion, turbulence is required. Turbulence which is required in SI engine implies disordered air motion with no general direction of flow to break up the surface of flame front and to distribute the shreds of flame thought-out in externally prepared homogeneous combustible mixture.	For effective combustion, swirl is required. Swirl which is required in CI engine implies an orderly movement of whole body of air with a particular direction of flow, to bring a continuous supply of fresh air to each burning droplets and sweep away the products of combustion which otherwise suffocate it.
4	In SI Engine ignition occurs at one point with a slow rise in pressure	In the CI engine, the ignition occurs at many points simultaneously with consequent rapid rise in pressure. There is no definite flame front.
5	In SI engine physical delay is almost zero and chemical delay controls combustion	In CI engine physical delay controls combustion.
6	In SI engine , A/F ratio remains close to stoichiometric value from no load to full load	In CI engine , irrespective of load, at any speed, an approximately constant supply of air enters the cylinder. With change in load, quantity of fuel is changed to vary A/F ratio. The overall A/F can Range from 18:1 to 80:1.
5	Delay period must be as long as possible. High octane fuel(low cetane) is required.	Delay period must be as short as possible. High cetane (low octane) fuel is required

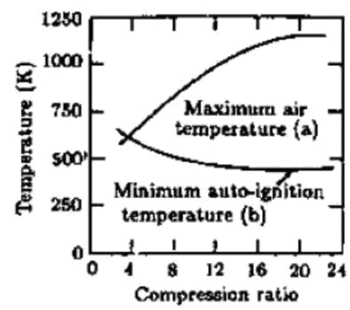
Home work : Good SI engine fuel is bad CI engine fuel – Justify this statement

EFFECT OF VARIOUS FACTORS ON DELAY PERIOD IN CI ENGINE (VTU July 06/July07)

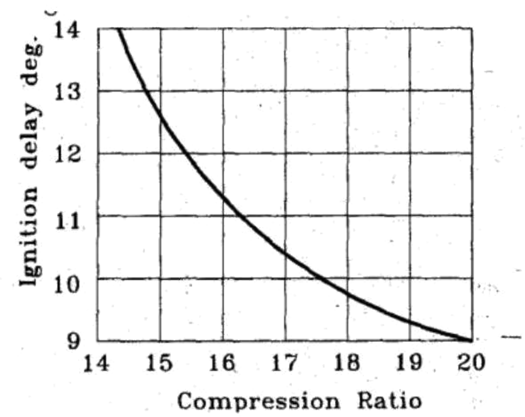
Many design and operating factors affect the delay period. The important ones are:

- compression ratio
- engine speed
- output
- injection timing
- quality of the fuel
- intake temperature
- intake pressure

1.Compression Ratio. The increase in the compression temperature of the air with increase in compression ratio evaluated at the end of the compression stroke is shown in Fig. It is also seen from the same figure that the minimum auto ignition temperature of a fuel decreases due to increased density of the compressed air. This results in a closer contact between the molecules of fuel and oxygen reducing the time of reaction. The increase in the compression temperature as well as the decrease in the minimum auto ignition temperature decrease the delay period. The maximum peak pressure during the combustion process is only marginally affected by the compression ratio (because delay period is shorter with higher compression ratio and hence the pressure rise is lower).



Effect of Compression Ratio on Maximum Air Temperature and Minimum Autoignition Temperature



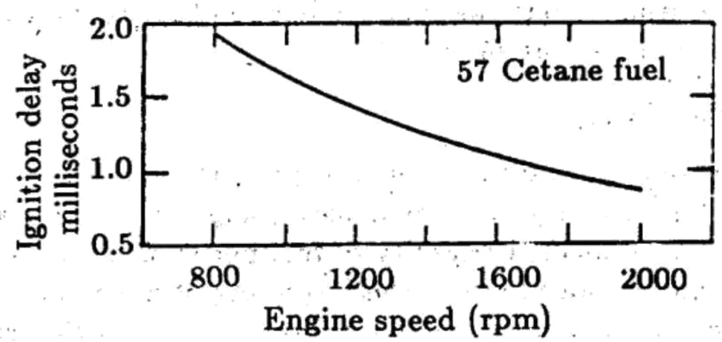
Then why we do not use very high compression ratio in CI?

One of the practical disadvantages of using a very high compression ratio is that the mechanical efficiency tends to decrease due to increase in weight of the reciprocating parts. Therefore, engine designers always try to use a lower compression ratio which helps in easy cold starting and light load running at high speeds.

2.Engine Speed:

The delay period could be given either in terms of absolute time (in milliseconds) or in terms of crank angle degrees

With increase in engine speed, the loss of heat during compression decreases, resulting in the rise of both the temperature and pressure of the compressed air thus reducing the delay period in milliseconds. However,



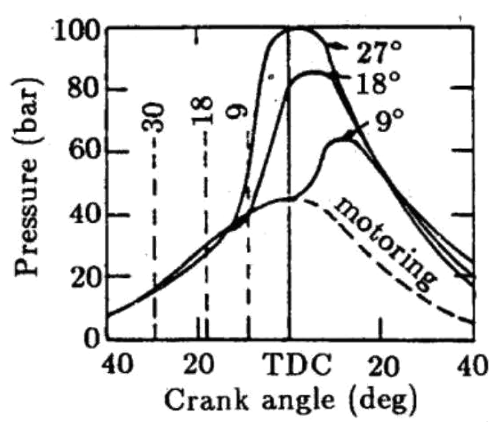
in degrees of crank travel the delay period increases as the engine operates at a higher rpm. The fuel pump is geared to the engine, and hence the amount of fuel injected during the delay period depends on crank degrees and not on absolute time. Hence, at high speeds, there will be more fuel present in the cylinder to take part in the second stage of uncontrolled combustion resulting in high rate of pressure rise.

3 Outputs

With an increase in engine output the air-fuel ratio decreases, operating temperatures increase and hence delay period decreases. The rate of pressure rise is unaffected but the peak pressure reached may be high.

4. Injection timing:

The effect of injection advance on the pressure variation is shown in Fig. for three injection advance timings of 90°, 18°, and 27° before TDC. The injected quantity of fuel per cycle is constant. As the pressure and temperature at the beginning of injection are lower for higher ignition advance, the delay period increases with increase in injection advance. The optimum angle of injection advance depends on many factors but generally it is about 20°bTDC.



Effect of Injection Timing on Indicator Diagram

5. Quality of Fuel used:

The physical and chemical properties of fuel play very important role in delay period. The most important property of fuel which is responsible for chemical delay is its self-ignition temperature. Lower the self-ignition temperature, lower the delay period.

The cetane number (CN) of the fuel is another important parameter which is responsible for the delay period. A fuel of higher cetane number gives lower delay period and provides smoother engine operation.

The effect of cetane number on the indicator diagram when injection timing is same is shown in adjacent figure.

The delay period for a fuel having CN = 50 is lowest and pressure rise is also smooth and maximum pressure rise is least as most of the fuel burns during controlled combustion.

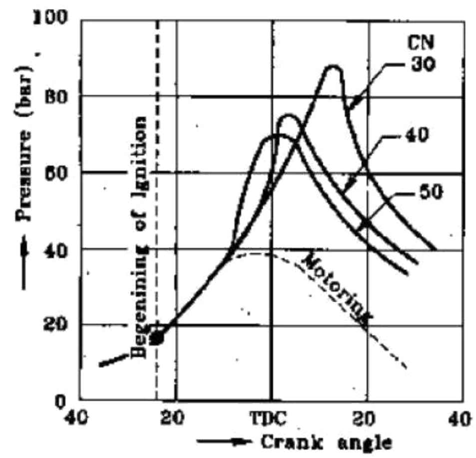
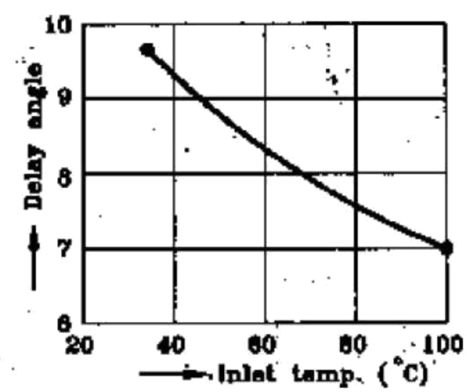


Fig. 18.2. Effect of CN on indicator diagram.

The other properties of fuel which affects the physical delay period are volatility, latent heat, viscosity and surface tension. The viscosity and surface tension are responsible for the better atomization whereas latent heat and viscosity are responsible for the rapid evaporation of fuel.

6. Intake Temperature

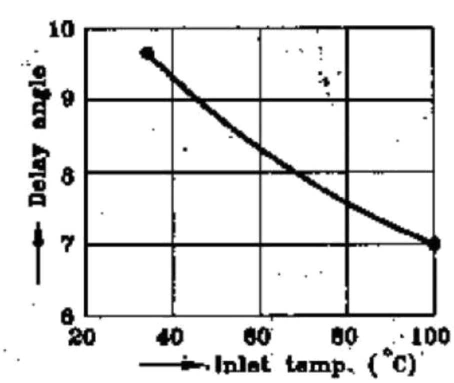
The delay period is reduced either with increased temperature. However, preheating of charge for this purpose is not desirable because it reduces the density of charge and volumetric efficiency and power output.



(a)

7. Intake pressure

Increase in intake pressure or supercharging reduces the auto ignition temperature and hence reduces the delay period. The peak pressure will be higher since the compression pressure will increase with intake pressure.



(a)

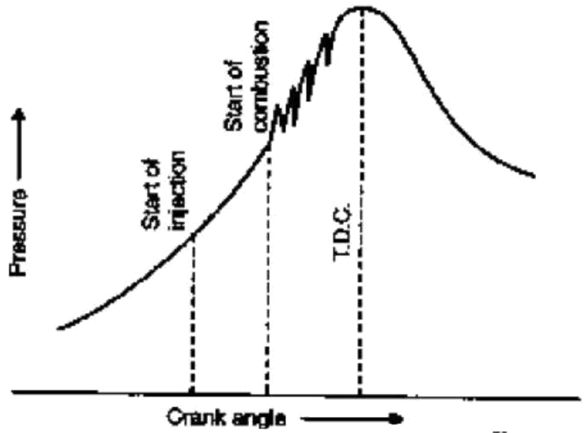
The following table gives the summary of the factors which influence the delay period in CI engine.

EFFECT OF VARIABLE ON DELAY PERIOD – SUMMARY

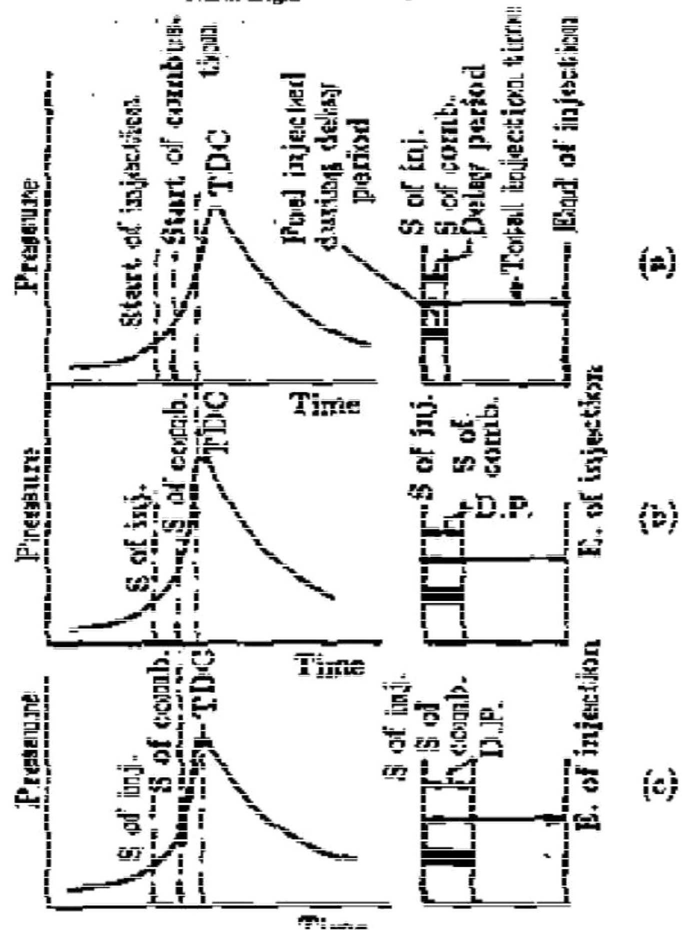
SL No	Increase in variables	Effect on Delay period	Reason
1	Cetane Number of fuel	Reduce	Reduces the self ignition temperature
2	Injection pressure	Reduce	Reduces the physical delay due to greater surface to volume ratio
3	Injection timing advance	Increase	Reduces the pressure and temperature when the injection begins
4	Compression ratio	Reduce	Increases air temperature and pressure and reduces auto ignition temperature
5	Intake temperature	Reduce	Increase air temperature
6	Jacket water temperature	Reduce	Increase wall and hence air temperature
7	Fuel temperature	Reduce	Increases chemical reaction due to better vaporization
8	Intake pressure	Reduce	Increases the density and also reduces the auto ignition temperature
9	Speed	Increase in terms of crank angle but reduces in terms of milliseconds.	Reduce loss of heat
10	Load (Fuel/air ratio)	Decrease	Increase the operating temperature
11	Engine size	Increase in terms of crank angle but little effect in terms of milliseconds.	Larger engines operate at normally slow speeds.
12	Type of combustion chamber	Lower for engines with pre-combustion chamber	Due to compactness of the chamber.

PHENOMENON OF DIESEL KNOCK (VTU Feb 2006)

Knocking is violet gas vibration and audible sound produced by extreme pressure differentials leading to the very rapid rise during the early part of uncontrolled second phase of combustion.



In C.I. engines the injection process takes place over a definite interval of time. Consequently, as the first few droplets injected are passing through the ignition lag period, additional droplets are being injected into the chamber. If the ignition delay is longer, the actual burning of the first few droplets is delayed and a greater quantity of fuel droplets gets accumulated in the chamber. When the actual burning commences, the additional fuel can cause too rapid a rate of pressure rise, as shown on pressure crank angle diagram above, resulting in Jamming of forces against the piston (as if struck by a hammer) and rough engine operation. If the ignition delay is quite long, so much fuel can accumulate that the rate of pressure rise is almost instantaneous. Such, a situation produces extreme pressure differentials and violent gas vibration known as knocking (diesel knock), and is evidenced by audible knock. The phenomenon is similar to that in the SI engine. However, in SI Engine knocking occurs near the end of combustion whereas in CI engine, knocking the occurs near the beginning of combustion.



Delay period is directly related to Knocking in CI engine. An extensive delay period can be due to following factors:

- A low compression ratio permitting only a marginal self ignition temperature to be reached.
- A low combustion pressure due to worn out piston, rings and bad valves
- Low cetane number of fuel
- Poorly atomized fuel spray preventing early combustion
- Coarse droplet formation due to malfunctioning of injector parts like spring
- Low intake temperature and pressure of air

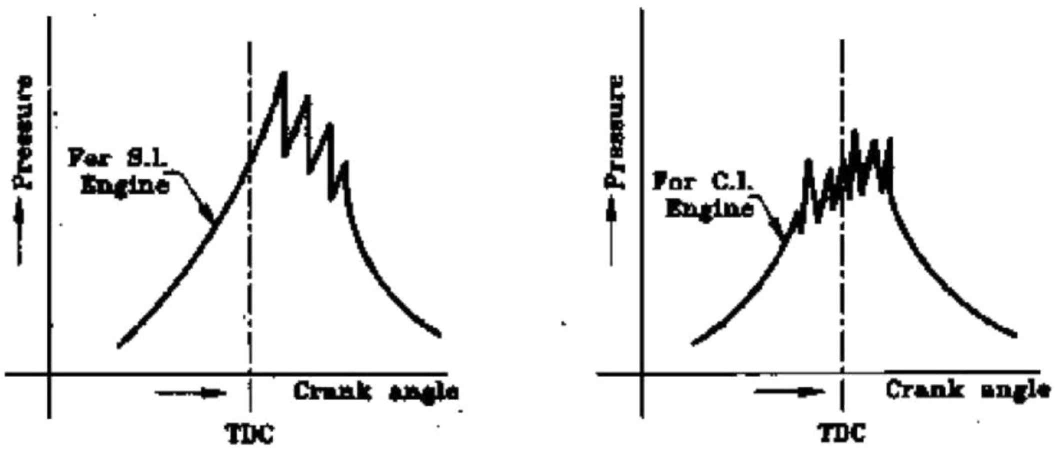
METHODS OF CONTROLING DIESEL KNOCK (VTU Feb 2006)

We have discussed the factors which are responsible for the detonation in the previous sections. If these factors are controlled, then the detonation can be avoided.

- Using a better fuel. Higher CN fuel has lower delay period and reduces knocking tendency.
- Controlling the Rate of Fuel Supply. By injecting less fuel in the beginning and then more fuel amount in the combustion chamber detonation can be controlled to a certain extent. Cam shape of suitable profile can be designed for this purpose.
- Knock reducing fuel injector : This type of injector avoid the sudden increase in pressure inside the combustion chamber because of accumulated fuel. This can be done by arranging the injector so that only small amount of fuel is injected first. This can be achieved by using two or more injectors arranging in out of phase.
- By using Ignition accelerators : C N number can be increased by adding chemical called dopes. The two chemical dopes are used are ethyl-nitrate and amyle –nitrate in concentration of 8.8 gm/Litre and 7.7 gm/Litre. But these two increase the NO_x emissions
- Increasing Swirl : Knocking can be greatly reduced by increasing swirl (or reducing turbulence). Swirl helps in knock free combustion.

COMPARISON OF KNOCK IN SI AND C ENGINES

It may be interesting to note that knocking in spark-ignition engines and compression-ignition engines is fundamentally due to the auto ignition of the fuel-air mixture. In both the cases, the knocking depends on the auto ignition lag of the fuel-air mixture. But careful examination of knocking phenomenon in SI and CI engines reveals the following differences:



(a) S.I. engine (b) C.I. engine

1.In spark ignition engines, auto ignition of end gas away from the spark plug, most likely near the end of combustion causes knocking. But in compression engines the auto ignition of charge causing knocking is at the start of combustion.

2.In order to avoid knocking in SI engine, it is necessary to prevent auto ignition of the end gas to take place at all. In CI engine, the earliest auto –ignition is necessary to avoid knocking

3.The knocking in SI engine takes place in homogeneous mixture, therefore , the rate of pressure rise and maximum pressure is considerably high. In case of CI engine, the mixture is not homogenous and hence the rate of pressure is lower than in SI engine.

4.In CI engine only air is compressed, therefore there is no question of Pre-ignition in CI engines as in SI engines.

5.It is lot more easy to distinguish between knocking and non-knocking condition in SI engines as human ear easily finds the difference. However in CI engines, normal ignition itself is by auto-ignition and rate of pressure rise under the normal conditions is considerably high (10 bar against 2.5 bar for SI engine) and causes high noise. The noise level becomes excessive under detonation condition. Therefore there is no definite distinction between normal and knocking combustion.

6.SI fuels should have long delay period to avoid knocking. CI fuels should have short delay period to avoid knocking.

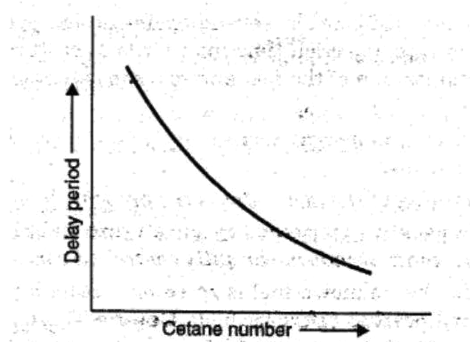
The following table gives a comparative statement of various characteristics that reduce knocking in SI and CI engines

S. No.	Factors Affecting Knock	S.I. Engines	C.I. Engines
1.	Self ignition temperature	High	Low
2.	Delay period of fuel	Long	Short
3.	Compression Ratio	Low	High
4.	Inlet Temperature	Low	High
5.	Inlet Pressure	Low	High
6.	Speed	High	Low
7.	Cylinder Size	Small	Large
8.	Combustion chamber wall Temperature	Low	High

Knock rating of CI fuels (CETANE NUMBER) (VTU July 2007/ Jan 07.)

The cetane number is a numerical measure of the influence the diesel fuel has in determining the ignition delay. Higher the cetane rating of the fuel lesser is the propensity for diesel knock. The cetane number of a diesel fuel is a measure of its ignition quality.

The cetane number of a fuel is the percentage by volume of cetane in a mixture of cetane [$C_{16}H_{34}$] and α-methylnapthalane [$C_{10}H_7\ CH_3$] that has same performance in the standard test engine as that of the fuel. Cetane is arbitrarily assigned a number 100 and originally α-methylnapthalane was given a number 0 but now reference fuels is heptamethylnonane (HMN) which is given a value of 15. HMN is used because it is more stable compound and has slightly better ignition quality.

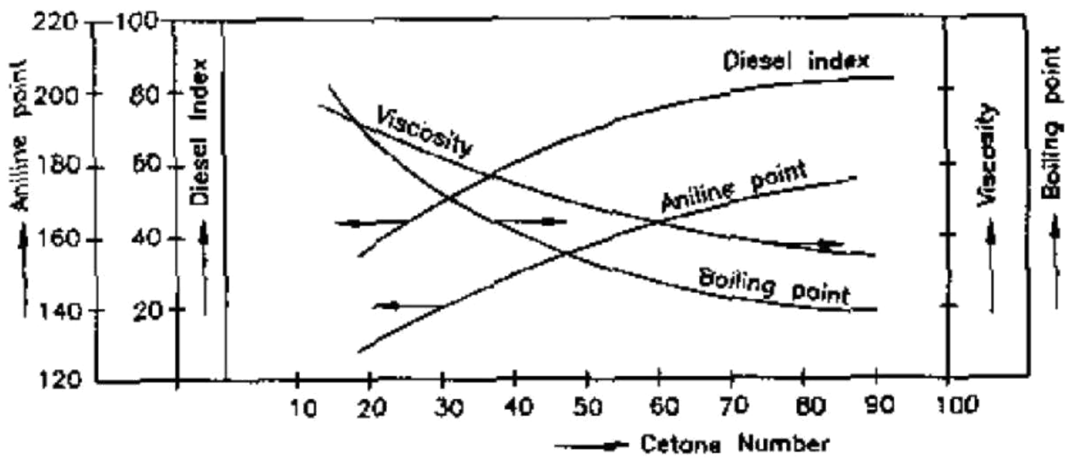


The relation between the cetane number and delay period is shown in adjacent figure

Cetane number 40 means a mixture containing 40 % cetane and 60 % of heptamethylnonane (HMN) by volume which gives same ignition delay as tested fuel. For high sped engine, cetane number of 50 is required, for medium speed engine about 30.

High octane number implies low cetane number . In other words good CI engine fuel is bad CI engine fuel. An approximate relationship between Cetane (CN) and octane (ON) number is given by $\boxed{CN = 60 - \frac{ON}{2}}$

The following graph shows relationship of other properties of fuel with CN



DIESEL INDEX (DI) (VTU Jan 2007)

Diesel index is a cheap method of predicting ignition quality. This scale is made possible because ignition quality is quite sensitive to hydrocarbons compositions. That is paraffin have high ignition quality and aromatic compounds have low ignition quality. Thus the diesel index gives an indication of ignition quality obtained from certain physical characteristics of fuel as opposed to an actual determination in the test engine. The index is derived from knowledge of aniline point and American petroleum Institute (API) gravity.

$$DI = \frac{\text{Aniline point}\ (^{\circ}F) \times \text{API gravity}}{100}$$

Aniline point of fuel is the temperature at which equal parts of fuel and pure aniline dissolve each other. It therefore gives an indication of chemical composition of fuel since the more "parafinnic" the fuel the higher solution temperature. Likewise, a higher API gravity reflects a low specific gravity and indicates a high paraffinic content, which corresponds to a good ignition quality.

Good SI engine fuel is a bad CI engine fuel

To reduce knocking Diesel oil should have low self ignition temperature and short time lag, whereas petrol should have high self ignition temperature and a long ignition lag.

In SI engine knocking occurs near the end of combustion, where as in CI engine this occurs in the beginning of combustion. Because of this dissimilarity in the time of starting of knock in SI and CI engines . The conditions which reduce the knock tendency in SI engine will increase the knocking tendency in CI engine.

Diesel has a high cetane number (40-60) and low octane number(30) and petrol has high Octane number (80-90) ad low cetane number(20).

Figure shows typical indicator diagram of a diesel engine with sharp pressure oscillating during the combustion caused by shock waves when using petrol

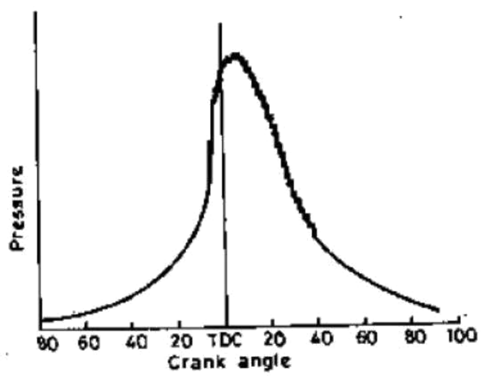
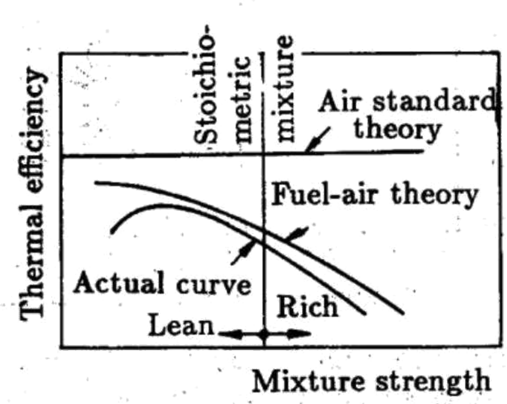


Fig. 6.11. Indicator diagram of diesel engine when using petrol.

Weak mixture gives better efficiency in CI engine- (July 2007)

As the mixture is made lean (less fuel) the temperature rise due to combustion will be lowered as a result of reduced energy input per unit mass of mixture. This will result in lower specific heat. Further, it will lower the losses due to dissociation and variation in specific heat. The efficiency is therefore, higher and, in fact, approaches the air-cycle efficiency as the fuel-air ratio is reduced as shown in adjacent figure.



Thermodynamic analysis of the engine cycles has clearly established that operating an engine with a leaner air-fuel ratio always gives a better thermal efficiency but the mean effective pressure and the power output reduce. Therefore, the engine size becomes bigger for a given output if it is operated near the stoichiometric conditions, the A/F ratio in certain regions within the chamber is likely to be so rich that some of the fuel molecules will not be able to find the necessary oxygen for combustion and thus produce a noticeably black smoke. Hence the CT engine is always designed to operate with an excess air, of 15 to 40% depending upon the application. The power output curve for a typical CI engine operating at constant speed is shown in Fig. given below. The approximate region of A/F ratios in which visible black smoke occurs is indicated by the shaded area.

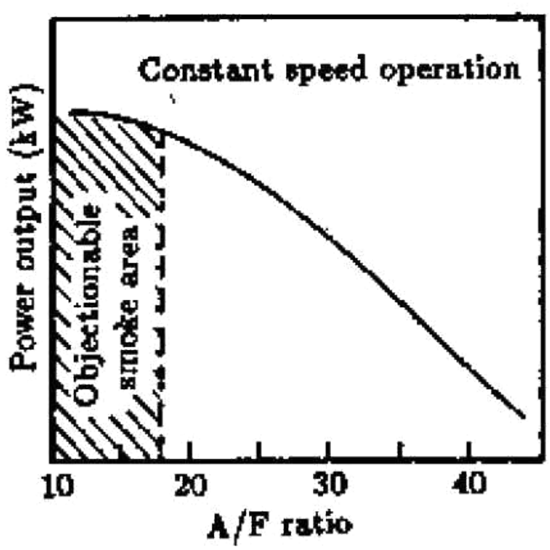


Fig.10.10 Effect of A/F Ratio on Power Output of a CI Engin

Testing of I.C.Engines

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

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

(i) Friction Power,

(ii) Indicated Power,

(iii) Brake Power,

(iv) Specific Fuel Consumption,

(v) Air – Fuel ratio

(vi) Thermal Efficiency

(vii) Mechanical Efficiency,

(viii) Volumetric Efficiency,

(ix) Exhaust gas emissions,

(x) Noise

1.3. Measurement of Performance Parameters in a Laboratory

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

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

(i) Willan's line method;

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

(iii) Motoring test;

(iv) Retardation test;

(v) Morse Test.

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

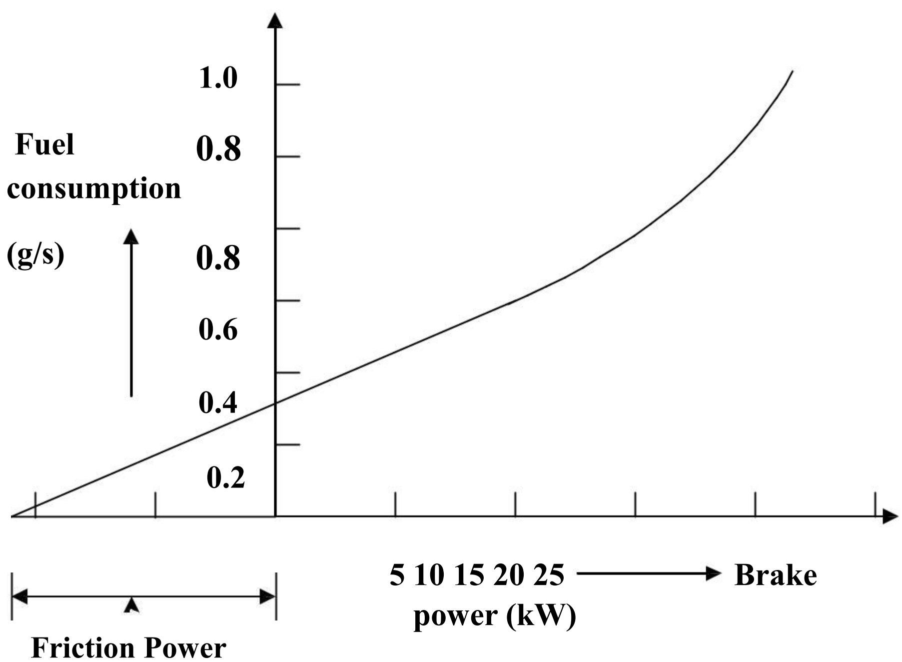


Figure.1 Willan's line method

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

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

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

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

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

If there are *k* cylinders, then

Total indicated power
when all the cylinders are working = $ip_1 + ip_2 + ip_3 + \ldots\ldots\ldots\ldots.. + ip_k = \sum_{j=1}^{k} ip_j$

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

where ip_j is the indicated power produced by j th cylinder, k is the number of cylinders,

B_t is the total brake power when all the cylinders are producing power and F_t is the total frictional power for the entire engine.

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

we can write $\sum_{j=2}^{k} ip_j = B_1 - F_t$……………………………………..(2)

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

F_t = Total frictional power.

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

$$ip_1 = B_t – B_1 \quad \text{……………………………………..(3).}$$

Similarly we can find the indicated power of all the cylinders, viz., ip_2, ip_3, …..ip_k. Then the total indicated power is calculated as

$$(ip)_{total} = \sum_{j=1}^{k} ip_j \quad \text{……………………………………(4)}$$

The frictional power of the engine is therefore given by

$$F_t = (ip)_{total} – B_t \quad \text{……………………………………(5)}$$

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

1.4. MEASUREMENT OF INDICATED POWER

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

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

A typical *p-v* diagram taken by a mechanical indicator is shown in Figure 2.

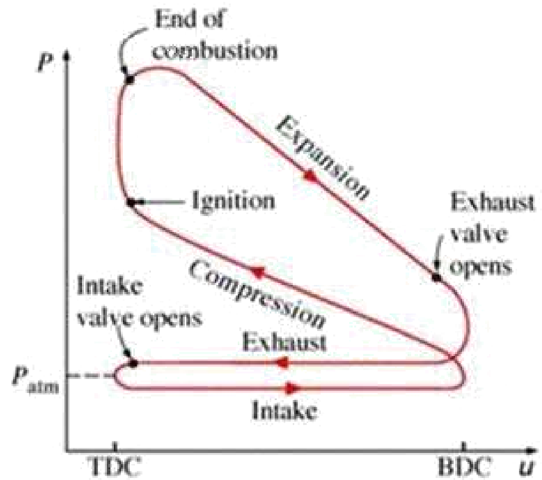


Figure.2 p-v diagram taken by mechanical indicator

The areas, the positive loop and negative loop, are measured with the help of a planimeter and let these be A_p and A_n cm^2 respectively, the net positive area is $(A_p - A_n)$. Let the actual length of the diagram as measured be L cm, then the average height of the net posiive area is given by

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

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

$$Imep=(A_p-A_n)*S/L \qquad .....(6)$$

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

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

$$P_m=A_p*S_p/L-A_n*S_n/L \quad .....(7)$$

Where S_p = Spring strength used for taking *p-v* diagram of positive loop, (N/m^2 per cm)

S_n = Spring strength used for taking *p-v* diagram of negative loop, (N/m^2 per cm)

A_p = Area in Cm2 of positive loop taken with spring of strength S_p

A_n = Area in Cm2 of positive loop taken with spring of strength S_n

Sometimes spring strength is also noted as spring constant.

The IP developed by the engine is given by

$$IP=P_mLAn/L \quad .....(8)$$

Where ‘n’ is the number of working strokes per second.

The explanation of this expression is already given in the last chapter.

1.5. MEASUREMENT OF B.P

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

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

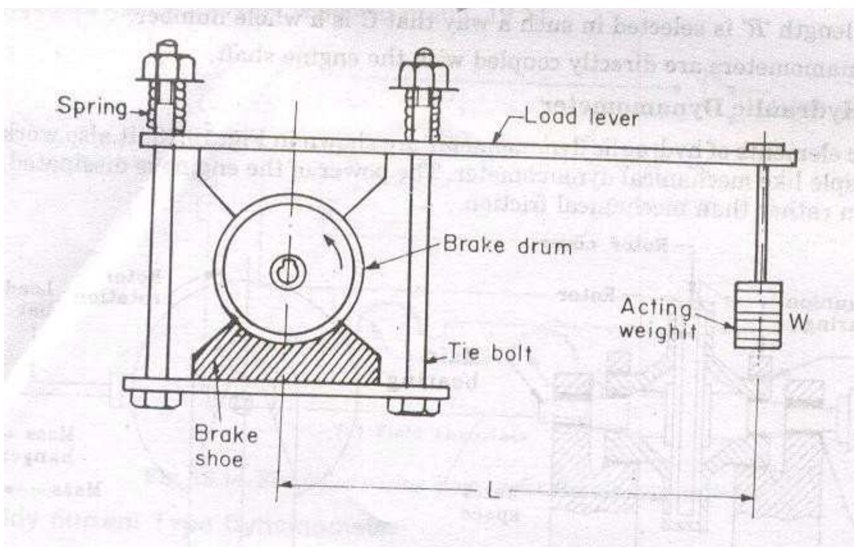


Fig. Prony-brake arrangement.

Figure.3

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

The BP of the engine is given by

B.P (brake power) = 2* *N*T/60 watts =2* *N*T/60*1000 Kw ….. (9)

Where T = ($W.L$) (N-m)

Where W = Weight on load carrier, (N)

And L = Distance from the centre of shaft to the point of load-meter in meters.

The prony brake is inexpensive, simple in operation and easy to construct. It is, therefore, used extensively for testing of low speed engines. At high speeds, grabbing and chattering of the band occur and lead to difficulty in maintaining constant load. The main disadvantage of the prony brake is its constant torque at any one band pressure and therefore its inability to compensate for varying conditions.

1.5.1 Hydraulic Dynamometer.

The BP of an engine coupled to the dynamometer is given by

B.P (brake power) = 2* *N*W*R/60*1000 = WN(2* *R/60*1000) Kw

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

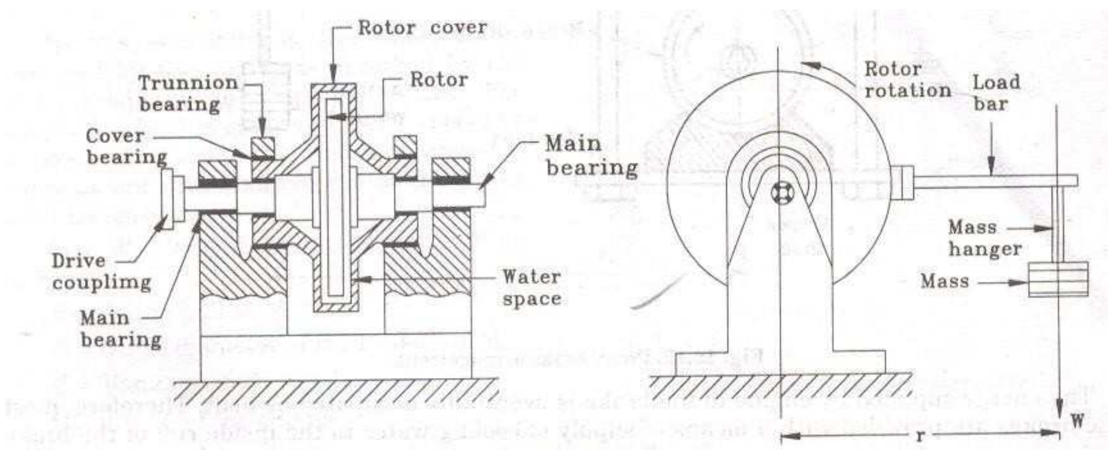


Figure.4 Hydraulic dynamometer

In the hydraulic dynamometer, as the arm length (*R*) is fixed, the factor [2 *R*/(60x1000)] is constant and its value is generally given on the name plate of the dynamometer by the manufacturer and is known as brake or dynamometer constant. Then the BP measured by the dynamometer is given by

$$B.P = \frac{WN}{K} \qquad \ldots..(10)$$

Where *W* = Weight measured on the dynamometer, N

K = Dynamometer constant (60*1000/2*pi*R)

and *N* = RPM of the engine.

The arm length '*R*' is selected in such a way that *K* is a whole number.
These dynamometers are directly coupled with the engine shaft.

1.5.2 Electric Dynamometer:

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

equal and opposite to the torque exerted on the armature, which is driven by the engine which is shown in
Figure 5.

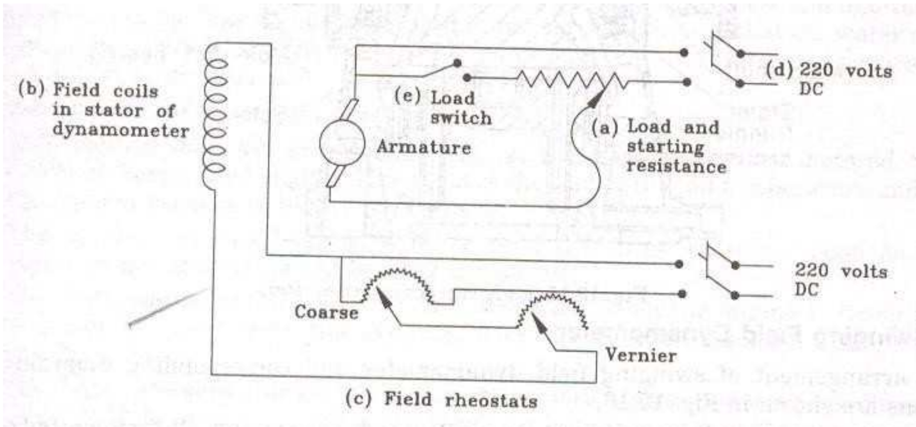


Fig. Simplified wiring diagram for electric dynamometer.

Figure.5

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

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

1.5.3 Eddy current Type Dynamomter

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

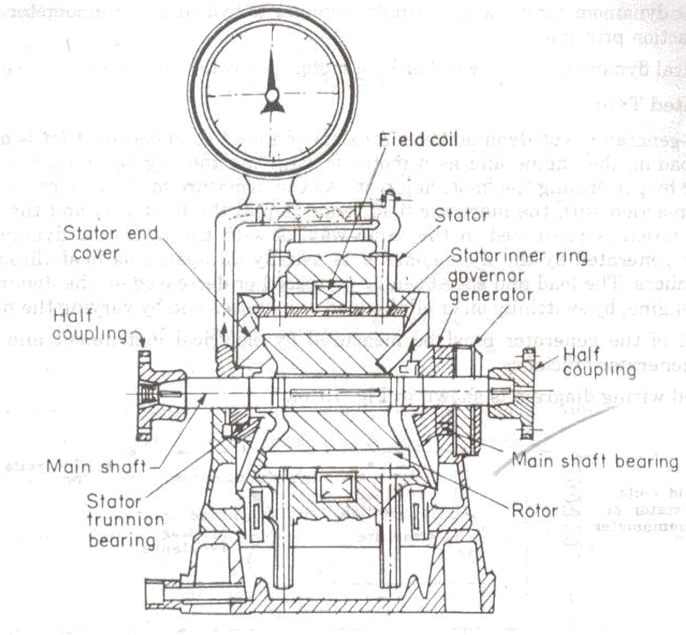


Fig. Eddy-current dynamometer.

Figure.6

The power output of eddy-current dynamometer is given by the equation where *C* is eddy-current dynamometer constant.

The advantages of eddy-current dynamometer are listed below:

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

1.5.4 Swinging Field Dynamometer

The arrangement of swinging field dynamometer and corresponding diagram of electric connections are shown in Figure 7.

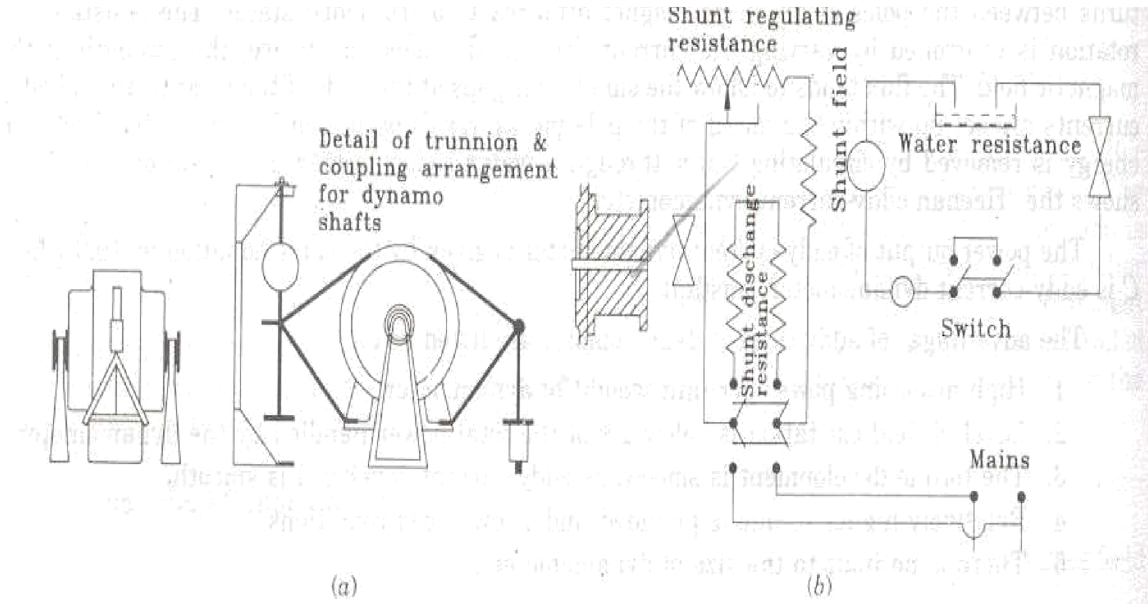


Fig. (*a*) Swinging-field dynamometer (*b*) Diagram of connections for swinging-field electrical dynamometer.

Figure.7

A swinging field DC dynamometer is basically a DC shunt motor. It is supported on trunnion bearings to measure the reaction torque that the outer casing and field coils tend to rotate with the magnetic drag. Therefore, it is named as “Swinging field”. The Torque is measured with an arm and weighting equipment in the usual manner.

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

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

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

This method is used in multi-cylinder engines to measure I.P with out the use of indicator. The BP of the engine is measured by cutting off each cylinder in turn. If the engine consists of 4-cylinders, then the BP of the engine should be measured four times cutting each cylinder turn by turn. This is applicable to petrol as well as for diesel engines. The cylinder of a petrol engine is made inoperative by “shorting” the spark plug whereas in case of diesel engine, fuel supply is cut-off to the required cylinder.

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

then $(B.P)_n = (I.P)_n - (F.P)_n$ …..(11)

Where F.P is the frictional power per cylinder.

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

When cylinder "1" is cut off; then

$$(B.P)_{n-1} = (I.P)_{n-1} - (F.P)_n \quad \ldots..(12)$$

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

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

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

1.7 MEASUREMENT OF AIR-CONSUMPTION

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

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

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

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

The pressure different causing the flow through the orifice is measured with the help of a water monometer. The pressure difference should be limited to 10cm of water to make the compressibility effect negligible. Let

A_o = Area orifice in m^2.; h_w = Head of water in cm causing the flow.

C_d = Coefficient of discharge for orifice. ; d = Diameter of orifice in cm.

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

Head in terms of meters of air is given by

$$H.\rho_a = \frac{h_w}{100}\rho_w \; ; \; H = \frac{h_w}{100} \times \frac{\rho_w}{\rho_a} = \frac{h_w}{100} \times \frac{1000}{\rho_a} = \frac{10\,h_w}{\rho_a} \text{ m of air}$$

Th

e velocity of air passing through the orifice is g given by

$$v = \sqrt{2gH} \text{ m/Sec} = \sqrt{2g \times \frac{10\, h_w}{\rho_a}} \text{ m/Sec}$$

The volume of air passing through the orfice is given by

$$V_a = A_0 \times v \times C_d = A_0 \times C_d \times \sqrt{2g \times \frac{10\, h_w}{\rho_a}} = 14.01 \times A_0 \times C_d \sqrt{\frac{h_w}{\rho_a}} \text{ cu.m/Sec}$$

$$= 840.428\, A_0 \times C_d \sqrt{\frac{h_w}{\rho_a}} \text{ m}^3\text{/min}$$

The volum

etric efficiency of the engine

= Actual volume of air taken in

$14.01\, A_0 \times C_d \; \frac{h_w}{}$

$\frac{a}{D^2 L \times N n}$

60

4

Displacement volume

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

Mass of air passing through the orifice is given by

$m_a = V_a \times \rho_a = \ldots$ 14.01 ´ 400$_2$ ... d^2 ... $\times C_d$ ... h_w ... $\times \rho_a$ = 11.003 ... $C_d \times d$ ... $\sqrt{\rho_a h_w}$

$$= 0.0011 C_d \times d^2 \sqrt{h_w \rho_a}\ Kg / Sec = 0.066 \times C_d \times d^2 \sqrt{h_w \rho_a}\ kg / min \ldots\ldots(13)$$

Where d is in cm; h_w is in cm of water and Pa is in

kg/m^3 The density of atmospheric air is given by

$$\rho_a = \frac{p_a \times 10^5}{287 \times T_a}$$

Where P_a is the atmospheric pressure in bar and T_a is the atmospheric temperature in

K. Substituting the value of ρ_a in Eq. (13)

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

$$= 1.23 \times C \ldots \sqrt{\frac{h_w\, p_a}{\ldots}}\ \text{kg/mi}\ldots$$

d — n

$\times$

d

2 T_a $\sqrt{\ }$

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

The measurement of air consumption by the orifice chamber method is used for:

(a) The determination of the actual A : F ratio of the engine at running condition.
(b) The weight of exhaust gases produced, and
(c) The volumetric efficiency of the engine at the running condition.

The mass of air supplied per kg of fuel used can also be calculated by using the following formula if the volumetric analysis of the exhaust gases is known.

$$m_a \text{ / Kg of fuel} = \frac{N\ C}{33\,(C_1 + C_2)} \quad \ldots(14)$$

Where N = Percentage of nitrogen by volume in exhaust gases.

C_1 = Percentage of carbon dioxide by volume in exhaust gases.

C_2 = Percentage of carbon monoxide by volume in exhaust gases.

C = Percentage of carbon in fuel by weight.

If $C_2 = 0$ then; $m_a = \dfrac{N\ C}{33 \times C_1}$ …(15)

1.8 MEASUREMENT OF FUEL CONSUMPTION

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

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

$$\text{Fuel consumption kg/hr} = \frac{X_{cc} \text{ X Sp. gravity of fuel}}{1000 \text{ x } t}$$

1.9 MEASUREMENT OF HEAT CARRIED AWAY BY COOLING WATER

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

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

$$Q_w = C_p m_w (T_{wo} - T_{wi}) \text{ kJ/min.}$$

Where

M_w = mass of water/min.

T_{wi} = Inlet temperature of water, ^oC

T_{wo} = Out let temperature of water, ^oC

C_p = Specific heat of water.

1.10 MEASUREMENT OF HEAT CARRIED AWAY BY EXHAUST GASES

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

m

$=$

$N X$
C

33

And heat carried away by the exhaust gas per kg of fuel supplied can be calculated as

$Q_g = (m_a + 1)\, C_{pg}\, (T_{ge} - T_a)$ kJ/kg of fuel ….(16)

Where $(m_a + 1)$ = mass of exhaust gases formed per kg of fuel supplied to engine C_{pg} = Specific heat of exhaust gases

T_{ge} = Temperature of exhaust gases coming out from the engine ^oC.

T_a = Ambient temperature ^oC or engine room temperature.

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

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

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

1.10.1Exhaust Gas Calorimeter

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

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

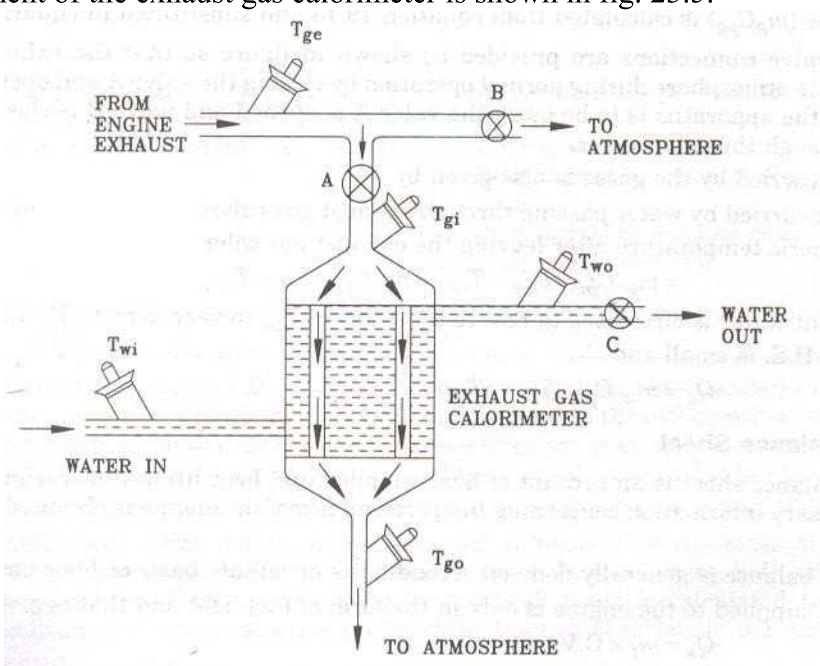


Fig. Exhaust gas calorimeter.

Figure.8

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

If it is assumed that the calorimeter is well insulated, there is no heat loss except by heat transfer from the exhaust gases to the circulating water, then
Heat lost by exhaust gases = Heat gained by circulating water.
Therefore $m_g , C_{pg} (T_{gi} – T_{go}) = m_w. C_{pw} (T_{wo} – T_{wi})$
Where T_{gi} = The temperature of the exhaust gases entering the calorimeter, ^oC
T_{go} = The temperature of the exhaust gases leaving the calorimeter, ^oC
T_{wi} = The temperature of water entering the calorimeter, ^oC
T_{wo} = The temperature of water leaving the calorimeter, ^oC
m_w = Mass of water circulated through the exhaust gas calorimeter, generally measured.
m_g = Mass of exhaust gases (unknown)
C_{pg} = specific heat of exhaust gases.
C_{pw} = Specific heat of water.

$$m_g = \frac{m_w C_{pw} (T_{wo} - T_{wi})}{C_{pg} (T_{gi} - T_{go})} \quad \ldots(17)$$

As all the quantities on the RHS are known the gas flow rate can be determined.
Then the heat carried away by the exhaust gases is given by

$$Q_g = m_g \ C_{pg.} \ (T_{ge} - T_a) \quad \ldots.(18)$$

Where T_{ge} = Temperature of exhaust gases just leaving the engine exhaust valve, ^oC T_a = Ambient temperature, ^oC

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

The heat carried by the gases is also given by

Q_g = Heat carried by water passing through exhaust gas calorimeter + Heat in exhaust gases above atmoshpheric temperature after leaving the exhaust gas calorimeter.

$$= m_w C_{pw} (T_{wo} - T_{wi}) + m_g C_{pg} (T_{go} - T_a) \quad \ldots(19)$$

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

$$Q_g = m_w C_{pw} (T_{wo} - T_{wi}) \quad \ldots.(20)$$

1.11 HEAT BALANCE SHEET

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

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

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

$$Q_s = m_f \text{ X CV}$$

Where mf is the mass of fuel supplied per minute or per sec. and CV is the lower calorific value of the fuel.

The various ways in which heat is used up in the system is given by

(a) Heat equivalent of BP = kW = kJ/sec. = 0 kJ/min.
(b) Heat carried away by cooling water

$= C_{pw} \times m_w (T_{wo} - T_{wi})$ kJ/min.

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

(c) Heat carried away by exhaust gases

$= m_g C_{pg} (T_{ge} - T_a)$ (kJ/min.) or (kJ/sec)

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

T_g = Temperature of burnt gases coming out of the engine.

T_a = Ambient Temperature.

C_{pg} = Sp. Heat of exhaust gases in (kJ/kg-K)

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

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

Heat input per minute	*kcal (kj)*	*%*	*Heat expenditure per minute*	*kcal (kj)*	*%*
Heat supplied by the combustion fuel	Q_s	100%	(a) Heat in BP. (b) Heat carried by jacket cooling water (c) Heat Carried by exhaust gases (d) Heat unaccounted for $= Q_s - (a + b + c)$	-- -- -- --	-- -- -- --
Total	Q_s	100%			100%

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

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

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

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

Thus, indicated specific fuel consumption = isfc = $\dfrac{3600\,\dot{m}}{ip}$ kg/kWh …..(13)

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

Thus, brake specific fuel consumption = bsfc = $\dfrac{3600\,\dot{m}}{bp}$ kg/kWh ……..(14)

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

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

Thus $_{ith}$ = $\dfrac{\text{Indicated Power in kW}}{\text{(Massflowrateoffuelinkg/s)x(CalorificvalueoffuelinkJ/kg)}}$

Or$_{ith}$ = $\dfrac{ip}{m \times CV}$ ……………………………………………..(15)

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

Or $_{bth}$ = $\dfrac{bp}{m \times CV}$ ……………………………………………..(16)

1.15.Mechanical Efficiency: Mechanical efficiency takes into account the mechanical losses in an engine. The mechanical losses include (i) frictional losses, (ii) power absorbed by engine auxillaries like fuel pump, lubricating oil pump, water circulating pump, magneto and distributor, electric generator for battery charging, radiator fan etc., and (iii) work requited to charge the cylinder with fresh charge and work for discharging the exhaust gases during the exhaust stroke. It is defined as the ratio of brake power to indicated power. Thus

$$_{mech} = \frac{bp}{ip} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(17)$$

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

Thus $$_{v} = \frac{V_{actual}}{V_{th}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(18)$$

where n is the number of intake strokes per minute and V_S is the stroke volume of the piston.

2. Illustrative examples:

Example 1:- *The following observations have been made from the test of a four cylinder, two – stroke petrol engine. Diameter of the cylinder = 10 cm; stroke = 15 cm; speed = 1600 rpm; Area of indicator diagram = 5.5 cm^2; Length of the indicator diagram = 55 mm; spring constant = 3.5 bar/cm; Determine the indicated power of the engine.*

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

Solution: Indicated mean effective pressure = $p_{im} = \dfrac{a\,k_s}{l_d}$

$$\text{or} \quad p_{im} = \frac{5.5 \times 3.5}{5.5} = 3.5 \text{ bar} = 3.5 \times 10^{5} \text{ N / m}^2$$

$$\text{Indicated power} = ip = \frac{p_{im}\,LAnK}{60{,}000} = \frac{3.5 \times 10^5 \times 0.15 \times (\ /4) \times 0.1^2 \times 1600 \times 4}{60{,}000}$$

$= 43.98$ kW

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

Given:- Volume of fuel consumed/hour = y/t = $8 \times 10^{3} / 3600$ cc/s ;

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

Specific gravity of petrol = s = 0.75

To find: $_{ith}$;

Solution: Mass of fuel consumed = $m = \dfrac{y\,s}{1000\,t} = \dfrac{8 \times 10^{3} \times 0.75}{1000 \times 3600} = 1.67 \times 10^{-3}$ kg/s.

Indicated thermal efficiency = $_{ith} = \dfrac{ip}{m\,CV} = \dfrac{25}{1.67 \times 10^{-3} \times 44000}$

$= 0.3402 = 34.02$ %.

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

Given:- Diameter = d = 80 $x10^{-3}$ = 0.008 m ; stroke = L = 0.110 m; T = 23.5 N-m;

To find (i) bmep ; (ii) bmep if bp = 4 kw and N= 1500 rpm.

Solution: (i) Relation between brake power (bp) and brake mean effective pressure (bmep) is given by

2
N
T (bmep)LAn
bp =

----- = ---------------
6
0
,
0
0
0 60,000

Hence bmep =
(2 NT) / (LAn)
= (2 NT) / {(L
d^2 /4) N/2}

16T ---------------16x23.5 = 5.34 x 10 ^{5} N / m^2 = 5.34
= ------------- = bar
d^2L 0.08 ^{2} x 0.11

(ii) when bp = 4
kw and N =
1500 rpm, we
have

60,000 bp 60,000 x 4
bmep = -------------- = ---
0.110 x (/
4) x 0.08
^{2} x
L
An (1500 / 2)

= 5.79 x
10 ^{5} N/m^2
= 5.79
bar.

60,000 bp 60,000 x 4
Also bp = 2 or T = ---------------- = ------------------- =
NT / 60,000
2
5

.
4
6

N

–

m
.

2 N 2 x x 1500

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

Given:- y = 10 cc ; t = 20.4 s ; V_a = 0.1 m^3; t_a = 16.3 s ; W = 7 N ; N = 3000 rpm;

$_a$ = 1.175 kg/m^3; s = 0.7 ; CV = 43 x 10^3 kJ/kg; Dynamometer constant = C = 5000.

To find:- (i) m_a / m_f ; (ii) bsfc ; (iii) $_{bth.}$

0.1 x 1.175

Solution: (i) $^{-3}$ kg/s.

Mass of air consumed = m_a = ---------------- = 7.21 x 10 $^{-3}$ kg/s

16. 3

y s 10 x 0.7

Mass of fuel consumed = m_f = --------- = ------------------= 0.343 x 10

1000 t 1000 x 20.4

$$\text{Air fuel ratio} = \frac{m_a}{m_f} = \frac{7.21 \times 10^{-3}}{0.343 \times 10^{-3}} = 21$$

(ii) Brake power

$$= bp = WN / C = \frac{7 \times 3000}{5000} = 4.2 \text{ kW}$$

$$bsfc = \frac{m_f \times 3600}{bp} = \frac{0.343 \times 10^{-3} \times 3600}{4.2} = 0.294 \text{ kg/kWh}$$

$$\text{(iii) } b_{ith} = \frac{bp}{m_f\, CV} = \frac{4.2}{0.343 \times 10^{-3} \times 43 \times 10^{3}} = 0.2848 = 28.48\ \%.$$

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

Given:- K = 6 ; n = N /2 ; d = 8 cm ; L = 10 cm ; V_c = 70 cc ; N = 4000 rpm ; m_f = 20 kg/h ; T = 150 N-m ; CV = 43000 kJ/kg ;

To find:- (i) bp ; (ii) bmep ; (iii) $_{bth}$; (iv) $_{Relative.}$

Solution:

(i) bp = $\dfrac{2\ \ NT}{60{,}000} = \dfrac{2 \text{ x } \ \text{ x } 4000 \text{ x } 150}{60{,}000}$

$= 62.8$ kW

(ii) bmep = $\dfrac{60{,}000 \text{ bp}}{L\,A\,n\,K} = \dfrac{60{,}000 \text{ x } 62.8}{0.1 \text{ x } (\ \ /4) \text{ x } 0.08^2 \text{ x } (4000/2) \text{ x } 6}$

$= 6.25 \text{ x } 10^5 \text{ N/m}^2 = 6.25$ bar

(iii) $_{bth} = \dfrac{\text{bp}}{m_f\,CV} = \dfrac{62.8}{(20 / 3600) \text{ x } 43{,}000} = 0.263 = 26.3$ %.

(iv) Stroke volume = $V_s = (\ \ /4)\, d^2 L = (\ \ /4) \times 8^2 \times 10 = 502.65$ cc

$$\text{Compression Ratio of the engine} = R_c = \frac{V_s + V_c}{V_c} = \frac{502.65 + 70}{70} = 8.18$$

Air standard efficiency of Otto cycle = $\ _{Otto} = 1 - (1/R_c^{\ \ -1})$

$$= 1 - \frac{1}{8.18^{0.4}} = 0.568 = 56.8\ \%$$

Hence Relative efficiency = $\ _{Relative} = \ _{bth} / \ _{Otto} = 0.263 / 0.568 = 0.463 = 46.3$ %.

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

Given:- K = 8 ; Four stroke hence n = N/2 ; d = 0.09 m; L = 0.08 m; $R_c = 7$; N = 4500 rpm; Brake arm = R = 0.54 m ; t = 10 min ; Brake load = W = (42 x 9.81) N

$m_f = 4.4$ kg ; CV = 44,000 kJ/kg ; $T_a = 27 + 273 = 300$ K ; $p_a = 1$ bar; $\dot{m}_a = 6$ kg/min;

To find:- (i) bp ; (ii) bmep ; (iii) bsfc ; (iv) bsac ; (v) $\ _v$; (vi) $\ _{bth}$; (vii) $\dot{m}_a / \dot{m}_f$

Solution:

$$\text{(i) bp} = \frac{2\ \ NT}{60{,}000} = \frac{2\ \ NWR}{60{,}000} = \frac{2 \times \ \ \times 4500 \times (42 \times 9.81) \times 0.54}{60{,}000}$$

$= 104.8$ kW

$$\text{(ii) bmep} = \frac{60{,}000\ bp}{L\,A\,n\,K} = \frac{60{,}000 \times 104.8}{0.08 \times (\ \ /4) \times 0.09^2 \times (4500/2) \times 8}$$

$= 6.87 \times 10^5\ N/m^2 = 6.87$ bar.

(iii) mass of fuel consumed per unit time = $\dot{m}_f = m_f / t = 4.4 \times 60 / 10$ kg/h

= 26.4 kg/h

Brake specific fuel consumption = bsfc = $\dfrac{\dot{m}_f}{bp} = \dfrac{26.4}{104.8}$ = 0.252 kg / kWh

(iv) brake specific air consumption = bsac = $\dfrac{\dot{m}_a}{bp} = \dfrac{6 \times 60}{104.8}$ = 3.435 kg / kWh

(v) $_{bth} = \dfrac{bp}{\dot{m}_f \; CV} = \dfrac{104.8}{(26.4 / 3600) \times 44{,}000}$ = 0.325 = 32.5 %.

(vi) Stroke volume per unit time = $\dot{V}_s = (d^2/4)\, L\, n\, K$

$= \dfrac{}{4} \times (0.09^2) \times 0.08 \times (4500 / 2) \times 8$

= 9.16 m^3 / min.

Volume flow rate of air per minute = $\dot{V}_a = \dfrac{\dot{m}_a R_a T_a}{} = \dfrac{6 \times 286 \times 300}{}$

$p_a \qquad\qquad 1 \times 10^5$

$= 5.17\ m^3 / min$

Volumetric efficiency = $_{v} = \dot{V}_a / \dot{V}_s =$ 5.17 / 9.16 = 0.5644 = 56.44 %.

(vii) Air fuel ratio = $\dot{m}_a / \dot{m}_f$ = 6 /(4.4 / 10) = 13.64

Example 2.7:- *A gasoline engine working on four- stroke develops a brake power of 20.9 kW. A Morse test was conducted on this engine and the brake power (kW) obtained when each cylinder was made inoperative by short circuiting the spark plug are 14.9, 14.3, 14.8 and 14.5 respectively. The test was conducted at constant speed. Find the indicated power, mechanical efficiency and brake mean effective pressure when all the cylinders are firing. The bore of the engine is 75mm and the stroke is 90 mm. The engine is running at 3000 rpm.*

Given:- brake power when all cylinders are working = B_t = 20.9 kW ;

Brake power when cylinder 1 is inoperative = B_1 = 14.9 kW ;

Brake power when cylinder 2 is inoperative = B_2 = 14.3 kW ;

Brake power when cylinder 3 is inoperative = B_3 = 14.8 kW ;

Brake power when cylinder 4 is inoperative = B_4 = 14.5 kW ;

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

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

Solution:

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

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

= 25.1 Kw

(ii) $_{mech} = \dfrac{B_t}{(ip)_{total}} = \dfrac{20.9}{25.1} = 0.833 = 83.3\ \%$

(iii) $\text{bmep} = \dfrac{60{,}000\ B_t}{L\ A\ n\ K} = \dfrac{60{,}000 \times 20.9}{0.09 \times (\ /4) \times 0.075^2 \times (3000/2) \times 4}$

$= 5.25 \times 10^5\ N/m^2 = 5.25$ bar.

Example 2.8:- *The following observations were recorded during a trail of a four – stroke, single cylinder oil engine.*
Duration of trial = 30 min ; oil consumed = 4 litres ; calorific value of oil = 43 MJ/kg ; specific gravity of fuel = 0.8 ; average area of the indicator diagram = 8.5 cm^2; length of the indicator diagram = 8.5 cm; Indicator spring constant = 5.5 bar/cm; brake load = 150 kg; spring balance reading = 20 kg; effective brake wheel diameter = 1.5 m ; speed = 200 rpm ; cylinder diameter = 30 cm ; stroke = 45 cm ; jacket cooling water = 10 kg/min ; temperature rise of cooling water = 36 C. Calculate (i) indicated power, (ii) brake power, (iii) mechanical efficiency, (iv) brake specific fuel consumption, (v) indicated thermal efficiency, and (vi) heat carried away by cooling water.

Given:- t = 30 min ; y = 4000 cc; CV = 43 x10 ^{3} kJ/kg; s = 0.8 ; area of the diagram = a = 8.5 cm^2; length of the diagram = l_d = 8.5 cm ; indicator spring constant = k_s = 5.5 bar / cm; W = 150 x 9.81 N ; Brake radius = R = 1.5 / 2 = 0.75 m; N = 200 rpm ; d = 0.3 m ;

L = 0.45 m ; $\dot{m}_w$ = 10 kg/min ; T_w = 36 C; Spring Balance Reading = S = 20 x 9.81 N

To find:- (i) ip ; (ii) bp ; (iii) $_{mech}$; (iv) bsfc ; (v) $_{ith}$; (vi) $\dot{Q}_w$

Solution:

(i) $p_{im} = \frac{a}{l_d} k_s = \frac{8.5}{8.5} \times 5.5 = 5.5 \text{ bar} = 5.5 \times 10^5 \text{ N/m}^2$

$$ip = \frac{p_{im} L A n K}{60{,}000} = \frac{5.5 \times 10^5 \times 0.45 \times (\ / 4) \times 0.3^2 \times (200 / 2) \times 1}{60{,}000}$$

= 29.16 kW

$$\text{(ii) bp} = \frac{2\ N(W - S) R}{60{,}000} = \frac{2 \times\ \times 200 \times (150 - 20) \times 9.81 \times 0.75}{60{,}000}$$

= 20.03 kW

(iii) $_{mech}$ = bp / ip = 20.03 / 29.16 = 0.687 = 68.7 %.

(iv) Mass of fuel consumed per hour = $\dot{m}_f = \frac{y\ s}{1000\ t} \times 60 = \frac{4000 \times 0.8}{1000 \times 30} \times 60$ = 6.4 kg / h.

. 6.4

bsfc $= \dot{m}_f / bp = \dfrac{}{20.03} = 0.3195$ kg/kWh

(v) $\eta_{ith} = \dfrac{ip}{\dot{m}_f\ CV} = \dfrac{29.16}{(6.4 / 3600) \times 43 \times 10^{3}} = 0.3814 = 38.14\ \%.$

(vi) $\dot{Q}_w = \dot{m}\ C_p\ \Delta T_w = (10 / 60) \times 4.2 \times 36 = 25.2$ kW

Example 2.9:- *A four stroke gas engine has a cylinder diameter of 25 cm and stroke 45 cm. The effective diameter of the brake is 1.6 m.The observations made in a test of the engine were as follows.*
Duration of test = 40 min; Total number of revolutions = 8080 ; Total number of explosions = 3230; Net load on the brake = 80 kg ; mean effective pressure = 5.8 bar; Volume of gas used = 7.5 m^3; Pressure of gas indicated in meter = 136 mm of water (gauge); Atmospheric temperature = 17 C; Calorific value of gas = 19 MJ/ m^3 at NTP; Temperature rise of cooling water = 45 C; Cooling water supplied = 180 kg. Draw up a heat balance sheet and find the indicated thermal efficiency and brake thermal efficiency. Assume atmospheric pressure to be 760 mm of mercury.

Given:- d = 0.25 m ; L = 0.45 m; R = 1.6 / 2 =0.8 m; t = 40 min ; N_{total} = 8080 ;

Hence N = 8080 / 40 = 202 rpm n_{total} = 3230 ;

Hence n = 3230 / 40 = 80.75 explosions / min; W = 80 x 9.81 N; p_{im} = 5.8 bar ;

V_{total} = 7.5 m^3; hence $\dot{V}$ = 7.5 / 40 = 0.1875 m^3/min; p_{gauge} = 136 mm of water

(gauge); T_{atm} = 17 + 273 = 290 K; $(CV)_{NTP}$ = 19 x 10 ^{3} kJ/ m^3 ; T_w = 45 C;

$\dot{m}_w$ = 180 / 40 = 4.5 kg/min; p_{atm} = 760 mm of

mercury ***To find:-*** (i) $_{ith}$; (ii) $_{bth}$; (iii) heat balance sheet

Solution:

(i) $$ip = \frac{p_{im} L A n K}{60,000} = \frac{5.8 \times 10^5 \times (\ /4) \times 0.25^2 \times 0.45 \times 80.75}{60,000}$$

= 17.25 kW.

$$bp = \frac{2 N W R}{60,000} = \frac{2 \times \times 202 \times (80 \times 9.81) \times 0.8}{60,000}$$

= 13.28 kW

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

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

$$= \frac{0.1875 \times 273 \times 770}{290 \times 760} = 0.17875 \text{ m}^3 / \text{min}$$

Heat supplied by fuel = $\dot{Q}_f = \dot{V}_{NTP}\,(CV)_{NTP} = 0.17875 \times 19 \times 10^3 = 3396.25$ kJ/min

Heat equivalent of bp in kJ/min = $13.28 \times 60 = 796.4$ kJ/min

Heat lost to cooling water in kJ/min = $\dot{m}_w C_p T_w = 4.5 \times 4.2 \times 45 = 846.5$ kJ/min

Friction power = ip – bp = 17.25 – 13.28 = 3.97 kW

Hence heat loss due to fiction, pumping etc. = $3.97 \times 60 = 238.2$ kJ/min

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

= 1465.15 kJ/min

Heat Balance Sheet:

Item No.		*Heat Energy Input (kJ/min)*	*(percent)*	*Heat Energy spent (kJ/min)*	*(percent)*
1	Heat supplied by fuel	3396.25	100.00		
2	Heat equivalent of bp			896.4	26.4
3	Heat lost to cooling Water			796.4	23.4
4	Heat equivalent of fp			238.2	7.0
5	Heat unaccounted (by difference)			1465.15	43.2
	Total	3396.25	100.0	3396.25	100.0

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

Speed = 350 rpm; Net brake load = 65 kg ; mean effective pressure = 3 bar ; Fuel consumption = 4 kg/h ; Jacket cooling water flow rate = 500 kg/h ; jacket water temperature at inlet = 20 C ; jacket water temperature at outlet = 40 C ; Test room temperature = 20 C ; Temperature of exhaust gases = 400 C; Air used per kg of fuel = 32 kg ; cylinder diameter = 22 cm ;stroke = 28 cm; effective brake diameter = 1 m ; Calorific value of fuel = 43 MJ/kg ; Mean specific heat of exhaust gases = 1 kJ/kg –K. Find indicated power, brake power and draw up a heat balance for the test in kW and in percentage.

Given:- Two stroke engine. Hence n = N ; N = 350 rpm ; W = (65 x 9.81) N ;

p_{im} = 3 bar ; $\dot{m}_f$ = 4 kg/h ; $\dot{m}_w$ = 500 kg/h ; T_{wi} = 20 C ; T_{wo} = 40 C ; T_{atm} = 20 C ;

T_{eg} = 400 C ; $\dot{m}_a / \dot{m}_f$ = 32 ; d = 0.22 m ; L = 0.28 m ; Brake radius = R = ½ m ;

CV = 43,000 kJ/kg ; $(C_p)_{eg}$ = 1.0 kJ/(kg-K) ;

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

Solution:

$$\text{(i) ip} = \frac{p_{im}\ LAn}{60{,}000} = \frac{3 \times 10^5 \times 0.28 \times (\ /4) \times 0.22^2 \times 350}{60{,}000}$$

= 18.63 kW.

$$\text{(ii) bp} = \frac{2\ \ N\,WR}{60{,}000} = \frac{2 \times\ \ \times 350 \times (65 \times 9.81) \times 0.5}{60{,}000}$$

= 11.68 kW.

(iii) Heat supplied in kW = $\dot{m}_f$ CV = (4 / 3600) x 43,000

= 47.8 kW

= (500 / 3600) x 4.2 x [40 – 20]

= 11.7 kW.

$$= \frac{(32 + 1) \times 4}{3600} \times 1.0 \times [400 - 20]$$

= 13.9 kW

Heat balance sheet:

Heat Input	kW	%	Heat Expenditure	kW	%
Heat supplied by fuel	47.8	100	Heat in bp	11.68	24.4
			Heat lost to cooling Water	11.70	24.5
			Heat lost to exhaust Gases	13.90	29.1
			Unaccounted heat (by difference)	10.52	22.0
Total	**47.80**	**100**	**Total**	**47.80**	**100.0**

UNIT-IV

Compressors:

Compressor is a mechanical device which converts mechanical energy into fluid energy. The compressor increases the air pressure by reducing its volume which also increases the temperature of the compressed air. The compressor is selected based on the pressure it needs to operate and the delivery volume.

Types of Air Compressors:

Compressors are classified in many ways out of which the common one is the classification based on the principle of operation.

Types of Compressors:

1. Positive Displacement and

2. Roto-Dynamic Compressors.

Positive displacement compressors cab be further divided into Reciprocating and rotary compressors.

Under the classification of reciprocating compressors, we have

1. In-line compressors,
2. “V"-shaped compressors,
3. Tandem Piston compressors.
4. Single-acting compressors,
5. Double-acting compressors,
6. Diaphragm compressors.

The rotary compressors are divided into

1. Screw compressors,
2. Vane type compressors,
3. Lobe and scroll compressors and other types.

Under the Roto-dynamic compressors, we have

1. Centrifugal compressors, and the
2. Axial flow compressors.

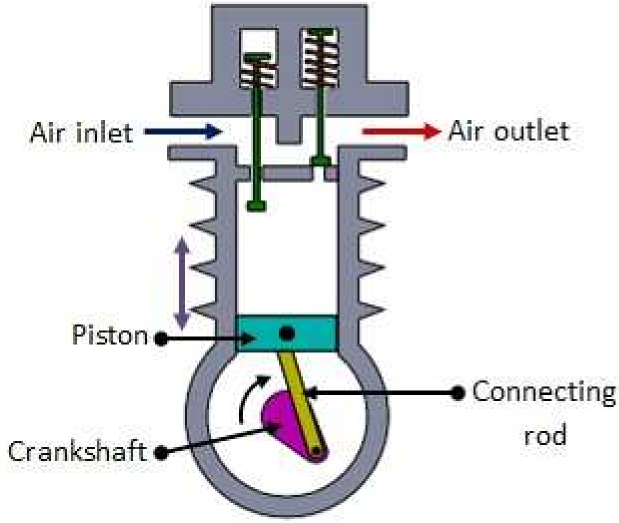
The compressors are also classified based on other aspects like

1. Number of stages (single-stage, 2-stage and multi-stage),
2. Cooling method and medium (Air cooled, water cooled and oil-cooled),
3. Drive types (Engine driven, Motor driven, Turbine driven, Belt, chain, gear or direct coupling drives),
4. Lubrication method (Splash lubricated or forced lubrication or oil-free compressors).
5. Service Pressure (Low, Medium, High)

Positive displacement

Positive-displacement compressors work by forcing air into a chamber whose volume is decreased to compress the air. Once the maximum pressure is reached, a port or valve opens and air is discharged into the outlet system from the compression chamber

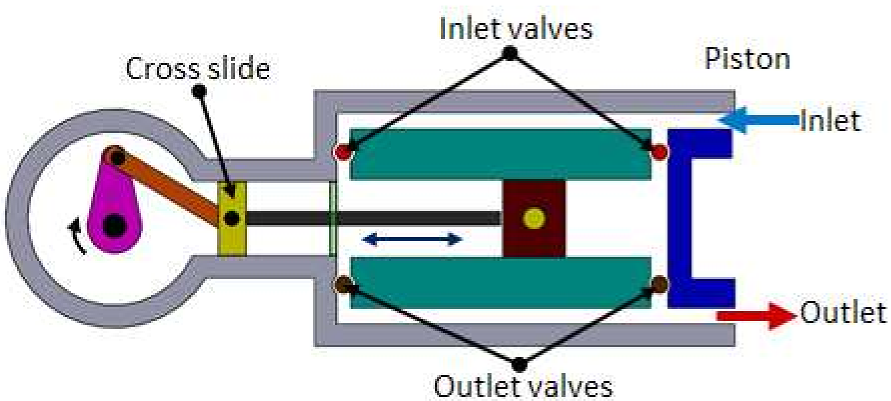
Reciprocating air compressors



Single acting reciprocating air compressor

Piston compressors are commonly used in pneumatic systems. The simplest form is single cylinder compressor. It produces one pulse of air per piston stroke. As the piston moves down during the inlet stroke the inlet valve opens and air is drawn into the cylinder. As the piston moves up the inlet valve closes and the exhaust valve opens which allows the air to be expelled. The valves are spring loaded. The single cylinder compressor gives significant amount of pressure pulses at the outlet port. The pressure developed is about 3-40 bar.

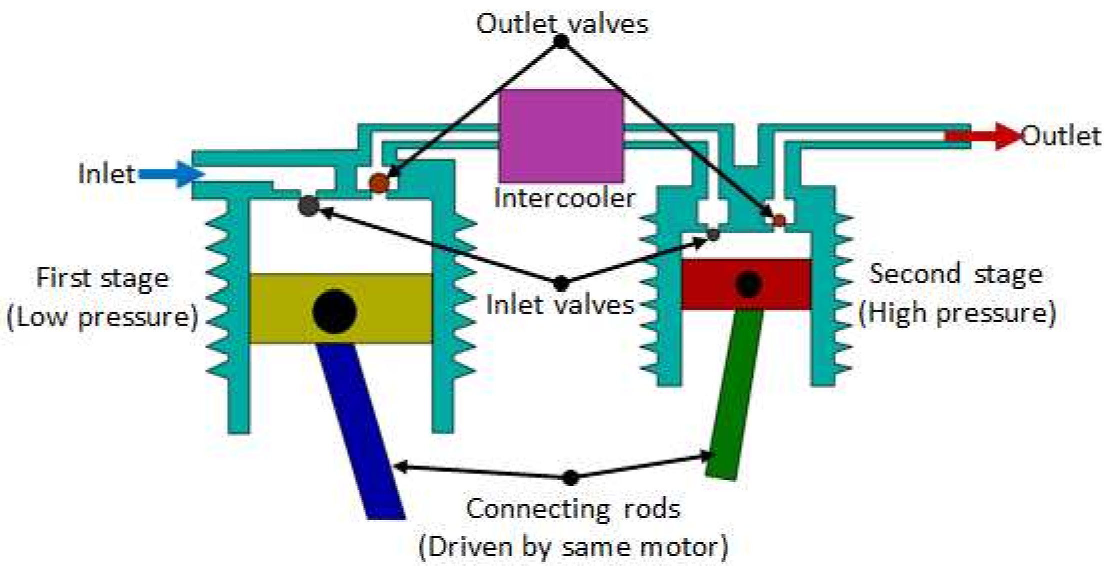
Double acting reciprocating air compressor



Double acting reciprocating air compressor

The pulsation of air can be reduced by using double acting compressor as shown in Figure 6.1.4. It has two sets of valves and a crosshead. As the piston moves, the air is compressed on one side whilst on the other side of the piston, the air is sucked in. Due to the reciprocating action of the piston, the air is compressed and delivered twice in one piston stroke. Pressure higher than 30bar can be produced.

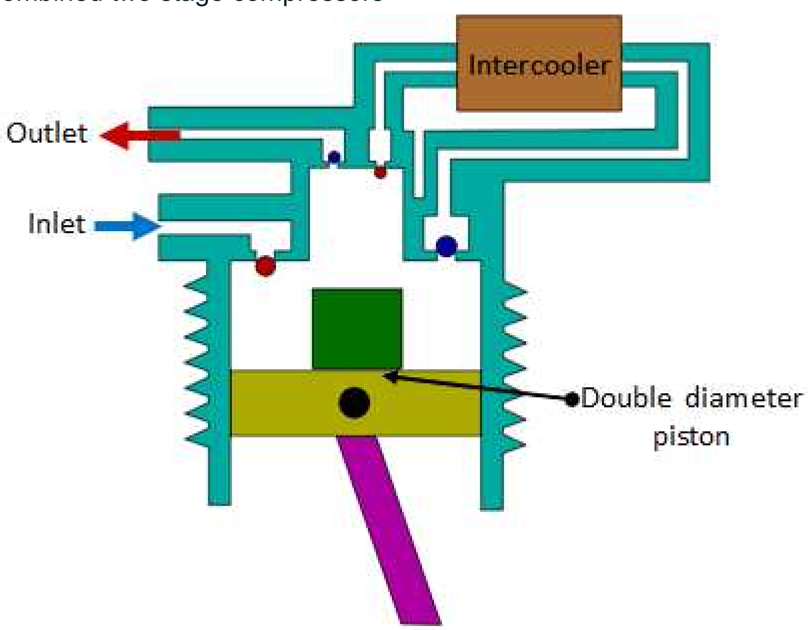
Multistage compressor



Multi-stage compressor

As the pressure of the air increases, its temperature rises. It is essential to reduce the air temperature to avoid damage of compressor and other mechanical elements. The multistage compressor with intercooler in-between is shown in fig. It is used to reduce the temperature of compressed air during the compression stages. The inter-cooling reduces the volume of air which used to increase due to heat. The compressed air from the first stage enters the intercooler where it is cooled. This air is given as input to the second stage where it is compressed again. The multistage compressor can develop a pressure of around 50bar.

Combined two stage compressors



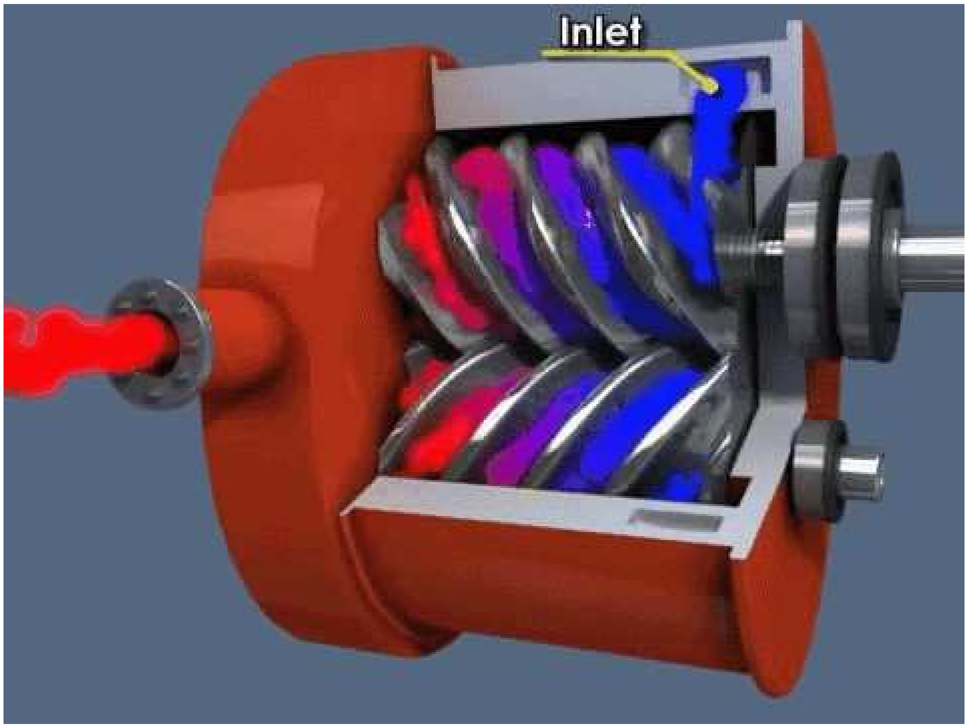
Combined two stage compressor

In this type, two-stage compression is carried out by using the same piston. Initially when the piston moves down, air is sucked in through the inlet valve. During the compression process, the air moves out of the exhaust valve into the intercooler. As the piston moves further the stepped head provided on the piston moves into the cavity thus causing the compression of air. Then, this is let out by the exhaust port.

Rotary screw compressors

Rotary compressors are the type of famous compressors. It uses two Asymmetrical rotors that are also called helical screws to compress the air.

The rotors have a very special shape and they turn in opposite directions with very little clearance between them. The rotors are covered by cooling jackets. Two shafts on the rotors are placed that transfer their motion with the help of timing gears that are attached at the starting point of the shafts/compressor(as shown in the image).



Rotary Compressor

Working principle-Air sucked in at one end and gets trapped between the rotors and get pushed to other side of the rotors .The air is pushed by the rotors that are rotating in opposite direction and compression is done when it gets trapped in clearance between the two rotors. Then it pushed towards pressure side.

Rotary screw compressors are of two types oil-injected and oil-free.

Oil-injected is cheaper and most common than oil-free rotary screw compressors.

Advantages

Less noisy in operation.

These are called the work-horses as they supply large amount of compress air.
More energy efficient as compared to piston type compressors.
The air supply is continuous as compared to reciprocating compressors.
Relatively low end temperature of compressed air.

Disadvantages

Expensive than piston-type compressors.
More complex design.
Maintenance is difficult.

Power producing and power absorbing machines

A power generating machine converts potential/kinetic energy of fluid into mechanical energy and later in electrical energy.

An IC engine is a power generating power generating and a compressor/pump is a power absorbing turbo machine.

Power Producing Machines

IC engines, Gas turbines, Steam turbines, Hydraulic turbines, Wind turbines etc.

Power Absorbing Machines

On the other hand power absorbing machines uses electric energy to do work on the fluid.

Fans, blowers, compressors, pump etc.

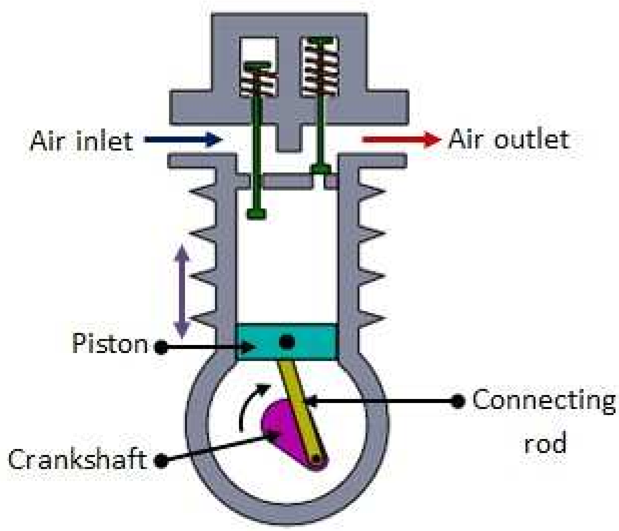
A fan moves large amounts of air with a low increase in pressure:

A blower is a machine used for moving air with a moderate increase of pressure: a more powerful fan, if you will. By changing the angle of the blades, a blower will be able to push air in any direction. A blower is a machine for moving volumes of a gas with moderate increase of pressure

A compressor is a machine for raising gas to a higher level of pressure, actually making the air denser by cramming air into a small space.

A compressor is a machine for raising a gas a compressible fluid - to a higher level of pressure

Reciprocating air compressors



Single acting reciprocating air compressor

Piston compressors are commonly used in pneumatic systems. The simplest form is single cylinder compressor. It produces one pulse of air per piston stroke. As the piston moves down during the inlet stroke the inlet valve opens and air is drawn into the cylinder. As the piston moves up the inlet valve closes and the exhaust valve opens which allows the air to be expelled. The valves are spring loaded. The single cylinder compressor gives significant amount of pressure pulses at the outlet port. The pressure developed is about 3-40 bar.

WORK DONE IN SINGLE-STAGE RECIPROCATING AIR COMPRESSOR NEGLECTING CLEARANCE VOLUME

Fig. 6.1 shows the theoretical air compression cycle without clearance on p-V diagram. The different operations are : suction, compression and discharge. The air is sucked at constant pressure p_1 and is deliverged at constant pressure p_2. The work done in the cycle is given by the area of the p-V diagram.

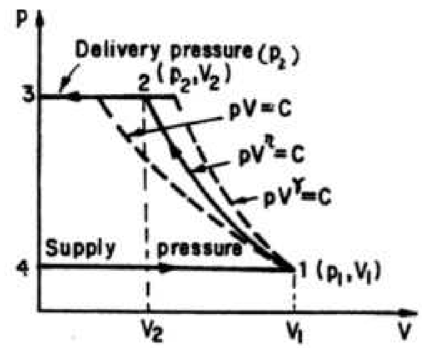


Fig. 6.1

Work done when compression follows the law $pV^n = C$.

Work done per cycle $= \text{Area of } p\text{-}V \text{ diagram 1-2-3-4}$

$= (\text{Area under 1-2}) + (\text{Area under 2-3}) - (\text{Area under 1-4})$

$$= \left(\frac{p_2V_2 - p_1V_1}{n-1}\right) - p_2V_2 - p_1V_1$$

$$= (p_2V_2 - p_1V_1)\left(\frac{1}{n-1} + 1\right) = \left(\frac{n}{n-1}\right)(p_2V_2 - p_1V_1)$$

$$= \left(\frac{n}{n-1}\right)p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \left(\frac{n}{n-1}\right) mRT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad (\because \; p_1V_1 = mRT_1)$$

6.3.2. Work done when compression follows the law $pV^{\gamma} = C$

The work done per cycle is obtained by changing n to γ.

$$\therefore \quad \text{Work done per cycle} = \left(\frac{\gamma}{\gamma-1}\right) mRT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \text{ or } \left(\frac{\gamma}{\gamma-1}\right) p_1V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right].$$

6.3.3. Work done when compression follows the isothermal law (*i.e.*, pV = C)

$$\text{Work done per cycle} = p_1V_1 \log_e\left(\frac{V_1}{V_2}\right) + p_2V_2 - p_1V_1$$

$$= p_1V_1 \log_e\left(\frac{V_1}{V_2}\right) \qquad (\because \; p_2V_2 = p_1V_1)$$

$$= p_1V_1 \log_e\left(\frac{p_2}{p_1}\right) = p_1V_1 \log_e r$$

where r = Compression ratio $= \frac{V_1}{V_2} = \frac{p_2}{p_1}$

The work done is minimum when compression follows isothermal law (*i.e.* $pV = C$ or $n = 1$) and is maximum when compression is adiabatic (*i.e.*, $n = \gamma$). Isothermal compression is not possible in practice as the compressor would need to run at very low speed. In practice, the value of n varies from 1.1 to 1.3.

The performance of a reciprocating air compressor is given by *isothermal efficiency* which is the ratio of isothermal work and actual indicator work.

EFFICIENCIES OF A COMPRESSOR

The efficiencies of a compressor are :

(*i*) isothermal efficiency, (*ii*) adiabatic efficiency,

(*iii*) mechanical efficiency, and (*iv*) volumetric efficiency.

(*i*) **Isothermal efficiency.** It is the ratio of isothermal work to actual indicator work. Mathematically,

$$\text{Isothermal efficiency} = \frac{p_1V_1 \log_e\left(\dfrac{p_2}{p_1}\right)}{\left(\dfrac{n}{n-1}\right)p_1V_1\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right]} = \frac{\log_e\left(\dfrac{p_2}{p_1}\right)}{\left(\dfrac{n}{n-1}\right)\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right]}$$

(*ii*) **Adiabatic efficiency**. It is the ratio of adiabatic work to actual work of a compressor. Mathematically,

$$\text{Adiabatic efficiency} = \frac{p_1V_1\left(\dfrac{\gamma}{\gamma-1}\right)\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}{p_1V_1\left(\dfrac{n}{n-1}\right)\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right]}$$

$$= \frac{\left(\dfrac{\gamma}{\gamma-1}\right)\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}{\left(\dfrac{n}{n-1}\right)\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right]}.$$

(*iii*) **Mechanical efficiency.** It is the ratio of I.H.P. to B.H.P. of the motor.

(*iv*) **Volumetric efficiency.** It is the ratio of the actual mass of air pumped by the compressor to the mass of air which the compressor would pump if it handled a volume of air equal to swept volume at the suction stroke at free air condition (*i.e.,* intake condition).

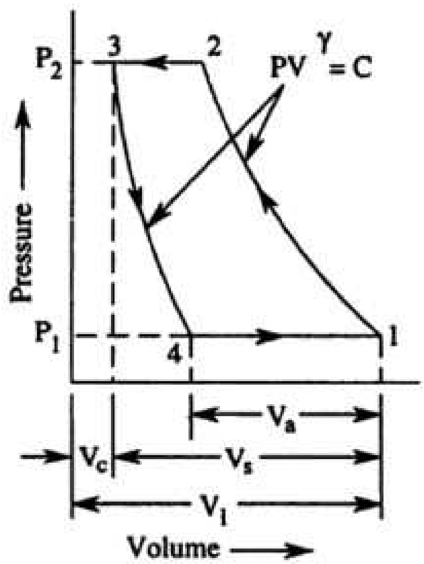
1. If the expansion and compression follows the same law, then volumetric efficiency in terms of clearance ratio and pressure ratio is given by

$$\text{Volumetric efficiency} = 1 + C - C\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \text{ or } 1 + C - C\left(\frac{V_1}{V_2}\right)$$

where C = Clearance ratio, and $\left(\dfrac{p_2}{p_1}\right)$ = Pressure ratio.

2. The volumetric efficiency decreases with the increase of pressure ratio. Also the volumetric efficiency decreases with the increase of clearance ratio.

EFFECT OF CLEARANCE VOLUME AND EXPRESSION FOR VOLUMETRIC EFFICIENCY



Single stage air compressor with clearance

In actual compressor, a certain clearance space is provided between the extreme travel of the piston and the cylinder cover to prevent the piston from striking the end or cover of the cylinder. The volume, thus left unswept by the piston is known as clearance volume. Therefore at the end of every delivery stroke the amount of air filling the clearance volume remains in the cylinder. The clearance volume is generally expressed as the percentage of piston displacement. Figure shows the indicator diagram for a single stage air compressor with clearance.

As already stated, at the end of the delivery stroke the amount of air filling the clearance volume will not be discharged but remains in the cylinder. At the begining of the forward stroke, air is not sucked in but the air in the clearance space expands till the pressure becomes P_1 and volume V_4, and then suction begins. The volume of air drawn in at the end of suction stroke is V_a.

EXPRESSION FOR WORK DONE HAVING CLEARANCE

Let the index n of expansion curve 3–4 and compression curve 1–2 is same.
Work required per cycle = area 1–2–3–4 = area 1–2–6–5– area 3–4–5–6

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right] - \frac{n}{n-1} P_1 V_4 \left[\left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right]$$

$$W = \frac{n}{n-1} P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right] = \frac{n}{n-1} P_1 V_a \left[\left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right] \quad \text{J/cycle}$$

Thus it is seen that the work required to compress and deliver same volume of air V_a with clearance and without clearance is same.

Indicated power of the compressor

$$= \frac{\text{Work required per cycle} \times \text{No. of cycles per minute}}{60}$$

$$= \frac{W \times N}{60} \ \text{J/s or Watt.} \qquad = \frac{W \times N}{60000} \ \text{kW.}$$

If P_1V_a of Eq. is replaced by mRT_1, then the work required per kg of air is given by

$$W = \frac{n}{n-1} RT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right] \quad \text{J}$$

Work required in J/s = W × mass of air delivered per second

POWER AND EFFICIENCY OF A COMPRESSOR

Single Stage Compressor without Clearance

Isothermal work required per cycle,

$$W = P_1V_1 \log_e \left(\frac{P_2}{P_1} \right) \quad \text{J}$$

(a) Isothermal power $= \dfrac{W \times N}{60}$ J/s or Watt

where N = No. of cycles per minute.

But $P_1 V_1 = mRT_1$ then isothermal work required per kg of air is given by,

$$W = RT_1 \log_e \left(\frac{P_2}{P_1}\right) \text{ J}$$

(b) Isothermal power = $W \times$ mass of air delivered per second. watt.

(c) Isothermal efficiency $= \dfrac{\text{Isothermal power in watts.}}{\text{Indicated or actual power in watts}}$

(d) Overall isothermal efficiency or compressor efficiency =

$$\frac{\text{Isothermal power in watts}}{\text{Brake power or shaft power required to drive the compressor in watts}}$$

(e) Mechanical efficiency $= \dfrac{\text{Indicated power in watts}}{\text{Brake power or shaft power in watts.}}$

Adiabatic work required per cycle,

$$W = \frac{\gamma}{\gamma - 1}(P_2 V_2 - P_1 V_1) \quad \text{J}, = \frac{\gamma}{\gamma - 1} P_1 V_1 \left(\frac{P_2 V_2}{P_1 V_1} - 1\right)$$

$$= \frac{\gamma}{\gamma - 1} mRT_1 \left(\frac{T_2}{T_1} - 1\right) = \frac{\gamma}{\gamma - 1} mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}} - 1\right] \text{J}$$

$$= \frac{\gamma}{\gamma - 1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}} - 1\right], \quad \left[\therefore \quad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}}\right]$$

(f) Adiabatic power $= \dfrac{\gamma}{\gamma - 1} P_1 V_1 \left[\left(\dfrac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}} - 1\right] \times \dfrac{N}{60}$ J/s or watt.

(g) Adiabatic efficiency =

$$= \frac{\text{Adiabatic power in watts}}{\text{Brake power or shaft power required to drive the compressor}}$$

Multi-stage compression

We have seen in section that as the pressure ratio increases the volumetric efficiency (and hence the capacity of the compressor) decreases. Therefore, for high pressure ratios, compression is carried out in stages in separate cylinders instead of compressing the air in a single cylinder, and such compression is called *multistage compression.* The advantages of multistage compression are :

i) higher volumetric efficiency for a given pressure ratio.

ii) less driving power for a given pressure ratio.

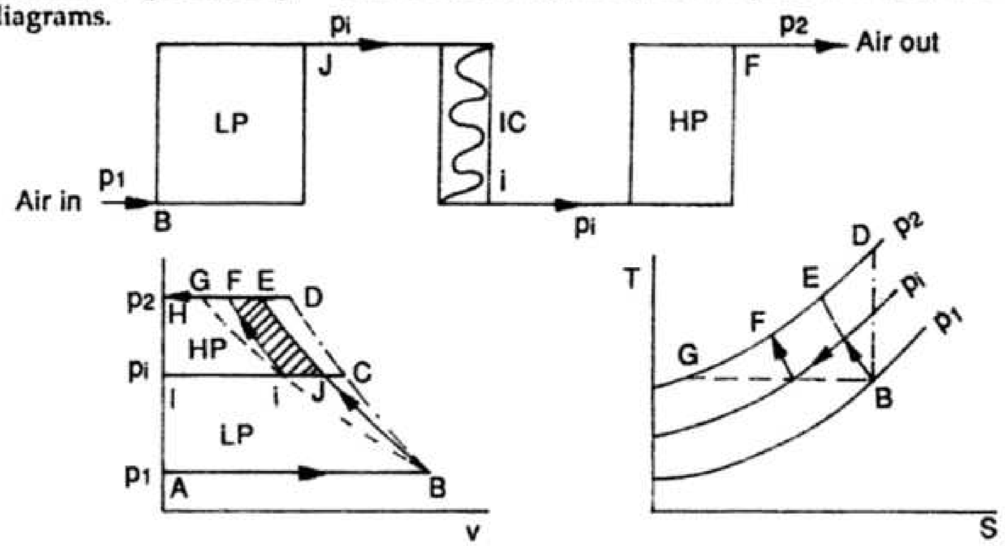
iii) better mechanical balancing due to more number of cylinders.

iv) exit air temperature is less.

In multistage compression an *intercooler* is used in between cylinders to cool the air at constant pressure. In the discussion below, it will be assumed that intercooling is perfect, i.e. the air coming out of a cylinder is cooled at constant pressure in the intercooler down to its initial temperature before it is admitted into the next cylinder. It is also assumed that the index n for compression is same for all cylinders and for simplicity we will also assume that there is no clearance space.

TWO-STAGE COMPRESSION

Consider 1 kg of air. Fig. shows the arrangement, and the process in (p-v) and (T-s) diagrams.



LP - low pressure cylinder
HP - high pressure cylinder
IC - intercooler

BiG - Isothermal $pv = c$
BCD - Isentropic $pv^{\gamma} = c$
BJE,iF - Polytropic $pv^{n} = c$

Multistage compression

i) Air is compressed from p_1 to some intermediate pressure (between suction pressure p_1 and delivery pressure p_2) p_i, say polytropically along *BJ* in the low pressure cylinder *LP*.

ii) Air from the *LP* cylinder is cooled in the intercooler *IC* at constant pressure p_i to its initial temperature T_B along *Ji*.

iii) Air from the intercooler is compressed from p_i to p_2 in the *HP* cylinder along *iF*.

For single stage compression, work required to compress and deliver 1 kg of air is given by:

area *ABDHA* - isentropic compression;
area *ABEHA* - polytropic compression; ($pv^n = c$)
area *ABGHA* - isothermal compression.

Work required for *two-stage* polytropic compression (could be isentropic compression) is given by,

W = work in *LP* + work in *HP*

= area *ABJIA* + area *IiFHI*

$= \{n/(n-1)\}\, p_B v_B \{1 - (p_i/p_1)^{(n-1)/n}\} + \{n/(n-1)\}\, p_i v_i \{1 - (p_2/p_i)^{(n-1)/n}\}$

$= \{n/(n-1)\}\, p_1 v_1 \{1 - (p_i/p_1)^{(n-1)/n}\} + \{n/(n-1)\}\, p_1 v_1 \{1 - (p_2/p_i)^{(n-1)/n}\}$

Taking v_B as v_1; and since *B* or *1*, and *i* are on the same isothermal $p_1 v_1 = p_i v_i$,

$$\therefore \quad W = \{n/(n-1)\}\, p_1 v_1 \{2 - (p_i/p_1)^{(n-1)/n} - (p_2/p_i)^{(n-1)/n}\}$$

For a given condition, n, p_1, v_1 and p_2 are fixed, therefore W will be minimum when the quantity within the second bracket is minimum.

Let $y = 2 - (p_i/p_1)^{(n-1)/n} - (p_2/p_i)^{(n-1)/n}$; and put $(n-1)/n = x$

$$y = 2 - (p_i/p_1)^x - (p_2/p_i)^x = 2 - (p_i^x/p_1^x) - (p_2^x/p_i^x)$$

Here, the variable is p_i. Differentiating with respect to p_i we get,

$$\therefore \quad dy/dp_i = -(xp_i^{(x-1)})/p_1^x + (xp_2^x)/p_i^{x+1} = 0$$

$$\therefore \quad p_i^{(x-1)}/p_1^x = p_2^x/p_i^{(x+1)}$$

or $\quad p_1^x p_2^x = p_i^{2x}$

or $\quad p_i^2 = p_1 p_2$

$\therefore \quad p_i = (p_1 p_2)^{0.5}$, for minimum work

From this we get $\quad p_i/p_1 = (p_1)^{0.5}(p_2)^{0.5}/p_1 = (p_2/p_1)^{0.5}$

and $\quad p_2/p_i = p_i/p_1 = (p_2/p_1)^{0.5}$

$\therefore$ Minimum Work $W_{min} = \{n/(n-1)\}\, p_1 v_1 \{2 - (p_2/p_1)^{(n-1)/2n} - (p_2/p_1)^{(n-1)/2n}\}$

$= \{2n/(n-1)\}\, p_1 v_1 \{1 - (p_2/p_1)^{(n-1)/2n}\}$

$= \{2n/(n-1)\} RT_1 \{1 - (p_2/p_1)^{(n-1)/2n}\}$

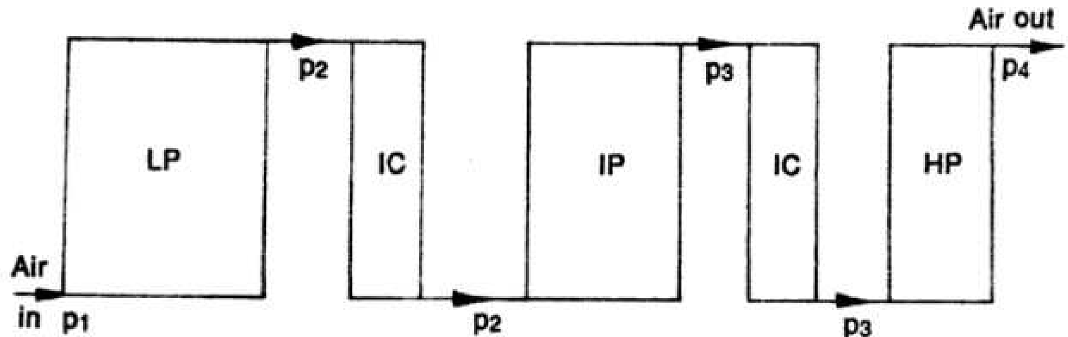
For temperature we get,

$$T_i/T_B = (p_i/p_1)^{(n-1)/n} = (p_2/p_1)^{(n-1)/2n}$$
$$T_F/T_i = (p_2/p_i)^{(n-1)/n} = (p_2/p_1)^{(n-1)/2n}$$

Work saved due to two-stage compression is given by the area *iJEF* shown by the shaded area.

For three-stage compression with perfect intercooling, it can be shown that

$$p_2/p_1 = p_3/p_2 = p_4/p_3 = (p_4/p_1)^{1/3}$$



IP -Intermediate pressure cylinder

$$W_{min} = \{3n/(n-1)\}\, p_1 v_1 \,\{1 - (p_4/p_1)^{(n-1)/3n}\}$$
$$= \{3n/(n-1)\}\, RT_1 \,\{1 - (p_4/p_1)^{(n-1)/3n}\}$$

In a three-stage compressor there are two intercoolers.

If intercooling is not perfect, i.e. if the air is not cooled down to its initial temperature, the work for the cylinders is to be calculated saparately and then added together. For a multistage air compressor with clearance volume, the volumetric efficiency is that of the *LP* cylinder of the compressor. Notice that saving of work due to multistaging is possible for isentropic and polytropic compressions only (i.e. for *n* greater than 1); as the number of stages increase the compression process approaches isothermal compression.

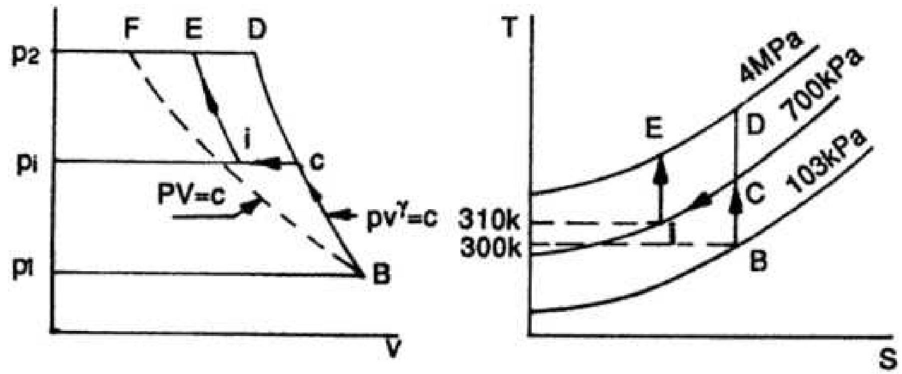
Ex.1

Air at 103 kPa and 27°C is drawn in the *LP* cylinder of a two-stage air compressor and is isentropically compressed to 700 kPa. The air is then cooled at constant pressure to 37°C in an intercooler and is then again compressed isentropically to 4MPa in the *HP* cylinder, and is delivered at this pressure. Determine the power required to run the compressor if it has to deliver 30 m^3 of air per hour measured at inlet condition.

$$p_1 = 103 \text{ kPa}; p_2 = 4\text{MPa}; p_i = 700 \text{ kPa}$$

$$T_1 = 273 + 27 = 300\text{K}; T_i = 273 + 37 = 310\text{K}$$

Fig. 4.8 shows the process in (p-v) and (T-s) diagrams.



Here the intercooling is not perfect, because, in the intercooler the air was cooled to 37°C, and not to the initial temperature of 27°C.

$W = W_{LP} + W_{HP}$

$= \{\gamma/(\gamma-1)\}RT_1 \{1-(p_i/p_1)^{(\gamma-1)/\gamma}\} + \{\gamma/(\gamma-1)\}RT_i \{1 - (p_2/p_i)^{(\gamma-1)/\gamma}\}$

$= \{1.4 \times 0.287 \times 300/(1.4-1)\} \{1 - (700/103)^{0.2857}\}$

$+ \{1.4 \times 0.287 \times 310 / (1.4 - 1)\} [1 - \{(4 \times 10^6) / (700 \times 10^3)\}^{0.2857}]$

$= 301.35 (- 0.729) + 311.4 (- 0.6454) = - 219.6 - 201 = - 420.6$ kJ/kg.

Mass of 30 m^3 of air at 103 kPa and 300 K, is given by

$m = (103 \times 10^3 \times 30)/(287 \times 300) = 35.888$ kg (per hour)

$\therefore$ mass per second $= 35.888/(60 \times 60) = 0.00997$ kg/s

$\therefore$ power required $= 0.00997 \times 420.6$ kJ/s $= 4.193$ kW

This is the theoretrical power; actual power required will be more.

P (actual) = P(theorectical)/mechanical efficiency.

A three-stage air compressor with perfect intercooling takes 15 m^3 of air per minute at 95 kPa and 27°C, and delivers the air at 3.5 MPa. If compression process is polytropic ($pv^{1.3} = c$), determine :

i) power required if mechanical efficiency is 90%.

ii) heat rejected in the intercoolers per minute.

iii) isothermal efficiency.

iv) heat rejected through cylinder walls per minute.

$T_1 = 273 + 27 = 300$K; intercooling is perfect; therefore,

$$p_2/p_1 = p_3/p_2 = p_4/p_3 = (p_4/p_1)^{1/3}$$

i) POWER REQUIRED

Mass of 15m^3 of air 95 kPa and 300K

$$m = (95 \times 10^3 \times 15)/(287 \times 300) = 16.55 \text{ kg}$$

$$W = \{3n/(n-1)\}RT_1 \{1 - (p_4/p_1)^{(n-1)/3n}\}$$

$$= \{(3 \times 1.3 \times 0.287 \times 300)/(1.3 - 1)\}\{1 - (3500/95)^{0.0769}\}$$

$$= 1119.3\ (-0.3196) = -357.73 \text{ kJ/kg}$$

$\therefore$ power required = $(357.73 \times 16.55)/(60 \times 0.9) = 109.6$ kW

ii) HEAT REJECTED PER MINUTE IN INTERCOOLERS

$$T_2/T_1 = (p_2/p_1)^{(n-1)/n} = (p_4/p_1)^{(n-1)/3n}$$

$$= (3500/95)^{(1.3-1)/(3 \times 1.3)}$$

$$= 36.84^{0.0769} = 1.3196$$

$\therefore$ $T_2 = 300 \times 1.3196 = 395.9$K

As the intercooling is perfect, the temperature range in both the intercoolers will be same, hence the heat rejected in each will also be same.

$\therefore$ Heat rejected per minute in the two intercoolers

$$= mc_p(T_2 - T_1) \times 2$$

$$= 16.55 \times 1.005\ (395.9 - 300) \times 2$$

$$= 3190 \text{ kJ/min}$$

iii) ISOTHERMAL EFFICIENCY

$$\text{isothermal work} = RT_1 \log_e (p_1/p_4)$$

$$= 0.287 \times 300 \log_e (95/3500) = -310.5 \text{ kJ/kg}$$

$\therefore$ isothermal efficiency $\eta_{iso} = 310.5/357.73 = 0.868$

iv) HEAT REJECTED THROUGH CYLINDER WALLS

As the compression of the air in the three cylinders is polytropic, some heat will be transferred from the air during compression through the cylinder walls; this rejection of heat is in addition to the heat rejected in the intercoolers. Again, while the heat rejected in the intercoolers is at constant pressure, the heat rejected through the cylinder walls is during compression.

For each cylinder, per kg of air (neglecting *KE* and *PE* effects)

$$h_1 + W = h_2 + Q; \text{ (energy in = energy out)} \quad \ldots (a)$$

or

$$Q = h_1 - h_2 + W$$
$$= c_p (T_1 - T_2) + W$$

where Q = heat rejected through cylinder walls in each cylinder, per kg of air, and W = work required by each cylinder per kg of air.

As the intercooling is perfect, pressure and temperature range in each cylinder will be same, and hence W and Q will also be same.

$$W = \{nR/(n-1)\}\,(T_1 - T_2)$$
$$= \{1.3 \times 0.287/(1.3 - 1)\}(300 - 395.9)$$
$$= -119.25 \text{ kJ/kg}$$

This is just 1/3 of the total compressor work, 357.73 kJ/kg as calcutated in (*i*).

$$\therefore \quad Q = 1.005\,(300 - 395.9) + 119.25$$
$$= -96.38 + 119.25$$
$$= 22.87 \text{ kJ per cylinder per kg of air.}$$

Notice that here we have just put in the value of W because the sign (negative) was already taken into consideration in (*a*) above.

Thus, heat rejected per minute in each cylinder

$$= 22.87 \times 16.55 \text{ kJ}$$

$\therefore$ Total heat rejected per minute $= 22.87 \times 16.55 \times 3$

$$= 1135.5 \text{ kJ}$$

EXAMPLE Air at 100 kPa and 280 K is compressed steadily to 600 kPa and 400 K. The mass flow rate of air is 0.02 kg/s and a heat loss of 16 kJ/kg occurs during the process.

Assuming the changes in kinetic and potential energies to be negligible, determine the change in enthalpy, work done per mole of air and necessary power input to the compressor. Assume air to behave as an ideal gas. For air

$$C_P^0 = 28.11 + 0.1967 \times 10^{-2}T + 0.4802 \times 10^{-5}T^2 - 1.966 \times 10^{-9}T^3 \text{ [T in K, } C_P^0 \text{ in J/mol-K]}$$

Solution: State 1: 100 kPa, 280 K, $\dot{m}$ = 0.02 kg/s

State 2: 600 kPa, 400 K, $\dot{m}$ = 0.02 kg/s

Energy balance around the compressor gives

$$\Delta H = q - w \qquad \text{(neglecting kinetic and potential energy changes)}$$

$$\Delta H^{ig} = \int C_P^0 dT$$

$$\int_{T_1}^{T_2} C_P^0 dT = 28.11 \times 120 + \frac{0.1967 \times 10^{-2}}{2}(400^2 - 280^2) + \frac{0.4802 \times 10^{-5}}{3}(400^3 - 280^3)$$

$$- \frac{1.966 \times 10^{-9}}{4}(400^4 - 280^4) = 3511.2 \text{ J/mol}$$

$$\Delta H^{ig} = 3511.2 \text{ J/mol}$$

Molecular weight of air = 0.21 × 32 + 0.79 × 28 = 28.84

$$q = -16\frac{J}{g} \times \frac{28.84\,g}{1\,\text{mol}} = -461.44 \text{ J/mol}$$

$w = q - \Delta H = -461.44 - 3511.2 = -3972.64$ J/mol. The negative sign implies that work is done on the compressor.

$$\dot{m} = 0.02 \text{ kg/s} = 20 \text{ g/s}$$

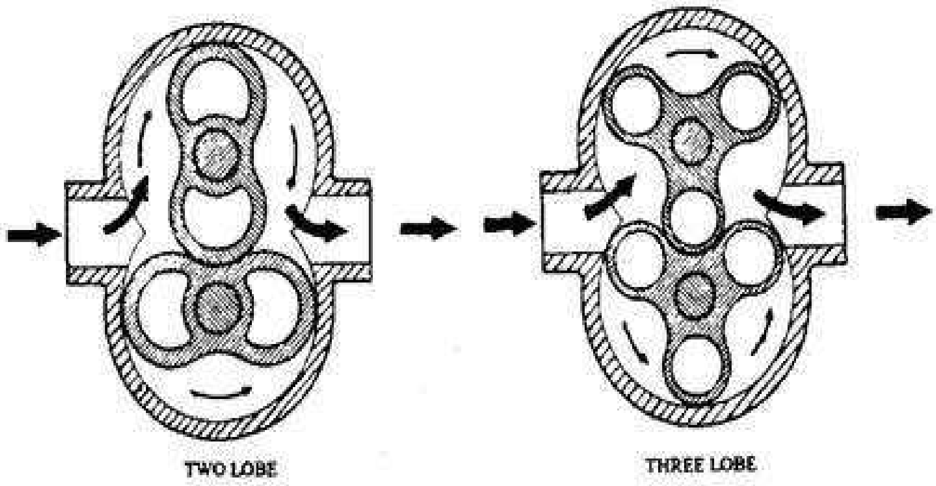
$$\dot{n} = \frac{20g}{s}\frac{1\,\text{mol}}{28.84\text{g}} = 0.6935 \text{ mol/s}$$

$$\dot{W} = -3972.64\frac{J}{\text{mol}} \times 0.6935\frac{\text{mol}}{s} = -2755.03 \text{ W} = -2.755 \text{ kW}.$$

Rotary air compressors:

Roots blower

A root blower consists of two rotors with lobes rotating in a air tight casing. The casing has inlet and outlet PORTS ON OPPOSITE SIDES. Root blower has two or three lobes as given in fig.



The lobes are so designed that they provide an air tight joint at point of their contact. One of the rotors is rotated by external means. The other is gear is driven by the first one. When the rotator rotates, the air at atmosphere pressure is trapped in the pockets formed between rotors and casing. The rotary motion of the lobes delivers the entrapped air into the receiver.
Thus more and more air is delivered in to the receiver. This increases the pressure of air in the receiver. Finally the air is used at required pressure from the receiver.

Roots-blower advantages

A roots-blower quickly attains the full number of revolutions

The power demand in the partial-load range is lower.

Roots-Blower Disadvantages

In the range of partial load, the conveying speed is higher, i.e. Wear on the conveyor pipe and breakage of the conveying material will be greater;

A conveying speed once chosen can only be altered with considerable expenditure;

Due to the low-frequency noises (pulsating conveying air flow), expensive noise dampening equipment will be necessary;

In order not to exceed the operating pressures, a control device must be installed; on account of the narrow piston clearance.

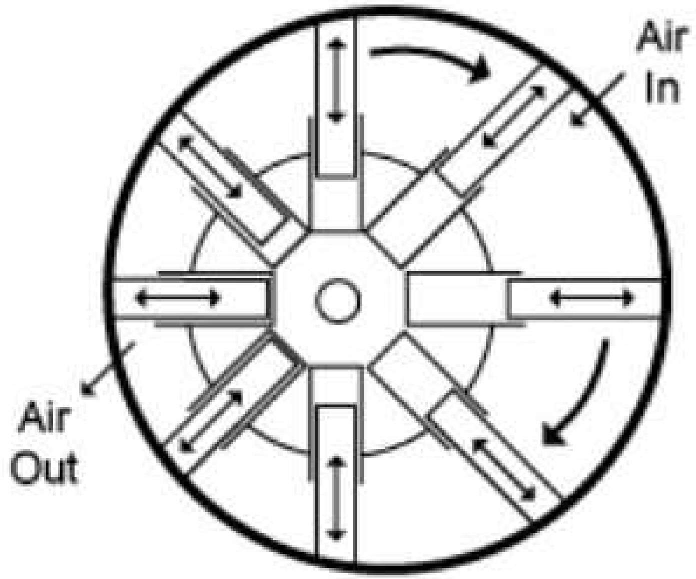
The roots-blower is sensitive to foreign matter, i.e., Filter cleaning of the conveying air is required;

After a longer period of use, the piston clearance becomes larger and leads to capacity losses.

The Vane Compressors:

In an air tool, compressed air enters the inlet port which is plumbed to the smallest compartment of the vane-housing inside. The compressed air should be at least at the minimum operating pressure for that air tool to work properly.

The compressed air is moving from an area of high pressure as it enters the air tool, to an area of relative low pressure, that being back to atmospheric pressure outside the air tool. As the air moves inside the tool, it too moves the vanes.



As the center shaft rotates, so to does the vane housing. The vanes slide in and out of the housing, keeping contact with the wall of the cylinder. Air enters at the largest opening and exits at the smallest, reducing volume and compressing the air.

As the shaft in the vane-housing rotates due to the air movement, the vanes inserted into that housing slide in or out, depending on where they are in the cycle.

Centrifugal force ensures that the vanes are always keeping contact with the inside of the outer cylinder, creating a seal. This forms air-tight compartments within the vane housing.

As mentioned, compressed air always flows from an area of high pressure to an area of low pressure, so the high pressure air in the small vane-compartment wants to get to the larger area vane, and ultimately, out of the tool back to a stable, lower, pressure.

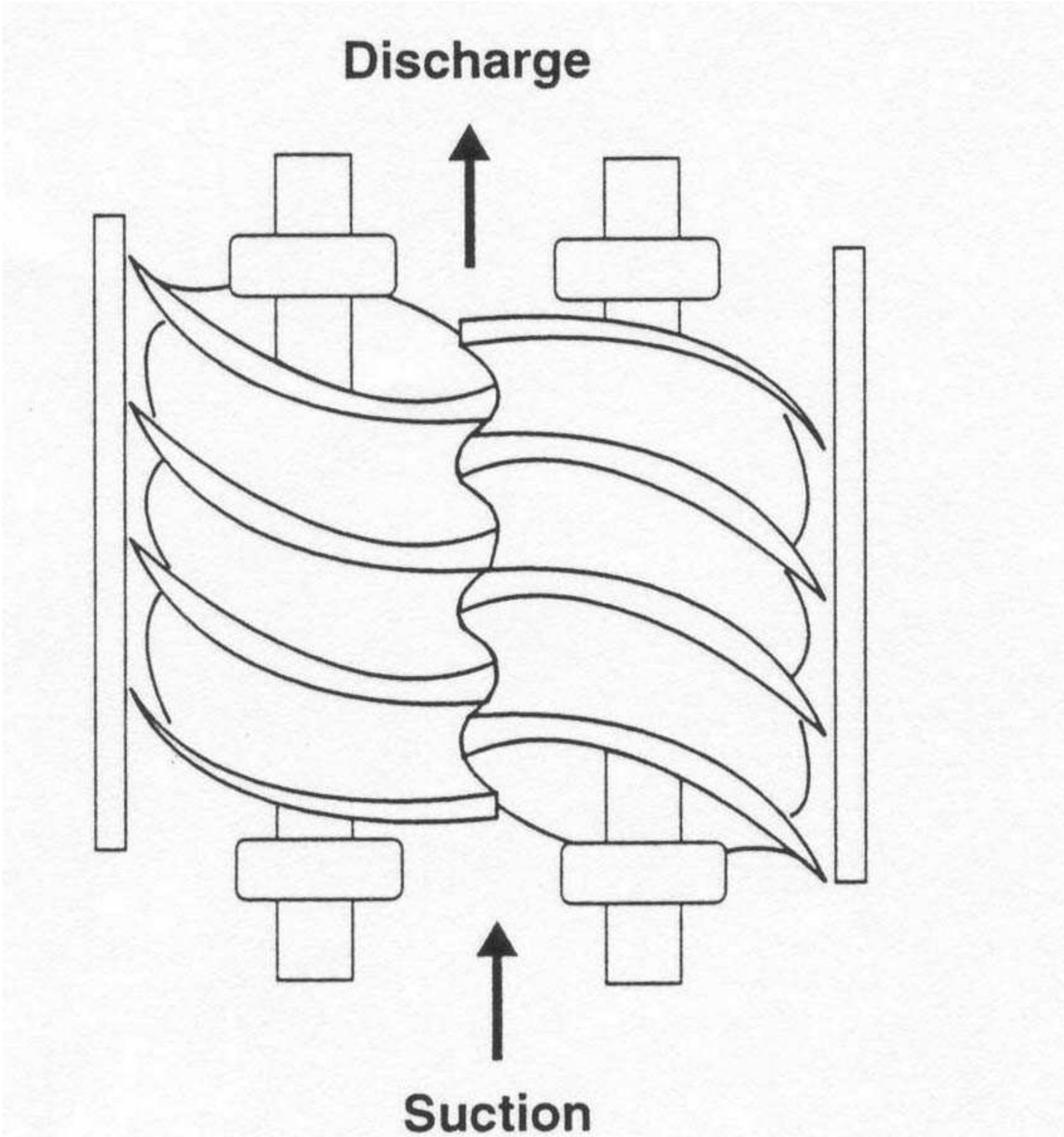
The shaft inside of the vane-housing extends through air-tight seals up into the air tool, and is attached to tooling on the end or to gearing of some sort. The result is rotary motion of that tooling.

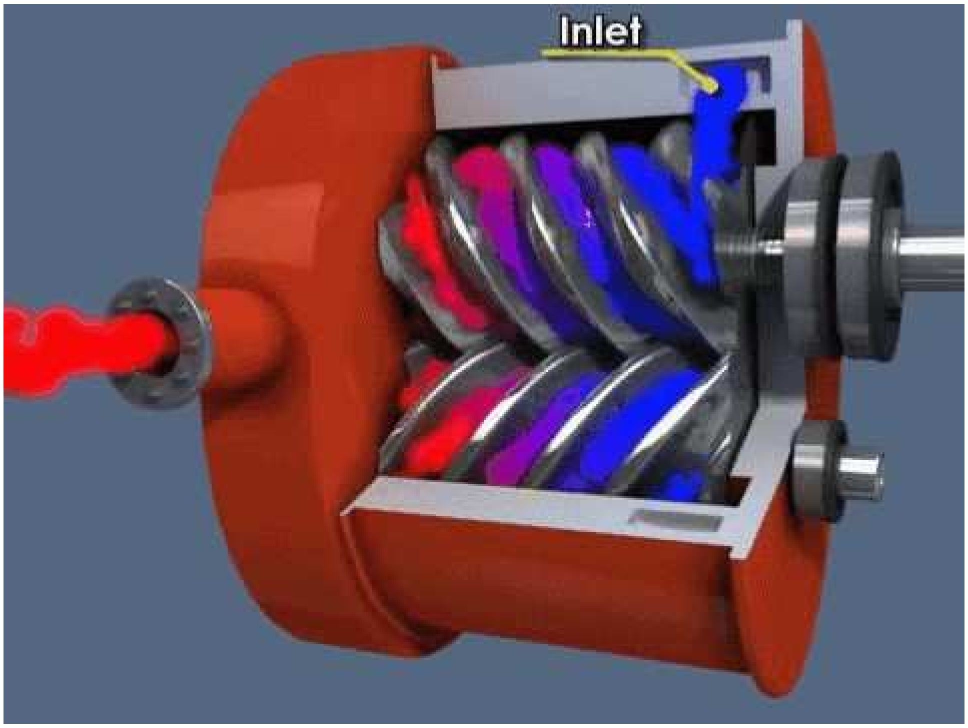
The power that drives the air tool is compressed air.

Rotary screw compressors

Rotary compressors are the type of famous compressors. It uses two Asymmetrical rotors that are also called helical screws to compress the air.

The rotors have a very special shape and they turn in opposite directions with very little clearance between them. The rotors are covered by cooling jackets. Two shafts on the rotors are placed that transfer their motion with the help of timing gears that are attached at the starting point of the shafts/compressor(as shown in the image).


Discharge
Suction



Rotary Compressor

Working principle-Air sucked in at one end and gets trapped between the rotors and get pushed to other side of the rotors .The air is pushed by the rotors that are rotating in opposite direction and compression is done when it gets trapped in clearance between the two rotors. Then it pushed towards pressure side.Rotary screw compressors are of two types oil-injected and oil-free.

Oil-injected is cheaper and most common than oil-free rotary screw compressors.

Advantages

Less noisy in operation.
More efficient compared to reciprocating air compressors.
Supplies large amount of compressed air.
The air supply is continuous.
Relatively low end temperature of compressed air.

Disadvantages

Expensive than reciprocating (piston-type) compressors.
More complex design.
Maintenance is difficult.

1.A single stage reciprocating compressor takes 1m^3 of air per minute at 1.013 bar and 15^{o}C and delivers it at 7 bar. Assuming that the law of compression is $P_v^{1.35}$= constant, and the clearance is negligible, calculate the indicated power?

Solution

Volume of air taken in, $V_1 = 1$ m^3/ min

Intake pressure, $p_1 = 1.013$ bar

Initial temperature, $T_1 = 15 + 273 = 288$ K

Delivery pressure, $P_2 = 7$ bar

Law of compression: $P_V^{1.35}$ = constant

Indicated power I.P.:

Mass of air delivered per min.,

$$m = \frac{p_1 v_1}{RT_1} = \frac{1.013\times10^5\times1}{287\times288} = 1.266 kg/\min$$

Delivery temperature, $T_2 = T_1\left(\frac{p2}{p1}\right)^{(n-1/n)}$

$$= 288\left(\frac{7}{1.013}\right)^{(1.35-1)/1.35} = 475.2K$$

Indicated work $= \frac{n}{n-1} mR(T_2 - T_1) kJ/\min$

$$= \frac{1.35}{1.35-1}\times 1.226\times 0.287(475.2-288 = 254 kJ/\min$$

i.e., Indicated power I.P = $\frac{254}{60} = 4.23kW.(Ans)$

2.An air compressor cylinder has 150mm bore and 150mm stroke and the clearance is 15%. It operates between 1 bar, 27°C and 5 bar. Take polytrophic exponent n=1.3 for compression and expansion processes find?

i. **Cylinder volume at the various salient points of in cycle.**

ii. **Flow rate in m^3/min at 720 rpm and .**

iii. **The deal volumetric efficiency.**

Given

$D = 150\times10^{-3}m$	$P_2 = 5\times10^5\ N/m^2$
$L = 150\times10^{-3}\ m$	$T_1 = 27+273 = 300K$
$V_c = 0.15V_s$	$N = 720rpm$
$P_1 = 1\times10^5\ N/m^2$	$pv^n = Cn=1.3$

Find

i. V_1, V_2, V_3, V_4

ii. FAD (V_a)

iii. η_v

Solution

$V_1 = V_c + V_s$

$$V_s = \frac{\pi}{4} D^2 L.N = \frac{\pi}{4}(0.15)^2 \times .0.15 \times 720 = 1.9085 m^3 / \text{min}$$

$V_c = 0.15\ V_s$

$= 0.15 \times 1.9085$

$V_c = 0.2862\ m^3/min$

$V_1 = V_c + V_s$

$= 0.2862 + 1.9085$

$\boxed{V_1 = 2.1948}$ m^3/min

$$P_1 V_1^{\,n} = P_2 V_2^{\,n}$$

$$V_2 = V_1 \left(\frac{P_1}{P_2} \right)^{1/n}$$

$$= 2.1948\left(\frac{1\times10^5}{5\times10_5}\right)^{1/1.3}$$

$$\boxed{V_2 = 0.6366\ m^3/min}$$

$$\boxed{V_3 = 0.2862\ m^3/min} = V_c$$

$V_c = V_3 \quad \therefore$

$$P_3V_3^n = P_4V_4^n$$

$$V_4 = V_3\left(\frac{P_3}{P_4}\right)^{1/4}$$

WKT

$P_2 = P_3$

$P_1 = P_4$

$$\therefore V_4 = V_3\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$= 0.2862\left[\frac{5\times10^5}{1\times10^5}\right]^{1/1.3}$$

$$\boxed{V_4 = 0.98674\ m^3/min}$$

$\therefore$ Volumetric efficiency (η_v) = $1+k-k\left(\frac{P^2}{P_1}\right)^{1/n}$

$$k = \text{Clearance Raito} = \frac{V_c}{V_s} = \frac{0.2862}{1.9085}$$

$$\boxed{K = 0.1499}$$

$$\therefore \ \eta_v = 1+0.1499 - 0.1499\left[\frac{5}{1}\right]^{1/1.3}$$

$$\eta_v = 0.633 = 63.3\%$$

$$\boxed{\eta_v = 63.3\%}$$

$\therefore$ **WKT**

$$\eta_v = \frac{FAD}{V_s}$$

$$\therefore FAD = \eta_v \times V_s$$

$$= 0.633 \times 1.9085$$

$$\boxed{FAD = 1.2083 \ m^3/min}$$

3.Calcute the diameter and stroke for a double acting single stage reciprocating air compressor of 50kW having induction pressure 100 kN/m^2 and temperature 150^{o}C. The law of compression is $PV^{1.2}$ = C and delivery pressure is 500 kN/ m^2. The revolution/sec =1.5 and mean piston speed in 150 m/min. Clearance is neglected.

Given:

Double acting single stage

Compressor

IP = 50kW

$P_1 = 100\times10^3\ N/m^2$

$T_1 = 15 + 273 = 288K$

$PV^{1.2} = C \qquad \therefore n=1.2$

$P2 = 500\times10^3\ N/m^2$

$N = 1.5$ rps $= 1.5\times60$ rpm

2LN = 150m/min (Double acting)

Find

i. D and L

Solution

For double acting compressor average piston speed = 2LN

$\therefore$ 2LN = 150 m/min

$$\therefore L= \frac{150}{2\times1.5\times60} = 0.833m$$

To Find D

IP = W.N_w

where

N_w = Number of working stroke

For Double acting N_w = 2N

For single acting N_w = N

$\therefore N_w = 2\times1.5\times60 = 180$ rpm

$$\therefore \text{W.D/cycle} = \frac{n}{n-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right]$$

$$=\frac{1.2}{1.2-1}\times100\times10^3\left(\frac{\pi}{4}D^2\times0.833\right)\times\left[\left(\frac{(500)}{100}\right)^{\frac{0.2}{1.2}}-1\right]$$

$\boxed{W = 120764.2D^2}$ N-m

$$\therefore \text{IP} = \frac{W.N_w}{60}$$

$$50\times10^3 = \frac{1207642D^2\times180}{60}$$

$D^2 = 0.1380$

$\boxed{D = 0.371 \text{ m}}$

UNIT-V

Dynamic Compressors:

Centrifugal compressors; also known as turbo-compressors belong to the roto-dynamic type of compressors. In these compressors the required pressure rise takes place due to the continuous conversion of angular momentum imparted to the refrigerant vapour by a high-speed impeller into static pressure. Unlike reciprocating compressors, centrifugal compressors are steady-flow devices hence they are subjected to less vibration and noise.

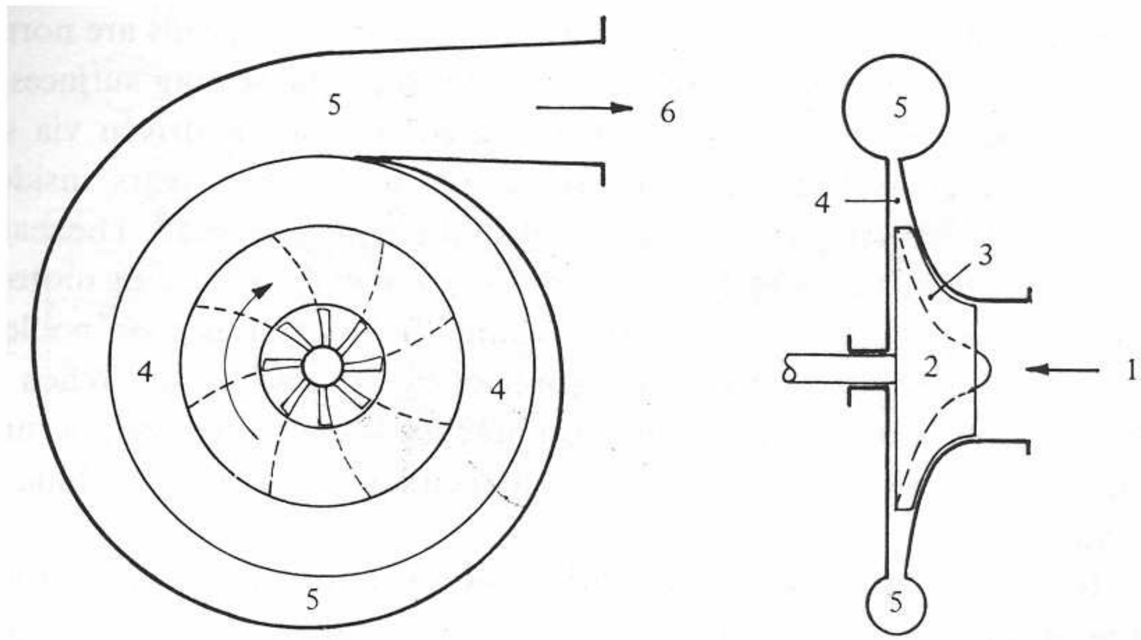
Figure 21.1 shows the working principle of a centrifugal compressor. As shown in the figure, low-pressure refrigerant enters the compressor through the eye of the impeller (1). The impeller (2) consists of a number of blades, which form flow passages (3) for refrigerant. From the eye, the refrigerant enters the flow passages formed by the impeller blades, which rotate at very high speed. As the refrigerant flows through the blade passages towards the tip of the impeller, it gains momentum and its static pressure also increases. From the tip of the impeller, the refrigerant flows into a stationary diffuser (4). In the diffuser, the refrigerant is decelerated and as a result the dynamic pressure drop is converted into static pressure rise, thus increasing the static pressure further. The vapour from the diffuser enters the volute casing (5) where further conversion of velocity into static pressure takes place due to the divergent shape of the volute. Finally, the pressurized refrigerant leaves the compressor from the volute casing (6).

The gain in momentum is due to the transfer of momentum from the high-speed impeller blades to the refrigerant confined between the blade passages. The increase in static pressure is due to the self-compression caused by the centrifugal action. This is analogous to the gravitational effect, which causes the fluid at a higher level to press the fluid below it due to gravity (or its weight). The static pressure produced in the impeller is equal to the static head, which would be produced by an equivalent gravitational column. If we assume the impeller blades to be radial and the inlet diameter of the impeller to be small, then the static head, h developed in the impeller passage for a single stage is given by:

$h = (V^2/g)$ ---- (1).

Where h = static head developed, m
V = peripheral velocity of the impeller wheel or tip speed, m/s
g = acceleration due to gravity, m/s^2
Hence increase in total pressure, ΔP as the refrigerant flows through the passage is given by:

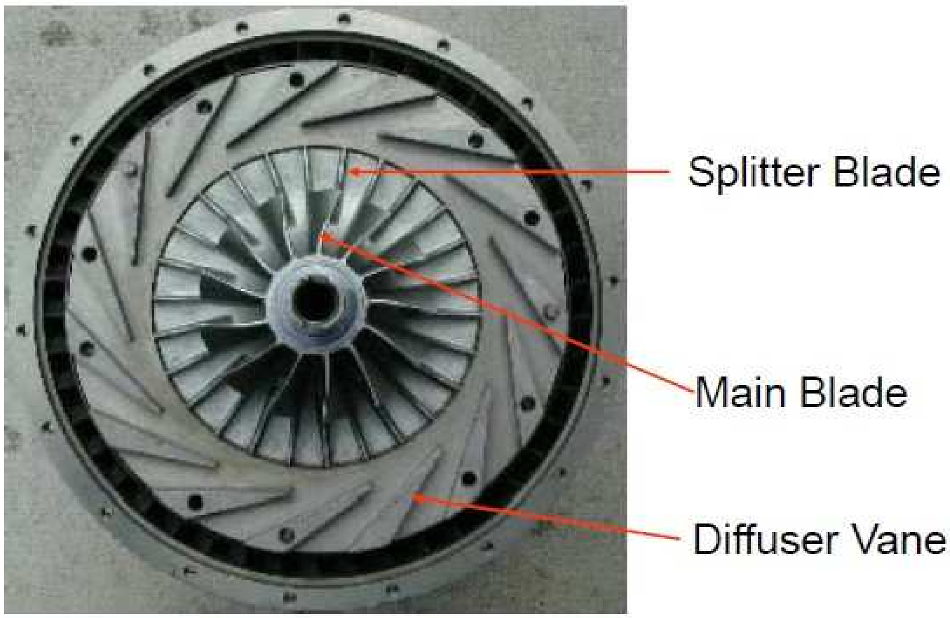
$\Delta P=\rho gh=\rho V^2$ ----- (2)



Centrifugal Compressor

1: Eye; 2: Impeller; 3: Air passages 4: Vane less diffuser; 5: Volute casing; 6: Air discharge

Radial Impeller with Diffuser Vanes



Thus it can be seen that for a given mass flow, the pressure rise depends only on the peripheral velocity or tip speed of the blade. The tip speed of the blade is proportional to the rotational speed (RPM) of the impeller and the impeller diameter. The maximum permissible tip speed is limited by the strength of the structural materials of the blade (usually made of high speed chrome-nickel steel) and the sonic velocity of the fluid. Under these limitations, the maximum achievable pressure rise (hence maximum achievable temperature lift) of single stage centrifugal compressor is limited for a given refrigerant. Hence, multistage centrifugal compressors are used for large temperature lift applications. In multistage centrifugal compressors, the discharge of the lower stage compressor is fed to the inlet of the next stage compressor and so on. In multistage centrifugal compressors, the impeller diameter of all stages remains same, but the width of the impeller becomes progressively narrower in the direction of flow as refrigerant density increases progressively.

The blades of the compressor or either forward curved or backward curved or radial. Backward curved blades were used in the older compressors, whereas the modern centrifugal compressors use mostly radial blades. The stationary diffuser can be vaned or vaneless. As the name implies, in vaned diffuser vanes are used in the diffuser to form flow passages. The vanes can be fixed or adjustable. Vaned diffusers are compact compared to the vaneless diffusers and are commonly used for high discharge pressure applications. However, the presence of vanes in the diffusers can give rise to shocks, as the refrigerant velocities at the tip of the impeller blade could reach sonic velocities in large, high-speed centrifugal compressors. In vaneless diffusers the velocity of refrigerant in the diffuser decreases and static pressure increases as the radius increases. As a result, for a required pressure rise, the required size of the vaneless diffuser could be large compared to vaned diffuser. However, the problem of shock due to supersonic velocities at the tip does not arise with vaneless diffusers as the velocity can be diffused smoothly.

Analysis of centrifugal compressors:

Applying energy balance to the compressor, we obtain from steady flow energy equation:

$$-Q + m\left(h_i + \frac{V_i^2}{2} + gZ_i\right) = -W_c + m\left(h_e + \frac{V_e^2}{2} + gZ_e\right)$$

Where
Q = heat transfer rate from the compressor
W = work transfer rate to the compressor
m = mass flow rate of the refrigerant
V_i,V_e = Inlet and outlet velocities of the refrigerant
Z_i,Z_e = Height above a datum in gravitational force field at inlet and outlet
Neglecting changes in kinetic and potential energy, the above equation becomes:

$$-Q + mh_i = -W_c + mh_e$$

In a centrifugal compressor, the heat transfer rate Q is normally negligible (as the area available for heat transfer is small) compared to the other energy terms, hence the rate of compressor work input for adiabatic compression is given by:

$$W_c = m(h_e - h_i)$$

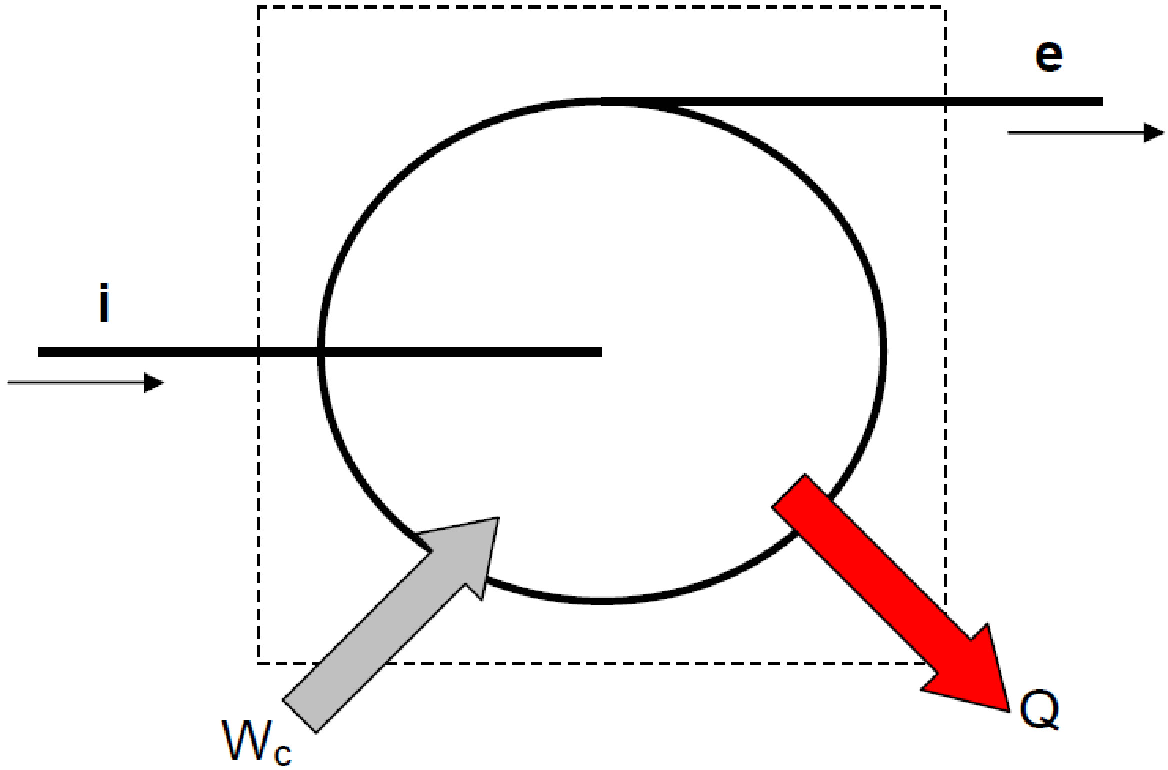
The above equation is valid for both reversible as well as irreversible adiabatic compression, provided the actual enthalpy is used at the exit in case of irreversible compression. In case of reversible, adiabatic compression, the power input to the compressor is given by:

$$W_{c,isen} = m(h_e - h_i)_{isen}$$

Then using the thermodynamic relation, *Tds=dh–vdp*; the isentropic work of compression is given by:

$$w_{c,isen} = (h_e - h_i)_{isen} = \int_{Pi}^{Pe} vdp \Big|_{isen}$$

Thus the expression for reversible, isentropic work of compression is same for both reciprocating as well as centrifugal compressors. However, the basic difference between actual reciprocating compressors and actual centrifugal compressors lies in the source of irreversibility.



Energy balance across a compressor

In case of reciprocating compressors, the irreversibility is mainly due to eat transfer and pressure drops across valves and connecting pipelines. However, in case of centrifugal compressors, since the fluid has to flow at entry with high velocities through the impeller blade passages for a finite pressure rise, the major source of irreversibility is due to the viscous shear stresses at the interface between the fluid and the impeller blade surface.

In reciprocating compressors, the work is required to overcome the normal forces acting against the piston, while in centrifugal compressors; work is required to overcome both normal pressure forces as well as viscous shear forces. The specific work is higher than the area of P-v diagram in case of centrifugal compressors due to irreversibilities and also due to the continuous increase of specific volume of refrigerant due to fluid friction.

To account for the irreversibilities in centrifugal compressors, a polytropic efficiency η_{pol} is defined. It is given by:

$$\eta_{pol} = \frac{w_{pol}}{w_{act}} = \frac{\int_{Pi}^{Pe} vdp}{(h_e - h_i)}$$

Where w_{pol} and w_{act} are the polytropic and actual works of compression, respectively. The polytropic work of compression is usually obtained by the expression:

$$w_{pol} = \int_{Pi}^{Pe} vdP = f\left(\frac{n}{n-1}\right)Pivi\left[\left(\frac{Pe}{Pi}\right)^{\frac{n-1}{n}} - 1\right]$$

Where n is the index of compression, f is a correction factor which takes into account the variation of n during compression. Normally the value of f is close to 1 (from 1.00 to 1.02), hence it may be neglected in calculations, without significant errors.

If the refrigerant vapour is assumed to as an ideal gas, then it can be shown that the polytropic efficiency is equal to:

$$\eta_{pol} = \left(\frac{n}{n-1}\right)\left(\frac{\gamma - 1}{\gamma}\right)$$

Losses in a Centrifugal Compressor

The losses in a centrifugal compressor are almost of the same types as those in a centrifugal pump. However, the following features are to be noted.

Frictional losses: A major portion of the losses is due to fluid friction in stationary and rotating blade passages. The flow in impeller and diffuser is decelerating in nature. Therefore the frictional losses are due to both skin friction and boundary layer separation. The losses depend on the friction factor, length of the flow passage and square of the fluid velocity. The variation of frictional losses with mass flow is shown in Figure. 8.1.

Incidence losses: During the off-design conditions, the direction of relative velocity of fluid at inlet does not match with the inlet blade angle and therefore fluid cannot enter the blade passage smoothly by gliding along the blade surface. The loss in energy that takes place because of this is known as incidence loss. This is sometimes referred to as shock losses. However, the word shock in this context should not be confused with the aerodynamic sense of shock which is a sudden discontinuity in fluid properties and flow parameters that arises when a supersonic flow decelerates to a subsonic one.

Clearance and leakage losses: Certain minimum clearances are necessary between the impeller shaft and the casing and between the outlet periphery of the impeller eye and the casing. The leakage of gas through the shaft clearance is minimized by employing glands. The clearance losses depend upon the impeller diameter and the static pressure at the impeller tip. A larger diameter of impeller is necessary for a higher peripheral speed (U_2) and it is very difficult in the situation to provide sealing between the casing and the impeller eye tip.

The variations of frictional losses, incidence losses and the total losses with mass flow rate are shown in Figure.8.1

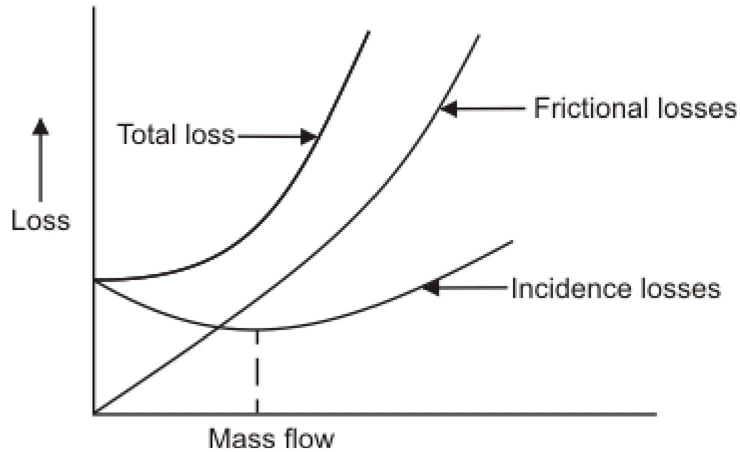


Figure 8.1 Dependence of various losses with mass flow in a centrifugal compressor

The leakage losses comprise a small fraction of the total loss. The incidence losses attain the minimum value at the designed mass flow rate. The shock losses are, in fact zero at the designed flow rate. However, the incidence losses, as shown in Fig. 8.1, comprises both shock losses and impeller entry loss due to a change in the direction of fluid flow from axial to radial direction in the vaneless space before entering the impeller blades. The impeller entry loss is similar to that in a pipe bend and is very small compared to other losses. This is why the incidence losses show a non zero minimum value (Figure. 8.1) at the designed flow rate.

forward-curved blades.

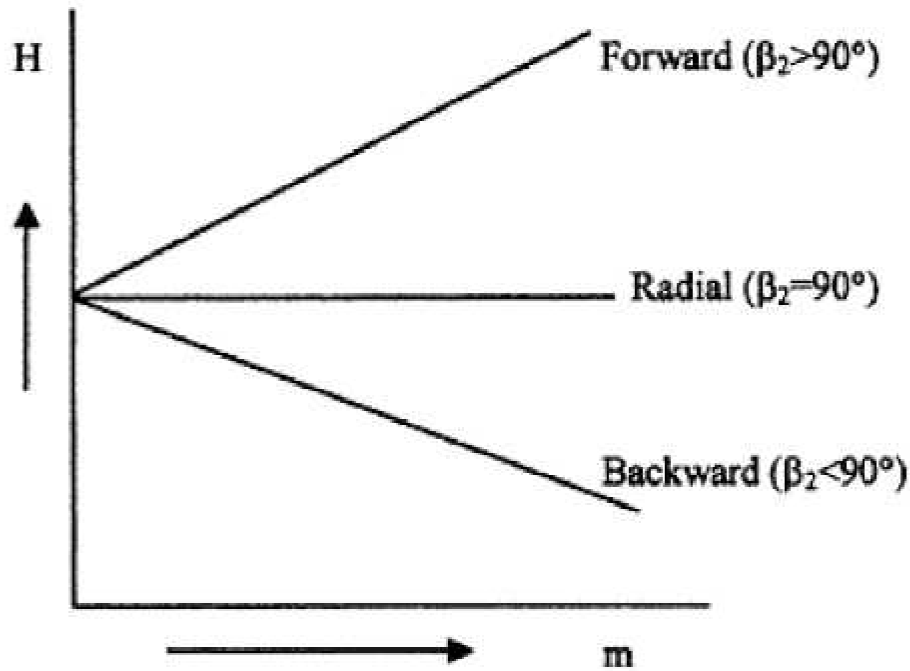
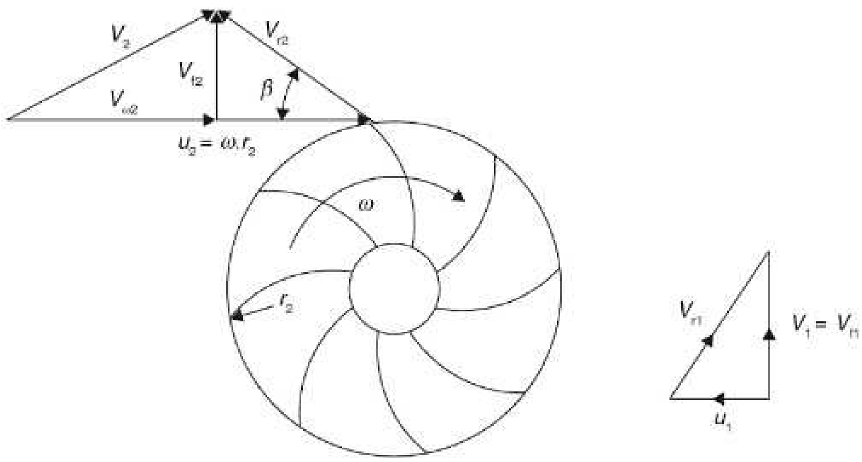


Figure 4.4 Pressure ratio or head versus mass flow or volume flow, for the three blade shapes.



(a) Velocity triangle at outlet (b) Velocity triangle at inlet

Velocity Diagram at the Outlet of the Impeller of a Centrifugal Compressor

the Figure low pressure enters the compressor through the eye of the impeller. The impeller consists of a number of blades, which form flow passages for gas.

From the eye, the gas enters the flow passages formed by the impeller blades, which rotate at very high speed. As the gas flows through the blade passages towards the tip of the impeller, it gains momentum and its static pressure also increases. From the tip of the impeller, the gas flows into a stationary diffuser. In the diffuser, the gas is decelerated and as a result the dynamic pressure drop is converted into static pressure rise, thus increasing the static pressure further. The gas from the diffuser enters the volute casing where further conversion of velocity into static pressure takes place due to the divergent shape of the volute. Finally, the pressurized gas leaves the compressor from the volute casing. The velocity triangle for the centrifugal compressor is shown in Figure

Here,

- V = Absolute velocity of gas;
- u = Blade velocity;
- V_r = Relative velocity;
- V_ω = Whirl component of absolute velocity; and
- V_f = Flow or normal component of absolute velocity.

Further, suffix 1 and 2 represent the conditions at inlet and outlet of the impeller.

For inlet velocity diagram, it has been assumed that gas enters the impeller eye in an axial direction, i.e., the whirl component of absolute velocity, $V_{\omega 1}$ is zero. Flow component of absolute velocity, $V_{f1} = V_1$ (Figure 10.10b).

In general, we consider the flow of a gas through a rotor of any shape; the rate of change of angular momentum is given by $(V_{\omega 2}r_2 - V_{\omega 1}r_1)$ m/skg.

$$\text{Work done} = (V_{\omega 2}r_2 - V_{\omega 1}r_1)\,\omega, \text{ as } V_{\omega 1} = 0$$

$$V_{\omega 2}r_2\,\omega = V_{\omega 2}u_2 \text{ J/kg.}$$

It has been observed that for backward curved vanes ($\beta < 90°$), the tangential component of absolute velocity is much reduced and consequently for a given impeller speed, the impeller will have a low energy transfer. In case of forward curved vanes ($\beta > 90°$), the tangential component

Centrifugal-flow compressors have the following advantages:

High pressure rise per stage.

Efficiency over wide rotational speed range.

Simplicity of manufacture with resulting low cost.

Low weight.

Low starting power requirements.

They have the following disadvantages:

Large frontal area for given airflow.

Impracticality if more than two stages because of losses in turns between stages.

Applications:

- In gas turbines and auxiliary power units.
- In automotive engine and diesel engine turbochargers and superchargers.
- In pipeline compressors of natural gas to move the gas from the production site to the consumer.

- Air-conditioning and refrigeration and HVAC: Centrifugal compressors quite often supply the compression in water chillers cycles.
- In industry and manufacturing to supply compressed air for all types of pneumatic tools.

Slip factor:

The slip factor is a measure of the fluid slip in the impeller of a centrifugal compressor. Fluid slip is the deviation in the angle at which the fluid leaves the impeller from the impeller's blade/vane angle.

Slip factor denoted by 'σ' is defined as the ratio of the actual & ideal values of the whirl velocity components at the exit of impeller. The ideal values can be calculated using analytical approach while the actual values should be observed experimentally.

where,

σ = /

V'w2 : Actual Whirl Velocity Component ,

Vw2 : Ideal Whirl Velocity Component

Usually, σ varies from 0-1 with an average ranging from 0.8-0.9 .

The Slip Velocity is given as:

Vs= Vw2- V'w2 = Vw2(1-σ)

The Whirl Velocity is given as:

V'w2 = σ Vw2

Power Input Factor

The power input factor takes into account of the effect of disk friction, windage, etc. for which a little more power has to be supplied than required by the theoretical expression. Considering all these losses, the actual work done (or energy input) on the air per unit mass becomes

$$w = \Psi \sigma U_2^2 \qquad (1)$$

Where Ψ is the power input factor. From steady flow energy equation and in consideration of air as an ideal gas, one can write for adiabatic work *w* per unit mass of air flow as

$$w = c_p (T_{0_2} - T_{0_1}) \qquad (2)$$

where T_{0_1} and T_{0_2} are the stagnation temperatures at inlet and outlet of the impeller, and c_p is the mean specific heat over the entire temperature range. With the help of Eq. (6.3), we can write

$$w = \Psi \sigma U_2^2 = c_p (T_{0_2} - T_{0_1}) \qquad (3)$$

The stagnation temperature represents the total energy held by a fluid. Since no energy is added in the diffuser, the stagnation temperature rise across the impeller must be equal to that across the whole compressor. If the stagnation temperature at the outlet of the diffuser is designated by T_{0_3}, then $T_{0_3} = T_{0_2}$. One can write from Eqn. (3)

$$\frac{T_{0_2}}{T_{0_1}} = \frac{T_{0_3}}{T_{0_1}} = 1 + \frac{\Psi \sigma U_2^2}{c_p T_{0_1}} \qquad (4)$$

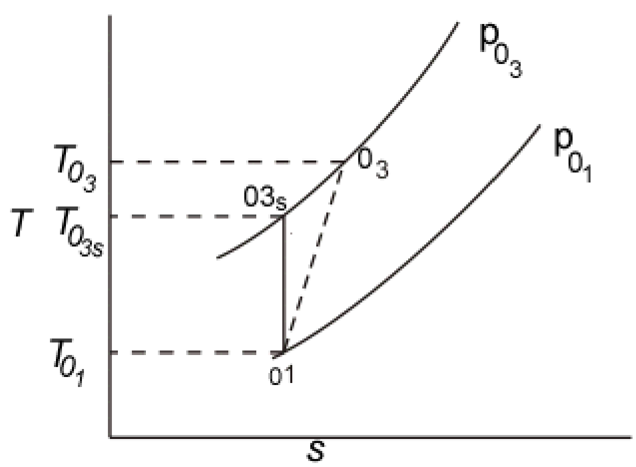
The overall stagnation pressure ratio can be written as

$$\frac{p_{0_3}}{p_{0_1}} = \left(\frac{T_{0_{3s}}}{T_{0_1}} \right)^{\frac{\gamma}{\gamma - 1}}$$

$$= \left[1 + \frac{\eta_c (T_{0_3} - T_{0_1})}{T_{0_1}} \right]^{\frac{\gamma}{\gamma - 1}} \qquad (5)$$

where, $T_{0_{3s}}$ and T_{0_3} are the stagnation temperatures at the end of an ideal (isentropic) and actual process of compression respectively, and η_c is the isentropic efficiency defined as

$$\eta_c = \frac{T_{0_{3s}} - T_{0_1}}{T_{0_3} - T_{0_1}} \qquad (6)$$



Ideal and actual processes of compression on T-s plane

Pressure coefficient

The pressure coefficient is a parameter for studying the flow of incompressible fluids such as water, and also the low-speed flow of compressible fluids such as air. The relationship between the dimensionless coefficient and the dimensional numbers is

$$C_p = \frac{p - p_\infty}{\frac{1}{2}\rho_\infty V_\infty^2} = \frac{p - p_\infty}{p_0 - p_\infty}$$

where:

p is the static pressure at the point at which pressure coefficient is being evaluated

p_∞ is the static pressure in the freestream (i.e. remote from any disturbance)

p_0 is the stagnation pressure in the freestream (i.e. remote from any disturbance)

ρ_∞ is the freestream fluid density (Air at sea level and 15 °C is 1.225 $\mathrm{kg/m^3}$)

V_∞ is the freestream velocity of the fluid, or the velocity of the body through the fluid

The pressure coefficient is is the ratio of ***pressure forces*** to ***inertial forces*** and can be expressed as

$C_p = dP / (\rho\ v^2 / 2)$

$= dh\ (\rho\ v^2 / 2\ g)$ (1)

where

C_p = pressure coefficient

dp = pressure difference (N)

ρ = fluid density (kg/m^3)

v = flow velocity (m/s)

dh = head (m)

g = acceleration of gravity (= 9.81 m/s^2)

Differences between Centrifugal and Axial Flow Compressors

S.no	Centrifugal Compressors	Axial Flow Compressors
1	In centrifugal compressors air flows radially in the compressor	In Axial flow compressors air flows parallel to the axis of shaft
2	Low maintenance and running cost	High maintenance and running cost
3	Low starting torque is required	Requires high starting torque
4	Not suitable for multi staging	Suitable for multi staging
5	Suitable for low pressure ratios up to 4	Suitable for only multi staging ratio of 10
6	For given mass flow rate, it requires a larger frontal area.	For a given mass flow rate, it requires less Frontal area.
7	Isentropic efficiency is 80 to 82%	Isentropic efficiency is 86 to 88%
8	Better performance at part load	Poor performance at part load

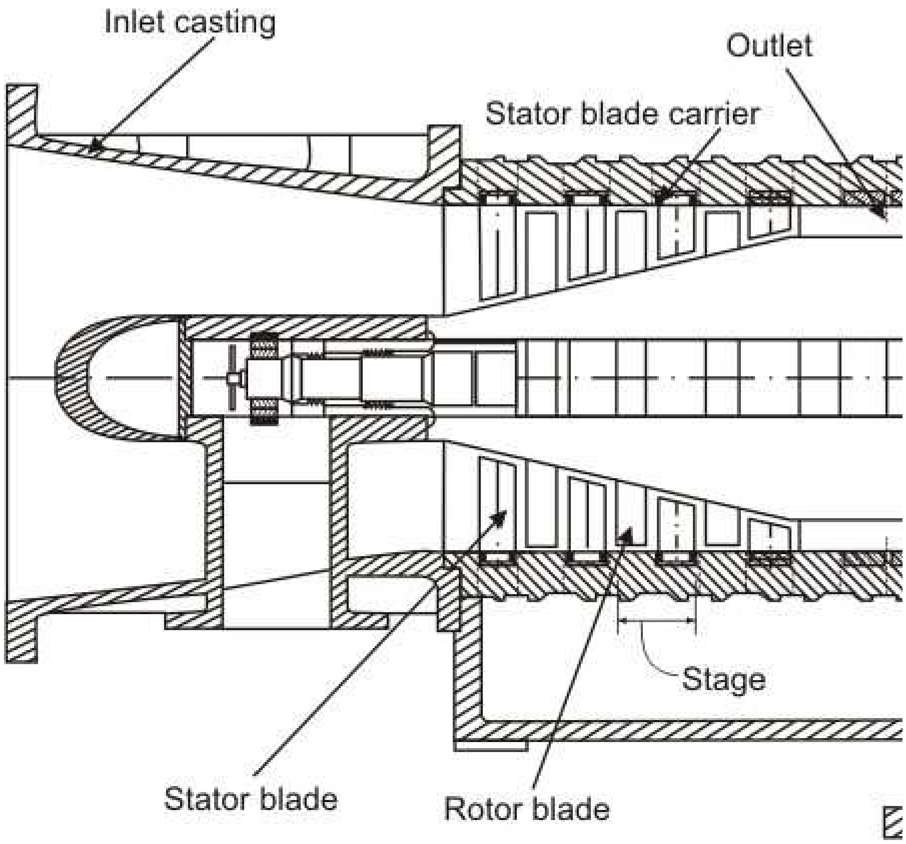
Comparison between Reciprocating and Centrifugal Compressors

S.no	Reciprocating Compressors	Centrifugal Compressors
1	Reciprocating compressors have poor mechanical efficiency due to large sliding parts.	Centrifugal compressors have better mechanical efficiency due to absence of sliding parts.
2	Installation cost for setting up reciprocating compressors is higher.	Installation cost for setting up centrifugal compressors is lower.
3	Reciprocating compressors produce greater noise and vibrations.	Centrifugal compressors have comparatively salient operation.
4	Pressure ratio up to 5 to 8.	Pressure ratio up to 4.
5	Higher pressure ratio up to 500 atmosphere is possible with multistage of compressor.	It is not suitable for multistage.
6	It runs intermittently and delivers pulsating air.	It runs continuously and delivers steady and pulsating free air.
7	Less amount of volume is handled.	Large amount of volume is handled.
8	More maintenance is required.	Less maintenance is required.
9	Weight of reciprocating compressor is more.	Less weight compared to other compressors.

10	Reciprocating compressor operates at low speed.	Centrifugal compressors operates at high speed.
11	Isothermal efficiency should be better.	Isentropic efficiency should be better.
12	Higher compression efficiency at pressure ratio more than 2.	Higher compression efficiency, if pressure ratio is less than 2.
13	Suitable for low discharge and high pressure ratio.	Suitable for high discharge and low pressure ratio.

Axial Flow Compressors:

The basic components of an axial flow compressor are a rotor and stator, the former carrying the moving blades and the latter the stationary rows of blades. The stationary blades convert the kinetic energy of the fluid into pressure energy, and also redirect the flow into an angle suitable for entry to the next row of moving blades. Each stage will consist of one rotor row followed by a stator row, but it is usual to provide a row of so called inlet guide vanes. This is an additional stator row upstream of the first stage in the compressor and serves to direct the axially approaching flow correctly into the first row of rotating blades. For a compressor, a row of rotor blades followed by a row of stator blades is called a stage. Two forms of rotor have been taken up, namely drum type and disk type. A disk type rotor illustrated in Figure 1. The disk type is used where consideration of low weight is most important. There is a contraction of the flow annulus from the low to the high pressure end of the compressor. This is necessary to maintain the axial velocity at a reasonably constant level throughout the length of the compressor despite the increase in density of air. Figure 9.2 illustrate flow through compressor stages. In an axial compressor, the flow rate tends to be high and pressure rise per stage is low. It also maintains fairly high efficiency.



axial flow compressor

The basic principle of acceleration of the working fluid, followed by diffusion to convert acquired kinetic energy into a pressure rise, is applied in the axial compressor. The flow is considered as occurring in a tangential plane at the mean blade height where the blade peripheral velocity is U. This two dimensional approach means that in general the flow velocity will have two components, one axial and one peripheral denoted by subscript w, implying a whirl velocity. It is first assumed that the air approaches the rotor blades with an absolute velocity, V_1, at and angle α_1 to the axial direction. In combination with the peripheral velocity U of the blades, its relative velocity will be V_{r1} at and angle β_1 as shown in the upper velocity triangle (Figure 9.3). After passing through the diverging passages formed between the rotor blades which do work on the air and increase its absolute velocity, the air will emerge with the relative velocity of V_{r2} at angle β_2 which is less than β_1. This turning of air towards the axial direction is, as previously mentioned, necessary to provide an increase in the effective flow area and is brought about by the camber of the blades. Since V_{r2} is less than V_{r1} due to diffusion, some pressure rise has been accomplished in the rotor. The velocity V_{r2} in combination with *U* gives the absolute velocity V_2 at the exit from the rotor at an angle α_2 to the axial direction. The air then passes through the passages formed by the stator blades where it is further diffused to velocity V_3 at an angle α_3 which in most designs equals to α_1 so that it is prepared for entry to next stage. Here again, the turning of the air towards the axial direction is brought about by the camber of the blades.

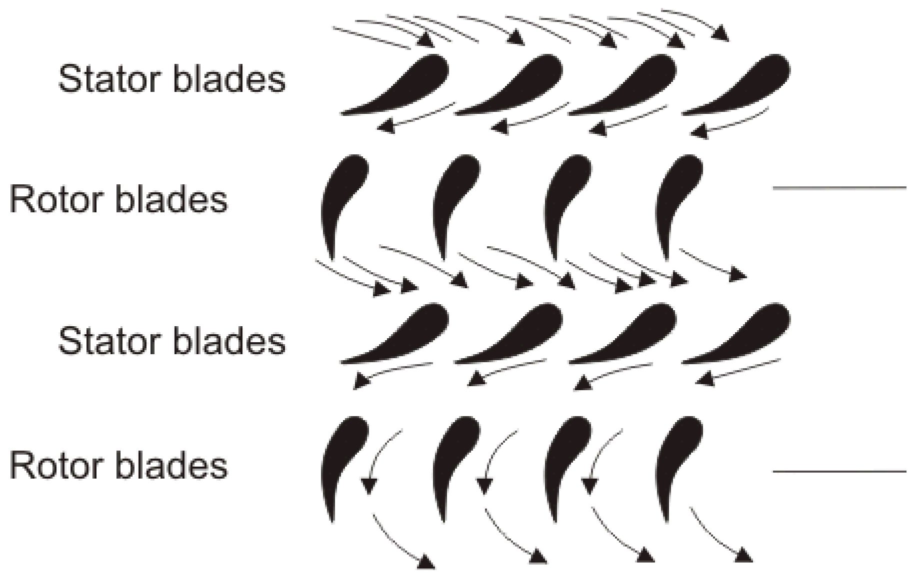
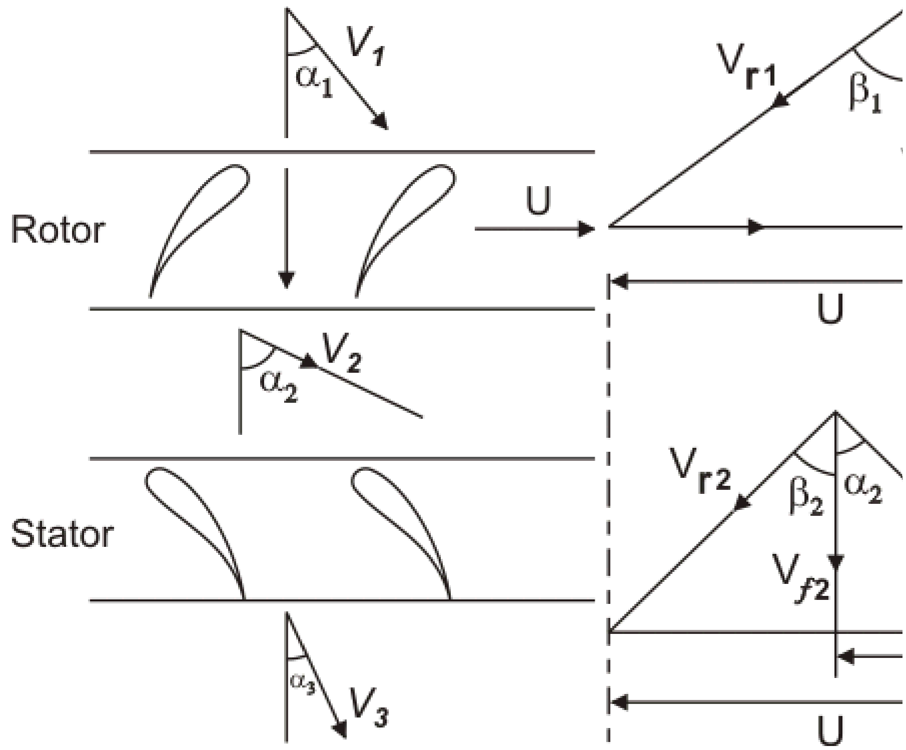


Figure 2 Flow through stages



Velocity triangles

Two basic equations follow immediately from the geometry of the velocity triangles. These are:

$$\frac{U}{V_f} = \tan\alpha_1 + \tan\beta_1 \qquad (1)$$

$$\frac{U}{V_f} = \tan\alpha_2 + \tan\beta_2 \qquad (2)$$

In which $V_f = V_{f1} = V_{f2}$ is the axial velocity, assumed constant through the stage. The work done per unit mass or specific work input, *w* being given by

$$w = U(V_{w2} - V_{w1}) \qquad (3)$$

This expression can be put in terms of the axial velocity and air angles to give

$$w = UV_f(\tan\alpha_2 - t\,an\,\alpha_1) \qquad (4)$$

or by using Eqs. (9.1) and (9.2)

$$w = UV_f(\tan\beta_1 - t\text{an}\,\beta_2) \tag{5}$$

This input energy will be absorbed usefully in raising the pressure and velocity of the air. A part of it will be spent in overcoming various frictional losses. Regardless of the losses, the input will reveal itself as a rise in the stagnation temperature of the air ΔT_0. If the absolute velocity of the air leaving the stage V_3 is made equal to that at the entry. V_1, the stagnation temperature rise ΔT_0 will also be the static temperature rise of the stage, ΔT_s, so that

$$\Delta T_0 = \Delta T_s = \frac{UV_f}{c_p}(\tan\beta_1 - \tan\beta_2) \tag{6}$$

In fact, the stage temperature rise will be less than that given in Eq. (9.6) owing to three dimensional effects in the compressor annulus. Experiments show that it is necessary to multiply the right hand side of Eq. (9.6) by a work-done factor λ which is a number less than unity. This is a measure of the ratio of actual work-absorbing capacity of the stage to its ideal value.

The radial distribution of axial velocity is not constant across the annulus but becomes increasingly peaky (Figure. 9.4) as the flow proceeds, settling down to a fixed profile at about the fourth stage. Equation (9.5) can be written with the help of Eq. (9.1) as

$$\begin{aligned} w &= U[(U - V_f \tan\alpha_1) - V_f \tan\beta_2] \qquad (7) \\ &= U(U - V_f(\tan\alpha_1 + \tan\beta_2)) \end{aligned}$$

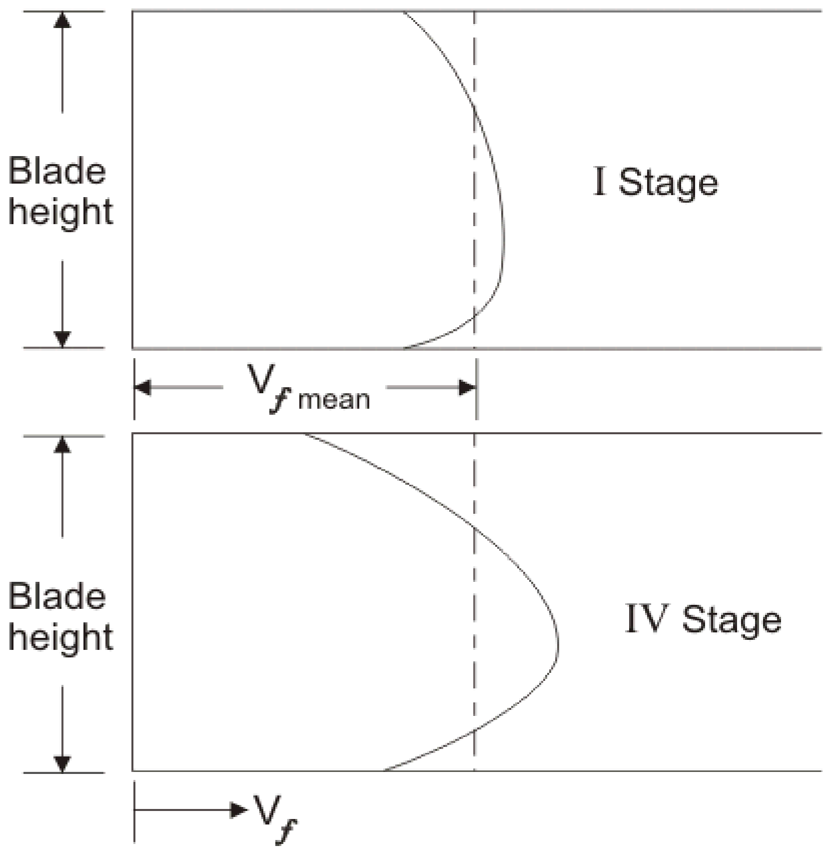


Figure 4 Axial velocity distibutions

Since the outlet angles of the stator and the rotor blades fix the value of α_1 and β_2 and hence the value of . Any increase in $V_f(\tan\alpha_1 + \tan\beta_2)$ V_f will result in a decrease in ω and vice-versa. If the compressor is designed for constant radial distribution of as shown by the dotted line in Figure (9.4), the effect of an increase in V_f in the central region of the annulus will be to reduce the work capacity of blading in that area. However this reduction is somewhat compensated by an increase in ω in the regions of the root and tip of the blading because of the reduction of V_f *at* these parts of the annulus. The net result is a loss in total work capacity because of the adverse effects of blade tip clearance and boundary layers on the annulus walls. This effect becomes more pronounced as the number of stages is increased and the way in which the mean value varies with the number of stages. The variation of λ with the number of stages is shown in Figure. 9.5. Care should be taken to avoid confusion of the work done factor with the idea of an efficiency. If ω is the expression for the specific work input (Equation. 9.3), then $\lambda\varpi$ is the actual amount of work which can be supplied to the stage. The application of an isentropic efficiency to the resulting temperature rise will yield the equivalent isentropic temperature rise from which the stage pressure ratio may be calculated. Thus, the actual stage temperature rise is given by

$$\Delta T_0 = \frac{\lambda U V_f}{c_p}(\tan\beta_1 - \tan\beta_2) \qquad (8)$$

and the pressure ratio R_s by

$$R_s = \left[1 + \frac{\eta_s \Delta T_0}{T_{0_1}}\right]^{\frac{\gamma}{\gamma-1}} \qquad (9)$$

where, T_{0_1} is the inlet stagnation temperature and η_s is the stage isentropic efficiency.

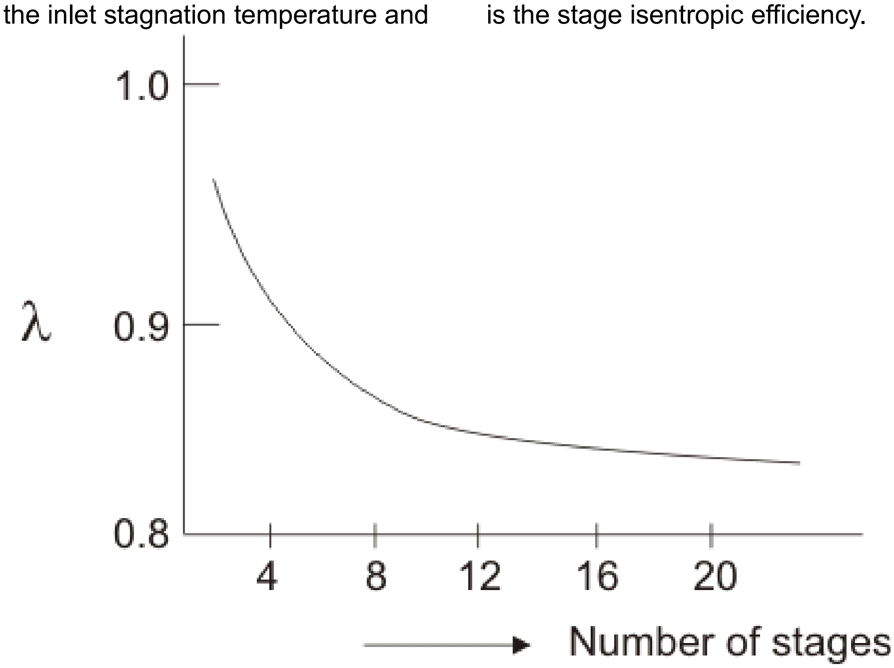


Figure 5 Variation of work-done factor with number of stages

Degree of reaction

Diffusion takes place in both rotor and the stator.

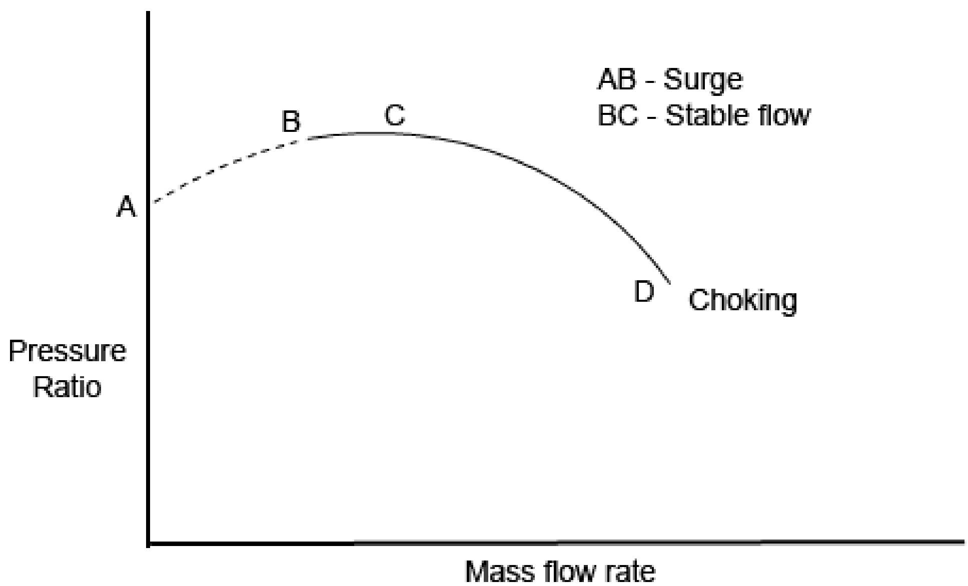
Static pressure rises in the rotor as well as the stator.

Degree of reaction provides a measure of the extent to which the rotor contributes to the overall pressure rise in the stage.

Degree of reaction

$$R_x = \frac{\text{Static enthalpy rise in the rotor}}{\text{Stagnation enthalpy rise in the stage}}$$
$$= \frac{h_2 - h_1}{h_{03} - h_{01}} \approx \frac{h_2 - h_1}{h_{02} - h_{01}}$$

Surging:



Surge is a complete breakdown of steady flow through the compressor. It is defined as the lower limit of stable operation of the compressor. The course of the surge is decreasing mass flow rate and increases a rotational speed of the impeller. It imposes stress on a bearing of compressor and motor and may damage it.

Stalling:

In stall the flow direction along the wall is reversed and approaching streamlines are deflected from the surface due to the overpowering effects of viscous shear and the adverse pressure gradient. The flow then becomes reoriented and large viscous shear stresses predominate, at least locally. Noise maybe generated.

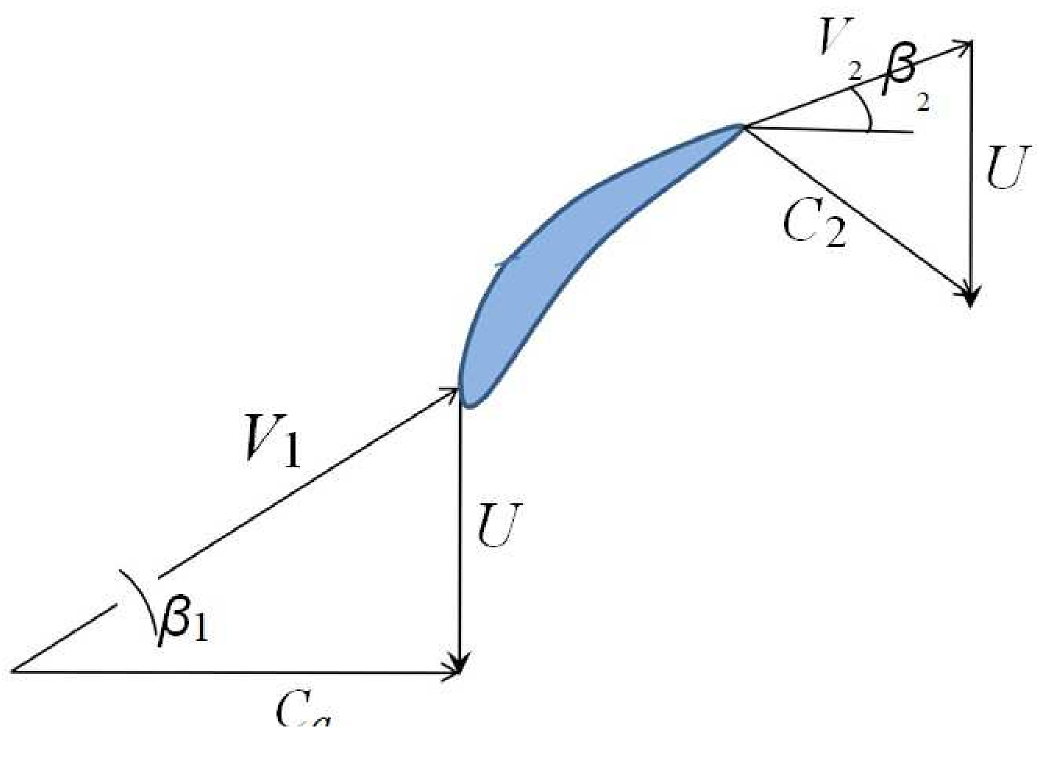
It is possible for several elements of a compressor stage to stall without the entire stage stalling or similar events occurring. When a stage either has a very strong stall in one of its elements, or a number of elements together collectively stall that the overall pressure ratio vs. flow characteristic is no longer stable (negatively sloped), then the stage has entered into a stage stall.

Choking

The choking occurs in the compressor which operates at a low discharge pressure and maximum flow rate. It is the limit of the performance curve, at this point velocity of the impeller reaches a velocity of sound of the gas at that condition within the compressor (Mach number reaches Unity). This point also is known as stonewalling of compressor.

The gas flow rate and velocity cannot go beyond value at a choke point. It normally causes no damage in single stage compressor, but causes serious damage to the multistage compressor. Artichoke valves are used to prevent shock when the flow increases to a certain value the anti-choke valve close and develops resistance to flow.

Eg. 1. Air at 1.0 bar and 288 K enters an axial flow compressor with an axial velocity of 150 m/s. There are no inlet guide vanes. The rotor stage has a tip diameter of 60 cm and a hub diameter of 50 cm and rotates at 100rps. The air enters the rotor and leaves the stator in the axial direction with no change in velocity or radius. The air is turned through 30.2 degree as it passes through the rotor. Assume a stage pressure ratio of 1.2 and overall pressure ratio of 6. Find a) the mass flow rate of air, b) the power required to drive the compressor, c) the degree of reaction at the mean diameter, d) the number of compressor stages required if the isentropic efficiency is 0.85



$$U = \pi \times \underline{d_t + d_h} \times N = \pi \times \underline{0.6 + 0.5} \times 100 = 172.76 m/s$$

$$\beta_1 = \tan^{-1} \frac{U}{C_a} = 49.2°$$

$$\beta_2 = 49.2 - 30.2 = 19°$$

$$\beta_1 = \tan^{-1} \frac{U}{C_a} = 49.2°$$

$$\beta_2 = 49.2 - 30.2 = 19°$$

$$\tan\alpha_2 \quad \frac{U - C_a \tan \beta_2}{C_a} = 80.75$$

$$\alpha_2 = 38.92°$$

$$\dot{m} = \frac{\pi}{4} \times \left(d_t^2 - d_h^2\right) \times C_2 \times \rho_2 \qquad \& \; T_1 = T_{01} - \frac{C_a^2}{2C_p} = 276.8\ K$$

$$T_{02} = T_{01} \times \left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma - 1}{\gamma}} \quad \therefore \quad T_{02} = 303.41K$$

$$T_2 = 303.41 - \frac{C_2^2}{2C_p} \quad \& \; \cos\alpha_2 = \frac{C_a}{C_2}$$

$$\therefore C_2 = \frac{C_a}{\cos\alpha_2} = \frac{150}{\cos 38.92} = 192.79 m/s$$

$$T_2 = 303.41 - \frac{192.79}{2010}^2 =$$

$$284.91 K P_2 = 1.216 bar$$

$$\rho_2 = \frac{1.216 \times 101325}{287 \times 284.9} = 1.507\ Kg/m^3$$

$$\dot{m} = 19.53\ Kg/s$$

$$P = U \times C_a \times \dot{m} \times (\tan\beta_1 - \tan\beta_2)$$

= 172.76×150×19.53× (tan 49.2 − tan19) = 412 KW

$$R_X = 1 - \frac{C_a}{2U} \times (\tan\beta_1 + \tan\beta_2)$$

$$= 1 - \frac{150}{2\times172.76} \times (\tan 49.2 + \tan 19) = 1 - 0.65 = 0.35$$

$$\Delta T_{0s} = \frac{U \times C_a}{C_p} \times (\tan\beta_1 - \tan\beta_2)$$

$$= \frac{172.76\times150}{1005} \times (\tan 49.2 - \tan 19) = 20.99\ K$$

$$\Delta T_{0overall} = \frac{T_1}{\eta_C} \times \pi_C^{\frac{\gamma-1}{\gamma}} - 1$$

$$= \frac{288}{0.85} \times (6^{0.286} - 1) = 226.5K$$

$$n = \frac{226.5}{20.99} = 10.79 \approx 11$$

Therefore the number of stages required for the given pressure ratio is 11.0.