DIGITAL NOTES

THERMAL ENGINEERING

(R17A0308)

B.Tech II Year II Semester

DEPARTMENT OF MECHANICAL ENGINEERING

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

(An Autonomous Institution – UGC, Govt.of India)

Recognizes under 2(f) and 12(B) of UGC ACT 1956

(Affiliated to JNTUH, Hyderabad, Approved by AICTE –Accredited by NBA & NAAC-“A” Grade-ISO 9001:2015 Certified)

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

(R17A0308) THERMAL ENGINEERING

COURSE OBJECTIVES:

- Introduction, Engine Types and their Operation. Application of the principles of thermodynamics to components and systems.
- Understand and describe the gas exchange and combustion processes in diesel engines. Good understanding of the various gas turbine, steam turbine and combined cycles for electricity generation.
- The purpose of this course is to enable the student to gain an understanding of how thermodynamic principles govern the behavior of various systems and have knowledge of methods of analysis and design of complicated thermodynamic systems

UNIT – I

Actual Cycles and their Analysis: Introduction, Comparison of Air Standard and Actual Cycles, Time Loss Factor, Heat Loss Factor, Exhaust Blowdown-Loss due to Gas exchange process, Volumetric Efficiency. Loss due to Rubbing Friction, Actual and Fuel-Air Cycles of CI Engines.

I.C. ENGINES : Classification - Working principles, Valve and Port Timing Diagrams, Air – Standard, air-fuel and actual cycles - Engine systems – Fuel, Carburetor, Fuel Injection System, Ignition, Cooling and Lubrication

UNIT – II

Combustion in S.I. Engines : Normal Combustion and abnormal combustion – Importance of flame speed and effect of engine variables – Type of Abnormal combustion, pre-ignition and knocking (explanation of) – Fuel requirements and fuel rating, anti knock additives – combustion chamber – requirements, types.

Combustion in C.I. Engines : Four stages of combustion – Delay period and its importance – Effect of engine variables – Diesel Knock– Need for air movement, suction, compression and combustion induced turbulence – open and divided combustion chambers and nozzles used – fuel requirements and fuel rating.

UNIT – III

Testing and Performance of IC Engines : Parameters of performance - measurement of cylinder pressure, fuel consumption, air intake, exhaust gas composition, Brake power – Determination of frictional losses and indicated power – Performance test – Heat balance sheet and chart.

UNIT-IV

Compressors – Classification –positive displacement and roto dynamic machinery – Power producing and power absorbing machines, fan, blower and compressor – positive displacement and dynamic types – reciprocating and rotary types.

Reciprocating: Principle of operation, work required, Isothermal efficiency volumetric efficiency and effect of clearance, stage compression, undercooling, saving of work, minimum work condition for stage compression.

Rotary (Positive displacement type) : Roots Blower, vane sealed compressor, Lysholm compressor – mechanical details and principle of working – efficiency considerations

UNIT – V:

Dynamic Compressors : Centrifugal compressors: Mechanical details and principle of operation – velocity and pressure variation. Energy transfer-impeller blade shape-losses, slip factor, power input factor, pressure coefficient and adiabatic coefficient – velocity diagrams – power.

Axial Flow Compressors : Mechanical details and principle of operation – velocity triangles and energy transfer per stage degree of reaction, work done factor - isentropic efficiency- pressure rise calculations – Polytropic efficiency.

COURSE OUTCOMES:

- To be able to recognize main and supplementary elements of SI and CI engines and define operational principles. To be able to describe the most important combustion concepts and problems in concern with SI engines and CI engines.

- To be able to analyze energy distribution in an internal combustion engine. Develop problem solving skills through the application of thermodynamics. Solve problems associated with Rotodynamic compressors.
- Solve problems associated with reciprocating compressors and expanders and internal combustion engines .To understand the velocity triangles in compressors

TEXT BOOKS:

1. I.C. Engines / V. GANESAN- TMH
2. Thermal Engineering / Rajput / Lakshmi Publications.
3. IC Engines – Mathur & Sharma – Dhanpath Rai & Sons.

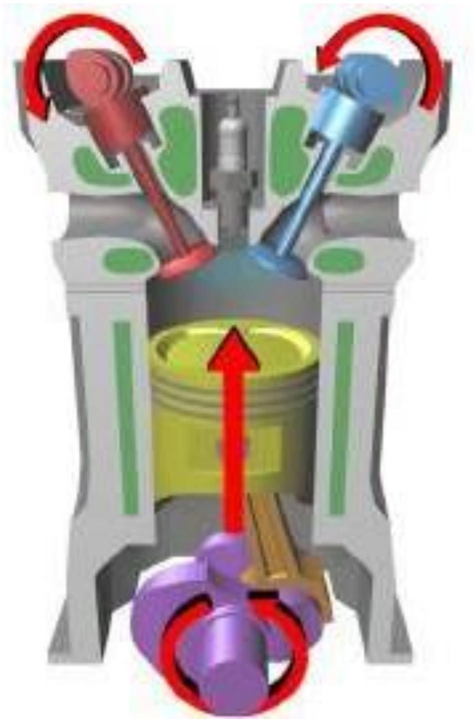
REFERENCE BOOKS:

1. Thermal Engineering / Rudramoorthy - TMH
2. Thermodynamics & Heat Engines / R.S. Yadav/ Central Book Depot., Allahabad
3. Thermal Engineering – R.S. Khurmi & J.K.Gupta – S.Chand

CONTENTS

Introduction

Course Contents



Introduction

- Once man discovered the use of heat in the form of fire, it was just a step to formulate the energy interactions. With this, human beings started to use heat energy for cooking, warming up living spaces, drying and so on.
- Further, due to the development of civilization and increase in population, man had to move from one place to another. Animals were used in transportation between the 4^{th} and 5^{th} centuries BC, and spread to Europe and other countries in the 5^{th} century BC and China in about 1200 BC.
- Gradually, man replaced the animals with motive power that was used in transportation. The use of power vehicles began in the late 18th century, with the creation of the steam engine. The invention of Otto (1876) and Diesel (1892) cycles in the 19th century transformed the method of propulsion from steam to petroleum fuel.
- **ENGINE:** Engine is a device which converts one form of Energy into another form
- **HEAT ENGINE:** Heat engine is a device which transforms the chemical energy of a fuel into thermal energy and utilizes this thermal energy to perform useful work. Thus, thermal energy is converted to mechanical energy in a heat engine.
- Heat engines can be broadly classified into two categories:

  a) Internal Combustion Engines (IC Engines)

  b) External Combustion Engines (EC Engines)

  Classification of heat engines
- Engines whether Internal Combustion or External Combustion are of two types:

  (i) Rotary engines

  (ii) Reciprocating engines
- A detailed classification of heat engines is given in Fig. 1.1.

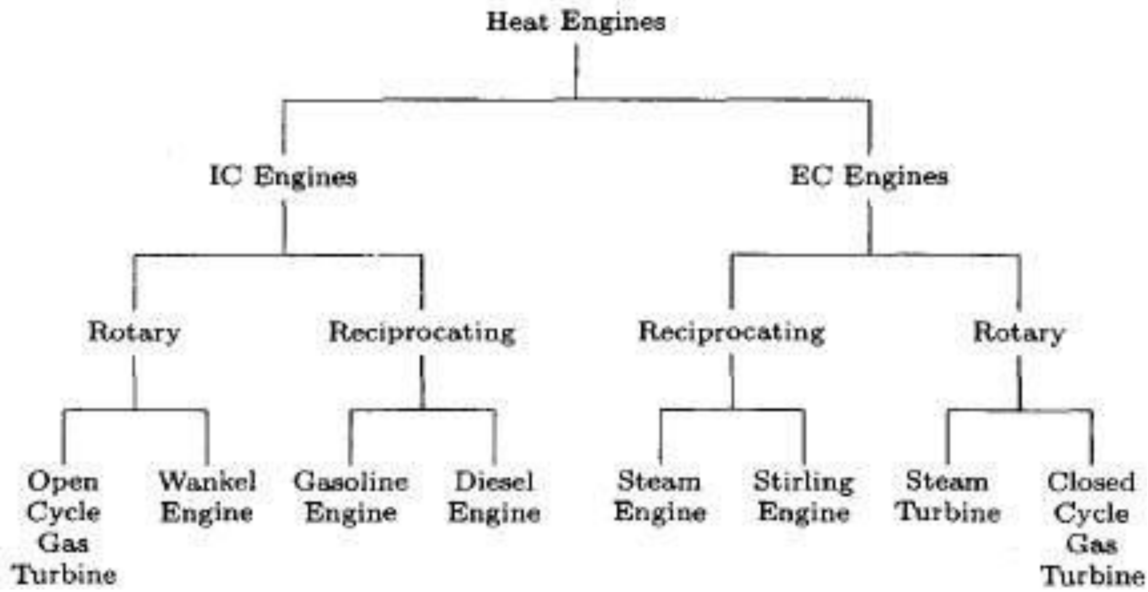


Fig 1.1 Classification of heat engines

Comparison of I.C. Engines and E.C. Engines

- Comparison of IC engine and EC engine is given in table 1.1.

Table 1.1 Comparison of IC engine and EC engine

I.C. Engine	E.C. engine
1. Combustion of fuel takes place inside the cylinder	1. Combustion of fuel takes place outside the cylinder
2. Working fluid may be Petrol, Diesel & Various types of gases	2. Working fluid is steam
3. Require less space	3. Require large space
4. Capital cost is relatively low	4. Capital cost is relatively high
5. Starting of this engine is easy & quick	5. Starting of this engine requires time
6. Thermal efficiency is high	6. Thermal Efficiency is low
7. Power developed per unit weight of these engines is high	7. Power Developed per unit weight of these engines is low
8. Fuel cost is relatively high	8. Fuel cost is relatively low

Basic components and terminology of IC engines

- Even though reciprocating internal combustion engines look quite simple, they are highly complex machines. There are many components which have to perform their functions effectively to produce output power.
- There are two types of engines, viz., spark-ignition (SI) and compression-ignition (Cl) engine.

Engine Components

- A cross section of a single cylinder spark-ignition engine with overhead valves is shown in Fig.1.2. The major components of the engine and their functions are briefly described below.

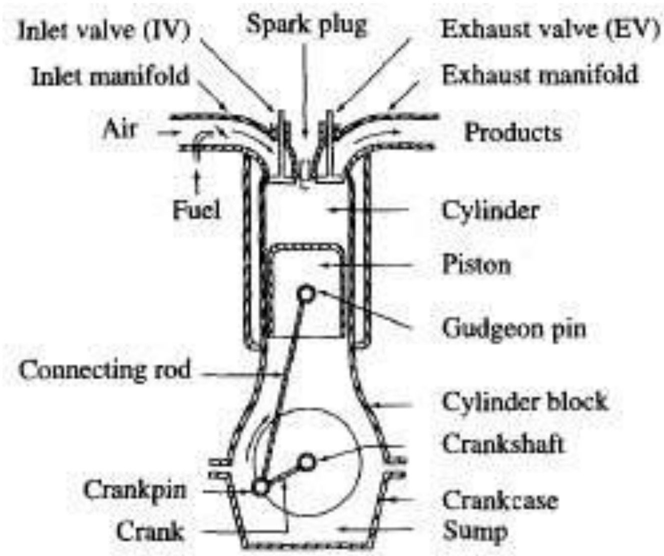


Fig. 1.2 Cross-section of spark-ignition engine

a) Cylinder block

- The cylinder block is the main supporting structure for the various components. The cylinder of a multicylinder engine are cast as a single unit, called cylinder block. The cylinder head is mounted on the cylinder block. The cylinder head and cylinder block are provided with water jackets in the case of water cooling or with cooling fins in the case of air cooling.

b) Cylinder

– As the name implies it is a cylindrical vessel or space in which the piston makes a reciprocating motion. The varying volume created in the cylinder during the operation of the engine is filled with the working fluid and subjected to different thermodynamic processes. The cylinder is supported in the cylinder block.

c) Piston

– It is a cylindrical component fitted into the cylinder forming the moving boundary of the combustion system. It fits perfectly (snugly) into the cylinder providing a gas-tight space with the piston rings and the lubricant. It forms the first link in transmitting the gas forces to the output shaft.

d) Combustion chamber

– The space enclosed in the upper part of the cylin•der, by the cylinder head and the piston top during the combustion process, is called the combustion chamber. The combustion of fuel and the consequent release of thermal energy results in the building up of pressure in this part of the cylinder.

e) Inlet manifold

– The pipe which connects the intake system to the inlet valve of the engine and through which air or air-fuel mixture is drawn into the cylinder is called the inlet manifold.

f) Exhaust manifold

– The pipe which connects the exhaust system to the exhaust valve of the engine and through which the products of combustion escape into the atmosphere is called the exhaust manifold.

g) Inlet and Exhaust valves

– Valves are commonly mushroom shaped pop•pet type. They are provided either on the cylinder head or on the side of the cylinder for regulating the charge coming into the cylinder (inlet valve) and for discharging the products of combustion (exhaust valve) from the cylinder.

h) Spark Plug

– It is a component to initiate the combustion process in Spark- Ignition (SI) engines and is usually located on the cylinder head.

i) Connecting Rod

– It interconnects the piston and the crankshaft and trans•mits the gas forces from the piston to the crankshaft. The two ends of the connecting rod are called as small end and the big end (Fig.1.3). Small end is connected to the piston by gudgeon pin and the big end is connected to the crankshaft by crankpin.

j) Crankshaft

– It converts the reciprocating motion of the piston into useful rotary motion of the output shaft. In the crankshaft of a single cylinder engine there are a pair of crank arms and balance weights. The balance weights are provided for static and dynamic balancing of the rotating system. The crankshaft is enclosed in a crankcase.

k) Piston rings

– Piston rings, fitted into the slots around the piston, provide a tight seal between the piston and the cylinder wall thus preventing leakage of combustion gases.

l) Gudgeon pin

– It links the small end of the connecting rod and the piston.

m) Camshaft

– The camshaft (not shown in the figure) and its associated parts control the opening and closing of the two valves. The associated parts are push rods, rocker arms, valve springs and tappets. This shaft also provides the drive to the ignition system. The camshaft is driven by the crankshaft through timing gears.

n) Cams

– These are made as integral parts of the camshaft and are so de•signed to open the valves at the correct timing and to keep them open for the necessary duration.

o) Flywheel

– The net torque imparted to the crankshaft during one complete cycle of operation of the engine fluctuates causing a change in the angular velocity of the shaft. In order to achieve a uniform torque an inertia mass in the form of a wheel is attached to the output shaft and this wheel is called the flywheel.

p) Carburetor

– Carburetor is used in petrol engine for proper mixing of air and petrol.

q) Fuel pump

– Fuel pump is used in diesel engine for increasing pressure and controlling the quantity of fuel supplied to the injector.

r) Fuel injector

– Fuel injector is used to inject diesel fuel in the form of fine atomized spray under pressure at the end of compression stroke.

Terminologies used in IC engine

– **Cylinder Bore (d):** The nominal inner diameter of the working cylinder is called the cylinder bore and is designated by the letter d and is usually expressed in millimeter (mm).

– **Piston Area (A):** The area of a circle of diameter equal to the cylinder bore is called the piston area and is designated by the letter A and is usually expressed in square centimeter (cm^2).

– **Stroke (L):** It is the linear distance traveled by the piston when it moves from one end of the cylinder to the other end. It is equal to twice the radius of the crank. It is designated by the letter L and is expressed usually in millimeter (mm).

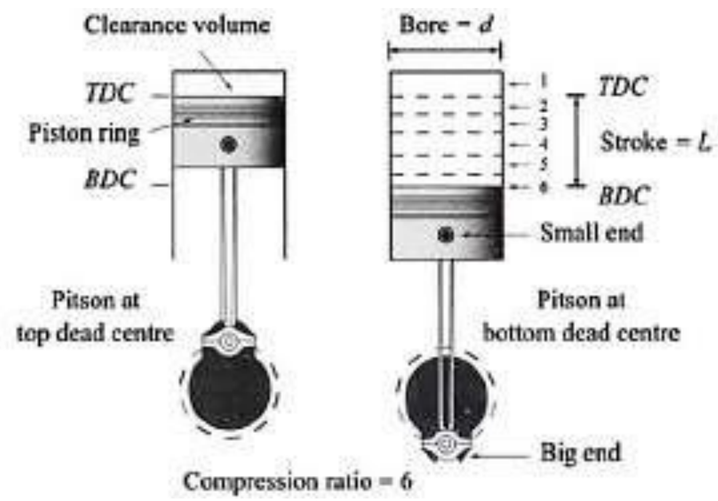


Fig 1.3 IC Engine nomenclature

- **Stroke to Bore Ratio (L/d):** L / d ratio is an important parameter in classifying the size of the engine.
  - If d < L, it is called under-square engine.
  - If d = L, it is called square engine.
  - If d > L, it is called over-square engine.

  An over-square engine can operate at higher speeds because of larger bore and shorter stroke.
- Dead Centre:
  In the vertical engines, top most position of the piston is called Top Dead Centre (TDC). When the piston is at bottom most position, it is called Bottom Dead Centre (BDC).
  In horizontal engine, the extreme position of the piston near to cylinder head is called Inner Dead Centre (IDC.) and the extreme position of the piston near the crank is called Outer Dead Centre (O.D.C.).
- **Displacement or Swept Volume (V_s)**: The volume displaced by the piston in one stroke is known as stroke volume or swept volume. It is expressed in terms of cubic centimeter (cc) and given by

$$V_s = A \times L = \frac{\pi}{4} d^2 L$$

- **Cubic Capacity or Engine Capacity:** The displacement volume of a cylinder multiplied by number of cylinders in an engine will give the cubic capacity or the engine capacity. For example, if there are K cylinders in an engine, then

$$\text{Cubic capacity} = V_s \times K$$

- **Clearance Volume (V_c):** It is the volume contained between the piston top and cylinder head when the piston is at top or inner dead center.
- **Compression Ratio (r):** The ratio of total cylinder volume to clearance volume is called the compression ratio (r) of the engine.

$$r = \frac{Total\ cylinder\ volume}{Clearance\ volume}$$

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

For petrol engine r varies from 6 to 10 and for Diesel engine r varies from 14 to 20.

- **Piston speed (V_P):** It is average speed of piston. It is equal to 2LN, where N is speed of crank shaft in rev/sec.

$$V_p = \frac{2LN}{60}\ m/_{sec}$$

where, L = Stroke length, m

N = Speed of crank shaft, RPM

Working of Four Stroke Spark-Ignition Engine

- In a four-stroke engine, the cycle of operations is completed in four strokes of the piston or two revolutions of the crankshaft.
- During the four strokes, there are five events to be completed, viz., suction, compression, combustion, expansion and exhaust. Each stroke consists of 180° of crankshaft rotation and hence a four-stroke cycle is completed through 720° of crank rotation.
- The cycle of operation for an ideal four-stroke SI engine consists of the following four strokes: (i) suction or intake stroke; (ii) compression stroke; (iii) expansion or power stroke and (iv) exhaust stroke.

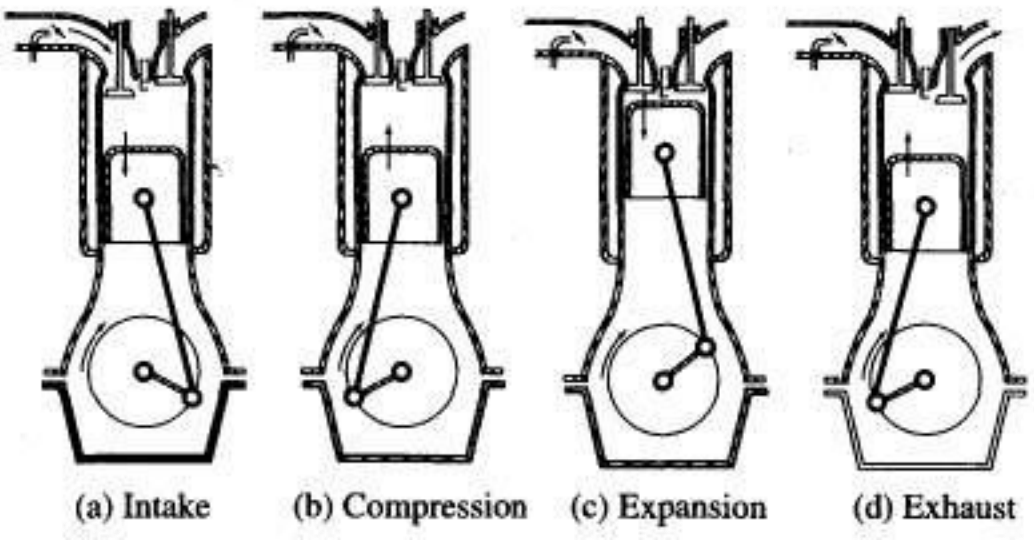


Fig. 1.4 Working principle of a four-stroke SI engine

- The details of various processes of a four-stroke spark-ignition engine with overhead valves are shown in Fig. 1.4 (a-d). When the engine completes all the five events under ideal cycle mode, the pressure-volume (p-V) diagram will be as shown in Fig.1.5.

a) Suction or Intake Stroke: Suction stroke 0→1 (Fig.1.5) starts when the piston is at the top dead centreand about to move downwards. The inlet valve is assumed to open instantaneously and at this time the exhaust valve is in the closed position, Fig.1.4 (a).

- Due to the suction created by the motion of the piston towards the bottom dead centre, the charge consisting of fuel-air mixture is drawn into the cylinder. When the piston reaches the bottom dead centre the suction stroke ends and the inlet valve closes instantaneously.

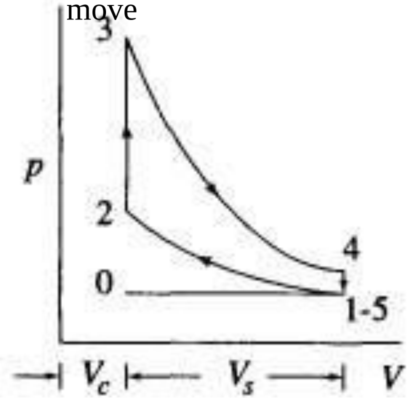


Fig. 1.5 Ideal p-V diagram of a four-stroke SI engine

b) Compression Stroke: The charge taken into the cylinder during the suction stroke is compressed by the return stroke of the piston 1→2, (Fig.1.5). During this stroke both inlet and exhaust valves are in closed position, Fig. 1.4(b).

- The mixture which fills the entire cylinder volume is now compressed into the clearance volume. At the end of the compression stroke the mixture is ignited with the help of a spark plug located on the cylinder head.
- In ideal engines it is assumed that burning takes place instantaneously when the piston is at the top dead centre and hence the burning process can be approximated as heat addition at constant volume.
- During the burning process the chemical energy of the fuel is converted into heat energy producing a temperature rise of about 2000 °C (process 2→3), Fig.1.5. The pressure at the end of the combustion process is considerably increased due to the heat release from the fuel.

c) Expansion or Power Stroke: The high pressure of the burnt gases forces the piston towards the BDC, (stroke 3→4) Fig .1.5. Both the valves are in closed position, Fig. 1.4(c). Of the four-strokes only during this stroke power is produced. Both pressure and temperature decrease during expansion.

d) Exhaust Stroke: At the end of the expansion stroke the exhaust valve opens instantaneously and the inlet valve remains closed, Fig. 1.4(d). The pressure falls to atmospheric level a part of the burnt gases escape. The piston starts moving from the bottom dead centre to top dead centre (stroke 5→0), Fig.1.5 and sweeps the burnt gases out from the cylinder almost at atmospheric pressure. The exhaust valve closes when the piston reaches TDC.

- At the end of the exhaust stroke and some residual gases trapped in the clearance volume remain in the cylinder. These residual gases mix with the fresh charge coming in during the following cycle, forming its working fluid.

- Each cylinder of a four-stroke engine completes the above four operations in two engine revolutions, first revolution of the crankshaft occurs during the suction and compression strokes and the second revolution during the power and exhaust strokes.
- Thus for one complete cycle there is only one power stroke while the crankshaft makes two revolutions. For getting higher output from the engine the heat addition (process $2 \rightarrow 3$) should be as high as possible and the heat rejection (process $3 \rightarrow 4$) should be as small as possible. Hence, one should be careful in drawing the ideal p - V diagram (Fig.1.5), which should represent the processes correctly.

Working of Four Stroke Compression-Ignition Engine

- The four-stroke Cl engine is similar to the four-stroke SI engine but it operates at a much higher compression ratio. The compression ratio of an SI engine is between 6 and 10 while for a Cl engine it is from 16 to 20.
- In the Cl engine during suction stroke, air, instead of a fuel-air mixture, is inducted. Due to higher compression ratios employed, the temperature at the end of the compression stroke is sufficiently high to self-ignite the fuel which is injected into the combustion chamber.
- In Cl engines, a high pressure fuel pump and an injector are provided to inject the fuel into the combustion chamber. The carburetor and ignition system necessary in the SI engine are not required in the Cl engine.
- The ideal sequence of operations for the four-stroke Cl engine as shown in Fig. 1.6 is as follows:

a) **Suction Stroke:** In the suction stroke piston moves from TDC to BDC. Air alone is inducted during the suction stroke. During this stroke inlet valve is open and exhaust valve is closed, Fig.1.6 (a).

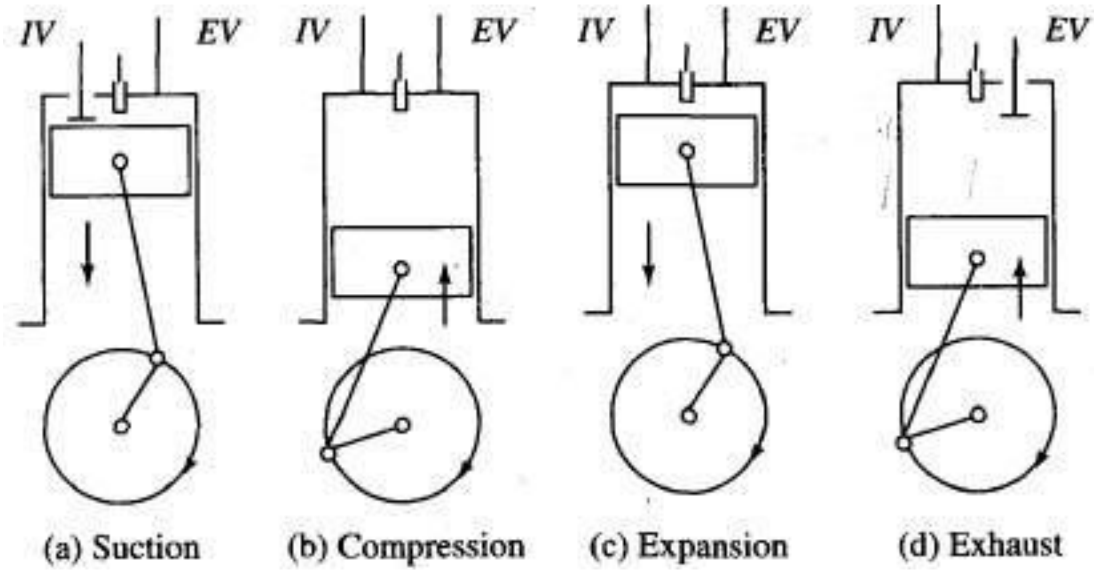


Fig. 1.6 Cycle of operation of CI engine

b) **Compression Stroke:** In this stroke piston moves from BDC to TDC. Air inducted during the suction stroke is compressed into the clearance volume. Both valves remain closed during this stroke, Fig. 1.6 (b).

c) **Expansion Stroke:** Fuel injection starts nearly at the end of the com•pression stroke. The rate of injection is such that combustion maintains the pressure constant in spite of the piston movement on its expansion stroke increasing the volume. Heat is assumed to have been added at constant pressure. After the injection of fuel is completed (i.e. after cut-off) the products of combustion expand. Both the valves remain closed during the expansion stroke, Fig. 1.6(c).

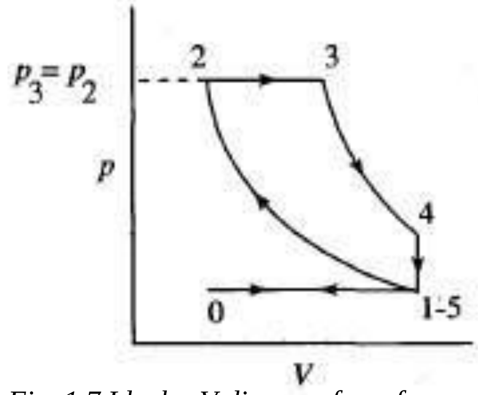


Fig. 1.7 Ideal p-V diagram for a four stroke CI engine

d) **Exhaust Stroke:** The piston travelling from BDC to TDC pushes out the products of combustion. The exhaust valve is open and the intake valve is closed during this stroke, Fig. 1.6 (d). The ideal p - V diagram is shown in Fig. 1.7.

– Due to higher pressures in the cycle of operations the Cl engine has to be sturdier than a SI engine for the same output. This results in a Cl engine being heavier than the SI engine. However, it has a higher thermal efficiency on account of the high compression ratio (of about 18 as against about 8 in SI engines) used.

Comparison of SI and Cl Engines

– The detailed comparison of SI and CI engine is given in table 1.2

Table 1.2 Comparison of SI and CI Engines

Description	SI Engine	CI Engine
Basic cycle	Works on Otto cycle or constant volume heat addition cycle.	Works on Diesel cycle or constant pressure heat addition cycle.
Fuel	Gasoline, a highly volatile fuel. Self-ignition temperature is high.	Diesel oil, a non-volatile fuel. Self- ignition temperature is comparatively low
Introduction of fuel	A gaseous mixture of fuel-air is introduced during the suction stroke. A carburetor and an ignition system are necessary. Modern engines have gasoline injection.	Fuel is injected directly into the combustion chamber at high pressure at the end of the compression stroke. A fuel pump and injector are necessary.
Load control	Throttle controls the quantity of fuel-air mixture to control the load.	The quantity of fuel is regulated to control the load. Air quantity is not controlled.

Ignition	Requires an ignition system with spark plug in the combustion chamber. Primary voltage is provided by either a battery or a magneto.	Self-ignition occurs d u e to high temperature of air because of the high compression. Ignition system and spark plug are not necessary.
Compression ratio	6 to 10. Upper limit is fixed by anti- knock quality of the fuel.	16 to 20. Upper limit is limited by weight increase of the engine.
Speed	Due to light weight and also due to homogeneous combustion, they are high speed engines.	Due to heavy weight and also due to heterogeneous combustion, they are low speed engines.
Thermal efficiency	Because of the lower CR, the maximum value of thermal efficiency that can be obtained is lower.	Because of higher CR, the maximum value of thermal efficiency that can be obtained is higher.
Weight	Lighter due to comparatively lower peak pressures.	Heavier due to comparatively higher peak pressures.

Two-Stroke Engine

- In two-stroke engines the cycle is completed in one revolution of the crankshaft. The main difference between two-stroke and four-stroke engines is in the method of filling the fresh charge and removing the burnt gases from the cylinder.
- In the four-stroke engine these operations are performed by the engine piston during the suction and exhaust strokes respectively.
- In a two- stroke engine, the filling process is accomplished by the charge compressed in crankcase or by a blower. The induction of the compressed charge moves out the product of combustion through exhaust ports. Therefore, no separate piston strokes are required for these two operations.
- Two strokes are sufficient to complete the cycle, one for compressing the fresh charge and the other for expansion or power stroke. It is to be noted that the effective stroke is reduced.
- Figure 1.8 shows one of the simplest two-stroke engines, viz., the crankcase scavenged engine. Figure 1.9 shows the ideal p - V diagram of such an engine.
- The air-fuel charge is inducted into the crankcase through the spring loaded inlet valve when the pressure in the crankcase is reduced due to upward motion of the piston during compression stroke. After the compression and ignition, expansion takes place in the usual way.
- During the expansion stroke the charge in the crankcase is compressed. Near the end of the expansion stroke, the piston uncovers the exhaust ports and the cylinder pressure drops to atmospheric pressure as the combustion products leave the cylinder.
- Further movement of the piston uncovers the transfer ports, permitting the slightly compressed charge in the crankcase to enter the engine cylinder.

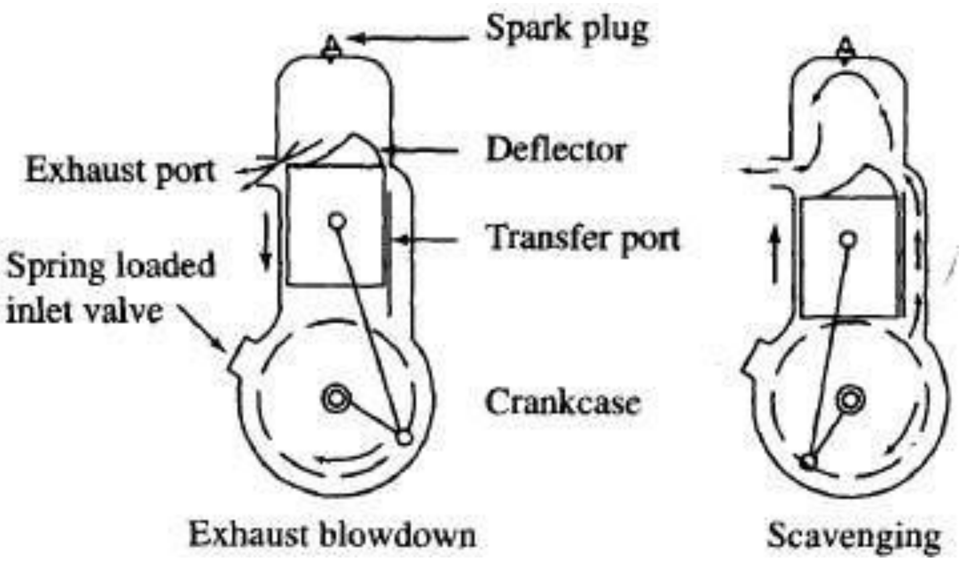


Fig. 1.8 Crankcase scavenged two-stroke SI engine

- The piston top usually has a projection to deflect the fresh charge towards the top of the cylinder preventing the flow through the exhaust ports. This serves the double purpose of scavenging the combustion products from the upper part of the cylinder and preventing the fresh charge from flowing out directly through the exhaust ports.
- The same objective can be achieved without piston deflector by proper shaping of the transfer port. During the upward motion of the piston from B D C the transfer ports close first and then the exhaust ports, thereby the effective compression of the charge begins and the cycle is repeated.

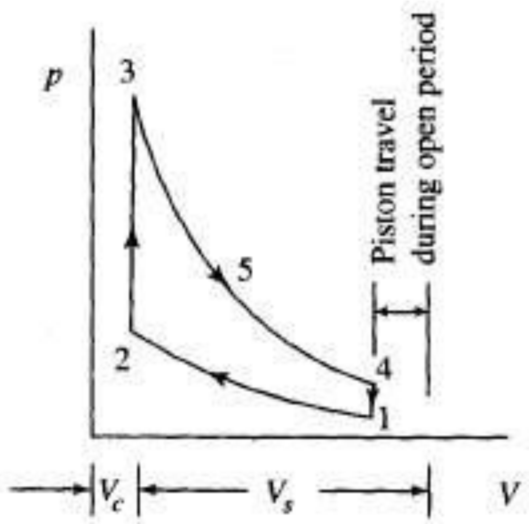


Fig 1.9 Ideal p-V diagram of a two-stroke SI engine

IC engine Classification

- I.C. Engines may be classified according to,
  a) Type of the fuel used as :
     (1) Petrol engine (2) Diesel engine
     (3) Gas engine (4) Bi-fuel engine (Two fuel engine)
  b) Nature of thermodynamic cycle as :
     (1) Otto cycle engine (2) Diesel cycle engine
     (3) Duel or mixed cycle engine
  c) Number of strokes per cycle as :
     (1) Four stroke engine (2) Two stroke engine

d) Method of ignition as :
   (1) Spark ignition engine (S.I. engine)
      Mixture of air and fuel is ignited by electric spark.
   (2) Compression ignition engine (C.I. engine)
      The fuel is ignited as it comes in contact with hot compressed air.

e) Method of cooling as :
   (1) Air cooled engine   (2) Water cooled engine

f) Speed of the engine as :
   (1) Low speed   (2) Medium speed
   (3) High speed
   Petrol engine are high speed engines and diesel engines are low to medium speed engines

g) Number of cylinder as :
   (1) Single cylinder engine (2) Multi cylinder engine

h) Position of the cylinder as :
   (1) Inline engines   (2) V – engines
   (3) Radial engines   (4) Opposed cylinder engine
   (5) X – Type engine   (6) H – Type Engine
   (7)U – Type Engine   (8) Opposed piston engine
   (9) Delta Type Engine

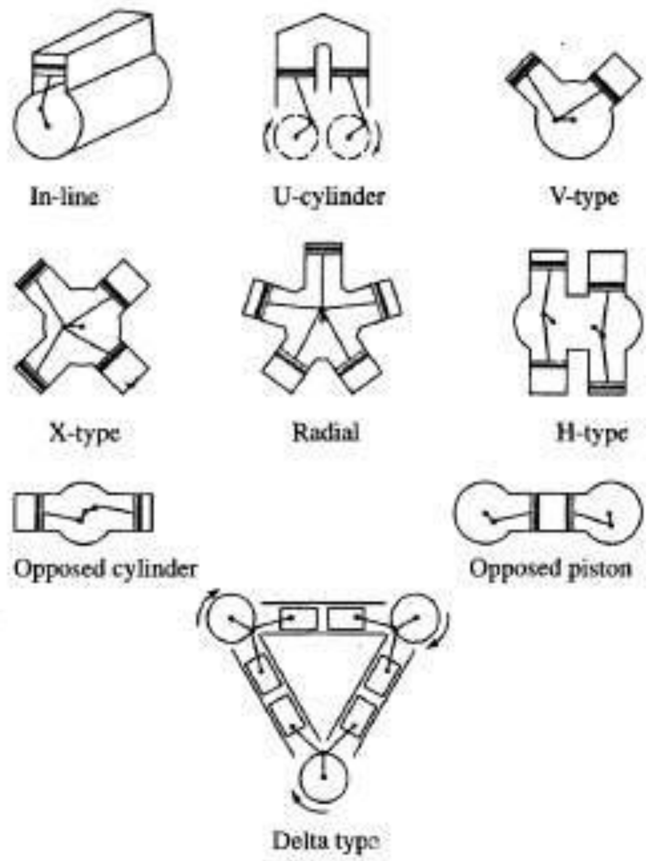


Fig. 1.10 Engine classification by cylinder arrangements

Application of IC Engines

- The most important application of IC engines is in transport on land, sea and air. Other applications include industrial power plants and as prime movers for electric generators. Table 1.3 gives, in a nutshell, the applications of both IC and EC engines.

Table 1.3 Application of Engines

IC Engine		EC Engine	
Type	Application	Type	Application
Gasoline engines	Automotive, Marine, Aircraft	Steam Engines	Locomotives, Marine
Gas engines	Industrial power	Stirling Engines	Experimental Space Vehicles
Diesel engines Railways,	Automotive, Power, Marine	Steam Turbines	Power, Large Marine
Gas turbines	Power, Aircraft, Industrial, Marine	Close Cycle Gas Turbine	Power, Marine

Engine Performance Parameters

- The engine performance is indicated by the term efficiency, η. Five important engine efficiencies and other related engine performance parameters are discussed below.

Indicated Power

- The power produced inside the engine cylinder by burning of fuel is known as Indicated power (I.P.) of engine. It is calculated by finding the actual mean effective pressure.

$$Actual\ meaneffective\ pressure,\ P_m = \frac{sa}{l}\ N/m^2 \quad (1.1)$$

where,
a = Area of the actual indicator diagram, cm^2
l = Base width of the indicator diagram, cm
s = Spring value of the spring used in the indicator, N/m^2/cm

$$ip = \frac{P_m LAn}{60000}\ kW \quad (1.2)$$

where,
P_m = Mean effective pressure N/m^2
L = Length of stroke, m
A = Area of cross section of the cylinder,
m^2 N = RPM of the engine crank shaft

$n = \frac{N}{2}$ for 4-stroke

$n = N$ for 2-stroke

Brake power

- It is the power available at engine crank shaft for doing useful work. It is also known as engine output power. It is measured by dynamometer.

$$B.P. = \frac{2\pi NT}{60000} = \frac{P_{mb}LAn}{60000} kW \quad (1.3)$$

where

$$T = W \times R \quad (1.4)$$

W = Net load acting on the brake drum, N
R = Effective radius of the brake drum, m
N = RPM of the crank shaft
T = Resisting torque, Nm
P_{mb} = Brake mean effective pressure

Indicated Thermal Efficiency (η_{ith})

- Indicated thermal efficiency is the ratio of energy in the indicated power, ip, to the input fuel energy in appropriate units.

$$\eta_{ith} = \frac{ip\left[kJ/s\right]}{\text{energy in fuel per second }\left[kJ/s\right]} \quad (1.1)$$

$$\eta_{ith} = \frac{ip}{\text{mass of fuel/s} \times \text{CV of fuel}} = \frac{ip}{m_f \times CV} \quad (1.2)$$

Brake Thermal Efficiency (η_{bth})

- Brake thermal efficiency is the ratio of power available at crank shaft, bp, to the input fuel energy in appropriate units.

$$\eta_{bth} = \frac{bp}{\text{mass of fuel/s} \times \text{CV of fuel}} = \frac{bp}{m_f \times CV} \quad (1.3)$$

Mechanical Efficiency (η_m)

- Mechanical efficiency is defined as the ratio of brake power (delivered power) to the indicated power (power provided to the piston).

$$\eta_m = \frac{bp}{ip} = \frac{bp}{bp + fp} \quad (1.4)$$

$$fp = ip - bp \quad (1.5)$$

Volumetric Efficiency (η_v)

- Volumetric efficiency indicates the breathing ability of the engine. It is to be noted that the utilization of the air is that determines the power output of the engine. Intake system must be designed in such a way that the engine must be able to take in as much air as possible.
- Volumetric efficiency is defined as the ratio of actual volume flow rate of air into the intake system to the rate at which the volume is displaced by the system.

$$\eta_v = \frac{\text{Actual volume of charge or air sucked at atm. condition}}{\text{Swept volume}} \quad (1.6)$$

Air standard efficiency

- It is the efficiency of the thermodynamic cycle of the engine.

- For petrol engine,

$$\eta_{air} = 1 - \frac{1}{(r)^{\gamma-1}} \tag{1.7}$$

- For diesel engine,

$$\eta_{air} = 1 - \frac{1}{(r)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\gamma(\rho-1)}\right] \tag{1.8}$$

Relative Efficiency or Efficiency Ratio

- Relative efficiency or efficiency ratio is the ratio of thermal efficiency of an actual cycle to that of the ideal cycle. The efficiency ratio is a very useful criterion which indicates the degree of development of the engine.

$$\eta_{rel} = \frac{\eta_{th}}{\eta_{air}} \tag{1.9}$$

Specific output

- The specific output of the engine is defined as the power output per unit area.

$$Specific output = \frac{B.P.}{A} \tag{1.10}$$

Specific fuel consumption

- Specific fuel consumption (SFC) is defined as the amount of fuel consumed by an engine for one unit of power production. SFC is used to express the fuel efficiency of an I.C. engine.

$$SFC = \frac{m_f}{B.P.}\ \mathrm{k}g / \mathrm{k}Wh \tag{1.11}$$

Air Standard Cycles

- In most of the power developing systems, such as petrol engine, diesel engine and gas turbine, the common working fluid used is air. These devices take in either a mixture of fuel and air as in petrol engine or air and fuel separately and mix them in the combustion chamber as in diesel engine
- The mass of fuel used compared with the mass of air is rather small. Therefore the properties of mixture can be approximated to the properties of air.
- Exact condition existing within the actual engine cylinder are very difficult to determine, but by making certain simplifying assumptions, it is possible to approximate these conditions more or less closely. The approximate engine cycles thus analysed are known as theoretical cycles.
- The simplest theoretical cycle is called the air-cycle approximation. The air-cycle approximation used for calculating conditions in internal combustion engine is called the air-standard cycle.

- The analysis of all air-standard cycles is based upon the following assumption:

a) The gas in the engine cylinder is a perfect gas, i.e. it obeys the gas laws and has constant specific heats.

b) The physical constants of the gas in the cylinder are the same as those of air at moderate temperatures i.e., the molecular weight of cylinder gas is 29 and $C_p = 1.005$ kJ/kg K and $C_v = 0.718$ kJ/kg K.

c) The compression and expansion processes are adiabatic and they take place without internal friction, i.e., these processes are isentropic.

d) No chemical reaction takes place in the cylinder. Heat is supplied or rejected by bringing a hot body or a cold body in contact with cylinder at appropriate points during the process.

e) The cycle is considered closed, with the same 'air' always remaining in the cylinder to repeat the cycle.

- Because of many simplifying assumptions, it is clear that the air-cycle approximation does not closely represent the conditions within the actual cylinder. Due to the simplicity of the air-cycle calculation, it is often used to obtain approximate answers to complex engine problems.

The Otto Cycle *OR* Constant Volume Cycle (Isochoric)

- The cycle was successfully applied by a German scientist Nicolous A. Otto to produce a successful 4 – stroke cycle engine in 1876.

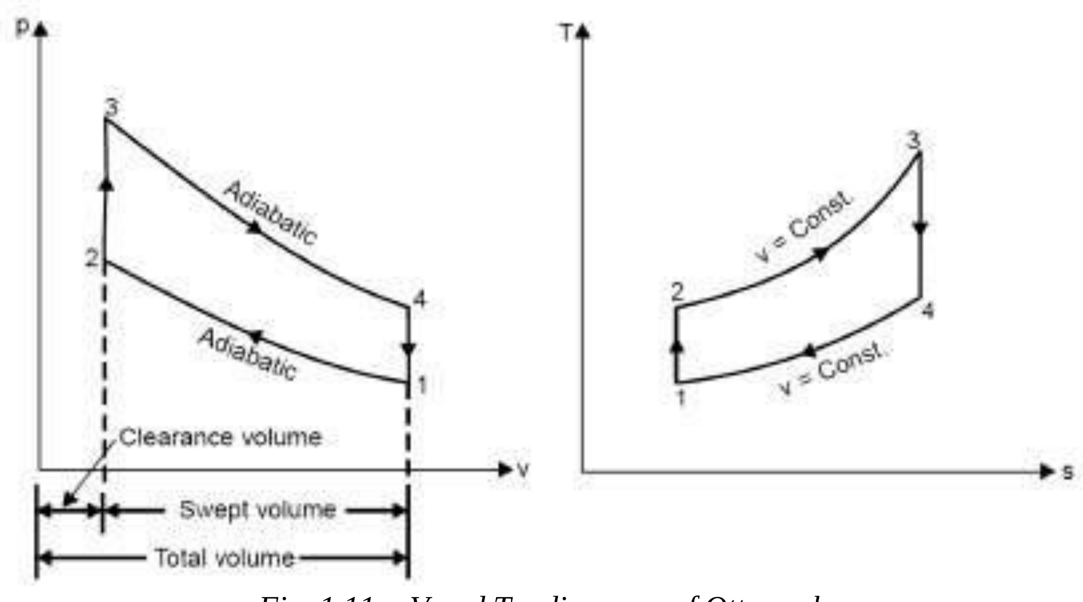


Fig. 1.11 p-V and T-s diagrams of Otto cycle

- The thermodynamic cycle is operated with isochoric (constant volume) heat addition and consists of two adiabatic processes and two constant volume changes.

- Fig. 1.11 shows the Otto cycle plotted on p – V and T – s diagram.

<u>Adiabatic Compression Process (1 – 2):</u>

- At pt. 1 cylinder is full of air with volume V_1, pressure P_1 and temp. T_1.

- Piston moves from BDC to TDC and an ideal gas (air) is compressed isentropically to state point 2 through compression ratio,

$$r = \frac{V_1}{V_2}$$

Constant Volume Heat Addition Process (2 – 3):

- Heat is added at constant volume from an external heat source.
- The pressure rises and the ratio $r_p \; or \; \alpha = \frac{p_3}{p_2}$ is called expansion ratio or pressure ratio.

Adiabatic Expansion Process (3 – 4):

- The increased high pressure exerts a greater amount of force on the piston and pushes it towards the BDC.
- Expansion of working fluid takes place isentropically and work done by the system.
- The volume ratio $\frac{V_4}{V_3}$ is called isentropic expansion ratio.

Constant Volume Heat Rejection Process (4 – 1):

- Heat is rejected to the external sink at constant volume. This process is so controlled that ultimately the working fluid comes to its initial state 1 and the cycle is repeated.
- Many petrol and gas engines work on a cycle which is a slight modification of the Otto cycle.
- This cycle is called constant volume cycle because the heat is supplied to air at constant volume.

Air Standard Efficiency of an Otto Cycle:

- Consider a unit mass of air undergoing a cyclic change.
- **Heat supplied** during the process 2 – 3,

$$q_1 = C_V \left(T_3 - T_2\right)$$

- **Heat rejected** during process 4 – 1 ,

$$q_2 = C_V \left(T_4 - T_1\right)$$

- Work done,

$$\therefore W = q_1 - q_2$$

$$\therefore W = C_V \left(T_3 - T_2\right) - C_V \left(T_4 - T_1\right)$$

- **Thermal efficiency,**

$$\eta = \frac{Work\,done}{Heat\,supplied} = \frac{W}{q_1}$$

$$= \frac{C_V\left(T_3 - T_2\right) - C_V\left(T_4 - T_1\right)}{C_V\left(T_3 - T_2\right)}$$

$$= 1 - \frac{\left(T_4 - T_1\right)}{\left(T_3 - T_2\right)} \qquad (1.12)$$

- For Adiabatic compression process (1 – 2),

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = r^{\gamma-1}$$

$$\therefore T_2 = T_1 r^{\gamma-1} \qquad (1.13)$$

- For Isentropic expansion process (3 – 4),

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1}$$

$$\therefore T_3 = T_4\left(\frac{V_4}{V_3}\right)^{\gamma-1}$$

$$\therefore T_3 = T_4\left(\frac{V_1}{V_2}\right)^{\gamma-1} \left(\because\ V_1 = V_4,\ V_2 = V_3\right)$$

$$\therefore T_3 = T_4\left(r\right)^{\gamma-1} \qquad (1.14)$$

- From equation 1.16, 1.17 & 1.18, we get,

$$\eta_{otto} = 1 - \frac{\left(T_4 - T_1\right)}{T_4 r^{\gamma-1} - T_1 r^{\gamma-1}}$$

$$\therefore \eta_{otto} = 1 - \frac{\left(T_4 - T_1\right)}{r^{\gamma-1}\left(T_4 - T_1\right)}$$

$$\therefore \eta_{otto} = 1 - \frac{1}{r^{\gamma-1}} \qquad (1.15)$$

- Expression 1.19 is known as the air standard efficiency of the Otto cycle.
- It is clear from the above expression that efficiency increases with the increase in the value of r (as γ is constant).
- We can have maximum efficiency by increasing r to a considerable extent, but due to practical difficulties its value is limited to 8.
- In actual engines working on Otto cycle, the compression ratio varies from 5 to 8 depending upon the quality of fuel.

- At compression ratios higher than this, the temperature after combustion becomes high and that may lead to spontaneous and uncontrolled combustion of fuel in the cylinder.
- The phenomenon of uncontrolled combustion in petrol engine is called detonation and it leads to poor engine efficiency and in structural damage of engine parts.

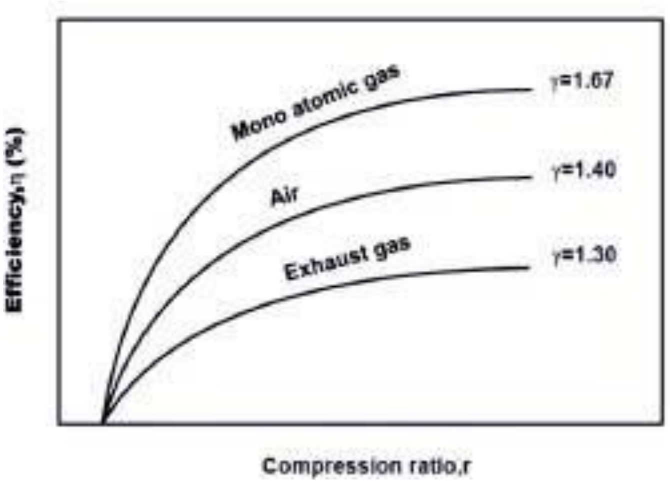


Fig. 1.12 Variation of Otto cycle efficiency with compression ratio

- Fig. 1.12 shows the variation of air standard efficiency of Otto cycle with compression ratio.

Mean Effective Pressure:

- **Net work done** per unit mass of air,

$$W_{net} = C_V\left(T_3 - T_2\right) - C_V\left(T_4 - T_1\right) \tag{1.16}$$

- Swept volume,

$$Swept\ volume = V_1 - V_2 = V_1\left(1 - \frac{V_2}{V_1}\right) = \frac{RT_1}{P_1}\left(1 - \frac{1}{r}\right)$$

$$= \frac{RT_1}{P_1 r}\left(r - 1\right) \tag{1.17}$$

- Mean effective pressure,

$$mep = \frac{Work\ done\ per\ cycle}{swept\ volume}$$

$$= \frac{C_V\left(T_3 - T_2\right) - C_V\left(T_4 - T_1\right)}{\frac{RT_1}{P_1 r}\left(r - 1\right)}$$

$$= \frac{C_V}{R}\frac{P_1 r}{\left(r-1\right)}\left[\frac{\left(T_3 - T_2\right) - \left(T_4 - T_1\right)}{T_1}\right] \tag{1.18}$$

- For process 1 – 2,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1}$$

$$T_2 = T_1 r^{\gamma-1}$$

- Process 2 – 3,

$$\frac{T_3}{T_2} = \frac{P_3}{P_2} \quad \because (V_2 = V_3)$$

$$\therefore T_3 = T_2\, \alpha \qquad (\alpha = explosion\ pressureratio)$$

$$\therefore\ T_3 = T_1\, \alpha\, r^{\gamma-1}$$

- Process 3 – 4,

$$T_4 = T_3 \left(\frac{V_3}{V_4}\right)^{\gamma-1}$$

$$T_4 = T_3 \left(\frac{V_3}{V_4}\right)^{\gamma-1}$$

$$\therefore T_4 = T_1\, \alpha\, r^{\gamma-1} \left(\frac{V_2}{V_1}\right)^{\gamma-1}$$

$$\therefore T_4 = T_1\, \alpha\, r^{\gamma-1} \times \frac{1}{r^{\gamma-1}}$$

$$\therefore T_4 = T_1 \cdot \alpha$$

- Substituting all these temperature values in equation 1.22, We get,

$$mep = \frac{C_V\ P_1 r}{R\ (r-1)} \left[\frac{\left(T_1\, \alpha\, r^{\gamma-1} - T_1 r^{\gamma-1}\right) - \left(T_1\, \alpha - T_1\right)}{T_1}\right]$$

$$\therefore mep = \frac{C_V\ P_1 r}{R\ (r-1)} \left[\frac{T_1 r^{\gamma-1}(\alpha-1) - T_1(\alpha-1)}{T_1}\right]$$

$$\therefore mep = \frac{C_V\ P_1 r}{R\ (r-1)} \left[\left(r^{\gamma-1}-1\right)(\alpha-1)\right]$$

$$\therefore mep = \frac{P_1 r}{(r-1)(\gamma-1)} \left[\left(r^{\gamma-1}-1\right)(\alpha-1)\right] \qquad (1.19)$$

$$\left(\because \frac{C_V}{R} = \frac{1}{\gamma-1}\right)$$

$$\left[\begin{array}{ll} \dfrac{C_P}{C_V} = \gamma, & C_P - C_V = R, \\ C_V\left(\dfrac{C_P}{C_V} - 1\right) = R, & \dfrac{C_V}{R} = \dfrac{1}{\gamma-1} \end{array}\right]$$

The Diesel Cycle *OR* Constant Pressure Cycle (Isobaric)

– This cycle was discovered by a German engineer Dr. Rudolph Diesel. Diesel cycle is also known as ***constant pressure heat addition cycle.***

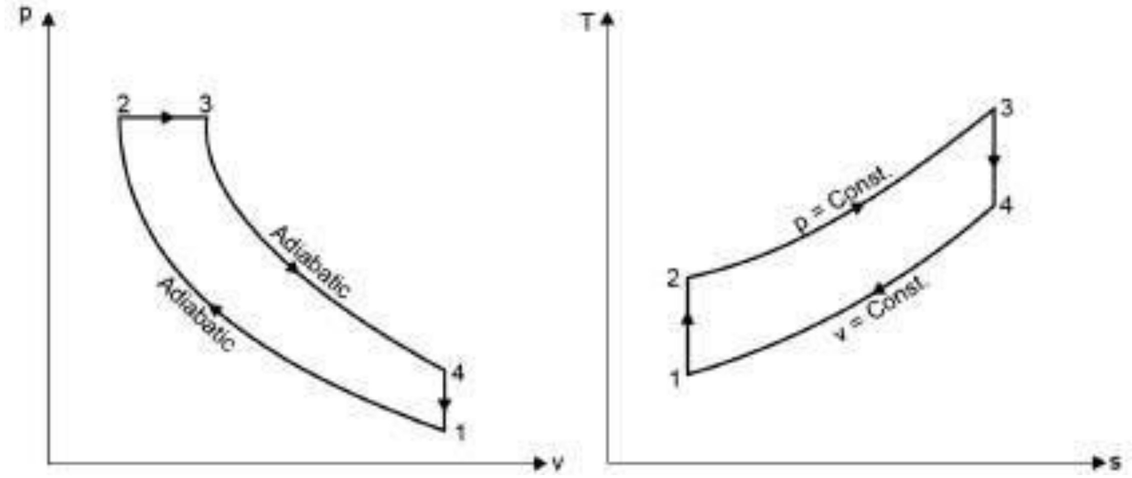


Fig. 1.13 p-V and T-s diagrams of Diesel cycle

Adiabatic Compression Process (1 – 2):

– Isentropic (Reversible adiabatic) compression with $r = \frac{V_1}{V_2}$.

Constant Pressure Heat Addition Process (2 – 3):

– The heat supply is stopped at point 3 which is called the cut – off point and the volume ratio $\rho = \frac{V_3}{V_2}$ is called **cut off ratio** or Isobaric expansion ratio.

Adiabatic Expansion Process (3 – 4):

– Isentropic expansion of air $\frac{V_4}{V_3}$ = isentropic expansion ratio.

Constant Volume Heat Rejection Process (4 – 1):

– In this process heat is rejected at constant volume.

– This thermodynamics cycle is called constant pressure cycle because heat is supplied to the air at constant pressure.

Air Standard Efficiency for Diesel Cycle:

– Consider unit mass of air.

– **Heat supplied** during process 2 – 3,

$$q_1 = C_P\left(T_3 - T_2\right)$$

– **Heat rejected** during process 4 – 1,

$$q_2 = C_V\left(T_4 - T_1\right)$$

– Work done,

$$W = q_1 - q_2$$
$$W = C_P(T_3 - T_2) - C_V(T_4 - T_1)$$

- **Thermal efficiency,**

$$\eta = \frac{Work\ done}{Heat\ supplied}$$
$$\therefore \eta = \frac{C_P(T_3 - T_2) - C_V(T_4 - T_1)}{C_P(T_3 - T_2)}$$
$$\therefore \eta = 1 - \frac{C_V}{C_P}\frac{(T_4 - T_1)}{(T_3 - T_2)}$$
$$\therefore \eta = 1 - \frac{1}{\gamma}\frac{(T_4 - T_1)}{(T_3 - T_2)} \qquad (1.20)$$

- For adiabatic compression process (1 – 2),

$$r = \frac{V_1}{V_2} \qquad (1.21)$$
$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^\gamma$$
$$P_2 = P_1 \cdot r^\gamma \qquad (1.22)$$
$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1 r^{\gamma-1} \qquad (1.23)$$

- For constant pressure heat addition process (2 – 3)

$$P_3 = P_2 = P_1 \cdot r^\gamma \qquad (1.24)$$
$$\rho = \frac{V_3}{V_2}\ (Cutoff\ ratio) \qquad (1.25)$$
$$T_3 = T_2\frac{V_3}{V_2} \qquad (1.26)$$
$$= T_2 \cdot \rho$$
$$\therefore T_3 = T_1 \cdot r^{\gamma-1} \cdot \rho \qquad (1.27)$$

- For adiabatic expansion process (3 – 4),

$$P_4 = P_3(V_3/V_4)^\gamma = P_3(V_3/V_1)^\gamma$$
$$\therefore P_4 = P_3\left(\frac{V_3/V_2}{V_1/V_2}\right)^\gamma = P_3(\rho/r)^\gamma \qquad (1.28)$$
$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{\gamma-1} = T_3\left(\frac{\rho}{r}\right)^{\gamma-1}$$

$$\therefore T_4 = \frac{T_1 \cdot r^{\gamma-1} \cdot \rho \cdot \rho^{\gamma-1}}{r^{\gamma-1}}$$

$$\therefore T_4 = T_1 \cdot \rho^{\gamma} \tag{1.29}$$

- Using above equations in equation 1.24

$$\eta = 1 - \frac{1}{\gamma}\frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$\therefore \eta = 1 - \frac{1}{\gamma}\frac{\left(T_1 \rho^{\gamma} - T_1\right)}{\left(T_1 r^{\gamma-1}\rho - T_1 r^{\gamma-1}\right)}$$

$$\therefore \boldsymbol{\eta} = 1 - \frac{1}{\boldsymbol{r}^{\gamma-1}}\left[\frac{\left(\boldsymbol{\rho}^{\gamma} - 1\right)}{\gamma\left(\boldsymbol{\rho} - 1\right)}\right] \tag{1.30}$$

- Apparently the efficiency of diesel cycle depends upon the compression ratio (r) and cutoff ratio (ρ) and hence upon the quantity of heat supplied.
- Fig. 1.14 shows the air standard efficiency of diesel cycle for various cut off ratio.
- Further,

$$K = \frac{\rho^{\gamma} - 1}{\gamma(\rho - 1)}$$

reveals that with an increase in the cut – off ratio (ρ) the value of factor K increases.

- That implies that for a diesel engine at constant compression ratio, the efficiency would increase with decrease in ρ and in the limit ρ → 1 , the efficiency would become

$$1 - \frac{1}{r^{\gamma-1}}$$

- Since the factor $K = \frac{\rho^{\gamma} - 1}{\gamma(\rho - 1)}$ is always greater than unity, the Diesel cycle is always less efficient than a corresponding Otto cycle having the same compression ratio.

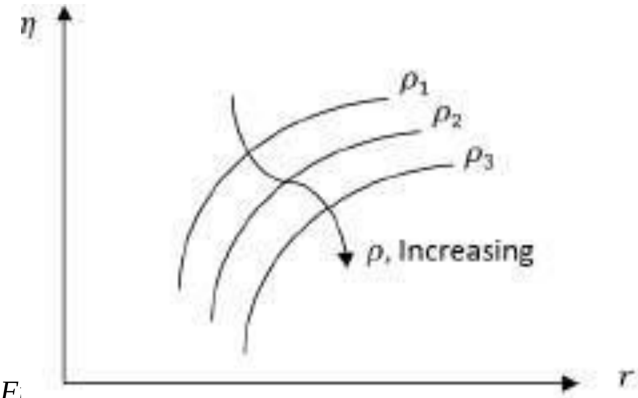


Fig. 1.14 Efficiency of Diesel cycle for various cut off ratio

- However Diesel engine operates on much higher compression ratio (14 to 18) compared to those for S.I. Engines operating on Otto cycle.
- High compression ratios for Diesel engines are must not only for high efficiency but also to prevent diesel knock; a phenomenon which leads to uncontrolled and rapid combustion in diesel engines.

Mean Effective Pressure:

- **Net work done** per unit mass of air,

$$W_{net} = C_p\left(T_3 - T_2\right) - C_V\left(T_4 - T_1\right) \tag{1.31}$$

- Swept volume,

$$Swept\ volume = V_1 - V_2 = V_1\left(1 - \frac{V_2}{V_1}\right) = \frac{RT_1}{P_1}\left(1 - \frac{1}{r}\right)$$

$$= \frac{RT_1}{P_1 r}\left(r - 1\right) \tag{1.32}$$

- Mean effective pressure,

$$mep = \frac{Work\ done\ per\ cycle}{swept\ volume}$$

$$\therefore mep = \frac{C_P\left(T_3 - T_2\right) - C_V\left(T_4 - T_1\right)}{\frac{RT_1}{P_1 r}\left(r - 1\right)}$$

$$\therefore mep = \frac{C_V}{R}\frac{P_1 r}{\left(r - 1\right)}\left[\frac{\gamma\left(T_3 - T_2\right) - \left(T_4 - T_1\right)}{T_1}\right] \tag{1.33}$$

- From equation 1.27, 1.31 and 1.33,

$T_2 = T_1 r^{\gamma - 1}$

$T_3 = T_1 r^{\gamma - 1}\rho$

$T_4 = T_1 \rho^{\gamma}$

$$\therefore mep = \frac{C_V}{R}\frac{P_1 r}{\left(r - 1\right)}\left[\frac{\gamma\left(T_1 r^{\gamma-1}\rho - T_1 r^{\gamma-1}\right) - \left(T_1\rho^{\gamma} - T_1\right)}{T_1}\right]$$

$$\therefore \boldsymbol{mep} = \frac{\boldsymbol{P_1 r}}{\left(\gamma - 1\right)\left(\boldsymbol{r} - 1\right)}\left[\gamma \boldsymbol{r}^{\gamma-1}\left(\boldsymbol{\rho} - 1\right) - \left(\boldsymbol{\rho}^{\gamma} - 1\right)\right] \tag{1.34}$$

The Dual Combustion Cycle OR The Limited Pressure Cycle

- This is a cycle in which the addition of heat is partly at constant volume and partly at constant pressure.

Adiabatic Compression Process (1 – 2):

- Isentropic (Reversible adiabatic) compression with $r = \frac{V_1}{V_2}$.

Constant Volume Heat Addition Process (2 – 3):

- The heat is supplied at constant volume with explosion ratio or pressure ratio $\alpha = \frac{P_3}{P_2}$.

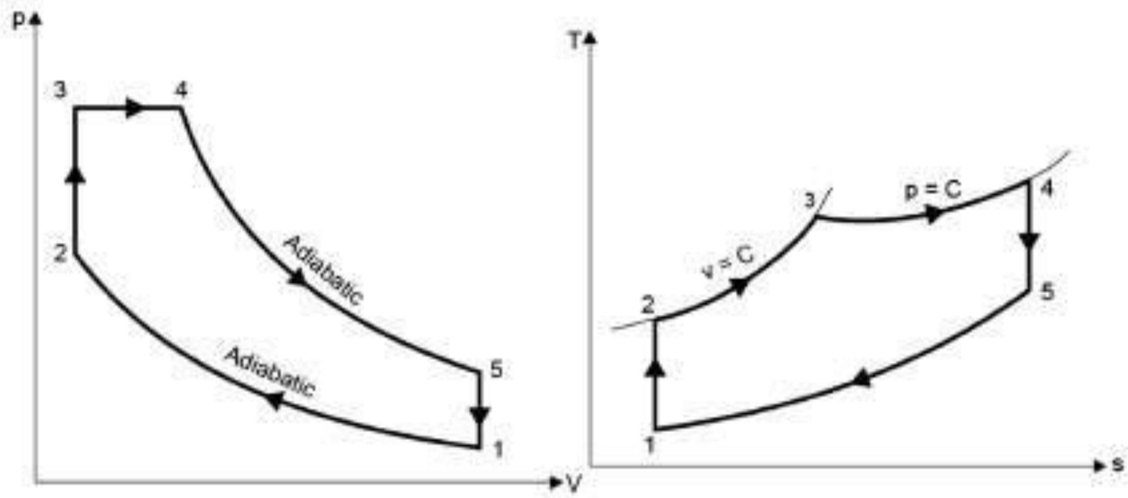


Fig. 1.15 p-V and T-s diagrams of Diesel cycle

Constant Pressure Heat Addition Process (3 – 4):

– The heat supply is stopped at point 4 which is called the cut – off point and the volume ratio $\rho = \frac{V_4}{V_3}$ is called **cut off ratio**.

Adiabatic Expansion Process (4 – 5):

– Isentropic expansion of air with $\frac{V_5}{V_4}$= isentropic expansion ratio.

Constant Volume Heat Rejection Process (5 – 1):

– In this process heat is rejected at constant volume.

– The high speed Diesel engines work on a cycle which is slight modification of the Dual cycle.

Thermal Efficiency for Dual Cycle:

– Consider unit mass of air undergoing the cyclic change.

– Heat supplied,

$$q_1 = q_{2-3} + q_{3-4}$$

$$q_1 = C_V \left(T_3 - T_2\right) + C_P \left(T_4 - T_3\right)$$

– **Heat rejected** during process 5 – 1,

$$q_2 = C_V \left(T_5 - T_1\right)$$

– Work done,

$$W = q_1 - q_2$$

$$W = C_V \left(T_3 - T_2\right) + C_P \left(T_4 - T_3\right) - C_V \left(T_5 - T_1\right)$$

– Thermal efficiency,

$$\eta = \frac{Work\ done}{Heat\ supplied}$$

$$\therefore \eta = \frac{C_V\left(T_3 - T_2\right) + C_P\left(T_4 - T_3\right) - C_V\left(T_5 - T_1\right)}{C_V\left(T_3 - T_2\right) + C_P\left(T_4 - T_3\right)}$$

$$\therefore \eta = 1 - \frac{\left(T_5 - T_1\right)}{\left(T_3 - T_2\right) + \gamma\left(T_4 - T_3\right)} \tag{1.35}$$

- For adiabatic compression process (1 – 2),

$$r = \frac{V_1}{V_2} \tag{1.36}$$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{\gamma}$$

$$P_2 = P_1 r^{\gamma} \tag{1.37}$$

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1 r^{\gamma-1} \tag{1.38}$$

- For constant volume heat addition process (2 – 3)

$$V_3 = V_2 = \frac{V_1}{r}$$

$$\alpha = \frac{P_3}{P_2}\ (Pressure ratio) \tag{1.39}$$

$$\therefore P_3 = P_2 \cdot \alpha = P_1 \cdot r^{\gamma} \cdot \alpha$$

$$T_3 = T_2 \frac{P_3}{P_2}$$

$$= T_2\,\alpha$$

$$\therefore T_3 = T_1 r^{\gamma-1} \alpha \tag{1.40}$$

- For constant pressure heat addition process (3 – 4)

$$P_3 = P_4 = P_1 r^{\gamma} \alpha \tag{1.41}$$

$$\rho = \frac{V_4}{V_3}\ (Cutoff\ ratio) \tag{1.42}$$

$$T_4 = T_3 \frac{V_4}{V_3}$$

$$\therefore T_4 = T_3\,\rho$$

$$\therefore T_4 = T_1 r^{\gamma-1} \rho \alpha \tag{1.43}$$

- For adiabatic expansion process (4 – 5),

$$P_4 V_4^{\gamma} = P_5 V_5^{\gamma}$$

$$P_5 = P_4 \left(V_4 / V_5\right)^{\gamma} = P_3 \left(V_4 / V_1\right)^{\gamma} \qquad \left(\because V_1 = V_5 \ \& P_3 = P_4\right)$$

$$P_5 = P_3 \left(\frac{V_4 V_3}{V_1 V_3}\right)^{\gamma} = P_3 \left(\frac{V_4 V_2}{V_1 V_3}\right)^{\gamma} \qquad \left(\because V_3 = V_2\right)$$

$$\therefore P_5 = P_3 \left(\frac{V_4 / V_3}{V_1 / V_2}\right)^{\gamma} = P_3 \left(\rho / r\right)^{\gamma} --------(i) \tag{1.44}$$

and

$$T_5 = T_4 \left(\frac{V_4}{V_5}\right)^{\gamma-1}$$

$$\therefore T_5 = T_4 \left(\frac{\rho}{r}\right)^{\gamma-1}$$

$$\therefore T_5 = \frac{T_1 r^{\gamma-1} \rho\alpha\, \rho^{\gamma-1}}{r^{\gamma-1}}$$

$$\therefore T_5 = T_1 \alpha \rho^{\gamma} \tag{1.45}$$

- From equation 1.39,

$$\eta = 1 - \frac{\left(T_5 - T_1\right)}{\left(T_3 - T_2\right) + \gamma\left(T_4 - T_3\right)}$$

$$\therefore \eta = 1 - \frac{\left(T_1 \alpha \rho^{\gamma} - T_1\right)}{\left(T_1 r^{\gamma-1}\alpha - T_1 r^{\gamma-1}\right) + \gamma\left(T_1 r^{\gamma-1}\alpha\rho - T_1 r^{\gamma-1}\alpha\right)}$$

$$\therefore \eta = 1 - \frac{\left(\rho^{\gamma}\alpha - 1\right)}{\left[r^{\gamma-1}\left\{\left(\alpha - 1\alpha\right) + \gamma\alpha\left(\rho - 1\right)\right\}\right]}$$

$$\therefore \boldsymbol{\eta} = 1 - \frac{1}{\boldsymbol{r}^{\gamma-1}}\left[\frac{\left(\boldsymbol{\alpha\rho}^{\gamma} - 1\right)}{\left(\boldsymbol{\alpha} - 1\right) + \boldsymbol{\gamma\alpha}\left(\boldsymbol{\rho} - 1\right)}\right] \tag{1.46}$$

- It can be seen from the equation 1.50 that the thermal efficiency of a Dual cycle can be increased by supplying a greater portion of heat at constant volume (high value of α) and smaller portion at constant pressure (low value of ρ).
- In the actual high speed Diesel engines operating on this cycle, it is achieved by early fuel injection and an early cut-off.
- It is to be noted that Otto and Diesel cycles are special cases of the Dual cycle.
- If $\rho = 1$ ($V_3 = V_4$)
- Hence, there is no addition of heat at constant pressure. Consequently the entire heat is supplied at constant volume and the cycle becomes the Otto cycle.
- By substituting $\rho = 1$ in equation 1.50, we get,

$$\eta = 1 - \frac{1}{r^{(\gamma-1)}} = Efficiency\ of\ Otto\ cycle$$

- Similarly if $\alpha = 1$, the heat addition is only at constant pressure and cycle becomes Diesel cycle.

- By substituting $\alpha = 1$ in equation 1.50, we get,

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{(\rho^{\gamma}-1)}{\gamma(\rho-1)}\right] = Efficiency\ of\ Diesel\ cycle$$

- **Mean Effective Pressure:**

- **Net work done** per unit mass of air,

$$W_{net} = C_V\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right) - C_V\left(T_5 - T_1\right) \tag{1.47}$$

- Swept volume,

$$\text{Swept Volume} = V_1 - V_2 = V_1\left(1 - \frac{V_2}{V_1}\right) = \frac{RT_1}{P_1}\left(1 - \frac{1}{r}\right)$$

$$= \frac{RT_1}{P_1 r}(r-1) \tag{1.48}$$

- Mean effective pressure,

$$mep = \frac{Work\ done\ per\ cycle}{swept\ volume}$$

$$\therefore mep = \frac{C_V\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right) - C_V\left(T_5 - T_1\right)}{\frac{RT_1}{P_1 r}(r-1)}$$

$$\therefore mep = \frac{C_V}{R}\frac{P_1 r}{(r-1)}\left[\frac{\left(T_3 - T_2\right) + \gamma\left(T_4 - T_3\right) - \left(T_5 - T_1\right)}{T_1}\right]$$

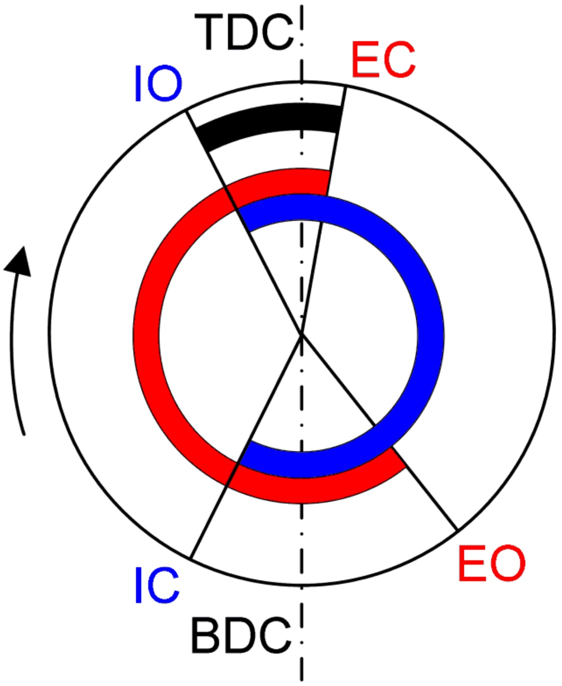
- From equation 1.42, 1.44, 1.47 and 1.49,

$$T_2 = T_1 \cdot r^{\gamma-1}$$
$$T_3 = T_1 \cdot r^{\gamma-1} \cdot \alpha$$
$$T_4 = T_1 \cdot r^{\gamma-1} \cdot \alpha \cdot \rho$$
$$T_5 = T_1 \cdot \alpha \cdot \rho^{\gamma}$$

$$\therefore mep = \frac{C_V}{R}\frac{P_1 r}{(r-1)}\left[\frac{\gamma\left(T_1 r^{\gamma-1}\alpha - T_1 r^{\gamma-1}\right) + \gamma\left(T_1 r^{\gamma-1}\alpha\rho - T_1 r^{\gamma-1}\alpha\right) - \left(T_1\alpha\rho^{\gamma} - T_1\right)}{T_1}\right]$$

$$\therefore \boldsymbol{mep} = \frac{\boldsymbol{P_1 r}}{(\boldsymbol{\gamma}-1)(\boldsymbol{r}-1)}\left[(\boldsymbol{\alpha}-1)\boldsymbol{r}^{\gamma-1} + \boldsymbol{\gamma\alpha r}^{\gamma-1}(\boldsymbol{\rho}-1) - (\boldsymbol{\alpha\rho}^{\gamma}-1)\right] \tag{1.49}$$

FUEL AIR CYCLES & ACTUAL AIR CYCLES



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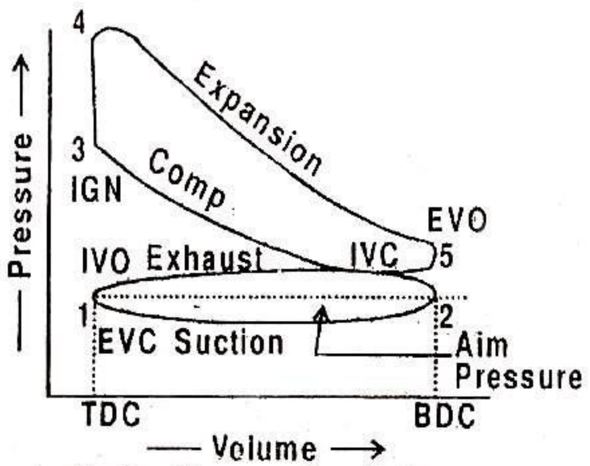


Indicator Diagram For A Four Stroke Cycle Petrol Engine.

Fuel-Air cycle

Introduction

- The air cycle approximation of air standard theory has highly simplified assumptions. The air standard theory gives an estimate of engine performance which is much greater than the actual performance. For example the actual indicated thermal efficiency of a petrol engine of, say compression ratio 7:1, is of the order of 30% whereas the air standard efficiency is of the order of 54%.
- This large divergence is partly due to non-instantaneous burning and valve operation, incomplete combustion, etc. But the main reason of divergence is the oversimplification in using the values of the properties of the working fluid for cycle analysis.
- In the air cycle analysis it was assumed that the working fluid is nothing but air and this air was a perfect gas and had constant specific heats.
- In actual engine the working fluid is not air but a mixture of air, fuel and residual gases. Furthermore, the specific heats of the working fluid are not constant but increase as temperature rises, and finally, the products of combustion are subjected to dissociation at high temperature.

Factors considered for Fuel-Air cycle calculations

The following factors are taken into consideration while making fuel-air cycle calculations:

- **The actual composition of the cylinder gases:** The cylinder gases con•tains fuel, air, water vapour and residual gas. The fuel-air ratio changes during the operation of the engine which changes the relative amounts of CO_2, water vapour, etc.
- **The variation in the specific heat with temperature:** Specific heats increase with temperature except for mono-atomic gases. Therefore, the value of γ also changes with temperature.
- **The effect of dissociation:** The fuel and air do not completely combine chemically at high temperatures (above 1600 K) and this leads to the presence of CO, H_2, H and O_2 at equilibrium conditions.
- **The variation in the number of molecules:** The number of molecules present after combustion depends upon fuel-air ratio and upon the pressure and temperature after the combustion.

Assumptions made for Fuel-Air cycle analysis

- There is no chemical change in either fuel or air prior to combustion.
- Subsequent to combustion, the charge is always in chemical equilibrium.
- There is no heat exchange between the gases and the cylinder walls in any process, i.e. they are adiabatic. Also the compression and expansion processes are frictionless.
- In case of reciprocating engines it is assumed that fluid motion can be ignored inside the cylinder.
- With particular reference to constant- volume fuel-air cycle, it is also assumed that

- The fuel is completely vaporized and perfectly mixed with the air, and
- The burning takes place instantaneously at top dead centre (at constant volume).

Importance of Fuel-Air cycle

- The air-standard cycle analysis shows the general effect of only compression ratio on engine efficiency whereas the fuel-air cycle analysis gives the effect of variation of fuel-air ratio, inlet pressure and temperature on the engine performance. It will be noticed that compression ratio and fuel-air ratio are very important parameters of the engine while inlet conditions are not so important.
- The actual efficiency of a good engine is about 85 per cent of the estimated fuel-air cycle efficiency. A good estimate of the power to be expected from the actual engine can be made from fuel-air cycle analysis. Also, peak pressures and exhaust temperatures which affect the engine structure and design can be estimated reasonably close to an actual engine. Thus the effect of many variables on the performance of an engine can be understood better by fuel-air cycle analysis.

Variable Specific Heats

- All gases, except mono-atomic gases, show an increase in specific heat with temperature. The increase in specific heat does not follow any particular law. However, over the temperature range generally encountered for gases in heat engines (300 K to 2000 K) the specific heat curve is nearly a straight line which may be approximately expressed in the form

$$C_p = a_1 + K_1 T$$
$$C_v = b_1 + K_1 T \tag{2.1}$$

where a_1,b_1 and K_1 are constants. Now,

$$R = C_p - C_v = a_1 - b_1 \tag{2.2}$$

where R is the characteristic gas constant.

- Above 1500 K the specific heat increases much more rapidly and may be expressed in the form

$$C_p = a_1 + K_1 T + K_2 T^2 \tag{2.3}$$

$$C_v = b_1 + K_1 T + K_2 T^2 \tag{2.4}$$

- In above equations if the term T^2 is neglected it becomes same as Eqn.2.1. Many expressions are available even upto sixth order of T (i.e. T^6) for the calculation of C_p and C_v.
- The physical explanation for increase in specific heat is that as the temperature is raised, larger fractions of the heat would be required to produce motion of the atoms within the molecules. Since temperature is the result of motion of the molecules, as a whole, the energy which goes into moving the atoms does not contribute to proportional temperature rise. Hence, more heat is required to raise the temperature

of unit mass through one degree at higher levels. This heat by definition is the specific heat. The values for C_p and C_v for air are usually taken as

C_p =1.005 kJ/kg K , C_v =0.717 kJ/kg K at 300 K

C_p =1.345 kJ/kg K , C_v =1.057 kJ/kg K at 2000 K

- Since the difference between C_p and C_v is constant, the value of γ decreases with increase in temperature. Thus, if the variation of specific heats is taken into account during the compression stroke, the final temperature and pressure would be lower than if constant values of specific heat are used. This point is illustrated in Fig.2.1.

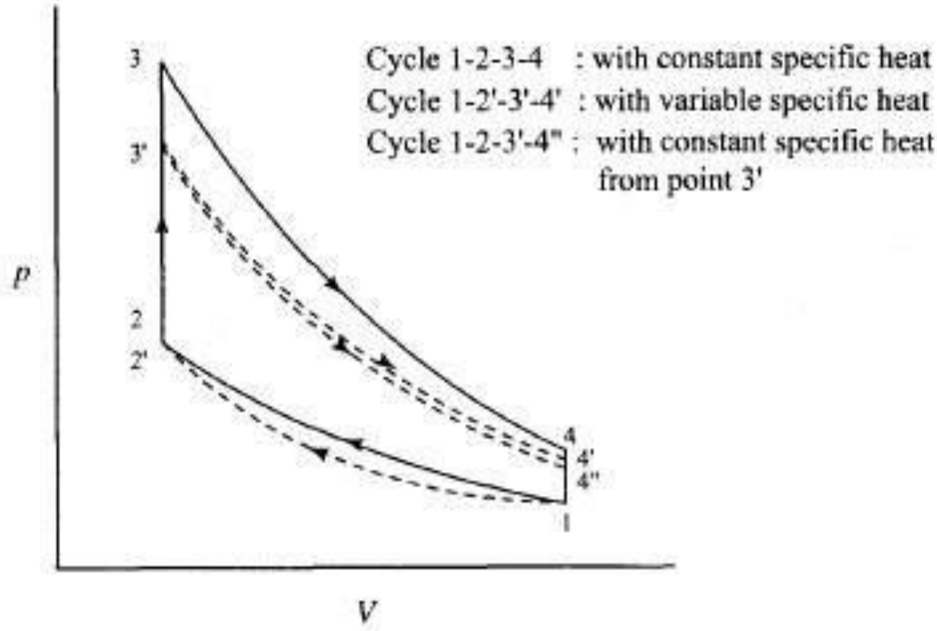


Fig. 2.1 Loss of power due to variation of specific heat

- With variable specific heats, the temperature at the end of compression will be 2', instead of 2. The magnitude of drop in temperature is proportional to the drop in the value of ratio of specific heats. For the process 1—>2, with constant specific heats

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{\gamma-1} \tag{2.5}$$

with variable specific heats,

$$T_{2'} = T_1\left(\frac{v_1}{v_{2'}}\right)^{k-1} \tag{2.6}$$

where $k = \frac{C_p}{C_v}$. Note that $v_{2'}$ = v_2 and v_1/v_2 = $v_1/v_{2'}$ = r.

- For given values of T_1, p_1 and r, the magnitude of T_2 depends on k. Constant volume combustion, from point 2 ' will give a temperature $T_{3'}$ instead of T_3. This is due to the fact that the rise in the value of C_v because of variable specific heat, which reduces the temperature as already explained.
- The process, 2'-3' is heat addition with the variation in specific heat. From 3', if expansion takes place at constant specific heats, this would result in the process 3'-4"

whereas actual expansion due to variable specific heat will result in 3'-4' and 4' is higher than 4". The magnitude in the difference between 4' and 4 " is proportional to the reduction in the value of γ.

- Consider the process 3'-4"

$$T_{4''} = T_{3'}\left(\frac{v_3}{v_4}\right)^{k-1} \tag{2.7}$$

For the process 3'-4'

$$T_{4'} = T_{3'}\left(\frac{v_3}{v_4}\right)^{\gamma-1} \tag{2.8}$$

- Reduction in the value of k due to variable specific heat results in increase of temperature from $T_{4''}$ to $T_{4'}$.

Change of Internal energy and enthalpy during a process with variable specific heats

Change of Internal energy

- The small change in internal energy of a unit mass of a gas for small change in temperature (dT) is given by:

$$du = C_v\, dT$$

$$\therefore u_2 - u_1 = \int_{T_1}^{T_2} C_v\, dT$$

$$= \int_{T_1}^{T_2} (b + KT)\, dT$$

$$= \left[bT + K\frac{T^2}{2}\right]_{T_1}^{T_2} = b(T_2 - T_1) + \frac{K}{2}\left(T_2^2 - T_1^2\right)$$

$$= (T_2 - T_1)\left[b + K\frac{(T_2 + T_1)}{2}\right]$$

$$= (T_2 - T_1)(b + KT_m) \qquad \text{where, } T_m = \frac{T_1 + T_2}{2}$$

$C_{vm} = b + KT_m$ (C_{vm} mean specific heat at constant volume)

$$\therefore u_2 - u_1 = C_{vm}\,(T_2 - T_1) \tag{2.9}$$

Change of Enthalpy

- The small change in enthalpy of a unit mass of a gas for small change in temperature (dT) is given by:

$$dh = C_p\, dT$$

$$\therefore h_2 - h_1 = \int_{T_1}^{T_2} C_p\, dT$$

$$= \int_{T_1}^{T_2} (a + KT)\, dT$$

$$= \left[aT + K\frac{T^2}{2} \right]_{T_1}^{T_2} = a\left(T_2 - T_1\right) + \frac{K}{2}\left(T_2^2 - T_1^2\right)$$

$$= \left(T_2 - T_1\right)\left[a + K\frac{\left(T_2 + T_1\right)}{2} \right]$$

$$= \left(T_2 - T_1\right)\left(a + KT_m\right) \qquad \text{where, } T_m = \frac{T_1 + T_2}{2}$$

$C_{pm} = a + KT_m$ (C_{pm} mean specific heat at constant pressure)

$$\therefore h_2 - h_1 = C_{pm}\left(T_2 - T_1\right) \tag{2.10}$$

Isentropic expansion with variable specific heats

- Consider one kg of air, the heat transfer to a system using first law can be written as

$$dQ = du + dW$$

$$dQ = C_v\, dT + pdv$$

- For isentropic process, $dQ = 0$

$$\therefore C_v\, dT + pdv = 0$$

$$\therefore C_v \frac{dT}{T} + \frac{p}{T}dv = 0$$

$$\therefore C_v \frac{dT}{T} + R\frac{dv}{v} = 0 \qquad \left(\because pv = RT\right)$$

- Putting the values of R and C_v in the above equation, we get

$$\therefore \left(b + KT\right)\frac{dT}{T} + \left(a - b\right)\frac{dv}{v} = 0$$

- Integrating both sides we get

$$\therefore \int \left(b + KT\right)\frac{dT}{T} + \int \left(a - b\right)\frac{dv}{v} = \text{constant}$$

$$\therefore \int b\frac{dT}{T} + K\int dT + \left(a - b\right)\int \frac{dv}{v} = \text{constant}$$

$$\therefore b \log_e T + KT + (a - b) \log_e v = \text{constant}$$

$$\therefore \log_e T^b + \log_e e^{KT} + \log_e v^{(a-b)} = \text{constant}$$

$$\therefore T^b e^{KT} v^{(a-b)} = \text{constant}$$

$$\therefore T e^{\frac{K}{b}T} v^{\left(\frac{a}{b} - 1\right)} = \text{constant} \tag{2.11}$$

$$\therefore \frac{T}{v} e^{\frac{K}{b}T} v^{\frac{a}{b}} = \text{constant} \tag{2.12}$$

$$pv = RT \Rightarrow \frac{T}{v} = \frac{p}{R} = \frac{p}{a - b}$$

- Inserting the value of above equation in eq. 2.13.

$$\therefore \quad \frac{p}{a-b} e^{\frac{K}{b}T^{\frac{a}{b}}} v^{b} = \text{constant}$$

$$\therefore \quad pv^{b} e^{-\frac{a}{b}\frac{KT}{b}} = \text{constant} \qquad (2.13)$$

Effect of variable specific heats on air standard efficiency of Otto and diesel cycle

Otto cycle

- The air standard efficiency of Otto cycle is given by

$$\eta = 1 - \frac{1}{r^{\gamma-1}}$$

$$Now,\ C_p - C_v = R$$

$$\therefore \quad \frac{C_p}{C_v} - 1 = \frac{R}{C_v}$$

$$\therefore \quad \gamma - 1 = \frac{R}{C_v} \qquad \left(\because \frac{C_p}{C_v} = \gamma \right) \qquad (2.14)$$

$$\eta = 1 - \frac{1}{r^{\frac{R}{C_v}}} = 1 - r^{-\frac{R}{C_v}}$$

$$\therefore \quad 1 - \eta = (r)^{-\frac{R}{C_v}}$$

- Taking log on both sides, we have

$$\therefore \log_e (1-\eta) = -\frac{R}{C_v} \log_e (r)$$

- Differentiating the above equation, we have

$$\therefore -\frac{1}{1-\eta}\frac{d\eta}{dC_v} = -R \log_e r \left(-\frac{1}{C_v^2} \right)$$

$$\therefore \frac{d\eta}{1-\eta} = -\frac{R}{C_v} \cdot \log_e r \cdot \frac{dC_v}{C_v}$$

$$\therefore \frac{d\eta}{\eta} = -\frac{1-\eta}{\eta} \cdot (\gamma - 1) \cdot \log_e r \cdot \frac{dC_v}{C_v} \qquad (2.15)$$

- Negative sign indicates the decrease in efficiency with increase in C_v.
- The Eq. 2.15 gives the percentage variation in air standard efficiency of Otto cycle on account of percentage variation in C_v.

Diesel Cycle

- The air standard efficiency of diesel cycle is given by

$$\eta = 1 - \frac{1}{(r)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\gamma(\rho-1)}\right]$$

$$\therefore 1-\eta = \frac{1}{(r)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\gamma(\rho-1)}\right]$$

- Taking log on both sides, we get

$$\therefore \log(1-\eta) = \log\left(\rho^{\gamma}-1\right) - \log(r)^{\gamma-1} - \log\gamma - \log(\rho-1)$$

$$\therefore \log(1-\eta) = \log\left(\rho^{\gamma}-1\right) - (\gamma-1)\log r - \log\gamma - \log(\rho-1)$$

- Differentiating the above equation with respect to γ

$$\therefore -\frac{1}{1-\eta}\cdot\frac{d\eta}{d\gamma} = \frac{1}{\rho^{\gamma}-1}\cdot\rho^{\gamma}\log_e\rho - \log_e r - \frac{1}{\gamma}$$

$$\therefore \frac{d\eta}{d\gamma} = (1-\eta)\left[\log_e r - \frac{\rho^{\gamma}\log_e\rho}{\rho^{\gamma}-1} + \frac{1}{\gamma}\right]$$

- Multiplying the above equation by $\frac{d\gamma}{\eta}$

$$\therefore \frac{d\eta}{\eta} = \left(\frac{1-\eta}{\eta}\right)\left[\log_e r - \frac{\rho^{\gamma}\log_e\rho}{\rho^{\gamma}-1} + \frac{1}{\gamma}\right]\cdot d\gamma \qquad (2.16)$$

- Eq. 2.14 is $\gamma - 1 = \frac{R}{C_v}$, differentiating this equation with respect to C_v

$$\therefore \frac{d\gamma}{dC_v} = -\frac{R}{C_v^2} \Rightarrow \qquad d\gamma = -\frac{R}{C_v}\cdot\frac{dC_v}{C_v}$$

$$d\gamma = -(\gamma-1)\cdot\frac{dC_v}{C_v} \qquad (2.17)$$

- Inserting the value of Eq. 2.17 into Eq. 2.16, we get

$$\therefore \frac{d\eta}{\eta} = -\frac{1-\eta}{\eta}\cdot(\gamma-1)\left[\log_e r - \frac{\rho^{\gamma}\log_e\rho}{\rho^{\gamma}-1} + \frac{1}{\gamma}\right]\cdot\frac{dC_v}{C_v} \qquad (2.18)$$

Dissociation

- Dissociation process can be considered as the disintegration of combustion products at high temperature.
- Dissociation can also be looked as the reverse process to combustion. During dissociation the heat is absorbed whereas during combustion the heat is liberated.
- In IC engines, mainly dissociation of CO_2 into CO and O_2 occurs, whereas there is a very little dissociation of H_2O.
- The dissociation of CO_2 into CO and O_2 starts commencing around 1000 °C and the reaction equation can be written as

$$2CO_2 + Heat \rightleftharpoons 2CO + O_2$$

- Similarly, the dissociation of H_2O occurs at temperatures above 1300 °C and written as

$$2H_2O + Heat \rightleftharpoons 2H_2 + O_2$$

- The presence of CO and O_2 in the gases tends to prevent dissociation of CO_2; this is noticeable in a rich fuel mixture, which, by producing more CO, suppresses dissociation of CO_2.
- On the other hand, there is no dissociation in burnt gases of a lean fuel-air mixture. This is mainly due to the fact that temperature produced is too low for this phenomenon to occur.
- Hence, the maximum extent of dissociation occurs in the burnt gases of the chemically correct fuel-air mixture when the temperatures are expected to be high but decreases with the leaner and richer mixtures.
- In case of internal combustion engines heat transfer to the cooling medium causes a reduction in the maximum temperature and pressure. As the temperature falls during the expansion stroke the separated constituents recombine; the heat absorbed during dissociation is thus again released, but it is too late in the stroke to recover entirely the lost power. A portion of this heat is carried away by the exhaust gases.

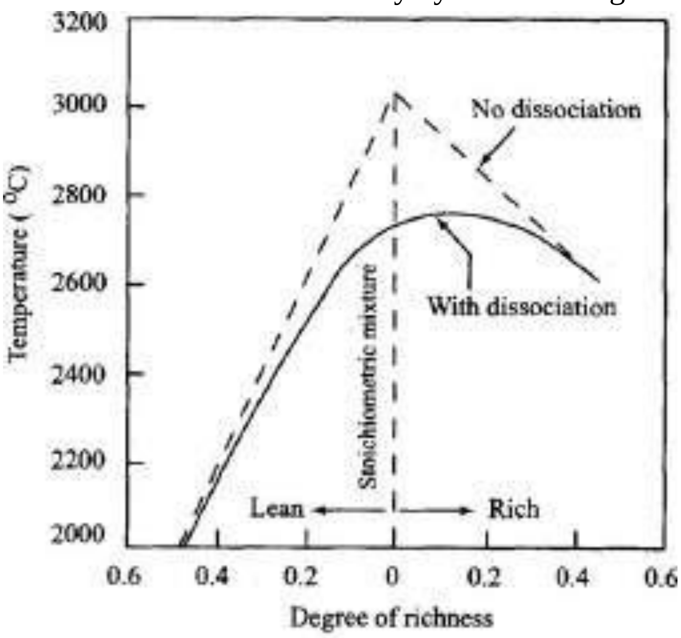


Fig. 2.2 Effect of dissociation on temperature

- Figure 2.2 shows a typical curve that indicates the reduction in the temperature of the exhaust gas mixtures due to dissociation with respect to air-fuel ratio. With no dissociation maximum temperature is attained at chemically correct air-fuel ratio.
- With dissociation maximum temperature is obtained when mixture is slightly rich. Dissociation reduces the maximum temperature by about 300 °C even at the chemically correct air-fuel ratio. In the Fig. 2.2, lean mixtures and rich mixtures are marked clearly.

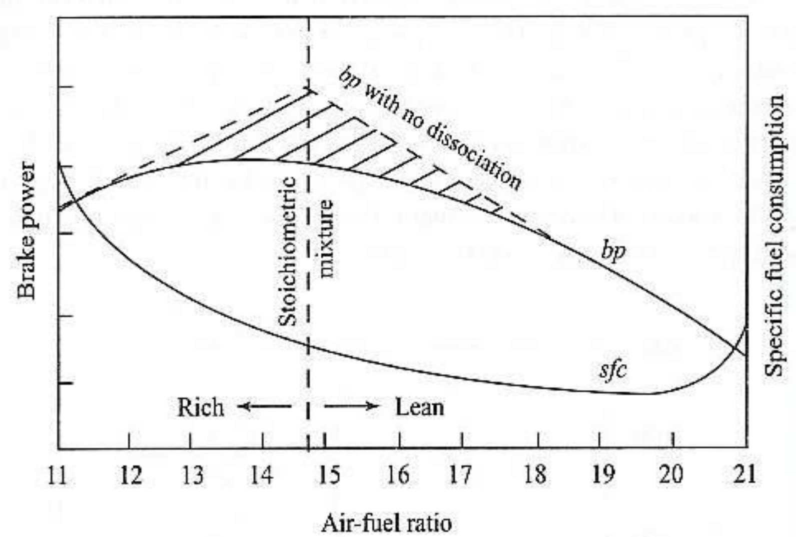


Fig. 2.3 Effect of dissociation on power

- The effect of dissociation on output power is shown in Fig.2.3 for a typical four-stroke spark-ignition engine operating at constant speed. If there is no dissociation the brake power output is maximum when the mixture ratio is stoichiometric.
- The shaded area between the brake power graphs shows the loss of power due to dissociation. When the mixture is quite lean there is no dissociation. As the air-fuel ratio decreases i.e., as the mixture becomes rich the maximum temperature raises and dissociation commences.
- The maximum dissociation occurs at chemically correct mixture strength. As the mixture becomes richer, dissociation effect tends to decline due to incomplete combus•tion.
- Dissociation effects are not so pronounced in a Cl engine as in an SI engine. This is mainly due to

  (i) the presence of a heterogeneous mixture and

  (ii) excess air to ensure complete combustion.

  Both these factors tend to reduce the peak gas temperature attained in the Cl engine.

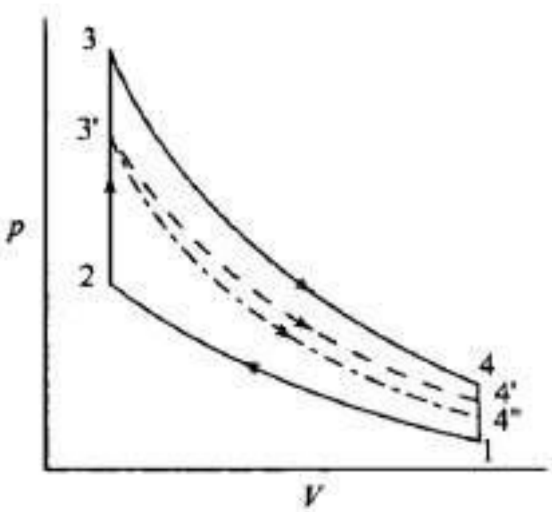


Fig. 2.4 Effect of dissociation shown on a p-V diagram

- Figure 2.4 shows the effect of dissociation on p-V diagram of Otto cycle. Because of lower maximum temperature due to dissociation the maximum pressure is also reduced and the state after combustion will be represented by 3' instead of 3. If there was no reassociation due to fall of temperature during expansion the expansion process would be represented by 3'-4" but due to reassociation the expansion follows the path 3'-4'.

- By comparing with the ideal expansion 3-4, it is observed that the effect of dissociation is to lower the temperature and consequently the pressure at the beginning of the expansion stroke. This causes a loss of power and also efficiency. Though during recombining the heat is given back it is too late to contribute a convincing positive increase in the output of the engine.

Effect of operating variables

The effect of common engine operating variables on the pressure and temperature within the engine cylinder is better understood by fuel-air cycle analysis. The details are discussed in this section:

Compression Ratio

- The fuel-air cycle efficiency increases with the compression ratio in the same manner as the air-standard cycle efficiency, principally for the same reason (more scope of expansion work. This is shown in fig 2.5.

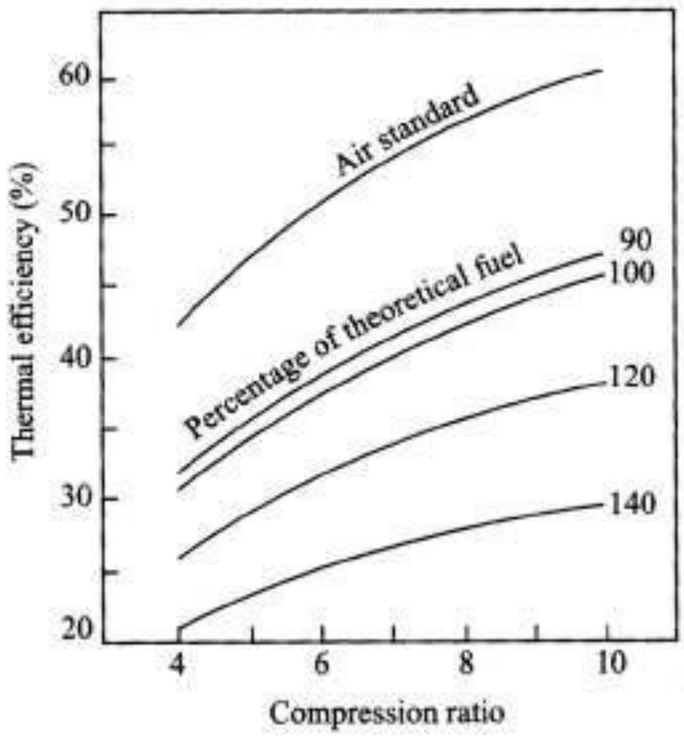


Fig. 2.5 Effect of compression ratio and mixture strength on efficiency

- The variation of indicated thermal efficiency with respect to the equivalence ratio for various compression ratios is given in fig 2.6. The equivalence ratio, ϕ , is defined as ratio of actual fuel-air ratio to chemically correct fuel-air ratio on mass basis.

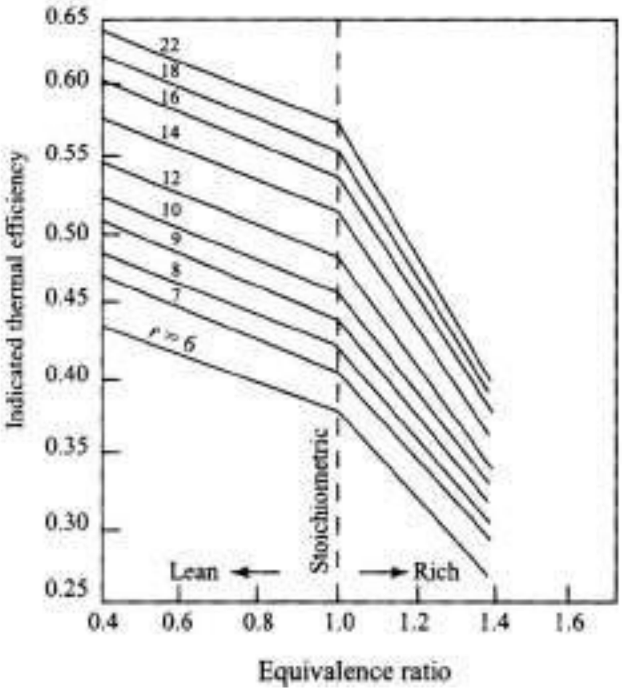


Fig. 2.6 Effect of mixture strength on thermal efficiency for various compression ratios

- The maximum pressure and temperature increase with compression ratio since the temperature, T_2, and pressure, p_2, at the end of compression are higher. However, it can be noted from the experimental results that the ratio of fuel-air cycle efficiency to air-standard efficiency is independent of the compression ratio for given equivalence ratio for the constant volume fuel-air cycle.

Fuel Air ratio

a) Efficiency

- 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 Fig. 2.7.

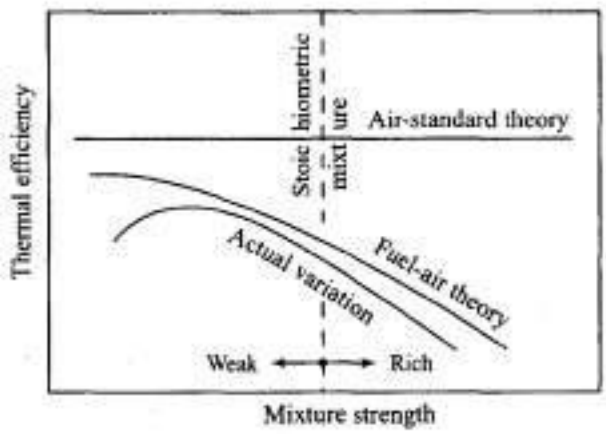


Fig. 2.7 Effect of mixture strength on thermal efficiency

b) Maximum Power

- Fig. 2.8 gives the cycle power as affected by fuel-air ratio. By air-standard theory maximum power is at chemically correct mixture, but by fuel-air theory maximum power is when the mixture is about 10% rich. As the mixture becomes richer the efficiency falls rapidly.

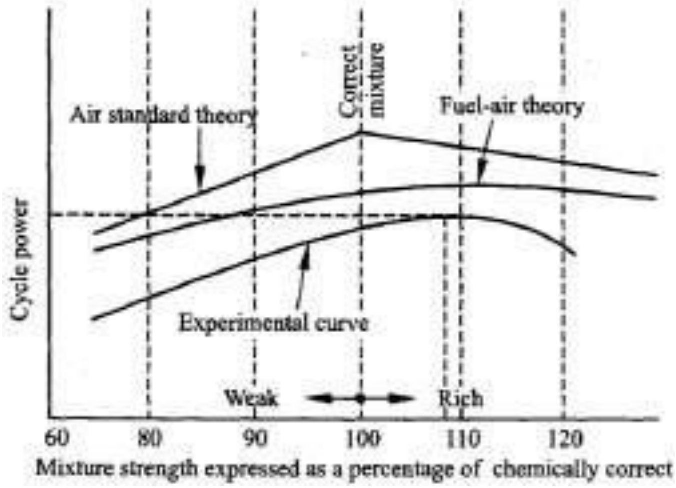


Fig. 2.8 Effect of fuel-air ratio on power

- This is because in addition to higher specific heats and chemical equilibrium losses, there is insufficient air which will result in formation of CO and H_2 in combustibles, which represents a direct wastage of fuel.

c) Maximum temperature

- At a given compression ratio the temperature after combustion reaches a maximum when the mixture is slightly rich, i.e., around 6 % or so (F/A = 0.072 or A/F = 14:1) as shown in Fig. 2.9.
- At chemically correct ratio there is still some oxygen present at the point 3 because of chemical equilibrium effects a rich mixture will cause more fuel to combine with oxygen at that point thereby raising the temperature T_3. However, at richer mixtures increased formation of CO counters this effect.

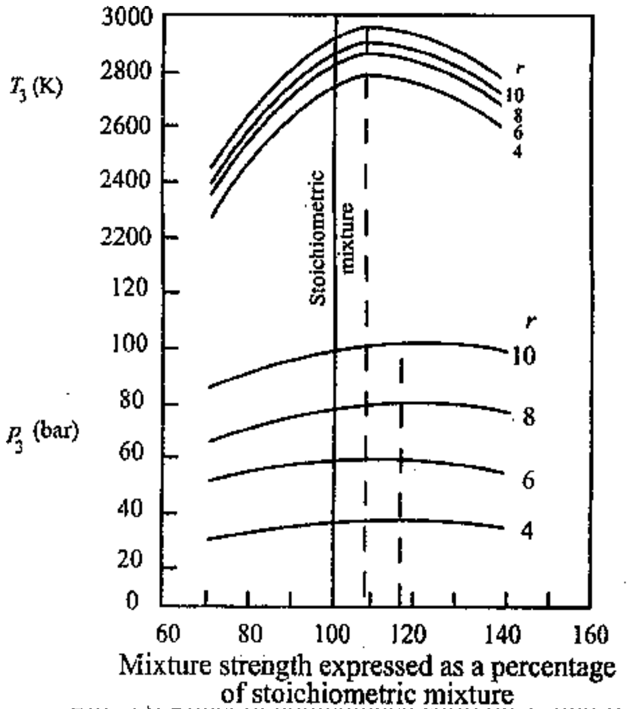


Fig. 2.9 Effect of equivalence ratio on T_3 and P_3

d) Maximum Pressure

- The pressure of a gas in a given space depends upon its temperature and the number of molecules The curve of p_3, therefore follows T_3, but because of the increasing number of molecules p_3 does not start to decrease until the mixture is somewhat richer than that for maximum T_3 (at F/A = 0.083 or A/F = 12:1), i.e. about 20 per cent rich (Fig.2.9).

e) Exhaust Temperature

- The exhaust gas temperature, T_4 is maximum at the chemically correct mixture as shown in Fig. 2.10. At this point there is reassociation as the temperature decrease so heat will be released these heat cannot be used in engine cylinder so the exhaust gases carry these heat with them and it result in higher exhaust temperature.

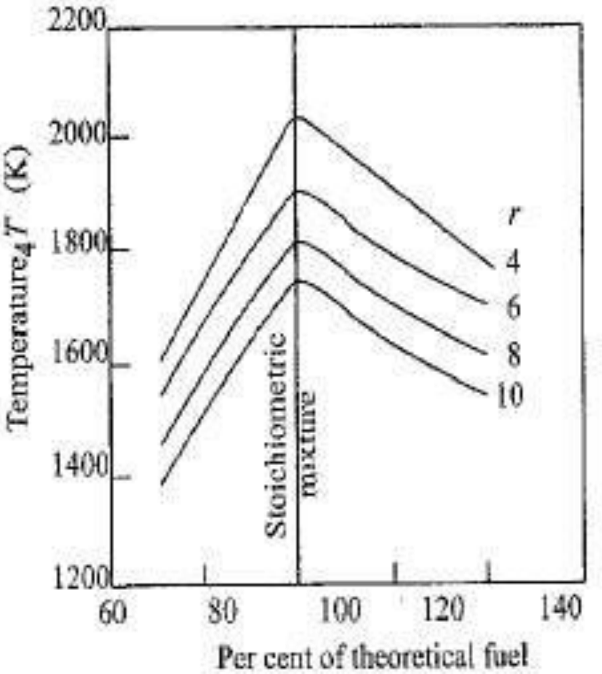


Fig. 2.10 Effect of fuel-air ratio on the exhaust gas temperature

- At lean mixtures, because of less fuel, T_3 is less and hence T_4 is less. At rich mixtures less sensible energy is developed and hence T_4 is less. That is, T_4 varies with fuel-air ratio in the same manner as T_3 except that maximum T_4 is at the chemically correct fuel-air ratio in place of slightly rich fuel-air ratio (6 %) as in case of T_3.
- However, the behaviour of T_4 with compression ratio is different from that of T_3 as shown in Fig. 2.10 Unlike T_3, the exhaust gas temperature, T_4 is lower at high compression ratios, because the increased expansion causes the gas to do more work on the piston leaving less heat to be rejected at the end of the stroke. The same effect is present in the case of air-cycle analysis also.

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:

- The working substance being a mixture of air and fuel vapour or finely atomized liquid fuel in air combined with the products of combustion left from the previous cycle.
- The change in chemical composition of the working substance.
- The variation of specific heats with temperature.
- The change in the composition, temperature and actual amount of fresh charge because of the residual gases.
- The progressive combustion rather than the instantaneous combustion.
- The heat transfer to and from the working medium
- The substantial exhaust blowdown loss, i.e., loss of work on the expan•sion stroke due to early opening of the exhaust valve.
- Gas leakage, fluid friction etc., in actual engines.

Most of the factors listed above tend to decrease the thermal efficiency and power output of the actual engines. On the other hand, the analysis of the cycles while taking these factors into account clearly indicates that the estimated thermal efficiencies are not very different from those of the actual cycles.

Deviation of Actual cycle from Fuel-Air cycle

- Major deviation from of actual cycle from the Fuel air cycle is due to
  - Variation in Specific heats
  - Dissociation
  - Progressive combustion
  - Incomplete combustion of fuel
  - Time loss factor
  - Heat loss factor
  - Exhaust blowdown factor

Time losses

Time losses may be burning time loss and spark timings loss.

Burning time loss

- In theoretical cycle, the burning is assumed to be instantaneous but actually burning takes some time. The time required depends upon F:A ratio, fuel chemical structure and its ignition temperature. This also depends upon the flame velocity and the distance from the ignition point to the opposite side of combustion chamber.
- During combustion, there is always increase in volume. The time internal between the spark and complete burning of the charge is approximately 40° crank rotation.

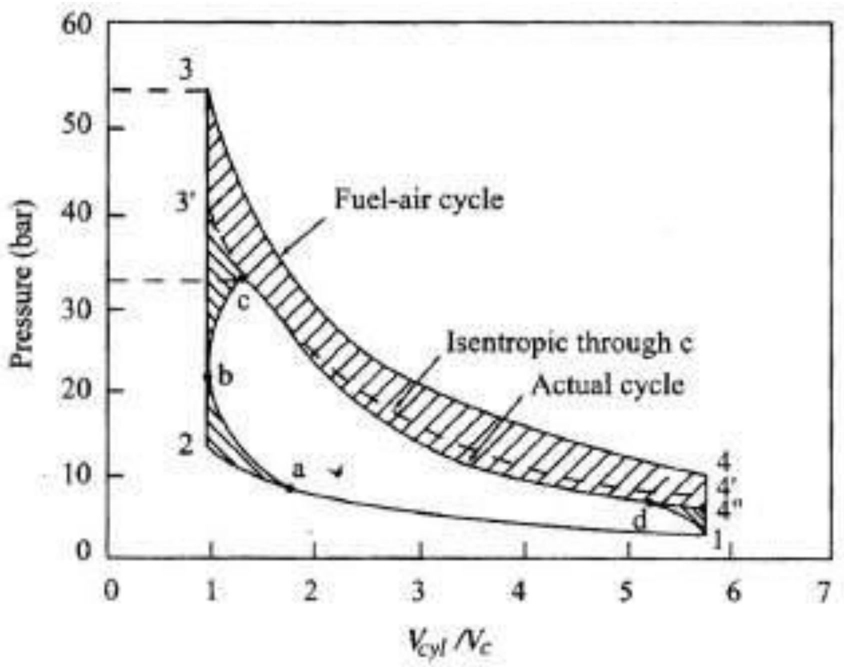


Fig. 2.11 Effect of time losses on p-V diagram

- The effect of time required for combustion; the maximum pressure is not produced when volume is minimum (v_c) as expected. It is produced some time after TDC. Therefore, the pressure rises from b to c as shown in Fig. 2.11.
- The point 3 represents the maximum pressure if the combustion should have taken place instantly. The difference in area of actual cycle and fuel-air cycle shows the loss of power as shown in Fig. 2.11. This loss of work is called burning time loss. This time loss is defined as the loss of power due to time required for mixing the fuel with air and for complete combustion.

b) Spark Timing Loss

- A definite time is required to start the burning of fuel after generating the spark in the cylinder. The effect of this, the maximum pressure is not reached at TDC and it reaches late during the expansion stroke. The time at which the burning starts is varied by varying the angle of advance (spark advance).
  - If the spark is given at T.D.C., the maximum pressure is low due to expansion of gases.
  - If the spark is advanced by 40° to start combustion at T.D.C., the combustion takes place at T.D.C. But the heat loss and the exhaust loss may be higher and again work obtained is not optimum.
- In the above two cases, the work area is less, and, therefore, power developed per cycle and efficiency are lower.

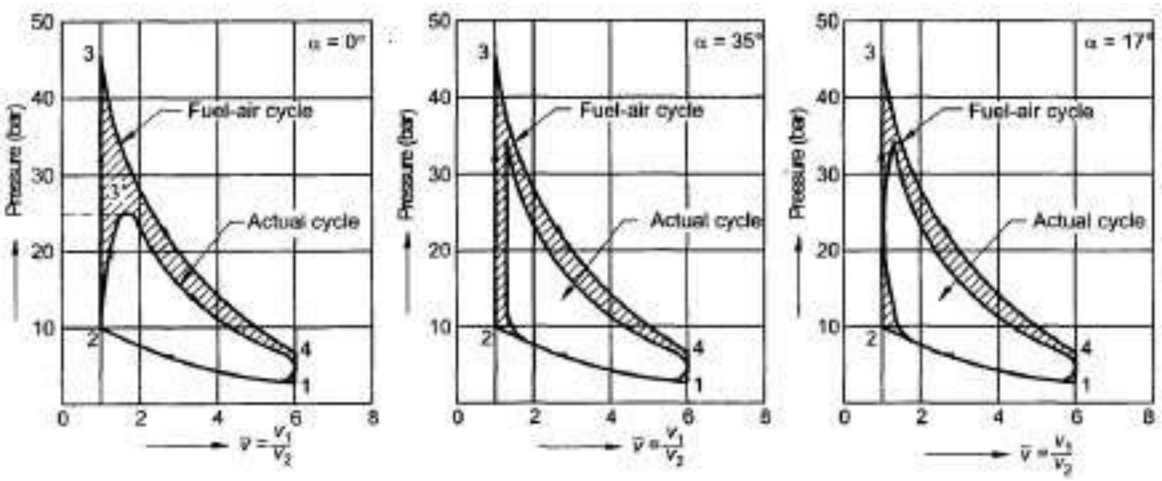


Fig. 2.12 Effects of angle of advance a on p - v diagram

- Thus for getting maximum work output, a moderate spark advance of 15° to 25° is the best.

c) Incomplete Combustion Loss

- The time loss always includes a loss due to incomplete combustion. It is impossible obtain perfect homogeneous air-fuel mixture. Fuel vapour, air, and residual gas is present in the cylinder before ignition takes place. Under these circumstances it is possible to have excess oxygen in one part of the cylinder and excess fuel in another part of it. Therefore, some fuel does not burn or burns partially. Both CO and O_2 will appear in the exhaust.

- It should be noted that it is necessary to use a lean mixture to eliminate fuel wastage while a rich mixture is required to utilize all the oxygen. Slightly leaner mixture will give maximum efficiency but too lean a mixture will burn slowly, increasing the losses or will not burn at all causing total waste. In the rich mixture some of the fuel will not get oxygen and will be completely wasted. Also, the flame speed in the rich mixture is low, thereby increasing the time losses and lowering the efficiency.

Direct heat loss

- During the combustion process and expansion process, the gases inside the engine cylinder are at a considerably higher temperature, so the heat is lost to the jacket cooling water or air. Some heat is lost to the lubricating oil where splash lubrication system is used for lubricating cylinder and piston.
- The loss of heat which takes place during combustion has the maximum effect, while that lost before the end of the expansion stroke has little effect, since it can do very small amount of useful work.
- During combustion and expansion, about 15% of the total heat is lost. Out of this, however, much is lost too late in the cycle to have done any useful work.
- In case all heat loss is recovered, about 20 percent of it may appear as useful work.

Exhaust blowdown loss

- At the end of exhaust stroke, the cylinder pressure is about 7 bar. If the exhaust valve is opened at B.D.C., the piston has to do work against high cylinder pressure costing part of the exhaust stroke. When the exhaust valve is opened too early entire part of the expansion stroke is lost.
- Thus, best compromise is that exhaust valve be opened 40° to 70° before B.D.C., thereby, reducing the cylinder pressure to halfway to atmosphere before the start of the exhaust stroke.

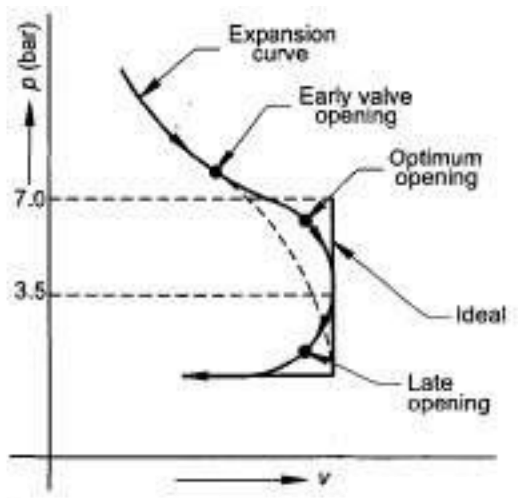


Fig. 2.13 Effect of blow down

Pumping losses

- In case of ideal cycles the suction and exhaust processes were assumed to be at atmospheric pressure. However some pressure differential is required to carry out the suction and exhaust processes between the fluid pressure and cylinder pressures.
- During suction the cylinder pressure is lower them the fluid pressure in order to induct the fluid into the cylinder and the exhaust gases are expelled at a pressure higher than the atmospheric pressure.

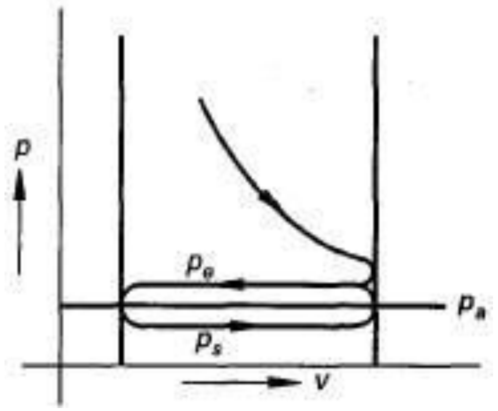


Fig. 2.14 Pumping Loss

- Therefore some work is done on the gases during suction and exhaust stroke. This work is called pumping work as shown in Fig. 2.14 by shaded area.

Rubbing Friction Losses

- The rubbing friction losses are caused due to
  - Friction between piston and cylinder walls
  - Friction in various bearings
  - Friction in auxiliary equipment such as pumps and fans.
- The piston friction increases rapidly with engine speed and to small extent by in•creases in m.e.p.
- The bearing and auxiliary friction also increase with engine speed.
- The engine efficiency is maximum at full load and reduces with the decrease in load. It is due to the fact that direct heat loss, pumping loss and rubbing friction loss increase at lower loads.

Valve and port timing diagrams

- The valve timing diagram shows the position of the crank when the various operations i.e., suction, compression, expansion exhaust begin and end.
- The valve timing is the regulation of the positions in the cycle at which the valves are set to at open and close
- The poppet valves of the reciprocating engines are opened and closed by cam mechanisms. The clearance between cam, tappet and valve must be slowly taken up and valve slowly lifted, at first, if noise and wear is to be avoided. For the same reasons the valve cannot be closed abruptly, else it will bounce on its seat. (Also, the cam contours should be so designed as to produce gradual and smooth changes in directional acceleration).
- Thus, the valve opening and closing periods are spread over a considerable number of crankshaft degrees. As a result, the opening of the valve must commence ahead of the time at which it is fully opened (i.e. before dead centres). The same reasoning applies for the closing time and the valves must close after the dead centres.

Valve timing diagram of 4-Stroke Petrol engine

- The actual valve timings used for low speed and high speed engines are shown in Fig. 2.15 (a) and (b).

a) Inlet valve

- The inlet valve opening occurs a few degrees prior to the arrival of the piston at TDC during the exhaust stroke. This is necessary to insure that the valve will be fully open and fresh charge starts to flow into the cylinder as soon as the piston starts to move down.
- If the inlet valve is allowed to close at BDC, the cylinder would receive less charge than its capacity and the pressure of the charge at the end of suction stroke will be below atmosphere. To avoid this, the inlet valve is kept open for 40°-50° rotation of the crank after the suction stroke for high speed engine and 20° to 25° for low speed engine.
- The kinetic energy of the charge produces a ram effect which packs more charge into the cylinder during this additional valve opening. Therefore, the inlet valve closing is delayed.
- Higher the speed of the engine, the inlet valve closing is delayed longer to take an advantage of ram effect.

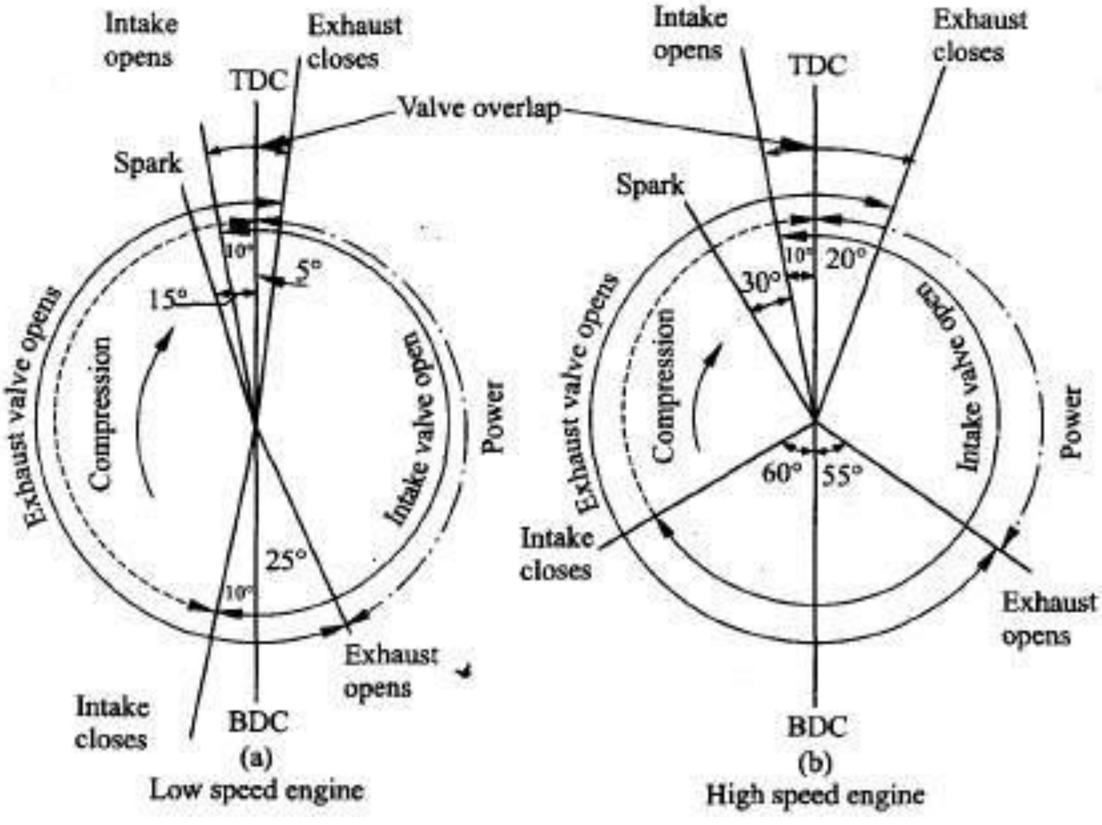


Fig. 2.15 Valve timing diagram for low and high speed 4-stroke SI engine

b) Exhaust valve

- The exhaust valve is set to open before BDC (say about 25° before BDC in low speed engines and 55° before BDC in high speed engines).

- If the exhaust valve did not start to open until BDC, the pressures in the cylinder would be considerably above atmospheric pressure during the first portion of the exhaust stroke, increasing the work required to expel the exhaust gases. But opening the exhaust valve earlier reduces the pressure near the end of the power stroke and thus causes some loss of useful work on this stroke.
- However, the overall effect of opening the valve prior to the time the piston reaches BDC results in overall gain in output.
- The closing time of exhaust valve effects the volumetric efficiency. By closing the exhaust valve a few degrees after TDC (about 15° in case of low speed engines and 20° in case of high speed engines) the inertia of the exhaust gases tends to scavenge the cylinder by carrying out a greater mass of the gas left in the clearance volume. This results in increased volumetric efficiency.

c) Ignition

- Theoretically it is assumed that spark is given at the TDC and fuel burns instantaneously. However, there is always a time lag between the spark and ignition of the charge. The ignition starts some time after giving the spark, therefore it is necessary to produce the spark before piston reaches the TDC to obtain proper combustion without losses. The angle through which the spark is given earlier is known as "**Ignition Advance**" or "**Angle of Advance**".

d) Valve Overlap

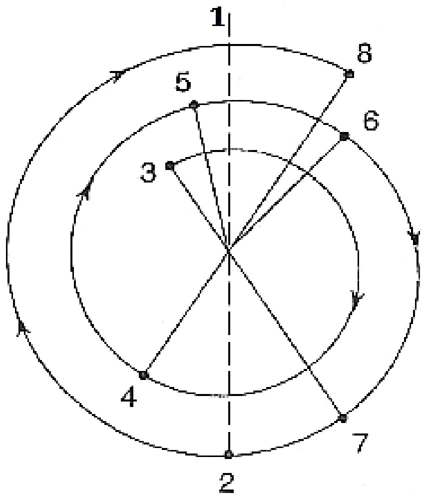
- From the valve timing diagram it is obvious that there will a period when both the intake and exhaust valves are open at the same time. This is called **valve overlap** (say about 15° in low speed engine and 30° in high speed engines). This overlap should not be excessive otherwise it will allow the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

Valve timing diagram of 4-Stroke Diesel engine

- The actual valve timing diagram of 4-Stroke Diesel cycle engine is shown in fig. 2.16. The various strokes are modified for similar reasons as explained in case of petrol engine.

Fuel Injection Timing

- The opening of fuel valve is necessary for better evaporation and mixing of the fuel. As there is always lag between ignition and supply of fuel, it is always necessary to supply the fuel little earlier.
- In case of diesel engine, the overlapping provided is sufficiently large compared with the petrol engine. More overlapping is not advisable in petrol engine because the mixture of air and petrol may pass out with the exhaust gases and it is highly uneconomical. This danger does not arise in case of diesel engine because only air is taken during the suction stroke.



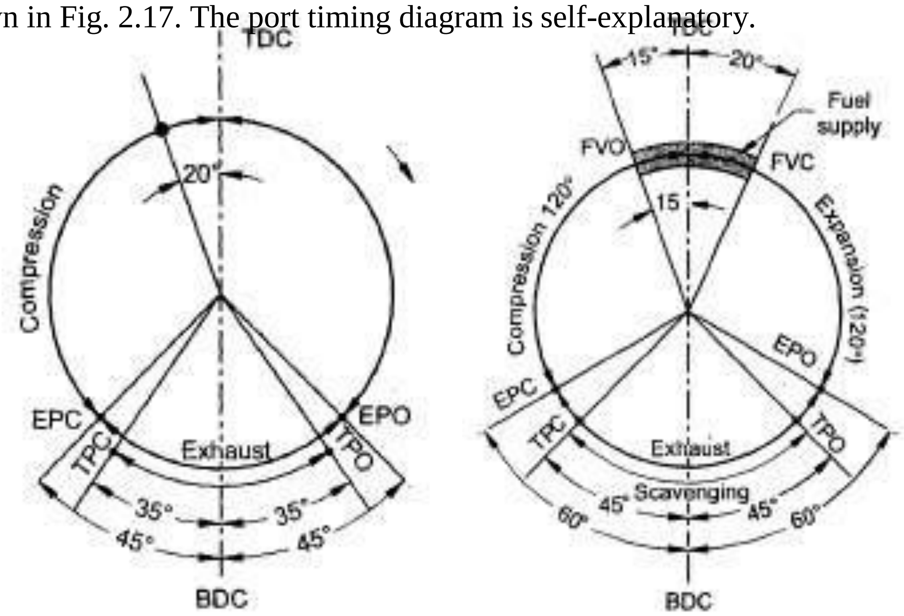
1 – TDC
2 – BDC
3 – Inlet Valve Opens 4 – Inlet Valve Closes 5 – Fuel Valve Opens 6 – Fuel Valve Closes
7 – Exhaust Valve Opens 8 – Exhaust Valve Closes

2.16 Valve Timing Diagram of 4-Stroke Diesel Cycle Engine

- The valve timing of diesel engine have to be adjusted depending upon the speed of the engine. The typical valve timings are as follows:
  - IV opens at 25^0 before TDC
  - IV closes at 30^0 after BDC
  - Fuel injection starts at 5^0 before TDC
  - Fuel injection closes at 25^0 after TDC
  - EV opens at 45^0 before BDC
  - EV closes at 15^0 after TDC

Port Timing Diagram of 2-stroke engine

- The port timing diagram for actual working of the two-stroke petrol and diesel engine is shown in Fig. 2.17. The port timing diagram is self-explanatory.



(a) Petrol *(b) Diesel*

Fig. 2.17 Port Timing Diagram for 2-stroke Engine

COMBUSTION

Course Contents

3.1. – Combustion
3.2. – Combustion equation
3.3. – Composition of air
3.4. – Mass fraction and mole fraction
Stoichiometry
3.6. – Determination of minimum air for complete combustion
Enthalpy of formation
Adiabatic flame temperature 3.9. – Calorific value of fuel and its determination

Course Contents
1.1 Introduction
1.2 Static Balancing
1.3 Types of Balancing
1.4 Balancing of Several Masses Rotating in the Same Plane
1.5 Dynamic Balancing
1.6 Balancing of Several Masses Rotating in the different Planes

Combustion

- In chemical thermodynamics the study of systems involving chemical reactions is important topic. A chemical reaction may be defined as the rearrangement of atoms due to redistribution of electrons. In a chemical reaction the terms, reactants and the products are frequently used.
- ‘Reactants’ comprise of initial constituents which start the reaction while ‘products’ comprise of final constituents which are formed by the chemical reaction. Although the basic principles which will be discussed in this chapter apply to any chemical reaction, here main attention will be focused on an important type of chemical reaction—“combustion”.
- Combustion is a chemical reaction that occurs when oxygen combines with other substances to produce heat and usually light.

Combustion equations

- All fuels contain combustible elements such as C, H_2 and S which readily combine with oxygen and evolve heat during combustion. It is always necessary to supply sufficient air for the complete combustion of fuels.
- The following chemical equations are used to calculate the amount of oxygen required, and the amount of gases produced by the combustion of fuel,
- The oxygen supplied for combustion is usually provided by atmospheric air, and it is necessary to use accurate and consistent analysis of air by mass and by volume. It is usual in combustion calculations to take air as 23.3% O_2, 76.7% N_2 by mass, and 21% O_2, 79% N_2 by volume.
- The small traces of other gases in dry air are included in nitrogen, which is sometimes called ‘atmospheric nitrogen'. Some important combustion equations are given below:

1) Combustion of Hydrogen:

$$2H_2 + O_2 \rightarrow 2H_2O$$

$$\text{2 moles of } H_2 + \text{1 mole of } O_2 \rightarrow \text{2 moles of } H_2O$$

$$\text{4 kg of } H_2 + \text{32 kg of } O_2 \rightarrow \text{36 kg of } H_2O$$

$$\text{1 kg of } H_2 + \text{8 kg of } O_2 \rightarrow \text{9 kg of } H_2O$$

2) Combustion of Carbon

a. Complete combustion of carbon to carbon dioxide

$$C + O_2 \rightarrow CO_2$$

$$\text{1 mole of C} + \text{1 mole of } O_2 \rightarrow \text{1 moles of } CO_2$$

$$\text{12 kg of C} + \text{32 kg of } O_2 \rightarrow \text{44 kg of } CO_2$$

$$\text{1 kg of C} + \frac{8}{3}\text{ kg of } O_2 \rightarrow \frac{11}{3}\text{kg of } H_2O$$

b. Incomplete combustion of carbon to carbon monoxide

2C	+	O_2	$\rightarrow$	2CO
2 mole of C	+	1 mole of O_2	$\rightarrow$	3 moles of CO
24 kg of C	+	32 kg of O_2	$\rightarrow$	56 kg of CO_2
1 kg of C	+	$\frac{4}{3}$ kg of O_2	$\rightarrow$	$\frac{7}{3}$ kg of H_2O

3) Combustion of Sulphur

S	+	O_2	$\rightarrow$	SO_2
1 mole of S	+	1 mole of O_2	$\rightarrow$	1 moles of SO_2
32 kg of S	+	32 kg of O_2	$\rightarrow$	64 kg of SO_2
1 kg of S	+	1 kg of O_2	$\rightarrow$	2 kg of SO_2

4) Combustion of Methane

CH_4	+	$2O_2$	$\rightarrow$	CO_2	+	$2H_2O$
1 mole of CH_4	+	2 mole of O_2	$\rightarrow$	1 moles of CO_2	+	2 mole of H_2O
16 kg of CH_4	+	64 kg of O_2	$\rightarrow$	44 kg of CO_2	+	36 kg of H_2O
1 kg of CH_4	+	4 kg of O_2	$\rightarrow$	$\frac{11}{4}$ kg of CO_2	+	$\frac{9}{4}$ kg of H_2O

Composition of Air

- Approximate composition of air by mass and by volume is shown in below table. Here we assume that air contains nitrogen and oxygen only the rests are negligible.

Table 3.3.1 Composition of Air

	By Volume	By Mass	Molecular Mass
N_2	79 % 3.76 moles	77 % 3.347 kg	32
O_2	21 % 1 mole	23 % 1 kg	28
Air	100 % **4.76 moles**	100 % **4.347 kg**	28.97 $\simeq$ 29

Mass fraction and Mole fraction

Mass Fraction

- It is defined as the ratio of mass of a constituent of a mixture or compound to the total mass of mixture or compound.

$$\text{Mass Fraction} = \frac{\text{Mass of a constituent in the mixture}}{\text{Mass of mixture}}$$

Mole fraction

- The ratio of moles of a constituent gas to the total moles of mixture of gases is called the mole fraction.

$$\text{Mole Fraction} = \frac{\text{Moles of a constituent in the mixture}}{\text{Total moles of mixture of gases}}$$

Stoichiometry

Air fuel Ratio:

- It is expressed in a mass basis and defined as A/F ratio = $\frac{m_a}{m_f}$.

Stoichiometric (Theoretical) Air

- The minimum amount of air needed for the complete combustion of a fuel is called the stoichiometric or theoretical air.
- A mixture of theoretical air and fuel is called Stoichiometric or Chemically correct mixture.

Excess Air

- In practice the combustion of fuel is never complete due to non-homogeneity of mixture.
- In order to ensure complete combustion of fuel, usually, actual air supplied is more than the theoretical air required for complete combustion of fuel. Then,

  Excess air = Actual air - Stoichiometric (theoretical) air

$$\%\ \text{excess air} = \frac{\text{Actual A/F ratio - Stoichiometric A/F ratio}}{\text{Stoichiometric A/F ratio}} \times 100$$

- The magnitude of excess air supplied depends on the homogeneity of mixture, turbulence and maximum pressure and temperature to be attained in combustion process.

Equivalence Ratio (Ø)

- It is a measure of the stoichiometric AF ratio relative to actual AF ratio.

$$\phi = \frac{AF_{st}}{AF_a} = \frac{FA_a}{FA_{st}}$$

  where AF = m_a/m_f, FA = m_f/m_a and FA = 1/AF.

- When $\phi = 1$, mixture is stoichiometric.
- **Lean mixture ($\phi < 1$):** If actual A.F. ratio is more than stoichiometric A.F. ratio, the mixture is said to **lean or weak mixture.**
- **Rich mixture ($\phi >$ (ϕ 1):** If actual A.F. ratio is less than stoichiometric A.F. ratio, the mixture is said to **rich** mixture.

Determination of Minimum Air Required Per kg of Solid or Liquid Fuel for Complete Combustion

- Let us consider 1 kg of fuel, the ultimate analysis of which shows that carbon is ‘C’ kg, hydrogen is ‘H’ kg, sulphur is ‘S’ kg and oxygen is ‘O’ kg. The amount of oxygen required can be computed with the help of combustion.
- From the combustion equations in section 3.2

  1 kg of carbon requires $\frac{8}{3}$ kg of O_2

  C kg of carbon requires $\left(\frac{8}{3}\right)$ kg of O_2

- Similarly,

  H kg of hydrogen requires (8 H) kg of O_2

  S kg of hydrogen requires (S) kg of O_2

  Total oxygen required = (8/3)C + 8H + S kg

- Since O kg of oxygen is already present in the fuel. Therefore, minimum oxygen required for complete combustion

  Min. Oxygen required for complete combustion

$$= \left(\frac{8}{3}\right) C + 8H + S - O = \frac{8}{3} C + 8\left(H - \frac{O}{8}\right) + S$$

- Air contains 23% of oxygen by mass and therefore kg of air required per kg of fuel for complete combustion is given by

$$= \frac{1}{100}\left[\frac{8}{3}C + 8\left(H - \frac{O}{8}\right) + S\right] \times \frac{100}{23}$$

$$= \frac{1}{23}\left[\frac{8}{3}C + 8\left(H - \frac{O}{8}\right) + S\right]$$

Enthalpy of formation

- A combustion reaction is a particular kind of chemical reaction in which products are formed from reactants with the release or absorption of energy as heat is transferred to and from the surroundings.
- In some substances like hydrocarbon fuels which are many in number and complex in structure the heat of reaction or combustion may be calculated on the basis of known values of the enthalpy of formation, ΔH_f of the constituent of the reactants and products at the temperature T_0 (reference temperature).
- The enthalpy of formation (ΔH_f) is the increase in enthalpy when a compound is formed from its constituent elements in their natural form and in a standard state.
- The standard state is 25°C, and 1 atm. pressure, but it must be borne in mind that not all substances ran exist in natural form, e.g. H20 cannot be a vapour at 1 atm. and 25°C.
- The expression of a particular reaction, for calculation purposes, may be given as:

$$\Delta H_0 = \sum_P n_i \Delta H_{f_i} - \sum_R n_i \Delta H_{f_i}$$

- Typical values of ΔH_f for different substances at 25℃ (298 K) in kJ/mole are given below:

Table 3.2 Values of Enthalpy of formation for different substances

Sr. No.	Substance	Formula	State	ΔH_f
1	Oxygen	O	Gas	249143
		O_2	Gas	0
2	Water	H_2O	Liquid	-285765
			Vapour	-241783
3	Carbon	C	Gas	714852
			Diamond	1900
			Graphite	0
4	Carbon Monoxide	CO	Gas	-111508
5	Carbon Dioxide	CO_2	Gas	-393443
6	Methane	CH_4	Gas	-74855
7	Methyl alcohol	CH_3OH	Vapour	-240532
8	Ethyl alcohol	C_2H_5OH	Vapour	-281102
9	Ethane	C_2H_6	Gas	-83870
10	Ethene (Ethylene)	C_2H_4	Gas	51780
11	Propane	C_3H_8	Gas	-102900
12	Butane	C_4H_{10}	Gas	-125000
13	Octane	C_8H_{18}	Liquid	-247600

Adiabatic flame temperature

- In a given combustion process that takes place adiabatically and with no work or changes in kinetic or potential energy involved, the temperature of the products is referred to as the adiabatic flame temperature.
- With the assumptions of no work and no changes in kinetic or potential energy, this is the maximum temperature that can be achieved for the given reactants because any heat transfer from the reacting substances and any incomplete combustion would tend to lower the temperature of the products.
- The following points are worth noting
  - i) The maximum temperature achieved through adiabatic complete combustion varies with the type of reaction and percent of theoretical air supplied.
  - ii) For a given fuel and given pressure and temperature of the reactants, the maximum adiabatic flame temperature that can be achieved is with a stoichiometric mixture.
  - iii) The adiabatic flame temperature can be controlled by the amount of excess air that is used. This is important, for example, in gas turbines, where the maximum permissible temperature is determined by metallurgical consideration in turbine, and close control of the temperature of the product is essential.

Calorific values of fuels and its determination

- The calorific value of the fuel is the amount of heat generated by burning unit mass or unit volume of the fuel. A fuel is more desirable whose heat generating capacity is high.
- If a fuel contains hydrogen, water will be formed as one of the products of combustion. If this water is condensed, a large amount of heat will be released than if water exist in the vapour phase.
- For this reason two heating values are defined; the higher or gross heating value and the lower or net heating value.
- The C.V. of the fuel determined by an experiment gives H.C.V. as the products of combustion are cooled to atmospheric temperature at which most water-vapour in the exhaust is cooled and condensed and its latent heat is given to the cooling medium.
- In actual practice, the latent heat of vapour in exhaust is carried with the water vapour and is not available. Therefore the L.C.V. of the fuel is calculated by deducting latent heat of water vapour from HCV (determined by experiment).

$$\therefore \qquad LCV = HCV - m_{H_2O} \cdot h_{fg}$$

where m_{H_2O} is the mass of water vapour formed per kg of fuel burned and h_{fg} is the latent heat of water vapour at partial water-vapour pressure in the exhaust gases.

- In actual practice, for all practical purposes, the LCV of the fuel is considered.
- Two different methods for finding HCV of liquid fuel and gaseous fuel are described here.

Bomb calorimeter

- The calorific value of powdered and liquid fuels is determined at constant volume in the bomb calorimeter. It resembles the shape of a bomb, and thus it is known as the bomb calorimeter. It is shown in Fig. 3.1.

Construction

- The fuel is burnt in a strong steel chamber, known as bomb, which is immersed in a known mass of water. The fuel sample is placed in a crucible inside the bomb, which is filled with oxygen under a pressure above 25 atm. It is then ignited by an electrically heated platinum wire. The combustion thus takes place at constant volume, the fuel bums almost in a constant-pressure environment due to the high pressure of oxygen. To reduce any losses of heat, the calorimeter is also provided with additional water jacket and air. A motor-driven stirrer is used to keep the water temperature uniform around the bomb and an accurate thermometer (Beckman type) is immersed in water to measure the temperature accurately.

Procedure

- A known quantity of fuel sample as a briquette is placed into the crucible and a fuse wire is connected with the electrodes as shown in Fig. 16.6. The bomb is then placed in a calorimeter with a weighed quantity of water. After making necessary connections, the stirrer is started and temperature measurements are taken every minute. At the end of the fifth minute, a charge is fired and temperature readings are taken carefully every 10 seconds during this period. When the temperature readings begin to fall, the frequency of readings may be reduced to one every minute.

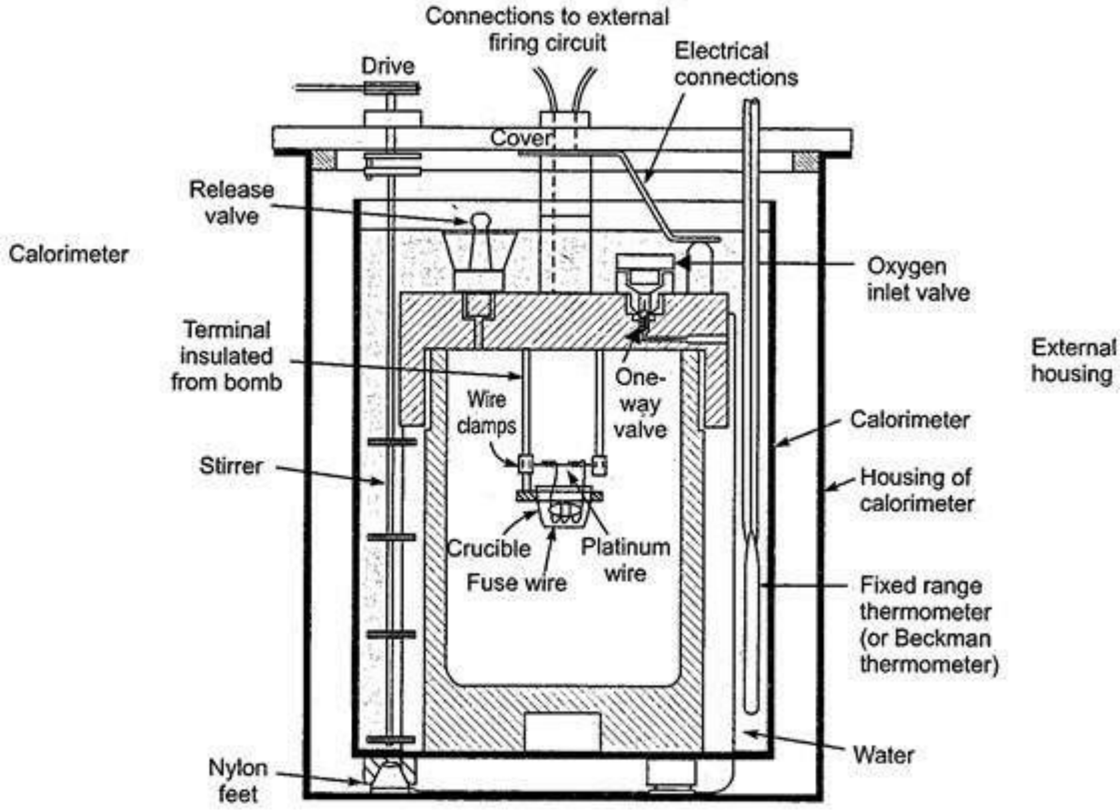


Fig.3.1 Sectional view of a bomb calorimeter

- After experimentation, the bomb is taken out from its housing. The products of combustion are released through the release valve. Then it is opened, and the unburnt fuse wire, if any, is collected and weighed. A temperature-time curve is plotted.
- The measured temperature rise is corrected for various losses. The allowance for combustion of fuse wire is determined from the weight of the fuse and its known calorific value. The water equivalence of a calorimeter must be used in calculation to accommodate its allowance.
- The heat released by combustion of fuel is absorbed by water surrounding the bomb and calorimeter. Thus an energy balance yields to

$$\left.\begin{array}{l}\text{Mass of fuel} \times \text{calorific}\\ \text{value + mass of fuse}\\ \text{wire burn} \times \text{caloric}\\ \text{value of fuse wire}\end{array}\right\} = \left\{\begin{array}{l}\text{Mass of water equivalent}\\ \text{of calorimeter} \times \text{specific}\\ \text{heat of water} \times \text{corrected}\\ \text{temperature rise}\end{array}\right\}$$

$$m_f CV + m_{fuse} CV_1 = (m_w + m_e) C_{pw} \left[(T_2 - T_1) + T_c \right]$$

$$CV = \frac{(m_w + m_e) C_{pw} \left[(T_2 - T_1) + T_c \right] - m_{fuse} CV_1}{m_f} \qquad (3.1)$$

where

T_c = radiation correction to temperature, it is obtained from graphical presentation of observation before and after firing

m_f = mass of fuel

m_{fuse} = mass of fuse water

m_w = mass of water filled in calorimeter

m_e = water equivalent of calorimeter

CV_1 = Calorific value of fuse wire

$T_2 - T_1$ = Observed temperature difference

- The bomb calorimeter measures a higher calorific value of fuel. If a liquid fuel is being tested, it is contained in a gelatin capsule and the firing may be assisted by paraffin of known calorific value in the crucible.

Junkers gas calorimeter

- Junkers gas calorimeter is shown in fig. 3.2. It is designed to burn a gaseous fuel under a steady flow conditions at atmospheric pressure.

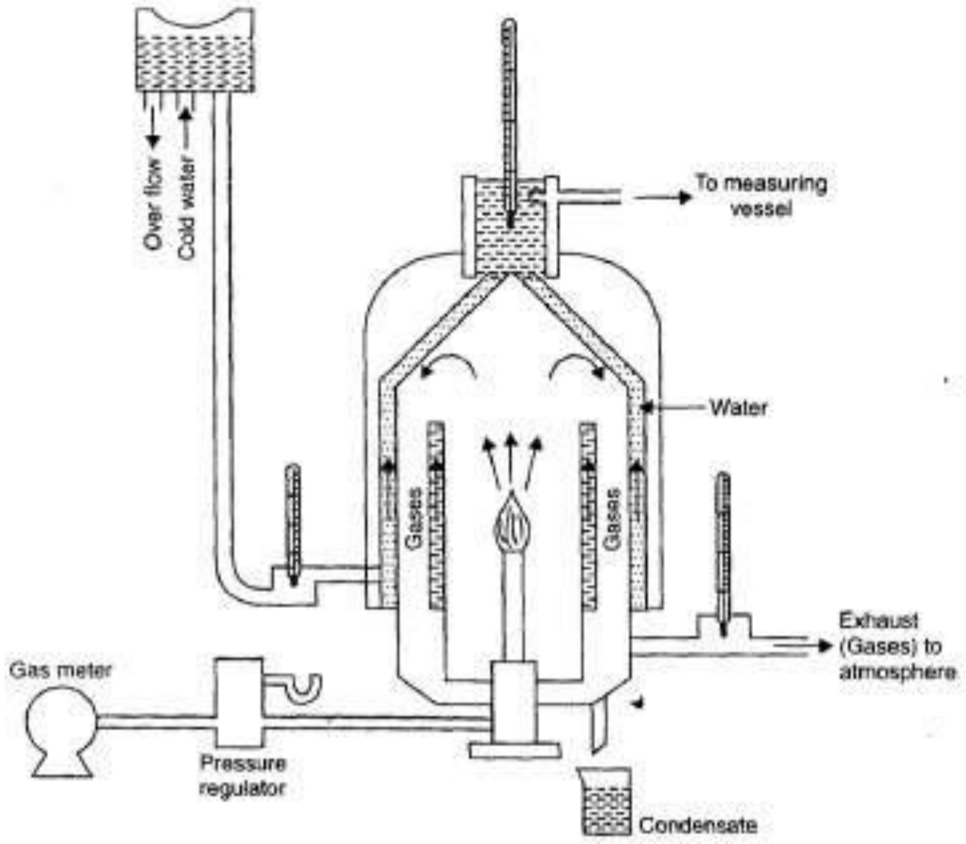


Fig. 3.2 Junkers Gas Calorimeter

- Heat is transferred from the products to water flowing steadily through the outer jacket of the calorimeter. The operating conditions are adjusted to obtain a gas outlet temperature equal to the inlet temperature of the fuel and combustion air.

- From observed water temperatures and measured quantities of fuel and jacket water, the heating value is calculated and reduced to the corresponding value for 25°C operation.
- Some of the water vapour in the products condenses and drains from the calorimeter in to a collecting vessel. This measured quantity of condensate is used in the sub sequent conversion of the calorimetric heating value to the constant pressure higher and lower heating values that are based, respectively, upon complete and zero condensation of the water vapour formed during the combustion reaction.

FUELS & ITS SUPPLY SYSTEM FOR SI AND CI ENGINE

Course Contents

Important qualities of IC engine fuels
Rating of Fuels
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Gasoline Injection
MPFI
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Classification of Injection System
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Fuels

- Heat engine is a device which converts heat energy into mechanical work. In an internal combustion engine the heat energy is released by burning fuel in the engine cylinder. The chemical reactions which permit the release of heat energy are quite fast but the time taken in preparing a proper mixture of fuel and air depends mainly upon the nature of fuel and method of introducing it into combustion chamber.

Important qualities of IC Engine fuels

- A good I.C. engine fuel must possess the following properties.
- It must have high energy density (kJ/kg).
- It should be easy to handle.
- It must possess good combustion qualities.
- It must have thermal stability.
- It must have low deposit forming tendency.
- It should be easy to handle and store.
- It should be free from hazard.
- It should not have chemical reactions with engine components, through which it flows.
- It should easily mix with air and evaporate as quickly as possible (low h_{fg}).
- Products of combustion should not be corrosive to the engine parts.
- It must possess low toxicity.
- Its effect on air-pollution should be minimum.
- It should be economically available in very large quantities.
- The basic requirement of I.C. engine fuel is, the combustion should be fast with maximum amount of heat release without forming any deposits and should not have destructive effects on the engine parts and atmospheric air by exhaust gases.

Rating of Fuels

Rating of SI engine fuels

- Fuels differ widely in their ability to resist the knocking and detonation in S.l. engines. It is expressed in terms of octane number.
- Fuel rating specifications require the standard engines operating under prescribed standard conditions.
- The rating of a particular fuel is compared on the standard engine with that of a standard reference fuel which is usually the combination of iso-octane (2,2,4 trimethyl pentane) and n-heptane (C7 H16) by volume.
- Arbitrarily the iso-octane is assigned a rating of 100 octane number since this fuel has minimum knocking tendency and the n-heptane is assigned a rating of zero octane because of its high tendency to knock.
- The percentage volume of iso-octane in the mixture of iso-octane and n-heptane represents the octane number rating of a fuel.

- If a fuel is assigned a knock rating of 80, it means the fuel has the same tendency to knock under standard operating conditions as the mixture of standard fuel having 80% iso-octane and 20% of n-heptane fuels by volume.
- Hence the octane number rating of the fuel is an expression which indicates the ability of a fuel to resist knock or detonation in SI Engines.

Method of Determination Octane Rating of Fuel:

- Octane rating of fuels is determined by testing a fuel on a variable compression co-operative fuel research (CFR) engine under specified conditions.
- The fuel to be tested for knock rating is used in this engine to produce standard knock by varying the compression ratio under standard operating conditions. Knockmeter reading is noted.
- The fuel to be tested for knock rating is used in this engine to produce standard knock by varying the compression ratio under standard operating conditions. Knockmeter reading is noted.

Performance Number (PN)

- Certain fuels show even less tendency to knock than iso-octane fuel i.e. they have octane number more than 100. In order to extend the octane scale, the knock rating of fuel is measured in terms Army-Navy performance number represented by "PN".
- It is defined as the ratio of knock limited indicated mean effective pressure ($KL_{i.m.e.p}$) of the fuel under test to the knock limited indicated mean effective pressure of iso-octane.

$$PN = \frac{KL_{i.m.e.p} \text{ of test fuel}}{KL_{i.m.e.p} \text{ of iso-octane}} \tag{4.1}$$

- In certain cases the knock rating of fuel can be improved by adding tetra ethyl lead (TEL) and when added to iso-octane it shows improved anti-knock characteristics.
- If 'x' ml of TEL is added to a U.S. gallon of iso-octane, the octane number (ON > 100) is expressed as (100 + x) ml of TEL.
- Another method of octane scale is given by Wiese, and expressed as,

$$ON = 100 + \frac{PN - 100}{3} \tag{4.2}$$

Rating of CI engine fuels

- Increased delay period or ignition lag promotes knocking in C.I. engines. The property of ignition lag is generally measured by cetane number.
- The fuel cetane ($C_{16}H_{34}$) is straight chain paraffin with good ignition qualities and it is arbitrarily assigned a rating of 100 cetane number.
- While the hydrocarbon fuel alpha-methyl-naphthalene ($C_{10}H_7CH_3$) has poor ignition quality and it is assigned zero cetane number.
- These two fuels are mixed by volume and the mixture is matched with a fuel under test in a standard engine running under prescribed conditions.

- The cetane number of a fuel is defined as the percentage by volume of Cetane in a mixture of cetane ($C_{16}H_{34}$) and alpha-methyl-naphthalene that produces the same delay period or ignition lag as the fuel being tested under same operating conditions on the same engine.
- Higher the cetane number of fuel lesser will be the tendency for diesel knock.

Diesel Index

- An alternative method of expressing the quality of diesel fuel is called diesel index.
- It is defined as

$$\text{Diesel Index} = \frac{\text{Aniline point }(^{\circ}\text{F}) \times \text{API gravity at 15 °C}}{100} \tag{4.3}$$

where

Aniline point represents the lowest temperature at which the diesel fuel is completely miscible with an equal volume of aniline.

American Petroleum Institute (API) gravity is the density of diesel oil and it is expressed as

$$\text{API gravity} = \frac{141.5}{\text{Specific gravity at 15 °C}} - 131.5 \tag{4.4}$$

Fuel supply system for SI Engine

- Spark ignition engine normally use volatile liquid fuels. Preparation of fuel-air mixture is done outside the engine cylinder and formation of a homogeneous mixture is normally not completed in the inlet manifold. Fuel droplets which remain in suspension continue to evaporate and mix with air even during suction and compression processes. The process of mixture preparation is extremely important for SI engines. The purpose of carburetion is to provide a combustible air-fuel mixture in the required quantity and quality for efficient operation of the engine under all conditions.

Carburetion

- The process of formation of a combustible fuel-air mixture by mixing the proper amount of fuel with air before admission to engine cylinder is called carburetion and the device which does this job is called a carburetor.

Factors affecting carburetion

a) The engine speed

- Since modern engines are high speed type, the time available for mixture formation is very limited. For example, an engine sunning at 3000 rpm has only 10 milliseconds for mixture induction during intake stroke. When the speed becomes 6000 rpm the time available is only 5 ms.

b) The vapourization characteristics of the fuel

- The factors which ensure high quality carburetion within short period are the presence of highly volatile hydrocarbons in the fuel. Therefore, suitable evaporation characteristics of the fuel, indicated by its distillation curve, are necessary for efficient carburetion especially at high speed.

c) The temperature of the incoming air

– The temperature and pressure of the surrounding air has large influence on efficient carburetion. Higher atmospheric air temperature increases the vapourization of fuel and produce a more homogeneous mixture. An increase in atmospheric temperature, however leads to a decrease in power output of the engine when the air-fuel ratio is constant due to reduced mass flow into the cylinder or, in other words, reduced volumetric efficiency.

d) The design of carburetor

– The design of the carburetor, the intake system and the combustion chamber have considerable influence on uniform distribution of mixture to the various cylinders of the engine. Proper design of carburetor elements alone ensures the supply of desired composition of the mixture under different condition of the engine.

Automotive engine mixture requirement at different loads and speeds

– Actual air-fuel mixture requirements in an automotive engine is not constant but varies with load and speed.

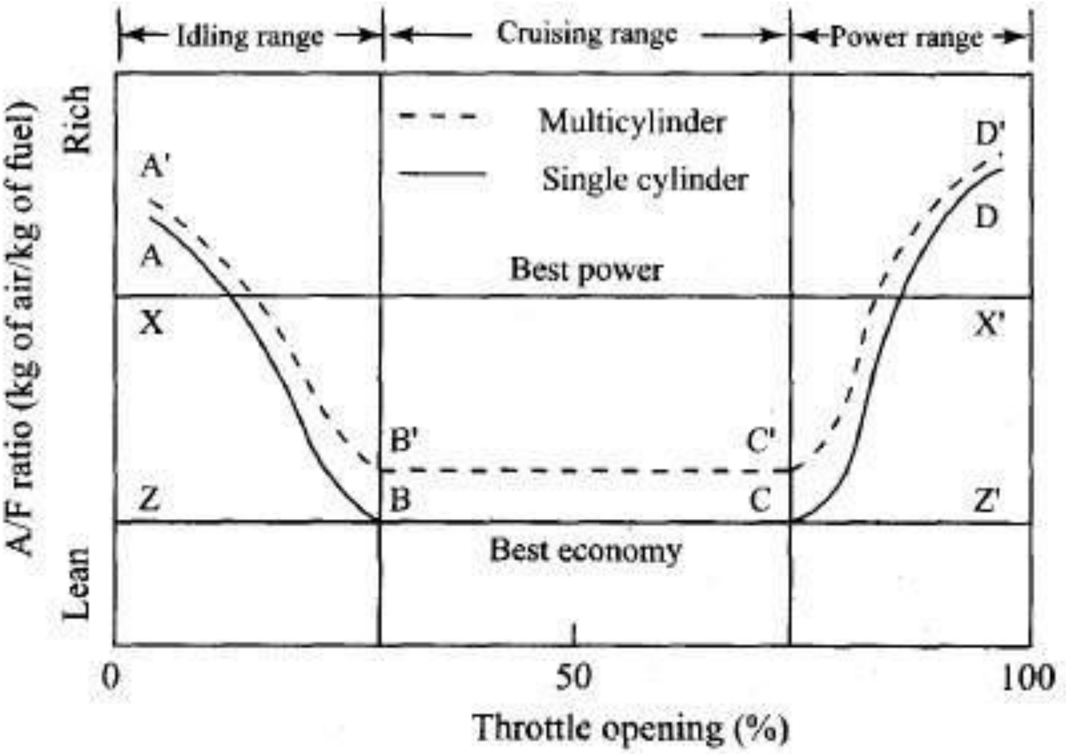


Fig. 4.1 Carburetor Performance to fulfill engine requirements

– For successful operation of the engine, the carburetor has to provide mixtures which follow the general shape of the curve ABCD (single cylinder) and A'B'C'D' (multi cylinder) in Fig. 4.1 which represents a typical automotive engine requirement. The carburetor must be suitably designed to meet the various engine requirements.

- As indicated in Fig. 4.1 there are three general ranges of throttle operation. In each of these, the automotive engine requirements differ. As a result, the carburetor must be able to supply the required air-fuel ratio to satisfy these demands. These ranges are:
  (i) Idling (mixture must be enriched)
  (ii) Cruising (mixture must be leaned)
  (iii) High Power (mixture must be enriched)

i) Idling Range

- An idling engine is one which operates at no load and with nearly closed throttle. Under idling conditions, the engine requires a rich mixture, as indicated by point A in Fig. 4.1.
- This is due to the existing pressure conditions within, the combustion chamber and the intake manifold which cause exhaust gas dilution of the fresh charge. The pressures indicated in Fig. 4.2 are representative values which exist during idling. The exhaust gas pressure at the end of the exhaust stroke does not vary greatly from the value indicated in Fig. 4.2, regardless of the throttle position.

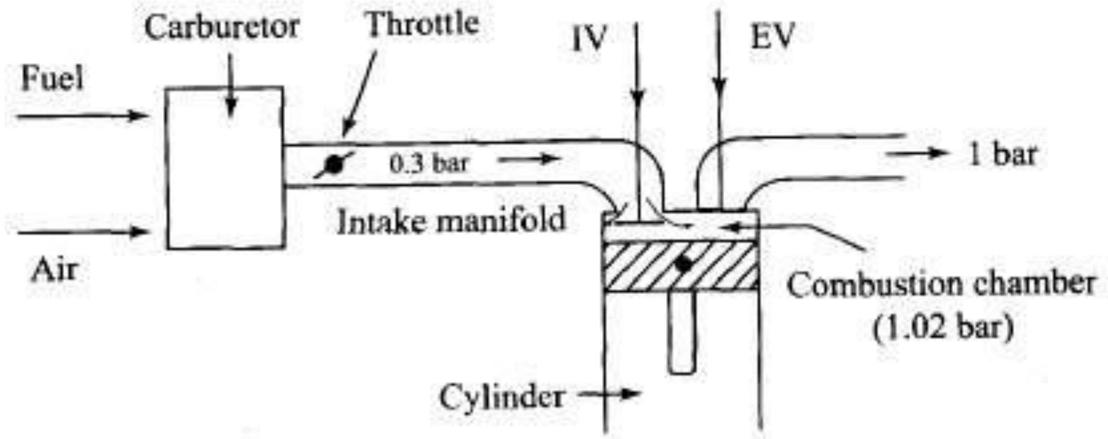


Fig. 4.2 Schematic diagram of combustion chamber and induction system at the start of intake stroke

- Since, the clearance volume is constant, the mass of exhaust gas in the cylinder at the end of the exhaust stroke tends to remain fairly constant throughout the idling range. The amount of fresh charge brought in during idling, however, is much less than that during full throttle operation, due to very small opening of the throttle (Fig. 4.2).
- This results in a much larger proportion of exhaust gas being mixed with the fresh charge under idling conditions. Further, with nearly closed throttle the pressure in the intake manifold is considerably below atmospheric due to restriction to the air flow.
- When the intake valve opens, the pressure differential between the combustion chamber and the intake manifold results in initial *backward* flow of exhaust gases into the intake manifold. As the piston proceeds down on the intake stroke, these exhaust gases are drawn back into the cylinder, along with the fresh charge.
- As a result, the final mixture of fuel and air in the combustion chamber is diluted more by exhaust gas. The presence of this exhaust gas tends to obstruct the contact of fuel

and air particles - a requirement necessary for combustion. This results in poor combustion and, as a result, in loss of power.

- It is, therefore, necessary to provide more fuel particles by richening the air-fuel mixture. This richening increases the probability of contact between fuel and air particles and thus improves combustion.
- As the throttle is gradually opened from A to B, (Fig. 4.1), the pressure differential between the inlet manifold and the cylinder becomes smaller and the exhaust gas dilution of the fresh charge diminishes. Mixture requirements then proceed along line AB (Fig. 4.1) to a leaner *A/F* ratio required for the cruising operation.

ii) Cruising Range

- In the cruising range from B to C (Fig. 4.1), the exhaust gas dilution problem is relatively insignificant.
- The primary interest lies in obtaining the maximum fuel economy. Consequently, in this range, it is desirable that the carburetor provides the engine with the best economy mixture.

iii) Power Range (From about 75% to 100% rated power)

- The mixture requirement for maximum power is a rich mixture, of *A/F* about 14 : 1 or ($F/A \approx 0.07$).
- Besides providing maximum power, a rich mixture also prevents overheating of exhaust valve at high load and inhibits detonation. At high load there is greater heat transfer to engine parts.
- Enriching the mixture reduces the flame temperature and the cylinder temperature, thereby reducing the cooling problem and lessening the chances of damaging the exhaust valves. Also, reduced temperature tends to reduce detonation. Aircraft engines have elaborate arrangement far enrichment of mixture, as detonation can wreck the engine in a matter of seconds.

Simple carburetor

- A simple carburetor is shown in the figure 4.3.
- It mainly consist of a float chamber, fuel discharge nozzle and a metering orifice, a venturi, a throttle valve and a choke.
- The float and a needle valve system maintains a constant level of gasoline in the float chamber.
- If the amount of fuel in the float chamber falls below the designed level, the float goes down, thereby opening the fuel supply valve and admitting fuel. When the designed level has been reached, the float closes the fuel supply valve thus stopping additional fuel flow from the supply system.
- Float chamber is vented either to the atmosphere or to the upstream side of the venturi.

- During suction stroke air is drawn through the venturi. Venturi is a tube of decreasing cross-section with a minimum are at the throat. Venturi tube is also known as choke tube and is so shaped that it offers minimum resistance to the air flow.

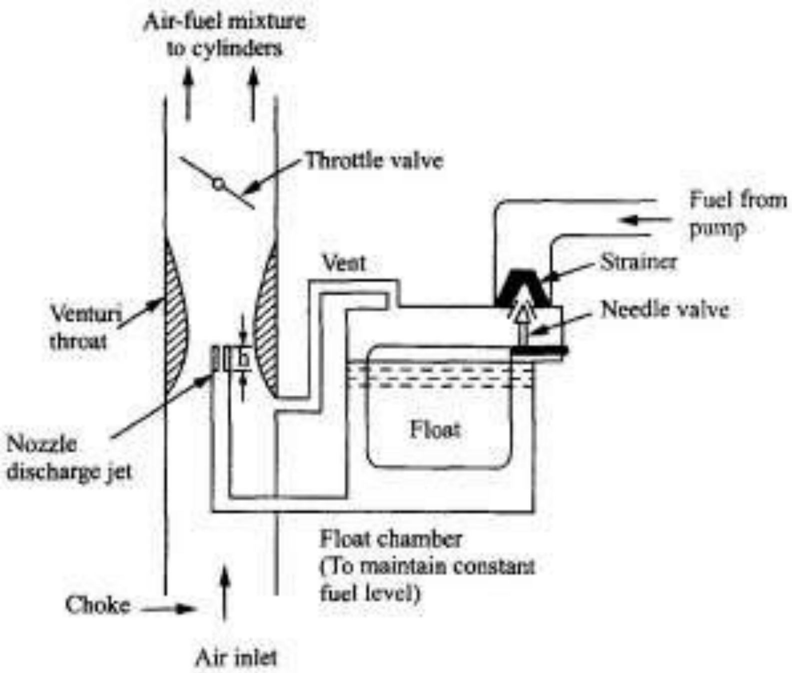


Fig. 4.3 A simple or elementary carburetor

- As air is passes through the venturi the velocity increases reaching a maximum at the venturi throat. Correspondingly, the pressure decrease reaching a minimum.
- From the float chamber, the fuel is fed to a discharge jet, the tip of which is located in the throat of venturi.
- Because of the differential pressure between the float chamber and the throat of the venturi, known as carburetor depression, fuel is discharged into the air stream. The fuel discharge is affected by the size of the discharge jet and it is selected to give the required air-fuel ratio.
- The pressure at the throat at the fully open throttle condition lies between 4 to 5 cm of Hg, below atmospheric and seldom exceeds 8 cm Hg below atmospheric. To avoid overflow of fuel through the jet, the level of the liquid in the float chamber is maintained at a level slightly below the tip of the discharge jet. This is called the ***lip of the nozzle***. The difference in the height between the top of the nozzle and the float chamber level is marked *h* in Fig. 4.3.
- The gasoline engine is ***quantity governed***, which means that when power output is to be varied at a particular speed, the amount of charge delivered to the cylinder is varied. This is achieved by means of a throttle valve usually of the butterfly type which is situated after the venturi tube.
- As the throttle is closed less air flows through the venturi tube and less is the quantity of air-fuel mixture delivered to the cylinder and hence power output is reduced. As the throttle is opened, more air flows through the choke tube resulting in increased

quantity of mixture being delivered to the engine. This increases the engine power output.

Drawbacks:

- It provides the required *A/F* ratio only at one throttle position. At the other throttle positions the mixture is either leaner or richer depending on whether the throttle is opened less or more.
- It provides increasing richness of A/F mixture as the speed of the engine increases. Reason behind that is as the throttle valve is opened gradually, the pressure at the venturi throat decreases, which decreases density of air with increase in its air velocity. Whereas, the quantity of fuel flow remains constant. Therefore, A/F ratio decreases with increase in speed of engine.
- If the speed is too low, we get very lean mixtures which may not be sufficient to ignite the mixture.
- So the simple carburettor is only suitable for small stationary engines to run at constant speed.

Complete Carburetor (Modification in Simple carburetor)

- For meeting the demand of the engine under all conditions of operation, the following addi-tional devices/systems are added to the simple carburetor:
  1. Main metering system
  2. Idling system
  3. Power enrichment or economiser system
  4. Acceleration pump system
  5. Choke.

Main metering system

- The main metering system of a carburetor is designed to supply a nearly constant basis fuel-air ratio over a wide range of speeds and loads. This mixture corresponds approximately to best economy at full throttle (A/F ratio ≈ 15.6 or F/A ratio 0.064).
- Since a simple or elementary carburetor tends to enrich the mixture at higher speeds automatic compensating device are incorporated in the main metering system to correct this tendency. These devices are:

a) Compensating Jet device

- The principle of compensating jet device is to make the mixture leaner as the throttle opens progressively. In this method, as can be seen from Fig. 4.4 in addition to the main jet, a compensating jet is incorporated. The compensating jet is connected to the compensation well. The compensating well is also vented to atmosphere like the main float chamber.
- The compensating well is supplied with fuel from the main float chamber through a restricting orifice. With the increase in air flow rate, there is decrease of fuel level in the compensating well, with the result that fuel supply through the compensating jet decreases.

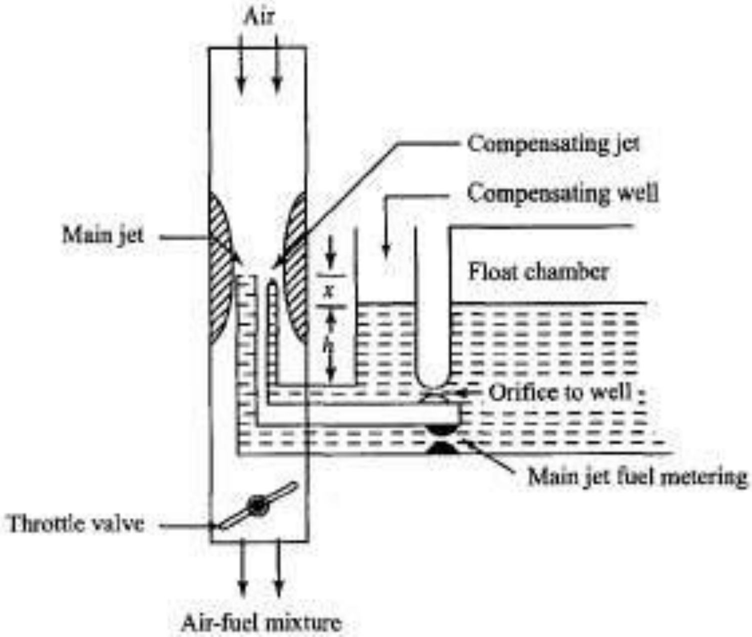


Fig 4.4 Compensating jet device

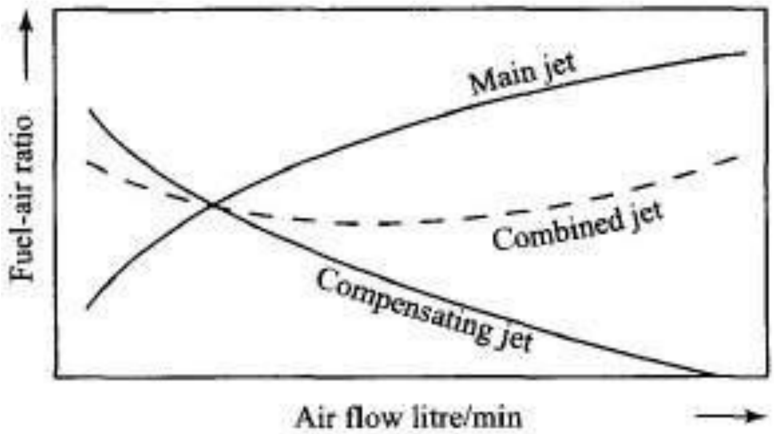


Fig 4.5 Effect of compensating device on fuel-air ratio

- The compensating well is supplied with fuel from the main float chamber through a restricting orifice. With the increase in air flow rate, there is decrease of fuel level in the compensating well, with the result that fuel supply through the compensating jet decreases.
- The compensating jet thus progressively makes the mixture leaner as the main jet progressively makes the mixture richer. The sum of the two tends to keep the fuel-air mixture more or less constant as shown in Fig. 4.5. The main jet curve and the compensating jet curve are more or less reciprocals of each other.

b) Emulsion tube or air bleeding device

- In the modem carburetors the mixture correction is done by air bleeding alone. In this arrangement the main metering jet is fitted about 25 mm below the petrol level and it is called a submerged jet (see Fig. 4.6).
- The jet is situated at the bottom of a well, the sides of which have holes which are in communication with the atmosphere.

- Air is drawn through the holes in the well, the petrol is emulsified, and the pressure difference across the petrol column is not as great as that in the simple or elementary carburetor.
- Initially, the petrol in the well is at a level equal to that in the float chamber. On opening the throttle this petrol, being subject to the low throat pressure, is drawn into the air. This continues with decreasing mixture richness as the holes in the central tube are progressively uncovered. Normal flow then takes place from the main jet.

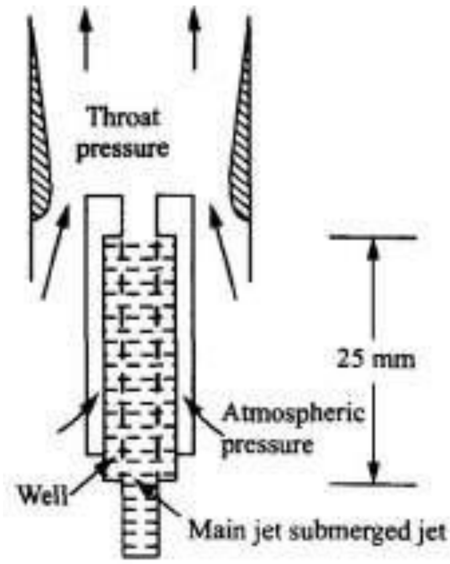


Fig. 4.6 Emulsion Tube

c) Auxiliary valve carburetor

- Fig. 4.7 (a) shows a simplified picture of an auxiliary valve device for under-standing the principle. When the engine is not operating the pressure, p_1 acting on the top of the auxiliary valve is atmospheric. The vacuum at the venturi throat increases (the throat pressure, p_2 decreases) with increase in load. This pressure differential ($p_1 - p_2$) lifts the valve against the tension of the spring. And as a result, more air is admitted and the mixture is prevented from becoming rich.

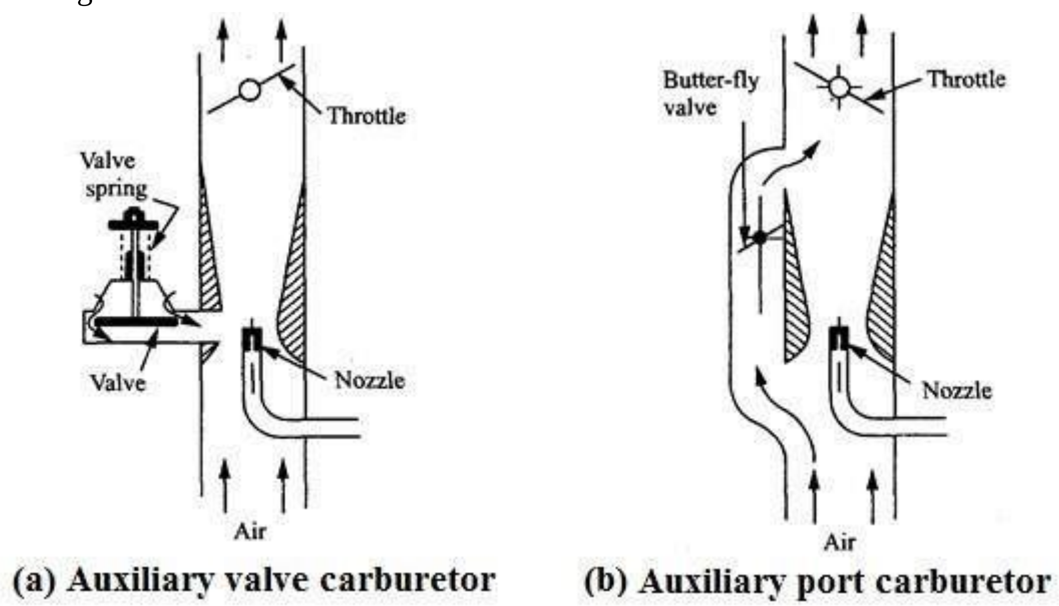


Fig. 4. Auxiliary Valve and Port carburetor

d) Auxiliary port carburetor

- An auxiliary port carburetor is illustrated in fig. 4.6(b).
- By opening the butterfly valve, additional air is admitted and at the same time the depression at the venturi throat also reduced, decreasing the quantity of fuel drawn in. This method is used in aircraft carburetors for altitude compensation.

Idling system

- During starting or idling, engine runs without load and throttle valve remains in closed position. Engine produces power only to overcome friction between the parts, and a rich mixture is to be fed to the engine to sustain combustion.
- The idling system as shown consist of an idling fuel passage and an idling port. When the throttle is partially closed, a depression past the throttle allows the fuel to go into the intake through the idle tube.
- The depression also draws air through the idle air bleed and mixes with fuel. As the throttle is opened, the main jet gradually takes over while the idle jet becomes ineffective.

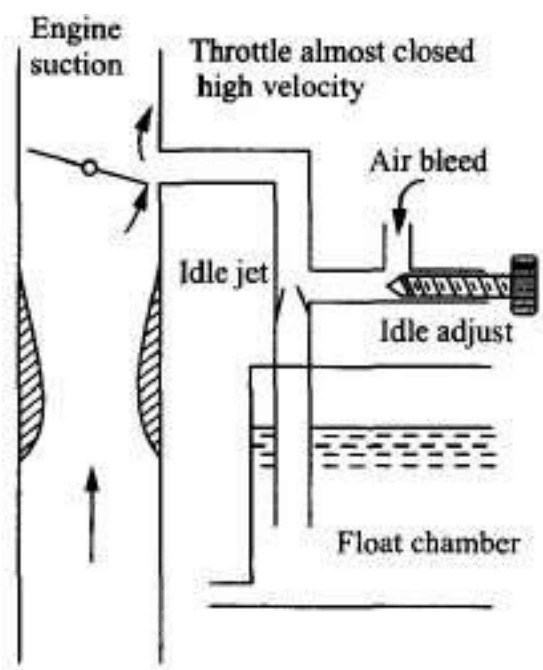
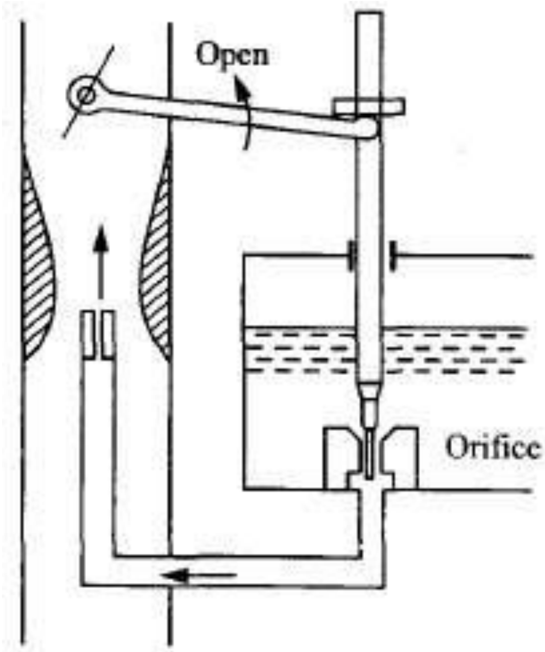


Fig. 4.8 Idling Jet

- The fuel flow depends on the location of the idle nozzle and the adjustment of the idle screw.

Power enrichment or Economiser system

- As the maximum power range of operation (75% to 100% load) is approached, some device must allow richer mixture (A/F about 13:1, F/A 0.08) to be supplied. Such a device is the meter rod economiser shown in Fig. 4.8. The name economiser is rather misleading.
- The meter rod economizer shown in Fig. 4.9, simply provides a large orifice opening to the main jet as the throttle is opened beyond a certain point. The rod may be tapered or stepped.



Acceleration Pump System

- When it is desired to accelerate the engine rapidly, a simple carburetor will not provide the required rich mixture. Rapid opening of the throttle will be immediately followed by an increased airflow, but the inertia of the liquid fuel will cause at least a momentarily lean mixture just when richness is desired for power.

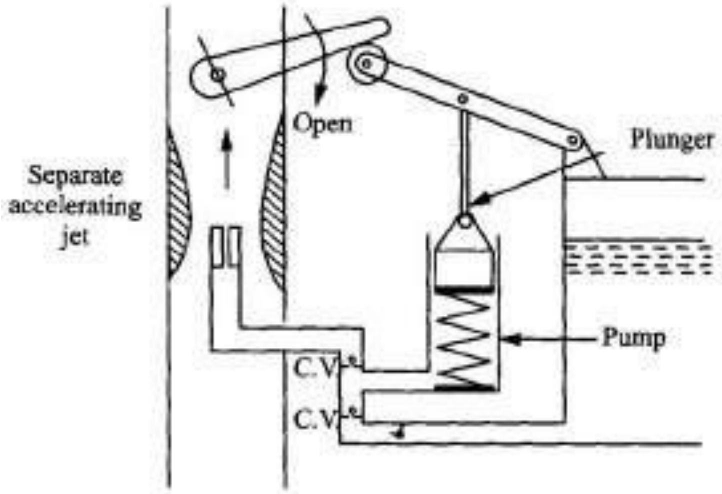
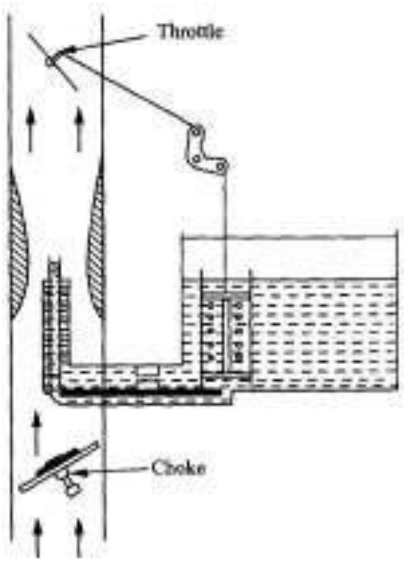


Fig. 4.10 Acceleration Pump

- To overcome this deficiency an acceleration pump is provided, one example of which is shown in Fig. 4.10. The pump consists of a spring loaded plunger. A linkage mechanism is provided so that when the throttle a rapidly opened the plunger moves into the cylinder and forces an additional jet of fuel into the venturi.
- The plunger is raised again against the spring force when the throttle is partly closed. Arrangement is provided so that when the throttle is opened slowly, the fuel in the pump cylinder is not forced into the venturi but leaks past plunger or some holes into the float chamber.

Choke

- During cold starting period, at low cranking speed and before the engine has warmed up, a mixture much richer than usual mixtures (almost 5 to 10 times more fuel) must be supplied simply because a , large fraction of the fuel will remain liquid even in the cylinder, and only the vapour fraction forms a combustible mixture with the air.
- The most common means of obtaining this rich mixture is by the use of a choke, which is a butterfly type of valve placed between the entrance to the carburetor and the Venturi throat as shown in Fig. 4.11.
- By partially closing the choke, a large pressure drop can be produced at the venturi throat that



loaded bypass

would normally result from the amount of air flowing through the venturi. This strong suction at the throat will draw large quantities of fuel from the main nozzle and supply a sufficiently rich mixture so that the ratio of evaporated fuel to air in the cylinders is within combustible limits.

- Choke valves are sometimes made with a spring loaded by-pass so that high pressure drops and excessive choking will not result after the engine has started and has attained a higher speed. Some manufacturers make the choke operate automatically by means of a thermostat such that when the engine is cold the choke is closed by a bimetallic element. After starting and as the engine warms up the bimetallic element gradually opens the choke to its fully open position.

Calculation of the Air-fuel ratio supplied by simple carburetor

- A simple carburetor with the tip of the fuel nozzle z metres above the fuel level in the float chamber is shown in fig. 4.12. The expression for air-fuel ratio for the carburetor can be found **(a) accurately**, by taking compressibility of air into account, or **(b) approximately**, by neglecting the change in density of air from inlet to the throat of the carburetor, i.e. neglecting the compressibility of the air.

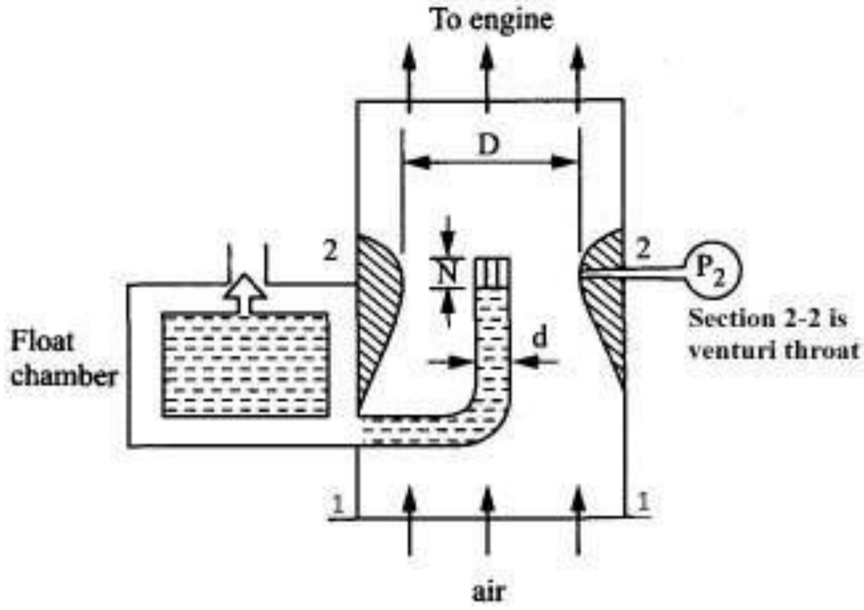


Fig. 4.12 Principles of a simple carburetor

Accurate analysis

- Applying the steady flow energy equation to section AA and BB and assuming unit mass flow of air, we have

$$q - w = (h_2 - h_1) + \frac{1}{2}(C_2^2 - C_1^2) \qquad (4.5)$$

- Here q, w are the heat and work transfers from entrance to throat and h and C stand for enthalpy and velocity respectively.
- Assuming an adiabatic flow, we get $q = 0$, $w = 0$ and $C_1 \approx 0$,

$$C_2 = \sqrt{2(h_1 - h_2)} \tag{4.6}$$

- Assuming air to behave like ideal gas, we get $h = C_P T$. Hence, eq. 4.6 can be written as,

$$C_2 = \sqrt{2C_P(T_1 - T_2)} \tag{4.7}$$

- As the flow process from inlet to the venturi throat can be considered to be isentropic, we have

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \tag{4.8}$$

$$T_1 - T_2 = T_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right] \tag{4.9}$$

- Substituting eq. 4.9 in eq. 4.7, we get

$$C_2 = \sqrt{2C_P T_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]} \tag{4.10}$$

- Now, mass for air,

$$m_a = \rho_1 A_1 C_1 = \rho_2 A_2 C_2 \tag{4.11}$$

where A_1 and A_2 are the cross-sectional area at section 1-1 and section 2-2.
- To calculate the mass flow rate of air at venturi throat, we have

$$\frac{p_1}{\rho_1^{\gamma}} = \frac{p_2}{\rho_2^{\gamma}} \tag{4.12}$$

$$\rho_2 = \rho_1\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}$$

$$m_a = \left(\frac{p_2}{p_1}\right)^{1/\gamma} \rho_1 A_2 \sqrt{2C_P T_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]} \tag{4.13}$$

$$= \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \frac{p_1}{RT_1} A_2 \sqrt{2C_P T_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]}$$

$$m_a = \frac{A_2 p_1}{R\sqrt{T_1}} \sqrt{2C_P\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma+1}{\gamma}\right)}\right]} \tag{4.14}$$

- The above equation gives theoretical mass flow of air. The actual mass flow is obtained by multiplying by the coefficient of discharge of the venturi.

$$m_{a_{actual}} = C_{da} \frac{A_2 p_1}{R\sqrt{T_1}} \sqrt{2C_P \left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1} \right)^{\left(\frac{\gamma+1}{\gamma} \right)} \right]} \qquad (4.15)$$

- In order to calculate the air-fuel ratio, fuel flow rate is to be calculated. As the fuel is incompressible, applying Bernoulli's Theorem we get

$$\frac{p_1}{\rho_f} - \frac{p_2}{\rho_f} = \frac{C_f^2}{2} + gz \qquad (4.16)$$

where ρ_f is the density of fuel, C_f is the fuel velocity at the nozzle exit and z is the height of the nozzle exit above the level of fuel in the float chamber

$$C_f = \sqrt{2 \left[\frac{p_1 - p_2}{\rho_f} - gz \right]} \qquad (4.17)$$

- Mass flow rate of fuel,

$$m_f = A_f C_f \rho_f \qquad (4.18)$$

$$= A_f \sqrt{2\rho_f \left(p_1 - p_2 - gz\rho_f \right)} \qquad (4.19)$$

where A_f is the area of cross-section of the nozzle and ρ_f is the density of the fuel

- Above equation gives theoretical mass flow of the fuel to find actual mass flow coefficient of discharge of fuel nozzle must be taken into consideration.

$$m_{f_{actual}} = C_{df} A_f \sqrt{2\rho_f \left(p_1 - p_2 - gz\rho_f \right)} \qquad (4.20)$$

$$A/F \text{ ratio} = \frac{m_{a_{actual}}}{m_{f_{actual}}}$$

$$\frac{A}{F} = \frac{C_{da}}{C_{df}} \times \frac{A_2}{A_f} \times \frac{p_1}{R\sqrt{T_1}} \times \frac{\sqrt{2C_P \left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1} \right)^{\left(\frac{\gamma+1}{\gamma} \right)} \right]}}{\sqrt{2\rho_f \left(p_1 - p_2 - gz\rho_f \right)}} \qquad (4.21)$$

Approximate analysis

- When air is considered as incompressible, Bernoulli's Theorem is applicable to air flow also. Hence, assuming $C_1 \approx 0$

$$\frac{p_1}{\rho_f} - \frac{p_2}{\rho_f} = \frac{C_2^2}{2} \qquad (4.22)$$

$$C_2 = \sqrt{2 \left[\frac{p_1 - p_2}{\rho_f} \right]} \qquad (4.23)$$

$$m_a = A_2 C_2 \rho_a = A_2 \sqrt{2\rho_a \left(p_1 - p_2 \right)} \qquad (4.24)$$

$$m_{a_{actual}} = C_{da} A_2 \sqrt{2\rho_a \left(p_1 - p_2 \right)} \qquad (4.25)$$

$$A/F \text{ ratio} = \frac{m_{a_{actual}}}{m_{f_{actual}}}$$

$$\frac{A}{F} = \frac{C_{da}}{C_{df}} \times \frac{A_2}{A_f} \times \sqrt{\frac{\rho_a \left(p_1 - p_2 \right)}{\rho_f \left(p_1 - p_2 - gz\rho_f \right)}} \qquad (4.26)$$

- If nozzle lip can be neglected then, $z = 0$

$$\frac{A}{F} = \frac{C_{da}}{C_{df}} \times \frac{A_2}{A_f} \times \sqrt{\frac{\rho_a}{\rho_f}} \qquad (4.27)$$

Air-Fuel ratio provided by a simple carburetor

- It is clear from expression for m_f (Eq. 4.20) that if $(p_1 - p_2)$ is less than $gz\rho_f$ there is no fuel flow and this can happen at very low air flow. As the air flow increases, $(p_1 - p_2)$ increases and when $(p_1 - p_2) > gz\rho_f$ the fuel flow begins and increases with increase in the differential pressure.
- At high air flows where $(p_1 - p_2)$ is large compared to $gz\rho_f$ the fraction $gz\rho_f / (p_1 - p_2)$ becomes negligible and the air-fuel ratio approaches

$$\frac{C_{da}}{C_{df}} \times \frac{A_2}{A_f} \times \sqrt{\frac{\rho_a}{\rho_f}}$$

- A decrease in the density of air reduces the value of air-fuel ratio (i.e., mixture becomes richer). It happens at
  (a) High air flow rates where $(p_1 - p_2)$ becomes large and ρ_2 decreases.
  (b) High altitudes where the density of air is low.

Types of carburetors

- Depending upon the direction of air and fuel flow, the carburettors are classified as:
  (i) Updraft carburettors (ii) Downdraft carburettors (iii) Side draft or horizontal carburettors.
- Fig. 4.13(a) shows the updraft carburettor in which the air enters the carburettor against the gravity from bottom in the upward direction.
- The disadvantage of such a carburettor is that it has to lift the sprayed fuel droplets by air friction.
- Since the fuel droplets have the tendency to separate out from air stream due to high inertia, it becomes necessary to design the jet tube and throat of relatively smaller area in order to increase the air velocity to an extent it carries the fuel particles along even at low engine speeds, otherwise, the mixture reaching the engine will be lean.
- However, with relatively smaller cross-section of jet tube, the carburettor cannot supply the mixture at the required rapid rate at high engine speeds. Due to this the updraft carburettors have now become almost obsolete.

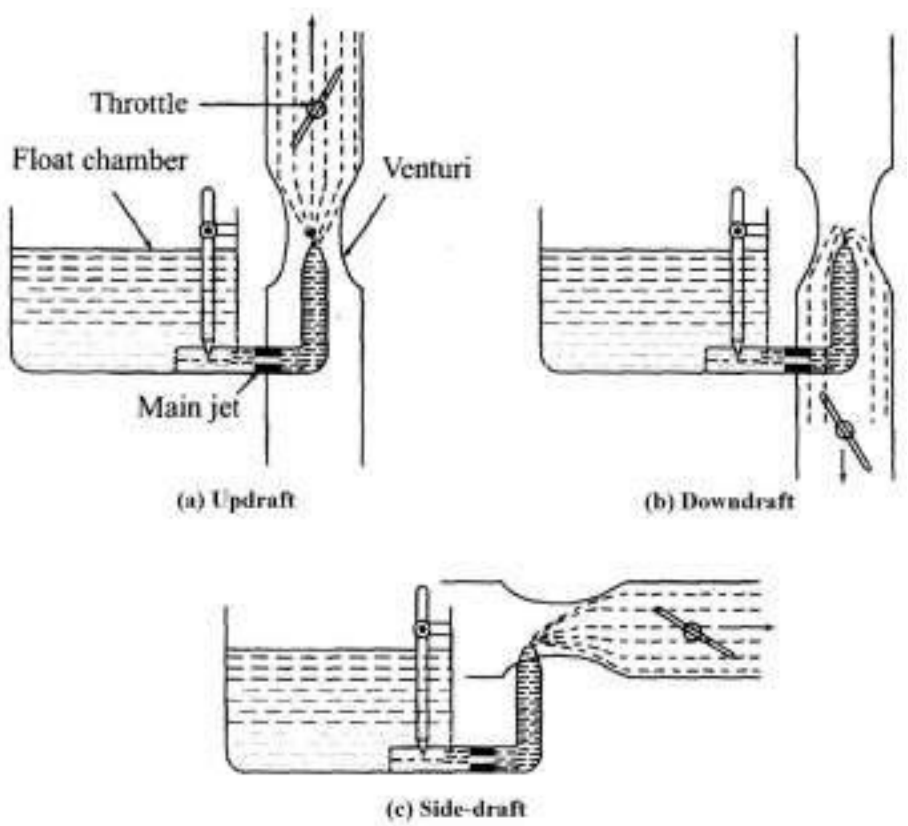


Fig. 4.13 Types of Carburetors

- Fig. 4.13(b) shows the down-draft carburettor. These are usually installed at a level higher than the intake manifolds. In these carburettors the flow of mixture is assisted by the gravity in its passage into intake manifolds. This allows the proper flow of mixture even at low engine speeds and at the same time the carburettor is reasonably accessible.
- Fig. 4.13(c) shows the side-draft carburettor. It consists of a horizontal jet tube. Such a carburettor has the advantage where under bonnet space is limited and also the resistance to flow is reduced due to elimination of one right angled turn in the intake passages.
- Most automotive carburetors are either downdraft or side-draft. In the United States, downdraft carburetors were almost ubiquitous, partly because a downdraft unit is ideal for V engines. In Europe, side-draft carburetors are much more common in performance applications. Small propeller-driven flat airplane engines have the carburetor below the engine (updraft).

Gasoline Injection

- In a carburettor engine, uniformity of mixture strength is difficult to realize in each cylinder of a multicylinder engine. Figure 4.14 shows a typical pattern of mixture distribution in an intake manifold of a multicylinder engine.

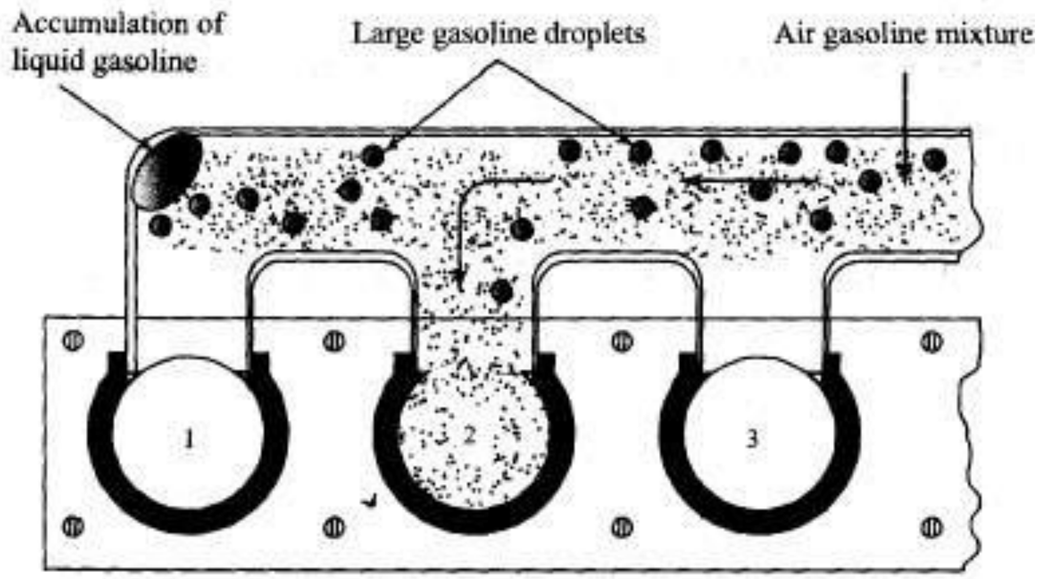


Fig. 4.14 Typical pattern of mixture distribution in a multi-cylinder engine

- As may be noticed that the intake valve is open in cylinder 2. As can also be observed the gasoline moves to the end of the manifold and accumulates there. This enriches the mixture going to the end cylinders.
- However, the central cylinders, which are very close to the carburettor, get the leanest mixture. Thus the various cylinders receive the air-gasoline mixture in varying quantities and richness. This problem is called the maldistribution and can be solved by the port injection system by having the same amount of gasoline injected at each intake manifold.
- Therefore, there is an urgent need to develop injection systems for gasoline engines. By adopting gasoline injection each cylinder can get the same richness of the air-gasoline mixture and the maldistribution can be avoided to a great extent.
- Gasoline injection system is preferred a carburetion for one or more of the following reasons:

  (i) To have uniform distribution of fuel in a multicylinder engine.

  (ii) To improve breathing capacity i.e. volumetric efficiency.

  (iii) To reduce or eliminate detonation.

  (iv) To prevent fuel loss during scavenging in case of two-stroke engines.

Types of Gasoline Injection Systems

- The fuel injection system can be classified as:

  (i) Gasoline direct injection into the cylinder (GDI)

  (ii) Port injection

  (a) Timed, and (b) Continuous

  (iii) Manifold injection
- The above fuel injection systems can be grouped under two heads, viz., single-point and multi-point injection. In the single point injection system, one or two injectors are mounted inside the throttle body assembly. Fuel sprays are directed at one point or

at the center of the intake manifold. Another name of the single point injection is throttle body injection. Multipoint injection has one injector for each engine cylinder. In this system, fuel is injected in more than one location. This is more common and is often called port injection system.

- The gasoline fuel injection system used in a spark- ignition engine can be either of continuous injection or timed injection.

Continuous injection systems:

This system usually has a rotary pump. The pump maintains a fuel line gauge pressure of about 0.75 to 1.5 bar. The system injects the fuel through a nozzle located in the manifold immediately downstream of the throttle plate. In a supercharged engine, fuel is injected at the entrance of the supercharger. The timing and duration of the fuel injection is determined by Electronic Control Unit (ECU) depending upon the load and speed.

Timed fuel injection system:

- This system has a fuel supply pump which sends fuel at a low pressure of about 2 bar when the engine is running at maximum speed. A fuel metering or injection pump and a nozzle are the other parts of the system. The nozzle injects the fuel in the manifold or the cylinder head port at about 6.5 bar or into the combustion chamber at pressures that range from 16 to 35 bar.
- Timed injection system injects fuel usually during the early part of the suction stroke. During maximum power operation injection begins after the closure of the exhaust valve and ends usually after BDC. Direct in-cylinder injection is superior and always desirable and better compared to manifold injection. In this case both low and high volatile fuels can be used and higher volumetric efficiencies can be achieved.
- Typical fuel injection methods used in four stroke gasoline engines are shown in Fig. 4.15.

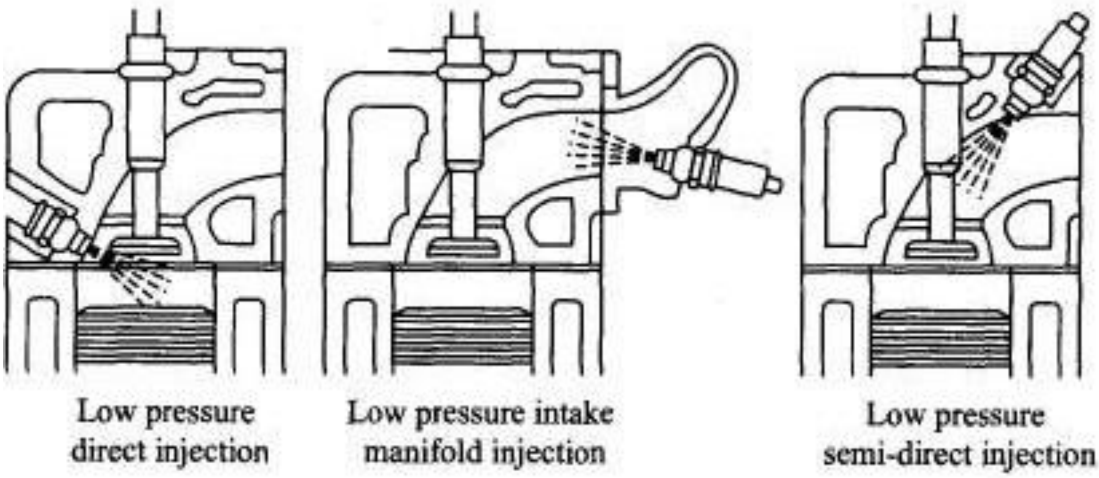


Fig. 4.15 Different methods of fuel injection

Components of Injection System

- The objectives of the fuel injection system are to meter, atomize and uniformly distribute the fuel throughout the air mass in the cylinder. At the same time it must maintain the required air-fuel ratio as per the load and speed requirement the engine.
- To achieve all the above tasks, a number of components are required in the fuel injection system, the functions of which are mentioned below.

  (i) Pumping element - moves the fuel from the fuel tank to the injector. This includes necessary piping, filter etc.

  ii) Metering element - measures and supplies the fuel at the rate demanded by load and speed conditions of the engine.

  iii) Mixing element - atomizes the fuel and mixes it with air to form a homogenous mixture.

  iv) Metering control - adjusts the rate of metering in accordance with load and speed of the engine.

  v) Mixture control - adjusts fuel-air ratio as demanded by the load and speed.

  vi) Distributing element - divides the metered fuel equally among the cylinders.

ELECTRONIC FUEL INJECTION SYSTEM

- Modern gasoline injection systems use engine sensors, a computer, and solenoid operated fuel injectors to meter and inject the right amount of fuel into the engine cylinders. These systems called **electronic fuel injection (EFI)** use electrical and electronic devices to monitor and control engine operation.
- An electronic control unit (ECU) or the computer receives electrical signals in the form of current or voltage from various sensors. It then uses the stored data to operate the injectors, ignition system and other engine related devices. As a result, less unburned fuel leaves the engine as emissions, and the vehicle rives better mileage.
- Typical sensors for an electronic fuel injection system includes the following:

  Exhaust gas or oxygen sensor - senses the amount of oxygen in the en-gine exhaust and calculates air-fuel ratio. Sensor output voltage changes in proportion to air-fuel ratio.

  Engine temperature sensor - senses the temperature of the engine coolant, and from this data the computer adjusts the mixture strength to rich side for cold starting.

  Air flow sensor - monitors mass or volume of air flowing into the intake manifold for adjusting the quantity of fuel.

  Air inlet temperature sensor - checks the temperature of the ambient air entering the engine for fine tuning the mixture strength.

  Throttle position sensor - senses the movement of the throttle plate so that the mixture flow can be adjusted for engine speed and acceleration.

  Manifold pressure sensor - monitors vacuum in the engine intake mani-fold so that the mixture strength can be adjusted with changes in engine load.

  Camshaft position sensor - senses rotation of engine camshaft/crankshaft for speed and timing of injection.

Knock sensor - microphone type sensor that detects ping or preignition noise so that the ignition timing can be retarded.

MPFI (Multi Point Fuel Injection System)

– The main purpose of the MPFI system is to supply a proper ratio of gasoline and air to the cylinders. These systems function under two basic arrangements namely

i) Port Injection
ii) Throttle body injection

Port Injection

– In the port injection arrangement, the injector is placed on the side of the intake manifold near the intake port (fig. 4.16), the injector sprays gasoline into the air, inside the intake manifold. The gasoline mixes with the air in a reasonably uniform manner. This mixture of gasoline and air then passes through the intake valve and enters into the cylinder.

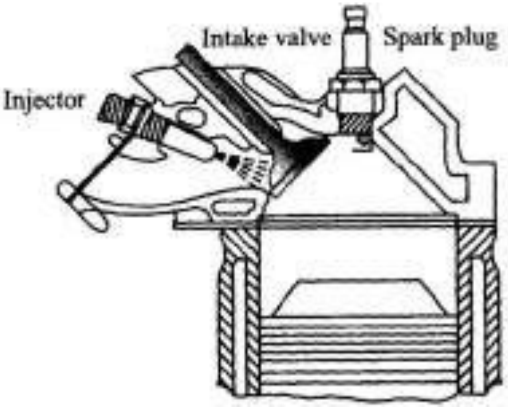


Fig. 4.16 Port Injection

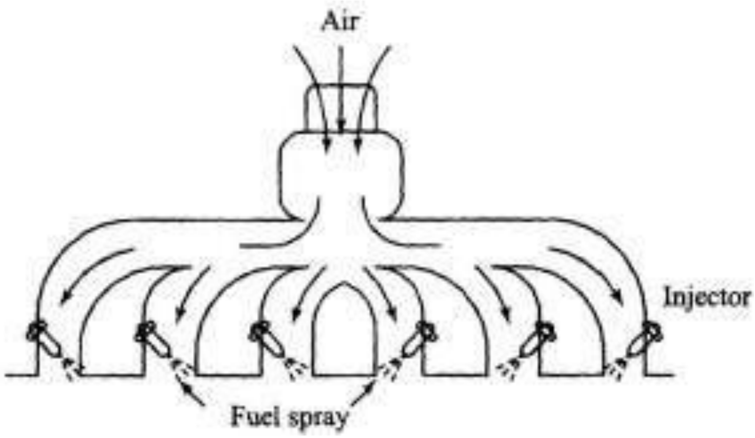


Fig. 4.17 Multi-point fuel injection (MPFI) near port

– Every cylinder is provide with an injector in its intake manifold. If there are six cylinders, there will be six injectors. Fig. 4.17 shows a simplified view of a port or multi point fuel injection (MPFI) system.

Throttle body injection System

– Fig. 4.18 illustrates the simplified sketch of throttle body injection system (Single point

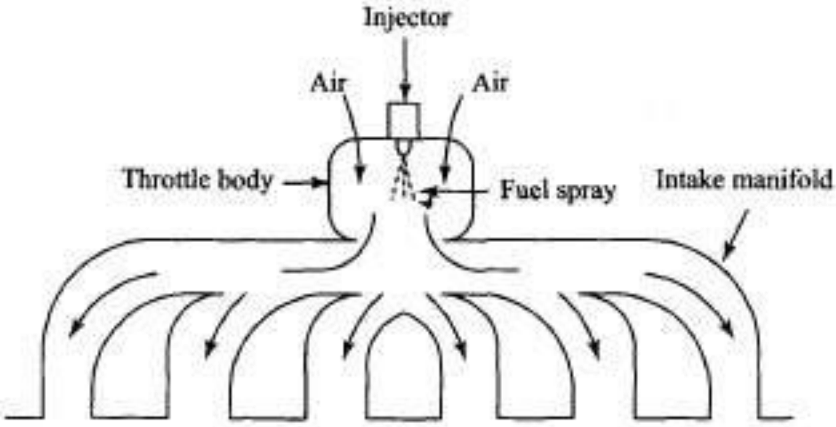


Fig. 4.18 Throttle body injection (single point)

-injection). This throttle body is similar to the carburettor throt-tle body, with the throttle valve controlling the amount of air entering the intake manifold.

- An injector is placed slightly above the throat of the throttle body. The injector sprays gasoline into the air in the intake manifold where the gasoline mixes with air. This mixture then passes through the throttle valve and enters into the intake manifold.
- As already mentioned, fuel-injection systems can be either timed or continuous. In the timed injection system, gasoline is sprayed from the injectors in pulses. In the continuous injection system, gasoline is sprayed continuously from the injectors.
- The port injection system and the throttle-body injection system may be either pulsed systems or continuous systems. In both systems, the amount of gasoline injected depends upon the engine speed and power demands. In some literature MPFI systems are classified into two types: D- MPFI and L-MPFI.

D- MPFI system

- The D-MPFI system is the manifold fuel injection system. In this type, the vacuum in the intake manifold is first sensed. In addition, it senses the volume of air by its density. Fig. 4.19 gives the block diagram regarding the functioning of the D-MPFI system.

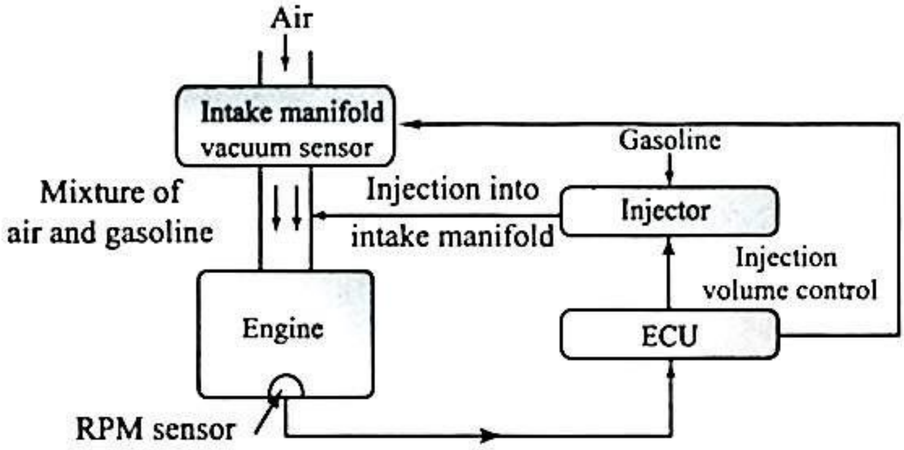


Fig 4.19 D-MPFI gasoline injection system

- As air enters into the intake manifold, the manifold pressure sensor detects the intake manifold vacuum and sends the information to the ECU. The speed sensor also sends information about the rpm of the engine to the ECU.
- The ECU in turn sends commands to the injector to regulate the amount of gasoline supply for injection. When the injector sprays fuel in the intake manifold the gasoline mixes with the air and the mixture enters the cylinder.

L-MPFI system

- The L-MPFI system is a port fuel-injection system. In this type the fuel metering is regulated by the engine speed and the amount of air that actually enters the engine. This is called air-mass metering or air-flow metering. The block diagram of an L-MPFI system is shown in Fig. 4.20.

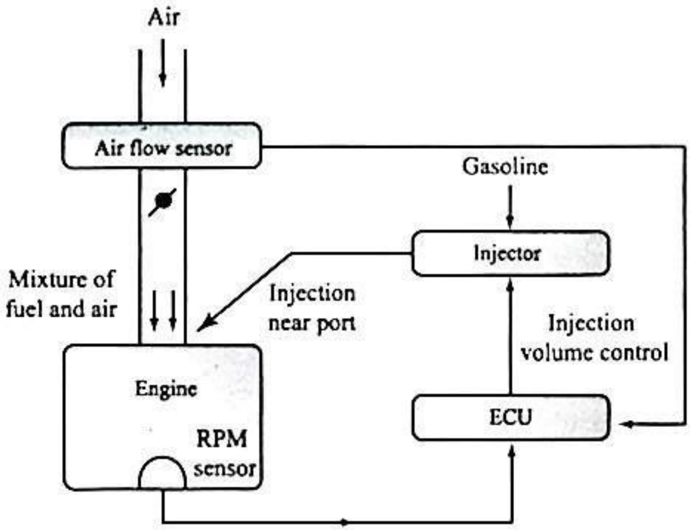


Fig. 4.20 L-MPFI gasoline injection system

- As air enters into the intake manifold, the air flow sensor measures the amount of air and sends information to the ECU. Similarly, the speed sensor sends information about the speed of the engine to the ECU.
- The ECU processes the information received and sends appropriate commands to the injector, in order to regulate the amount of gasoline supply for injection. When injection takes place, the gasoline mixes with the air and the mixture enters the cylinder.

Fuel supply system for CI Engine

- In C.I. engines, the air is taken in during the suction stroke and compressed to a high pressure (28 to 70 bar) and high temperature (520° to 720°C) according to the compres-sion ratio used (12 : 1 to 20 : 1). The high temperature of air at the end of stroke is sufficient to ignite the fuel.
- Fuel is injected into the cylinder at the end of the compression stroke; the pressure of fuel injected lies between 100 to 200 bar. During the process of injection the fuel is broken into very fine droplets. The droplets vaporise taking the heat from the hot air and form a combustible mixture and start burning. As the burning starts, the vaporisation of fuel is accelerated as more heat is available. As the combustion progresses, the amount of oxygen available for burning reduces and therefore heat release is reduced.
- The period between the start of injection and start of ignition, called the ignition delay, is about 0.001 second for high speed engines and 0.002 second for low speed engines. The injection period covers about 25° of crank rotation. After the ignition the temperature and pressure rise rapidly. The whole performance of engine is totally dependent on the delay period; the lesser the delay period better is the engine performance.

Functional Requirements of an Injection System

For a proper running and good performance from the engine, the following requirements must be met by the injection system:

- Accurate metering of the fuel injected per cycle. This is very critical due to the fact that very small quantities of fuel being handled. Metering errors may cause drastic variation from the desired output. The quantity of the fuel metered should vary to meet changing speed and load requirements of the engine
- Timing the injection of the fuel correctly in the cycle so that maximum power is obtained ensuring fuel economy and clean burning.
- Proper control of rate of infection so that the desired heat-release pattern is achieved during combustion.
- Proper atomization of fuel into very fine droplets
- Proper spray pattern to ensure rapid mixing of fuel and air
- Uniform distribution of fuel droplets in the combustion chamber
- To supply equal quantities of metered fuel to all cylinders in case of multi cylinder engines
- No lag during beginning and end of injection i.e., to eliminate dribbling of fuel droplets into the cylinder.

Classification of Injection Systems

- In a constant-pressure cycle or diesel engine, only air is compressed in the cylinder and then fuel is injected into the cylinder by means of a fuel-injection system. For producing the required pressure for atomizing the fuel either air or a mechanical means is used. Accordingly the injection systems can be classified as:

  (i) Air injection systems

  (ii) Solid injection systems

Air Injection System

- In this system, fuel is forced into the cylinder by means of compressed air.
- This system is little used nowadays, because it requires a bulky multi-stage air compressor. This causes an increase in engine weight and reduces the brake power output further.
- One advantage that is claimed for the air injection system is good mixing of fuel with the air with resultant higher mean effective pressure. Another is the ability to utilize fuels of high viscosity which are less expensive than those used by the engines with solid injection systems. These advantages are off-set by the requirement of a multistage compressor thereby making the air-injection system obsolete.

Solid Injection System

- In this system the liquid fuel is injected directly into the combustion chamber without the aid of compressed air. Hence, it is also called airless mechanical injection or solid injection system. Solid injection systems can be classified as:

  (i) Individual pump and nozzle system

(ii) Unit injector system
(iii) Common rail system
(iv) Distributor system

Typical fuel feed system for a CI engine

- Fuel feed system of CI engine comprise mainly of the following components.
  i) fuel tank
  ii) **fuel feed pump** to supply fuel from the main fuel tank to the injection system
  iii) **injection pump** to meter and pressurize the fuel for injection
  iv) **governor** to ensure that the amount of fuel injected is in accordance with variation in load
  v) **injector** to take the fuel from the pump and distribute it in the combus-tion chamber by atomizing it into fine droplets
  vi) **fuel filters** to prevent dust and abrasive particles from entering the pump and injectors thereby minimizing the wear and tear of the components.

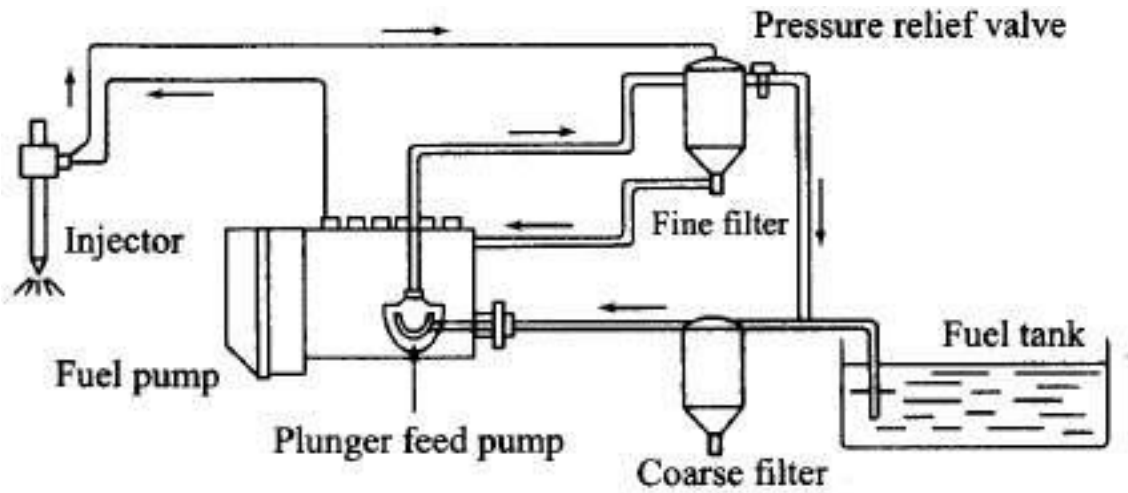


Fig. 4.21 Typical fuel feed system for a CI engine

- A typical arrangement of various components for the solid injection system used in a Cl engine is shown in Fig. 4.21.
- Fuel from the fuel tank first enters the coarse filter from which is drawn into the plunger feed pump where the pressure is raised very slightly. Then the fuel enters the fine filter where all the dust and dirt particles are removed.
- From the fine filter the fuel enters the fuel pump where it is pressurized to about 200 bar and injected into the engine cylinder by means of the injector. Any spill over in the injector is returned to the fine filter. A pressure relief valve is also provided for the safety of the system.
- The above functions are achieved with the components listed above. The types of solid injection system described in the following sections differ only in the manner of operation and control of the components mentioned above.

Types of Solid injection systems

a) Individual Pump and Nozzle System

- The details of the individual pump and nozzle system are shown in Fig. 4.22(a) and (b). In this system, each cylinder is provided with one pump and one injector.
- In this arrangement a separate metering and compression pump is provided for each cylinder. The pump may be placed close to the cylinder as shown in Fig. 4.22(a) or they may be arranged in a cluster as shown in Fig. 4.22(b).
- The high pressure pump plunger is actuated by a cam, and pro-duces the fuel pressure necessary to open the injector valve at the correct time. The amount of fuel injected depends on the effective stroke of the plunger.

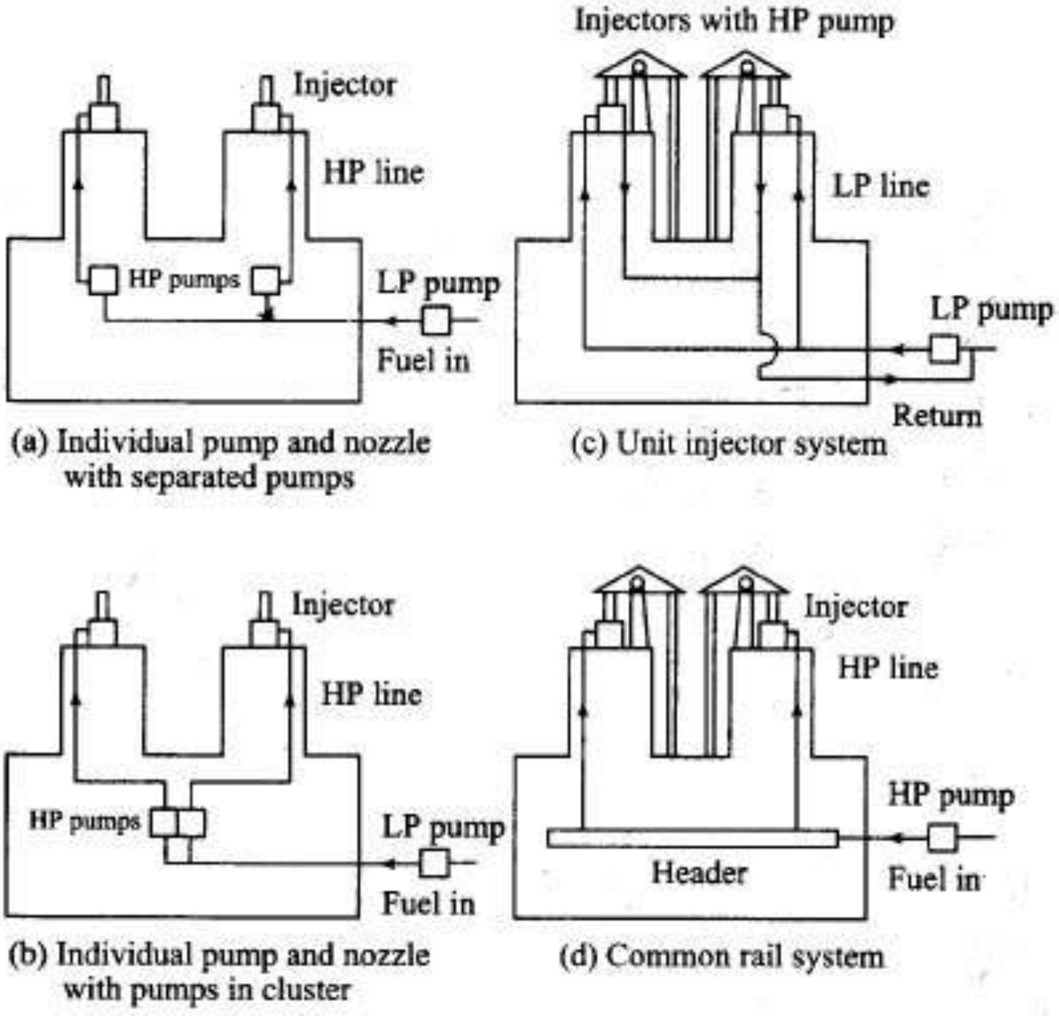


Fig. 4.22 Injection systems with pump and nozzle arrangements used in Cl engines

b) Unit Injector system

- The unit injector system, Fig. 4.22(c), is one in which the pump and the injector nozzle are combined in one housing. Each cylinder is provided with one of these unit injectors.
- Fuel is brought up to the injector by a low pressure pump, where at the proper time, a rocker arm actuates the plunger and thus injects the fuel into the cylinder. The

amount of fuel injected is regulated by the effective stroke of the plunger. The pump and the injector can be integrated in one unit as shown in Fig. 4.22(c).

c) Common Rail System

- In the common rail system, Fig. 4.22(d), a HP pump supplies fuel, under high pressure, to a fuel header. High pressure in the header forces the fuel to each of the nozzles located in the cylinders.
- At the proper time, a mechanically operated (by means of a push rod and rocker arm) valve allows the fuel to enter the proper cylinder through the nozzle. The pressure in the fuel header must be that, for which the injector system was designed, i.e., it must enable to penetrate and disperse the fuel in the combustion chamber.
- The amount of fuel entering the cylinder is regulated by varying the length of the push rod stroke. A high pressure pump is used for supplying fuel to a header, from where the fuel is metered by injectors (assigned one per cylinder). The details of the system are illustrated in Fig. 4.22(d).

d) Distributor System

- Fig. 4.23 shows a schematic diagram of a distributor system. In this system the pump which pressurizes the fuel also meters and times it. The fuel pump after metering the required amount of fuel supplies it to a rotating distributor at the correct time for supply to each cylinder.

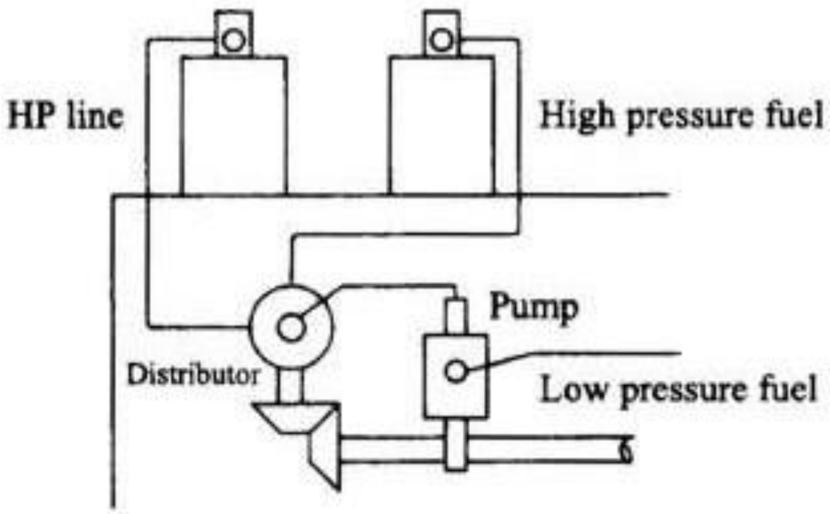


Fig. 4.23 Schematic diagram of distributor system

- The number of injection strokes per cycle for the pump is equal to the number of cylinders. The details of the system are given in Fig. 4.23. Since there is one metering element in each pump, a uniform distribution is automatically ensured. Not only that, the cost of the fuel-injection system also reduces to a value less than two-thirds of that for individual pump system. A comparison of various fuel-injection systems is given in Table 4.1.

Table 4.1 Comparison of various fuel injection system

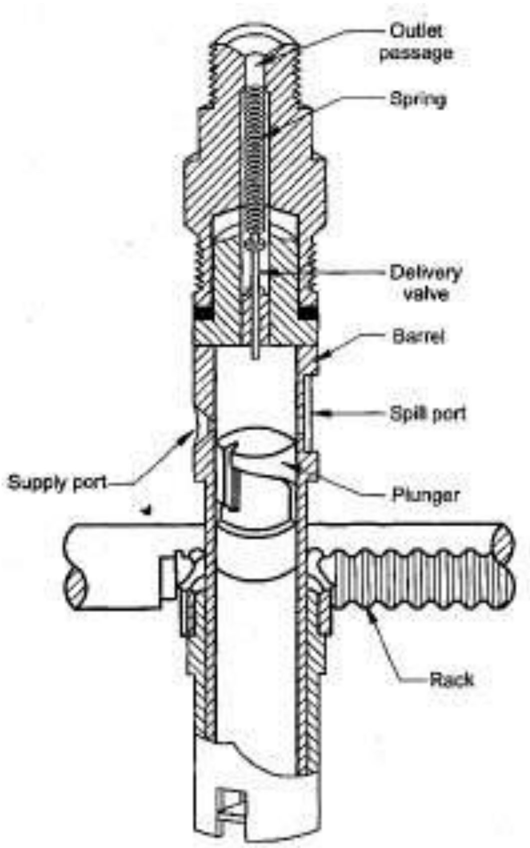
Job	Air Injection System	Solid Injection System		
		Individual Pump	Common Rail	Distributor
Metering	Pump	Pump	Injection valve	Pump
Timing	Fuel cam	Pump cam	Fuel cam	Fuel cam
Injection Rate	Spray valve	Pump cam	Spray valve	Fuel cam
Atomization	Spray valve	Spray tip	Spray tip	Spray tip
Distribution	Spray valve	Spray tip	Spray tip	Spray tip

Injection Pump

- The main objectives of fuel-injection pump is to deliver accurately metered quantity of high pressure fuel (in the range from 120 to 200 bar) at the correct time to the injector fitted on each cylinder. Injection pumps are of two types, viz. (i) Jerk type pumps (ii) Distributor type pumps

Jerk Type Pump (BOSCH fuel Injection Pump)

- Commonly used pump is shown in fig. 4.24.
- It consists of a reciprocating plunger inside a barrel. The plunger is driven by a camshaft. The working principle of jerk pump is illustrated in Fig. 4.25.
- A sketch of a typical plunger is shown.



- A schematic diagram of the plunger within the barrel is shown. Near the port A, fuel is always available under relatively low pressure. While the axial movement of the plunger is through cam shaft, its rotational movement about its axis by means of rack D. Port B is the orifice through which fuel is delivered to the injector. At this stage it is closed by means of a spring loaded check valve.
- When the plunger is below port A, the fuel gets filled in the barrel above it. As the plunger rises and closes the port A the fuel will flow out through port C. This is because it has to overcome the spring force of the check valve in order to flow through port B. Hence it takes the easier way out via port C.
- At this stage rack rotates the plunger and as a result port C also closes. The only escape route for the fuel is past the check valve through orifice B to the injector. This is the beginning of injection and also the effective stroke of the plunger.

- The injection continues till the helical indentation on the plunger un-covers port C. Now the fuel will take the easy way out through C and the check valve will close the orifice B. The fuel-injection stops and the effective stroke ends.
- Hence the effective stroke of the plunger is the axial distance traversed between the time port A is closed off and the time port A is uncovered.
- The plunger is rotated to the position shown. The same sequence of events occur. But in this case port C is uncovered sooner. Hence the effective stroke is shortened.
- It is important to remember here that though the axial distance traversed by the plunger is same for every stroke, the rotation of the plunger by the rack determining the length of the effective stroke and thus the quantity of fuel injected.

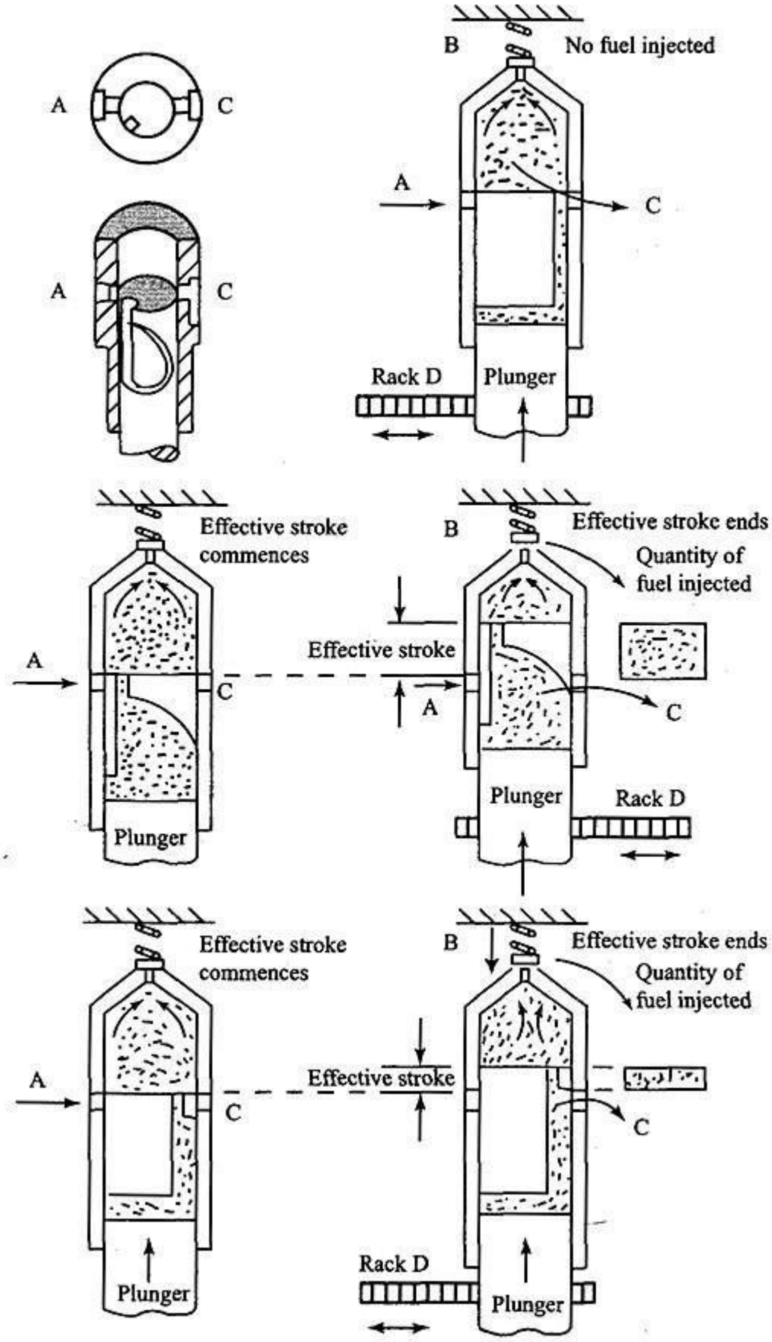


Fig. 4.25 Principle of helix bypass pump

Distributor Type Pump

- This pump has only a single pumping element and the fuel is distributed to each cylinder by means of a rotor (Fig. 4.26).
- The rotor has a central longitudinal passage two sets of radial holes (each equal to the number of engine cylinders) located at different heights. One set is connected to pump inlet via central passage whereas the second set is connected to delivery lines leading to injectors of the various cylinders.

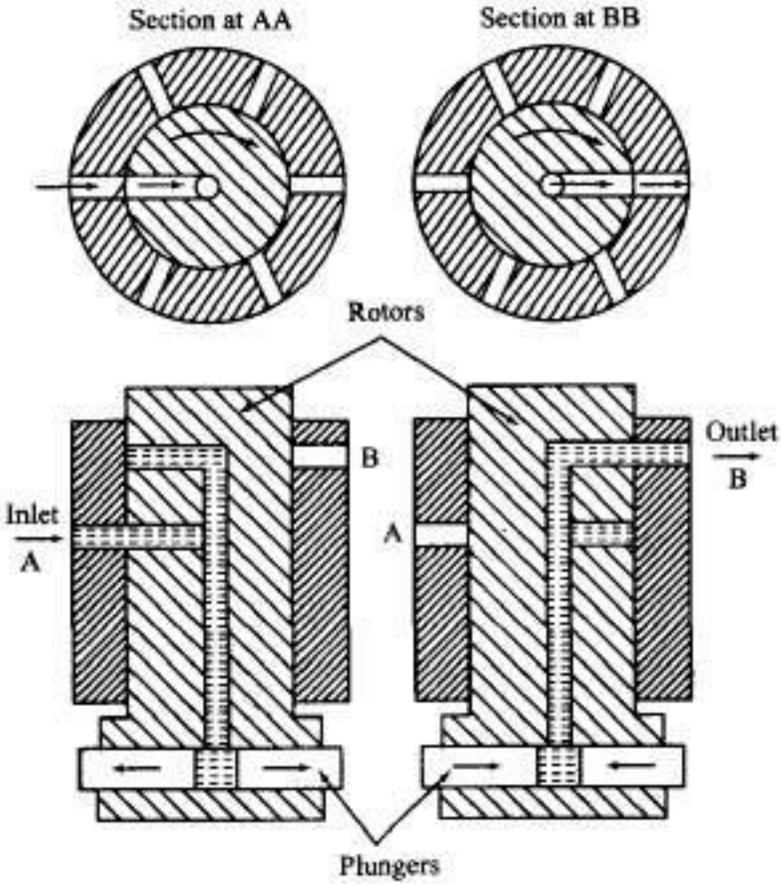


Fig. 4.26 Principle of working of distributor type fuel-injection pump

- The fuel is drawn into the central rotor passage from the inlet port when the pump plunger move away from each other. Wherever, the radial delivery passage in the rotor coincides with the delivery port for any cylinder the fuel is delivered to each cylinder in turn.
- Main advantages of this type of pump lies in its small size and its light weight.

Fuel Atomiser or Injector

- A fuel atomiser which is commonly used is shown, in Fig. 4.27. The high pressure fuel coming out of fuel pump enters into the atomiser as shown in the figure.
- The nozzle valve is lifted up due to high pressure fuel entering at the bottom of the valve and the fuel is injected into the cylinder through the nozzle.
- The pressure of the fuel falls as it is injected into the cylinder and the nozzle valve moves down under the spring force and closes the nozzle inlet to the inlet fuel passage. Thus the fuel supply to the engine is cut-off.
- Any high pressure fuel leaking past the plunger of the nozzle valve is fed back to the fuel tank through the outlet fuel passage. The adjusting screw helps to adjust the tension in the spring.

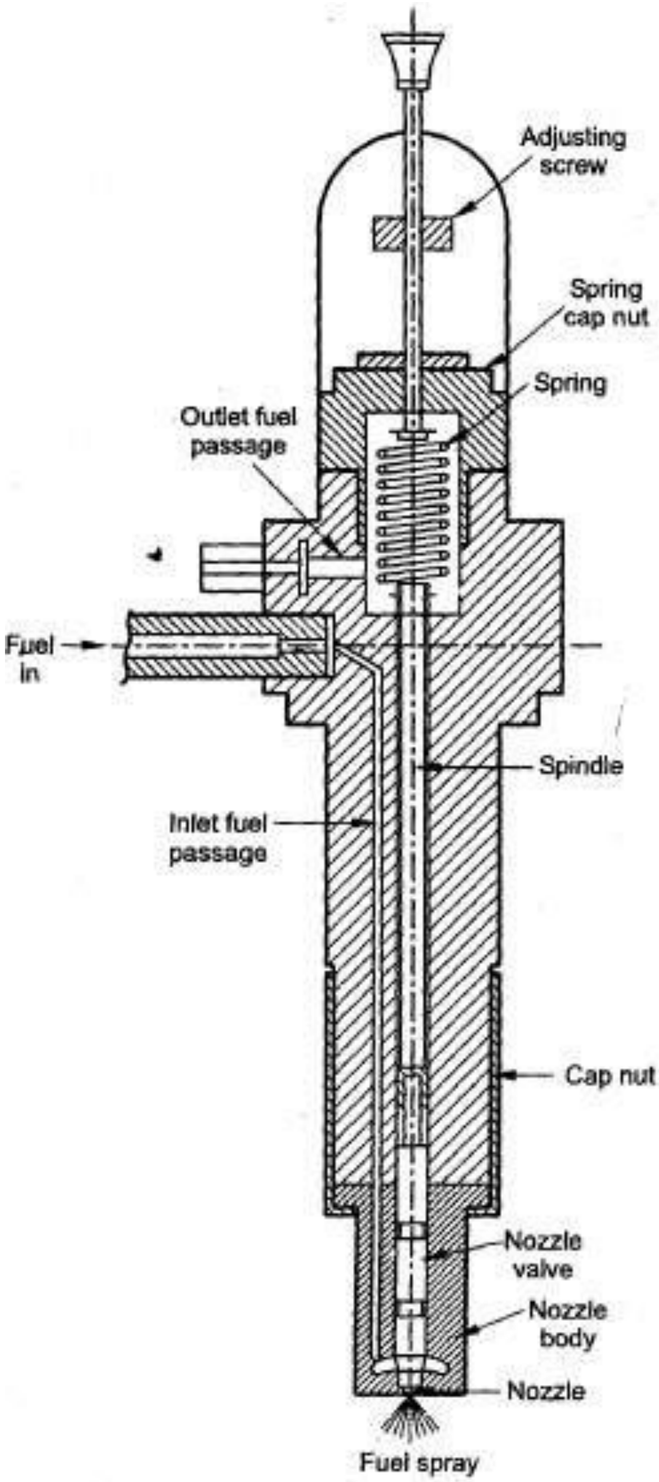


Fig. 4.27 Fuel Atomiser

Types of Nozzles

- The type of nozzle used is greatly dependent on the type of combustion chamber as open type or pre-combustion chamber. The nozzles are classified as per the type of orifice and its number used for injecting the fuel in the combustion chamber.
- The nozzles are classified as
  1. Single hole nozzle
  2. Multihole nozzle
  3. Pintal nozzle
  4. Pintaux nozzle.

Single Hole Nozzle

- This is the simplest type of nozzle and is used in open combustion chambers.
- It consists of a single hole bored centrally through the noz-zle body and closed by the needle valve. The size of the hole is usually larger than 0.2 mm.
- Its spray cone angle varies from 5 to 15°. In some cases, a cone is given a series of spiral grooves in order to impart a rotational motion to the fuel for better mixing with the air.
- The arrangement is shown in fig. 4.28.

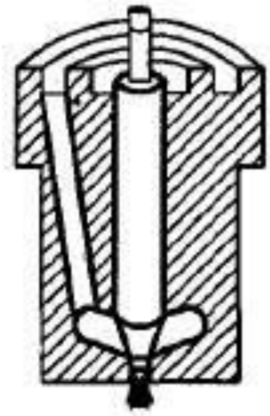


Fig. 4.28 Single hole nozzle

Advantages

- Simple in construction and operation.

Disadvantages

- Very high injection pressure is required because whole of the fuel passes through a single hole and, also, because the relative fuel velocity required is high.
- This type of nozzle has a tendency to dribble.
- As the spray angle is very narrow (usually about 15°), this does not facilitate good mixing unless higher air velocities are provided.

Multihole Nozzle

- This type of nozzle finds extensive use in automobile engines, particularly having open combustion chambers. It mixes the fuel with air properly even with slow air move-ment available with open combustion chambers.
- The number of holes varies from 4 to 18; the greater number provides better fuel distribution. The hole diameter lies between 0.25 to 0.35 mm and hole angle lies between 20° to 45°.
- Usually the holes are drilled symmetrically but many times they are non-symmetrical to meet certain specific requirements of the combustion chamber.

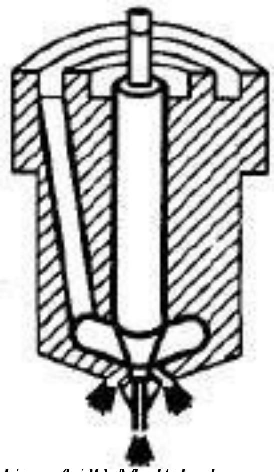


Fig. 4.29 Multihole nozzle

Advantages

- Gives good atomisation.
- Distributes fuel properly even with lower air motion available in open combustion chambers.

Disadvantages

- Holes are small and liable to clogging.
- Dribbling between injections
- Very high injection pressures (180 bar and above)
- Close tolerance in manufacture (due to small holes) and hence costly.

Pintle Nozzle

- The stem of the nozzle valve is extended to form a pin or pintle which protrudes through the mouth of the nozzle body. It may be either cylindrical or conical in shape.
- The size and shape of the pintle can be varied according to requirement. The spray core angle is gen-erally 60°.
- When the valve lifts, the pintle partially blocks the orifice and thus does not allow the pressure drop to be greater. As the lift of the valve increases the entire orifice is uncovered and full area for flow is available. Thus dribbling is avoided.

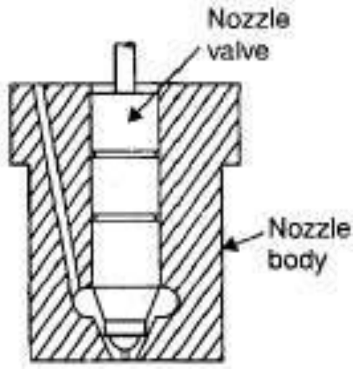


Fig. 4.30 Pintle nozzle

- The spray obtained by the pintle nozzle is hollow conical spray.

Advantages

- It is self-cleaning type and prevents the carbon deposition on the nozzle hole.
- It avoids weak injection and dribbling.
- It results in good atomisation.
- Its injection characteristics are more near the required one.

Disadvantages

- Distribution and penetration poor, hence not suitable for open combustion chambers.

Pintaux Nozzle

- It is a type of pintle nozzle which has an auxiliary hole drilled in the nozzle body [Fig. 4.31)].
- It injects a small amount of fuel through this additional hole which is called pilot injection in the upstream direction slightly before the main injection.
- The needle valve does not lift fully at low speeds and most of the fuel is injected through the auxiliary hole.

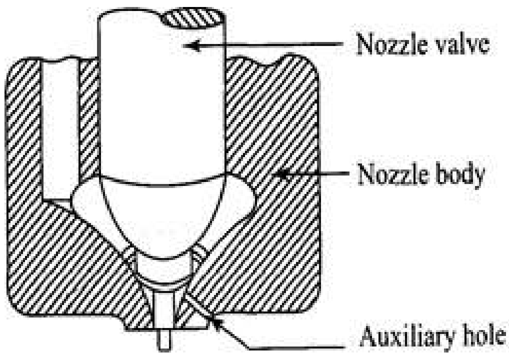


Fig. 4.31 Pintaux Nozzle

Advantage

- This nozzle gives better cold starting performance.

Disadvantage

- Its injection characteristics are poorer than the multihole nozzle.

Spray Formation

- The various phases of spray formation as the fuel is injected through the nozzle are shown in Fig. 4.32.

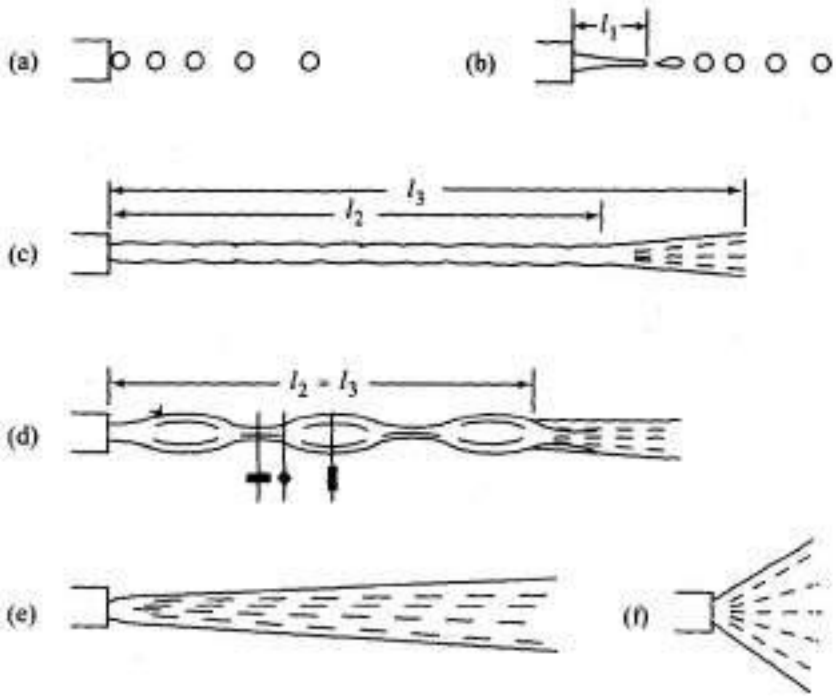


Fig. 4.32 Successive phases of spray formation

- At the start of the fuel-injection the pressure difference across the orifice is low. Therefore single droplets are formed as in Fig. 4.32(a). As the pressure difference increases the following process occur one after the other.
- A stream of fuel emerges from the nozzle, [Fig. 4.32(b)].
- The stream encounters aerodynamic resistance from the dense air present in the combustion chamber (12 to 14 times the ambient pressure) and breaks into a spray, say at a distance of l3, [Fig. 4.32(c)]. The distance of this point where this event occurs from the orifice is called the break-up distance.
- With further and further increase in the pressure difference, the break-up distance decreases and the cone angle increases until the apex of the cone practically coincides with the orifice [Fig. 4.32(d), (e) and (f)].
- At the exit of the orifice the fuel jet velocity, V/, is of the order of 400 m/s. It is given by the following equation

$$V_f = C_d \sqrt{\frac{2\left(p_{inj} - p_{cyl}\right)}{\rho_f}} \tag{4.28}$$

where C_d = coefficient of discharge for the orifice

p_{inj} = fuel pressure at the inlet to the injector, N/m^2

p_{cyl} = pressure of charge inside the cylinder, N/m^2

ρ_f = fuel density kg/m^3

- The spray from a circular orifice has a denser and compact core, surrounded by a cone of fuel droplets of various sizes and vaporized liquid.

- Larger droplets provide a higher penetration into the chamber but smaller droplets are required for quick mixing and evaporation of the fuel. The diameter of most of the droplets in a fuel spray is less than 5 microns.

Quantity of fuel and size of orifice

- The quantity of the fuel injected per cycle depends to a great extent upon the power output of the engine.
- The fuel is supplied into the combustion chamber through the nozzle holes and the velocity of the fuel for good atomization is of the order of 400 m/s. The velocity of the fuel through nozzle orifice in terms of h can be given by

$$V_f = C_d \sqrt{2gh} \tag{4.29}$$

  where h is the pressure difference between injection and cylinder pressure, measured in m of fuel column.
- The volume of fuel injected per second, Q, is given by

  Q = Area of all orifices × fuel jet velocity × time of one injection × number of injections per second for one orifice

$$Q = \left(\frac{\pi}{4} d^2 \times n\right) \times V_f \times \left(\frac{\theta}{360} \times \frac{60}{N}\right) \times \frac{N_i}{60} \tag{4.30}$$

  where N_i for four-stroke engine is rpm/2 and for a two-stroke engine N_i is rpm itself

  d is the diameter of one orifice in m,

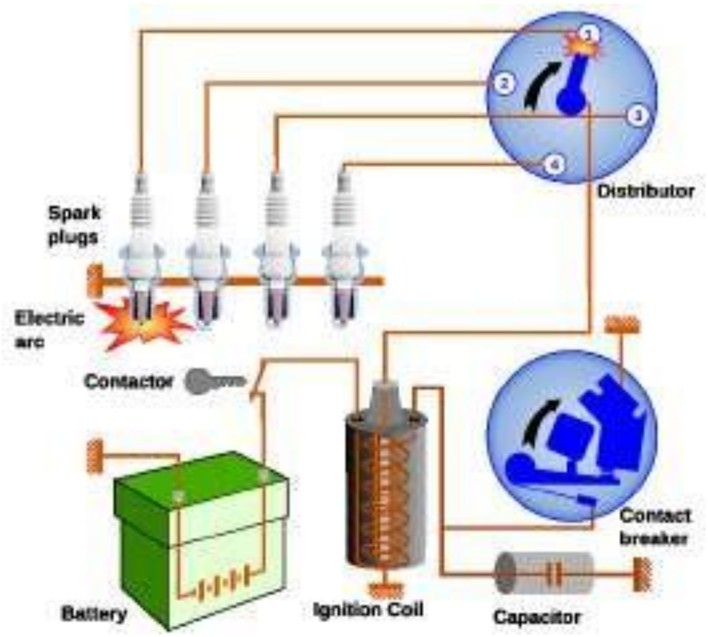
  n is the number of orifices,

  θ is the duration of injection in crank angle degrees and

  N_i is the number of injections per minute.
- Usually the rate of fuel-injection is expressed in mm^3/degree crank angle/litre cylinder displacement volume to normalize the effect of engine size.

5

IGNITION AND GOVERNING SYSTEM



Course Contents

Introduction

- In S.I. engine the combustion process is initiated by spark between the two electrodes of spark plug. This occurs just before the end of compression stoke. The ignition process must add necessary energy for starting and sustaining burning of the fuel till combustion takes place.
- Ignition is only a pre-requisite of combustion. It does not influence the gross combustion process. It is only a small scale phenomenon taking place within a specified small zone in the combustion chamber.
- Ignition only ensures initiation of combustion process and has no degree intensively or extensively.

Energy requirements for ignition:

- A spark energy below 10 millijoules is adequate to initiate combustion for A/F ratio 12-13 : 1 (Range of mixtures normally used); the duration of few micro-seconds is sufficient to start combustion.
- A spark can be struck between the gap in the in two electrodes of the spark plug by sufficiently high voltage. There is a critical voltage called *breakdown voltage* below which no sparking would occur. In practice the pressure, temperature and density have a profound influence on the voltage required to cause the spark. Also, the striking voltage is increased due to the fouling factor of the electrodes owing to deposits and abrasion.
- For automotive engines, in normal practice, the spark energy to the tune of 40 millijoules and duration of about 0.5 millisecond is sufficient over entire range of operation.

Requirements of an Ignition System

- For an ignition system to be acceptable it must be moderately priced, reliable and its performance must be adequate to meet all the demands imposed on it by various operating conditions.
- An ignition system should fulfil the following requirements:

i) It should have an adequate reserve of secondary voltage and ignition energy over the entire operating speed range of the engine.

ii) It should consume the minimum of power and convert it efficiently to a high-energy spark across the spark-plug electrode gap.

iii) It should have a spark duration which is sufficient to establish burning of the air-fuel mixture under all operating conditions.

iv) It should have an ability to produce an ignition spark when a shunt is established over the spark plug electrode insulator surface, due possibly to carbon, oil or lead deposit, liquid fuel or water condensation.

v) Good performance at high speed.

vi) Longer life of breaker points and spark plug.

vii) Good starting when the breaker points open slowly at cranking speed.

viii) Good reproducibility of secondary voltage rise and maximum rise.
ix) Adjustment of spark advance with speed and load.

- The basic source of electrical energy is either battery, a generator or magneto,
  - The battery and generator normally provide 6 V or 12 V direct current, while the magneto provides an alternating current of higher voltage.
  - The low voltage (6 V to 12 V) is boosted to a very high potential of about 10 kV to 20 kV, in order to overcome the spark gap resistance and to release enough energy to initiate self-propagating flame within the combustible mixture.

Battery ignition system

- It is a commonly used system because of its combined cheapness, convenience of mainte•nance, attention and general suitability.
- **Construction.** This system consists of the following components:

1. Battery (6 or 12 volts)
2. Ignition switch
3. Induction coil
4. Circuit/Contact breaker
5. Condenser
6. Distributor

- The system is shown in fig. 5.1.

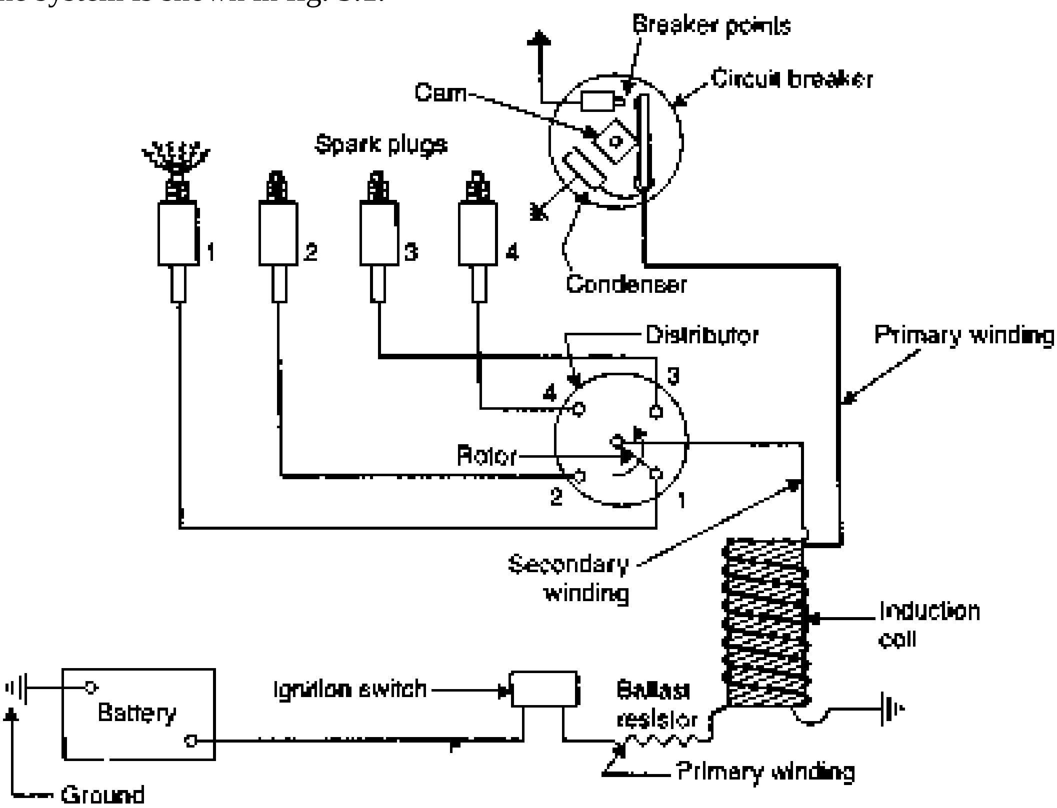


Fig. 5.1 Battery or Coil Ignition system.

- One terminal of the battery is ground to the frame of the engine, and other is connected through the ignition switch to one primary terminal of the ignition coil (consisting of a comparatively few turns of thick wire wound round an iron core).
- The other primary terminal is connected to one end of the contact points of the circuit breaker and through closed points to ground. The primary circuit of the ignition coil

thus gets completed when contact points of the circuit breaker are together and switch is closed.

- The secondary terminal of the coil is connected to the central contact of the distributor and hence to distributor rotor. The secondary circuit consists of secondary winding (consisting of a large number of turns of fine wire) of the coil, distributor and four spark plugs.
- The contact breaker is driven by a cam whose speed is half the engine speed (for four stroke engines) and breaks the primary circuit one for each cylinder during one complete cycle of the engine.
- The breaker points are held on contact by a spring except when forced apart by lobs of the cam.
- A ballast resistor is provided in series with the primary winding to regulate primary-current. For starting purposes this resistor is by passed so that more current can flow in the primary circuit.

Working:

- To start with, the ignition switch is made on and the engine is cranked i.e. turned by hand when the contacts touch, the current flows from battery through the switch, primary winding of the induction coil to circuit breaker points and the circuit is completed through the ground. A condenser connected across the terminals of the contact breaker points prevent the sparking at these points.
- The rotating cam breaks open the contacts immediately and breaking of this primary circuit brings about a change of magnetic field; due to which a very high voltage to the tune of 8000 to 12000 V is produced across the secondary terminals. (The number of turns in the secondary winding may be 50 to 100 times than in primary winding). Due to high voltage the spark jumps across the gap in the spark plug and air fuel mixture is ignited in the cylinder.
- Fig. 5.2 shows the gradually building up of the primary current from the time the points close until they open.

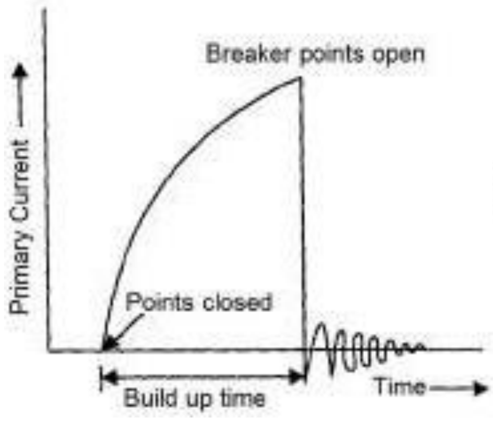


Fig. 5.2 Built up time for primary current

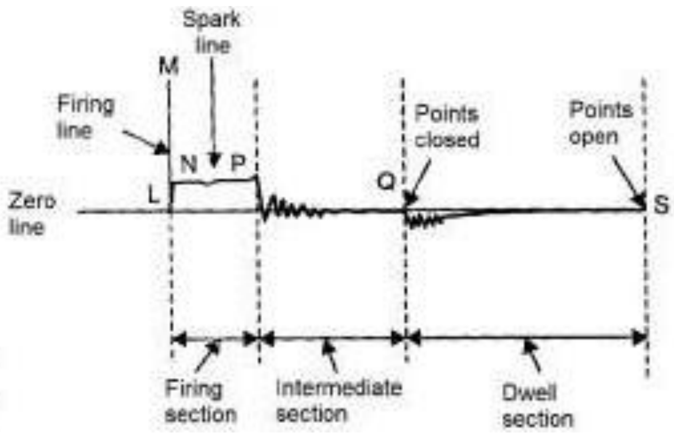


Fig. 5.3 Typical pattern of normal ignition action

- Fig 5.3 shows a typical wave-form or pattern of the normal ignition action.

- At point **L** the distributor opens and the magnetic field of the coil-primary winding collapses and consequently the secondary voltage, indicated by the firing line, rises to point **M**. The height of firing line shows the voltage needed to jump the rotor gap and to ionize the gap between the spark plug electrodes.
- After the spark is initiated the gap becomes ionized resulting in decreased gap resistance and a smaller voltage is then required to maintain the arc across the gap. The lower voltage and the spark duration is represented by the height and length of the spark line **NP**.
- At point **P** the major portion of the energy of the coil is expended and consequently there is a drop in the secondary voltage which result in extinguishing of the spark.
- Due to spark extinction the circuit becomes open, the current flow is stopped, and, hence the magnetic field (produced in the secondary winding, during the firing period **NP** while the current was flowing in the secondary winding and across the spark gap to ground) collapses, thereby, inducing a current in the primary winding, which eventually flows into the condenser and charge it.
- When voltage in the condenser becomes higher than that in the primary winding, it discharges back in the primary winding. This results in collapsing of the magnetic field and rebuilding up of voltage in the secondary winding. This pulsing back and forth, weaked each time, continues till whole of the energy is dissipated (Refer Fig. 5.3-intermediate section).
- At point **Q** the contact points close and remain so during dwell period. At the end of this period the points again open at S (there being no condenser action during the period, since it is shorted out across the closed points).

Advantages:

- It offers better sparks at low speeds, starting and for cranking purposes.
- The initial cost of the system is low.
- It is a reliable system and periodical maintenance required is negligible except for battery.
- Items requiring attention can be easily located in more accessible position than those of magnetos.
- The high speed engine drive is usually simpler than magneto drive.
- Adjustment of spark timing has no deterimental effect over the complete ignition timing range.

Disadvantages:

- With the increasing speed, sparking voltage drops.
- Battery, the only unreliable component of the system needs regular attention. In case battery runs down, the engine cannot be started as induction coil fails to operate.
- Because of battery, bulk of the system is high.

Components of Battery Ignition system

In this section we will discuss the essential components used in battery ignition system.

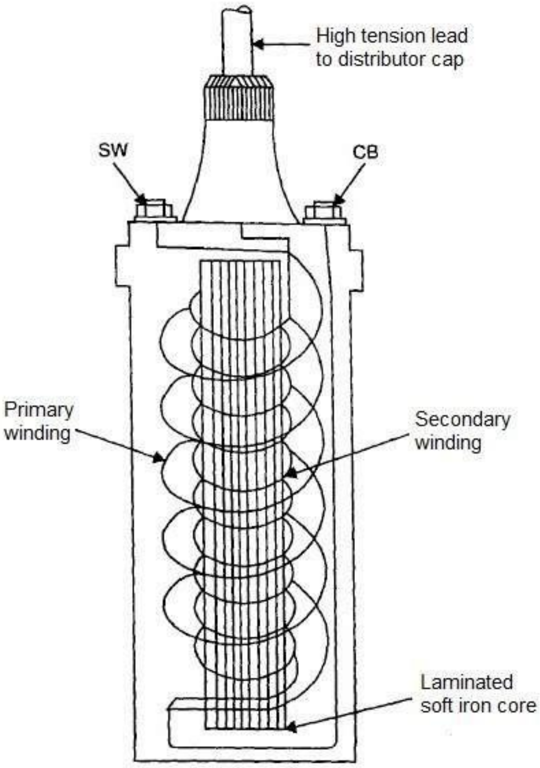
The battery:

- The function of battery is to store electricity in the form of chemical energy, when required to convert the latter back into electrical energy.
- Motor vehicles use lead-acid batteries which have a series of positive and negative plates which are interpersed, the plates being immersed in a solution of dilute sulphuric acid, called the electrolyte. For compactness the plates are placed close together and separators are used to reduce the chance of shorting taking place.

Advantage of 12 V ignition system over 6 V system:

- Considerably higher voltages are obtainable.
- For transmitting equal power with excessive voltage drop, the cable in a 6-V system needs theoretically to be four times the thickness of 12-V system, cables.
- Improved starting.
- Adequate electric power to supply the increasing number of electrical accessories used.

The ignition coil



- To create an adequate spark across the gap of sparking plug high electrical pressure is needed. Electrical pressure is measured in volts and the 12 volts supplied by the battery is totally inadequate.
- The function of ignition coil is to increase the voltage between 10,000 and 15,000 volts in some conditions, although the voltage which occurs under normal running conditions is of the order of 4000-5000 volts.
- Two coils of insulated wire are wound on a laminated soft iron core. The inner coil, called the secondary, has more turns than the outer primary coil. There are about 20000 turns on the secondary and 400 turns on the primary.
- If a low voltage passing through the primary coil is switched off a higher voltage is induced in the secondary coil, the increase being approximately in the same proportion to the number of turns of the two coils. The core and windings are placed in an iron sheath. The entire assembly being housed in a sealed container (Fig. 5.4).

- A high tension lead from the centre of the coil carries the supply to the distributor. Two small terminals are situated either side of the high tension lead, one being connected to the contact breaker and marked CB and the other to the ignition switch identified by the letters SW.

Contact breakers:

- This is a mechanical device for making [Fig. 5.5 (a)] and breaking [Fig. 5.5 (b)] the primary circuit of the ignition coil. It consists essentially of a fixed metal point against which, another metal point bears which is being on a spring loaded pivoted arm.

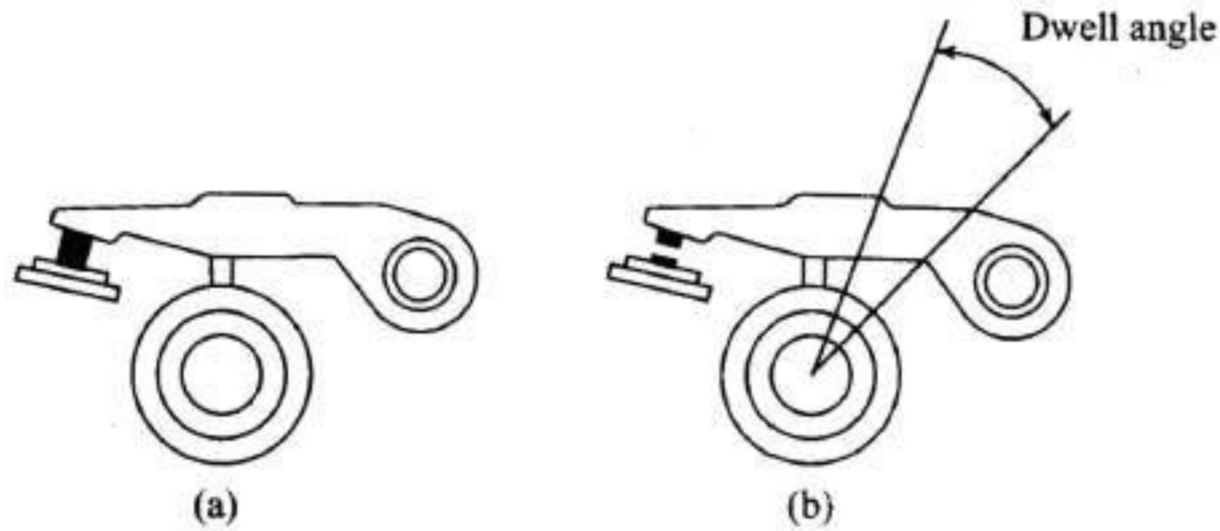


Fig. 5.5 Contact breaker

- The metal used is invariably one of the hardest metals, usually tungsten and each point has a circular flat face of about 3 mm diameter.
- The fixed contact point is earthed by mounting it on the base of the contact breaker assembly whereas the arm to which the movable contact point is attached, is electrically insulated. When the points are closed the current flows and when they are open, the circuit is broken and the flow of current stops. The pivoted arm has, generally, a heel or a rounded part of some hard plastic material attached in the middle and this heel bears on the cam which is driven by the engine. Consequently, every time the cam passes under the heel, the points are forced apart and the circuit is broken.
- The pivoted arm is spring loaded, so that when the points are not separated by the action of the cam, they are held together by the pressure of the spring thereby closing the primary circuit. The condition and adjustment of the contact breaker points are important. The points are subjected to a very severe hammering during their period of service. Uneven wear of the points may require a refacing or replacing depending upon the condition of the points.
- An eight cylinder engine running at 3000 rpm requires 12000 sparks per minute, i.e. 200 sparks per second. If the breaker is to operate satisfactorily at this speed, the

travel of the breaker arm must be held down to the minimum to ensure a positive spark and the breaker arm must be made very light.

The distributor

- The distributor includes the contact breaker points and the mechanism for automatically advancing or retarding the spark timing in accordance with the engine speed for optimum power to be developed by the engine.
- The function of the distributor is to distribute the ignition surges to the individual spark plugs in the correct sequence and at the correct instants in time. Depending on whether a particular engine has 4, 6 or 8 cylinders, there are 4, 6 or 8 ignition pulses (surges) generated for every rotation of the distributor shaft. The use of a distributor represents a considerable simplification in a battery ignition system because in most cases we want to use only a single ignition circuit.

The Condenser

- It consists of sheets of metal foil separated by an insulating material (e.g. mica) placed face to face.
- One sheet of metal foil is connected to condenser terminal, next to the metal case of the condenser and so on alternatively.
- The condenser terminal is connected to one side of contact breaker and the casing to the other side of contact breaker and usually earth so that the condenser remains in parallel with the contact breaker.
- As the contact breaker points separate in the distributor, the flow of current from the battery through the primary winding of the coil is interrupted.
- Instantly the magnetic field begins to collapse and this collapse attempts to re-establish the flow of current.
- If the condenser is not provided, the current would be re-established which would result into a heavy arc across the separating contact breaker points and the energy of the ignition coil will be consumed by the arc. This may bum the contact breaker points.
- The condenser prevents the arcing across the contact breaker points and prolongs its life.
- Therefore, the functions of condenser are:

  (i) To minimise arcing and pitting of contact breaker points.

  (ii) To intensify the spark

Magneto ignition system

- Magneto is a special type of ignition system with its own electric generator to provide the necessary energy for the system.
- It is mounted on the engine and replaces all the components of the coil ignition system except the spark plug. A magneto when rotated by the engine is capable of producing a very high voltage and does not need a battery as a source of external energy.
- A schematic diagram of a high tension magneto ignition system is shown in Fig. 5.6. The high tension magneto incorporates the windings to generate the primary voltage

as well as to step up the voltage and thus does not require a separate coil to boost up the voltage required to operate the spark plug.

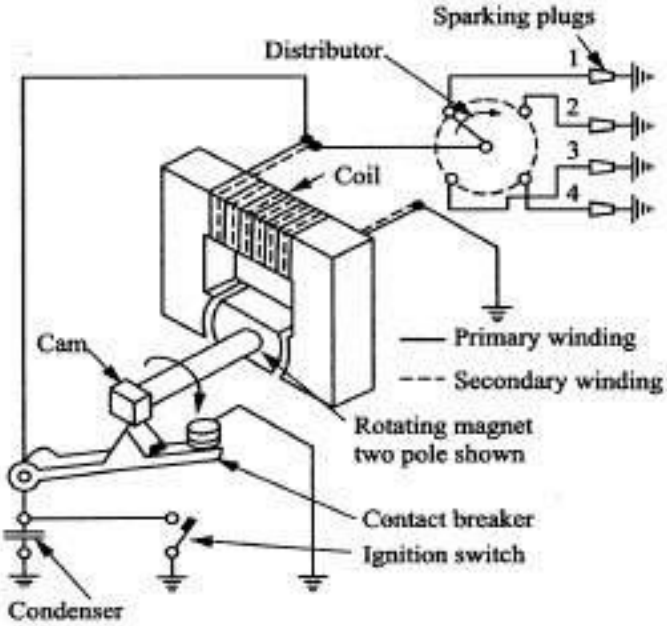


Fig. 5.6 High tension magneto ignition system

- Magneto can be either rotating armature type or rotating magnet type. In the first type, the armature consisting of the primary and secondary windings all rotate between the poles of a stationary magnet, whilst, in the second type the magnet revolves and the windings are kept stationary. A third type of magneto called the polar inductor type is also in use. In the polar inductor type magneto both the magnet and the windings remain stationary but the voltage is generated by reversing the flux field with the help of soft iron polar projections, called inductors.
- The working principle of the magneto ignition system is exactly the same as that of the coil ignition system. With the help of a cam, the primary circuit flux is changed and a high voltage is produced in the secondary circuit.
- Fig. 5.7 compares the breaker current vs. speed curve of the coil ignition system with that of the magneto ignition system.

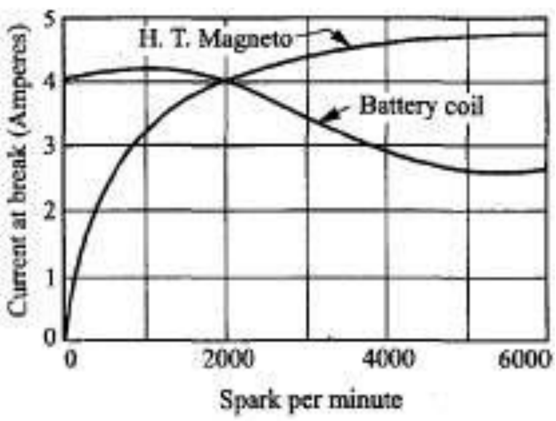


Fig. 5.7 Break current vs. speed for coil and magneto ignition systems

- It can be seen that since the cranking speed for starting is low the current generated by the magneto is very low. As the engine speed increases the current increases. Thus, with magneto there is almost always a starting difficulty and a separate battery is needed for starting. The magneto is best at high speeds, and therefore, is widely used for sports and racing cars, craft engines, etc.

- In comparison to the coil ignition system magneto system is more costly but highly reliable. However, due to the poor starting characteristics and due to the fact that voltage generated is effected with the changes in spark timing, almost invariably the coil ignition system is preferred to the magneto system.
- The coil or battery ignition system requires more maintenance than the magneto system. It is also heavier than the magneto system.

Low Tension Magneto Ignition system

- The main disadvantage of the high tension magneto ignition system lies in the fact that the wirings carry a very high voltage current and thus there is a high possibility of causing engine misfire due to leakage. To avoid this the high tension wires must be suitably shielded.
- The development of the low tension magneto system is an attempt to avoid this trouble.
- In the low tension magneto system the secondary winding is changed to limit the secondary voltage to a value of about 400 volts and the distributor is replaced by a brush contact. The high voltage is obtained with the help of a step-up transformer. All these changes have effect of limiting the high voltage current only in a small portion of the ignition system wiring and, thus, avoid the possibilities of leakage, etc.

Comparison between Battery (coil) Ignition System and Magneto Ignition System

Table 5.1 Comparison of Battery (coil) and Magneto Ignition system

	Battery (coil) Ignition System		Magneto Ignition System
1.	Battery is must. Impossible to start the engine when battery is discharged.	1.	No battery is needed hence no problem of battery discharge.
2.	Current for primary circuit is obtained from the battery.	2.	The required electric current is generated by the magneto.
3.	A good spark is available at spark plug at low speed.	3.	During staring quality of spark is poor due to low speed.
4.	Staring of engine is easier.	4.	Engine starting is rather difficult.
5.	Efficiency of the system falls with fine fall of spark intensity as engine speed rises.	5.	The intensity of spark keeps on improving as the speed goes on increasing. The efficiency of the system thus improves as the engine speed rises
6.	Occupies more space.	6.	Occupies less space.
7.	Mostly employed in petrol cars and buses.	7.	Used in racing cars, motor cycles, scooters, etc.

Spark plug

- The function of the spark plug is to generate the spark in the combustion chamber using a high voltage communicated by the secondary. The spark plug provides two electrodes with a proper gap across which, high potential is discharged and spark is generated.
- A sectional view of a conventional spark plug is shown in Fig. 5.8. It consists of a steel shell, an insulator, and two electrodes. The high voltage supply from secondary is given to the central electrode which is insulated with porcelain. The other electrode is welded to the steel shell of the plug and thereby automatically grounded when the plug is fitted in the cylinder head of the engine. The electrodes are made of high nickel alloy to withstand severe corrosion and erosion to which they are subjected.

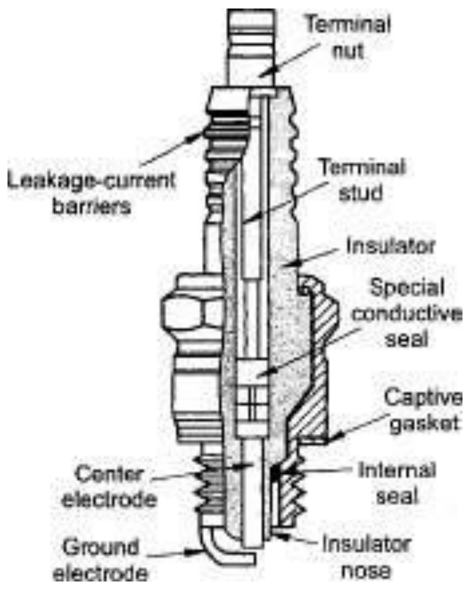


Fig. 5.8 Schematic of a Typical Spark Plug.

- The tips of central electrode and insulation are exposed to the burned gases. This results high thermal stresses and the insulator may crack. As the tips are subjected to high temperature (2000-2500°C), the heat must flow from the insulator and tip to the surrounding shell in order to cool the electrodes and prevent preignition.
- The spark plugs are classified as hot plug and cold plug depending upon the temperature at the tip of the electrodes. The operating temperature of the tip depends upon the amount of heat transferred and it depends upon the path followed by the heat to flow. A cold plug has a short heat flow path where as hot plug follows a long flow path for the heat to flow as shown in Fig. 5.9.

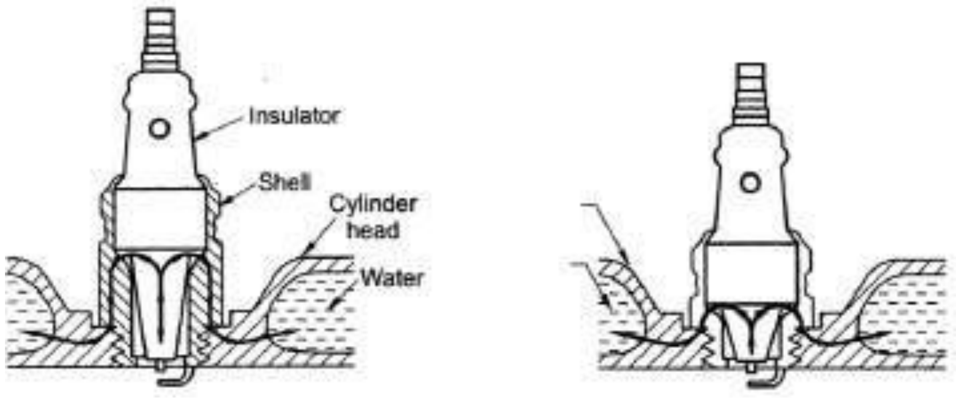


Fig. 5.9 Heat Transfer Path of Hot and Cold Spark Plug

- The hot plug is used to avoid cold fouling where combustion chamber temperatures are relatively low as during low power operation and continuous idling.
- A spark plug which runs satisfactory, the temperature at cruising speed may run cool at idling speed and tips will be fouled by unburned carbon deposits or excess

lubricating oil. The carbon deposits burns at 350°C where as lubricating deposits burn at 550°C. If the spark plug runs hot at idling speed to prevent carbon deposits, it may run too hot at high speed. This may cause undesirable preignition. If the plug runs above 800°C, then preignition generally occurs.

- Insulator tip length is the most important parameter which controls the operating temperature. Therefore, the tip temperature is generally controlled by varying insulator tip position and electrode material.
- It is necessary in practise to compromise in order to obtain a proper spark plug which would operate satisfactorily throughout the engine operating range. An improper spark plug has remained a major source of engine trouble as misfiring and preignition.

Firing order

- The sequence in which, firing impulses occur in multicylinder SI-engine is called the firing order. The angle between the successive crank throws in multi-cylinder engine govern the order in which successive pistons arrive at TDC.
- The firing order should be such that there must be always a proper balance and it does not cause vibrations.
- In 4-cylinder, 4-stroke engine, the firing in all cylinders will over in two revolutions of the crank-shaft. With crank throws at 180°, the cylinders 1 and 4 will reach TDC simultaneously. If the firing interval is made 180°, the firing in cylinder-1 cannot be followed by the cylinder-4. Similarly, the firing of the cylinder-2 cannot be followed by cylinder-3. Therefore, the possible firing order in 4-cylinder engine is 1-2-4-3 or 1-3-4- 2, the latter is more popular.
- In case of inline 6-cylinders engine, the cranks are set at 120° and the cylinders 1-6, 2-5 and 3-4 will be at TDC simultaneously. Therefore, the firing order should be arranged to take place in front and rear halves of the engine cylinder. The possible sequence of 6-cylinder engine is

  1-5-3-6-2-4 or 1-4-2-6-3-5
- In 8-cylinder engines, the cranks are set (360 x 2/8) to 90° and possible firing orders are

  1-6-2-5-8-3-7-4 and 1-5-4-8-6-3-7-2

  1-8-4-3-6-5-7-2 and 1-8-7-3-6-5-4-2
- To decide the sequence for V-arrangement is more complicated.

Governing of IC engine

- The purpose of governing is to maintain the speed of the engine constant regardless of the changes in the load on the engine. The mechanism used for this purpose is known as governor and method used is kwon as governing.
- If the load on the engine decreases, the speed of the engine will begin to increase if the fuel supply is not decreased. On the other hand, if the load on the engine increases, the speed of the engine will begin to decrease if the fuel supply is not

increased. The purpose of governing is to supply the fuel to the engine according to the load on the engine and to maintain the speed of the engine constant.

The methods of governing:

- The governing of speed of the engine according to the load is done by one of the following methods:
  i) The fuel supplied to the engine is completely cut off during few cycles of the engine. This is known as **Hit and Miss Governing**. This is generally used for gas engine.
  ii) The fuel supplied per cycle of the engine is varied according to the load on the engine. This is known as **Quality Governing**. The A : F ratio is changed according to the load on the engine. Rich mixture is supplied at high loads and lean mixture is supplied at low loads. This is used for diesel engines.
  iii) The quantity of air-fuel mixture supplied is varied according to the load on the engine. The A : F ratio of the mixture supplied to the engine at all loads remain merely constant, therefore it is known as **Quantity Governing**. This is used for petrol engine.
- All these methods are discussed in detail below.

Hit and Miss Governing

- This method is used for gas engines as well as for oil engines but is more popular in gas engines only.
- This system of governing omits the explosions occasionally when the speed of the engine rises above the mean speed of the engine. The number of omitted explosions are increased with the increase in speed.

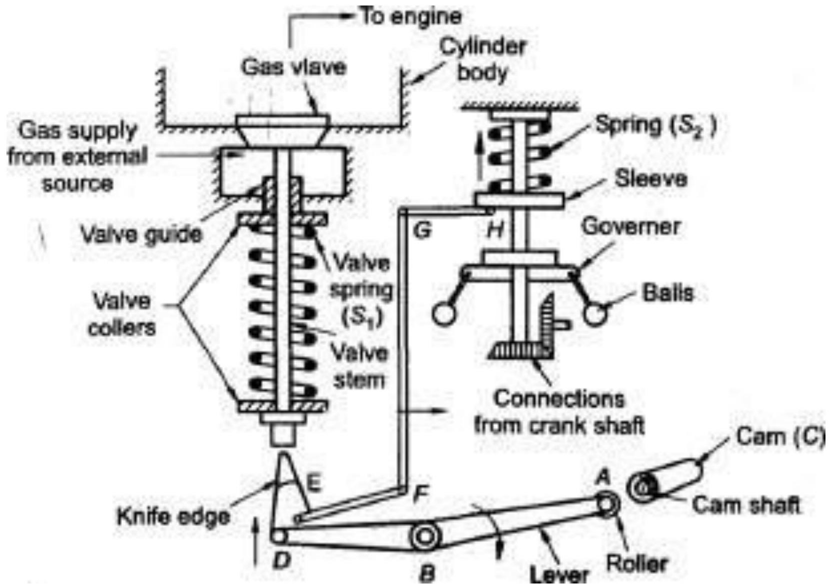


Fig. 5.10 hit and Miss Governing

- The usual method adopted for missing an explosion is to omit the opening of the gas valve in case of gas engine and putting the plunger of the oil pump out of action in the case of oil engines. During the missing cycle, the engine performs an idle cycle.
- The outline of the method used in gas engine is shown in Fig. 5.10.

- The position of all the components of the system are shown in the figure when the engine is running at full load. The cam 'C' rotates at half speed of the crankshaft. As the cam C pushes the point A, the point D is lifted upwards because the lever BD turns about the fulcrum B and hits the valve steam through the knife-edged point E and opens the valve to allow the gas to the engine cylinder. At full load condition, there is working stroke for every cycle of the engine.
- When the load on the engine is decreased, the speed of the crank-shaft increases and the speed of the governor also. The balls fly out as the speed of the spindle, on which the governor is fixed, increases. The governor sleeve is pushed up and the point 'H' of the lever GH also goes up and the point F on the lever GF moves towards the right as shown in figure.
- The point 'E' on the knife edge is also moved towards the right and misses the opening of the gas valve. The loss of power due to missing cycle decreases the speed of the engine. The point E is lifted up by the cam during the missing cycle also but as it is pushed away (towards the right) from the original position, it is not possible to open the gas valve.
- The number of the missing cycles increases with the further decrease in load. The missing cycles are zero when the engine is running at full load condition. The directions of motion of all components under low load condition are shown by an arrow on the figure.
- This method is known as **Hit and Miss** method because the valve is opened by giving the hit and speed control is achieved by missing the openings of the gas valve.
- The principle and mechanism of the method used for oil engine are exactly same but the plunger of the fuel pump is put out of action instead of gas valve.
- With this method of governing the engine, the engine either works under maximum efficiency condition or does not fire at all. This method gives better economy at light loads than any other method.
- The great disadvantage of this method is, the engine requires heavy flywheel as the absence of turning effort on the crankshaft during the idle cycle. This method is used for the engines of small B.P. (below 20 kW) and do not require close speed regulation.

Quality Governing

- The amount of fuel supplied to the diesel engine cylinder per cycle is varied according to the load on the engine in this method of governing.
- The quantity of fuel supplied according to the load on the engine is varied by one of the following methods.

a) The stroke of the fuel pump plunger is varied by the governor and quantity of oil supplied is varied according to load.

- The weight of fuel supplied by the fuel pump is given by

$$m_f = \frac{\pi}{4} d^2 L \rho_f$$

where d is the diameter of the fuel pump and L is the stroke of the fuel pump.

As L, is varied, the fuel supplied to the engine cylinder also varies.

b) A control valve is inserted to the delivery side of the fuel pump. The opening of the valve is controlled by the governor. It opens after a part of delivery stroke is performed. In this system, the oil is delivered to the engine cylinder from the fuel pump during some part of the delivery stroke and returned back to the suction side during the remaining part of the delivery stroke.

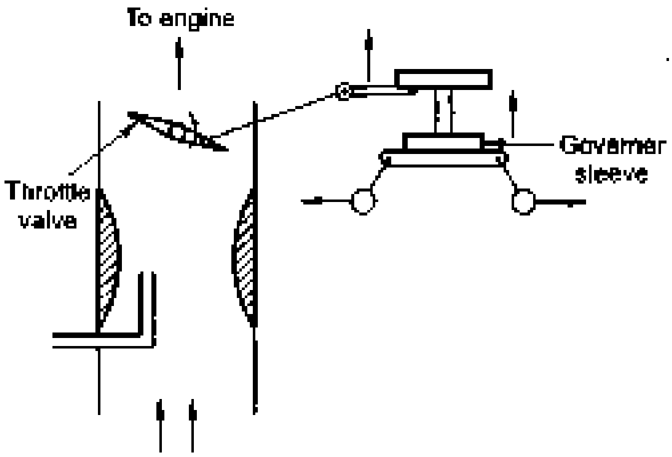
c) The closing of suction valve of the fuel pump may be delayed. The suction valve of the fuel pump remains open during first part of the delivery stroke, therefore part of the fuel oil is returned back to the fuel pump during some part of the delivery stroke and delivered to the engine during the remaining part of the delivery stroke. The method is known as "Spill Method”.

d) The stroke of the pump plunger remains constant at all loads but the effective stroke is changed according to the load on the engine. By changing the angular position of the helical groove on the plunger of the fuel pump relative to the suction port, the amount of fuel delivered can be changed. This is the common method used in all modem C.I. high speed engines.

- In all these systems, the air supply remains constant and the quantity of fuel supplied is changed, therefore the quality of the mixture (A : F) changes according to the load on engine.

Quantity Governing

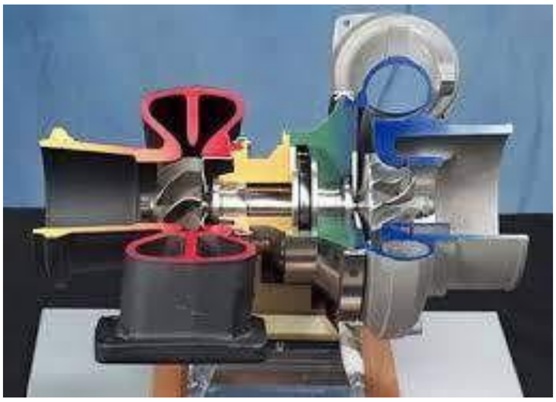
- This is used in many gas engines and is commonly used for all petrol engines. The mixture strength supplied to the engine is maintained constant but the quantity supplied to the engine is varied by means of a throttle valve. The movement of the throttle valve is regulated by the lift of the centrifugal governor.
- The arrangement is shown in Fig. 5.11.



- As the engine speed increases (due to decrease in load), the governor balls fly out and the governor sleeve lifted up and partly throttle valve is closed reducing the quantity of mixture supplied to the engine cylinder. This reduces the indicated mean effective pressure and ultimately the power developed by the engine.
- This system of governing can be used for gas engines in various ways. The air and gas supplied, each can be throttled by separate valves in the air and gas passages and then supplied to the engine or a mixture of air and gas of constant A : F ratio coming out from the mixing valve is supplied to the engine cylinder. The quantity of mixture supplied is controlled by varying the lift of the main inlet valve.

6

SUPERCHARGING



Course Contents

Objectives of super charging,

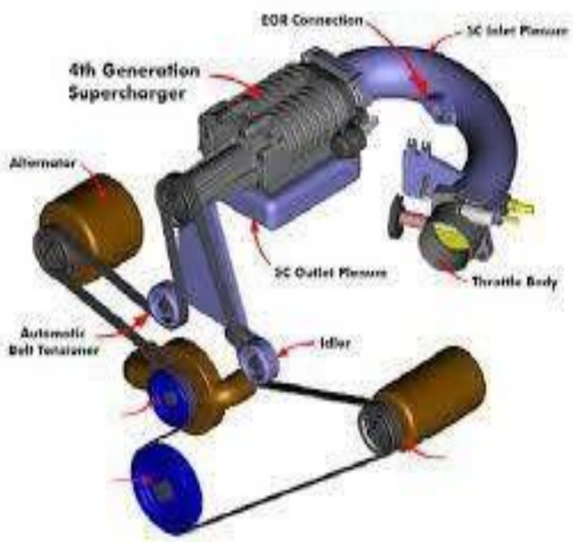
Types of Superchargers.

Supercharging of SI and CI engines.

Effects of supercharging, supercharging Limits.

Methods of supercharging.

Turbo Charging.



Objectives of super charging

- Supercharging of internal combustion engines has been used for many years as a method to improve engine performance and efficiency. Entering the millennium, a new trend is appearing. The trend points to small displacement engines in order to meet federal emission legislation on fuel consumption and emission control. The driver, however, still demands the same performance they're used to.
- A good way to meet these needs is supercharging otherwise known as forced induction. The purpose of supercharging an engine is to raise the density of the air charge, before it's delivered to the cylinders. Thus, the increased mass of air trapped and then compressed in each cylinder during each induction and compression stroke makes more oxygen available for combustion than the conventional method of drawing the fresh air charge into the cylinder (naturally aspirated). Consequently, more air and fuel per cycle will be forced into the cylinder, and this can be efficiently burnt during the combustion process to raise the engine power output to higher than would otherwise be possible.
- Generally, there are three basic types of "superchargers," the most popular being the exhaust-gas driven turbocharger, mechanically driven superchargers and the pressure-wave supercharger. The mechanically driven supercharger is broken up into two groups as well, the mechanically driven centrifugal supercharger and the mechanically-driven positive displacement supercharger such as the screw-type and roots-type.
- In automotive and marine applications, the pressure-wave supercharger is rarely used. The turbo and roots-type superchargers have been the most popular forced induction methods in the past. While the turbo creates great peak horsepower, turbo lag and high cold start emissions due to the thermal mass are severe drawbacks of the turbocharger. Small displacement engines need higher-pressure ratios to achieve the performance demanded by the driver.
- This fact increases the mentioned drawbacks of the turbo and makes the turbocharger a less desirable alternative for supercharging than the mechanical twin-screw supercharger. The Whipple twin-screw charger does not have the usual drawbacks of earlier mechanical superchargers such as the roots-type, such as poor efficiency especially at high-pressure ratios, high rpm, high noise level as well as high price.
- Comparative tests, made independently by Whipple Industries, show that the twin-screw compressor is the most effective supercharging method available.
- The purpose of supercharging in diesel engine is to compress the fresh air out of the working cylinder. And then the density of air in cylinder can be increased by increasing the air pressure. So more fuel can be combusted and the power of diesel engine is increased too. Lots of practice indicates that supercharging is the best way to increase the power and economy of diesel engine. So supercharging is widely used recently.

- The equipment which presses the air or mixed air to definite pressure is called compressor. The pressure of air which has been pressed is called supercharging pressure. And it is signed with Pk. Supercharging system of diesel engine contains compressor, compressor driving equipment and cooler. Supercharging system can be divided into 3 types by the difference of energy source in driving supercharge.

Types of Superchargers.

The working principle of centrifugal compressor

- The power and efficiency of an internal combusting engine can be increased with the use of an air compression device such as a supercharger or turbocharger. Increasing the pressure and density of the inlet air will allow additional fuel to be inducted into the cylinder, increasing the power produced by the engine.
- Spark ignition engines are knocking limited, restricting the allowable compressor pressure increase; in many cases the compression ratio of a SI engine is reduced. Superchargers and turbochargers are used extensively on a wide range of diesel engines, since they are not knocking limited.
- The types of compressors used on internal combustion engines are primarily of two types: positive displacement and dynamic. With a positive displacement compressor, a volume of gas is trapped, and compressed by movement of a compressor boundary element.
- Three types of positive displacement compressors are the roots, vane, and screw compressor, as shown in figure 6.1. Centrifugal compressor is based on that the action of high speed airflow and working impeller or fixed blade. Compared with the cubage compressor, centrifugal compressor has many merits.
- For example, the efficient of centrifugal compressor is higher than cubage compressor with consuming same power. And the supercharging pressure is usually high. There are also many other advantages, such as light weight and compact structure.
- At present, centrifugal compressor is used widely in supercharging of inner combustion engine. Though it has many advantage, its supercharging pressure will reduce sharply with the reduce of rotational speed.
- Centrifugal compressor is constituted with the following parts. Air flows into the gas passage from the filter. In order to increase the stability of airflow, the section of intake gas passage is smaller and smaller along the direction of airflow. Intake gas passage must transport air to the working impeller equably under the situation of the least flowing loss. Working impeller can be drive mechanically by crankshaft or turbine.
- Air flows into the working impeller along intake gas passage and rotates together with the working impeller. Air is compressed when it flows along the passage formed by the blade of working impeller with centrifugal effect. At the time, pressure changes from P1 to P2; speed of airflow changes from c1 to c2.

- The mechanical power which drives working impeller changes into kinetic energy and potential energy. The power at the outtake of working impeller is half of the whole energy of airflow. So, diffuser is used after working impeller to change the power to pressure energy.
- The main parameters of compressor characteristic are flux and supercharging rate. The air supplied for diesel engine in one unit time is called flux. And the supercharging rate is the ratio of air pressure at intake and outtake.

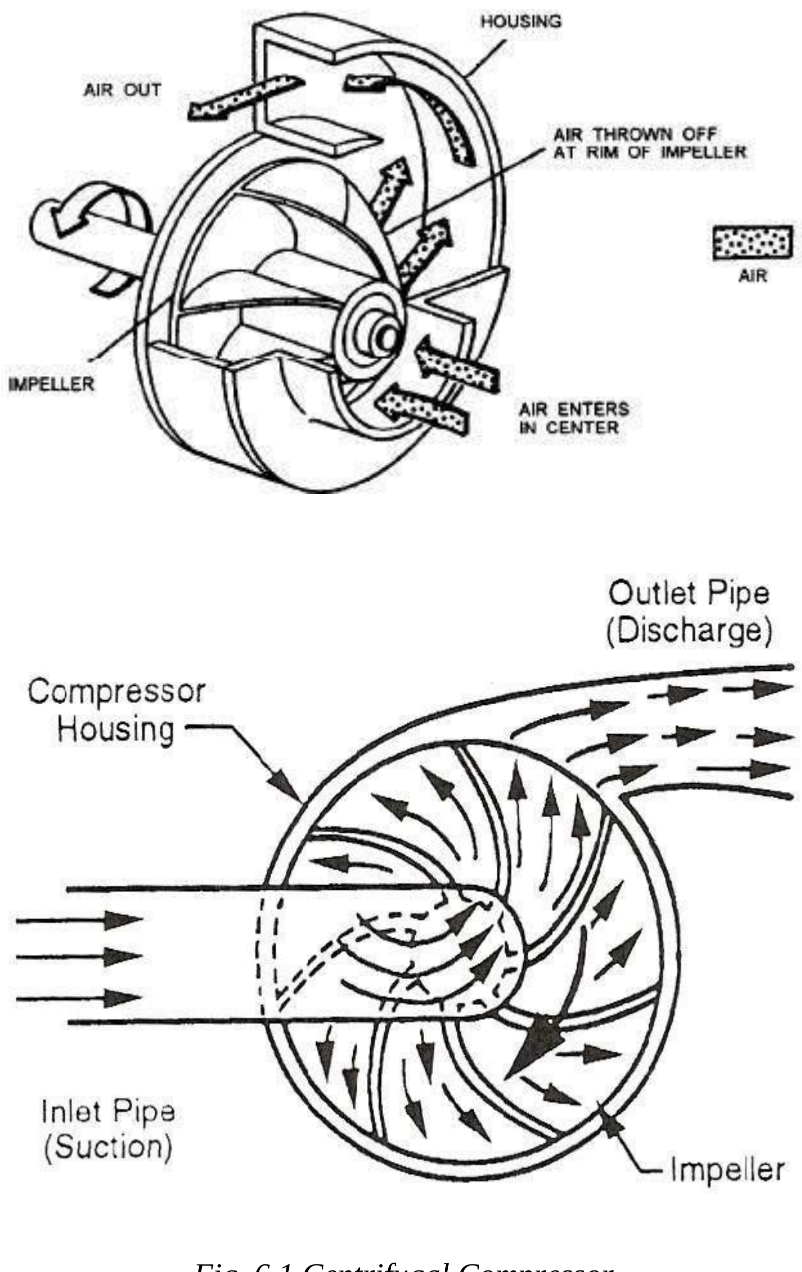


Fig. 6.1 Centrifugal Compressor

The working principle of root compressor

- Let us consider how the action of a displacement device like the Roots blower differs from that of a true compressor. Imagine a simple hypothetical reciprocating piston compressor in 2 versions, both delivering air through a valve into a chamber in which the pressure is significantly above atmospheric. Both inhale a fresh charge of air on the piston down-stroke but in one version the delivery valve opens at the commencement of the up-stroke whilst in the other the delivery valve opens only when the charge of air has been compressed to the same pressure as the chamber. Both convey the same amount of air into the chamber but the way in which it is done is quite different.
- In the first case the higher pressure air in the chamber will initially flow back into the compressor cylinder until pressures are equal and then it will be driven out again along with the new charge as the piston approaches the top of the stroke. Clearly unnecessary work is involved in this rather turbulent process but a further disadvantage is that the full pressure from the chamber is acting against the piston throughout the up stroke.
- In the second case no such back-flow from the chamber occurs and the pressure acting against the piston rises gradually as the up stroke proceeds and only reaches that of the chamber towards the end of the stroke, at which point the valve opens allowing the charge to be expelled into the chamber. It must be obvious that this involves considerably less pumping work and can be carried out with less power.
- In fact the excess power required by the first device is converted into heat which is added to the air transferred to the chamber. So this device not only consumes more power for the same job but it also heats up the air more as it does so. Hardly ideal for a detonation prone internal combustion engine - yet that is more or less how the Roots blower works, with adiabatic efficiency of barely 60%.
- There are a great many examples of true compressors analogous to our second hypothetical device but the most viable for use as an engine driven supercharger is the screw compressor devised in the 1930s by a noted Swedish engineer called Lysholm. It has two helical rotors, male and female, which intermesh very closely without ever making contact. The helix action is of a continuous movement from the entry port towards a closed face adjacent to which is the delivery port. Air is drawn in normally at the open end of the rotors and is trapped as the rotors mesh together, gradually compressed between the rotor lobes and the closed end face as it is propelled along, then expelled from the delivery port at the required pressure.
- An important point is that the area of the rotor lobes acting on the charge diminishes as compression takes place therefore the power requirement remains more or less constant throughout the process. This is a significant advantage over our hypothetical piston compressor where increasing effort is required to drive the piston against the rising pressure. Of course the Roots device fares badly in this

respect having the entire lobe area exposed to the higher pressure as delivery takes place.

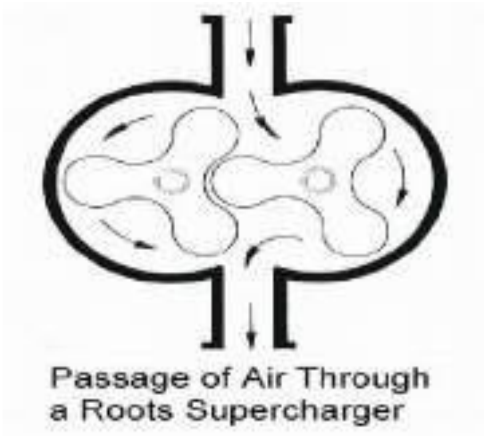


Fig. 6.2 Root Compressor

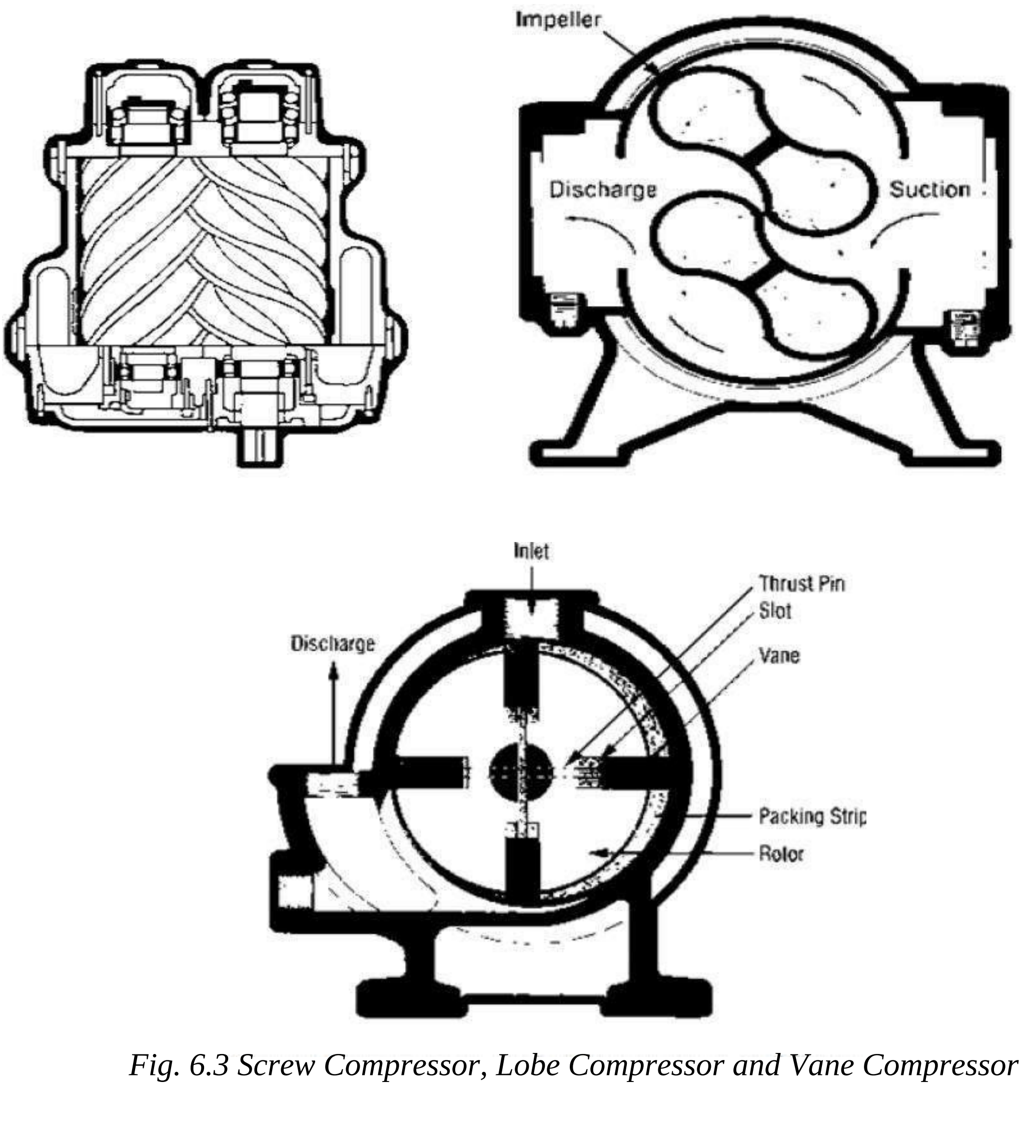


Fig. 6.3 Screw Compressor, Lobe Compressor and Vane Compressor

The working principle of vane compressor

- This is positive type supercharger. Its vanes are so designed as to crowd the air in a smaller space. The air blades slide in slots designed as to crowd the air in smaller space. The air blades slide in a slots or groove ion the hub of the supercharger.

Comparison between three superchargers

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

Effect of supercharging

- Air is supplied at high pressure which increases volumetric efficiency.
- Supercharged engines develop more power
- Mechanical efficiency is more than that of naturally aspirated engines.
- Supercharging gives better turbulence, proper air fuel ratio and efficient combustion of fuel
- The specific fuel consumption of super charged engines is less due to better turbulence and proper air fuel mixture.
- Super charging tends to increase the possibility of detonation in SI engines.
- Super-charging tends to decrease the possibility of knocking in CI engines.
- At high altitudes, it is possible to obtain sufficient air by supercharging only.
- Super-charging shortens the 'delay period'

Supercharging of SI and CI engines

- An engine that uses normal vacuum to draw in its air fuel fixture is called normally aspirated, that is not supercharged. Use of an air pump to deliver an air fuel mixture to the engine cylinders at a pressure greater than atmospheric pressure is called supercharging. Boost is the measure of the amount of air pressurization, above atmospheric, that a supercharger can deliver. The term supercharger usually applies to an air pressurizing pump, driven mechanically by the engine crankshaft through gears, shafts, chains or belts. These mechanical linkages consume a lot of power from the engine. Some superchargers are often driven at speeds of 50,000 to 90,000 rpm. Engine-driven superchargers, therefore, have not been popular for passenger car engines. Turbocharger is a super-charging device that uses exhaust gases to turn a turbine to force extra air into the cylinders.
- Turbo-charging has been very common on large CI engines for many years because this type of engine is particularly suited to pressure charging. Unlike the SI engine, the CI engine does not suffer from compression limitations. In addition, since only air being in the induction system and more under-bonnet space is available on a CI unit, fitting of a turbocharger is much easier

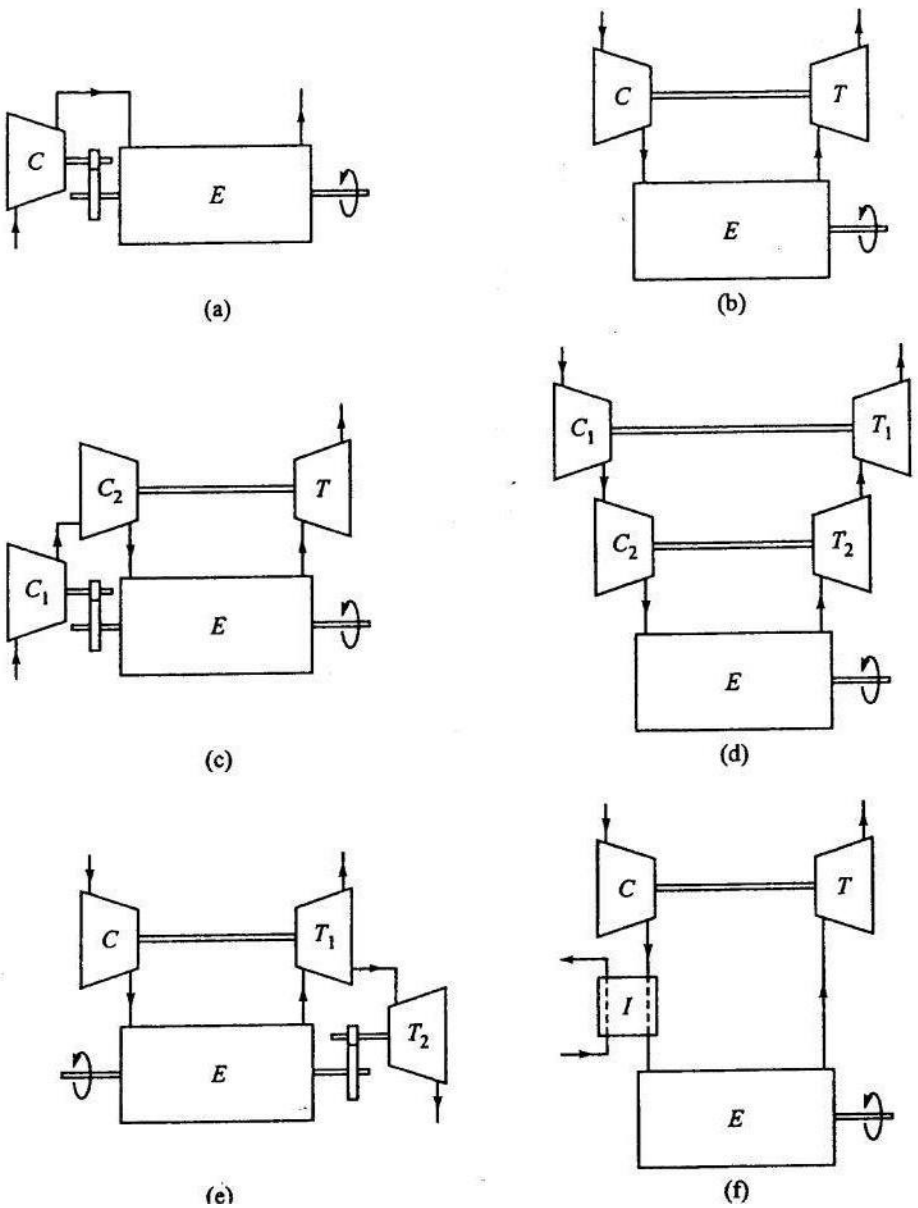


Fig. 6.4 Methods of Supercharging

TURBOCHARGERS:

- Turbochargers are a type of forced induction system whose function is same as that of Supercharger. They compress the air flowing into the engine. A turbocharged engine produces more power overall than the same engine without the charging. This can significantly improve the power-to-weight ratio for the engine.
- In order to achieve this boost, the turbocharger uses the exhaust flow from the engine to spin a turbine, which in turn spins an air pump. The turbine in the turbocharger spins at speeds of up to 150,000 RPM. The turbocharger is bolted to the exhaust manifold of the engine. The exhaust from the cylinders spins the turbine, which works like a gas turbine engine. The turbine is connected by a shaft to the

compressor, which is located between the air filter and the intake manifold. The compressor pressurizes the air going into the pistons.

- The exhaust from the cylinders passes through the turbine blades, causing the turbine to spin. The more exhaust that goes through the blades, the faster they spin. On the other end of the shaft that the turbine is attached to, the compressor which pumps air into the cylinders. The compressor is a type of centrifugal pump; it draws air in at the centre of its blades and flings it outward as it spins. In order to handle speeds of up to 150,000 rpm, the turbine shaft has to be supported very carefully. Most turbochargers use a 'Fluid Bearing'. This type of bearing supports the shaft on a thin layer of oil that is constantly pumped around the shaft. This serves two purposes: It cools the shaft and some of the other turbocharger parts and it allows the shaft to spin without much friction. Some turbochargers use 'Ball Bearings' instead of fluid bearings to support the turbine shaft. But these are not your regular ball bearings, they are super-precise bearings made of advanced materials to handle the speeds and temperatures of the turbocharger.
- Ceramic turbine blades are lighter than the steel blades used in most turbochargers. When air is compressed, it heats up; and when air heats up, it expands. So some of the pressure increase from a turbocharger is the result of heating the air before it goes into the engine. An intercooler or charge air cooler is an additional component that looks something like a radiator, except air passes through the inside as well as the outside of the intercooler. The intake air passes through sealed passage ways inside the cooler, while cooler air from outside is blown across fins by the engine cooling fan.

Methods of Turbo charging and their Advantages and Limits:

Constant Pressure Turbo charging:

The exhaust from various cylinders discharge into a common manifold at pressure at pressures higher than the atmospheric pressure. The exhaust gasses from all the expanded in the exhaust valves to an approximately constant pressure in common manifold from here it passes to turbine. Thus the blow-down energy, in the form of internal energy, is converted into work in the turbine. The exhaust gases are maintained at constant pressure during the whole cycle so that a pure Reaction turbine can be used.

Advantages:

- The exhaust piping is very simple for a multi-cylinder engine as well as single-cylinder; highly efficient turbine can be used.
- Engine speed is not limited by the pressure waves in the exhaust pipes.

Disadvantages:

- Scavenging is not efficient.
- At part load the efficiency of turbine reduces due to partial admissions to the turbine.

Pulse Turbo charging: Considerable part of the blow-down energy is converted into exhaust pulses as soon as the exhaust valve opens. Towards the end of exhaust the pressure in the exhaust pipe drops below the scavenging and large air pressure making scavenging quite easy. The rate of the exhaust gas at the various turbine inlets is different and variable in time.

Advantages:

- The space required is less due to short and smaller diameter pipes.
- Comparatively better scavenging is obtained at low loads due to reduced pressure.

Disadvantages:

- With large number of cylinders complicated inlet and exhaust pipe arrangements are needed.
- The length of the pipe or engine speed is limited.

Two Stage Turbo charging: Two –stage turbo charging is defined as use of two turbochargers of different sizes in series; for example a high-pressure stage operating on pulse system and a low-pressure stage on constant pressure operation.

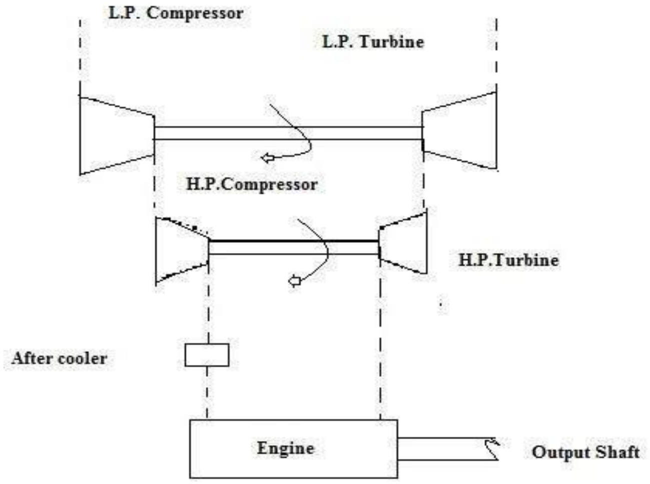


Fig. 6.5 Methods of Turbo Charging

Advantages:

- Better matching of the turbochargers to engine operating conditions possible.
- The efficiency of two-stage turbocharger is higher than that of a single stage turbocharger having a high boost ratio.

Disadvantages:

- The space requirement is higher.
- The total system is heavier.

Limitations of Turbo charging:

- The use of turbochargers requires special exhaust manifolds.

- Fuel injection has to be modified to inject more fuel per unit time.
- The efficiency of the turbine blades is very sensitive to gas velocity so that it is very difficult to obtain good efficiency over a wide range of operations.
- One of the main problems with turbochargers is that they do not provide an immediate power boost. It takes a second for the turbine to get up to speed before boost is produced. This results in a lag known as 'Turbo Lag'.

Methods to overcome Turbo lag

- One way to decrease turbo lag is to reduce the inertia of the rotating parts, mainly by reducing their weight. This allows the turbine and compressor to accelerate quickly, and start providing boost earlier.
- A small turbocharger will provide boost more quickly and at lower engine speeds.
- Most automotive turbochargers have a waste gate, which allows the use of a smaller turbocharger to reduce lag.
- Some engines use two turbochargers of different sizes. The smaller one spins up to speed very quickly, reducing lag, while the bigger one takes over at higher engine speeds to provide more boost.

7

COMBUSTION IN SI ENGINE

Course Contents

7.1 Ignition Lag and the Factors Affecting the Ignition Lag.

7.2 Flame Propagation and Factors Affecting Flame Propagation.

Abnormal combustion and knocking in SI engines,

Factors Affecting Knocking, Effects Of Knocking, Its Control.

Combustion chambers for SI Engines

Auto Ignition

Effect Of Detonation

Effect of engine operating variables on the engine knocking

7.9 Combustion chamber for SI engine

INTRODUCTION

- 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.

7.2. 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 neighboring unburnt mixture. For hydrocarbons fuel the stoichiometric fuel air ratios 1:15 and hence the fuel air ratio must be about 1:30 and 1:7.

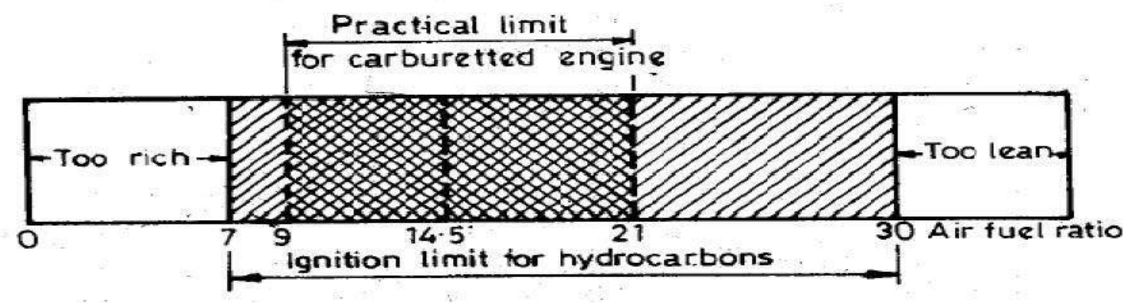


Fig7.1 Ignition Limits For Hydrocarbon

THEORIES OF COMBUSTION IN SI ENGINE

- Combustion in SI engine may roughly divide 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 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.

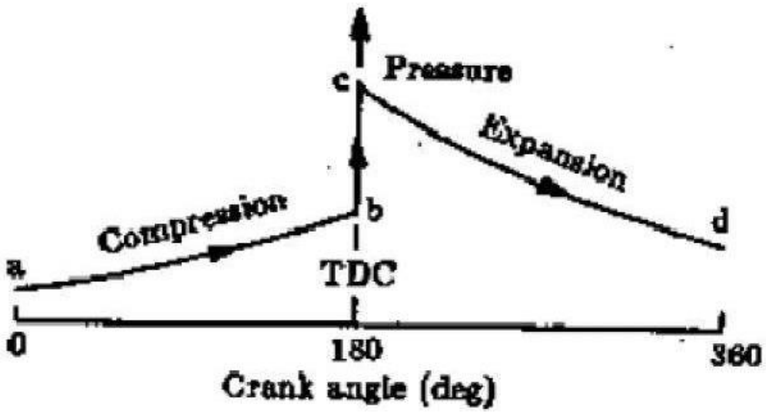


Fig. 7.2 theoretical p-Ө Diagram

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

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

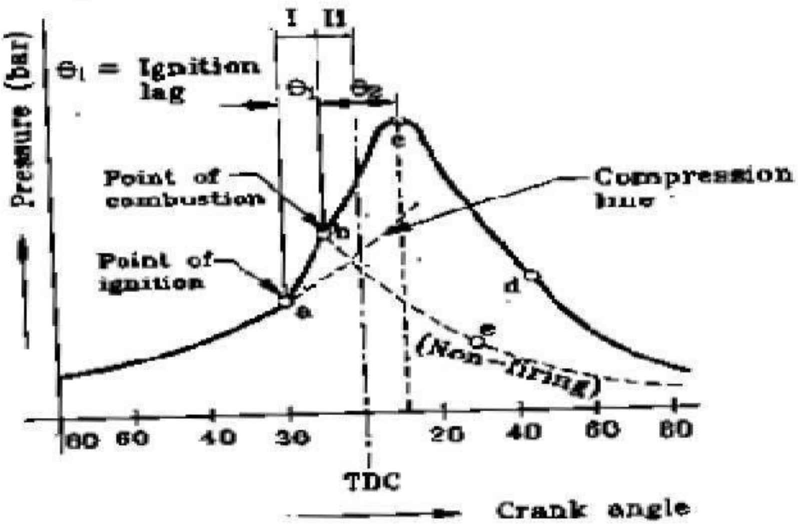


Fig. 7.3p-Ө Diagram

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 (1). 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 q2 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

- 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 between10 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

- 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 Coin stead 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:

- 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.
- 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 load 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 afterburning.
- 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.

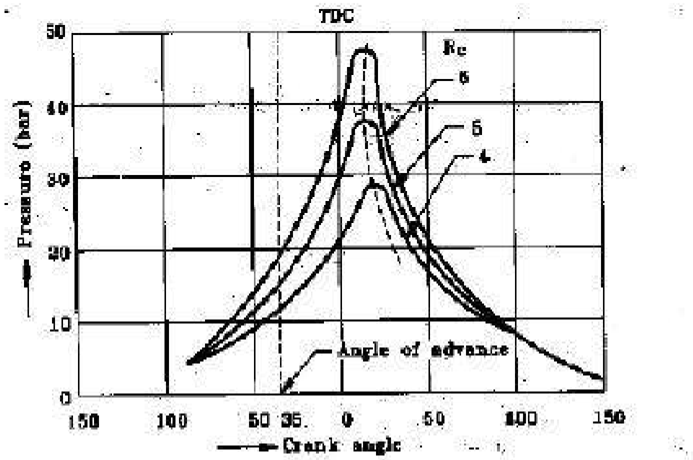


Fig. 7.4p-Ө Diagram

- **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 characteristic 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.

- **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

- 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 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. 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.

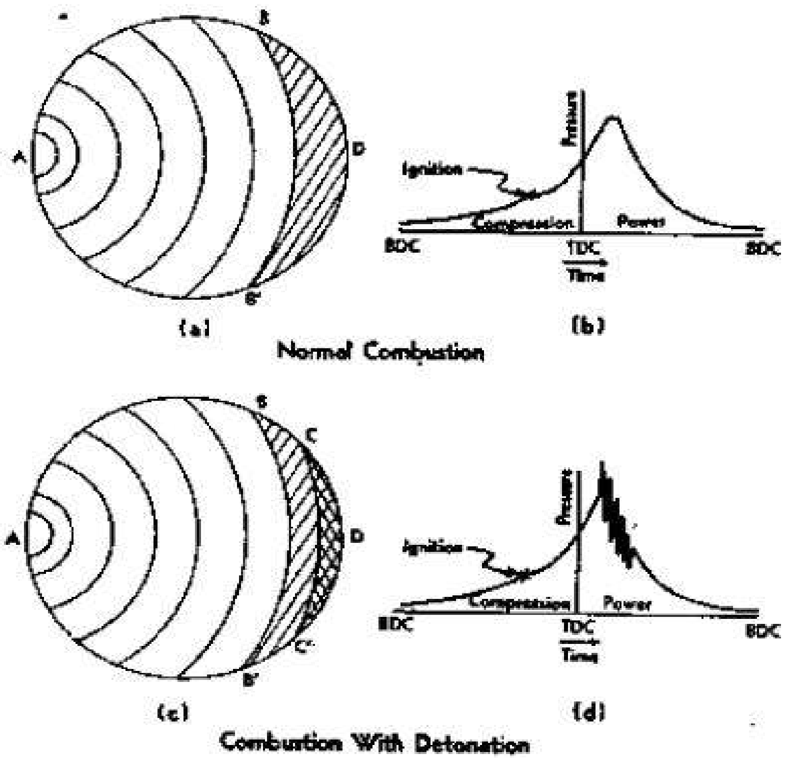


Fig. 7.4 Schematics Presentation of the Principle of Normal and Abnormal Combustion

AUTO IGINITION

- 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. 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.

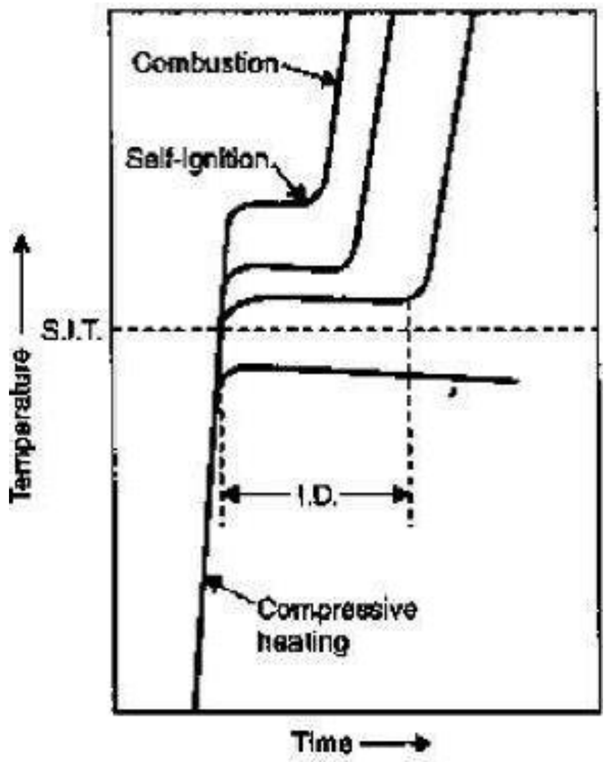


Fig. 7.5 Self-ignition characteristics of fuels

PRE -IGINITION

- 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 therefore provides a focal point for pre-ignition.
- Effects of Pre-ignition

1. It increases the tendency of denotation in the engine.
2. It increases heat transfer to cylinder walls because high temperature gas remains in contact with for a longer time.

3. Pre-ignition in a single cylinder will reduce the speed and power output.
4. Pre-ignition may cause seizer in the multi-cylinder engines, only if only cylinders have pre-ignition.

EFFECT OF DETONATION

The harmful effects of detonation are as follows:

- 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.
- 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 cavitations), 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.
- Carbon deposits.
    - Detonation results in increased carbon deposits.
- 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.
- 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.
- 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

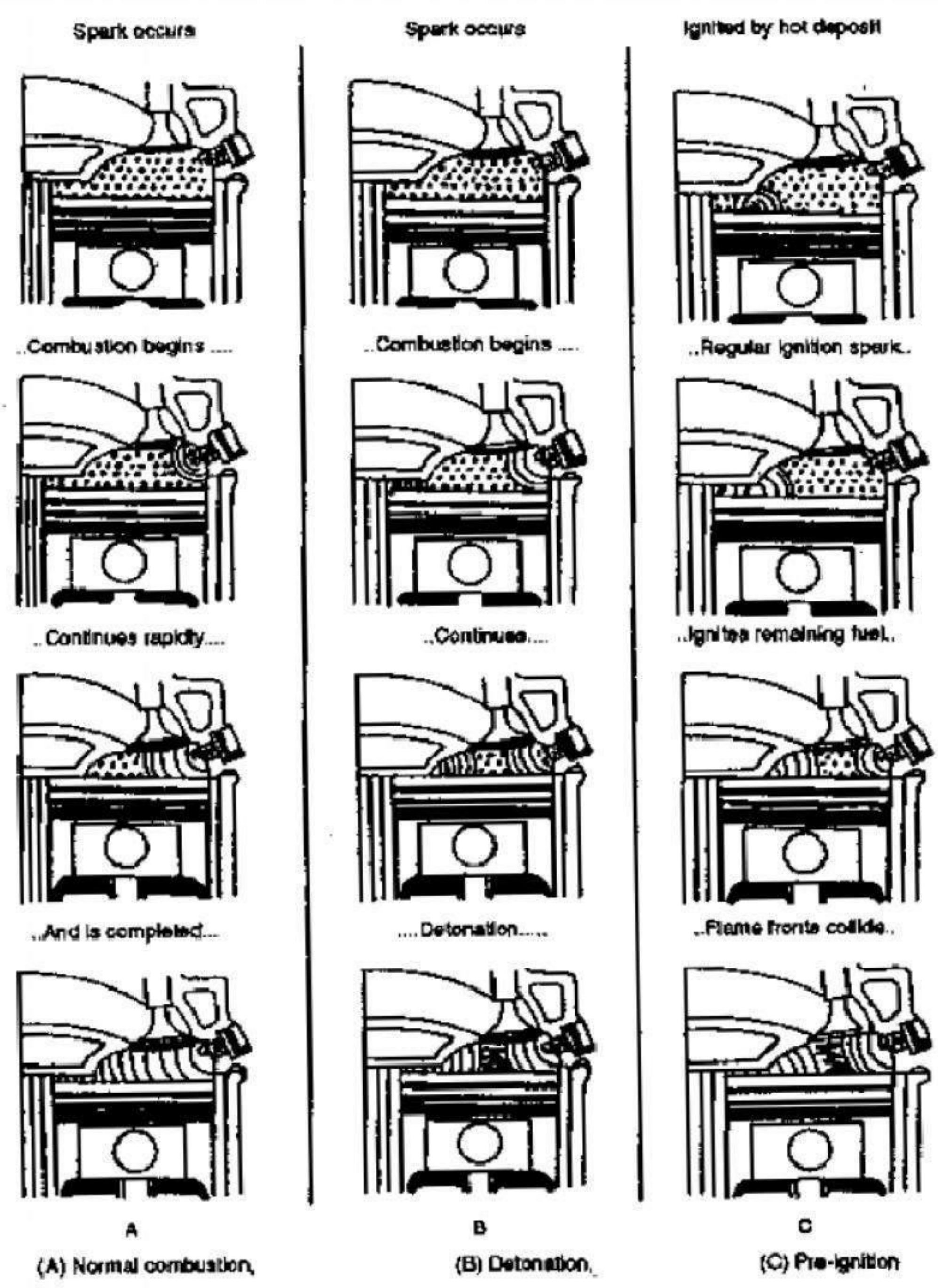


Fig. 7.6 normal and abnormal combustion

EFFECT OF ENGINE OPERATING VARIABLES ON THE ENGINE KNOCKING

The various engine variables 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.

- o 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 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.

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 hydrocarbons.

- Paraffin’s
  - Increasing the length of carbon chain increases the knocking tendency.
  - Centralizing 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
- Nathans and Aromatics
  - Nathans 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 stoichiometric (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

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 CHAMBER FOR SI ENGINE

- The design of combustion chamber has an important influence upon the engine Performance and its knock properties. The design of combustion chamber involves the shape of the combustion chamber, the location of the sparking plug and the disposition of inlet and exhaust valves. Because of the importance of combustion chamber design, it has been a subject of considerable amount of research and development in the last fifty years. It has resulted in raising the compression ratio from 4: 1 before the First World War period to 8: 1 to 11:1 in present times with special combustion Chamber designs and suitable anti-knock fuels.

BASIC REQUIREMENTS OF A GOOD COMBUSTION CHAMBER

The basic requirements of a good combustion chamber are to provide:

- High power output
- High thermal efficiency and low specific fuel consumption
- Smooth engine operation
- Reduced exhaust pollutants.

HIGHER POWER OUTPUT REQUIRES THE FOLLOWING:

- High compression ratio. The compression ratio is limited by the phenomenon of detonation. Detonation depends on the design of combustion chamber and fuel quality. Any change in design that improves the anti-knock characteristics of combustion chamber permits the use of a higher compression ratio which should result in higher output and efficiency.
- Small or no excess air.
- Complete utilization of the air – no dead pockets.
- An optimum degree of turbulence. Turbulence is induced by inlet flow configuration or ‘squish’. Squish is the rapid ejection of gas trapped between the piston and some flat or corresponding surface in the cylinder head. Turbulence induced by squish is preferable to inlet turbulence since the volumetric efficiency is not affected.
- High Volumetric Efficiency. This is achieved by having large diameter valves with ample clearance round the valve heads, proper valve timing and straight passage ways by streamlining the combustion chamber so that the flow is with lesser pressure drop. This means more charge per stroke and proportionate increase in the power output. Large valves and straight passageways also increase the speed at which the maximum power is obtained. This further increases the power by increasing the displacement per minute.

(C) SMOOTH ENGINE OPERATION REQUIRES THE FOLLOWING:

- Moderate rate of pressure rise during combustion.
- Absence of detonation which in turn means:
- Compact combustion chamber, short distance of flame travel from the sparking plug to the farthest point in the combustion space. Pockets in which stagnant gas may collect should be avoided.
- Proper location of the spark plug and exhaust valve.

- Satisfactory cooling of the spark plug points (to avoid pre ignition) and of exhaust valve head which is the hottest region of the combustion chamber.

(D) Reduced exhaust pollutants

- Exhaust pollutants can be reduced by designing a combustion chamber that produces a faster burning rate of fuel. A faster burning chamber with its shorter burning time permits operation with substantially higher amounts of Exhaust Gas Recirculation (EGR), which reduces the oxides of nitrogen (NOX) in the exhaust gas without substantial increase in the hydrocarbon emissions.
- It can also burn very lean mixtures within the normal constraints of engine smoothness and response. A faster burning chamber exhibits much less cyclic variations, permitting the normal combustion at part load to have greater dilution of the charge.

DIFFERENT TYPES OF COMBUSTION CHAMBERS

A few representative types of combustion chambers of which there are many more Variations are enumerated and discussed below:

1. T-head combustion chamber.
2. L-head combustion chamber.
3. I-head (or overhead valve) combustion chamber.
4. F-head combustion chamber.

It may be noted that these chambers are designed to obtain the objectives namely:

- A high combustion rate at the start.
- A high surface-to-volume ratio near the end of burning.
- A rather centrally located spark plug.

T Head Type Combustion chambers

This was first introduced by Ford Motor Corporation in 1908. This design has following disadvantages.

- Requires two cam shafts (for actuating the in-let valve and exhaust valve separately) by two cams mounted on the two cam shafts.
- Very prone to detonation. There was violent detonation even at a compression ratio of 4. This is because the average octane number in 1908 was about 40 -50.

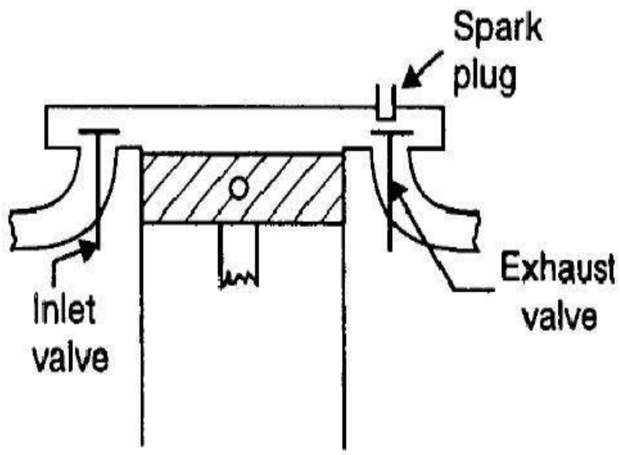


Fig. 7.7T-head type combustion chamber

L Head Type Combustion chambers

It is a modification of the T-head type of combustion chamber. It provides the two values on the same side of the cylinder, and the valves are operated through tappet by a single camshaft. This was first introduced by Ford motor in 1910-30 and was quite popular for some time. This design has an advantage both from manufacturing and maintenance point of view.

Advantages:

- Valve mechanism is simple and easy to lubricate.
- Detachable head easy to remove for cleaning and decarburizing without Disturbing either the valve gear or main pipe work.
- Valves of larger sizes can be provided.

Disadvantages:

- Lack of turbulence as the air had to take two right angle turns to enter the cylinder and in doing so much initial velocity is lost.
- Extremely prone to detonation due to large flame length and slow combustion due to lack of turbulence.
- More surface-to-volume ratio and therefore more heat loss.
- Extremely sensitive to ignition timing due to slow combustion process
- Valve size restricted.
- Thermal failure in cylinder block also. In I-head engine the thermal failure is confined to cylinder head only

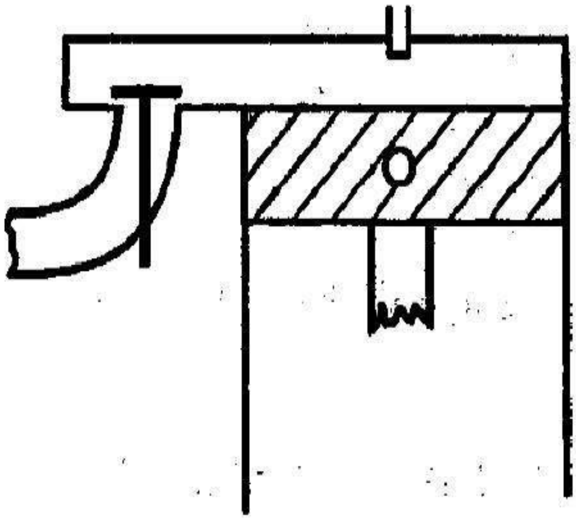


Fig. 7.8 L-head type combustion chamber

RICARDO'S TURBULENT HEAD– SIDE VALVE COMBUSTION CHAMBER

- Ricardo developed this head in 1919. His main objective was to obtain fast flame speed and reduce knock in L design. In Ricardo's design the main body of combustion chamber was concentrated over the valves, leaving slightly restricted passage communicating with cylinder.

Advantages:

- Additional turbulence during compression strokes possible as gases are forced back through the passage.

- By varying throat area of passage designed degree of additional turbulence is possible.
- This design ensures a more homogeneous mixture by scoring away the layer of stagnant gas clinging to chamber wall. Both the above factors increase the flame speed and thus the performance.
- Deign make engine relatively insensitive to timing of spark due to fast combustion
- Higher engine speed is possible due to increased turbulence
- Ricardo’s design reduced the tendency to knock by shortening length of effective flame travel by bringing that portion of head which lay over the further side of piston into as close a contact as possible with piston crown.
- This design reduces length of flame travel by placing the spark plug in the centre of effective combustion space.

Disadvantages:

- With compression ratio of 6, normal speed of burning increases and turbulent head tends to become over turbulent and rate of pressure rise becomes too rapid leads to rough running and high heat losses.
- To overcome the above problem, Ricardo decreased the areas of passage at the expense of reducing the clearance volume and restricting the size of valves. This reduced breathing capacity of engine; therefore these types of chambers are not suitable for engine with high compression ratio. Over head valve or I head combustion chamber.

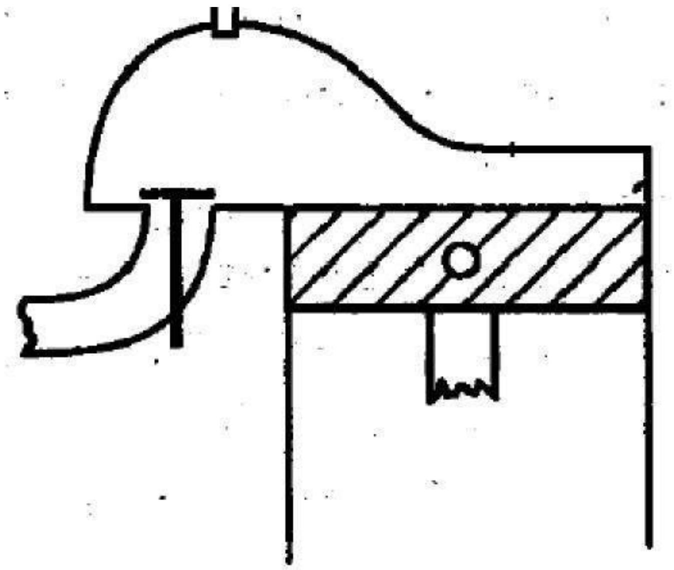


Fig. 7.9Ricardo’s Turbulent Head combustion Chamber

Over head valve or I head combustion chamber

The disappearance of the side valve or L-head design was inevitable at high compression ratio of 8: 1 because of the lack of space in the combustion chamber to accommodate the valves. Diesel engines, with high compression ratios, invariably used overhead valve design. Since 1950 or so mostly overhead valve combustion chambers are used. This type of combustion chamber has both the inlet valve and the exhaust valve located in the cylinder head. An overhead engine is superior to side valve engine at high compression ratios. The overhead valve engine is superior to side valve or L head engine at high compression ratios, for the following reasons:

- Lower pumping losses and higher volumetric efficiency from better breathing of the engine from larger valves or valve lifts and more direct passageways.
- Less distance for the flame to travel and therefore greater freedom from knock, or in other words, lower octane requirements.
- Less force on the head bolts and therefore less possibility of leakage (of compression gases or jacket water). The projected area of a side valve combustion chamber is inevitably greater than that of an overhead valve chamber.
- Removal of the hot exhaust valve from the block to the head, thus confining heat failures to the head. Absence of exhaust valve from block also results in more uniform cooling of cylinder and piston.
- Lower surface-volume ratio and, therefore, less heat loss and less air pollution.
- Easier to cast and hence lower casting cost.

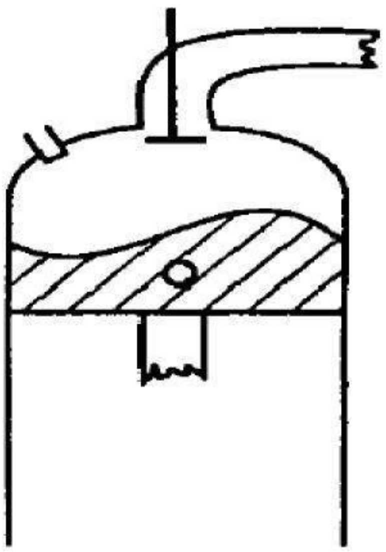


Fig. 7.10 I-Head combustion Chamber

Two important designs of overhead valve combustion chambers are used.

- Bath Tub Combustion Chamber.
- This is simple and mechanically convenient form. This consists of an oval shaped chamber with both valves mounted vertically overhead and with the spark plug at the side. The main drawback of this design is both valves are placed in a single row along the cylinder block. This limits the breathing capacity of engine, unless the overall length is increased. However, modern engine manufactures overcome this problem by using unity ratio for stroke and bore size.
- Wedge Type Combustion Chamber. In this design slightly inclined valves are used. This design also has given very satisfactory performance. A modern wedge type design can be seen in for Plymouth V-8 engine. It has a stake of 99 mm and bore of 84mm with compression ratio 9:1

F- Head combustion chamber

In such a combustion chamber one valve is in head and other in the block. This design is a compromise between L-head and I-head combustion chambers. One of the most Head engines (wedge type) is the one used by the Rover Company for several years. Another successful design of this type of chamber is that used in Walleyes jeeps

Its advantages are:

- High volumetric efficiency
- Maximum compression ratio for fuel of given octane rating
- High thermal efficiency
- It can operate on leaner air-fuel ratios without misfiring.

The drawback

- This design is the complex mechanism for operation of valves and expensive special shaped piston.

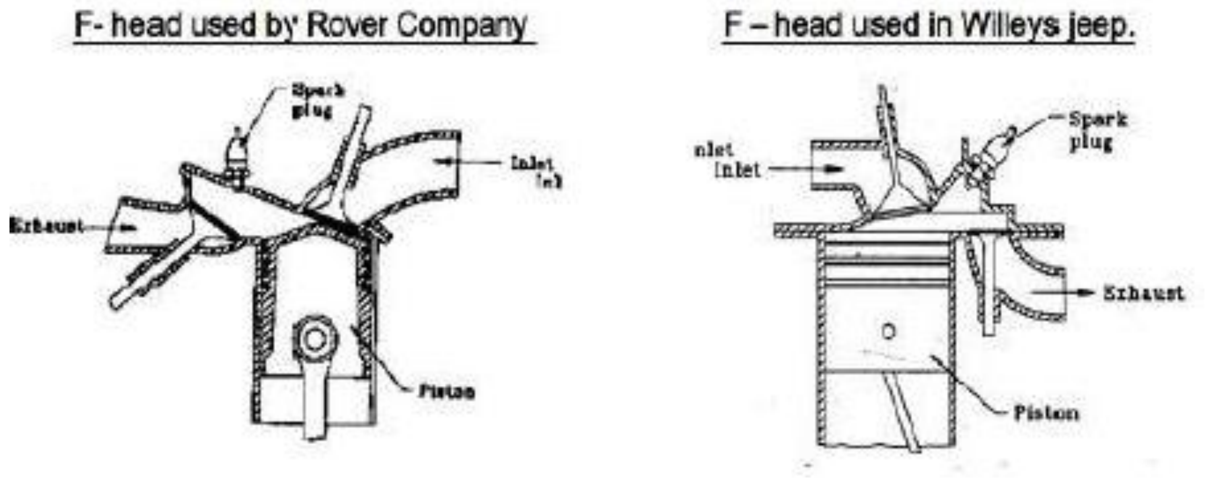


Fig. 7.11 F-Head combustion Chamber

Divided Combustion Chamber

- In this type of chambers usually with about 80 percent of the clearance volume in the main chamber above the piston and about 20 percent of the volume as a secondary chamber. Main chamber is connected to secondary chamber through a small orifice Combustion is started in the small secondary chamber.
- As the gases in secondary chambers are consumed by combustion, pressure rises and flaming gas expands back through orifice and act as torch ignition for main chamber. Secondary chamber has high swirl and designed to handle rich mixture
- The rich mixture with very high swirl in secondary chamber will ignite readily and burn very quickly. The flame gas expands through orifice and ignites the lean mixture in the main chamber. The net result is an engine that has good ignition and combustion and yet operates mostly lean to give good fuel economy.

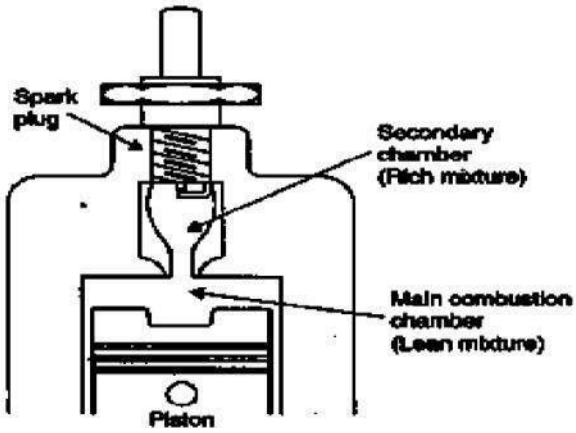
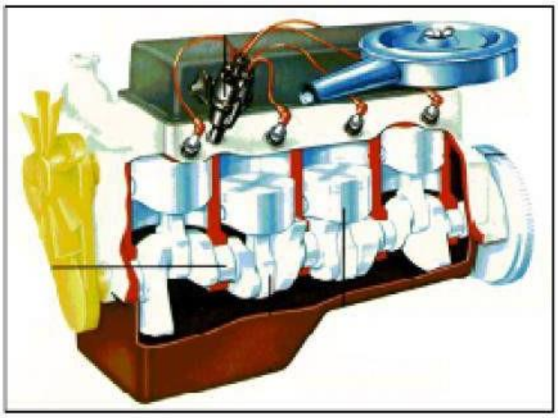


Fig. 7.12 divided combustion Chamber

8

COMBUSTION IN CI ENGINE



Course Contents

STAGES OF COMBUSTION IN CI ENGINE

- 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 stoichiometric 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, whereas 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.

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

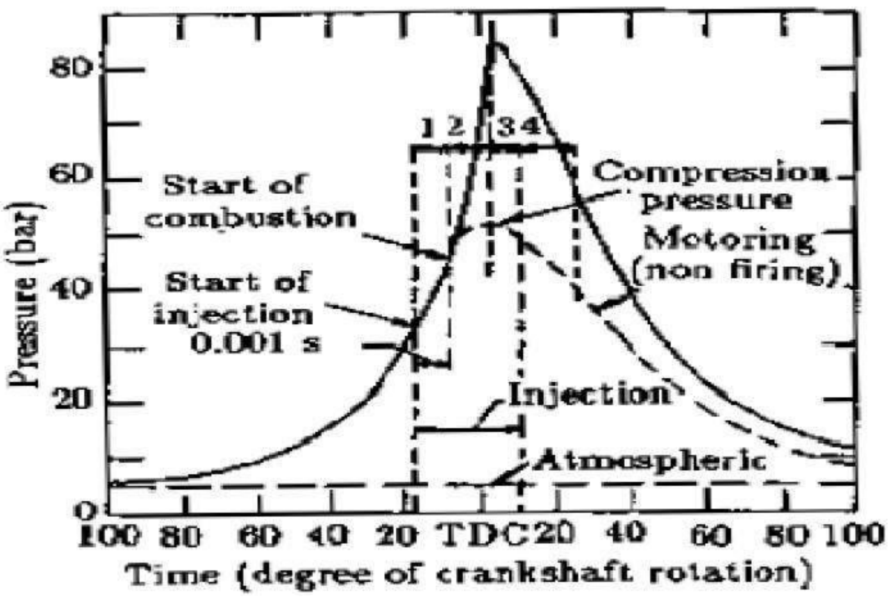


Fig. 8.1 Stage of Combustions

1. 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

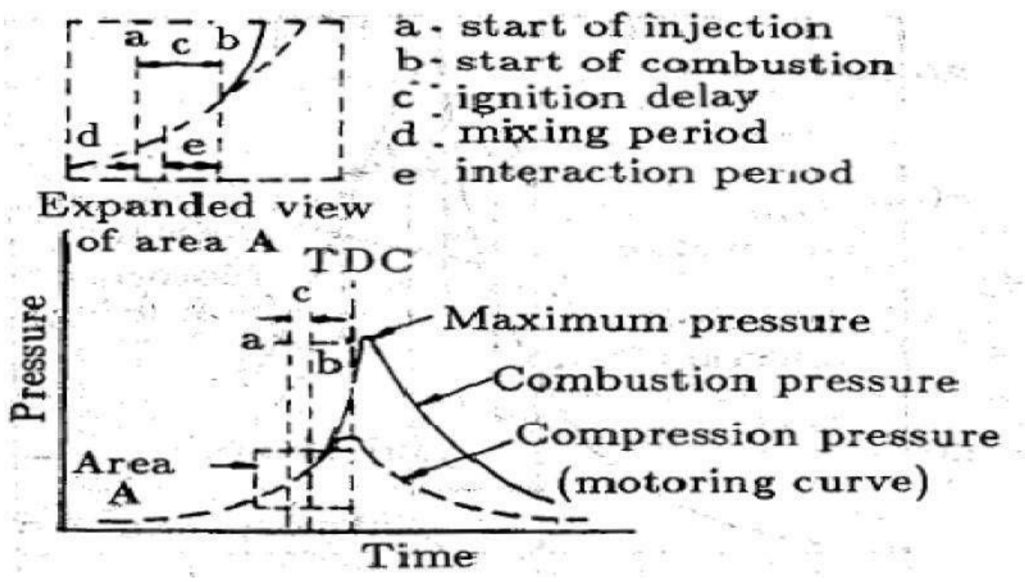


Fig. 8.2 Pressure Time diagram illustrating Ignition delay

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.

delay period /ignition lag and the factors affecting it

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

Table 8.1 Combustion phenomenon in CI engine V/s combustion in SI engine

Sr NO	COMUSTION IN SI ENGINE	COMBUSTION IN CI ENGINE
1	Homogeneous mixture of petrol vapor and air is compressed (CR 6:1to 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 incombustible 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
3	For effective combustion, turbulence is required. Turbulence which is required in SI engine implies disordered air motion with no general direction off low 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 orderly movements 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 allow 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 tostoichiometric 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.
7	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

EFFECT OF VARIOUS FACTORS ON DELAY PERIOD IN CI ENGINE

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

- compression ratio
- ngine speed

-output
injection timing q
uality 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).

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.
- 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.

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. 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.

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

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

8.2 EFFECT OF VARIABLE ON DELAY PERIOD – *SUMMARY*

Sr 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

- 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

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/Liter and 7.7 gm/Liter. But these two increase the NOx emissions
- **Increasing Swirl**: Knocking can be greatly reduced by increasing swirl (or reducing turbulence). Swirl helps in knock free combustion.

Combustion chambers for C.I. engines.

Primary Considerations in the Design of Combustion Chambers for C.I Engines

- In C engines fuel is injected into the combustion chamber at about 15°C before .D.C.during the compression stroke. For the best efficiency the combustion must complete within 15° to 20° of crank rotation after T.D.C. in the working stroke. Thus it is clear that injection and combustion both must complete in the short time. For best combustion mixing should be completed in the short time.
- In S.I engine mixing takes place in carburetor; however in C.I engines this has to be done in the combustion chamber. To achieve this requirement in a short period is an extremely difficult job particularly in high speed Cl. engines.
- From combustion phenomenon of C.I. engines it is evident that fuel-air contact must be limited during the delay period in order to limit, the rate of pressure rise in the second stage of combustion. This result can be obtained by shortening the delay to achieve high efficiency and power the combustion must be completed when piston is nearer to T.D.C., it is necessary to have rapid mixing of fuel and air dun the third stage of combustion.
- The design of combustion chamber for C.I. engines must also take consideration of injection system and nozzles to be used.

The considerations can be summarized as follows:

1. High thermal efficiency.
2. Ability to use less expensive fuel (multi-fuel).
3. Ease of starting.
4. Ability to handle variations in speed.
5. Smoothness of operation i.e. avoidance of diesel knock and noise.
6. Low exhaust emission.
7. Nozzle design.

8. High volumetric efficiency.
9. High brake Mean effective pressure.

Role of air swirls in Diesel engine

Most important function of CI engine combustion chamber is to provide proper mixing of fuel and air in short possible time. For this purpose an organized air movement called air swirl is to be produced to produce high relative velocity between the fuel droplets and air.

There are three basic methods of generating swirl in CI engine Combustion Chamber.

- By directing the flow of air during its entry to the cylinder known as Induction swirl. This method is used in open combustion chamber.
- By forcing air through a tangential passage into a separate swirl chamber during the compression stroke, known as Combustion swirl. This is used in swirl chamber.
- By use of initial pressure rise due to partial combustion to create swirl and turbulence, known as combustion induced swirl. This method is used in pre combustion chamber and air cell chambers.

INDUCTION SWIRL

Swirl refers to a rotational flow within the cylinder about its axes. In a four stroke engine induction swirl can be obtained either by careful formation of air intake passages or masking or shrouding a portion of circumference of inlet valve. The angle of mask is from 90° to 140° of the circumference. In two stroke engine, induction swirl is created by suitable inlet port forms. Induction swirl can be generated using following methods.

- Swirl is generated by constructing the intake system to give a tangential component to intake flow as it enters the cylinder. This is done by shaping and contouring the intake manifold,
- Swirl can be generated by masking one side of the inlet valve so that air is admitted only around a part of the periphery of the valve and in the desired direction.
- Swirl can also be generated by casting a lip over one side of the inlet valve.

Swirl generated by induction is very weak. Thus single orifice injection cannot provide the desired air fuel mixing. Therefore, with Induction swirl, it is advisable to use a multiple-orifice injector.

Advantages of Induction swirl.

- Easier starting (due to low intensity of swirl).
- High excess air (low temperature), low turbulence (less heat loss), therefore indicated thermal efficiency is high.

- Production of swirl requires no additional work.
- Used with low speeds, therefore low quality of fuel can be used.

Disadvantages of induction swirl:

- Shrouded valves, smaller valves, low volumetric efficiency.
- Weak swirl, low air utilization (60%), lower M.E.P. and large size (costly) engine.
- Weak swirl, multi-orifice nozzle, high induction pressure; clogging of holes, high maintenance.
- Swirl not proportional to speed; efficiency not maintained at variable speed engine.
- Influence minimum quantity of fuel. Complication at high loads and idling.

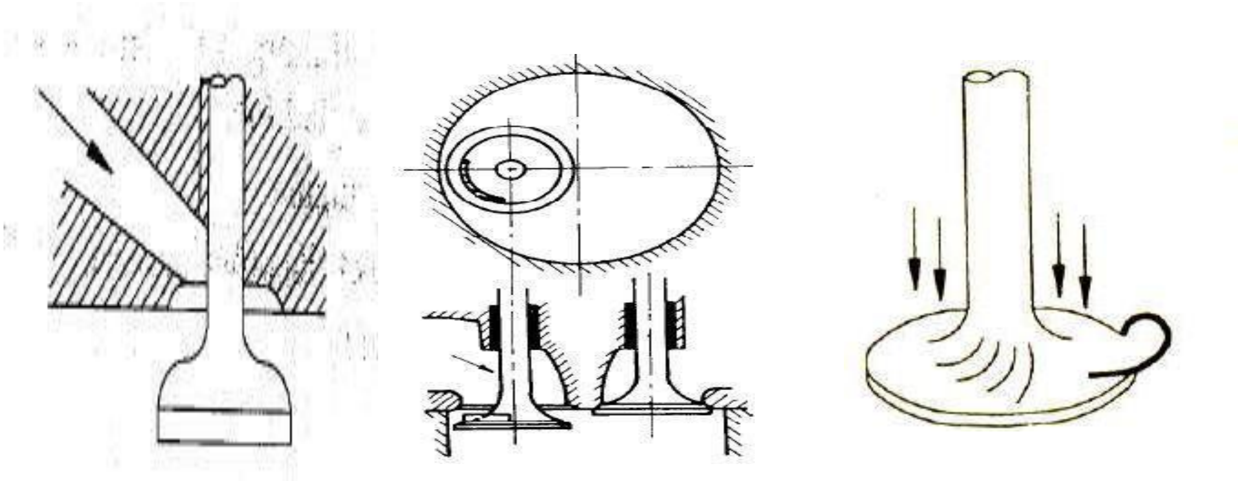


Fig. 8.3 Induction Swirl Process

COMPRESSION SWIRL

Compression swirl is generated using swirl chamber. A swirl chamber is a divided chamber. A divided combustion chamber is defined as one in which the combustion space is divided into two or more compartments. Pressure difference between these chambers is created by restrictions or throats. Very strong swirl can be generated using compression swirl.

Advantage of compression swirl:

- Large valves, high volumetric efficiency.
- Single injector, pintle type (self-cleaning), less maintenance.
- Smooth engine operation.
- Greater air utilization due to strong swirl. Smaller (cheaper) engine.
- Swirl proportional to speed, suitable for variable speed operation.

Disadvantage of compression swirl:

- Cold starting trouble due to high loss due to strong swirl, mechanical efficiency I
- Less excess air ; lower indicated efficiency; 5 to 8% more fuel consumption;
- decreased exhaust valve life
- Cylinder more expensive in construction.
- Work absorbed in producing swirl, mechanical efficiency lower.

COMBUSTION INDUCTION SWIRL

- This is created due to partial combustion, therefore known as combustion induced swirl. This method is used only in pre-combustion chamber. In this method, upward movement of piston during compression forces the air tangentially and fuel is injected in pre-combustion, when the combustion of the accumulated fuel during delay periods burns rapidly in pre-combustion chamber as A: F mixture is rich and forces the gases at a very high velocity in the main combustion chamber. This creates a good swirl and provides better combustion.

The requirement of air motion and swirl in a C.I engine combustion chamber is much more stringent than in an S.I. engine.-Justify this statement?

- Air motions are required in both S.I. and C.I. engines. In S.I. engine, we call it turbulence and in C.I. engine, we call it swirl. Turbulence which is required in S.I. engines implies disordered air motion with no general direction of flow, to break up the surface of flame front and to distribute flame throughout an externally prepared combustible mixture. Air swirl which is required in C.I. engines is 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 droplet and sweep away the products of combustion which otherwise would suffocate it.
- If there is no turbulence in S.I. engines, the time occupied by each explosion would be so great as to make high speed internal combustion engines impracticable. Insufficient turbulence lowers the efficiency due to incomplete combustion of fuel. In case of C.I. engines, it is impossible to inject fuel droplets so that they distribute uniformly throughout the combustion space, the fuel air mixture formed in combustion chamber is essentially heterogeneous. Under these conditions, if the air within the cylinder were motionless, only a small portion of fuel would find sufficient oxygen and even burning of this fuel would be slow or even choked. So it is essential to impart swirl to air so that a continuous supply of fresh air is brought to each burning droplet and the products of combustion are swept away.

The induction swirl in a C.I. engine helps in increasing indicated thermal efficiency.
Justify this statement.

- Induction swirl is used in direct injection type engines, where the entire combustion space is directly above the piston, and hence the surface-to-volume ratio of the combustion chamber is low. Further, the compressed air and the combustion products do not have to pass through a neck narrow connecting passage. Also, the mean combustion temperatures are lower, and there is less turbulence. All these factors result in less heat losses, and thus the indicated thermal efficiency is increased.

TYPES OF COMBUSTION CHAMBERS- CI Engines

- The most important function of CI engine combustion chamber is to provide proper mixing of fuel and air in short time. In order to achieve this, an organized air movement called swirl is provided to produce high relative velocity between the fuel droplets and the air.
- When the liquid fuel is injected into combustion chamber, the spray cone gets disturbed due to air motion and turbulence inside. The onset of combustion will cause an added turbulence that can be guide d by the shape of the combustion chamber, makes it necessary to study the combustion design in detail.

C I engine combustion chambers are classified into two categories:

- OPEN INJECTION (DI) TYPE: This type of combustion chamber is also called an Open combustion chamber. In this type the entire volume of combustion chamber is located in the main cylinder and the fuel is injected into this volume.
- INDIRECT INJECTION (IDI) TYPE: in this type of combustion chambers, the combustion space is divided into two parts, one part in the main cylinder and the other part in the cylinder head. The fuel –injection is affected usually into the part of chamber located in the cylinder head. These chambers are classified further into :

a) Swirl chamber in which compression swirl is generated
b) Pre combustion chamber in which combustion swirl is induced
c) Air cell in which both compression and combustion swirl are induced.

DIRECT INJECTION CHAMBERS – OPEN COMBUSTION CHAMBERS

- An open combustion chamber is defined as one in which the combustion space is essentially a single cavity with little restriction from one part of the chamber to the other and hence with no large difference in pressure between parts of the chamber during the combustion process. There are many designs of open chamber some of which are shown below :
- In four-stroke engines with open combustion chambers, induction swirl is obtained either by careful formation of the air intake passages or by masking a portion of the circumference of the inlet valve whereas in two-stroke engines it is created by suitable form for the inlet ports.

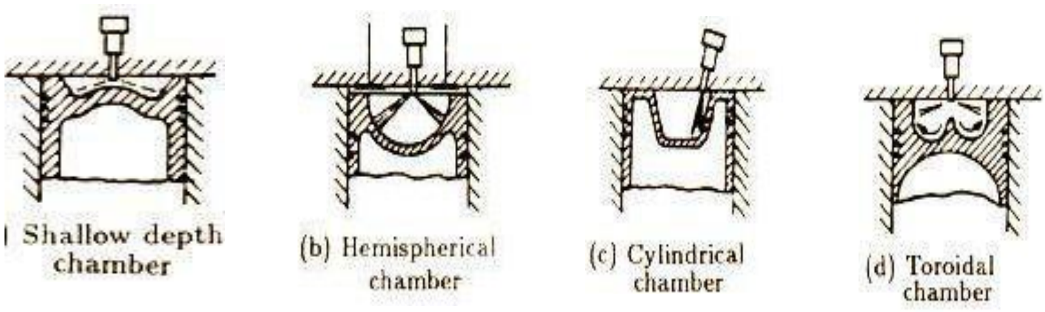


Fig. 8.4 Direct Injection Type Chamber

- These chambers mainly consist of space formed between a flat cylinder head and a cavity in the piston crown in different shapes. The fuel is injected directly into space. The injection nozzles used for this chamber are generally of multi whole type working at a relatively high pressure (about 200 bars).

The main advantages of this type of chambers are:

- Minimum heat loss during compression because of lower surface area to volume ratio and hence, better efficiency.
- No cold starting problems.
- Fine atomization because of multi hole nozzle.
- The drawbacks of these combustion chambers are:
- High fuel-injection pressure required and hence complex design of fuel injection pump.
- Necessity of accurate metering of fuel by the injection system, particularly for small engines.

- **Shallow Depth Chamber**: In shallow depth chamber the depth of the cavity provided in the piston is quite small. This chamber is usually adopted for large engines running at low speeds. Since the cavity diameter is very large, the squish is negligible.
- **Hemispherical Chamber:** This chamber also gives small squish. However, the depth to diameter ratio for a cylindrical chamber can be varied to give any desired squish to give better performance.
- **Cylindrical Chamber:** This design was attempted in recent diesel engines. This is a modification of the cylindrical chamber in the form of a truncated cone with base angle of 30°. The swirl was produced by masking the valve for nearly 1800 of circumference. Squish can also be varied by varying the depth.
- **Toroidal Chamber:** The idea behind this shape is to provide a powerful squish along with the air movement, similar to that of the familiar smoke ring, within the Toroidal chamber. Due to powerful squish the mask needed on inlet valve is small and there is better utilization of oxygen. The cone angle of spray for this type of chamber is 150° to 160°.

IN DIRECT INJECTION CHAMBERS

A divided combustion chamber is defined as one in which the combustion space is divided into two or more distinct compartments connected by restricted passages. This creates considerable pressure differences between them during the combustion process.

Ricardo's Swirl Chamber:

- Swirl chamber consists of a spherical shaped chamber separated from the engine cylinder and located in the cylinder head. Into this chamber, about 50% of the air is transferred during the compression stroke.
- A throat connects the chamber to the cylinder which enters the chamber in a tangential direction so that the air coming into this chamber is given a strong rotary movement inside the swirl chamber and after combustion, the products rush back into the cylinder through same throat at much higher velocity.
- This causes considerable heat loss to walls of the passage which can be reduced by employing a heat insulated passage. This type of combustion chamber finds its application where fuel quality is difficult to control, where reliability under adverse conditions is more important than fuel economy.
- The use of single hole of larger diameter for the fuel spray nozzle is often important consideration for the choice of swirl chamber engine.

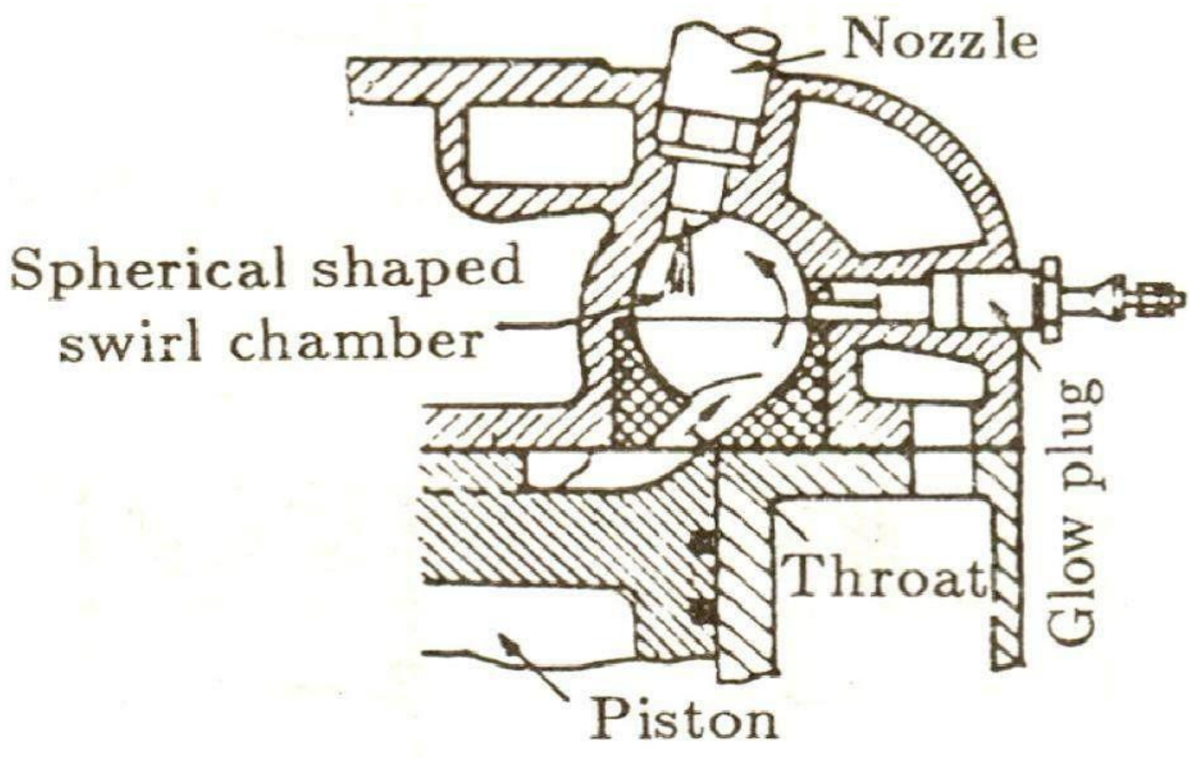


Fig. 8.5 Ricardo's Swirl Chamber

PRE COMBUSTION CHAMBER

- Typical pre-combustion chamber consists of an anti chamber connected to the main chamber through a number of small holes (compared to a relatively large passage in the swirl chamber). The pre-combustion chamber is located in the cylinder head and its volume accounts for about 40% of the total combustion, space. During the compression stroke the piston forces the air into the pre-combustion chamber.
- The fuels injected into the pre-chamber and the combustion is initiated. The resulting pressure rise forces the flaming droplets together with some air and their combustion products to rush out into the main cylinder at high velocity through the small holes. Thus it creates both strong secondary turbulence and distributes the flaming fuel droplets throughout the air in the main combustion chamber where bulk of combustion takes place.
- About 80% of energy is released in main combustion chamber. The rate of pressure rise and the maximum pressure is lower compared to those in open type chamber. The initial shock if combustion is limited to pre-combustion chamber only. The pre-combustion chamber has multi fuel capability without any modification in the injection system because the temperature of pre-chamber.
- The variation in the optimum injection timing for petrol and diesel operations is only 2 deg. for this chamber compared to 8 to 10 deg in other chamber design.

Advantages:

- Due to short or practically no delay period for the fuel entering the main combustion space, tendency to knock is minimum, and as such running is smooth.
- The combustion in the third stage is rapid.
- The fuel injection system design need not be critical. Because the mixing of fuel and air takes place in pre-chamber.

Disadvantages:

- The velocity of burning mixture is too high during the passage from pre-chambers, so the heat loss is very high. This causes reduction in the thermal efficiency, which can be offset by increasing the compression ratio.
- Cold starting will be difficult as the air loses heat to chamber walls during compression.

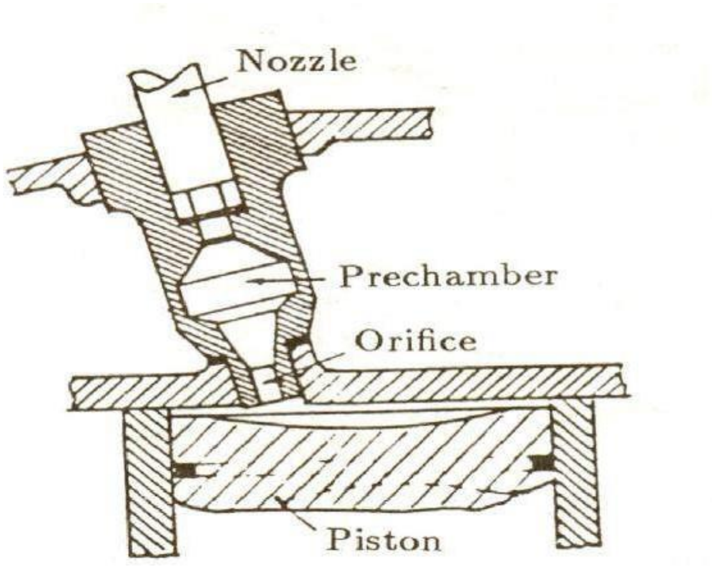


Fig. 8.6 Pre-combustion Chamber

Energy cell:

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

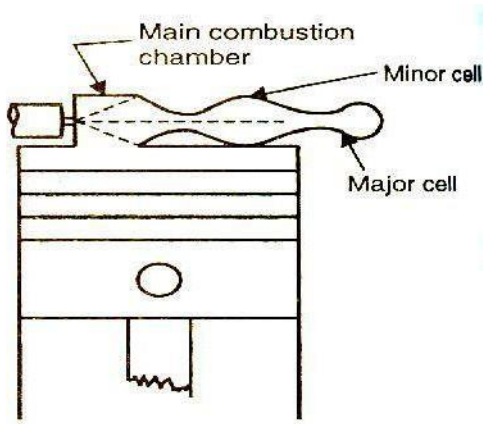


Fig.8.6 Energy cell

M COMBUSTION CHAMBER

- After twenty years of research in 1954, Dr. Meaner of M.A.N., Germany developed M process engine which ran without typical diesel combustion noise and hence it was named 'whisper engine'.
- The 'M-combustion chamber' is a special type of open combustion chamber, having combustion chamber in the piston cavity. It differs in principle from other open chamber designs in that the fuel spray impinges on and spreads over the surface of a spherical cavity in the piston.
- Earlier it had usually been assumed that fuel spray impingement was undesirable, though in most diesel engines some impingement always takes place at full load. The fuel is injected tangentially from a multi-hole nozzle on the surface of the chamber in the direction of the air swirl. Injected fuel forms a film, about 0.15 mm thick, on the surface of the chamber.
- The combustion is initiated by the auto-ignition of small portion of fuel which is air-borne at the very beginning. The amount of this airborne fuel is controlled by selecting a proper distance between the nozzle tip and the combustion chamber wall. Subsequently the fuel vapors rise from the hot wall and are mixed with the swirling air in successive layers and combustion takes place in a near homogeneous air-fuel mixture at the desired rate. The rate of energy release is thus almost equal to the rate of evaporation of fuel.
- Thus, even though the engine works on diesel cycle, once the ignition takes place, the combustion characteristics are similar to those of OTTO cycle combustion.

The advantages of 'M-chamber

- Low rates of pressure rise, low peak pressure.
- Low smoke level.
- Ability to operate on a wide range of liquid fuels (multi-fuel capability).
- No combustion noise is reported even for 80-octane petrol.

The disadvantages of M-chamber are:

- Since fuel vaporization depends upon the surface temperature of the combustion chamber, cold starting requires certain aids.
- Some white smoke, diesel odor, and high hydrocarbon emission may occur at starting and idling conditions.
- Volumetric efficiency is low.

8.3 THE FOLLOWING TABLE GIVES COMPARISON BETWEEN OPEN CHAMBER AND DIVIDED CHAMBER.

Sr No.	Aspects	Open Combustion Chamber (DIRECT INJECTION)	Divided combustion chamber (INDIRECT INJE.)
1.	Fuel Used	Can consume fuels of good Ignition quality. i.e., of shorter ignition delay or higher Cetane Number	Can consume fuels of poor ignition quality i.e. of longer ignition delay or lower Cetane Number
2.	Type of injection nozzles used	Requires multiple hole injection nozzles for proper mixing of fuel and air and also higher injection pressures	It is able to use single hole injection nozzles and moderate injection pressures It can also tolerate greater degree of nozzle fouling
3.	Sensitivity to fuel	Sensitive	Insensitive
4.	Mixing of fuel and air	Mixing of fuel and air is not coefficient and thus high fuel air ratios are not feasible without smoke	Ability to use higher fuel ratio without smoke due to proper mixing and consequent high air utilization factor
5.	Cylinder construction	Simple	More expensive cylinder Construction
6.	Starting	Easy cold starting	Difficult to cold start because of greater heat loss through throat
7.	Thermal efficienc y	Open combustion chambers are thermally more efficient	Thermal efficiency is lower duet throttling in throat areas leading to pressure losses and heat losses.

8.6.7.4 COLD STARTING OF CI ENGINES

- Easy starting from cold is a very important requirement of a CI engine. To ensure easy cold starting, frequently compression ratios higher than necessary are used. Even so, cold starting may become difficult in extreme cold climate like Himalayan region,
- when the cylinder liner is heavily worn, and
- When the valves are leaky.
- It is, therefore, sometimes necessary to provide some electrical aid for cold starting. Open chamber direct injection engines are easiest to cold start. The reasons for easy starting of open chamber direct injection engines are as follows
- They have smallest surface to volume (S/V) ratio. Because of this the loss of heat is minimum.

- They have lowest intensity of air swirl. Low intensity of swirl allows stagnant gas film to remain on cylinder walls, which reduces heat transfer.

Many methods have been used in the past to achieve easy cold starting. Few of them are described below:

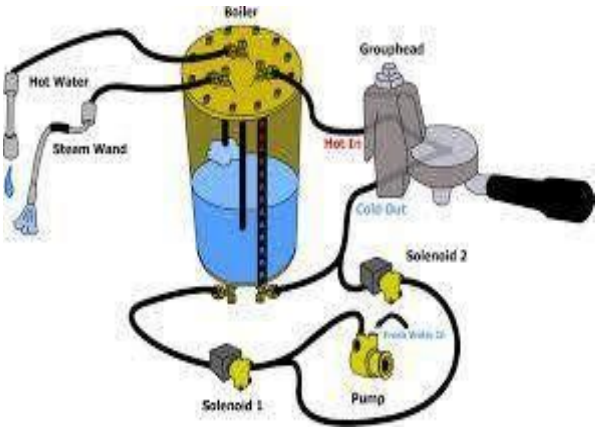
- Injection of a small quantity of lubricating oil or fuel oil. This ad hoc method helps by temporarily raising the compression ratio and sealing the piston rings and valves.
- Provision of cartridges. These may be self-igniting or requiring lighting before insertion into the combustion chamber.
- Starting as petrol engines. The engine is provided with a sparking plug and carburetor. At starting, compression ratio is reduced by providing an auxiliary chamber and the engine is started as a petrol engine.
- Preheating the engine cylinder by warm water.
- Modifying valve timings for starting.

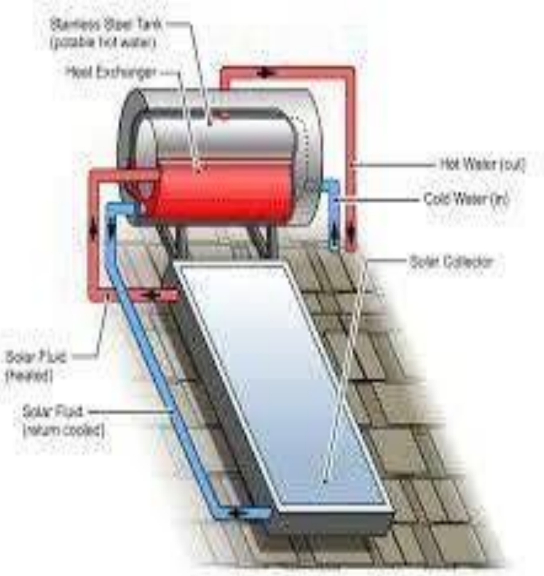
Modern starting aids of high speed engines. Basically three types of starting aids are used on modern high speed diesel engines:

- Electric glow plugs in. the combustion chamber.
- Manifold heaters which ignite a small feed of fuel.
- The injection into the intake, of controlled amounts of low- ignition temperature liquids, usually ethyl-ether with addition of other fuels.

9

ENGINE COOLING AND LUBRICATION





Course Contents

Significance:

- In a spark ignition engine, cooling must be satisfactory to avoid pre-ignition and knock. In a compression ignition engine, since a normal combustion is aided, cooling must be sufficient to allow the parts to operate properly. In short, cooling is a matter of equalization of internal temperature to prevent local overheating as well as to remove sufficient heat energy to maintain a practical overall working temperature.

Reasons for Cooling

- To promote a high volumetric efficiency.
- To ensure proper combustion, and
- To ensure mechanical operation & reliability.

Effect of Over-cooling

- The thermal efficiency is decreased due to more loss of heat carried by the coolant.
- The vaporization of the fuel is less resulting in lower combustion efficiency.
- Low temperature increases the viscosity of lubricant causing more loss due to friction.

Types of Cooling System

- Air cooling (or direct cooling) system.
- Liquid cooling (or indirect cooling) system.
- Remark: Aviation engines, motor cycle engines and scooter engines are air-cooled; while the stationery and automobile engines are liquid cooled.

Air cooling system

- Air cooled engines depend on airflow across their external surfaces of the engine cylinders to remove the necessary heat. The amount of heat dissipated depends upon:
- The area of cooling surface in contact with the air.
- Mass flow rate of air.
- Temperature difference between cylinder and air and
- Conductivity of metal.

Cooling fins in air cooled system

- The area of cooling surface is increased by forming thin fins, either integrally by machining them on the outer walls of the engine cylinder and cylinder head or by attaching separate fins to them.

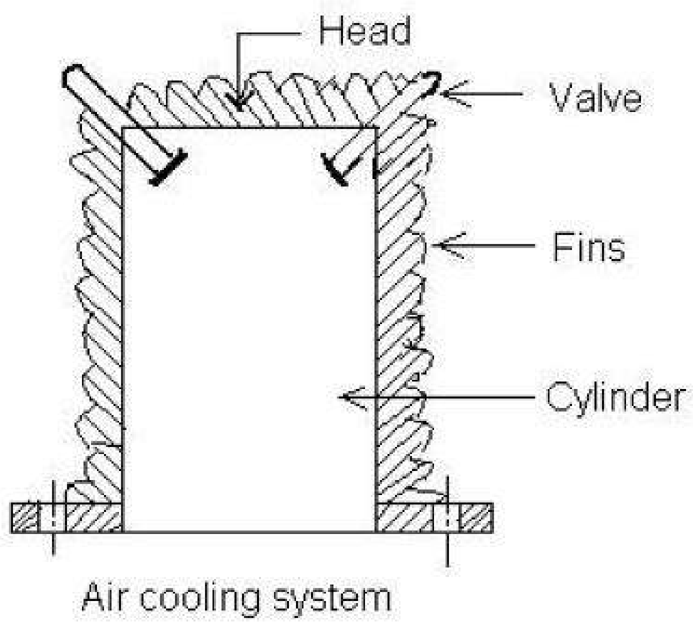


Fig. 9.1 fin air cooling system

Advantages

- The absence of radiator, cooling jackets, coolant and pumps make the engine lighter.
- The engine can be operated in cold climate where liquid may freeze.
- In places where water is scarce, air-cooled engine is an advantage.
- Handling of liquid coolant requires piping and pumping auxiliaries.
- Air cooled engines have no coolant leakage or freezing problems.

Disadvantages

- Relatively large amount of power issued to drive the cooling fan.
- Engines give low power output.
- Cooling fins under certain conditions may vibrate and amplify the noise level.
- Cooling is not uniform.
- Engines are subjected to high working temperature.

Liquid cooling systems – Types

1) Direct or non-return system
2) Thermo syphon system
3) Forced circulation cooling system
4) Evaporative cooling system

Direct or non-return system

- The heat released from the combustion of air-fuel mixture is transferred in all directions to the walls of the combustion chambers, cylinders and pistons by direct radiation, by convection

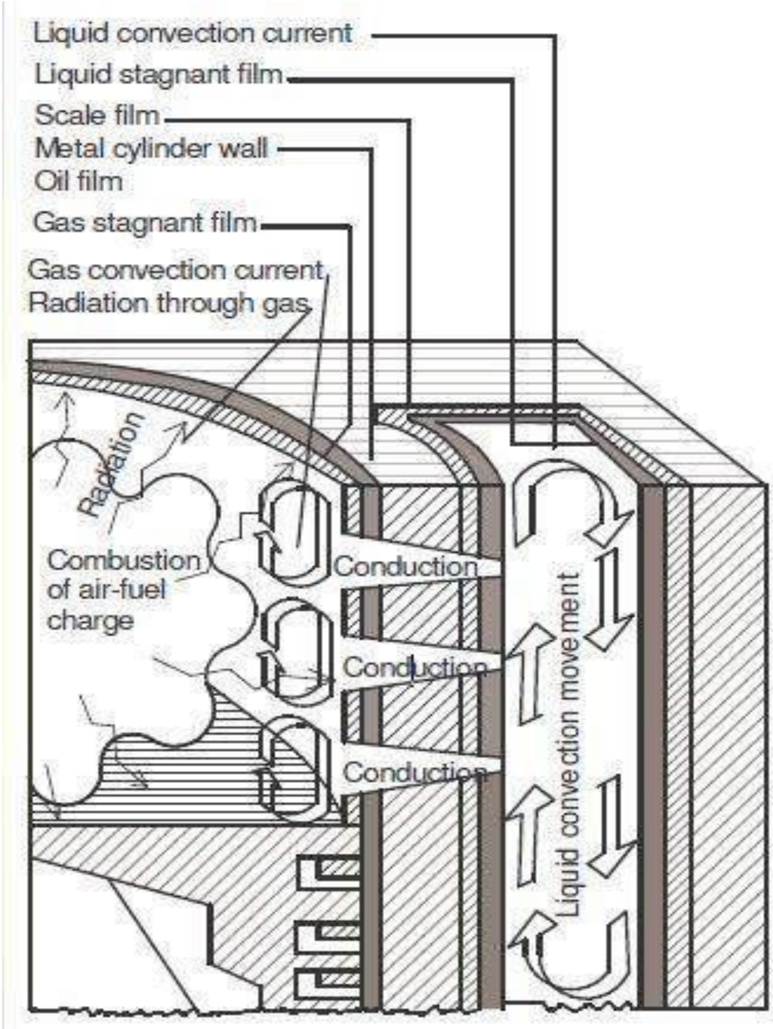


Fig. 9.2 Direct cooling system

- Current of gas rubbing against a stationary gas film, and then by conduction through this stagnant boundary layer of gas and an oil film to the metal wall.

Thermo-syphon system

- In this system, a fan rotated by the crankshaft draws cold air from outside through the radiator. The radiator is connected to the engine block by means of two pipes. The hot water passes through some thin pipes built in the radiator, where it gets cooled. Thus, the fluid circulates through the system in the form of convective currents.

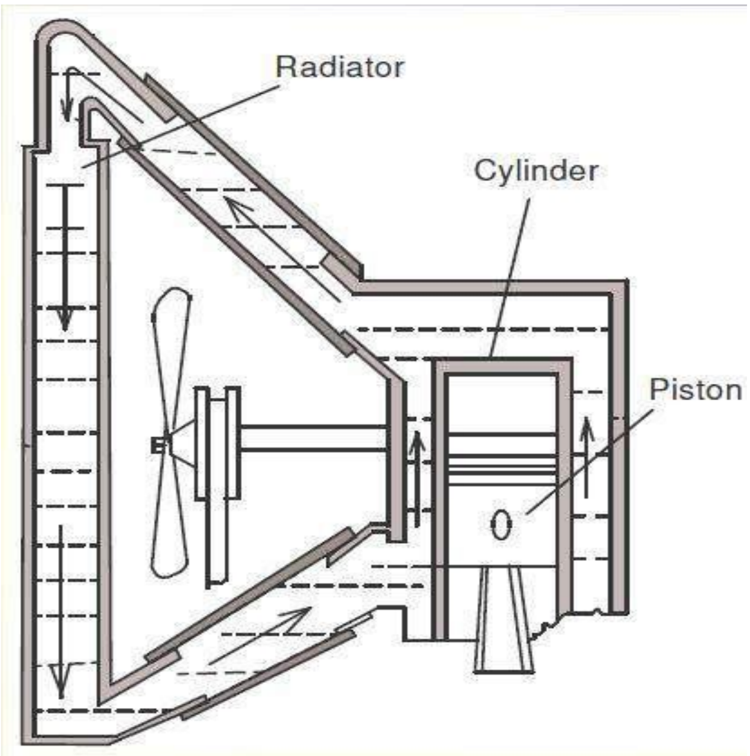


Fig.9.3 Schematic diagram of Thermosyphon system

Forced circulation cooling system

- This system is used in a large number of vehicles like cars, buses, trucks and other heavy vehicles. Here, circulation of water takes place with convection currents helped by a pump.
- The water or coolant is circulated through jackets around the parts of the engine to be cooled, and is kept in motion by a centrifugal pump, driven from the engine. A thermostat is used to control the water temperature required for cooing.

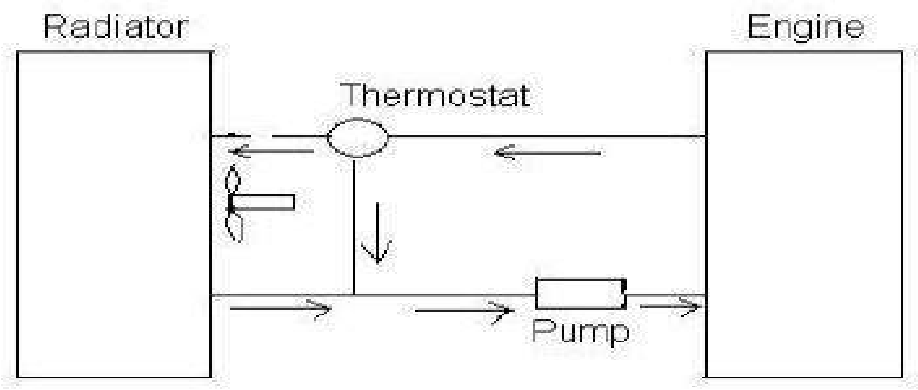


Fig.9.4 Schematic diagram of a Forced circulation system

Pump-cooling or forced cooling

- Pump is introduced between radiator and engine block.
- Rotated by crankshaft by means of a belt.
- Water is circulated with force => heat is removed quickly.

Limitation

- Cooling is independent of temp. =>Engine is overcooled (range of temp. =75-900C).
- Can be overcome by using thermostat.

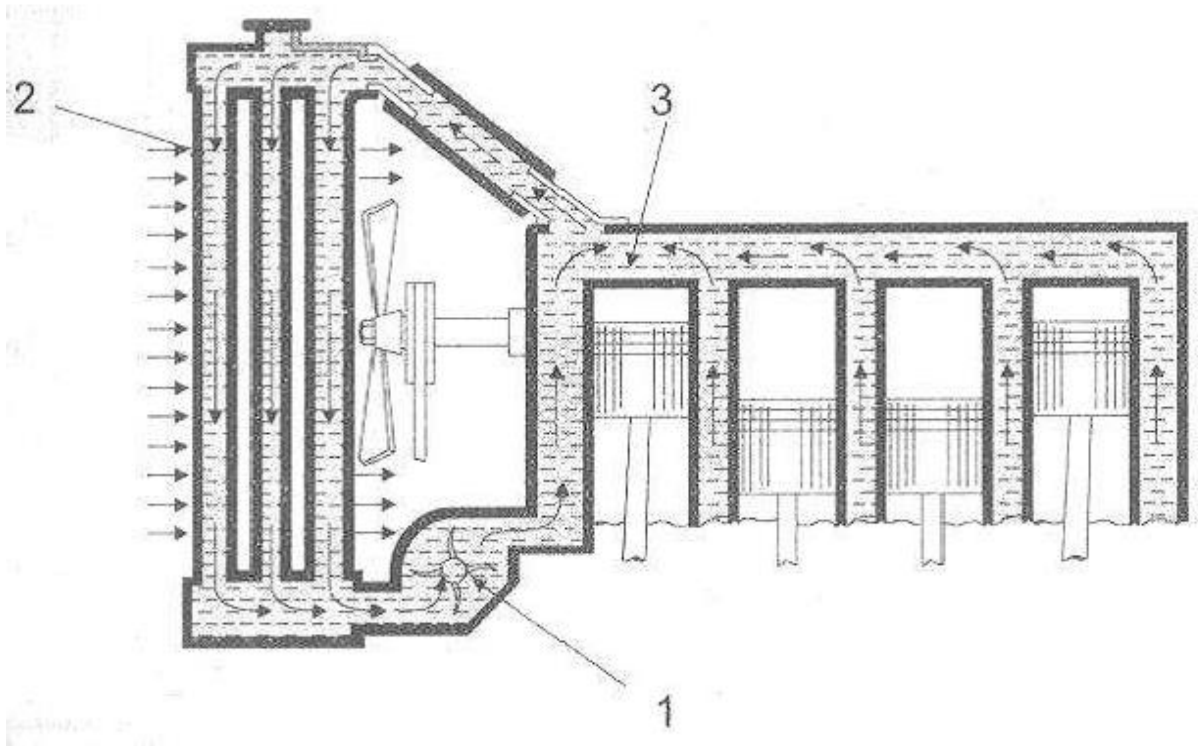


Fig.9.5 Pump Cooling System

Evaporative cooling system

- In this system, the engine will be cooled because of the evaporation of the water in the cylinder jackets into steams.
- The advantage is being taken from the high latent heat of vaporization of water by allowing it to evaporate in the cylinder jackets. This system is used for cooling of many types of industrial engines.

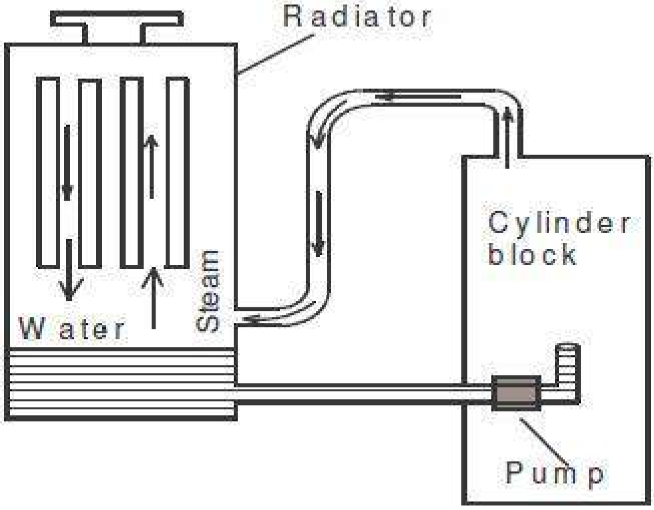


Fig.9.6 Schematic diagram of an evaporative cooling system

Parts of cooling systems – Radiator

- Rejects the coolant heat to the surrounding air
- Disperses the heated coolant into fine streams so that small quantities of coolant are brought in contact with large metal surface areas, which intern are cooled by air stream
- Two types of radiators
  1) Down-flow type
  2) Cross-flow type
- Availability of space dictates the choice of radiator, both are equally efficient.
- Various designs of radiator cores are used for cooling the water.

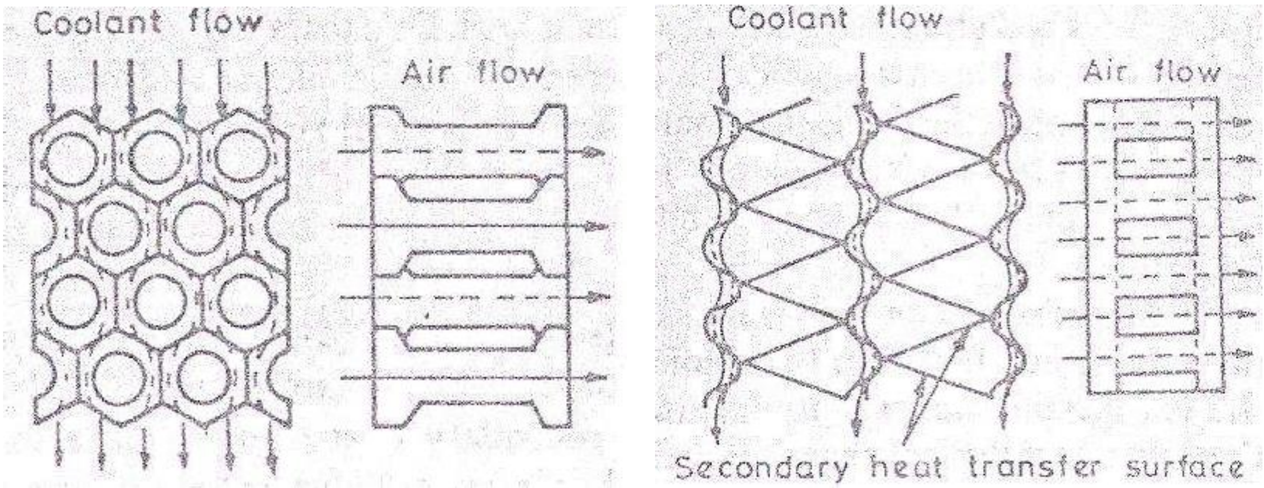
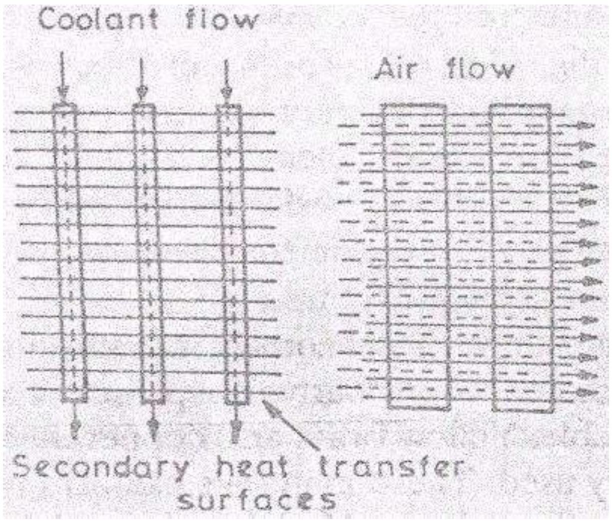


Fig.9.7 Honeycomb block type core and Film type of radiator core



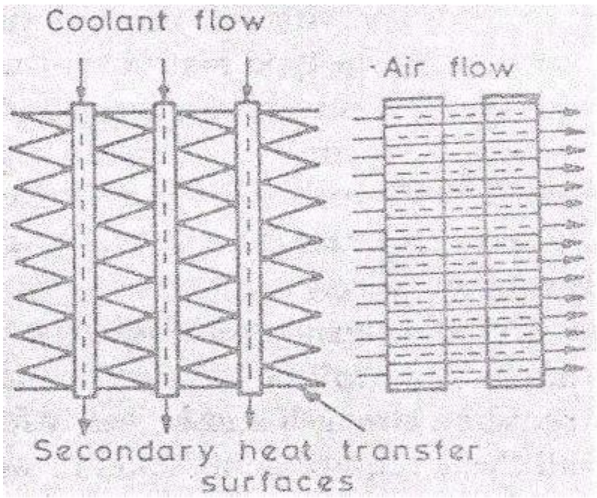


Fig.9.8 Tube and fin type core and Tube and corrugated type core

Materials for radiator

- Good corrosion resistance.
- Good thermal conductivity.
- Must possess the required strength
- Must be easily formable.
- Yellow brasses, copper are used (soldered easily => easy repair).
- Aluminum is used where weight is critical.

Wax Thermostat

- Can operate reliably within the specified temperature range.
- Heat is transmitted to wax, which has high coefficient of thermal expansion.
- Upon being heated, wax expands and the rubber plug presses the plunger forcing it to move vertically upwards.

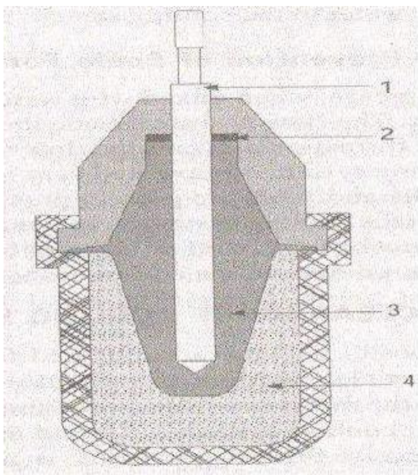


Fig.9.9 Wax Thermostat

Cooling fan

- Maintain an adequate air flow across the radiator matrix.
- Serves the purpose when natural draft is not sufficient to cool e.g., at low speed but heavy load, when vehicle ascends uphill etc.
- Driven by a belt run by crankshaft.

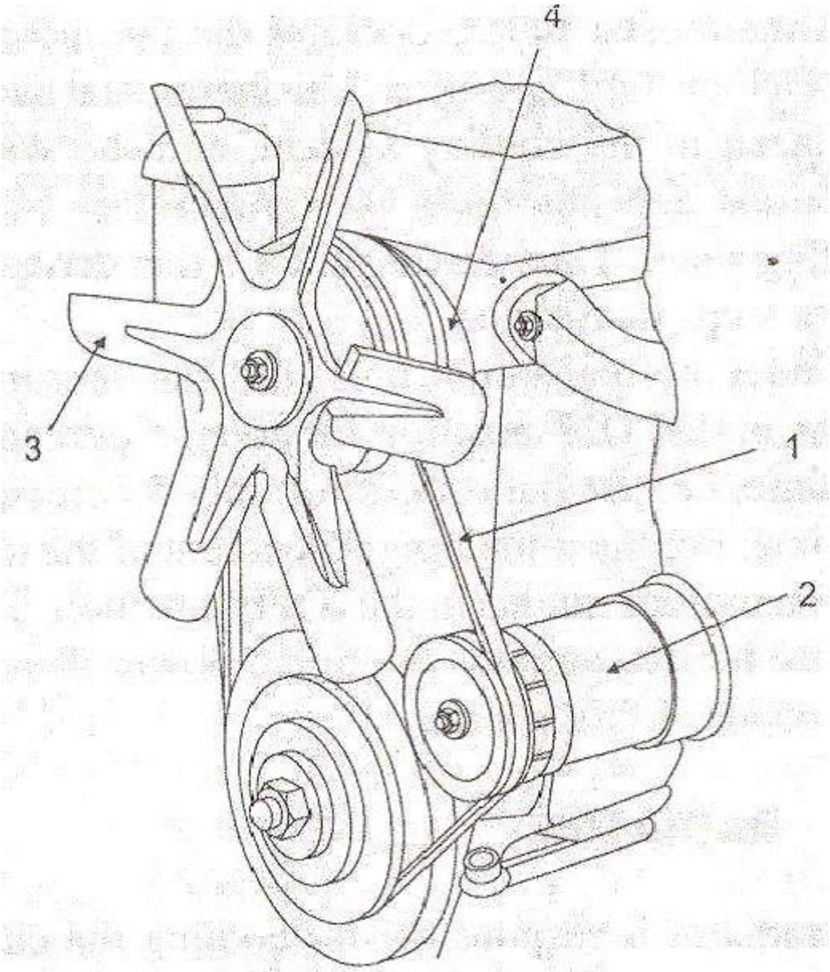


Fig. 9.10 cooling fan

Limitations

- Rising level of noise.
- Increasing power consumption with engine speed.
- Tendency to overcool.

Cooling Fan:

Advantages

- Because of even cooling of cylinder barrel and head (due to jacketing) makes it possible to reduce the cylinder head and valve seat temperatures.
- The volumetric efficiency of water cooled engines is higher than that of air- cooled engines.
- Compact design of engines with appreciably smaller frontal area is possible.
- In case of water cooled engines, installation is not necessarily at the front of the mobile vehicles, aircraft etc. as the cooling system can be conveniently located.

Disadvantages

- The system requires more maintenance.
- The engine performance becomes sensitive to climatic conditions.
- The power absorbed by the pump is considerable and affects the power output of the engine.
- In the event of failure of the cooling system serious damage may be caused to the engine.

Use of Anti-freezers

- During winter or when the engine is kept out of operation in cold places, the cooling water in the cylinder jackets, radiator tanks and leading pipes will freeze, expand and lead to their fracture.
- To prevent damage to the engine and radiator during winter weather, suitable liquids or compound substances which go into solution are added to the water to lower the freezing temperature of the coolant.
- Ethylene glycol is the most widely used automotive cooling-system antifreeze, although methanol, ethanol, isopropyl alcohol, and propylene glycol are also used.

Requirements of anti- freezers

- They should thoroughly mix with water.
- They should not corrode the surfaces with which they are in contact.
- Their boiling point should be high so that the loss due to evaporation is minimum.
- They should not deposit any foreign matter in the jackets, hose, pipes or radiator.
- It should be chemically stable, a good conductor of heat, and a poor conductor of electricity.

TABLE 9.1 PROPERTIES OF ANTIFREEZE SOLUTIONS

ETHYLENE GLYCOL–WATER MIXTURES

% ETHYLENE GLYCOL by Volume	SPECIFIC GRAVITY at 101 kPa and 15°C	FREEZING POINT at 101 kPa °C	FREEZING POINT at 101 kPa °F	BOILING POINT at 101 kPa °C	BOILING POINT at 101 kPa °F
0	1.000	0	32	100	212
10	1.014	−4	24		
20	1.029	−9	15		
30	1.043	−16	3		
40	1.056	−25	−14		
50	1.070	−38	−37	111	231
60	1.081	−53	−64		
100	1.119	−11	12	197	386

	ENTHALPY OF VAPORIZATION (kJ/kg)	SPECIFIC HEAT (kJ/kg-K)	THERMAL CONDUCTIVITY (W/m-K)
Water	2202	4.25	0.69
Ethylene Glycol	848	2.38	0.30
Ethylene Glycol–Water Mixture (50/50)	1885	3.74	0.47
Propylene Glycol	1823	3.10	0.15
Propylene Glycol–Water Mixture (50/50)		3.74	0.37

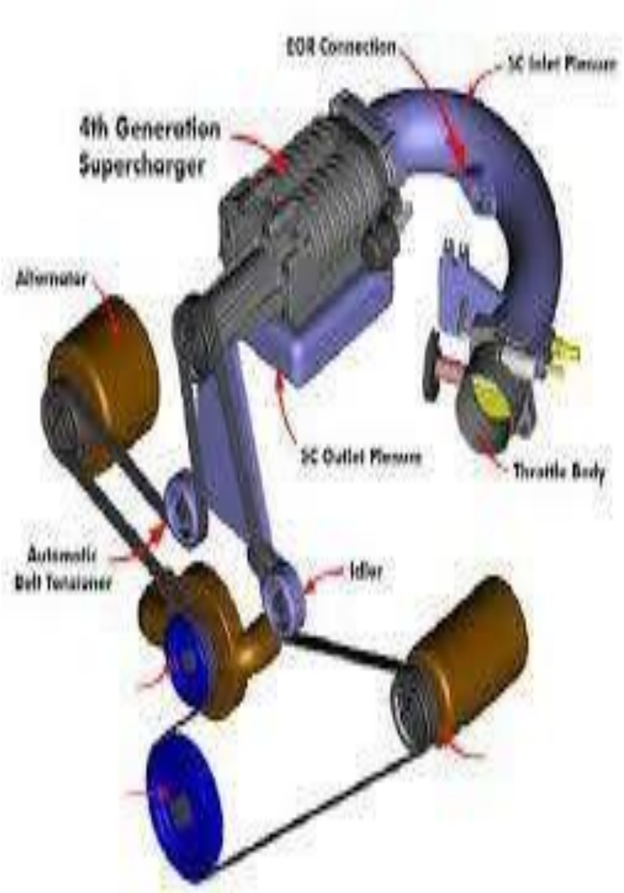
PROPYLENE GLYCOL–WATER MIXTURES					
% PROPYLENE GLYCOL by Volume	SPECIFIC GRAVITY at 101 kPa and 15°C	FREEZING POINT at 101 kPa		BOILING POINT at 101 kPa	
		°C	°F	°C	°F
0	1.000	0	32	100	212
10	1.006	−2	28		
20	1.017	−7	19		
30	1.024	−13	8		
40	1.032	−21	−6		
50	1.040	−33	−28	108	225
60	1.048	−48	55		
100	1.080	−14	6	188	370

Lubricating Oil as Coolant

- The lubricating oil used in an engine also helps to cool the engine. The hotter parts like piston face and back surface of piston crown is subjected to oil flow, usually done by spraying the oil by pressurized systems or by splash in non-pressurized system. Other components like camshaft and connecting rods are also cooled by oil circulation through oil passages.

10

TESTING OF IC ENGINE



Course Contents

Introduction

- Important performance parameters of i.c.engines
- Measurement of performance parameters in a laboratory
- Measurement of indicated power
- Measurement of b.p
- Hydraulic dynamometer
- Measurement of i.p of multi-cylinder Engine (morse test)
- Measurement of air-consumption
- Measurement of fuel consumption
- Measurement of heat carried away by Cooling water
- Measurement of heat carried away by Exhaust gases

Heat balance sheet

- Indicated specific fuel consumption:

Brake specific fuel consumption:

- Mechanical efficiency

Volumetric efficiency

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.

Important Performance Parameters of I.C.Engines

The important performance parameters of I.C. engines are as follows:

a. Friction Power
b. Indicated Power
c. Brake Power
d. Specific Fuel Consumption
e. Air – Fuel ratio
f. Thermal Efficiency
g. Mechanical Efficiency
h. Volumetric Efficiency
i. Exhaust gas emissions
j. Noise

Measurement of Performance Parameters in a Laboratory

Measurement of Friction Power

- Friction power includes the frictional lossesand 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:

a. Willan's line method;
b. From the measurement of indicated power and brake power;
c. Motoring test;
d. Retardation test;
e. Morse Test.

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 that speed.

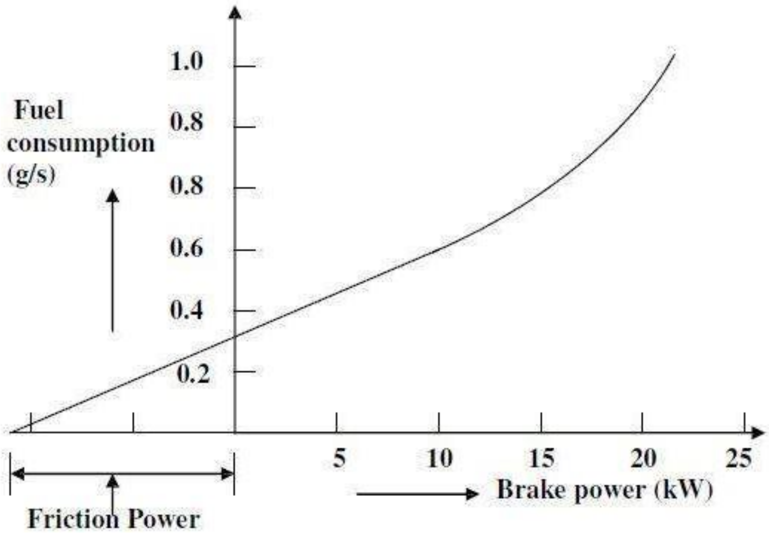


Fig.10.1 Willan's line method

- 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 drawback 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.

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.

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 off 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 = ip1 + ip2 + ip3 + ...+ ipk= $\sum_{j=1}^{k} iP_j$

- We can write $\sum_{j=1}^{k} iP_j = Bt + Ft$ (1)

  - Where ip is the indicated power produced by j the cylinder, k is the number of cylinders, BT is the total brake power when all the cylinders are producing power and Ft 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 $\sum_{j=2}^{k} iP_j = B1 - Ft$ (2)

- Where,

  B1 = total brake power when cylinder 1 is cut - off and
  Ft = Total frictional power.

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

  Thus $ip1 = Bt – B1$ (3)

- Similarly we can find the indicated power of all the cylinders, viz., ip2, ip3, …….., ipk. Then the total indicated power is calculated as

  $$(Ip)_{total} = \sum_{j=1}^{k} iP_j \quad (4)$$

- The frictional power of the engine is therefore given by

  $$Ft = (ip)_{total} – Bt \quad (5)$$

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

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.
- 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 positive area is given by

  $h = (A_p - A_n)/L$ in centimeter

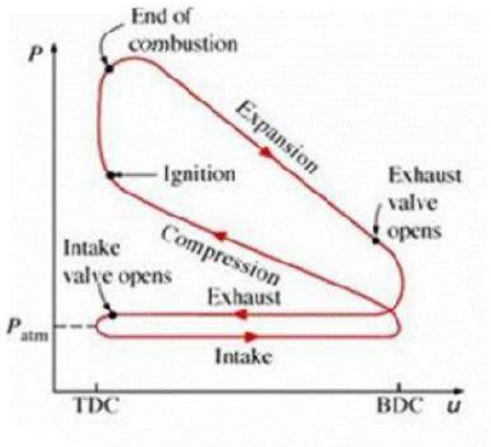


Fig.10.2 p-v diagram taken by mechanical indicator

- 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

$$P_{m=}A_p*S_p/L-A_n*S_n/L \qquad (7)$$

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

$$IP=P_mLAn/L \qquad (8)$$

Where 'n' is the number of working strokes per second.
The explanation of this expression is already given in the last chapter.

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.

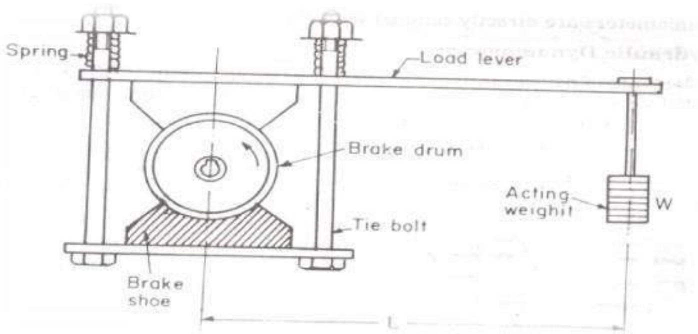


Fig.10.3 prony brake dynamometer

- 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

$$\text{B.P (brake power)} = \frac{2\pi NT}{60} \text{watts} = \frac{2\pi NT}{60\, X\, 1000}\ \text{Kw} \quad (9)$$

Where,

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

W = Weight on load carrier, (N)
L = Distance from the center 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.

Hydraulic Dynamometer.

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

$$\text{B.P (brake power)} = \frac{2\pi NWR}{60 \, X \, 1000} = WN \left(\frac{2\pi R}{60{,}000}\right) \text{Kw}$$

- The working of a prony brake dynamometer is shown in figure .

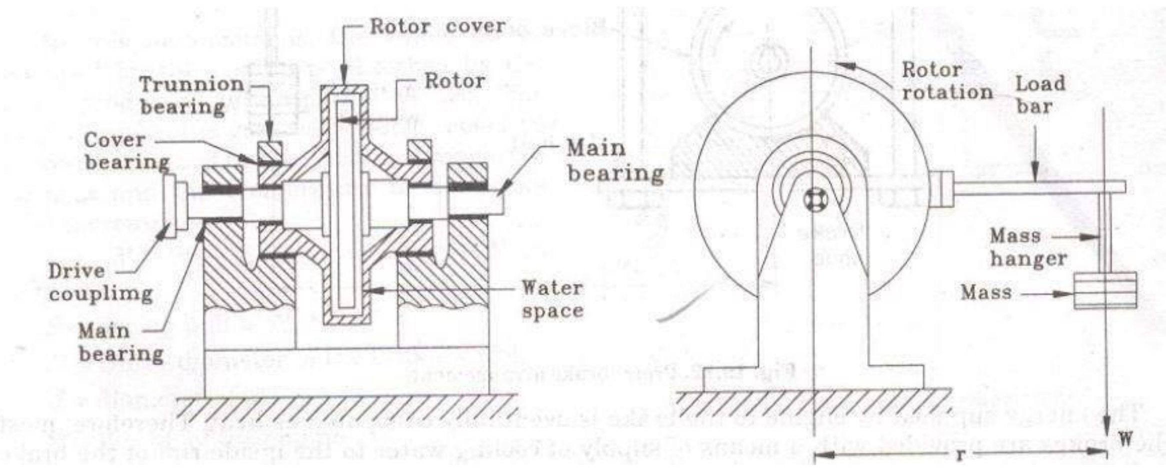


Fig.10.4 Hydraulic dynamometer

- 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 ake or dynamometer constant. Then the BP measured byte dynamometer is given

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

Where,
W = Weight measured on the dynamometer, N
K = Dynamometer constant (60*1000/2*pi*R)
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.

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 of the torque exerted on the armature, which is driven by the engine which is shown in Figure.

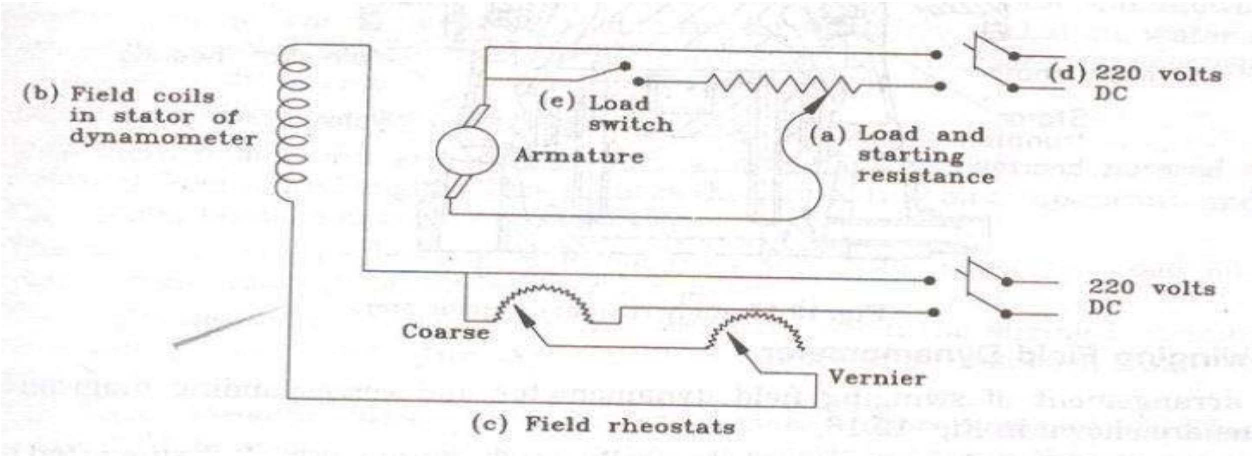


Fig.10.5 Electrical dynamometer

- The electric dynamometer can also use 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.

Eddy current Type Dynamometer

- 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 shows the “Heenan eddy-current dynamometer”.

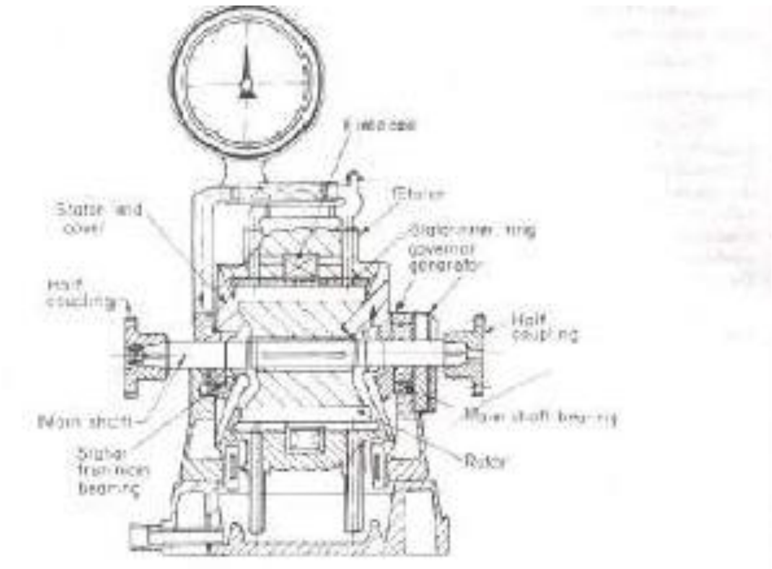


Fig.10.6 Eddy Current dynamometer

- 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.

Swinging Field Dynamometer

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

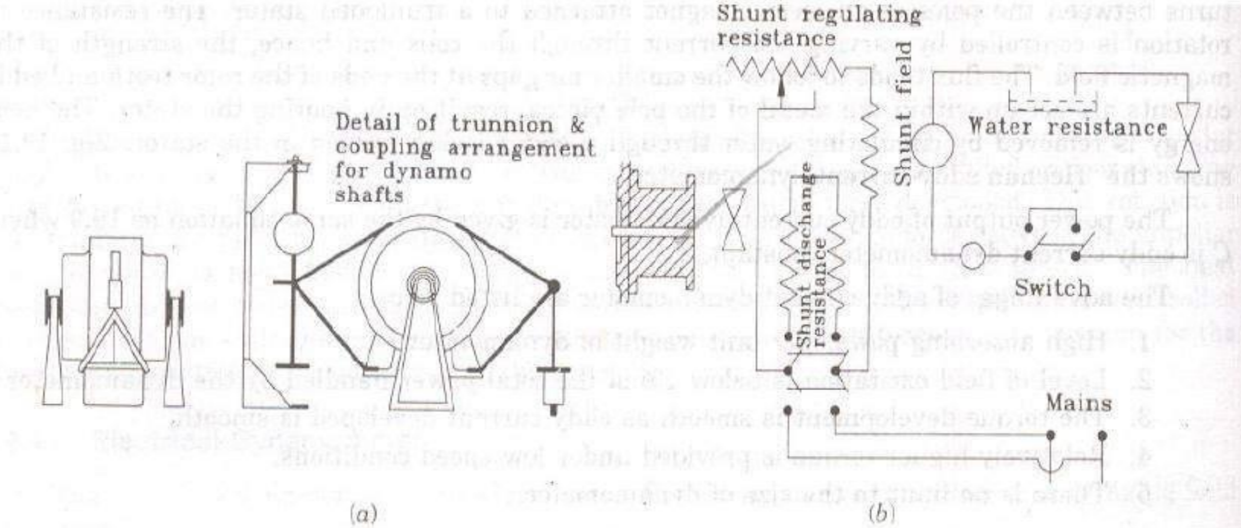


Fig.10.7 Swinging Field dynamometer

- 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 dynamometer and engine.

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

- This method is used in multi-cylinder engines to measure I.P without 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

$$(B.P)_n = (I.P)_n - (F.P)_n \qquad (11)$$

- 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

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

- 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 multi-cylinder engine is known as 'Morse Test'.

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. r$_a$= Density of air in kg/m$_3$ under atmospheric conditions.
- Head in terms of meters of air is given by

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

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

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

- The volume of air passing through the orifice is given by

$$v = A_0 v C_d\sqrt{2g\frac{10h_w}{\rho_a}} = 14.01.A_0.C_d\sqrt{\frac{h_w}{\rho_a}}cu.m/sec$$

- The volumetric efficiency of the engine = Actual volume of air taken in as measure

$$= \frac{14.01A_0, C_d\sqrt{\frac{h_w}{\rho_a}}}{\frac{\pi D^2}{4}.L\frac{N}{60}n}$$

- 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 give by

$$m_a = V_a.\rho_a = 14.01 \times \frac{\pi d^3}{4 \times 100^2}.C_d\sqrt{\frac{h_w}{\rho_a}}.\rho_a = 11.003 \times 10^{-4}C_d.d^2\sqrt{\rho_a h_w}$$

- Where d is in cm; h_w is in cm of water and ρ_a is in kg/m3
- 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)

$$m_a = 0.066 \cdot C_d \cdot d^2\sqrt{h_w\frac{p_a\times 10^5}{287\times T_a}} = 1.23 \cdot C_d \cdot d^2\sqrt{\frac{p_a\times h_w}{T_a}}\ Kg/min$$

- 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.

$$\frac{m_a}{kg} of\ fuel = \frac{N \times C}{33(C_1 + C_2)}$$

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;

$$m_a = \frac{N \times C}{33(C_1)}$$

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 tube measured, the valve is closed so that fuel is consumed from the burette. The time for known value of fuel consumption can be measured and fuel consumption rate can be calculated.
- Fuel consumption

$$\frac{kg}{hr} = \frac{X_{cc} \times Sp.gravity\ of\ fuel}{1000 \times t}$$

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

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

Where,

M_w = mass of water/min.

T_{wi} = Inlet temperature of water, C

T_{wo} = Out let temperature of water, C

C_p = Specific heat of water

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

$$m_a = \frac{N \times C}{33(C_1 + C_2)}$$

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 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 C.

T_a = Ambient temperature C 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.

9.10.1 Exhaust 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 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, C

T_{go}= The temperature of the exhaust gases leaving the calorimeter, C

T_{wi}= The temperature of water entering the calorimeter, C

T_{wo} = The temperature of water leaving the calorimeter, C

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.

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

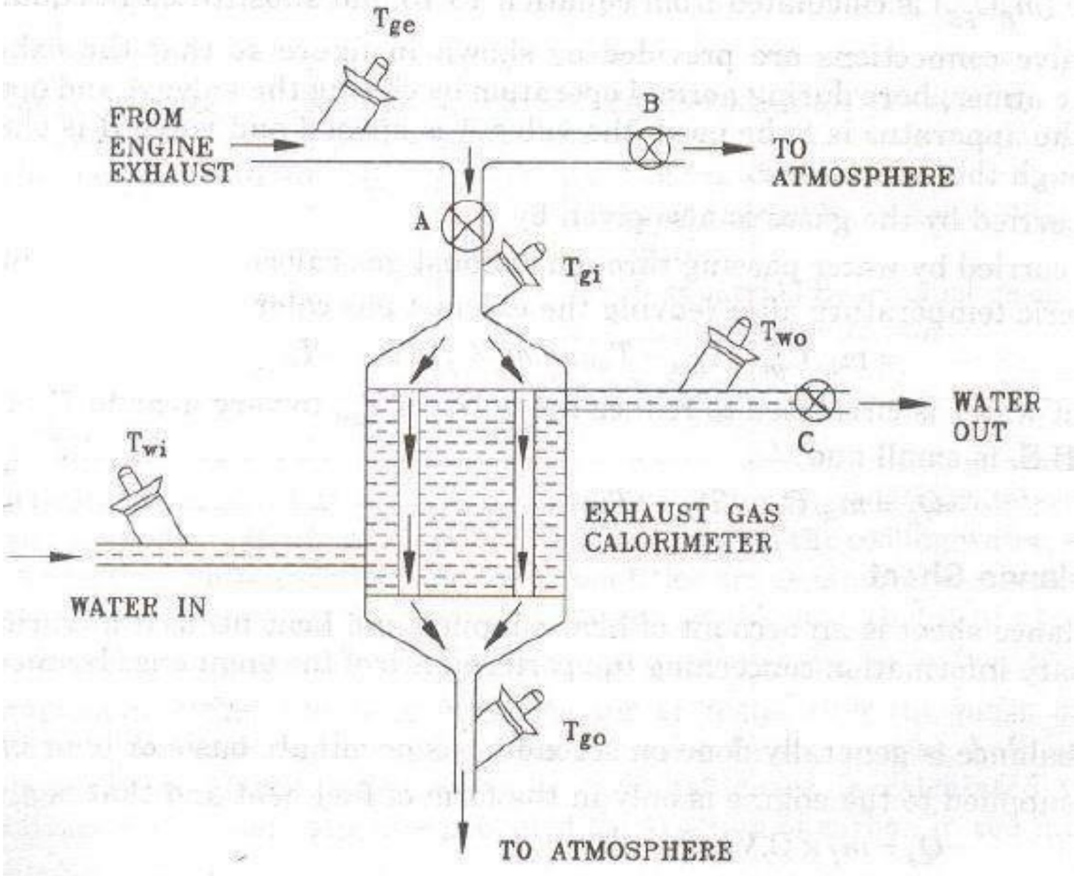


Fig.10.8 Exhaust Gas Calorimeter

$$\therefore m_g = \frac{C_{pw}}{C_{pg}}\left(\frac{T_{wo} - T_{wi}}{T_{gi} - T_{go}}\right)m_w \qquad (17)$$

- 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

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

Where, T_{ge} = Temperature of exhaust gases just leaving the engine exhaust valve, C

T_a = Ambient temperature, C

- 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 atmospheric temperature after leaving the exhaust gas calorimeter.

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

- 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,

$$Q_g = m_w C_{pw}(T_{wo} - T_{wi})$$

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

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

- 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

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

- 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

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

- 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.

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$= isfc = \frac{3600\,\dot{m}}{ip}$ kg/kWh

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$= bsfc = \frac{3600\,\dot{m}}{bp}$ kg/k\ h

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

$$\eta_{th} = \frac{Indicated\ power\ in\ kW}{(Mass\ flow\ rate\ of\ fuel\ in\ kg/s)\ x\ (Calorific\ value\ of\ fuel\ in\ kJ/kg)}$$

Or

$$\eta_{th} = \frac{ip}{n \times CV}$$

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

Or

$$\eta_{bth} = \frac{bp}{n \times CV}$$

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 auxiliaries like fuel pump, lubricating oil pump, water circulating pump, magneto and distributor, electric generator for battery charging, radiator fan etc., and (iii) work required 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

$$\eta_{mech} = \frac{bp}{ip}$$

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.

$$\eta_v = \frac{V_{actual}}{V_{theoritical}}$$

Where n is the number of intake strokes per minute and V_s is the stroke volume of the piston

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.

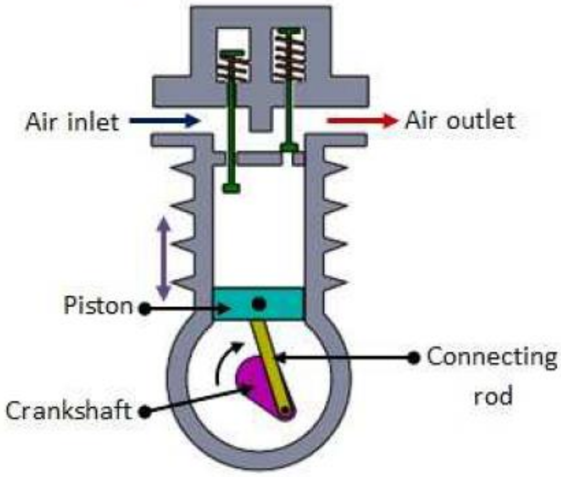
The compressors are also classified based on other aspects like

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

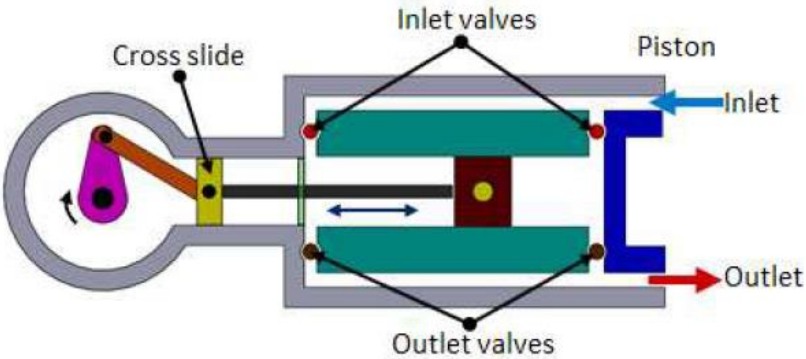
Reciprocating air compressors



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.

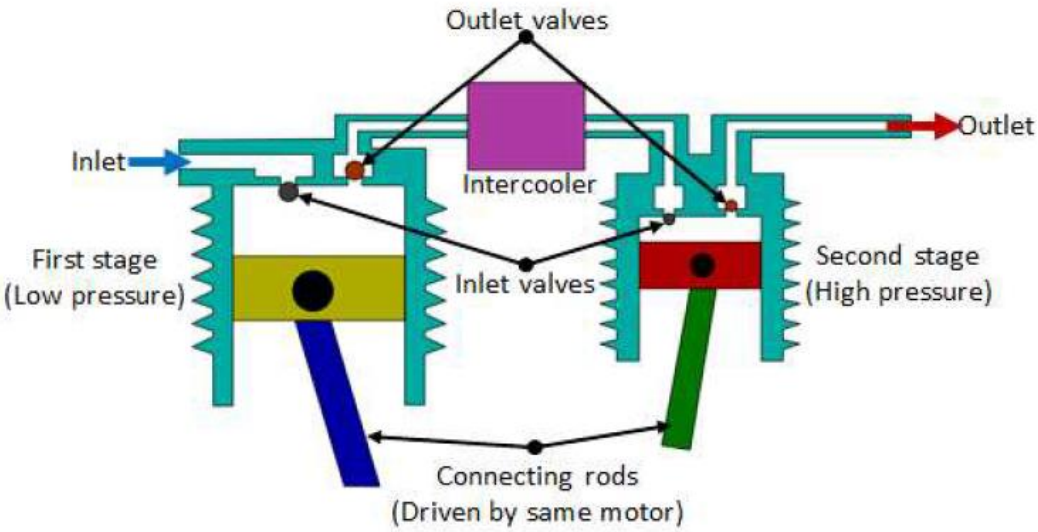
Double acting reciprocating air compressor



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.

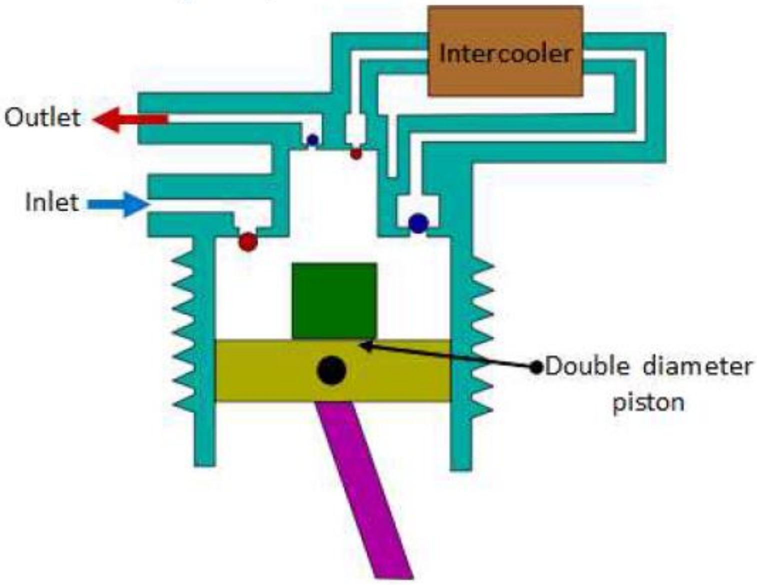
Multistage compressor



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



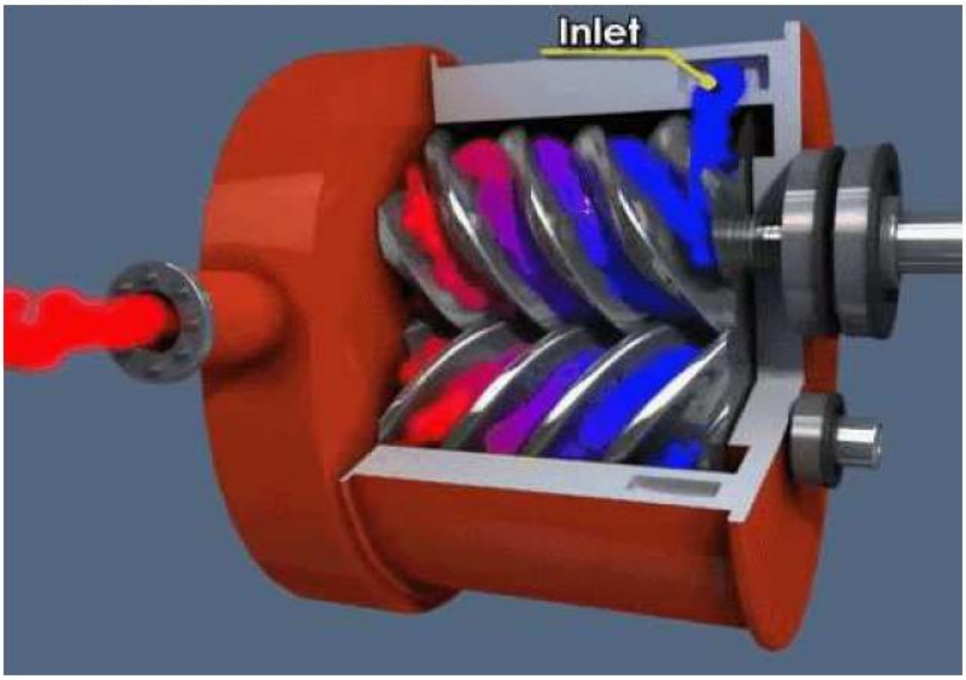
Combined two stage compressor

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.

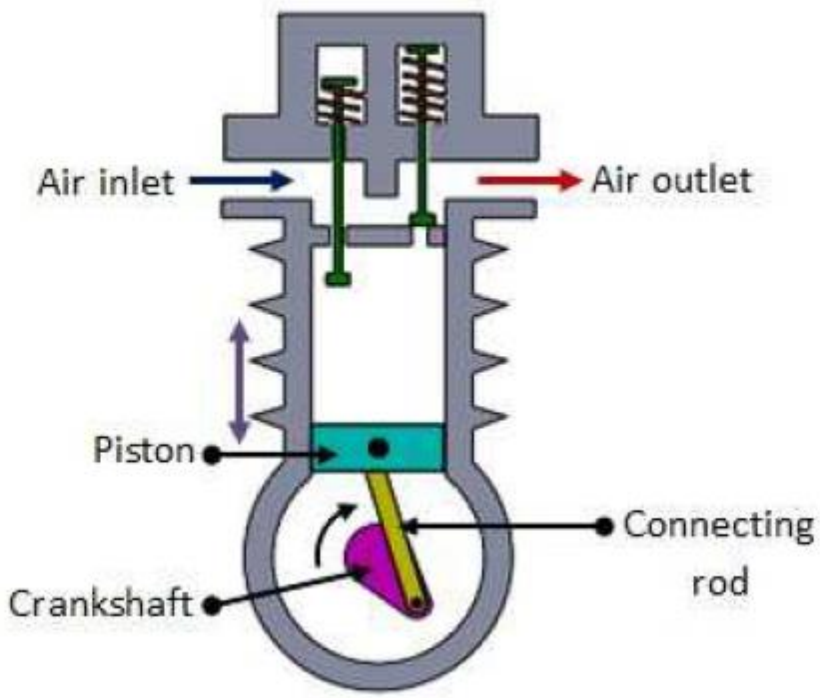
A fan moves large amounts of air with a low increase in pressure:

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.

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

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,

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

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

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

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

(*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).

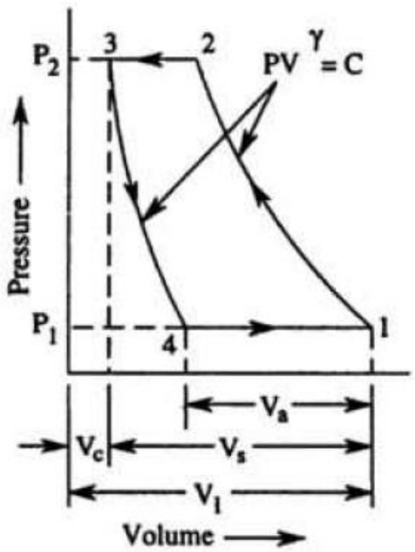
1. If the expansion and compression follows the same law, then volumetric efficiency in terms of clearance ratio and pressure ratio is given by

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

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

If P_1V_a of Eq.   is replaced by mRT_1, then the work required per kg of ai
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 = $W \times$ 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 $P_1V_1 = mRT_1$ then isothermal work required per kg of air is giv

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

(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,

$$W = \frac{\gamma}{\gamma - 1}(P_2V_2 - P_1V_1) \quad \text{J}, = \frac{\gamma}{\gamma - 1}P_1V_1\left(\frac{P_2V_2}{P_1V_1} - 1\right)$$

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

$$= \frac{\gamma}{\gamma - 1}P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}} - 1\right], \quad \left[\therefore \quad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}}\right]$$

(f) Adiabatic power = $\dfrac{\gamma}{\gamma - 1}P_1V_1\left[\left(\dfrac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}} - 1\right] \times \dfrac{N}{60}$ 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.

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

$$W=\frac{n}{n-1}P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}}-1\right]-\frac{n}{n-1}P_1V_4\left[\left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}}-1\right]$$

$$W=\frac{n}{n-1}P_1(V_1-V_4)\left[\left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}}-1\right]=\frac{n}{n-1}P_1V_a\left[\left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}}-1\right]\quad \text{J/cycle}$$

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.

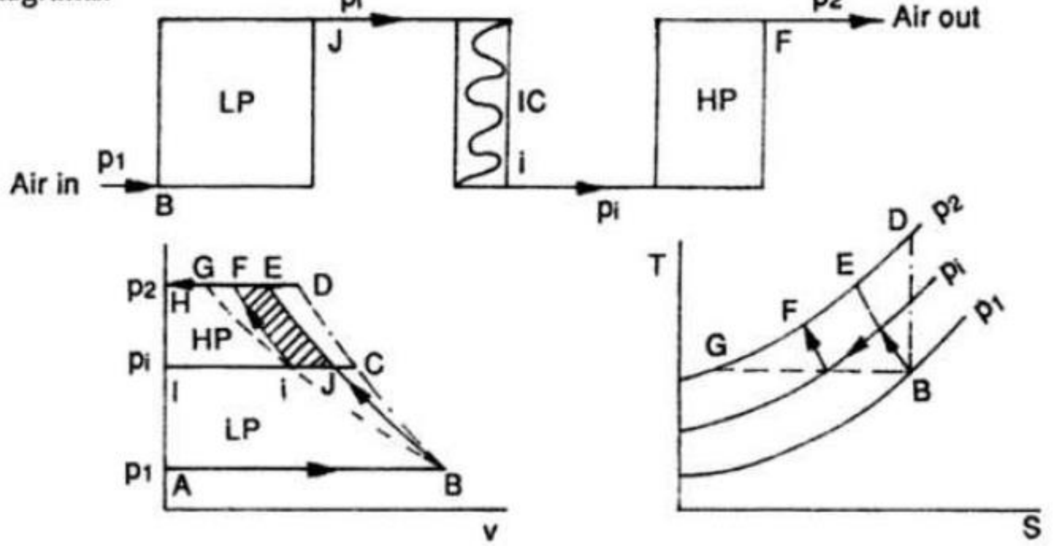
Indicated power of the compressor

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

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

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*

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

$= \{n/(n-1)\}\, p v \{1 - (p/p)^{(n-1)/n}\} + \{n/(n-1)\}\, p v \{1 - (p/p)^{(n-1)/n}\}$

Taking v_B as v_1; and since B or 1, and i are on the same isothermal $p_1v_1 = p_iv_i$

$\therefore$ $$W = \{n/(n-1)\}\, p_1v_1\{2 - (p_i/p_1)^{(n-1)/n} - (p_2/p_i)^{(n-1)/n}\}$$

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,

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

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

or $$p_1^xp_2^x = p_i^{2x}$$

or $$p_i^2 = p_1p_2$$

$\therefore$ $$p_i = (p_1p_2)^{0.5}, \text{ for minimum work}$$

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

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

$\therefore$ $$\text{Minimum Work } W_{min} = \{n/(n-1)\}\, p_1v_1\, \{2 - (p_2/p_1)^{(n-1)/2n} - (p_2/p_1)^{(n-1)/2n}\}$$

$$= \{2n/(n-1)\}\, p_1v_1\, \{1 - (p_2/p_1)^{(n-1)/2n}\}$$

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

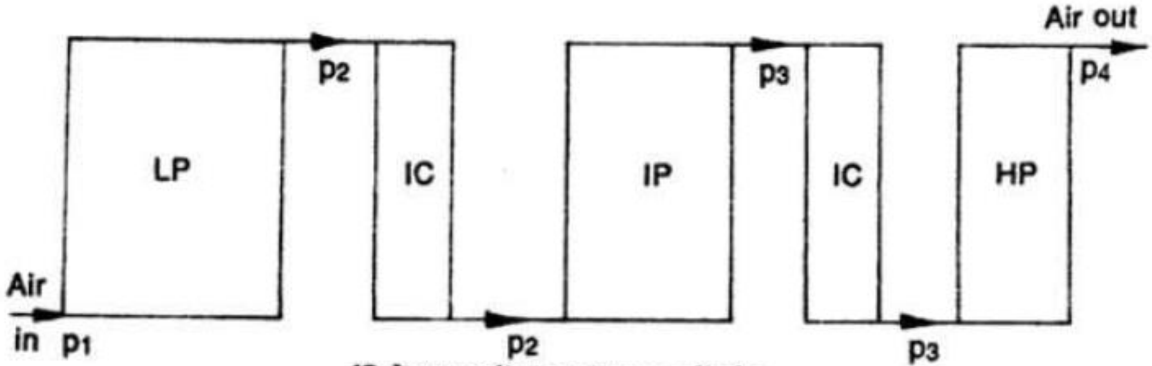
For temperature we get,

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

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.

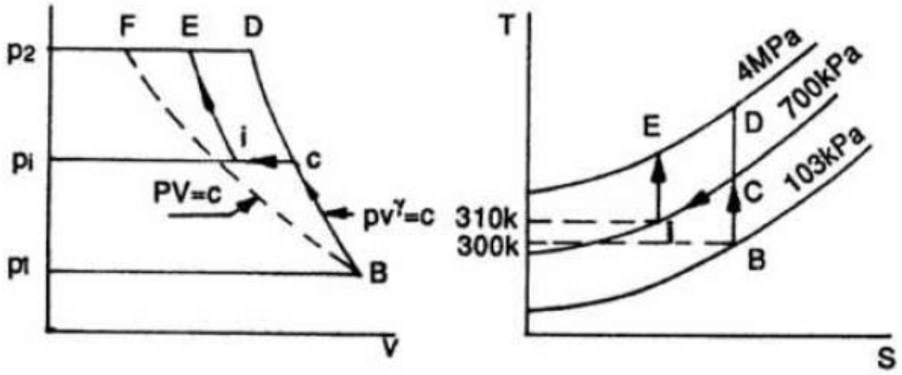
If intercooling is not perfect, i.e. if the air is not cooled down to its initial temperature, the work for the cylinders is to be calculated saparately and then added together. For a multistage air compressor with clearance volume, the volumetric efficiency is that of the *LP* cylinder of the compressor. 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.

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

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

$= \{1.4\times0.287\times300/(1.4-1)\}\, \{1 - (700/103)^{0.2857}\}$

$\qquad + \{1.4\times0.287\times310 / (1.4 - 1)\}\, [\, 1 - \{(4\times10^6) / (700\times10^3)\}^{0.2857}]$

$= 301.35\,(-0.729) + 311.4\,(-0.6454) = -219.6 - 201 = -420.6$ 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.

$T_1 = 273 + 27 = 300K$; 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

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

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

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

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

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

ii) HEAT REJECTED PER MINUTE IN INTERCOOLERS

$$T_2/T_1 = (p_2/p_1)^{(n-1)/n} = (p_4/p_1)^{(n-1)/3n}$$
$$= (3500/95)^{(1.3 - 1)/(3 \times 1.3)}$$
$$= 36.84^{0.0769} = 1.3196$$

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

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

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

$$\therefore \quad \text{Total heat rejected per minute} = 22.87 \times 16.55 \times 3$$
$$= 1135.5 \text{ kJ}$$

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

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

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

$$\int_{T_1}^{T_2} C_P^0 dT = 28.11 \times 120 + \frac{0.1967\times10^{-2}}{2}(400^2 - 280^2) + \frac{0.4802\times10^{-5}}{3}(400^3 - 280^3)$$

$$- \frac{1.966\times10^{-9}}{4}(400^4 - 280^4) = 3511.2 \text{ J/mol}$$

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

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.

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

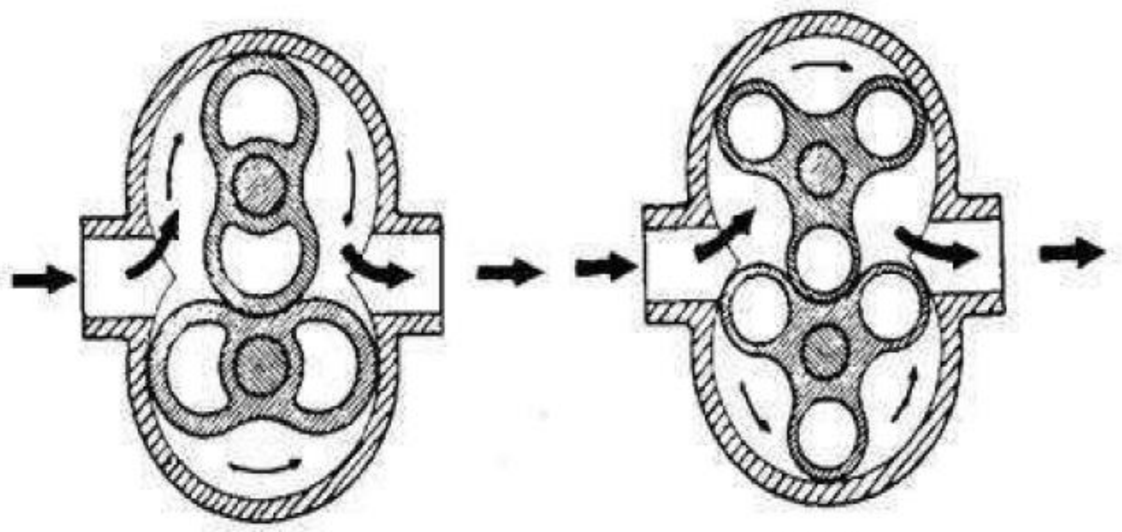
$$\dot{n} = \frac{20g}{s}\frac{1\,\text{mol}}{28.84\text{g}} = 0.6935 \text{ mol/s}$$

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

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.



TWO LOBE THREE LOBE

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.

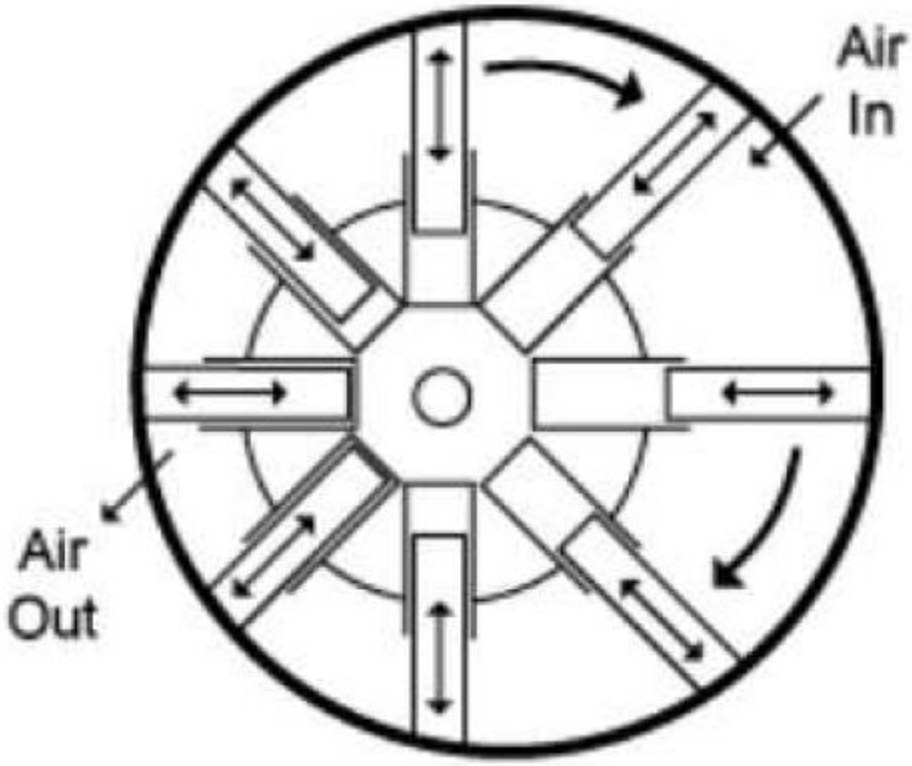
The roots-blower is sensitive to foreign matter, i.e., Filter cleaning of the conveying air is required;

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.

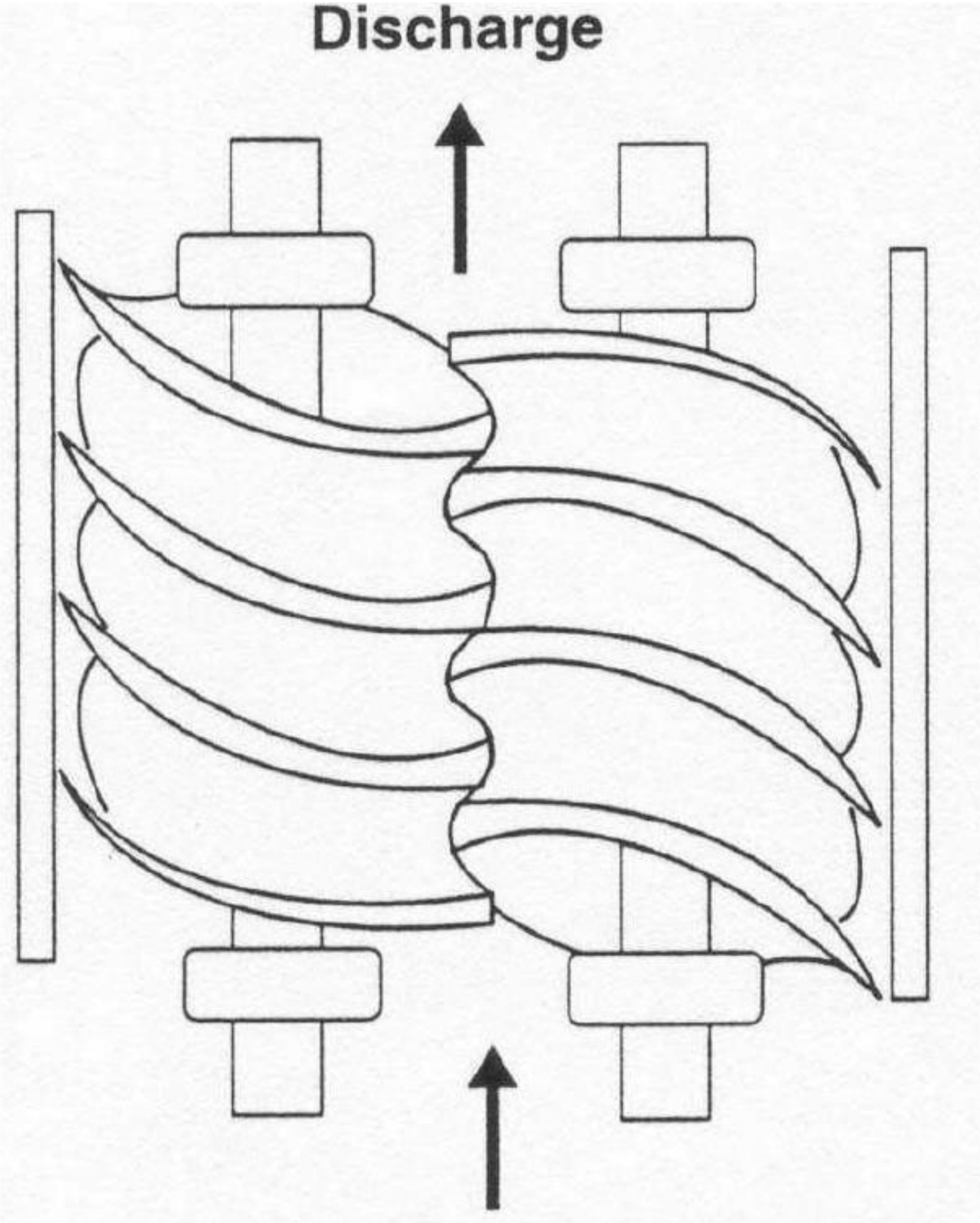
The shaft inside of the vane-housing extends through air-tight seals up into the air tool, and is attached to tooling on the end or to gearing of some sort. The result is rotary motion of that tooling.

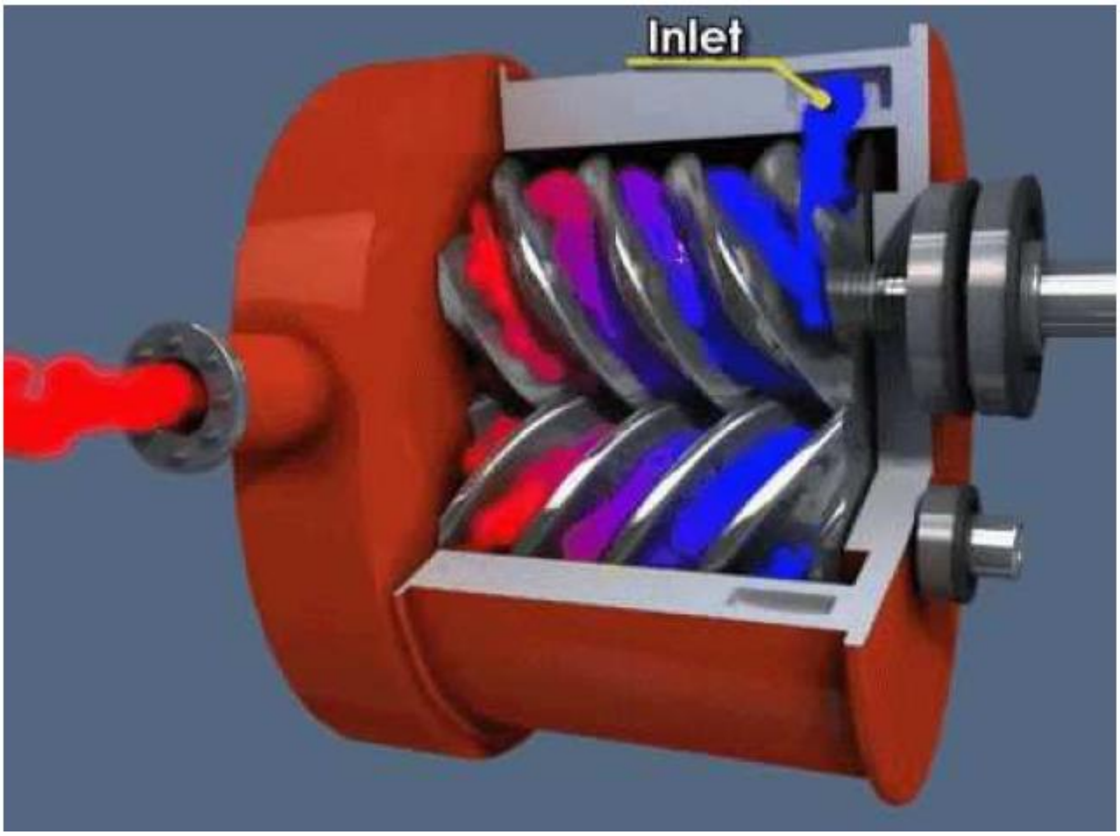
The power that drives the air tool is compressed air.

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



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^oC 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.,

$$m = \frac{p_1 v_1}{RT_1} = \frac{1.013 \times 10^5 \times 1}{287 \times 288} = 1.266 kg/\min$$

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

$D = 150\times10^{-3}m$	$P_2 = 5\times10^5\ N/m^2$
$L = 150\times10^{-3}\ m$	$T_1 = 27+273 = 300K$
$V_c = 0.15V_s$	$N = 720rpm$
$P_1 = 1\times10^5\ N/m^2$	$pv^n = C\ n=1.3$

Find

- i. V_1, V_2, V_3, V_4
- ii. FAD (V_a)
- iii. η_v

Solution

$$V_1 = V_c + V_s$$

$$V_s = \frac{\pi}{4}D^2 L.N = \frac{\pi}{4}(0.15)^2 \times .0.15 \times 720 = 1.9085 m^3 / \text{min}$$

$$V_c = 0.15\ V_s$$

$$= 0.15\times1.9085$$

$$V_c = 0.2862\ m^3/\text{min}$$

$$V_1 = V_c + V_s$$

$$= 0.2862 + 1.9085$$

$$\boxed{V_1 = 2.1948} \ m^3/min$$

$$P_1 V_1^{\,n} = P_2 V_2^{\,n}$$

$$V_2 = V_1 \left(\frac{P_1}{P_2} \right)^{1/n}$$

$$= 2.1948 \left(\frac{1 \times 10^5}{5 \times 10_5} \right)^{1/1.3}$$

$$\boxed{V_2 = 0.6366 \ m^3/min}$$

$$\boxed{V_3 = 0.2862 \ m^3/min} \ = V_c$$

$$V_c = V_3 \quad \therefore$$

$$P_3 V_3^{\,n} = P_4 V_4^{\,n}$$

$$V_4 = V_3 \left(\frac{P_3}{P_4} \right)^{1/4}$$

WKT

$$P_2 = P_3$$

$$P_1 = P_4$$

$$\therefore V_4 = V_3 \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}}$$

$$= 0.2862 \left[\frac{5 \times 10^5}{1 \times 10^5} \right]^{1/1.3}$$

$$\boxed{V_4 = 0.98674 \text{ m}^3/\text{min}}$$

To Find D

$$IP = W.N_w$$

where

N_w = Number of working stroke

For Double acting $N_w = 2N$

For single acting $N_w = N$

$$\therefore N_w = 2 \times 1.5 \times 60 = 180 \text{ rpm}$$

$$\therefore \text{W.D/cycle} = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$=\frac{1.2}{1.2-1}\times 100\times 10^{3}\left(\frac{\pi}{4}D^{2}\times 0.833\right)\times\left[\left(\frac{(500)}{100}\right)^{\frac{0.2}{1.2}}-1\right]$$

$\boxed{W = 120764.2D^2}$ N-m

$$\therefore \text{IP} = \frac{W.N_w}{60}$$

$$50\times 10^{3} = \frac{1207642D^{2}\times 180}{60}$$

$D^2 = 0.1380$

$\boxed{D = 0.371 \text{ m}}$

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:

$$h = (V^2/g) \text{ ---- } (1).$$

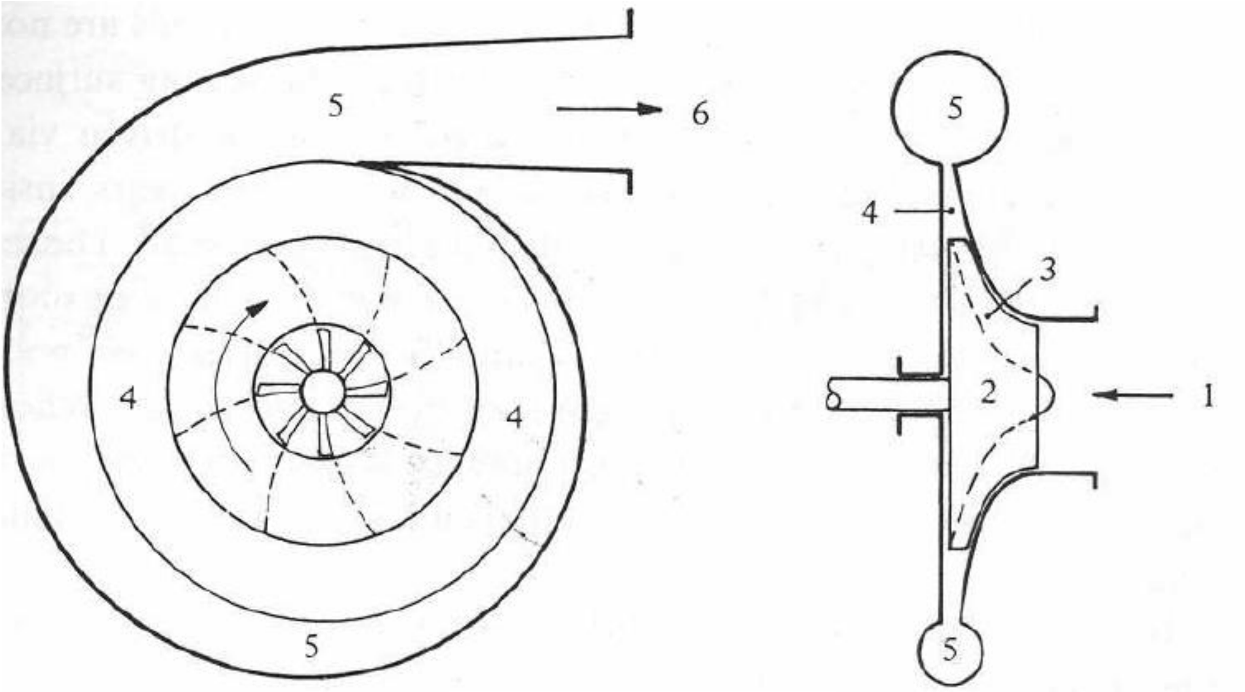
Where h = static head developed, m

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:

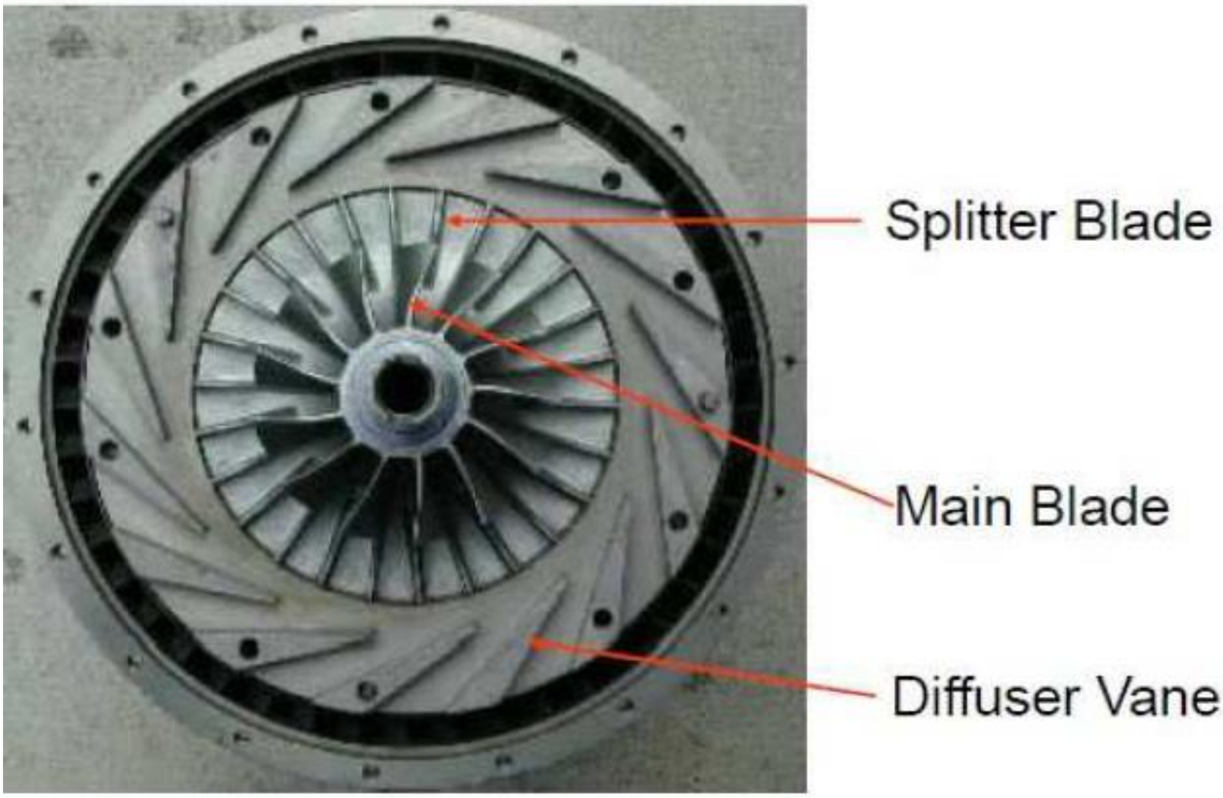
$$\Delta P=\rho g h=\rho V^2 \text{ ----- } (2)$$



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:

$$W_c = m(h_e - h_i)$$

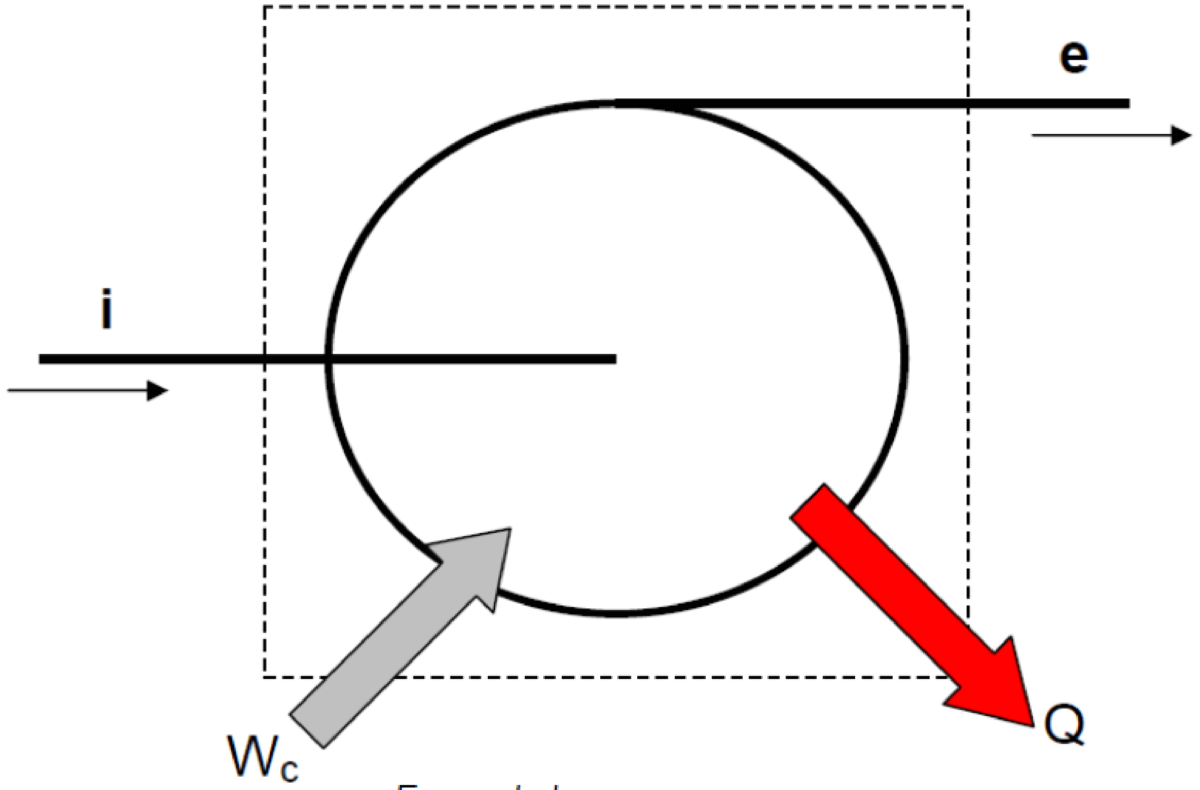
The above equation is valid for both reversible as well as irreversible adiabatic compression, provided the actual enthalpy is used at the exit in case of irreversible compression. In case of reversible, adiabatic compression, the power input to the compressor is given by:

$$W_{c,isen} = m(h_e - h_i)_{isen}$$

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 ηpol is defined. It is given by:

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:

$$\eta_{pol} = \frac{w_{pol}}{w_{act}} = \frac{\int_{Pi}^{Pe} vdp}{(h_e - h_i)}$$

$$w_{pol} = \int_{Pi}^{Pe} vdP = f\left(\frac{n}{n-1}\right)Pivi\left[\left(\frac{Pe}{Pi}\right)^{\frac{n-1}{n}} - 1\right]$$

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

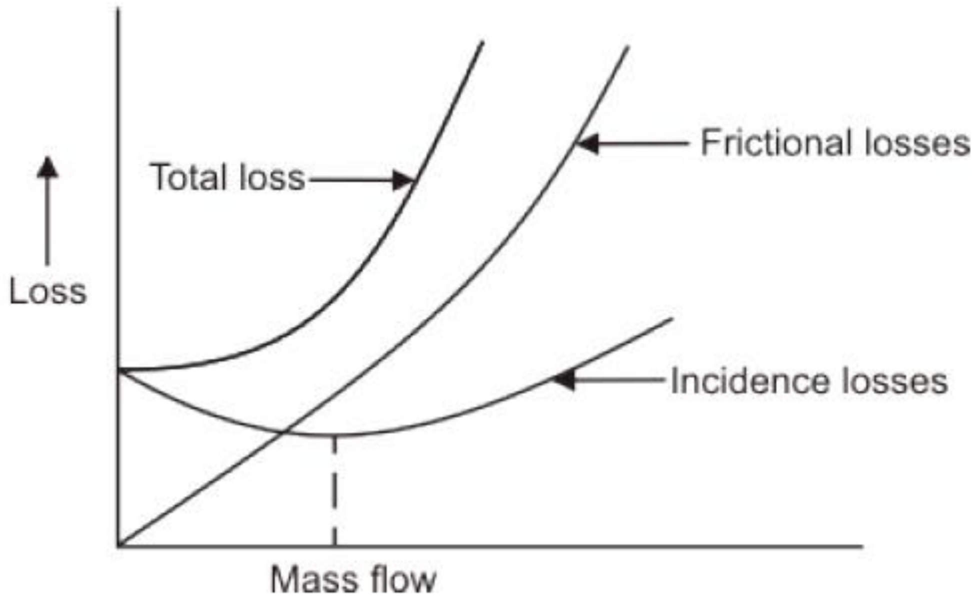


Figure 8.1 Dependence of various losses with mass flow in a centrifugal compressor

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.

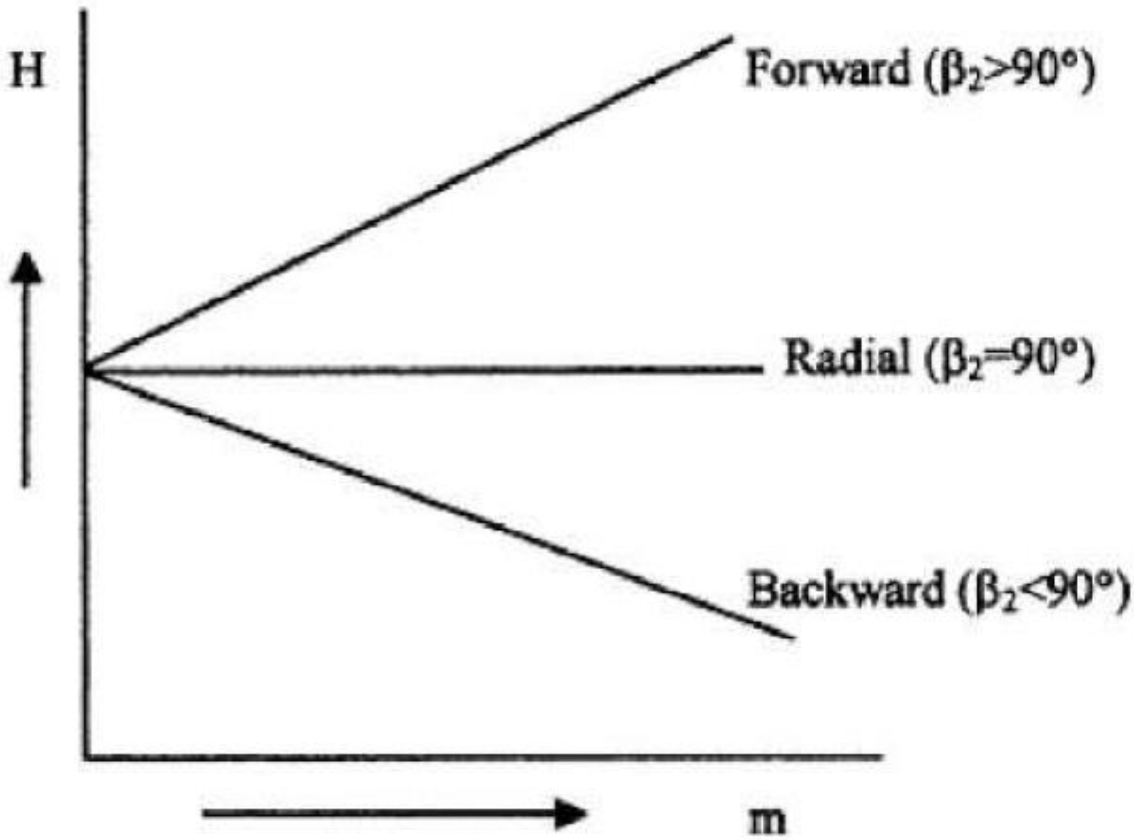
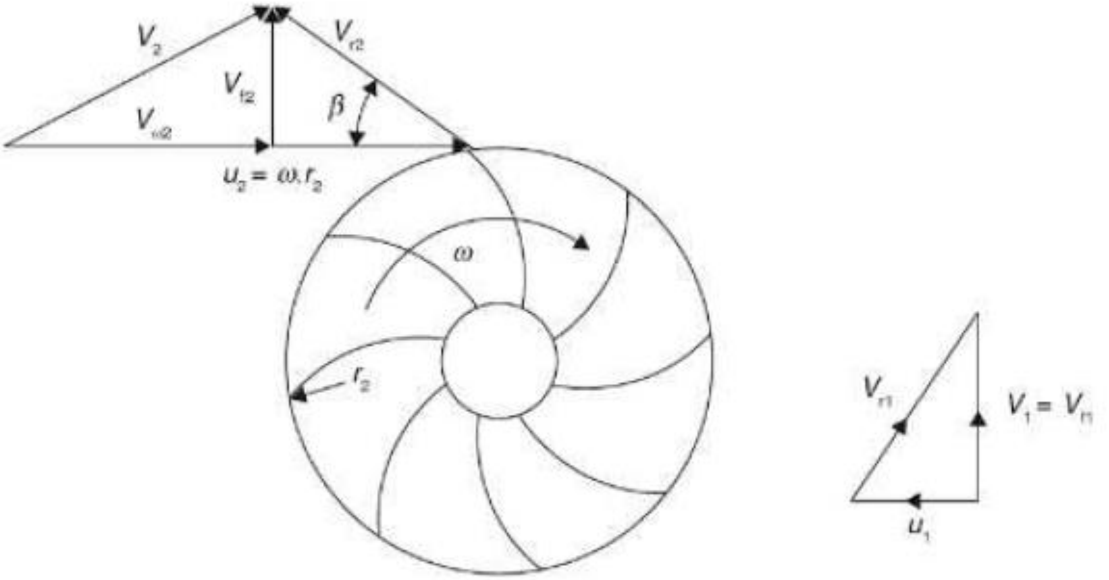


Figure 4.4 Pressure ratio or head versus mass flow or volume flow, for the three blade shapes.



(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_w = 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)

$$\frac{T_{0_2}}{T_{0_1}} = \frac{T_{0_3}}{T_{0_1}} = 1 + \frac{\Psi \sigma U_2^2}{c_p T_{0_1}} \tag{4}$$

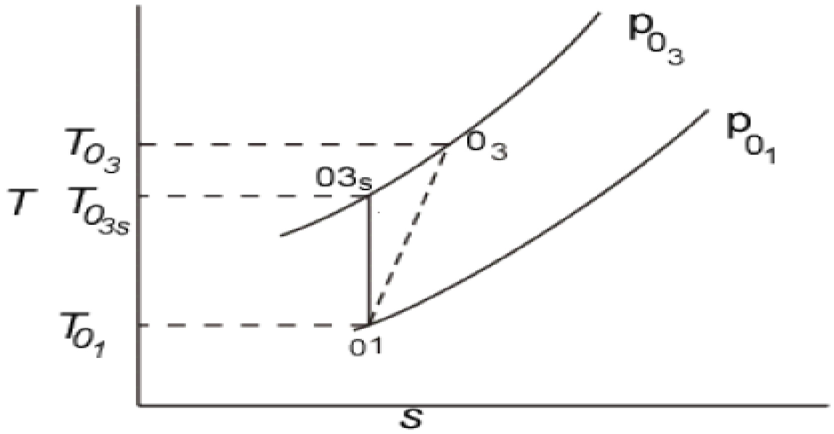
The overall stagnation pressure ratio can be written as

$$\frac{p_{0_3}}{p_{0_1}} = \left(\frac{T_{0_{3s}}}{T_{0_1}}\right)^{\frac{\gamma}{\gamma-1}}$$

$$= \left[1 + \frac{\eta_c (T_{0_3} - T_{0_1})}{T_{0_1}}\right]^{\frac{\gamma}{\gamma-1}} \tag{5}$$

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

$$\eta_c = \frac{T_{0_{3s}} - T_{0_1}}{T_{0_3} - T_{0_1}} \tag{6}$$



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

$C_p = dP / (\rho\, v^2 / 2)$

$\quad = dh\, (\rho\, v^2 / 2\, g) \qquad (1)$

where

C_p *= pressure coefficient*

dp = pressure difference (N)

ρ = fluid density (kg/m³)

v = flow velocity (m/s)

dh = head (m)

g = acceleration of gravity (= 9.81 m/s²)

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

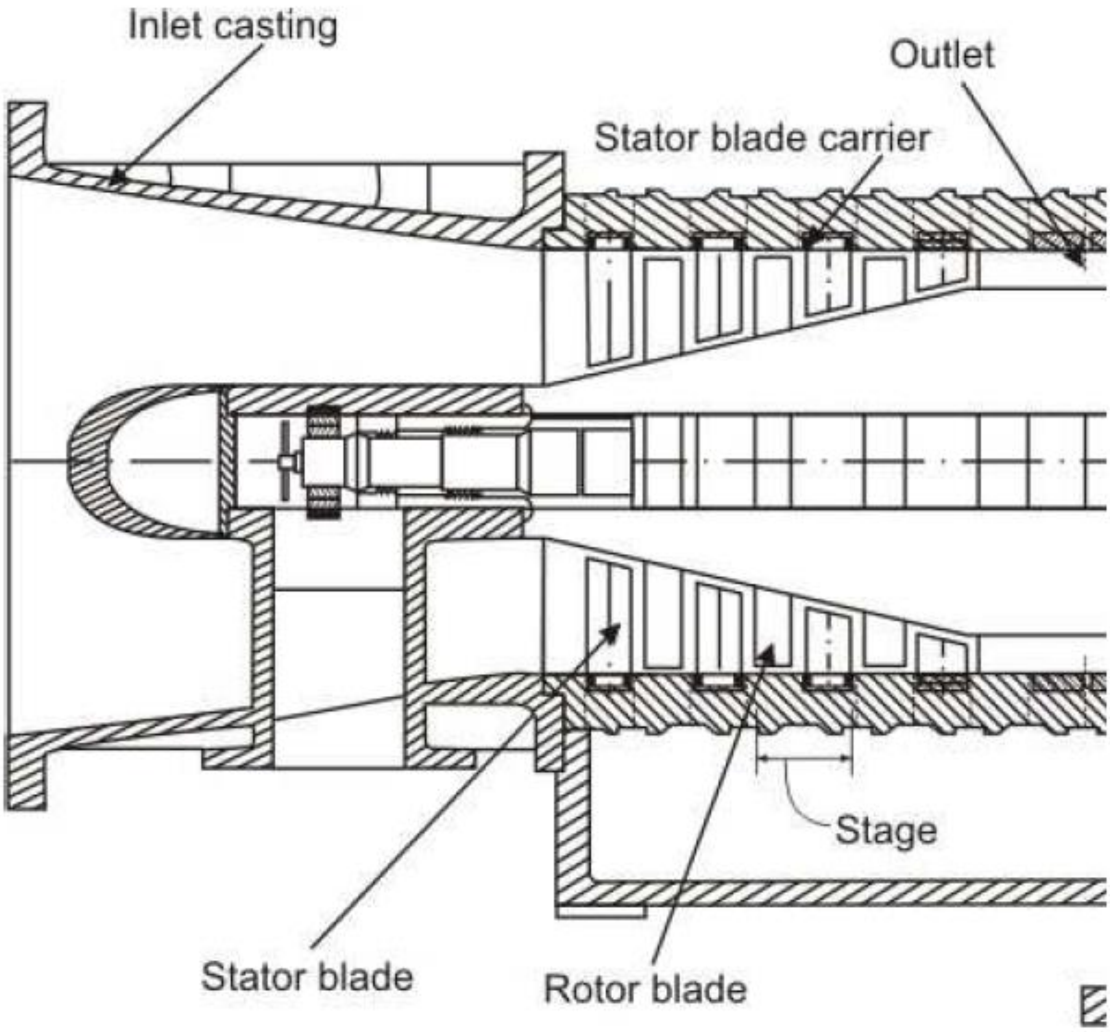
Comparison between Reciprocating and Centrifugal Compressors

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.

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.

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, $\bar{V}_1$, at and angle $\bar{\alpha}_1$ 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.

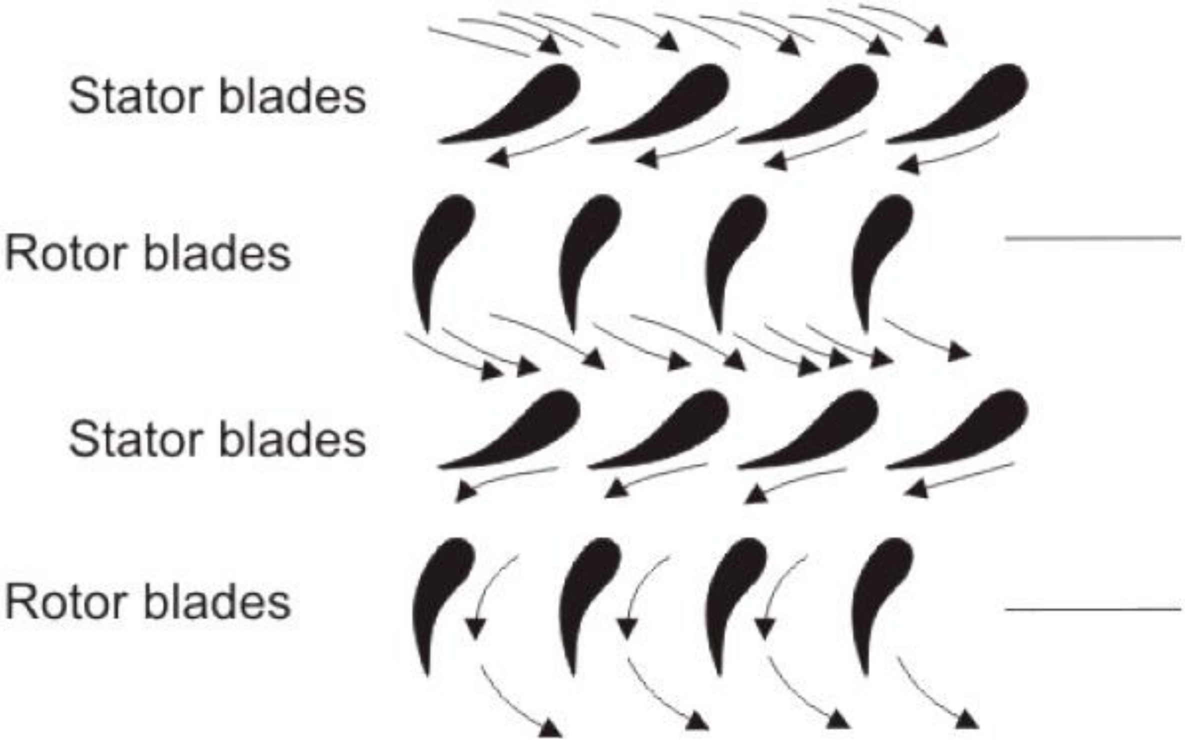
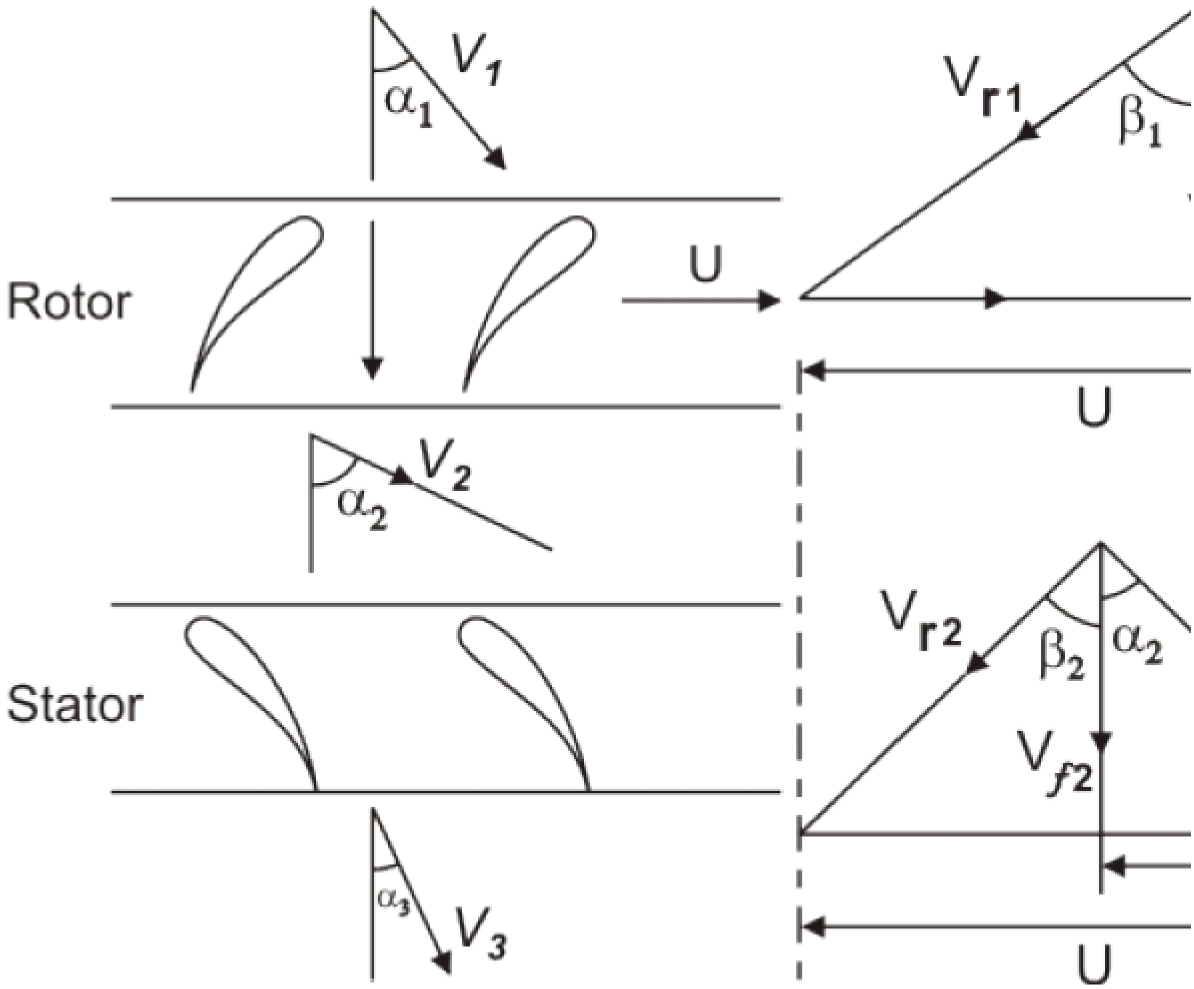


Figure 2 Flow through stages



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

$$w = UV_f(\tan\alpha_2 - \tan\alpha_1) \qquad (4)$$

or by using Eqs. (9.1) and (9.2)

$$w = UV_f(\tan\beta_1 - \tan\beta_2) \quad (5)$$

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

$$\Delta T_0 = \Delta T_s = \frac{UV_f}{c_p}(\tan\beta_1 - \tan\beta_2) \quad (6)$$

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

$$\begin{aligned} w &= U[(U - V_f\tan\alpha_1) - V_f\tan\beta_2] \quad (7) \\ &= U(U - V_f(\tan\alpha_1 + \tan\beta_2)) \end{aligned}$$

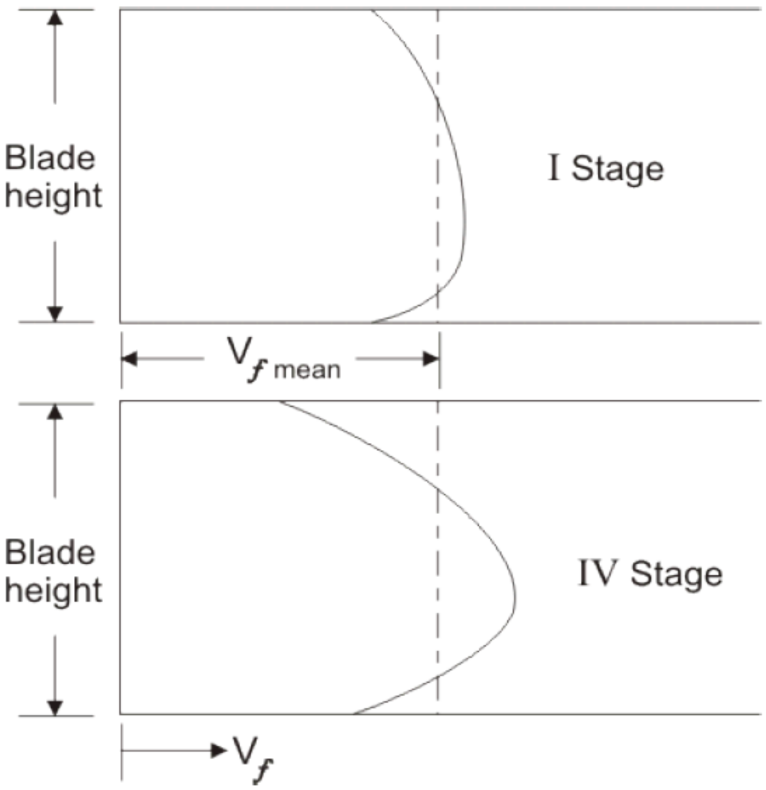


Figure 4 Axial velocity distibutions

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 $V_f(\tan \alpha_1 + \tan \beta_2)$. Any increase in 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.

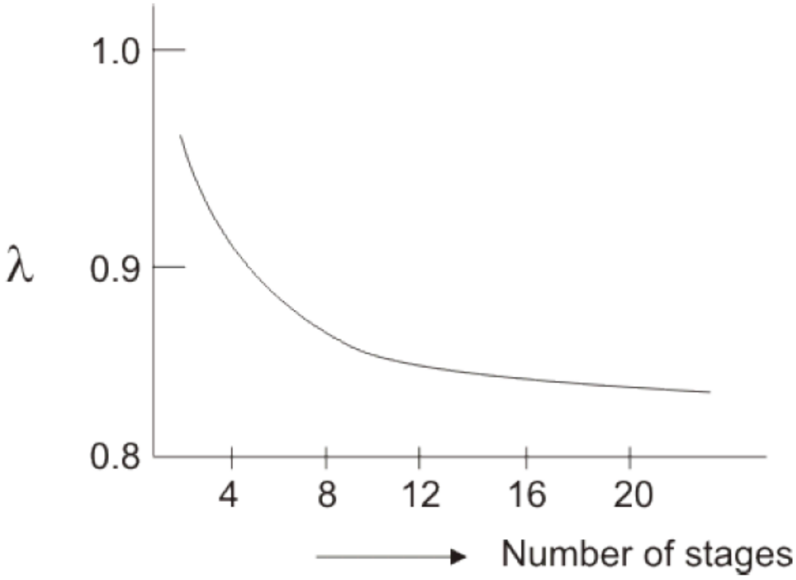


Figure 5 Variation of work-done factor with number of stages

Degree of reaction

Diffusion takes place in both rotor and the stator.

Static pressure rises in the rotor as well as the stator.

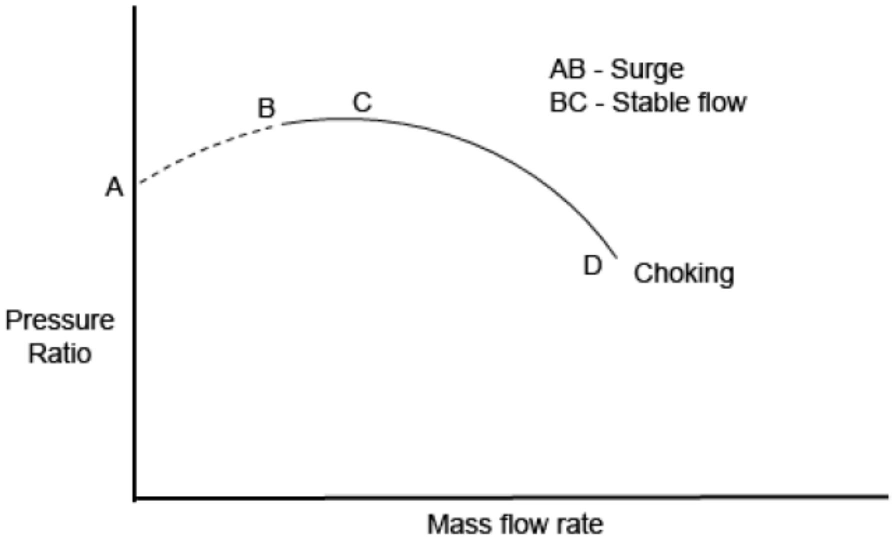
Degree of reaction provides a measure of the extent to which the rotor contributes to the overall pressure rise in the stage.

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.

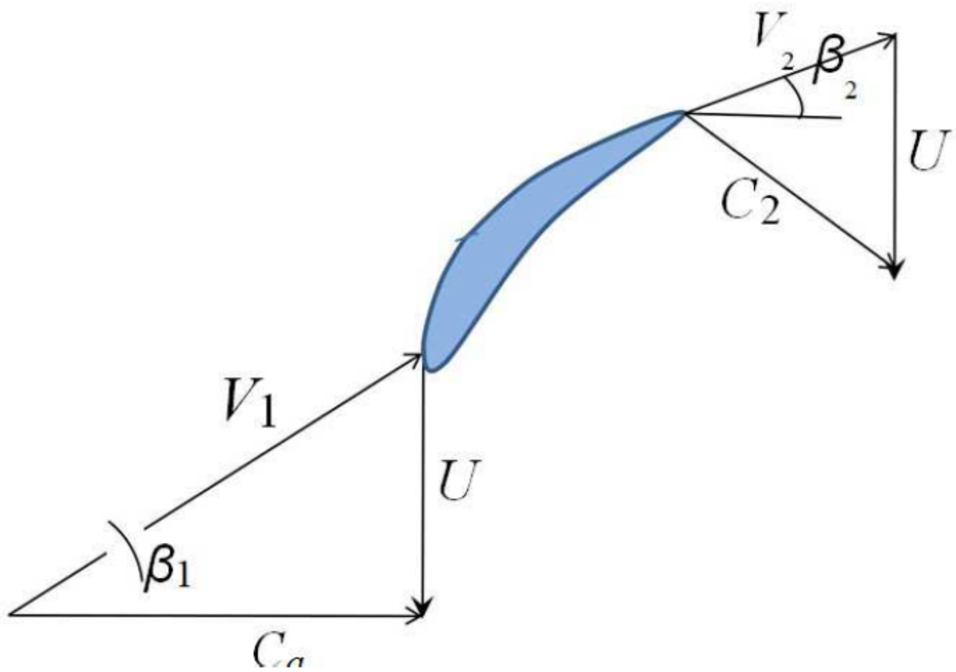
It is possible for several elements of a compressor stage to stall without the entire stage stalling or similar events occurring. When a stage either has a very strong stall in one of its elements, or a number of elements together collectively stall that the overall pressure ratio vs. flow characteristic 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



$$U = \pi \times \frac{d_t + d_h}{2} \times N = \pi \times \frac{0.6 + 0.5}{2} \times 100 = 172.76 m/s$$

$$\beta_1 = \tan^{-1} \frac{U}{C_a} = 49.2°$$

$$\beta_2 = 49.2 - 30.2 = 19°$$

$$\beta_1 = \tan^{-1} \frac{U}{C_a} = 49.2°$$

$$\beta_2 = 49.2 - 30.2 = 19°$$

$$\tan\alpha_2 \quad \frac{U - C_a \tan \beta_2}{C_a} = 80.75$$

$$\alpha_2 = 38.92°$$

$$\dot{m} = \frac{\pi}{4} \times \left(d_t^2 - d_h^2\right) \times C_2 \times \rho_2 \qquad \& \; T_1 = T_{01} - \frac{C_a^2}{2C_p} = 276.8\ K$$

$$T_{02} = T_{01} \times \left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore \quad T_{02} = 303.41K$$

$$T_2 = 303.41 - \frac{C_2^2}{2C_p} \quad \& \; \cos\alpha_2 = \frac{C_a}{C_2}$$

$$\therefore C_2 = \frac{C_a}{\cos\alpha_2} = \frac{150}{\cos 38.92} = 192.79 m/s$$

$$T_2 = 303.41 - \frac{192.79}{2010}^2 =$$

$$284.91 K \quad P_2 = 1.216 bar$$

$$\rho_2 = \frac{1.216 \times 101325}{287 \times 284.9} = 1.507\ Kg/m^3$$

$$\dot{m} = 19.53\ Kg/s$$

$$P = U \times C_a \times \dot{m} \times (\tan\beta_1 - \tan\beta_2)$$

= 172.76×150×19.53× (tan 49.2 − tan19) = 412 KW

$$R_X = 1 - \frac{C_a}{2U} \times (\tan\beta_1 + \tan\beta_2)$$

$$= 1 - \frac{150}{2\times 172.76} \times (\tan 49.2 + \tan 19) = 1 - 0.65 = 0.35$$

$$\Delta T_{0s} = \frac{U \times C_a}{C_p} \times (\tan\beta_1 - \tan\beta_2)$$

$$= \frac{172.76\times 150}{1005} \times (\tan 49.2 - \tan 19) = 20.99\,K$$

$$\Delta T_{0overall} = \frac{T_1}{\eta_C} \times \left(\pi_C^{\frac{\gamma-1}{\gamma}} - 1\right)$$

$$= \frac{288}{0.85} \times \left(6^{0.286} - 1\right) = 226.5K$$

$$n = \frac{226.5}{20.99} = 10.79 \approx 11$$

Therefore the number of stages required for the given pressure ratio is 11.0.