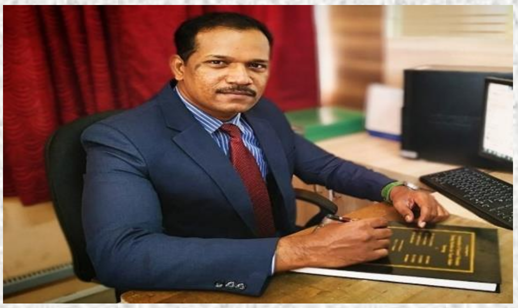
Pimpri Chinchwad Education Trust's
PIMPRI CHINCHWAD COLLEGE OF ENGINEERING & RESEARCH
Laxminagar, Ravet, Pune – 412101

TUTORIAL NOTES

Applied Thermodynamics

(Subject Code 202050)

SE Mechanical Engineering 2015 Pattern (Semester – IV)



Prepared by : Prof. Rahul Krishnaji Bawane
Assistant Professor, Department of Mechanical Engineering

PIMPRI CHINCHWAD EDUCATION TRUST'S

PIMPRI CHINCHWAD COLLEGE OF ENGINEERING & RESEARCH,

RAVET, PUNE 412101

VISION AND MISSION OF THE INSTITUTE:

VISION:

To be a Premier Institute of Technical Education and Research to serve the need of society and all the stakeholders.

MISSION:

To establish state-of-the-art facilities to create an environment resulting in individuals who are technically sound having professionalism, research and innovative aptitude with high moral and ethical values.

VISION AND MISSION OF THE DEPARTMENT OF MECHANICAL ENGINEERING:

VISION:

To be a Premier Department of Mechanical Engineering and research to serve the need of the society and all the stakeholders.

MISSION:

- To provide state of art facilities to impart quality education.
- To undertake various value added and add on courses to make students technically sound and thorough professionals.
- To collaborate with the industries and academia and strive to transform the research and innovative aptitude in the students and faculties.
- To inculcate high moral, ethical values and national pride in students and faculties.

202050: Applied Thermodynamics

Prerequisites: - 1. Engineering Thermodynamics.
2. Engineering Mathematics

Course Objectives:

- To get familiar with fundamentals of I. C. Engines, Construction and working Principle of an Engine and Compare Actual, Fuel-Air and Air standard cycle Performance.
- To study Combustion in SI and CI engines and its controlling factor in order to extract maximum power.
- To study emission from IC Engines and its controlling method, Various emission norms.
- Perform Testing of I. C. Engines and methods to estimate Indicated, Brake and Frictional Power and efficiencies
- To understand theory and performance Calculation of Positive displacement compressor.

Course Outcomes:

On completion of the course, learner will be able to–

CO-1 : Classify various types of Engines, Compare Air standard, Fuel Air and Actual cycles and make out various losses in real cycles.

CO-2 : Understand Theory of Carburetion, Modern Carburetor, Stages of Combustion in S. I. Engines and Theory of Detonation, Pre-ignition and factors affecting detonation.

CO-3 : Understand Fuel Supply system, Types of Injectors and Injection Pumps, Stages of Combustion in CI Engines, Theory of Detonation in CI Engines and Comparison of SI and CI Combustion and Knocking and Factors affecting, Criteria for good combustion chamber and types.

CO-4 : Carry out Testing of I. C. Engines and analyze its performance.

CO-5 : Describe construction and working of various I. C. Engine systems (Cooling, Lubrication, Ignition, Governing, and Starting) also various harmful gases emitted from exhaust and different devices to control pollution and emission norms for pollution control.

CO-6 : Describe construction, working of various types of reciprocating and rotary compressors with performance calculations of positive displacement compressors.

Unit I

Basics of IC Engines (5 Hrs)

Heat Engine, IC and EC engines, I.C. Engine construction - components and materials, Engine nomenclature, Valve timing diagram, Intake and exhaust system, Engine classification, Applications.

Fuel Air Cycle and Actual Cycle (5 Hrs)

Fuel air cycle, Assumptions, Comparison with air standard cycle, Effect of variables on performance, Actual cycle and various losses, Comparison of Air standard Vs Fuel Vs Actual cycle.

Unit II

SI Engines (5 Hrs)

Theory of Carburetion, Types of carburetors, Electronic fuel injection system, Combustion in spark Ignition engines, stages of combustion, flame propagation, rate of pressure rise, abnormal combustion, Phenomenon of Detonation in SI engines, effect of engine variables on Detonation. Combustion chambers, Rating of fuels in SI engines, Additives.

Unit III

CI Engines (5 Hrs)

Fuel supply system, types of fuel pump, injector and distribution system, Combustion in compression ignition engines, stages of combustion, factors affecting combustion, Phenomenon of knocking in CI engine. Effect of knocking, Methods of knock control, Types of combustion chambers, rating of fuels in CI engines. Dopes & Additives, Comparison of knocking in SI & CI engines.

Unit IV

Testing of IC Engines (6 Hrs)

Objective of testing, Various performance parameters for I.C. Engine - Indicated power, brake power, friction power, SFC, AF ratio etc. Methods to determine various performance parameters, characteristic curves, heat balance sheet.

Supercharging (2 Hrs)

Supercharging and turbo-charging methods and their limitations.

Unit V

I.C. Engine Systems (6 Hrs)

Cooling System, Lubrication System, Ignition System, Governing system, Starting System.

I.C. Engine Emissions and Control (4 Hrs)

Air pollution due to IC engine and its effect, Emissions from petrol/gas and diesel engines, Sources of emissions, Euro norms, Bharat stage norms, Emission control methods for SI and CI engines.

Unit VI

Positive Displacement Compressors (Reciprocating and Rotary) (10 Hrs)

Reciprocating Compressor - Single stage compressor – computation of work done, isothermal efficiency, effect of clearance volume, volumetric efficiency, Free air delivery, Theoretical and actual

indicator diagram, Multistaging of compressor, Computation of work done, Volumetric efficiency, Condition for maximum efficiency, Inter-cooling and after cooling, Capacity control of compressors

Rotary Compressor – Introduction, vane compressors, roots blower, screw compressor. (Numerical treatment on Reciprocating compressor single stage and multistage only)

Text:

1. V. Ganesan: Internal Combustion Engines, Tata McGraw-Hill
2. M.L. Mathur and R.P. Sharma: A course in Internal combustion engines, Dhanpat Rai
3. H.N. Gupta, Fundamentals of Internal Combustion Engines, PHI Learning Pvt. Ltd.

Reference:

1. Heywood: Internal Combustion Engine Fundamentals, Tata McGraw-Hill
2. Domkundwar & Domkundwar: Internal Combustion Engine, Dhanpat Rai
3. R. Yadav: Internal Combustion Engine, Central Book Depot, Ahmedabad.
4. S. Domkundwar, C. P. Kothandaraman, A. Domkundwar, Thermal Engineering, Dhanpat Rai & Co.

List of Practical's:

1. Study of Carburetor
2. Study of Fuel pump and injector
3. Study of Ignition System
4. Demonstration & study of commercial exhaust gas analyzers.
5. Morse Test on Multi cylinder Petrol/ Diesel engine for determination of Friction power.
6. Variable load test on diesel engine to determine various efficiencies, SFC and Heat balance sheet.
7. Test on variable compression ratio engine.
8. Visit to Automobile service station
9. Test on Positive Displacement Air Compressor
10. Assignment on any one advanced technology related to I.C. Engine such as VVT, VGT, HCCI
11. Assignment on alternative fuels used in I.C. Engines.

Notes:

1. Minimum 8 experiments should be performed.
2. Perform any 3 from 1 to 4.
3. Perform any 2 from 5, 6, and 7.
4. Experiment 8 and 9 are compulsory.

UNIT – I : BASIC OF IC ENGINE

Basic of IC Engines : *Heat Engine, IC and EC engines, I.C. Engine construction - components and materials, Engine nomenclature, Valve timing diagram, Intake and exhaust system, Engine classification, Applications.*
Fuel Air Cycle and Actual Cycle : *Fuel air cycle, Assumptions, Comparison with air standard cycle, Effect of variables on performance, Actual cycle and various losses, Comparison of Air Standard – Vs – Fuel – Vs – Actual Cycle.*

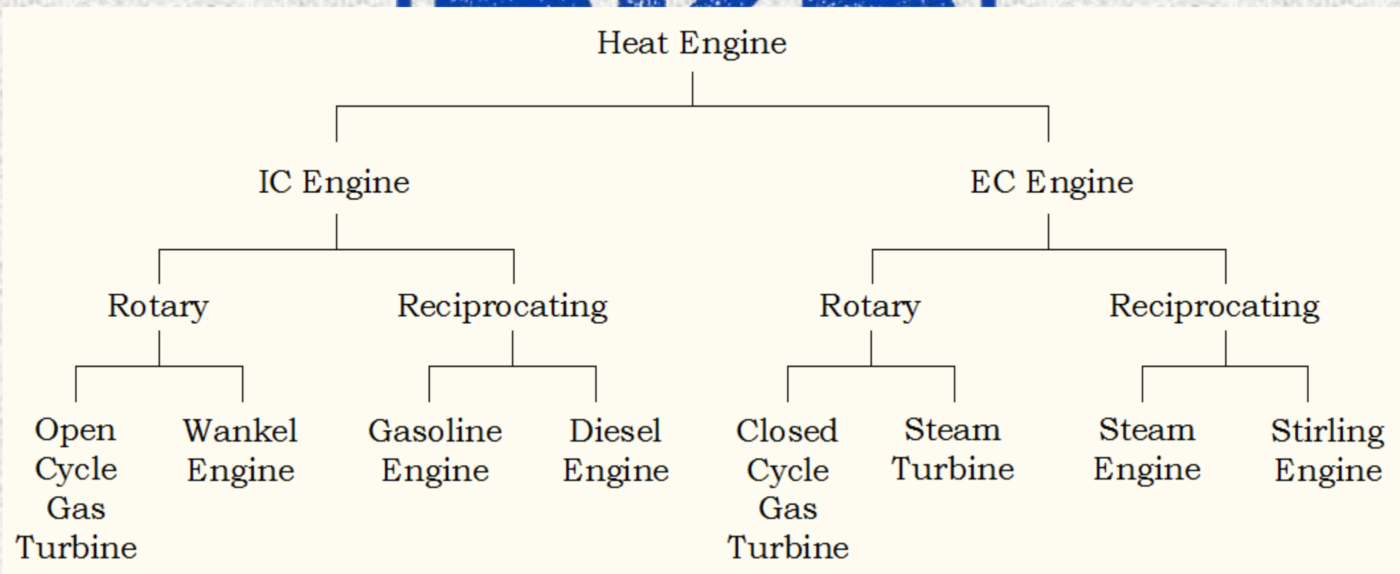
1. Heat Engine : SPPU : May-17, 6-Marks

An engine is a device which transforms one form of energy into another useful form of energy , during this transformation some energy is wasted, thus an engine never have 100% efficiency.

Heat engine is a device which transforms the chemical energy of a fuel into thermal energy and utilizes this thermal energy to perform some useful work, ie mechanical energy.

The conversion of all thermal energy into mechanical energy is not practically possible, hence heat engine always have efficiency less than one.

2. Classification of Heat Engines :



2.1. IC (Internal Combustion) Engines :

The Internal Combustion Engine (IC Engine) is a heat engine that converts chemical energy in a fuel into mechanical energy, usually in the form of a rotating output shaft.

Chemical energy of the fuel is first converted to thermal energy by means of combustion or oxidation with air inside the engine. This thermal energy raises the temperature and pressure of the gases within the engine, and the high-pressure gas then expands against the mechanical mechanisms of the engine. This expansion is converted by the mechanical linkages of the engine to a rotating crankshaft, which is the output of the engine.

Most internal combustion engines are reciprocating engines having pistons that reciprocate back and forth in cylinders internally within the engine.

2.2. EC (External Combustion) Engines :

The External Combustion Engines (EC Engine) is a heat engine that converts chemical energy in a fuel into Thermal energy which in turn utilized to raises the temperature and pressure of working fluid outside the engine system. The energy of the working fluid then converted into mechanical energy inside the engine system, usually in the form of a rotating output shaft.

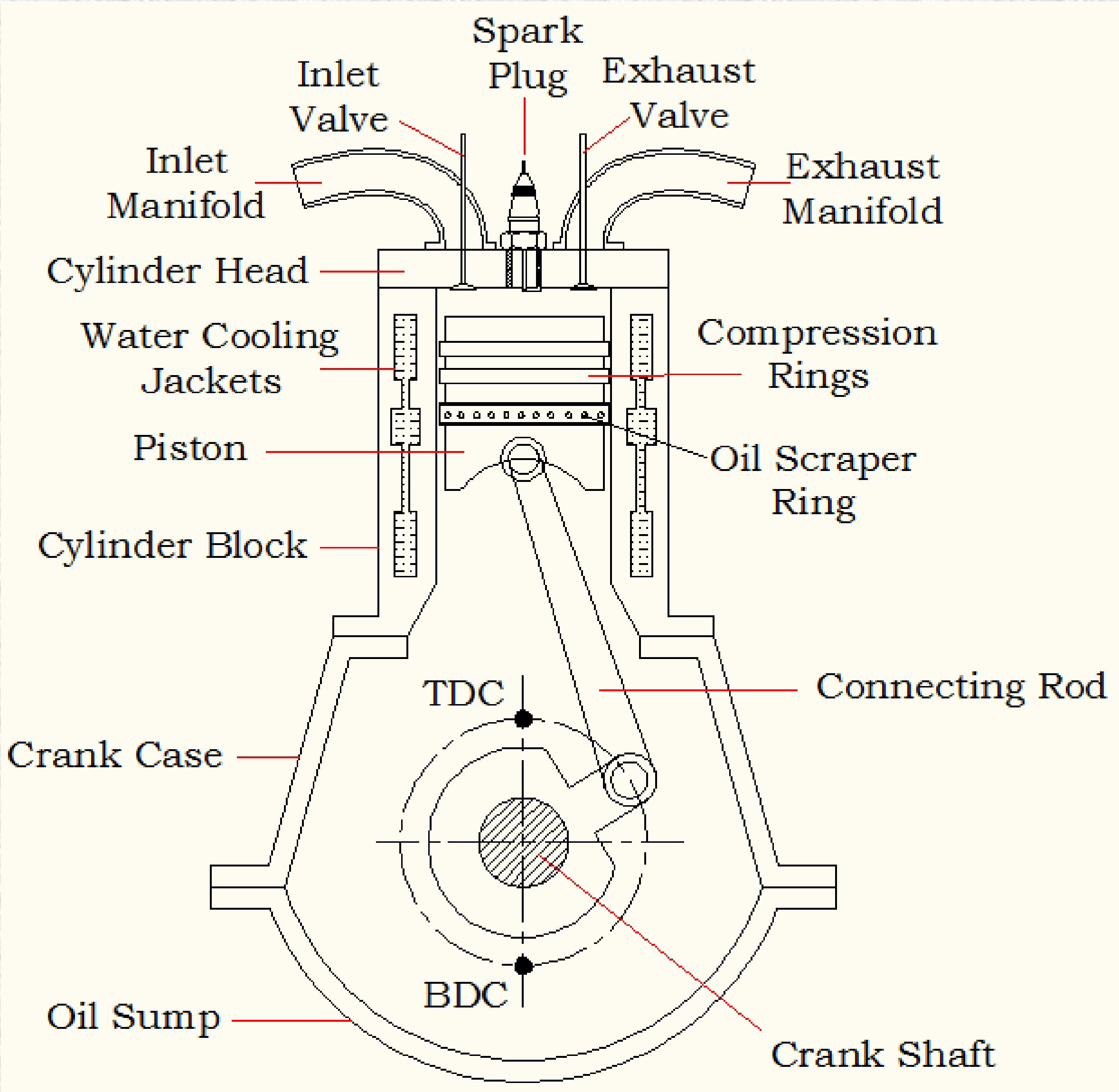
Chemical energy of the fuel is first converted to thermal energy by means of combustion or oxidation with air outside the engine system. This thermal energy raises the temperature and pressure of the gases of the working fluid (ie gases or steam), and the high-pressure working fluid then expands against the mechanical mechanisms of the engine. This expansion is converted by the mechanical linkages of the engine to a rotating crankshaft, which is the output of the engine.

2.3. Comparison of IC and EC Engines : SPPU : May-17, 6-Marks

IC Engine	EC Engine
1. Combustion of fuel take place inside the engine cylinder.	1. Combustion of fuel take place outside the engine cylinder.
2. Working fluid is itself the product of combustion of fuel.	2. Working fluid is other fluid than the product of combustion of fuel.
3. Power to Weight ratio is more as overall weight of engine is less due to absence of boiler and condenser units.	3. Power to Weight ratio is less as overall weight of engine is more due to presence of boiler and condenser units.
4. Compact in size as compared to EC engines.	4. Non compact in size as compared to IC engines.
5. Higher Thermal efficiency (35 to 40%), due to less heat loss.	5. Lower thermal efficiency (20%), due more heat loss.
6. Required high starting torque.	6. Do not required starting torque.
7. It required high grade fuels only.	7. It can use solid, liquid or gaseous fuels.
8. These engines are not self starting.	8. These engines are self starting.

9. Applications : Wankel engines, Open cycle gas turbine, Gasoline and Diesel Automobile engines etc.	9. Applications : Steam engines, closed cycle gas turbine, sterling engines etc.

3. IC Engine Construction :



3.1. IC Engine Components and its materials : SPPU : Dec.-14, 6-Marks

3.1.1. Cylinder and Cylinder Block :

Cylinder is a cylindrical vessel or space in which the piston moves. The change in volume during piston movement filled with working fluid and subjected to different thermodynamic processes of engine cycle.

The cylinder is supported in the cylinder block, which is provided with water cooling jackets in case of water cooled engine.

Cylinder block supported the cylinder heat at top and crank case at the bottom with the help of gasket and bolting.

Material required for the cylinder and cylinder bock must possess good casting properties, good thermal conductivity, corrosion resistance and creep resistance. These are usually made up of Grey Cast Iron and Aluminium Alloys.

3.1.2. Piston and Piston Rings :

It is a cylindrical component fitted into the cylinder forming the moving boundary of the combustion system. It fits perfectly into cylinder providing a gas tight space with the piston rings and layer of lubricating oil.

Pistons rings are of two types, compression rings mounted on top side slots on piston which provide a tight seal between the piston and the cylinder to prevent any pressure leakage to bottom side of piston. And other oil scrapper ring mounted on bottom side slot on piston which provide to remove excess lubricating oil from cylinder wall.

Material required for the piston must possess Good wearing quality, light weight, high thermal conductivity, high creep resistance. It is usually made up of Aluminium Alloys containing silicon.

Material required for the piston rings must possess good wear quality, high thermal conductivity and corrosive resistant. It is usually made up of fine grained alloy cast iron containing manganese and silicon.

3.1.3. Inlet and Exhaust 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.

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.

Inlet manifold is usually made up of alluminium alloys, where are exhaust manifold is usually made up on cast iron.

3.1.4. Inlet and Exhaust Valves :

Inlet and Exhaust valves are commonly mushroom shaped poppet 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.

Material required for the valves must possess good wear quality, high thermal conductivity and corrosive resistant. It is usually made up of fine grained alloy steel.

3.1.5. Spark Plug :

It is the component used in Spark Ignition Engine, which is used to ignite and initiate the combustion process toward the end of compression stroke.

It having two electrode with gap when high tension voltage passing through it during the jump through gap it produces spark which causes ignition of charge.

It is usually mounted on the cylinder head and covered with mica / porcelain material.

3.1.6. Connecting Rod :

It interconnects the piston and the crankshaft and transmits 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.

The small end is connected to the piston by gudgeon pin and the big end is connected to the crankshaft by crankpin.

Material required for the connecting rod must possess high strength, light weight, high fatigue strength etc. It is usually made up of steel, duralumin, malleable cast iron.

3.1.7. Crank Shaft :

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.

Crankshaft enclosed in crankcase. The material required for it must possess high strength, toughness, hardness and fatigue strength, and it is normally made up of chromium-vanadium steel or Cr-Mo steel.

3.1.8. Camshaft and Cams :

Cam shaft and its parts like push rods, rocker arms, valve springs and tappets etc. control the opening and closing of the two valves. This shaft also provide the drive to the ignition system. The camshaft is driven by the crankshaft through timing gears.

Cams are made as integral parts of the camshaft and are so designed to open the valves at the correct timing and to keep them open for the necessary duration.

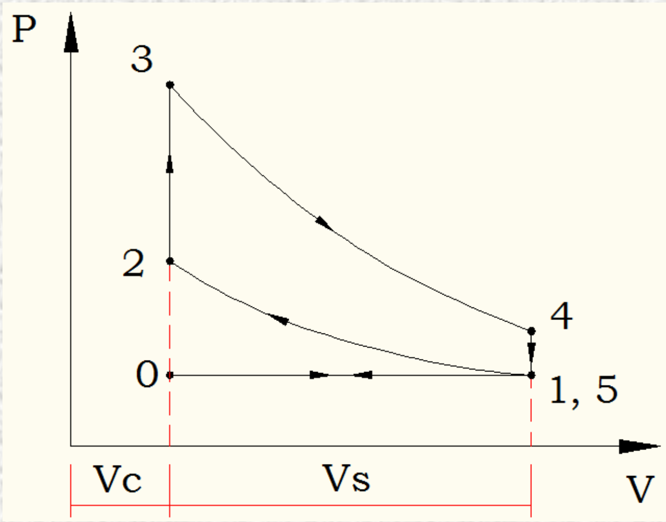
3.1.9. Flywheel :

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.

It is made up steel or cast iron disc.

5.1. Working Principle of 4-Stroke SI Engine :

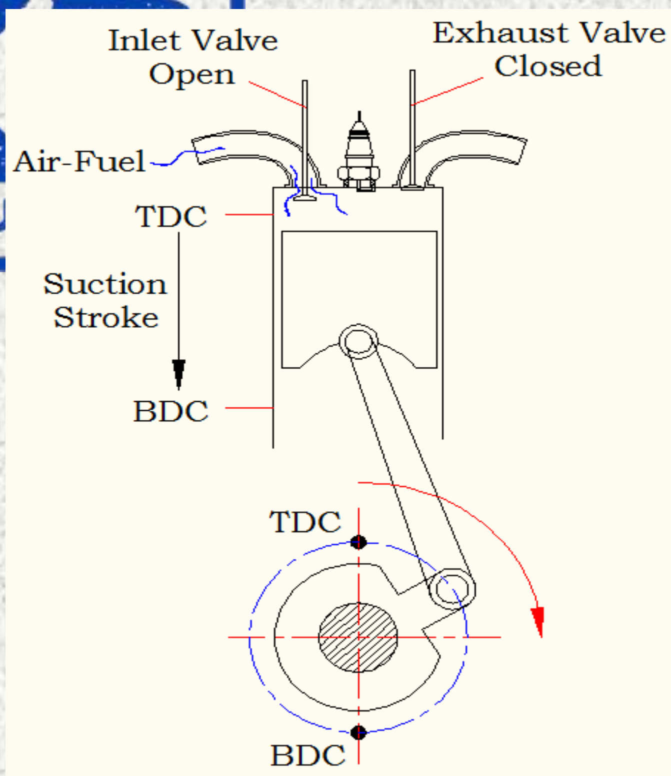
Four Stroke Spark Ignition (SI) engine working on the principle of Otto Cycle, as shown, during the operation it is assumed that inlet and exhaust valves are open and closed instantaneously which is not the actual case.



5.1.1. Suction / Intake Stroke :

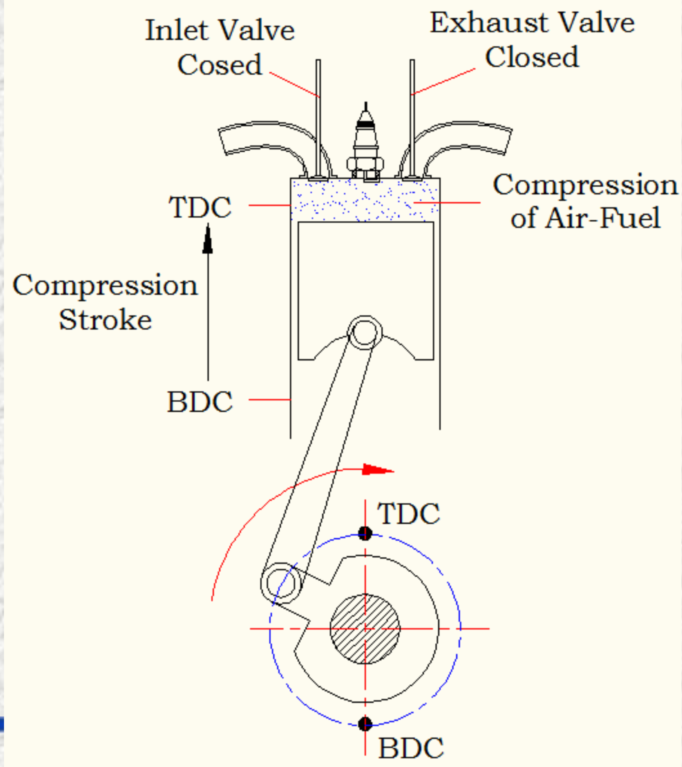
Suction / Intake stroke shown by process 0→1 on PV diagram, is started when piston is at the TDC and moves down to BDC, during this piston moves through one stroke and crank moves through 180° rotation or half revolution.

In suction stroke, inlet valve open and exhaust valve remain closed, thus when piston moves down, it create negative pressure causes air-fuel mixture to drawn into the cylinder. This is continue till piston reach the BDC, where inlet valve get closed and complete cylinder filled with entrapped air-fuel mixture.



5.1.2. Compression Stroke :

Compression stroke shown by process 1→2 on PV diagram, is started when piston is at the BDC at the end of suction stroke. Piston moves up to TDC, during this piston moves through one stroke and crank moves through 180° rotation more.

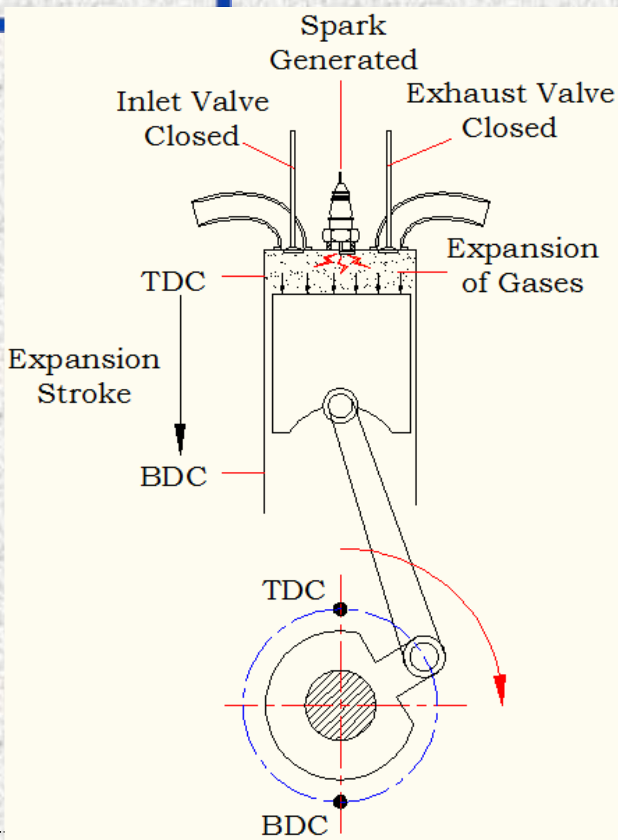


In compression stroke, both, inlet valve and exhaust valve remain closed, thus when piston moves up, it compresses the entrapped air-fuel mixture. This is continue till piston reach the TDC, where entire cylinder volume air-fuel mixture is compressed into the clearance volume, thus it attend high pressure and temperature.

At the end of the compression stroke the mixtures ignited with the help of a spark plug. The burning process converted chemical energy of the fuel into heat energy, thus it can be approximated as heat additional at constant volume, shown by process 2→3 on PV diagram.

5.1.3. Expansion / Power Stroke :

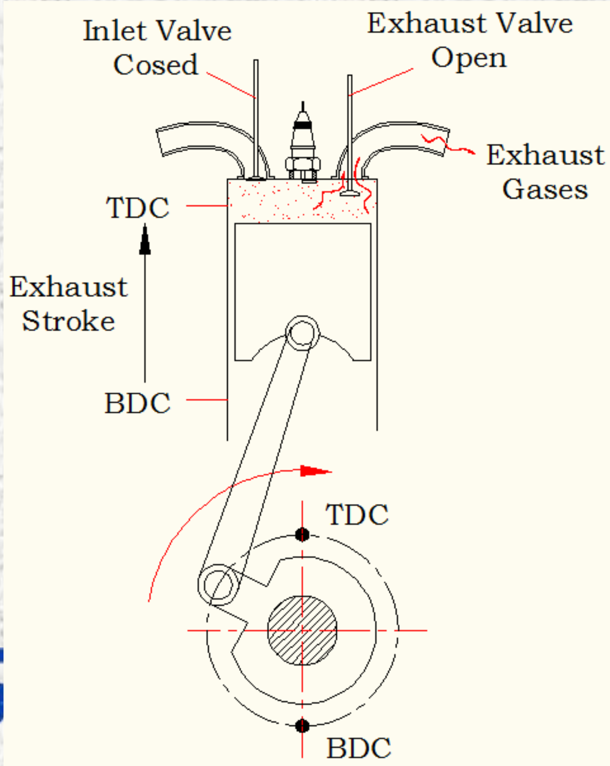
Expansion / Power stroke shown by process 3→4 on PV diagram, is started when pressure at the end of the combustion process is considerably increased due to the heat released from fuel. The high pressure of the burnt gases forces the piston to move from TDC to BDC. During this piston moves through one stroke and crank moves through 180° rotation more, upto this crank complete one and half revolution.



In expansion stroke, both, inlet valve and exhaust valve remain closed, thus high pressure gases expands it forces piston to move down to BDC, thus actual power is obtained in this stroke hence some time it is called as power stroke. This is continue till piston reach the BDC, where entire cylinder volume is filled with burnt gases.

5.1.4. Exhaust Stroke :

Exhaust stroke shown by process 5→0 on PV diagram. At the end of the expansion stroke high pressure of the burnt gases filled cylinder volume, at this exhaust valve is open, thus high pressure gases escape from the valve and the pressure falls to atmospheric level at the constant volume, this is shown by the process 4→5, on PV diagram.



The piston start to move from BDC to TDC. During this piston moves through one stroke and crank moves through 180° rotation more.

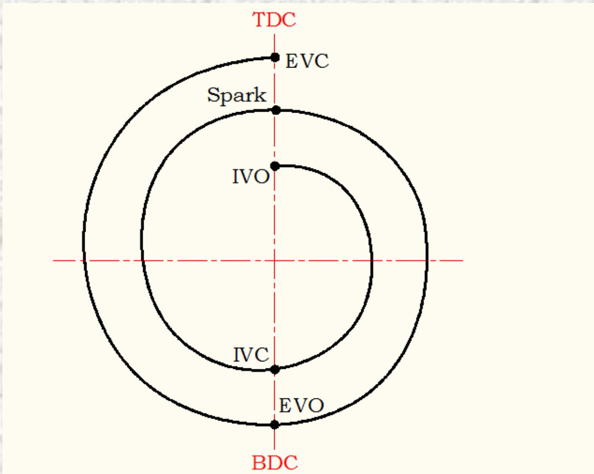
In exhaust stroke, inlet valve remain closed but exhaust valve get opened, thus piston upward motion pushes out the burnt gases from the cylinder. This is continue till piston reach the TDC, where exhaust valve get closed and new cycle suction stroke will start as piston moves from TDC again.

Thus Four Stroke SI engine completes one cycle through 720° rotation or two revolution of crank or in four stroke of piston.

5.2. Valve Timing Diagram of 4-Stroke SI Engine : SPPU : May-18, Dec.-15, May-14, 6-Marks

Valve timing is the regulation of the points in the cycle at which the valves are set to open and close.

In ideal cycle inlet and exhaust valves open and closed at dead centre, and each operation suction, compression, expansion and exhaust exhibits in one stroke of piston i.e. the 180° each. The spark is ignite at TDC and there is instantaneous combustion and expansion starts from TDC only.



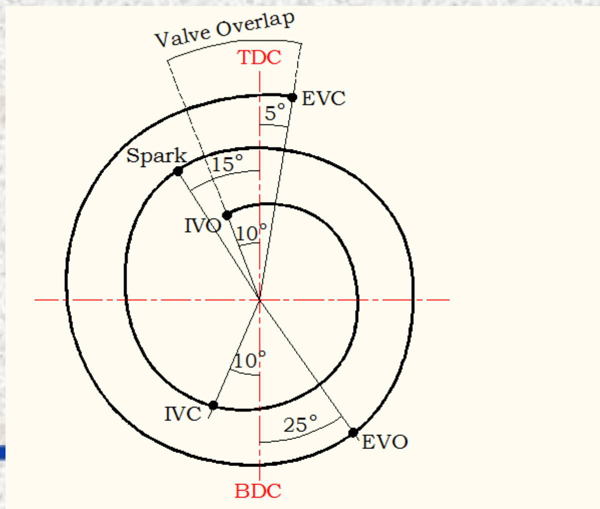
Ideal Valve Timing Diagram

IVO - Inlet Valve Open IVC - Inlet Valve Closed
EVO - Exhaust Valve Open EVC - Exhaust Valve Closed
Spark - Spark Ignition

But in actual cycle they open and closed before or after dead centre. Four Stroke SI engine working on the principle of Otto Cycle, and during the operation its inlet and exhaust valves, open and closed and spark plug actuate to ignite compressed air-fuel mixture, in a sequence and at a particular crank angle, this shown by the valve timing diagram as follow,

Low Speed Engine SI Engine -

Inlet Valve Open (IVO) 10° before Top Dead Centre (bTDC) and as piston moves down air-fuel mixture suck inside the cylinder, this continue till Inlet Valve Closed (IVC) at 10° after Bottom Dead Centre (aBDC). Thus Inlet Valve Opens for total 200° of crank rotation, during this suction take place.



Actual Valve Timing Diagram - Low Speed SI Engine
IVO - Inlet Valve Open IVC - Inlet Valve Closed
EVO - Exhaust Valve Open EVC - Exhaust Valve Closed
Spark - Spark Ignition

After Inlet Valve Closed (IVC) entrapped fresh Air-Fuel mixture get compressed due to piston upward motion toward TDC, as both valves are closed.

Near TDC Spark Plug actuated and spark is generated (Spark) at 15° before TDC. The period from IVC to Spark is a compression and from Spark to TDC air-fuel mixture start burning with the compression till the piston reach to the TDC.

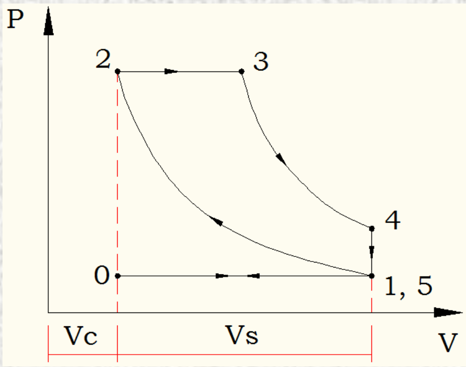
From the point TDC to Exhaust Valve Open (EVO) at 25° before BDC, the time period is called a expansion and as EVO the burnt gases start to escape through the opening of exhaust valve.

Exhaust Valve Closes at 5° after TDC, thus from BDC to EVC the piston movement push out the exhaust gases from the cylinder.

As Valve timing diagram shows, EVC at 5° after TDC but the next cycle IVO 10° before TDC, thus Inlet and Exhaust Valve, both remain open at common crank angle interval for the 15° this is called valve overlap.

6.1. Working Principle of 4-Stroke CI Engine :

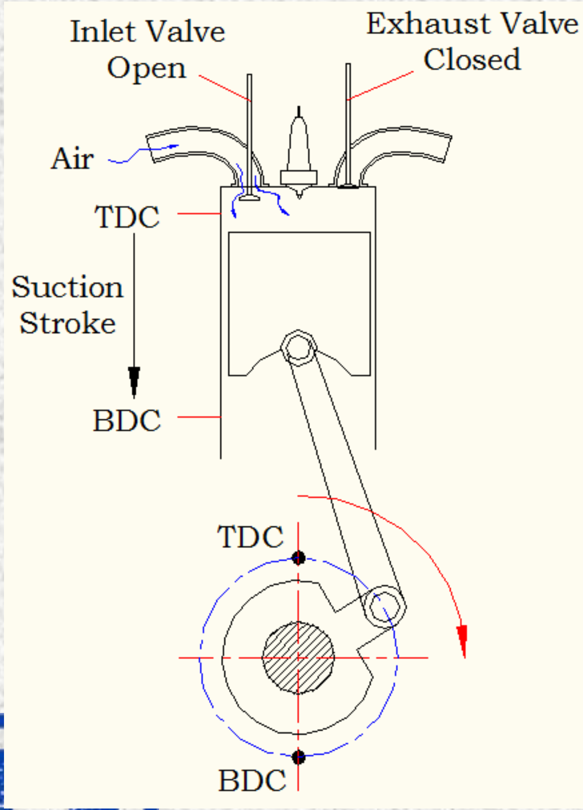
Four Stroke Compression Ignition (CI) engine working on the principle of Diesel Cycle, as shown, during the operation it is assumed that inlet and exhaust valves are open and closed instantaneously which is not the actual case.



6.1.1. Suction / Intake Stroke :

Suction / Intake stroke shown by process 0→1 on PV diagram, is started when piston is at the TDC and moves down to BDC, during this piston moves through one stroke and crank moves through 180° rotation or half revolution.

In suction stroke, inlet valve open and exhaust valve remain closed, thus when piston moves down, it create negative pressure causes air to drawn into the cylinder. This is continue till piston reach the BDC, where inlet valve get closed and complete cylinder filled with entrapped air.

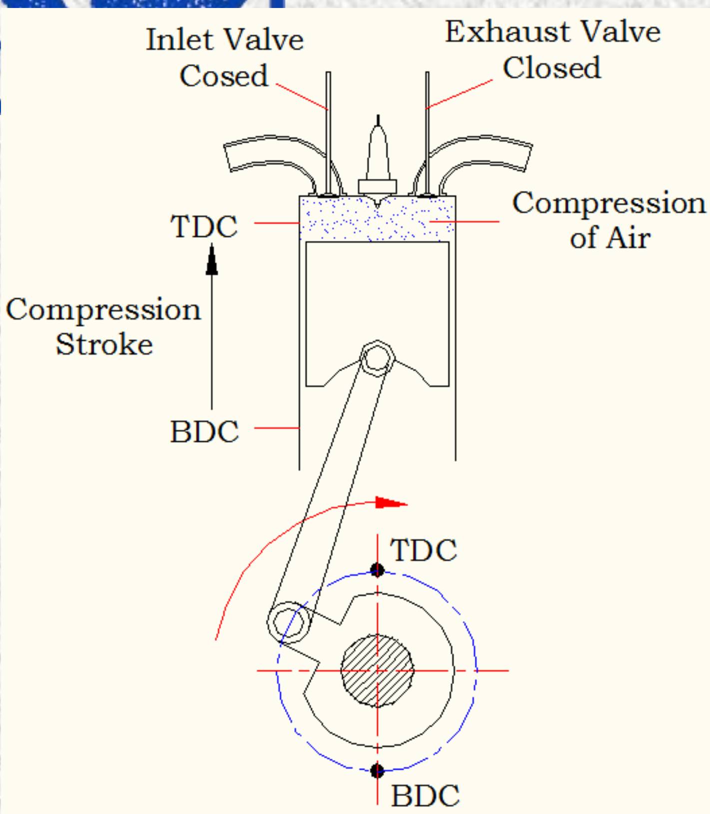


6.1.2. Compression Stroke :

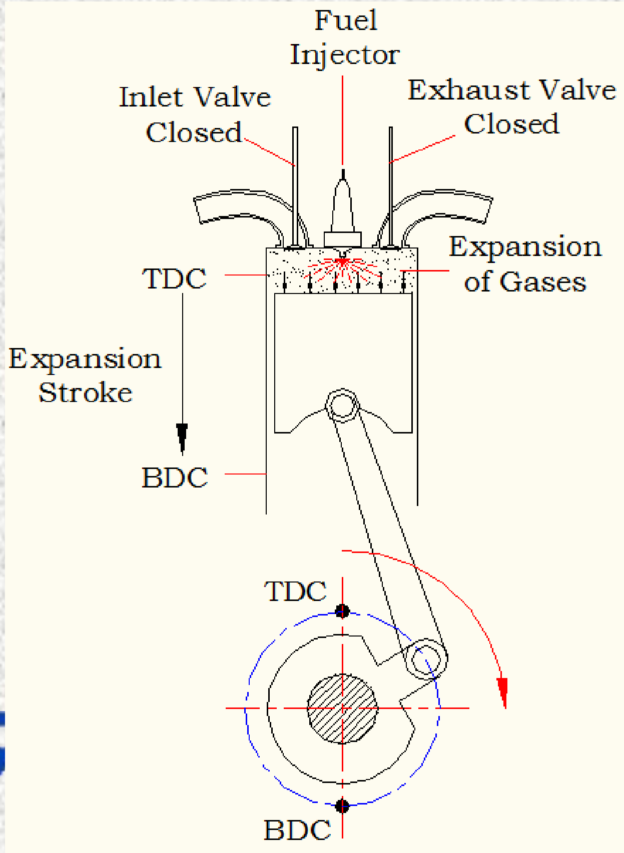
Compression stroke shown by process 1→2 on PV diagram, is started when piston is at the BDC at the end of suction stroke. Piston moves up to TDC, during this piston moves through one stroke and crank moves through 180° rotation more, thus crank complete one revolution.

In compression stroke, both, inlet valve and exhaust valve remain closed, thus when piston moves up, it compresses the entrapped air. This is continue till piston reach the TDC, where entire cylinder volume air is compressed into the clearance volume, thus it attend high pressure and temperature.

At the end of the compression stroke the fuel is injected with the help of fuel pump and fuel injector.



6.1.3. Expansion / Power Stroke :

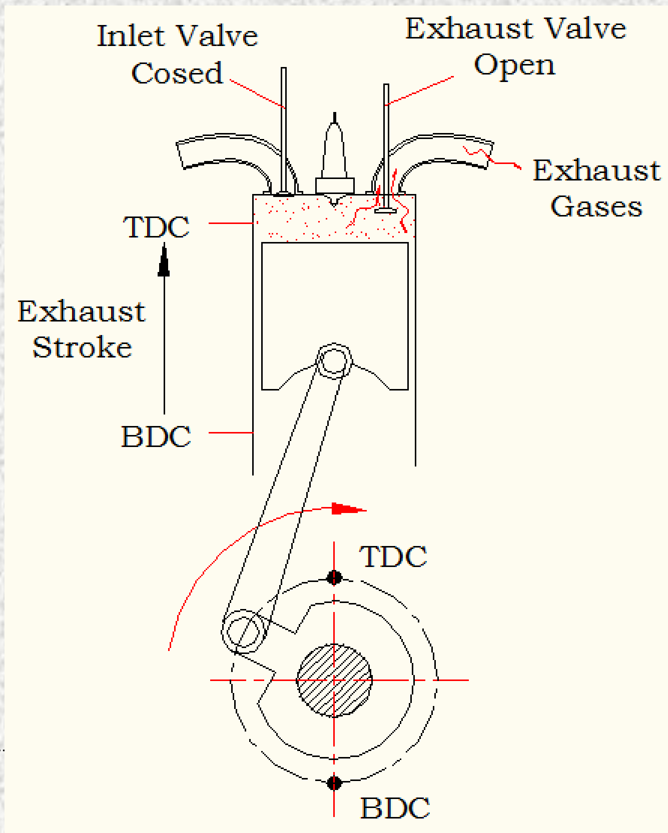


Fuel injection starts at the end of compression stroke, but the rate of injection of fuel is such that the combustion maintains the pressure constant. Thus heat is added at the constant pressure, which is shown by process 2→3 on PV diagram.

Expansion / Power stroke shown by process 3→4 on PV diagram, is started when fuel injection stops. Pressure of the product of combustion is considerably increased due to the heat released from fuel. The high pressure of the burnt gases forces the piston to move from TDC to BDC. During this piston moves through one stroke and crank moves through 180° rotation more, upto this crank complete one and half revolution.

In expansion stroke, both, inlet valve and exhaust valve remain closed, thus high pressure gases expands it forces piston to move down to BDC, thus actual power is obtained in this stroke hence some time it is called as power stroke. This is continue till piston reach the BDC, where entire cylinder volume is filled with burnt gases.

6.1.4. Exhaust Stroke :



Exhaust stroke shown by process 5→0 on PV diagram. At the end of the expansion stroke high pressure of the burnt gases filled cylinder volume, at this, exhaust valve is open, thus high pressure gases escape from the valve and the pressure falls to atmospheric level at the constant volume, this is shown by the process 4→5, on PV diagram.

The piston start to move from BDC to TDC. During this piston moves through one stroke and crank moves through 180° rotation more and crank complete two revolutions.

In exhaust stroke, inlet valve remain closed but exhaust valve get opened, thus piston upward motion pushes out the burnt gases from the cylinder. This is continue till piston reach the TDC, where exhaust valve get closed and new cycle suction stroke will start as piston moves from TDC again.

Thus Four Stroke SI engine completes one cycle through 720° rotation or two revolution of crank or in four stroke of piston.

6.2. Valve Timing Diagram of 4-Stroke CI Engine : SPPU : Dec.-17, 6-Marks

Valve timing is the regulation of the points in the cycle at which the valves are set to open and close.

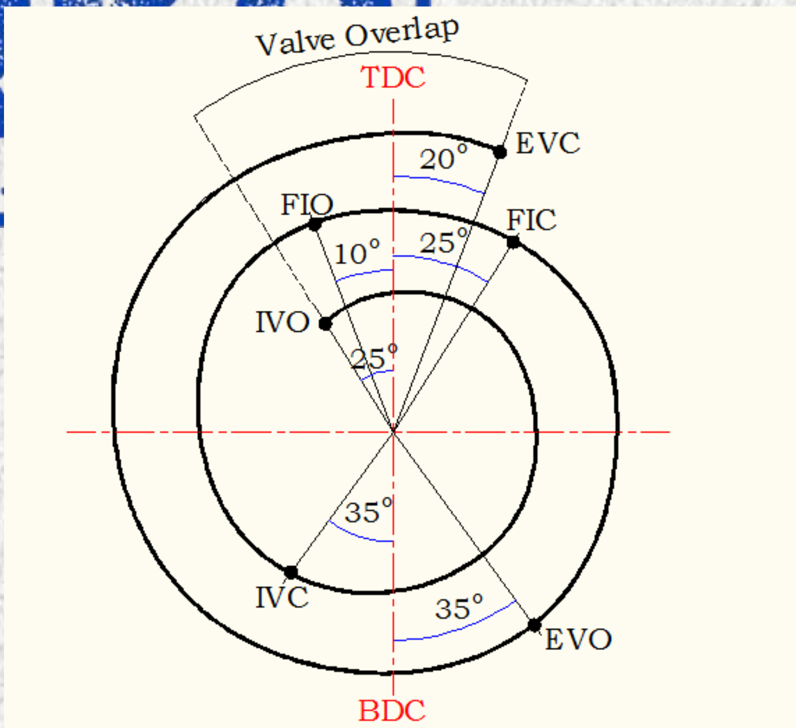
In ideal cycle inlet and exhaust valves open and closed at dead centre, but in actual cycle they open and closed before or after dead centre.

Four Stroke Compression Ignition (CI) engine working on the principle of Diesel Cycle, and during the operation it is inlet and exhaust valves, open and closed and fuel injection start and stop in a sequence and at a particular crank angle, this shown by the valve timing diagram.

Low Speed CI Engine -

Inlet Valve Open (IVO) 25° before Top Dead Centre (bTDC) and as piston moves down air suck inside the cylinder, this continue till Inlet Valve Closed (IVC) at 35° after Bottom Dead Centre (aBDC). Thus Inlet Valve Opens for total 240° of crank rotation, during this suction take place.

After Inlet Valve Closed (IVC) entrapped fresh Air get compressed due to piston upward motion toward TDC, as both valves are closed.



Actual Valve Timing Diagram - Low Speed CI Engine

IVO - Inlet Valve Open IVC - Inlet Valve Closed
EVO - Exhaust Valve Open EVC - Exhaust Valve Closed
FIO - Fuel Injection Open FIC - Fuel Injection Closed

Near TDC Fuel Injector Open (FIO) and start injecting fuel at 10° before TDC and Fuel Injector Closed (FIC) at 25° after TDC. The period from IVC to TDC is a compression and from FIO to FIC it is a fuel injection period, during this, fuel mixes with compressed air and start burning at constant pressure.

From the point FIC to Exhaust Valve Open (EVO) at 35° before BDC, the time period is called a expansion and as EVO the burnt gases start to escape through the opening of exhaust valve.

Exhaust Valve Closes at 20° after TDC, thus from BDC to EVC the piston movement push out the exhaust gases from the cylinder.

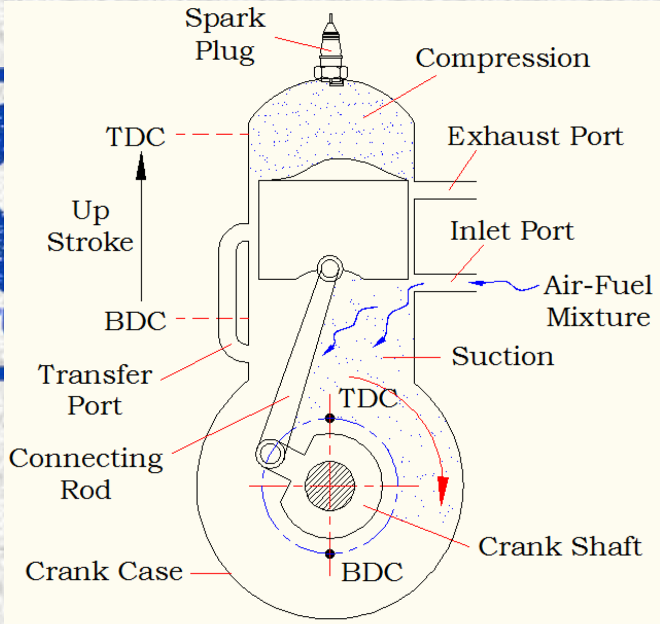
As Valve timing diagram shows, EVC at 20° after TDC but the next cycle IVO 25° before TDC, thus Inlet and Exhaust Valve, both remain open at common time interval for the 45° this is called valve overlap.

7.1. Working Principle of 2-Stroke SI Engine :

In two stroke engines one working cycle is completed into one revolution of a crank. Thus two piston stroke are required to generate a power in engine cycle. The two stroke of SI engine are explained as below.

7.1.1. Up Stroke :

In Two Stroke Engine Valves are replaced with ports, Inlet Port, Exhaust Port and Transfer Port. The piston have dome shape at its top. Its crank case is air tight and working as a suction unit. Initially the piston is at BDC.



The arrangement of the ports is such that the piston performs two operations simultaneously. When the piston starts rising from BDC it closes the transfer port and the upward movement of piston causes vacuum at crankcase, on further upward movement of piston opens Inlet Port and fresh charge get sucked inside the crankcase due to negative pressure inside the cylinder. As piston moves further it closes exhaust port and the already existing charge at top side of piston is get compressed.

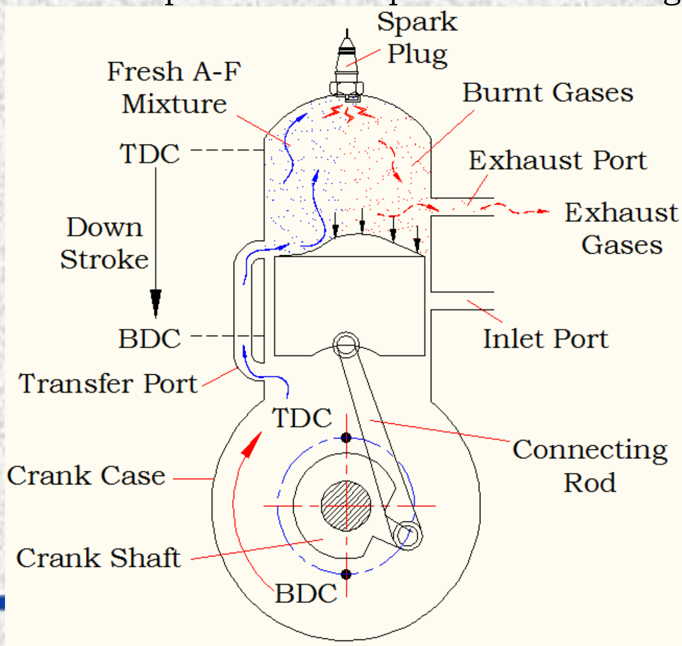
Thus in Up Stroke at bottom side of Piston suction in completed and at top side of piston compression is completed. At the end of this First / Up Stroke piston reaches to TDC, hence piston moves from BDC to TDC and crank rotates through 180° or Half Revolution.

7.1.2. Down Stroke :

In this stroke piston moves from TDC to BDC. Before the completion of compression stroke, the compressed charge is ignited using Spark Plug in Spark Ignition Engine (

and by using fuel injection in Compression Ignition Engine), and the gas pressure is exerted on the crown of the piston.

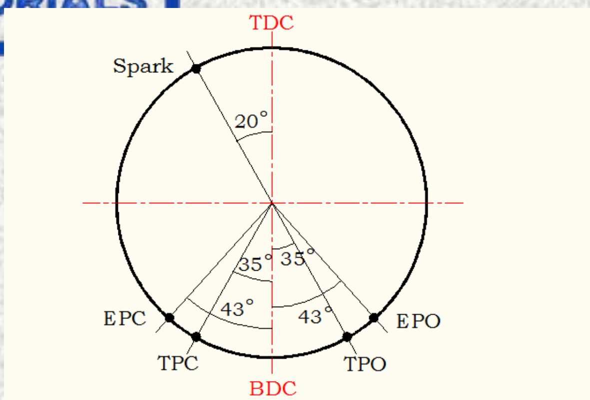
This pressure, forces the piston in downward direction which produces useful work. The downward movement of piston closes the inlet port and compresses the charge already sucked in the crankcase. When piston moves from TDC to BDC it first opens exhaust port and pressurized exhaust gases start to escape from it.



On further downward movement of piston Transfer Port is open the fresh charge from crankcase is entered into combustion chamber such that it forces residual exhaust gases to escape through exhaust port. And fill the combustion area with fresh charge. Thus during down stroke piston moves from TDC to BDC and Expansion / Power and Exhaust are performed. Crank rotates 360° or one complete rotation.

7.2. Port Timing Diagram of 2-Stroke SI Engine : SPPU : Dec.-17, Dec.-14, 6-Marks

Port timing is the regulation of the points in the cycle at which the port are set to open and close, this is done by movement of piston.



TPO - Transfer Port Open
EPO - Exhaust Port Open
Spark - Spark Ignition
TPC - Transfer Port Closed
EPC - Exhaust Port Closed

Two Stroke Spark Ignition (SI) engine working on the principle of Otto Cycle, and during the operation its inlet and exhaust port, open and closed and spark plug actuate and to ignite compressed air-fuel mixture, in a sequence and at a particular crank angle, this shown by the port timing diagram.

In Up stroke, piston moves from BDC to TDC, during its movement, piston closes first Transfer Port (TPC) at 35°after BDC, and on further upward movement, piston closes Exhaust Port (EPC), now the entrapped air-fuel mixture on top of the piston get compressed. This is continue till piston reach TDC.

Before the end of compression, Spark is actuated at 20° before TDC and compressed air-fuel mixture ignited.

Thus from EPC to TDC the compression of charge take place.

Burnt gases tried to expand and exerted pressure on piston and pushes it down, thus expansion starts and it continue till Exhaust Port Opened.

EPO at 43° before BDC and exhaust gases escape from the port. On further downward movement of piston, Transfer Port (TPO) get opened at 35° before BDC and fresh air-fuel mixture rushes inside the cylinder, which scavenges burnt gases from the cylinder, this continue till Transfer port closed (TPC).

8.1. Comparison of 4-stroke and 2-Stroke Engine :

4-Stroke Engine	2-Stroke Engine
1. The thermodynamic cycle is completed in four strokes of the piston or 720° of rotation or two revolution of crankshaft.	1. The thermodynamic cycle is completed in two strokes of the piston or 360° of rotation or one revolution of crankshaft.
2. Power obtained after four strokes, as, Suction, Compression, Expansion and Exhaust.	2. Power obtained after two strokes, as Up stroke – contained Suction and Compression operations, and Down Stroke – contained Expansion and Exhaust operation.
3. As one power stroke obtained after each two revolution of crankshaft, hence as compared to 2-stroke power produced for the same size of engine is less.	3. As one power stroke obtained after each revolution of crankshaft, hence as compared to 4-stroke power produced for the same size of engine is more.
4. As one power stroke obtained after each two revolution of crankshaft, hence as compared to 2-stroke engine lesser cooling and lubrication required.	4. As one power stroke obtained after each revolution of crankshaft, hence as compared to 4-stroke engine greater cooling and lubrication required.
5. It contains valves and valve actuating mechanisms to open and close the valves.	5. It have no valves, but only ports, and opening and closing of the port controlled by piston movement.
6. For the same power engine is heavier and bulkier, and complicated valve mechanism, causes initial cost of the engine more.	6. For the same power engine is lighter and more compact, simple port mechanism, causes initial cost of the engine is less.
7. Volumetric Efficiency is higher as compared to 2-stroke engine, due to more time for induction.	7. Volumetric Efficiency is lower as compared to 4-stroke engine, due to less time for induction.

8. Thermal Efficiency is higher as compared to 2-storke engine.	8. Thermal Efficiency is lower as compared to 4-storke engine.
9. Heavier flywheel is needed, as turning moment is not so uniform due to one power obtained after two revolution.	9. Lighter flywheel is needed, as turning moment is uniform due to one power obtained after each revolution.
10. Application – It is used where efficiency is important, like in cars , buses, trucks, tractors, industrial engines etc.	10. Application – It is used where low cost, light weight and compactness is important, like in mopeds, scooters, motorcycles, hand sprayers etc.

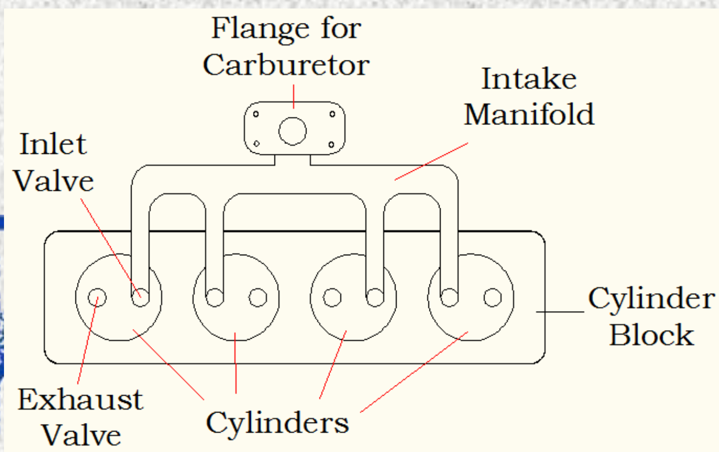
8.2. Comparison of SI and CI Engine :

Spark Ignition (SI) Engine	Compression Ignition (CI) Engine
1. It work on Otto Cycle or constant volume heat addition cycle.	1. It work on Diesel Cycle or constant pressure heat addition cycle.
2. It uses gasoline, which is highly volatile fuel, whose self ignition temperature is high as compared to diesel.	2. It uses diesel, which is non-volatile fuel, whose self ignition temperature is low as compared to gasoline.
3. A mixture of air-fuel is supplied to engine during suction stroke.	3. Only air is supplied to engine during suction stroke.
4. Throttle valve controls the quantity of air-fuel mixture introduced to cylinder.	4. Fuel pump controls the quantity of fuel injected into compressed air, inducted air quantity is not controlled.
5. These are quantity governed engines.	5. These are quality governed engines.
6. It requires ignition system with spark plug.	6. It is self-ignition due to which ignition system and spark plug are not required.
7. It have compression ratio, 6 to 10, upper limit is limited by knocking increases of the engine.	7. It have compression ratio, 16 to 20, upper limit is limited by weight increases of the engine.

8. It have homogeneous combustion, as air-fuel homogeneous mixture is supplied to cylinder.	8. It have heterogeneous combustion, as only air supplied to cylinder and fuel is injected and mixes to for combustible mixture.
9. For the same power, it is light weight, hence these are high speed engine.	9. For the same power, it is heavy weight, hence these are low speed engine.
10. Low thermal efficiency, because of lower compression ratio.	10. Higher thermal efficiency, because of higher compression ratio.

9.1. Intake System (Intake Manifold) :

The intake system / manifold is a casting or assembly of passages which carries the mixture of fuel and air in carbureted SI engine from the carburetor to the intake valve ports of the engine cylinder.



The carburetor is mounted on the inlet of the intake manifold. The outlet of intake manifold is mounted on the cylinder block at the entry of air-fuel mixture to cylinder.

The components of intake manifold system for four cylinder engine is as shown.

The function of the intake manifold are as follows,

1. It provide a branched path from single carburetor to intake of many cylinder of multi-cylinder engine.
2. It sized and shaped and orientation to cause maximum charge inducted into the engine cylinder to improve the volumetric efficiency, power and thermal efficiency.
3. It minimizes condensation of fuel and assist vaporization of the gasoline in the mixture, by accommodating heating arrangement, uses exhaust gases or cooling water from engine or by providing hot spots in the manifolds.
4. It supply almost equal amount of homogeneous air-fuel mixture to all cylinder.

The design requirement for intake manifold are as follows,

1. The friction in the manifolds increases the pressure losses in the pipe. Thus reduced pressure at intake to cylinders increases pumping losses. Therefore design should minimizes the pressure losses.

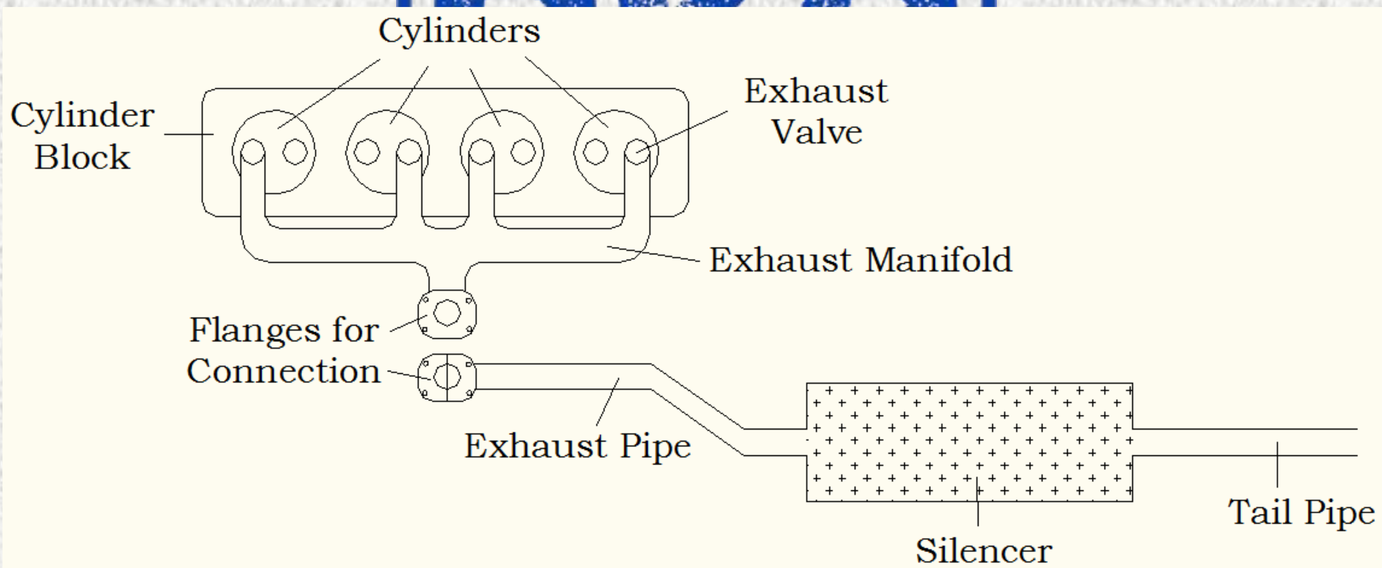
2. Mixture when passing through intake manifold at corner and direction change fuel particles get condensed. Therefore design of shape and size of manifold should prohibit the formation of fuel droplets without restricting the air flow.
3. Mixture having some heavy particles of fuel, which condensed and separate out at manifold surface, thus usually rich mixture is available at nearer cylinder and richness reduced toward farthest cylinder. Therefore design should distribute the air-fuel mixture uniformly to each cylinder over wide range of speeds and loads.
4. Mixture having some heavy liquid fuel particles thus mixture become heterogeneous. Therefore design should assist vaporization of fuel droplets by heating arrangement using exhaust gas or cooling water.

The material used for intake manifold are as follows,

1. Old engines were using cast iron material for intake manifold.
2. Modern engines use die-cast aluminium material for intake manifold.

9.2. Exhaust System (Exhaust Manifold) :

The exhaust system / manifold is a casting or assembly of passages which carries the burnt exhaust gases, during exhaust stroke, from the cylinders of engine and transfer to silence and then exhaust to atmosphere via tail pipe.



The exhaust manifold is mounted on the exhaust passage at exhaust valve. The outlet of exhaust manifold is connected to silencer by exhaust pipe. The components of intake manifold system for four cylinder engine is as shown below,

The function of the exhaust manifold are as follows,

1. It collect exhaust gases from all cylinder and exhausted to atmosphere.
2. It withstand the high temperature of exhaust gases, and transfer minimum heat to the vehicle body.

3. It reduces back pressure so as to reduced the power loss in discharging the gases.
4. It also reduces noise and vibration generated by the engine using the silencer in the system.
5. Also clean up the emissions which are hazardous to the environment with the help of silencer and muffler of the system.

The design requirement for exhaust manifold are as follows,

1. It should discharge burnt gases quickly from engine cylinder to the atmosphere without creating any back pressure.
2. It should avoid overlapping of exhaust strokes by dividing the manifolds into branches.
3. It should avoid restriction of flow, by using large radius bends with large cross-sectional area of pipe.
4. It should connected to the cylinder head by flexible connections to permit free expansion and contraction.
5. It should take care of the variation of exhaust gas temperature and speed of flow at different load and speed of engine.

The material used for exhaust manifold is a Cast Iron to withstand high temperature of exhaust gases and used as vibration and noise damping.

10. Engine Classification :

Internal combustion engines are classified under the various head as given below,

10.1. According to Fuel used –

a) **Volatile Liquid Fuel Engine –** Engine uses highly volatile liquid fuel like gasoline, alcohol, kerosene, benzene etc.

b) **Non-volatile Liquid Fuel Engine –** Engine uses low or non-volatile liquid fuel at normal atmospheric temperature, like diesel fuel.

c) **Gaseous Fuel Engine –** Engine uses gaseous fuel like, compressed natural gas (CNG), liquefied petroleum gas (LPG), blast furnace gas and biogas etc.

d) **Duel Fuel Engine –** These engines are used two fuel, homogeneous mixture of primary fuel with air is supplied to engine and near the end of compression pilot fuel is injected and carried out compression ignition.

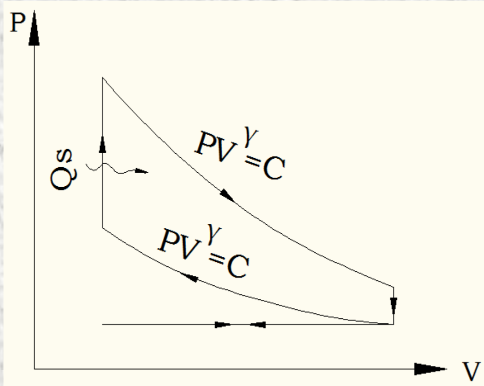
10.2. According to Mechanical Cycle used –

a) **Two Stroke Cycle Engine –** One cycle completed in two stroke (Up stroke and Down stroke) of the piston, or one revolution of crankshaft or 360° rotation of crank.

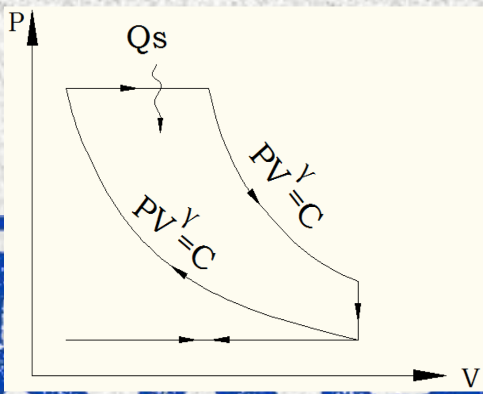
b) **Four Stroke Cycle Engine -** One cycle completed in four stroke (Suction, Compression, Expansion and Exhaust stroke) of the piston, or two revolution of crankshaft or 720° rotation of crank.

10.3. According to Thermodynamic Cycle used –

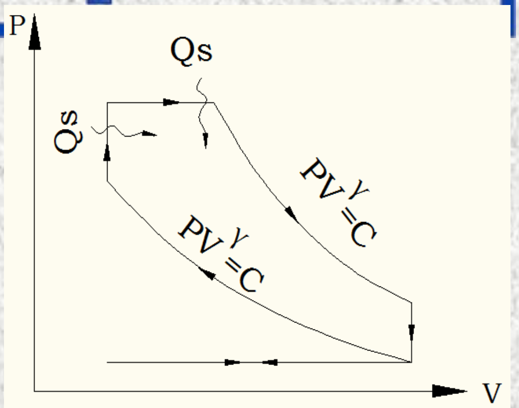
a) **Otto Cycle Engine –** It work on constant volume heat addition thermodynamic cycle, and used in spark ignition engines.



b) **Diesel Cycle Engine –** It work on constant pressure heat addition thermodynamic cycle, and used in compression ignition engines.



c) **Duel Cycle Engine –** It work on partially constant volume heat addition and partially constant pressure heat addition thermodynamic cycle.



10.4. According to Method of Ignition used –

a) **Spark Ignition Engine (SI engine) –** These uses highly volatile fuel, the mixture of air-fuel is ignited near the end of compression with the help of spark generated by spark plug.

b) **Compression Ignition Engine (CI engine) –** These uses low volatile fuel, fuel is injected near the end of compression of air, the mixture thus form reaches the self ignition temperature and ignited.

10.5. According to Method of Cooling used –

a) **Air Cooled Engine –** Heat of engine is dissipated from the engine with the help of air blow due to vehicle motion.

b) **Water Cooled Engine –** Heat of engine is dissipated from the engine with the help of water, using water jacket and radiator arrangement.

c) **Liquid (Coolant) Cooled Engine –** Heat of engine is dissipated from the engine with the help of coolant whose coefficient of heat convection is higher than water, using cooling jackets and radiator arrangement.

10.6. According to Movement of Piston used –

a) Reciprocating Piston Engine – This engine have a cylinder and piston arrangement, piston reciprocates inside the cylinder, and completing required operations (suction, compression, expansion and exhaust) for a cycle. Example of it is petrol, diesel and gas internal combustion engines etc.

b) Rotary Piston Engine – This engine have a casing and rotor arrangement, rotor rotates in a casing and complete the required operations for a cycle. Example of it is open cycle gas turbine engines etc.

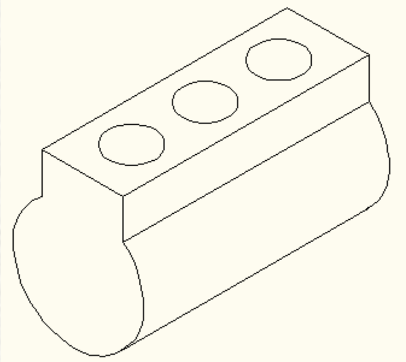
10.7. According to Numbers of Cylinders used –

a) **Single Cylinder Engine –** These engines have only one cylinder, thus producing less power, hence used in two wheeler vehicles like motor cycle, scooter etc.

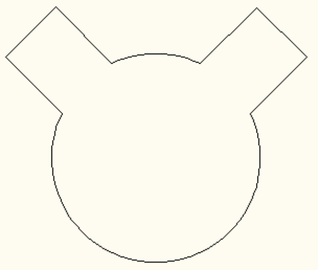
b) **Multi Cylinder Engine –** These engines have more than one cylinder, thus producing high power, hence used in all four wheeler vehicles like cars, buses, truck etc.

10.8. According to Cylinder Arrangement used –

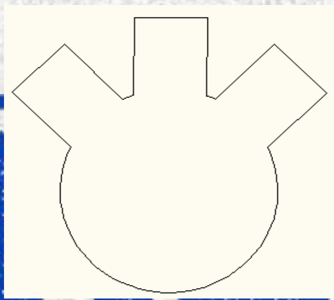
a) **Inline Cylinder Engine –** In this type of multi-cylinder engine all cylinders are arranged in a single row, thus number of cylinders are limited by the engine length size.



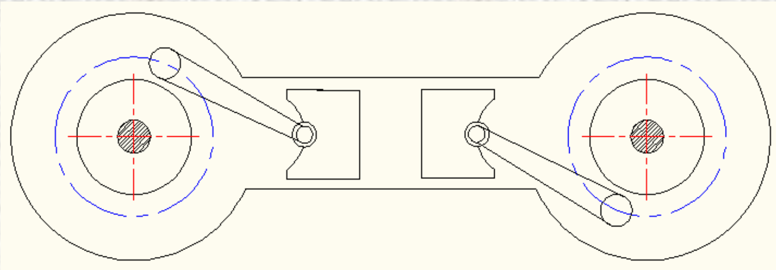
b) V-Type Engine – In this type of multi-cylinder engines all cylinders are arranged in two row in the form of V, thus for the same length of engine number of cylinder and in turn power produced is more than inline cylinder engine.



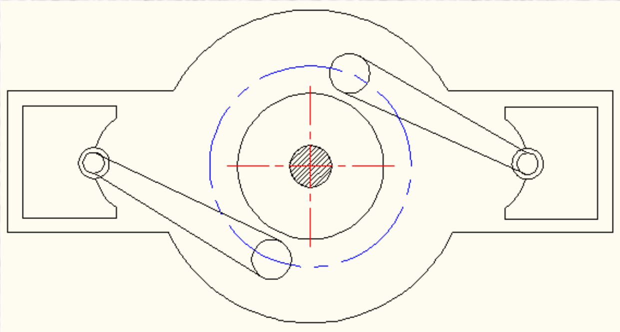
c) W-Type Engine - In this type of multi-cylinder engines all cylinders are arranged in three row in the form of W, thus for the same length of engine number of cylinder and in turn power produced is more than inline cylinder engine and V type engine.



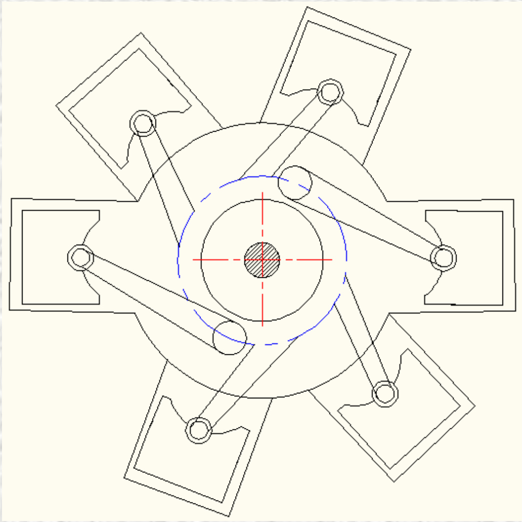
d) Opposed Piston Engine – In this type of engines, single cylinder houses two pistons, each of which driving a separate crankshaft.



e) Opposed Cylinder Engine - In this type of engines, two cylinder located in the same plane on opposite sides of the single crankshaft.



f) Radial Engine – Radial engine is one where more than two cylinders in each row are equally spaced around the crankshaft. Piston of all the cylinders are coupled to the same crankshaft.



g) Horizontal Engine - In this type of engines, have a cylinder and piston arrangement horizontally, piston reciprocates inside the cylinder from inner most (IDC - Inner Dead Centre) to outer most position (ODC – Outer Dead Centre).

h) Vertical Engine - In this type of engines, have a cylinder and piston arrangement vertically, piston reciprocates inside the cylinder from bottom most (BDC - Bottom Dead Centre) to top most position (TDC – Top Dead Centre).

10.9. According to Method of Governing used –

a) Quantity Governing Engine – These engines control the output of engine with the help of controlling quantity of homogeneous mixture supplied to engine. Usually used in SI engines.

b) Quality Governing Engine - These engines control the output of engine with the help of controlling quality of the air-fuel mixture formed, by controlling the fuel injected inside the cylinder. Usually used in CI engines.

c) Hit and Miss Governing Engine – These engines control the output of the engine with the help of short circuiting spark plug. When spark plug short circuited, then it can not provide spark to ignite mixture of the cylinder thus, overall power produced is reduced, in turn engine output reduced. Thus by hitting and missing the spark of cylinders engine output is control.

11. Air-Standard Cycle :

An internal combustion engine follows the sequence of processes, suction, compression, expansion and exhaust. It does not operate on thermodynamic closed cycle, as it suck fresh charge and exhausted burnt gases, thus it is open system.

The accurate analysis of IC engine is very complicated, due to working fluid entering and exit conditions are different and not follow the closed cycle.

In order to analyze the performance of an ideal closed cycle that closely approximate the real cycle, the approach used is air-standard cycle, in which air is assumed to be a working fluid, which have same state at entry and exit.

11.1. Assumptions made for air-standard cycle -

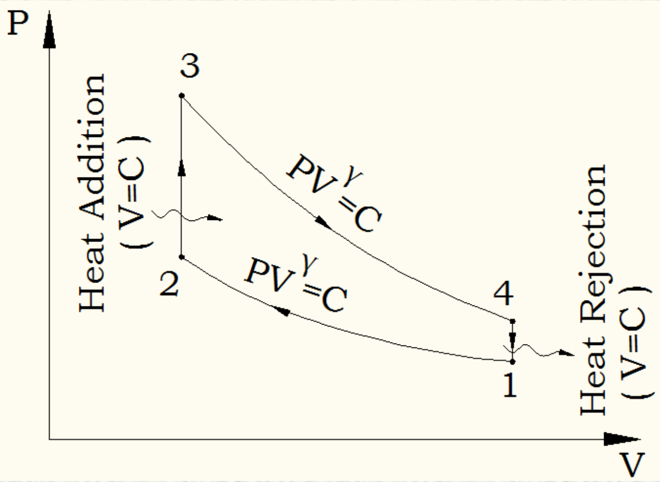
1. The working medium is assumed to be a perfect gas (air) and follows the relation, PV = mRT.
2. There is no change in the mass of the working medium at entry and exit.
3. All the processes of cycle are reversible.
4. Heat is supplied from heat reservoir (high temperature source) and not from chemical reactions during the cycle.
5. Some heat is assumed to be rejected to heat sink (low temperature sink).
6. It is assumed that there is no heat loss from system to the surroundings.
7. The working fluid (air) has constant specific heats throughout the cycle.
8. Working fluid constant, Cp, Cv, γ, M are assumed as of air at standard atmospheric conditions, Cp = 1.005 Kj/Kg K, Cv = 0.717 Kj/Kg K, γ = 1.4 and M = 29 Kg/Kmol.

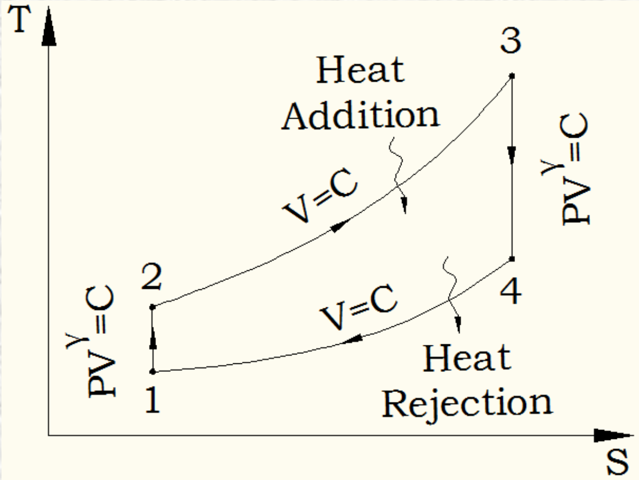
11.2. Limitations of Air-Standard Cycle -

1. Due to all above assumptions, the analysis becomes over simplified and thus the result do not match with those of the actual cycle.
2. Work output, peak pressure, peak temperature and thermal efficiency based on air-standard cycles will be the maximum that can be attained and will differ considerably from those of the actual cycle.
3. Its result are only the approximate to the complicated processes in internal combustion actual cycle engine.
4. Its working fluid is air and not a mixture of air-fuel, hence the effect of variation air-fuel ratio can not be approximated.

11.3. Analysis of Air-Standard Otto Cycle :

Spark Ignition (SI) engines operates on Otto cycle, which represented on P-V and T-S diagram as follow,





Process 1-2 : Reversible adiabatic compression or isentropic compression during which air is compressed from state-1 to state-2, thus volume is reducing and corresponding pressure and temperature increases but entropy remain constant.

Process 2-3 : Heat addition process to air from a heat reservoir from stat-2 to state-3, during this pressure, temperature and entropy increases but volume remain constant.

Process 3-4 : Reversible adiabatic expansion or isentropic expansion during which air is expanded from state-3 to state-4, thus volume increases and corresponding pressure and temperature decreases but entropy remain constant.

Process 4-1 : Heat rejection process from air to a heat sink from state-4 to state-1, thus system return to its original state, during this pressure, temperature and entropy reduces but volume remain constant.

1. Heat supplied = $m \times C_v \times \Delta T$

   $$Q_s = m \times C_v \times (T_3 - T_2)$$

2. Heat rejected = $m \times C_v \times \Delta T$

   $$Q_R = m \times C_v \times (T_4 - T_1)$$

3. Net Work done, $W = Heat\ supplied - Heat\ rejected$

   $$W = Q_s - Q_R$$

   $$W = m \times C_v \times (T_3 - T_2) - m \times C_v \times (T_4 - T_1)$$

4. Compression ratio, $r = \frac{V_1}{V_2}$

   For adiabatic compression 1-2,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \quad \ldots..Comp. Ratio, r = \frac{V_1}{V_2}$$

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

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

5. Expansion ratio, $r_e = \frac{V_4}{V_3}$

For adiabatic expansion 3-4,

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} \quad \ldots\ldots as, V_3 = V_2 \; and \; V_4 = V_1$$

$$\frac{T_4}{T_3} = \left(\frac{V_2}{V_1}\right)^{\gamma-1}$$

$$\frac{T_3}{T_4} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \quad \ldots..Comp. Ratio, r = \frac{V_1}{V_2}$$

$$\frac{T_3}{T_4} = (r)^{\gamma-1} \quad \ldots..Comp. Ratio, r = \frac{V_1}{V_2}$$

$$T_4 = \frac{T_3}{(r)^{\gamma-1}}$$

6. *Air Standard Efficiency,*

$$\eta_{air\ std} = \frac{Net\ Work\ done}{Heat\ supplied}$$

$$\eta_{air\ std} = \frac{W}{Q_s}$$

$$\eta_{air\ std} = \frac{m \times C_v \times (T_3 - T_2) - m \times C_v \times (T_4 - T_1)}{m \times C_v \times (T_3 - T_2)}$$

$$\eta_{air\ std} = 1 - \frac{m \times C_v \times (T_4 - T_1)}{m \times C_v \times (T_3 - T_2)}$$

$$\eta_{air\ std} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

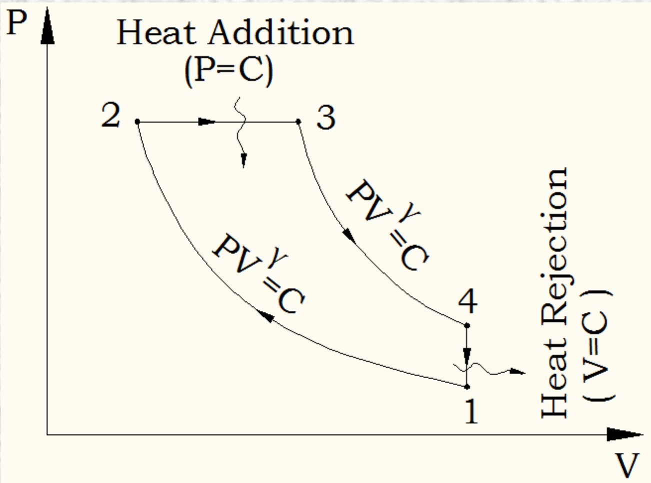
$$\eta_{air\ std} = 1 - \frac{\left(\frac{T_3}{(r)^{\gamma-1}} - \frac{T_2}{(r)^{\gamma-1}}\right)}{(T_3 - T_2)}$$

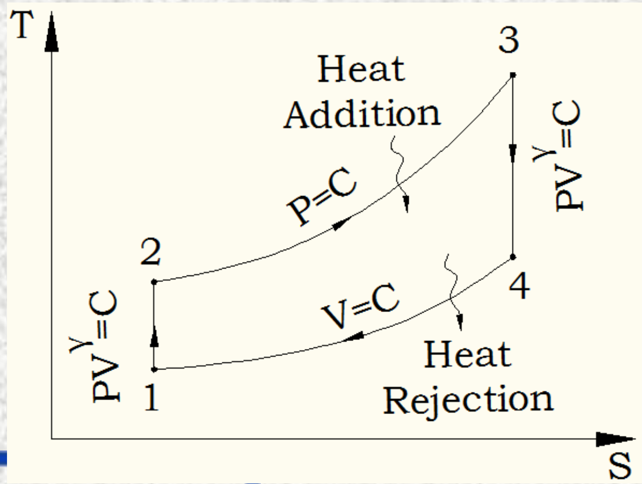
$$\eta_{air\ std} = 1 - \frac{(T_3 - T_2)}{(r)^{\gamma-1}\,(T_3 - T_2)}$$

$$\eta_{air\ std} = \eta_{otto} = 1 - \frac{1}{(r)^{\gamma-1}}$$

11.4. Analysis of Air-Standard Diesel Cycle :

Compression Ignition (CI) engines operates on Diesel cycle, which represented on P-V and T-S diagram as follow,





Process 1-2 : Reversible adiabatic compression or isentropic compression during which air is compressed from state-1 to state-2, thus volume is reducing and corresponding pressure and temperature increases but entropy remain constant.

Process 2-3 : Heat addition process to air from a heat reservoir from stat-2 to state-3, during this volume , temperature and entropy increases but pressure remain constant.

Process 3-4 : Reversible adiabatic expansion or isentropic expansion during which air is expanded from state-3 to state-4, thus volume increases and corresponding pressure and temperature decreases but entropy remain constant.

Process 4-1 : Heat rejection process from air to a heat sink from state-4 to state-1, thus system return to its original state, during this pressure, temperature and entropy reduces but volume remain constant.

1. Heat supplied = $m \times C_p \times \Delta T$

   $Q_s = m \times C_v \times (T_3 - T_2)$

2. Heat rejected = $m \times C_v \times \Delta T$

   $Q_R = m \times C_v \times (T_4 - T_1)$

3. Net Work done, $W = Heat\ supplied - Heat\ rejected$

   $W = Q_s - Q_R$

   $W = m \times C_p \times (T_3 - T_2) - m \times C_v \times (T_4 - T_1)$

4. Compression ratio, $r = \frac{V_1}{V_2}$

For Isentropic compression,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \quad \ldots..Comp.Ratio, r = \frac{V_1}{V_2}$$

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

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

5. Cut-off ratio, $r_c = \frac{V_3}{V_2}$

For constant pressure heat addition,

$$\frac{P_2\ V_2}{T_2} = \frac{P_3\ V_3}{T_3}$$

$$T_3 = T_2 \frac{P_3\ V_3}{P_2\ V_2} \ldots\ldots as, P_2 = P_3$$

$$T_3 = T_2 \frac{V_3}{V_2}$$

$$T_3 = T_2\,(r_c) \ldots\ldots..Put, T_2 = T_1\,(r)^{\gamma-1}$$

$$T_3 = T_1\,(r)^{\gamma-1}\,(r_c)$$

6. Expansion ratio, $r_e = \frac{V_4}{V_3}$

Multiply and Divided RHS by V_2

$$r_e = \frac{V_4}{V_3} \times \frac{V_2}{V_2}$$

$$r_e = \left(\frac{V_4}{V_2}\right) \times \frac{1}{\left(\frac{V_3}{V_2}\right)} \ldots\ldots V_4 = V_1$$

$$r_e = \left(\frac{V_1}{V_2}\right) \times \frac{1}{\left(\frac{V_3}{V_2}\right)}$$

$$r_e = \frac{r}{r_c}$$

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} \ldots\ldots cut-off\ ratio, r_e = \frac{V_4}{V_3}$$

$$\frac{T_4}{T_3} = \frac{1}{(r_e)^{\gamma-1}}$$

$$T_4 = T_3 \frac{1}{\left(\frac{r}{r_c}\right)^{\gamma-1}} \ldots\ldots\ldots as, r_e = \frac{r}{r_c}$$

$$T_4 = T_3 \frac{(r_c)^{\gamma-1}}{(r)^{\gamma-1}} \ldots. Put, T_3 = T_1 (r)^{\gamma-1} (r_c)$$

$$T_4 = T_1 (r)^{\gamma-1} (r_c) \times \frac{(r_c)^{\gamma-1}}{(r)^{\gamma-1}}$$

$$T_4 = T_1 (r_c) \times (r_c)^{\gamma-1}$$

$$T_4 = T_1 (r_c)^{1+\gamma-1}$$

$$T_4 = T_1 (r_c)^{\gamma}$$

7. $Air\ Standard\ Efficiency, \eta_{air\ std} = \frac{Net\ Work\ done}{Heat\ supplied}$

$$\eta_{air\ std} = \frac{W}{Q_s}$$

$$\eta_{air\ std} = \frac{m \times C_p \times (T_3 - T_2) - m \times C_v \times (T_4 - T_1)}{m \times C_p \times (T_3 - T_2)}$$

$$\eta_{air\ std} = 1 - \frac{m \times C_v \times (T_4 - T_1)}{m \times C_p \times (T_3 - T_2)}$$

$$\eta_{air\ std} = 1 - \frac{C_v (T_4 - T_1)}{C_p (T_3 - T_2)}$$

$$\eta_{air\ std} = 1 - \frac{1}{\left(\frac{C_p}{C_v}\right)} \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$\eta_{air\ std} = 1 - \frac{1}{\gamma} \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$\eta_{air\ std} = 1 - \frac{1}{\gamma} \frac{[T_1 (r_c)^{\gamma} - T_1]}{[T_1 (r)^{\gamma-1} (r_c)] - (T_1 (r)^{\gamma-1})}$$

$$\eta_{air\ std} = 1 - \frac{1}{\gamma} \frac{(r_c^{\gamma} - 1)}{(r)^{\gamma-1} (r_c - 1)}$$

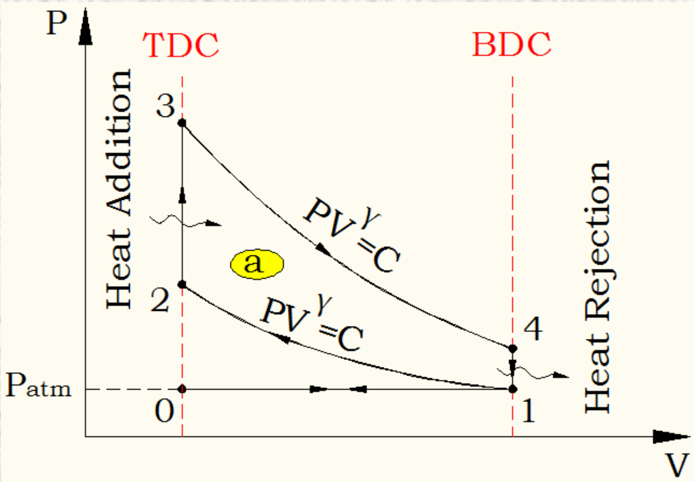
$$\eta_{air\ std} = 1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{r_c^{\gamma} - 1}{\gamma (r_c - 1)}\right]$$

$$\boldsymbol{\eta_{air\ std} = \eta_{diesel} = 1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{r_c^{\gamma} - 1}{\gamma (r_c - 1)}\right]}$$

12. Theoretical and Actual P-V diagrams for 4-Stroke SI Engine :

Theoretical and Actual P-V diagrams for SI engine working on Otto Cycle.

12.1. Theoretical Otto Cycle :



a) The suction stroke (0-1) and exhaust stroke (4-0) are at atmospheric pressure.

b) Each stroke, suction (0-1), compression (1-2), expansion (2-3) and exhaust (4-0) are take place during 180° of crank rotation.

c) Compression and expansion processes are reversible adiabatic.

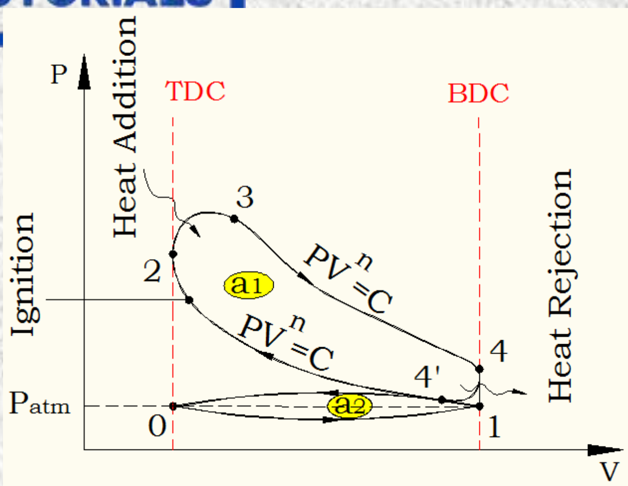
d) Heat addition process (2-3) and heat rejection process (4-1) are both at constant volume and takes place instantly. There is no unintended heat loss.

e) The air behave as a perfect gas.

f) It is assumed that there is no effect of friction and viscosity.

g) Work developed in theoretical cycle is equal to the area (a), There is no pumping losses.

12.2. Actual Otto Cycle :



a) During the suction stroke (0-1) the mixture of fuel and air is admitted instead of only air. And the process is carried below atmospheric pressure.

b) Bunt gases escape to the atmosphere, the exhaust stroke (4-0) is above atmospheric pressure.

c) The compression process (1-2) and expansion process (3-4) are polytrophic due to heat transfer between the system and surrounding.

d) The combustion process (2-3) is carried out by providing a spark to compressed mixture few degree before TDC. Pressure rise take place through some crank rotation.

e) Exhaust at point 4 is not instantaneous since there is a time interval in opening the exhaust valve.

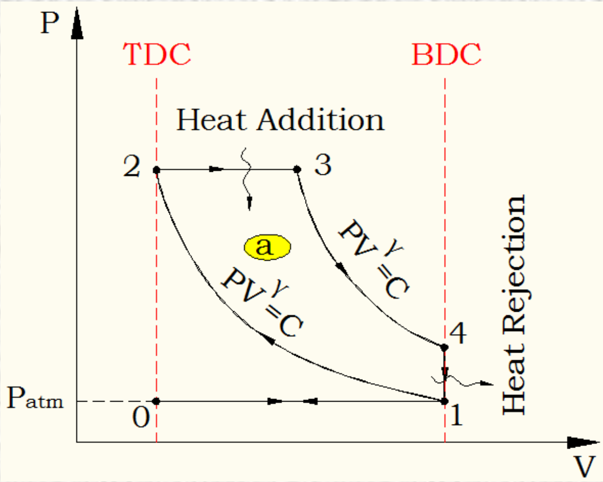
f) Work developed in actual cycle is equal to the area (a1 – a2), where 'a2' is the pumping losses. And processes are not ideal.

Prof. Rahul Krishnaji Bawane
Pimpri Chinchwad College of Engineering & Research, Ravet, Pune.

13. Theoretical and Actual P-V diagrams for 4-Stroke CI Engine :

Theoretical and Actual P-V diagrams for CI engine working on Diesel Cycle.

13.1. Theoretical Diesel Cycle :



a) The suction and exhaust strokes are at atmospheric pressure. During suction stroke (0-1) only air is sucked.

b) Each of the stroke, suction (0-1), compression (1-2), expansion (2-3) and exhaust (4-0) are take place during 180° of crank rotation.

c) Compression (1-2) and expansion (3-4) are reversible adiabatic.

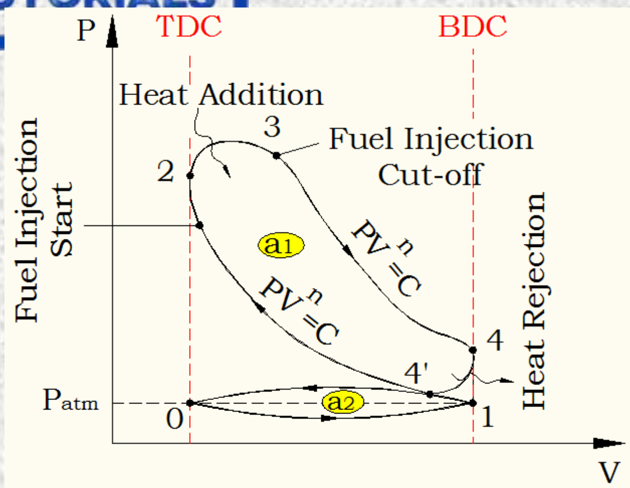
d) Heat addition process (2-3) at constant pressure from heat reservoir and heat rejection process (4-1) at constant volume to heat sink. There is no other intended heat loss.

e) Air behave like a perfect gas.

f) It is assumed that there is no effect of friction and viscosity.

g) Work developed in theoretical cycle is equal to the area (a), There is no pumping losses.

13.2. Actual Diesel Cycle :



a) During the suction stroke (0-1), air is admitted at a pressure below atmospheric pressure.

b) The exhaust stroke (4-0) is only possible above atmospheric pressure, since the burnt gases escape to atmosphere.

c) The compression process (1-2) and expansion process (3-4) are polytropic due to heat exchanged between system and surrounding.

d) Combustion process (2-3) is due to self ignition of high temperature of mixture fuel-air mixture formed caused by high compression ratio.

e) The fuel is injected few degree before TDC and continue upto the point of cut-off, after TDC. Hence combustion process is not at constant pressure due to continuous fuel injection.

f) Exhaust point 4 is not instantaneous due to time interval in opening the exhaust valve.

g) The processes are not ideal, and fuel-air mixture does not behave like a perfect gas.

h) Work developed in actual cycle is equal to the area (a1-a2), where area 'a2' is the pumping losses.

14. Fuel-Air Cycle :

Air-standard cycle analysis can not bring out the effect of air-fuel ratio on the thermal efficiency because the working medium was assumed to be air only, but in actual cycle, fuel in the cylinder is taken into account and accordingly the working medium will be a mixture of air-fuel.

By fuel-air cycle analysis it will be possible to bring out the effect of fuel-air ratio (air-fuel ratio) on thermal efficiency and also study how the peak pressures and temperatures during the cycle vary with respect to fuel-air ratio.

The fuel-air cycle analysis takes into account the followings,

1. The actual composition of the cylinder gases : the cylinder gases contains fuel, air, water vapour and residual gas. The fuel-air ratio changes during the operation of the engine which changes the relative amounts of CO2, water vapour etc.
2. The variation in the specific heat with temperature : Specific heats increase with temperature except for mono-atomic gases. Therefore the value of $'\gamma'$ also changes with temperature.
3. The effect of dissociation : The fuel and air do not completely combine chemically at high temperatures and this lead to the presence of CO, H_2, H and O_2 at equilibrium conditions.
4. The variations 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.

14.1. Assumptions made for fuel-air cycle -

1. There is no chemical changes in either fuel or air prior to combustion.
2. Subsequent to combustion, the change is always in chemical equilibrium.
3. There is no heat exchange between the gases and the cylinder walls in any process (adiabatic process).
4. There is frictionless compression and expansion processes.
5. Fluid motion inside the cylinder is ignored.
6. It is assumed that fuel is completely vaporized and perfectly mixed with air.
7. It is also assumed that, the burning takes place instantaneously at TDC (at constant volume).

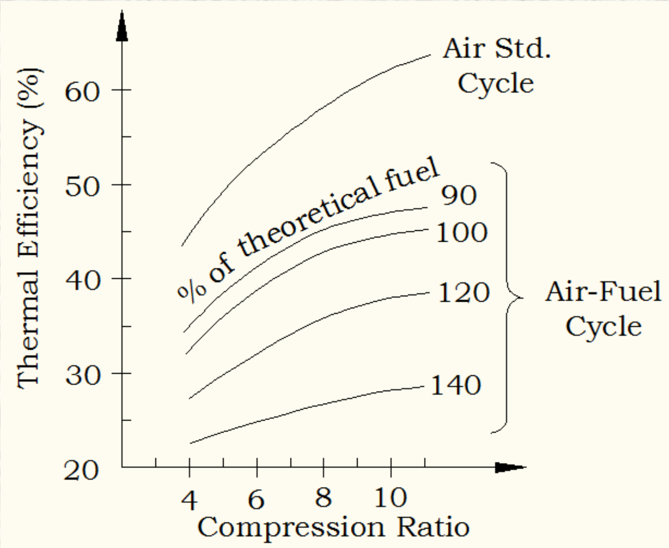
14.2. Comparison of Air-Standard and Fuel-Air cycle :

Air-Standard Cycle	Fuel-Air Cycle
1. Working fluid is assumed to be a pure air.	1. Working fluid is a mixture of fuel and air.
2. Heat is added by heat reservoir.	2. Heat is added due to combustion of fuel-air mixture.
3. Specific heats of gases assumed to be not changes with temperature.	3. Variation of specific heat with temperature is considered. Also dissociation effect at high temperature is considered.
4. There is no change in chemical composition of working fluid at inlet and exit.	4. There is change in chemical composition of working fluid at inlet and exit.
5. Compression and expansion are assumed to be adiabatic.	5. Compression and expansion are not adiabatic.
6. Suction and exhaust process are eliminated.	6. Suction and exhaust process are assumed to be carried out at atmospheric pressure.

15. Effect of Operating Variables on Fuel-Air Cycle :

15.1. Variation of Compression Ratio -

The fuel-air cycle efficiency increases with the compression ratio in the same manner as the air-standard cycle efficiency, because of increases in compression ratio, increases more scope of expansion work.

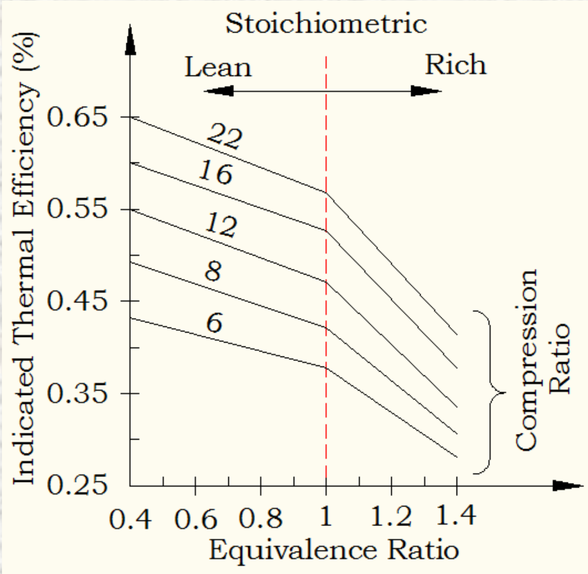


15.2. Variation of Air-Fuel Mixture Strength -

The equivalence ratio is defined as, the ratio of actual fuel-air ratio to chemically correct (stoichiometric) fuel-air ratio on the mass basis.

As the mixture is made lean 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. It will lower the losses due to dissociation and variation in specific heat.

The efficiency is therefore, higher and approaches the air-cycle efficiency as the fuel-air ratio is reduced



15.3. Variation of Specific Heat -

SPPU : Dec.-17, May-16, Dec.-15, 6-Marks

All gases (except mono-atomic gases), shows an increase in specific heat with temperature.

Over the temperature range 300K to 2000K, generally the specific heat curve is nearly a straight line.

$$C_p = a_1 + k_1 T$$

$$C_v = b_1 + k_1 T$$

$$R = C_p - C_v = a_1 - b_1 \quad \text{Also } \gamma = {C_p}/{C_v}$$

Where, a_1, b_1, k_1 are constants, and R is characteristic gas constant.

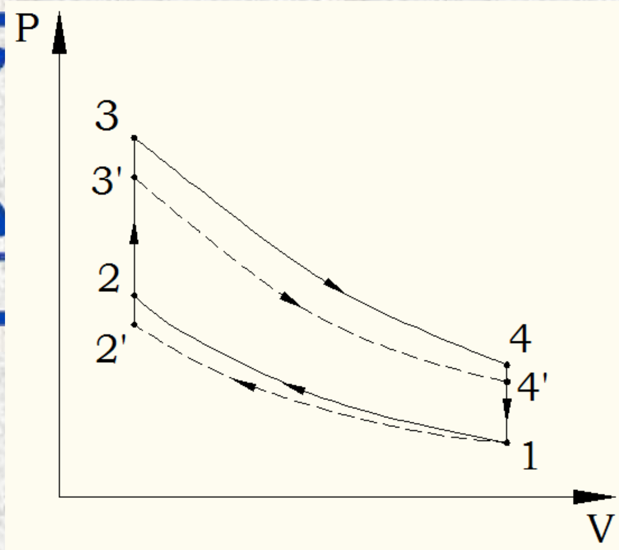


Figure shows of Otto Cycle, where cycle 1-2-3-4 represents the ideal Otto cycle. During the compression process the value of $'\gamma'$ decreases with the increase in temperature due to which the actual temperature of gas after compression will be lower than the ideal compression temperature.

Process 1-2' represents the actual compression process. For the same amount of heat supplied, the maximum pressure and temperature achieved is also lower and it corresponds to state-3' due to increase in specific heat at constant volume.

Process 3'-4' represents the actual expansion process, the value of $'\gamma'$ increases with the decrease in temperature.

Thus the actual cycle is represented by 1-2'-3'-4' with variation in specific heat. The work developed in actual cycle is less than the ideal cycle. The difference of ideal work and the actual work is called the loss of work due to variation in specific heat.

15.4. Variation of Number of Moles -

In an ideal cycle it is assumed that the number of moles of the working substance before and after combustion remains constant, since the heat transfer to and from the working substance were assumed with the help of heat reservoirs.

However, the number of moles present before and after the combustion would be different in case of fuel-air cycle.

Considering following combustion equations,

$$C \quad + \quad O_2 \quad \rightarrow \quad CO_2$$
$$1\,mole \quad 1\,mole \quad \rightarrow \quad 1mole$$

$$2\,H_2 \quad + \quad O_2 \quad \rightarrow \quad 2H_2O$$
$$2\,mole \quad 1\,mole \quad \rightarrow \quad 2mole$$

The total number of moles before and after combustion of fuel are different, the variation in number of moles is commonly known as molecular contraction or expansion.

$$P\,V = n\,R°\,T \;\ldots\ldots R° = Universal\ Gas\ Constant$$

The pressure of the gas is proportional to number of moles at a given temperature and volume. Therefore, the actual pressure in the combustion chamber will be different compared to theoretical cycles due to variation in number of moles caused by the combustion of fuel.

15.5. Variation of Dissociation - SPPU : May-16, 6-Marks

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 heat is liberated.

Dissociation of CO_2 start around 1000°C, and the reaction is as follows,

$$CO_2 \rightleftharpoons 2CO + O_2$$

And H_2O start around 1300°C, and the reaction is as follows,

$$H_2O \rightleftharpoons 2H_2 + O_2$$

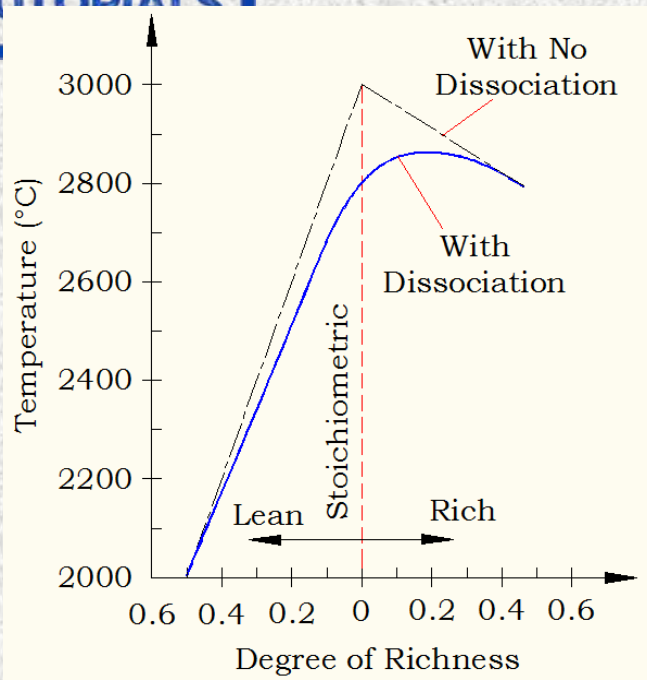
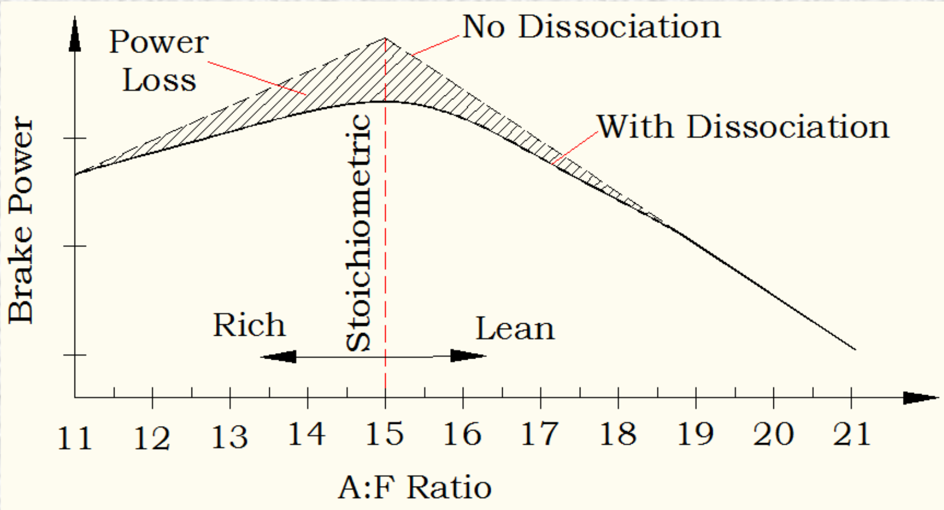


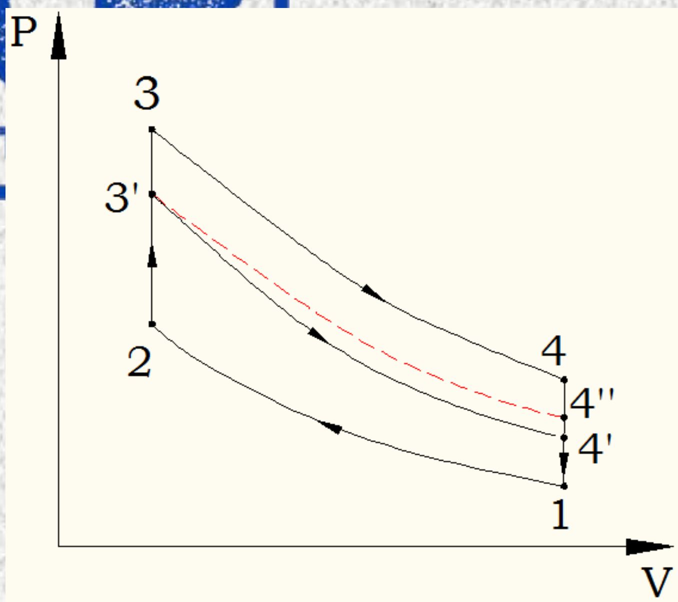
Figure shows the reduction in the temperature of the exhaust gas mixtures due to dissociation with respect to air-fuel ratio. With no dissociation maximum temperature is attained at stoichiometric air-fuel ratio. With dissociation maximum temperature is obtained when mixture is slightly rich.

The loss due to dissociation is shown in the figure given below, the brake power for four stroke SI engine is plotted against the A:F ratio when engine run at constant speed. The graph shows that the power output is higher at the stoichiometric A:F ratio. The shaded area between the graph of BP when there is no dissociation and when the dissociation is considered is the loss of power due to dissociation. When the A:F mixture is very lean then there is no dissociation so both the curve lines are overlap, but as A:F ratio decreases i.e. mixture become richer, the maximum temperature rises and dissociation starts and reach to maximum at stoichiometric mixture.



When mixture become more richer then the temperature of gases lowered due to incomplete combustion and this cause reduction in dissociation and thus power loss.

The effect of dissociation in case of Otto cycle engine is shown on the P-V diagram below. The cycle 1-2-3-4 represents the normal Otto cycle when there is no dissociation. When there is a dissociation then the temperature of gases falls and reach to the level T_3'which is lower than the T_3. And when the dissociated gases are not re-associated then expansion follow the isentropic process $3' - 4'$ and if dissociated gases are re-associated then expansion follow the actual process $3' - 4''$. Where temperature $T_4' < T_4''$, this is because of heat released during the re-association. The energy is carried away in the exhaust gases at point $4''$ without utilizing during the expansion of gases.



16. Actual Cycle : SPPU : May-17, 6-Marks

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

1. The working substance is not pure air but it is a mixture of air-fuel vapors in case of petrol engine and air-atomized fuel injected in case of diesel engine.
2. Heat addition is not by heat reservoir but it is due to combustion of fuel which changes the composition of working substance of the cycle.
3. Variation of specific heat with temperature and dissociation of products of combustion divert cycle from ideal cycle.
4. Burnt gases at outlet and fresh mixture at inlet have different composition, pressure and temperature.
5. Compression and expansion processes are not adiabatic since there is heat loss from the system to surrounding.
6. The valves do not open and close instantaneously, due to early opening of exhaust valve before BDC, there is loss of expansion work.
7. Suction and exhaust processes are carried out at below and above atmospheric pressure and not at atmospheric pressure.
8. There are always frictional and leakage power losses.

16.1. Actual Cycle losses : SPPU : May-18, May-17, Dec.-15, 6-Marks

16.1.1. Time Losses -

In ideal cycles the heat addition process was assumed at constant volume whereas in actual cycle the burning of fuel does not take place instantaneously and the entire combustion process take a definite time interval.

During the combustion time period, the gases experience a change in volume, the increasing volume due to motion of piston result in lowering maximum pressure and less work on the piston, the power loss is known as 'Time Losses'.

16.1.2. Heat Losses -

The ideal compression and expansion processes were assumed to be adiabatic, where in actual processes there is heat transfer from the working substance to the cylinder walls.

Thus there is considerable amount of heat loss particularly during the combustion and expansion processes due to which there is loss of work and efficiency, this is known as 'Heat Losses".

16.1.3. Exhaust Blow Down Losses -

In ideal cycle exhaust valve is assumed to open at BDC, but in actual engines it is opened at about 45° bBDC. Thus, using expansion of gases completely for torque, high pressure gases start to escape from exhaust valve, which reducing the pressure in the cylinder during the expansion stroke so that the work produced on piston is lowered.

Due to this escape of exhaust gases, lot of heat energy is carried away by it resulting into the loss of work, this is known as 'Exhaust Blow Down Losses'.

16.1.4. Pumping Losses -

In ideal cycle the suction and exhaust processes are assumed to be at atmospheric pressure, where in actual, suction take place at below atmospheric pressure and exhaust take place at above atmospheric pressure, as some pressure differential is required to carry out the suction and exhaust processes.

Therefore some work is done on the inducted charge during suction and on burnt gases during exhaust. This work is called as 'Pumping Losses'.

16.1.5. Friction Losses -

The loss of power to overcome the friction between the piston and cylinder wall, various bearings and other auxiliary equipments such as pumps and fans etc. is known as 'friction losses'.

Friction losses increases with speed rapidly which may be reduced by providing proper lubrication to moving parts.

17. Comparison of Air-Standard Cycle – Vs – Fuel-Air Cycle – Vs – Actual Cycle :

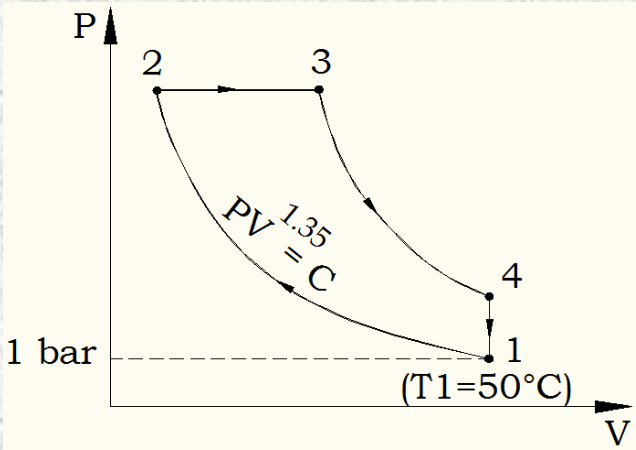
Air-Standard Cycle	Fuel-Air Cycle	Actual Cycle
Definition – the closed cycles with air as working substance with closely resembles with actual open cycle is called an air-standard cycle.	Definition – the closed cycle which take into account the variations of specific heat, molecular structure and the mixtures of fuel and air approximating to actual engine working substance, are called as fuel-air cycle.	Definition – the cycle working on fuel-air mixture and which take into account the various looses occurring in the actual engine operation, is called as actual cycle.
In air-standard cycle, the working medium is assumed to be a perfect gas, i.e. air and follows the relation $pV = mRT$	In fuel-air cycle, actual composition of the cylinder gas (fuel + air = water vapor in air + residual gases etc.) is considered as working fluid.	In actual cycle, the working fluid is a mixture of fuel-air and the residual product of combustion of previous cycle in the cylinder.
There is no change is the working fluid during the cycle.	There is a change in working fluid, at inlet it is fuel-air mixture and at outlet the exhaust gases due to burning which is assumed at constant volume and combustion is instantly.	There is also a change in composition of working fluid, at due to burning of fuel-air which is not at constant volume and it is progressive combustion.

Air-Standard Cycle	Fuel-Air Cycle	Actual Cycle
All processes in air-standard cycle are reversible adiabatic, and heat losses are zero.	All processes in fuel-air cycle are irreversible and not adiabatic, and no heat losses.	All processes are not adiabatic and heat losses are considered.
Heat is assumed to be supplied from a constant high temperature source and not from chemical reactions during the cycle, and heat is to be rejected to a constant low temperature sink.	There is no heat exchange between the gases and the cylinder wall in any process. The heat is supplied from the burning of fuel-air mixture by chemical reactions during cycle and heat is rejected to the atmosphere through exhaust gases.	There is a heat exchange between the gases and the cylinder wall as well as there are losses due to incomplete combustion of fuel and exhaled to atmosphere.
It is assumed that there is no loss of heat from the system to the surroundings.	There is losses of heat to surrounding, as exhaust gases are exhaled to atmosphere. But heat loss to cylinder wall is zero	There is losses of heat to surrounding, as exhaust gases are exhaled to atmosphere. But the heat loss to cylinder wall is also considered.
The working medium has constant specific heat throughout the cycle. And there is not burning process take place.	The working medium fuel-air, has variation in the specific heat with temperature during the cycle due to chemical reaction takes place. But complete combustion of fuel is considered	The working medium fuel-air mixture has variation in the specific heat with temperature during the cycle, but complete combustion of fuel is not considered.
Air-standard cycle analysis shows the general effect of only compression ratio on engine efficiency.	Fuel-air cycle analysis shows the effect of variation of fuel-air ratio, inlet pressure and temperature in addition to compression ratio.	Actual cycle analysis shows the effect of variation of fuel-air ratio, inlet pressure and temperature, compression ratio, various heat losses occurring in the actual operation.
In air-standard cycle, suction and exhaust strokes are eliminated.	In fuel-air cycle, suction and exhaust is at atmospheric pressure, thus there is no pumping losses.	In actual cycle, suction is at below the atmospheric pressure and exhaust is at above atmospheric pressure, thus it causes pumping losses.

Prob. 1 : In a diesel cycle, air at the beginning of compression is 1 bar and 50°C. The A/F ratio is 25 : 1 and compression ratio is 15. Assuming $C_v = 0.71 + 0.00021T$ and law of compression is $p\ V^{1.35} = Constant$. Calculate the percentage stroke at which the combustion is completed. Take Calorific Value of Fuel = 44000Kj/Kg, Characteristic Gas Constant = 287 J/Kg K. SPPU : Dec-07, 8-Marks

Ans. : Diesel Cycle, P1 = 1bar, T1 = 50°C = 50+273 = 323 K, A/F = 25, CR =15, $C_v = 0.71 + 0.00021T$, $p\ V^{1.35} = Constant$, CV = 44000Kj/Kg, R = 287 J/Kg K.

Calculate % stroke = $\frac{V_3 - V_2}{V_1 - V_2}\ x\ 100$ = ?



Hint :

$$\frac{V_3 - V_2}{V_1 - V_2}$$

$$\frac{(V_3/V_2) - 1}{(V_1/V_2) - 1}$$

$$(V_1/V_2) = CR = 15$$

$$(V_3/V_2) = Cut\ of\ Ratio$$

Process 2-3,

$$(V_3/V_2) = (T_3 - T_2)$$

T2 :Process 1-2,

$$p\ V^{1.35} = C$$

T3 : Heat Addition 2-3,

$$m_f\ CV = \int_{T2}^{T3} m\ C_p\ dT$$

m = 1 Kg, $m_f + m_a = 1$

Thus , $m_f = \frac{1}{26}$

$$C_p = C_v + R$$

We know, Compression Ratio (CR) is,

$$\boldsymbol{CR = \frac{V1}{V2} = 15}$$

And for heat addition process, 2-3,

$$\frac{V2}{T2} = \frac{V3}{T3}$$

$$\boldsymbol{\frac{V3}{V2} = \frac{T3}{T2}\ \ldots\ldots\ldots eq.1}$$

For compression process 1-2,

$$p_1\ {V_1}^{1.35} = p_2\ {V_2}^{1.35}$$

$$\frac{{V_1}^{1.35}}{{V_2}^{1.35}} = \frac{p_2}{p_1}$$

$$\left(\frac{V_1}{V_2}\right)^{1.35} = \frac{p_2}{p_1} \quad \ldots\ldots\ldots(a)$$

$$\frac{p1\ V1}{T1} = \frac{p2\ V2}{T2}$$

$$\frac{T2}{T1}\frac{V1}{V2} = \frac{p2}{p1} \Rightarrow \frac{T2}{T1}\left(\frac{V1}{V2}\right) = \frac{p2}{p1} \;\ldots\ldots put\ in\ eq.(a)$$

$$\left(\frac{V1}{V2}\right)^{1.35} = \frac{T2}{T1}\left(\frac{V1}{V2}\right)$$

$$\frac{T2}{T1} = \left(\frac{V1}{V2}\right)^{1.35}\left(\frac{V1}{V2}\right)^{-1}$$

$$\frac{T2}{T1} = \left(\frac{V1}{V2}\right)^{0.35}$$

$$\frac{T2}{323} = (15)^{0.35}$$

$$\boldsymbol{T2 = 833.36\ K}$$

We know,

$$Cp = Cv + R$$

$$Cp = (0.71 + 0.00021T) + R$$

$$Cp = (0.71 + 0.00021T) + 0.287$$

$$Cp = 0.997 + 0.00021T$$

For mass of charge, m = 1 Kg,

$$m_f + m_a = 1\ Kg$$

$$m_f + 25\ m_f = 1$$

$$26\ m_f = 1$$

$$m_f = (1/26)\ Kg$$

We know, heat addition at constant pressure process, 2-3 :

$$m_f\ CV = \int_{T2}^{T3} m\ C_p\ dT$$

$$\left(\frac{1}{26}\right) x\ 44000 = \int_{T2}^{T3} 1\ x\ (0.997 + 0.00021T)\ dT$$

$$1692.31 = \int_{T2}^{T3} 1\ x\ (0.997 + 0.00021T)\ dT$$

$$1692.31 = 0.997\ [T]_{T2}^{T3} + 0.00021\left[\frac{T^2}{2}\right]_{T2}^{T3}$$

$$1692.31 = 0.997\ (T3 - 833.36) + \frac{0.00021}{2}\ (T3^2 - 833.36^2)$$

$$1692.31 = 0.997\,T3 - 830.86 + 0.000105\,T3^2 - 72.92$$

$$0.000105\,T3^2 + 0.997\,T3 = 2596.09$$

$$\boldsymbol{T3 = 2127.3\ K}$$

From eq. 1,

$$\frac{V3}{V2} = \frac{2127.3}{833.36}$$

$$\boldsymbol{\frac{V3}{V2} = 2.55}$$

The percentage of stroke at which combustion is completed is,

$$= \frac{V3 - V2}{V1 - V2}\ x\,100$$

$$= \frac{(V3/V2) - 1}{(V1/V2) - 1}\ x\,100$$

$$= \frac{2.55 - 1}{15 - 1}\ x\,100$$

$$\boldsymbol{= 11.07\ \%\ \ldots\ldots\ Ans.}$$

Prob. 2 : Determine the effect of percentage change in efficiency of Otto cycle having a compression ratio 8 if the specific heat at constant volume increases by 2%. SPPU : Dec.-06, 8-Marks

Ans. : Compression Ratio, r = 8, Increase in C_v = 2%

We know, Efficiency of Otto Cycle is,

$$\eta = 1 - \frac{1}{r^{(\gamma-1)}}\ \ \ldots\ldots Eq.1.$$

$$Cp - Cv = R\ \ldots\ldots\ \frac{Cp}{Cv} = \gamma, Thus\ Cp = \gamma\ Cv$$

$$\gamma\,Cv - Cv = R$$

$$(\gamma - 1)Cv = R$$

$$(\gamma - 1) = \frac{R}{Cv}\ \ \ldots\ldots put\ in\ Eq.1$$

$$\boldsymbol{\eta = 1 - \frac{1}{r^{(R/Cv)}}\ \ \ldots\ldots Eq.A}$$

Case 1 : No increase in Cv –

$$\frac{R}{Cv} = (\gamma - 1)\ \ \ \ldots\ldots\ldots put\ \ \gamma = 1.4$$

$$\frac{R}{Cv} = (1.4 - 1)$$

$$\boldsymbol{\frac{R}{Cv} = 0.4}$$

From Eq. A, Efficiency when no increase in Cv is,

$$\eta_1 = 1 - \frac{1}{r^{(R/Cv)}}$$

$$\eta_1 = 1 - \frac{1}{8^{(0.4)}}$$

$$\boldsymbol{\eta_1 = 0.5647}$$

Case 2 : Increase in Cv by 2% -

$$\frac{R}{Cv} = (\gamma - 1)$$

Dividing both side by 1.02 as Cv increase to value 1.02 Cv

$$\frac{R}{1.02\ Cv} = \frac{(\gamma - 1)}{1.02}$$

$$\frac{R}{1.02\ Cv} = \frac{(1.4 - 1)}{1.02}$$

$$\boldsymbol{\frac{R}{1.02\ Cv} = 0.3921}$$

From Eq. A Efficiency when increase in Cv by 2% is,

$$\eta_2 = 1 - \frac{1}{r^{(R/1.02Cv)}}$$

$$\eta_2 = 1 - \frac{1}{8^{(0.3922)}}$$

$$\boldsymbol{\eta_2 = 0.5575}$$

Change in efficiency is,

$$Change\ in\ Efficiency = \frac{\eta_2 - \eta_1}{\eta_1}\ x\ 100$$

$$= \frac{0.5575 - 0.5647}{0.5647}\ x\ 100$$

$$\boldsymbol{= -1.28\ \%\ \ \ldots\ldots Negative\ sign\ shows\ decrease\ in\ efficiency\ \ \ \ldots Ans.}$$

Thus, there is decrease in efficiency by 1.28% if the specific heat at constant volume is increased by 2%.

Exercise

1. Define the Engine and Heat Engine.
2. Give the classification of Engine.
3. Compared the EC and IC engines.
4. Explain in brief the various components of IC engine.
5. Define the following terms used in IC engine,
    a. Bore
    b. Stroke
    c. Cubic Capacity / Breathing Capacity of Engine
    d. Clearance Volume and Swept Volume
    e. Compression Ratio and Expansion Ratio.
6. With neat sketch explain the working of 4-stroke SI engine.
7. Classify the IC engine on the basis of,
    a. Cylinder arrangement
    b. Types of ignition
    c. Types of cooling
    d. Types of fuel used
    e. Cycle of operation
8. Differentiate between four stroke CI engine and SI engine.
9. Differentiate between two stroke engine and four stroke engine.
10. List the various application of IC engines.
11. Compare SI engine and CI engine on the basis of following,
    a. Introduction of fuel
    b. Efficiency
    c. Basic cycle of operation
    d. Compression ratio
    e. Fuel used
    f. Ignition
    g. Speed

h. Weight

12. What is meant by mean piston speed.
13. List out the various assumption made in air-standard cycle analysis.
14. Draw the Otto cycle on P-V and T-S diagram and shows the various process on it.
15. Draw the Diesel cycle on P-V and T-S diagram and shows the various process on it.
16. Prove that the efficiency of Diesel cycle is lower than that of Otto cycle for the same compression ratio.
17. List out the various assumption made in fuel-air cycle analysis.
18. Differentiate between air-standard cycle and fuel-air cycle.
19. Explain with neat sketch the loss due to variation of specific heat.
20. Explain with neat sketch the effect of dissociation.
21. Briefly explain the followings,
    a. Time losses
    b. Heat losses
    c. Exhaust blow down losses
    d. Pumping losses
22. Compare the actual and fuel-air cycle.
23. An engine working on Otto cycle, have initial condition of pressure and temperature are 1 bar and 37 °C respectively. At the end of the adiabatic compression pressure reach to 15 bar and peak temperature of the cycle is 2000K. Calculate the thermal efficiency of the cycle and also find the heat supplied per unit of air, work done per unit of air and pressure at the end of adiabatic expansion. Take $C_v = 0.717\ Kj/Kg\ K$, and $\gamma = 1.4$. (***Ans.*** $\boldsymbol{\eta = 53.9\%,\ Q_S = 952.2\ Kj/Kg\ of\ air,\ WD = 513.2\ Kj/Kg\ of\ air,}$ ***pressure at the end of expansion = 2.98 bar).***
24. A diesel cycle engine is working on compression ratio 15 and expansion ratio 10. Calculate the air-standard efficiency of the cycle. Assume $\gamma = 1.4$. (***Ans.*** $\boldsymbol{= 63\%}$ ***).***
25. Dual cycle engine working on compression ratio 10 and the maximum pressure is limited to 70 bar. The heat supplied is 1680 Kj/Kg. At the beginning of the compression stroke the air is at 1 bar and 100 °C. Find the pressure and temperature at the various points of the cycle and the also find the cycle efficiency. Assume $C_v = 0.717\ Kj/Kg\ K\ of\ air$, and $C_p = 1.004\ Kj/Kg\ K\ of\ air$. (***Ans.*** $\boldsymbol{P_2 = 25.12\ bar,\ T_2 = 663.9\ K,\ T_3 = 2611.1\ K,\ Q_{S-Cont\ Vol} = 1200.4\ Kj/Kg\ of\ air,\ Q_{S-Cont\ Press} = 479.6\ Kj/Kg\ of\ air,\ T_4 = 3088.8\ K,\ T_5 = 1314.4\ K,\ P_5 = 3.53\ bar,\ Q_R = 674.98\ Kj/Kg\ of\ air,\ \eta = 59.82\%}$ ***).***

Unit – II : SI (Spark Ignition) Engines

Theory of Carburetion, Types of carburetors, Electronic fuel injection system, Combustion in spark Ignition engines, stages of combustion, flame propagation, rate of pressure rise, abnormal combustion, Phenomenon of Detonation in SI engines, effect of engine variables on Detonation. Combustion chambers, Rating of fuels in SI engines, Additives.

1. 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.

The device which supplies the mixture of correct amount of fuel and air for the efficient combustion in cylinder at all operating conditions is called Carburetor.

1.1. Factors Affecting Carburetion :

The process of carburetion is affected by the followings,

1.1.1. The engine speed –

The time available for formation of mixture by the carburetor is affected by the speed of the engine. If engine runs at @4000rpm, the time available for the process of carburetion is in the range of 0.0075sec to 0.01sec.

In this time period the fuel is to be atomized, mixed with air, vaporized and to be inducted into the engine cylinder.

1.1.2. The vaporization characteristics of the fuel –

Complete vaporization could be achieved by using highly volatile fuels, which are expensive to produced. It also achieve by using heat in intake manifold to promote vaporization.

At the same time, the excessive vaporization reduces the volumetric efficiency, thus in turn it reduces power output of the engine.

1.1.3. The temperature of the incoming air –

The temperature of air plays an important role in the vaporization process of fuel. Higher surrounding air temperature increases the vaporization of fuel and homogeneity of mixture, but increased temperatures reduce the volumetric efficiency, hence, the power output.

1.1.4. The design of the carburetor –

When the multi-cylinder engines supplied a partially vaporized mixture of fuel and air, each cylinder does not receive the same amount of fuel.

The cylinder nearer to intake manifold receive rich mixture and the cylinder farthest received comparatively lean mixture as heavy fuel particles are separate out from the mixture stream during its long path.

Therefore proper design of intake manifolds becomes essential to ensure proper distribution of homogeneous mixture with equal strength.

1.2. Air-Fuel Mixtures :

Air and fuel are mixed to form three different types of mixtures,

1.2.1 Chemically Correct (Stoichiometric) Mixture –

Chemically correct or stoichiometric mixture is one in which there is just enough air (oxygen) for complete combustion of the fuel. Thus with this mixture there is complete combustion of fuel and all carbon in the fuel is converted to carbon dioxides (CO_2) and all hydrogen to water vapors (H_2O) and there is no excess oxygen remains in combustion gases.

The hydrocarbon fuel which is used in SI engine is a petrol and it mainly octane ($C_8 H_{18}$). Chemical equation for combustion can be written as,

$$Fuel + \ Air \ \rightarrow Products\ of\ Combustions$$

$$C_8\, H_{18} + \ 12.5\,(O_2 + 3.76\, N_2) \ \rightarrow 8\, CO_2 + \ 9\, H_2O + (12.5 \times 3.76)\, N_2$$

$$C_8\, H_{18} + \ 12.5\,(4.76\, Air) \rightarrow Productof\ combustions$$

$$1\, kmole\ \ C_8\, H_{18} + \ (12.5 \times 4.76) kmole\ Air \rightarrow Product\ of\ Combustions$$

Mass of air,

$$m_{air} = No\ of\ Moles\ of\ air\ \times Molecular\ weight\ of\ air$$

$$= (12.5 \times 4.76) kmole \times 29\, Kg/kmole$$

$$= 1725.5\, Kg$$

Mass of fuel,

$$m_{fuel} = No\ of\ Moles\ of\ fuel\ \times Molecular\ weight\ of\ fuel$$

$$= (8\, kmole\ \times 12\, Kg/kmole)_C + (9\, kmole\ \times 2\, Kg/kmole)_{H2}$$

$$= (96\, Kg)_C + (18\, Kg)_{H2}$$

$$= 114\, Kg$$

The air fuel ratio on mass basis is,

$$A : F = \frac{m_{air}}{m_{fuel}}$$

$$A : F = \frac{1725.5}{114}$$

$$A : F = \frac{15.14}{1}$$

Thus, Chemically correct (stoichiometric) Air-Fuel mixture contains about 15 part of air by mass for 1 part of octane fuel.

$A : F = 15 : 1$

1.2.2. Rich Mixture –

A mixture which contains less air than the stoichiometric requirement is called a rich mixture. Thus with this mixture there is always incomplete combustion of fuel take place and all carbon can not converted to carbon dioxides (CO2) due to deficiency of oxygen, some carbon converted to carbon monoxide (CO).

It should be noted that there is extreme limit for richness beyond which mixture unable to burn due to lack of oxygen (air) comparatively fuel in the mixture.

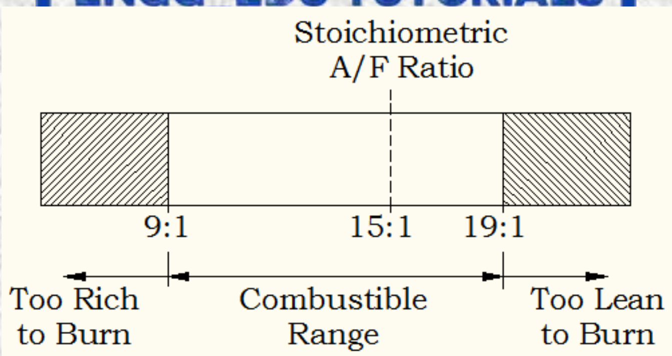
Thus rich mixture ranges the ratio of air-fuel, from 15:1 to 9:1, beyond 9:1 mixture becomes too rich to burn.

1.2.3. Lean Mixture –

A mixture which contains excess air than the stoichiometric requirement is called a lean mixture. Thus with this mixture there is always excess oxygen with result into complete combustion of fuel and all carbon converted into carbon dioxides but excess oxygen remain in the combustion gases as it is.

A mixture which contains excess air than the stoichiometric requirement is called a lean mixture. There is also a extreme limit for leanness beyond which mixture unable to burn due to lack of fuel comparatively oxygen (air) in the mixture.

Thus lean mixture ranges the ratio of air-fuel, from 15:1 to 19:1, beyond 19:1 mixture becomes too lean to burn.



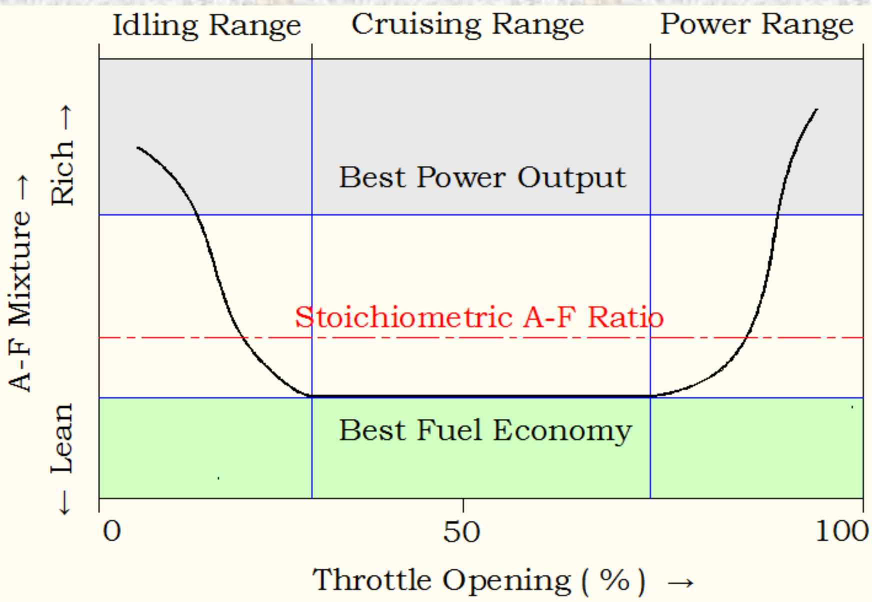
1.2.4. Air-Fuel Mixture Requirement for Engine : SPPU : Dec.-15, 6-Marks

a) Under Steady State Operation –

Steady state operation means continuous operation at a given speed and power output with normal engine temperature. There are three main areas which required different air-fuel mixture. It consists of idling, cruising range and power range.

Idling and low load – the no load or to about 20% of rated power i.e. low load running mode engine supplied with rich mixture as air supplied restricted by closed position of throttle valve but as load increases the mixture richness decreases and fuel economy enhances.

Cruising range – the cruising range is from about 20% to 75% of the rated power, as the load increases the throttle valve opens considerably and more air supply is available, thus mixture richness decreases to the best economy level. This is on leaner side than that of the stoichiometric A-F ratio. Hence in cruising range prime consideration is usually the best fuel economy only.



Power range – the power range is from about 75% to 100% of the rated power. In this range the mixture requirement is rich which provide maximum power but at the same time rich mixture reduces the flame temperature and the cylinder temperature, thus help to prevents overheating of various engine parts.

b) Under Transient Operation –

Transient operation means operation under which speed, load, temperature or pressure change rapidly. The transient operations are starting, warming up, acceleration and deceleration.

Starting and warm-up conditions – in this condition rich mixture required, as the maximum part of fuel supplied do not vaporize and remain in liquid form only and also some part of vaporized fuel when come into cold engine parts re-condensed. Therefore the total evaporated fuel and air mixture become too lean to burn, thus needed to supply very rich mixture so that enough evaporated fuel air mixture is prepared for proper ignition. As the engine warm up the amount of evaporated fuel increase and hence the mixture ratio should be progressively mad leaner.

Acceleration Conditions – the acceleration refer to an increase in engine speed resulting form opening the throttle. During gradually opening throttle valve increase the corresponding increased fuel supplied too which increases the richness of mixture. But when throttle valve is suddenly opened then this cause sudden increase in air supply but corresponding fuel rate done not changes, which result in lean mixture supplied this is compensated by using power enrichment and acceleration pump system.

1.5. Simple Carburetor :

SPPU : May-17, Dec.-17, May-15, Dec.-14, May-14, 6-Marks

A simple carburetor consists of a float with float chamber, venture tube and main jet, and throttle valve, as shown in figure below.

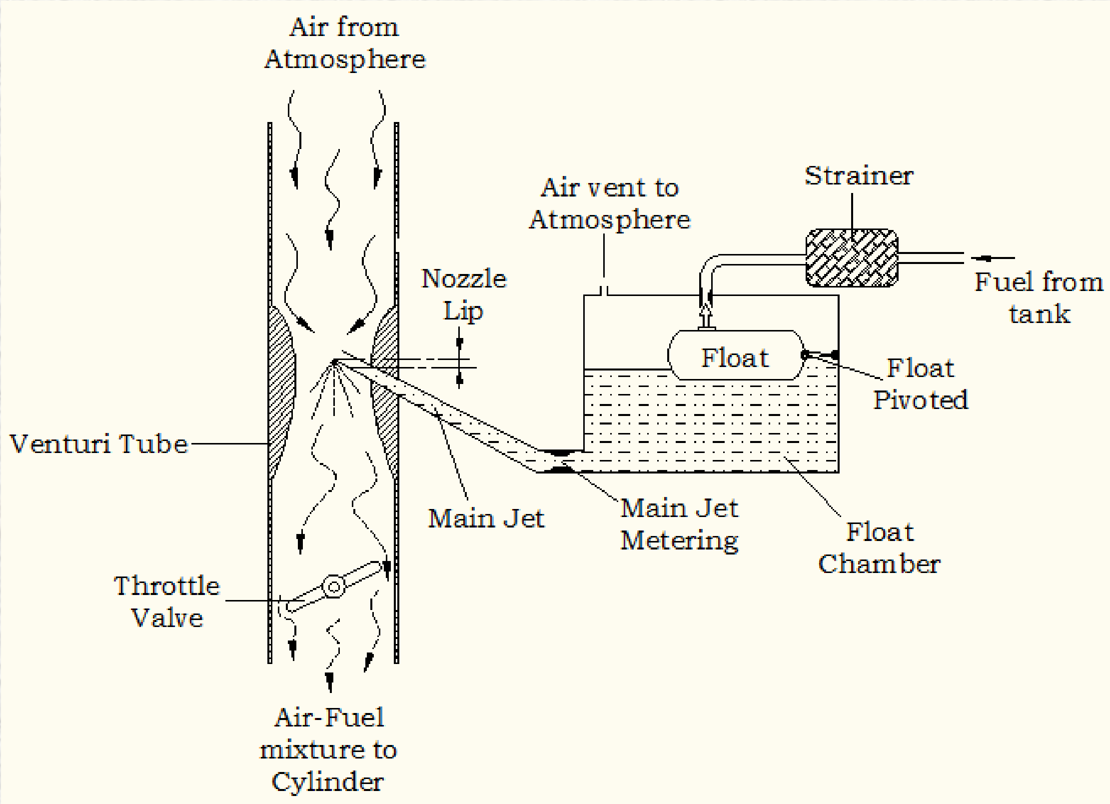
The float and a needle valve system maintains a constant level of gasoline in the float chamber.

During suction stroke air is drawn through the venturi, at venturi the velocity increases reaching a maximum at throat, and thus pressure decreases to minimum. Because of the differential pressure (carburetor depression) between the float chamber and the throat of venture, fuel is discharged into the air stream.

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.

The gasoline engine is quantity governed, which is achieved by means of a throttle valve usually a butterfly valve.

When throttle valve 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.



And when throttle valve is opened, more air flows through the venturi tube resulting in increased quantity of mixture being delivered to the engine and hence increases the engine power output.

1.6.1. Drawbacks of a Simple Carburetor –

1. Simple carburetor provide the required Air-Fuel ratio only at one throttle position, on changing position mixture become either leaner or richer depending on the throttle open less or more.
2. It provided richer mixture with increase in engine speed, as the throttle valve is opened the air flow increases which decreases pressure at venturi throat due to

this density of air decreases but density of fuel remain same. Therefore air-fuel ration decreases and mixture become richer with increase in speed.

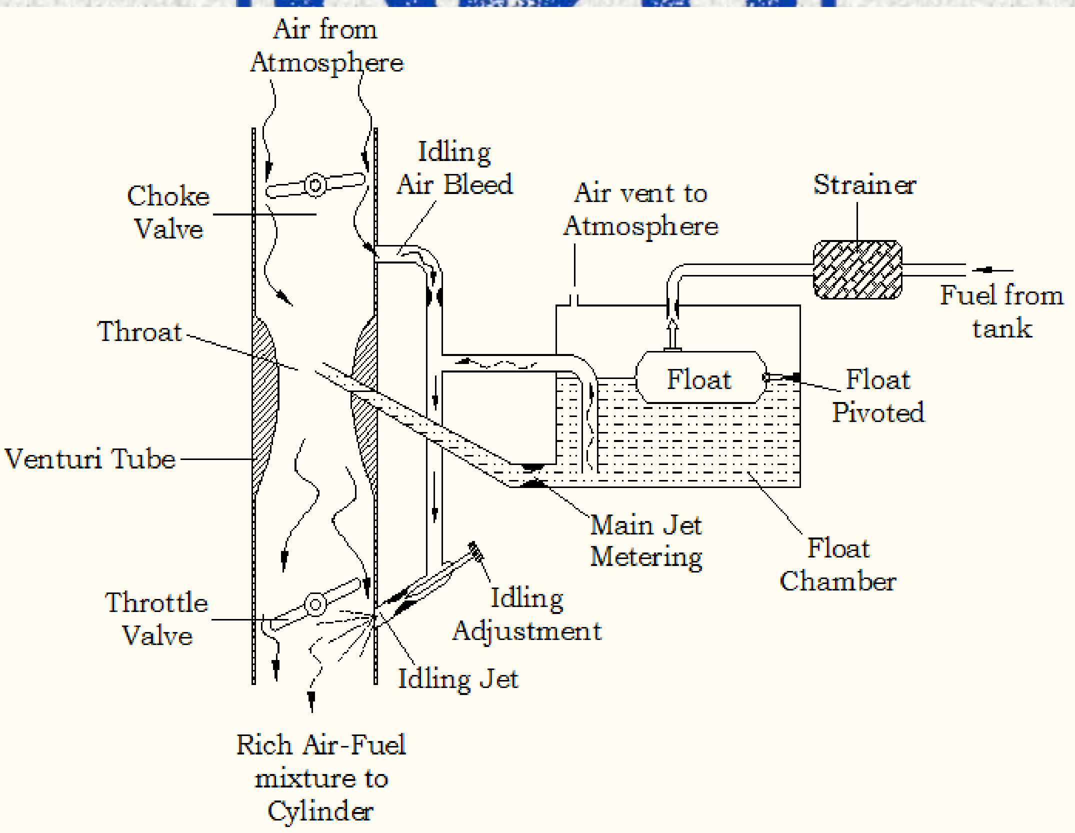
3. Similarly with decrease in engine speed, mixture become leaner, as decrease in air flow, pressure differential at throat decreases thus more dense air and less amount of fuel form lean mixture.
4. Simple carburetor provide lean mixture at low speed, where the requirement is of rich mixture at idling and low speed.
5. It provide rich mixture at cruising, where the requirement is of lean mixture.
6. Simple carburetor cannot work on engine operating at varying load and speed.
7. Simple carburetor can use only for stationary engines working at constant speed and load.

1.7. Modified Simple Carburetor :

To overcome drawback of simple carburetor, the idling system with choke valve, power enrichment (economizer) system and acceleration pump system are incorporated.

1.7.1. Main metering & Idling system / Choke system –

Engine required rich mixture at idling and low speed, due to cold condition of engine, which is fulfilled by incorporating idling system which contains choke valve, idling air bleed, and idling jet with adjustment screw, as shown in figure.



This system get operational at starting, idling and very low speed running of the engine and it become non-operational when throttle is opened beyond 15%.

When at starting, choke is partly closed, the very small quantity of air creases very little pressure depression at the throat of the venturi, and that is not enough to suck any fuel from the main jet.

But very low pressure caused on the downstream side of the throttle valve, the fuel rise in the idling tube and very small amount of air from the idling air bleed discharged through idling jet which mixes with venturi air and form rich mixture.

Thus the rich mixture below throttle valve is available for starting the engine. The desired air-fuel ration for idling can be regulated by idling adjustment screw.

When the throttle valve opens more than 15% the suction pressure at the idle jet is not sufficient to draw the fuel through the idling passage, thus it becomes non-operational. Thereafter of more than 15% opening of throttle valve, main air flow increases and the cruising range of operation is established.

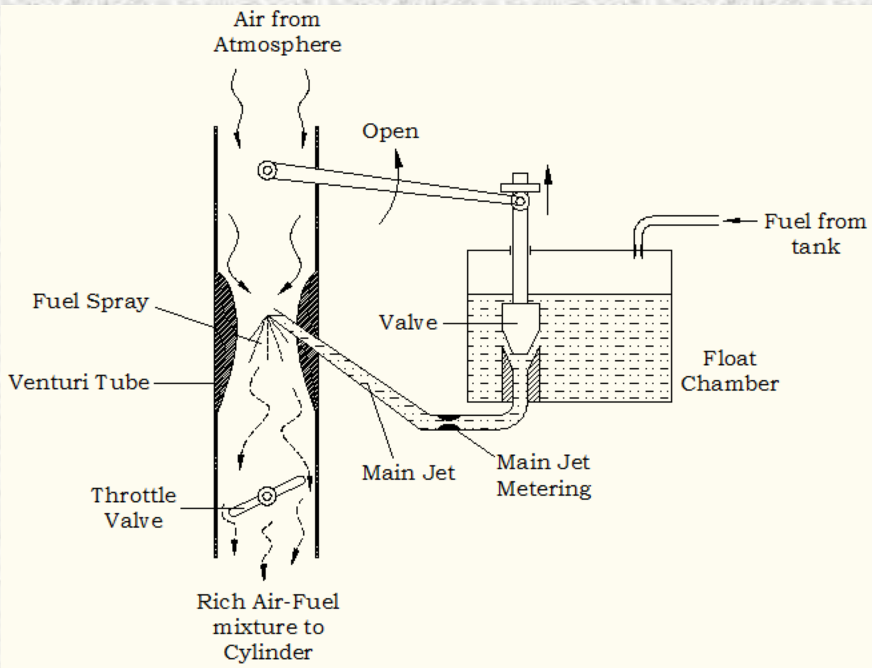
1.7.2. Power enrichment (Economizer) system –

At the maximum power range of operation from 80% to 100% load, richer air-fuel mixture is required. An economizer / power enrichment system valve remain partially closed at normal cruise operation and regular required fuel is supplied.

The power enrichment / economizer system incorporate a valve operated varying opening jet as shown in figure below,

The valve get opened to supply rich mixture at full throttle operation. It regulates the additional fuel supply during the full throttle operation.

The valve as shown is attached to the lever which operated by linking to throttle valve. Thus when full throttle operation the lever lifted and the opening to main jet fuel supply is increased and more fuel is supplied and thus increase the output power of engine.



This system does not interfere during cruising operation where economy mixture is supplied, it come in to role when more power is required at full throttle position, hence it is also known as power enrichment system.

1.7.3. Acceleration pump system –

In order to accelerate the vehicle, very rich mixture has to be supplied to the engine, and that richness of the mixture has to be obtained quickly and very rapidly.

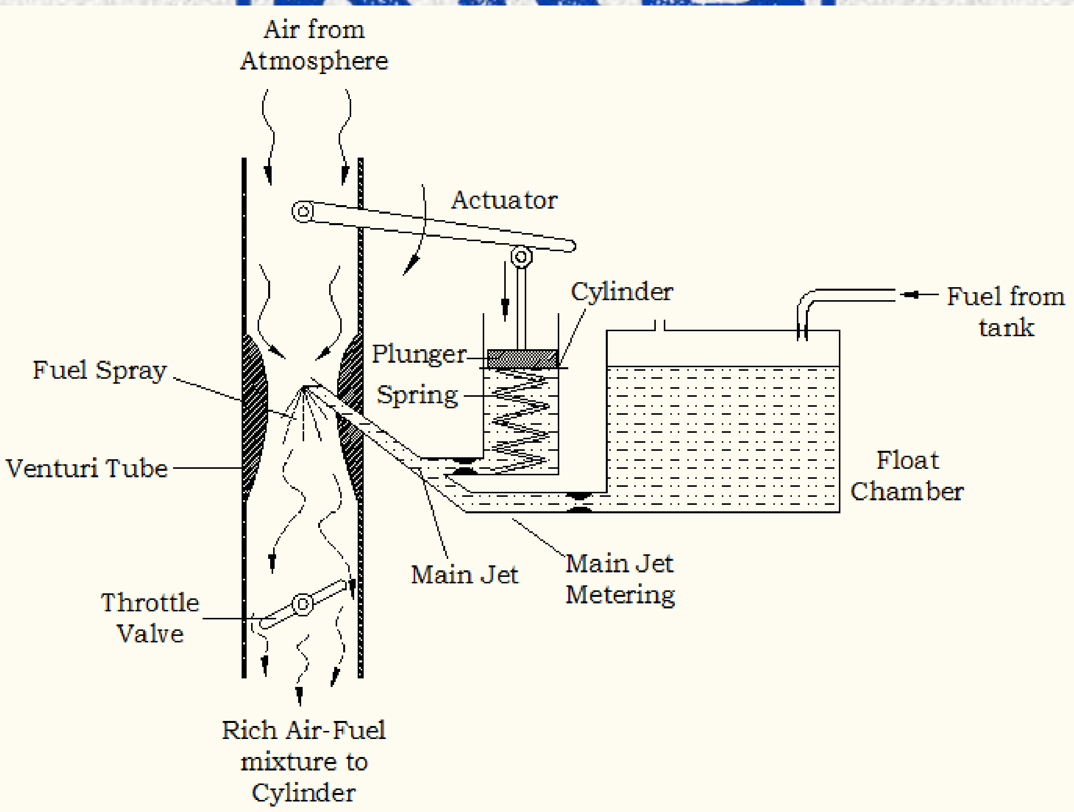
For acceleration, if the throttle valve is suddenly opened there is a increase in the air flow, but due to inertia, liquid fuel flow does not increase in that proportion so there is a temporary lean mixture causing the engine to misfire and a temporary reduction in power output.

To prevent this condition there are accelerating pump system incorporated in carburetor, as shown in figure below.

The pump comprises of a spring loaded plunger which takes care of the situation with the rapid opening of the throttle valve.

In crushing operation the cylinder get filled with fuel, when sudden acceleration, throttle movement link to actuator which pushes the plunger moves into the cylinder and forces an additional fuel at the venturi throat.

When the throttle is partly open the spring sets the plunger back, there is an arrangement which ensure that fuel in the pump cylinder is not forced through the jet when valve is slowly opened or leaks past the plunger.



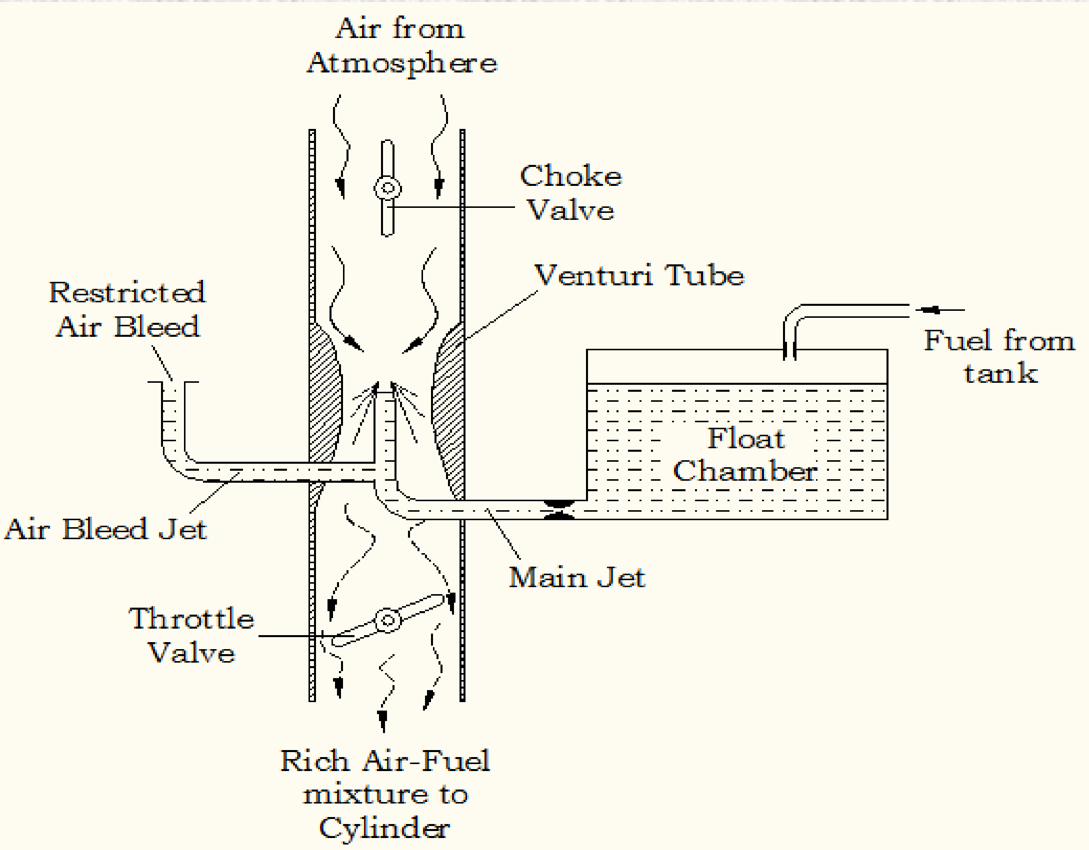
1.8. Compensating devices used in simple carburetor : SPPU : May-17, 6-Marks

A vehicle on city road runs at about 60% of the throttle only. During such condition the carburetor must be able to supply nearly constant air-fuel ratio mixture (about 16:1) which is economical.

A simple carburetor has the tendency to richen the mixture as the throttle starts opening. The main metering system alone will not be sufficient to take care of the needs of the engine. Therefore certain compensating devices are usually added in the carburetor along with the main metering system so as to supply a mixture with the required air-fuel ratio.

1.8.1. Air-Bleed Jet –

Air-bleed jet compensating device contains an air-bleed into the main nozzle as shown in the figure given. The flow of air through this bleed is restricted by an orifice and thus it is called as restricted air-bleed jet.

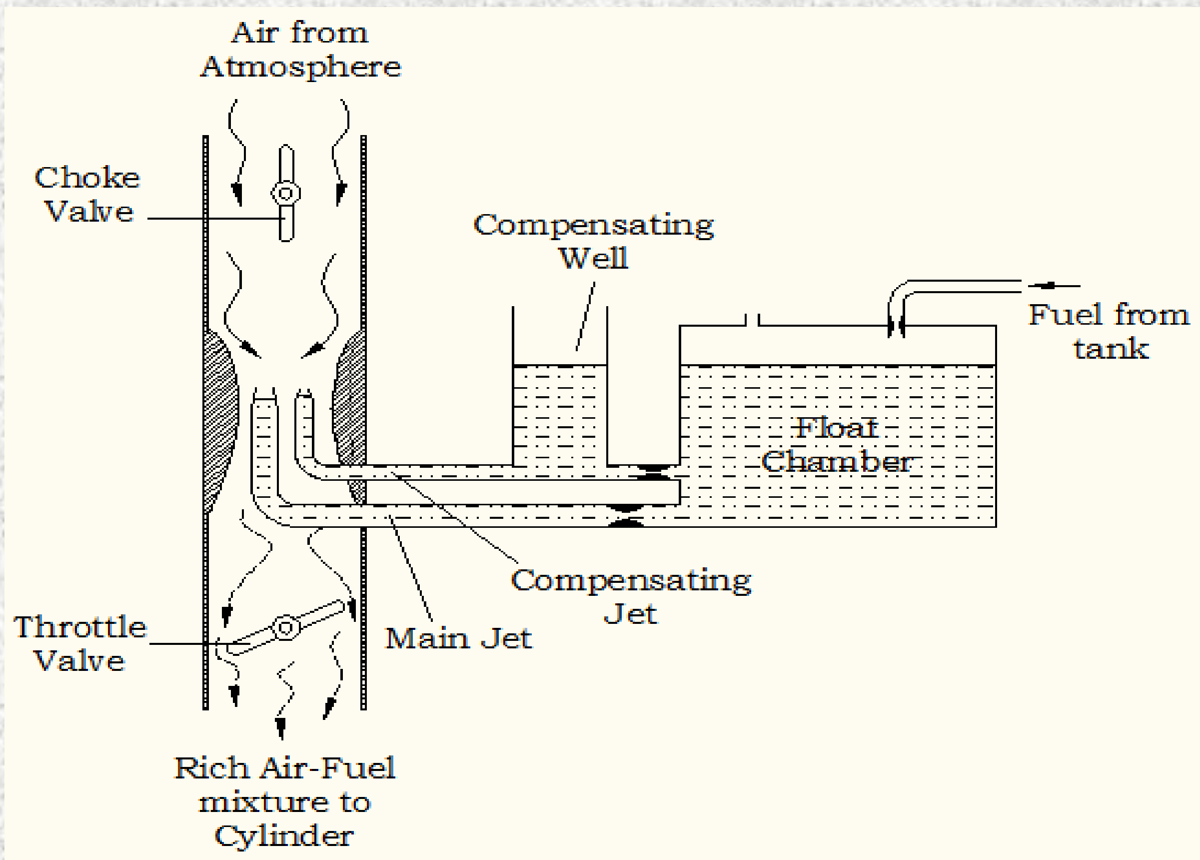


When the engine is not operating the main jet and the air bleed jet will be filled with fuel. When the engine starts, initially the fuel starts coming through the main as well as the air-bleed jet. As the engine picks up, only air starts coming through the air bleed and mixes with fuel at main jet, making a air-fuel emulsion. This emulsion of air-liquid has negligible viscosity and surface tension. Thus same fuel-air mixture for the entire power range of operation.

If the fuel flow nozzle of the air-fuel system is placed in the centre of the venture, both the air-bleed nozzle and the venture are subjected to same engine suction resulting approximately same fuel-air mixture for the entire power range of operation.

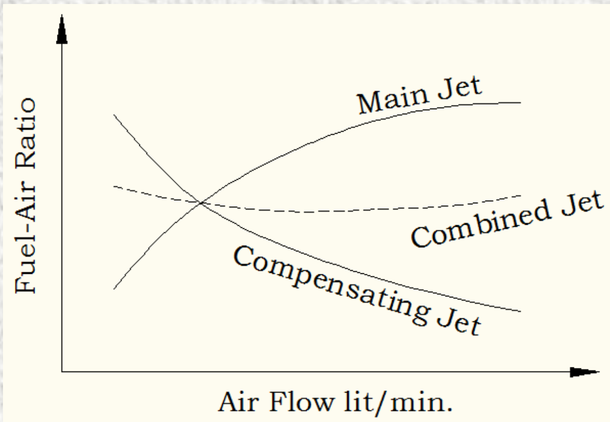
1.8.2. Compensating Jet –

The compensating jet device make the mixture leaner as the throttle opens progressively. In this system additional compensating jet is incorporated, which is connected to the compensation well which is also vented to atmosphere like float chamber.



With the increase in air flow rate, there is decrease of fuel level in the compensating well, with decrease fuel supply through the compensating jet. 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 air-fuel mixture more or less constant. The main jet curve and the compensating jet are more or less reciprocal of each other, but the combined jet curve almost constant for entire power range.

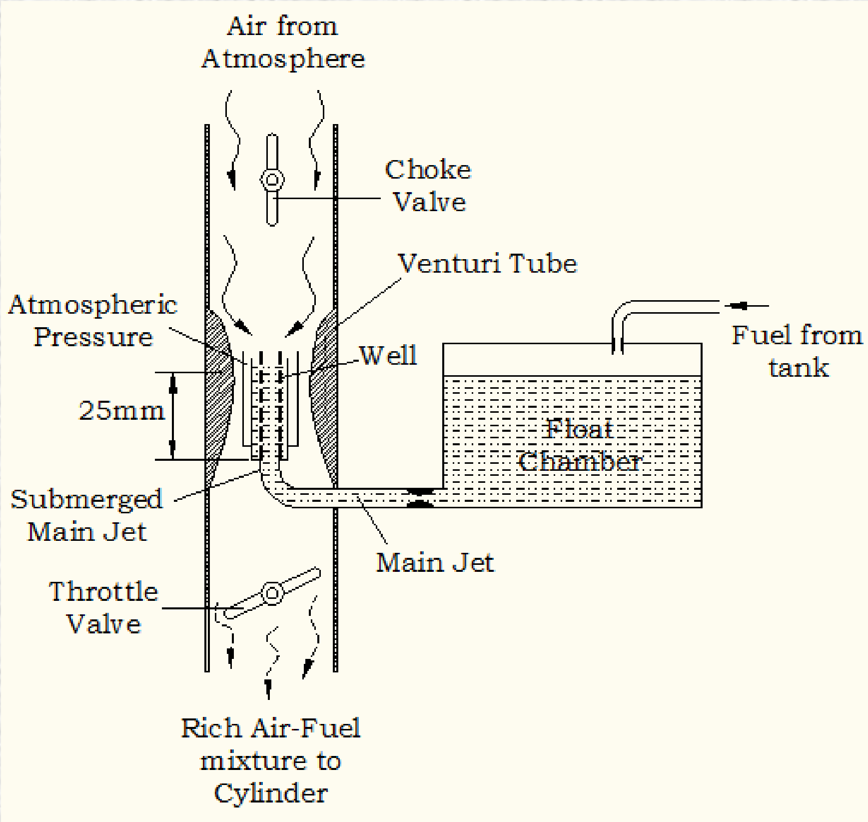


1.8.3. Emulsion Tube –

The mixture correction is done by air bleeding in emulsion tube compensating device. In this, the main metering jet is kept at a level of about 25mm below the fuel level in the float chamber, so that it is also called as submerged jet.

The jet is located at the bottom of a well and the sides of the well have holes and these holes are in communication with the atmosphere as shown in given figure.

In the beginning the level of fuel in the float chamber and the well is the same. When the throttle is opened the pressure at the venture throat decreases and fuel is drawn into the air stream. This result in progressively uncovering the holes in the central tube leading to increasing air-fuel ratio or decreasing richness of the mixture when all holes have been uncovered.



Normal flow takes place from the main jet. The air drawn through theses holes in the well, and the fuel is emulsified and the pressure differential across the column of fuel is not as high as that in simple carburetor.

1.9. Solex Carburetor :

The solex carburetor is famous for ease of starting, good performance, and reliability. It is made in various models, and is used in fiat and standard cars and wills jeep.

1.9.1. Normal Running –

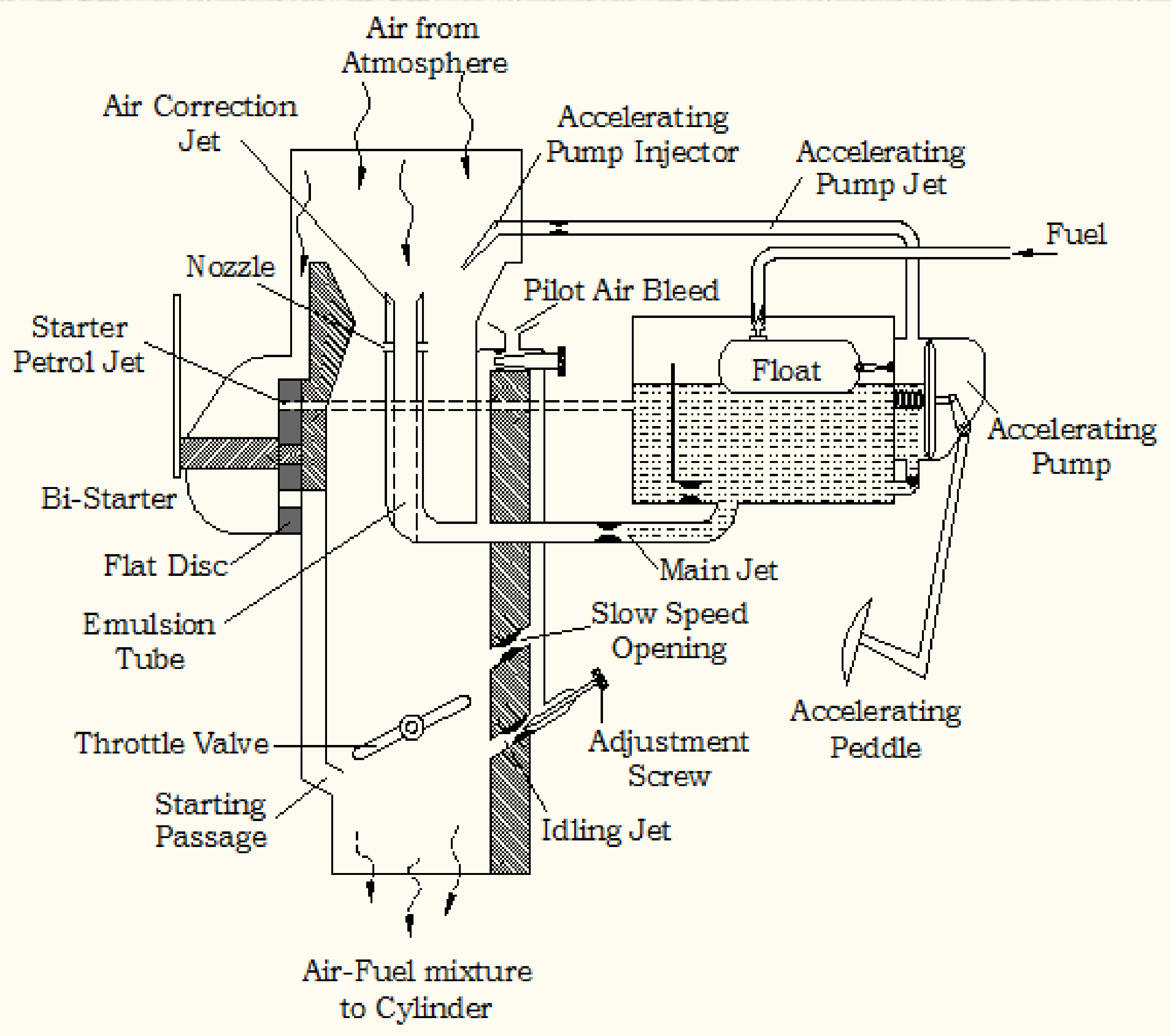
The solex carburetor has a conventional float in a float chamber. For normal running, the fuel is provided by the main jet passes into the well of the air bleed emulsion system, is the emulsion tube which has lateral holes.

The correct balance of air and fuel is automatically ensured by air entering through and being calibrated by the air correction jet. The metered emulsion of fuel and air is discharged through the spraying orifice or nozzles drilled horizontally in the vertical stand-pipe in the middle of choke tube or venturi, is the conventional butterfly valve.

1.9.2. Normal Running –

The solex carburetor has a conventional float in a float chamber. For normal running, the fuel is provided by the main jet passes into the well of the air bleed emulsion system, is the emulsion tube which has lateral holes.

The correct balance of air and fuel is automatically ensured by air entering through and being calibrated by the air correction jet. The metered emulsion of fuel and air is discharged through the spraying orifice or nozzles drilled horizontally in the vertical stand-pipe in the middle of choke tube or venturi, is the conventional butterfly valve.



1.9.3. Acceleration –

A diaphragm type acceleration pump is provided. It delivers spurts of extra fuel needed for acceleration through pump injector. Pump lever is connected to the accelerator so that when the pedal is pressed, the lever moves towards left, pressing the diaphragm thus forcing the fuel through pump jet and injector.

When the pedal is left free, the lever movers the diaphragm back towards right creating vacuum towards left which opens the pump inlet valve and thus admits the petrol from chamber into the pump.

1.9.4. Idling and Slow Running –

Idling and slow running system, from the lower part of the main jet to emulsion tube, fuel bleeds to pilot jet. At idling the fuel is drawn from the pilot jet and mixed with a small amount of air from pilot air bleed and discharged through idling jet during idling and slop speed opening during slow running.

1.9.5. Cold Starting –

Cold starting system consists of bi-starter. The starter valve is in the form of a flat disc with holes of different sizes. These holes connects the starter petrol jet and starter air jet sides to the passage which opens into a hole just below the throttle valve.

Depending on the position of the starter lever either bigger or smaller holes come opposite the passage. The starter lever, which rotates the starter valve, is operated from the dashboard control by means of a flexible cable.

Initially for starting richer mixture is required and after the engine starts the richness required decreases. So in the start position bigger holes are the connecting holes. The throttle valve being in the closed position the whole of the engine suction is applied to starting passage, sucking fuel from jet and air from jet. Thus the mixture supplied to the carburetor is rich enough for starting.

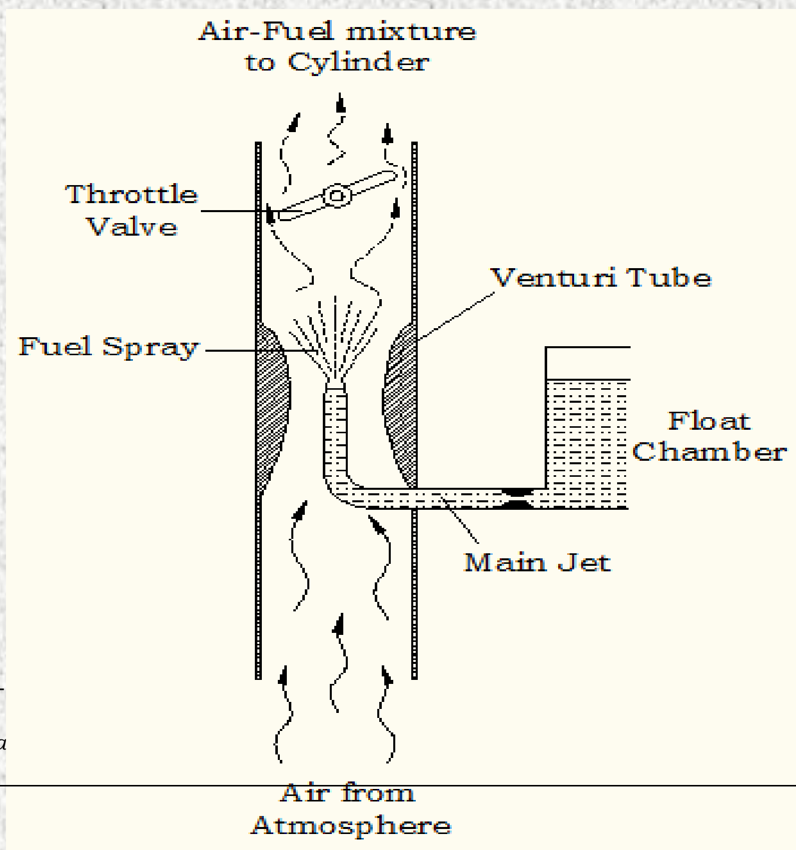
After the engine has started, the starter lever is brought to the intermediate position, bringing the smaller hole in the starter valve into circuit, thus reducing the amount of fuel. The reduced mixture supply from the starter system in this situation is sufficient to keep the engine running.

2. Types of Carburetor :

2.1. Up-Draught Carburetor –

In this type, air enters the carburetor against the gravity from bottom in the upward direction.

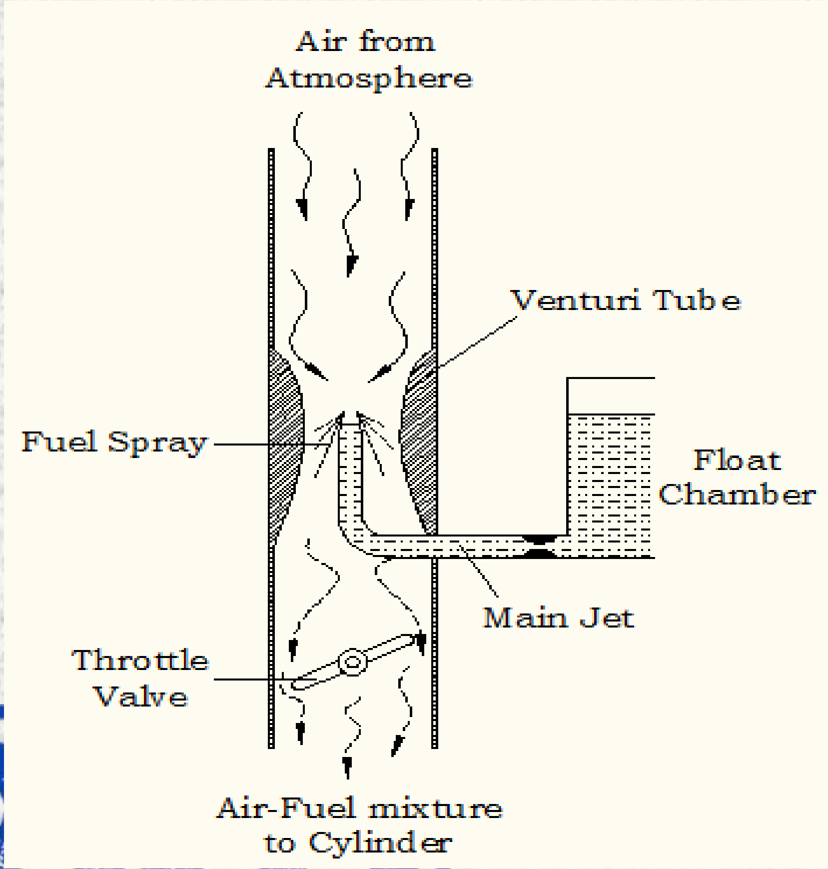
The disadvantage of such carburetor is that it has to lift the sprayed fuel droplets by air friction, against the tendency of fuel droplets to separate out from the air stream due to high inertia. Thus mixture tends to be leaner, due to this up-draught carburetor now a days become obsolete.



2.2. Down-Draught Carburetor –

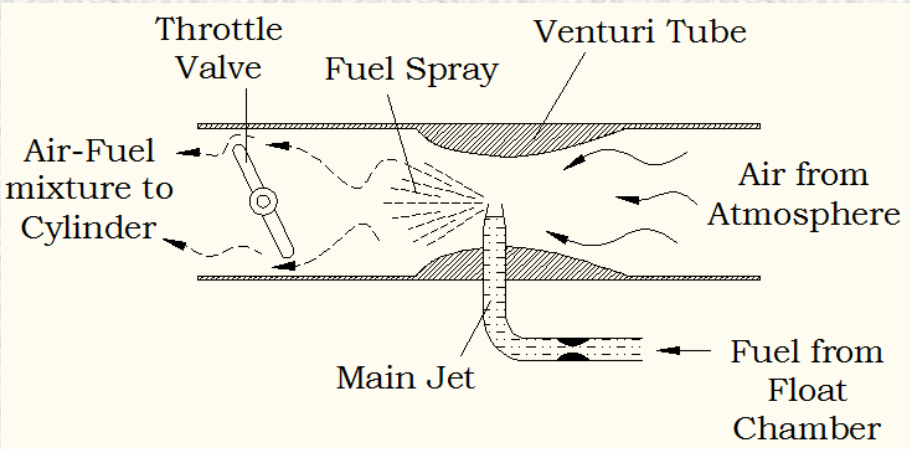
These are always at a level higher than the intake manifold. In this the flow is assisted by the gravity in its passage into intake manifold.

This allows the proper flow of mixture even at low engine speeds and at the same time the carburetor is reasonably accessible.



2.3. Side-Draught Carburetor –

It consists of a horizontal jet tube. Such a carburetor 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 passage.



2.4. Constant Choke Carburetor –

In this type, the air and fuel flow areas are always maintained to be constant. But the pressure difference or depression which causes the flow of fuel and air are being varied as per the demand on the engine. Automobile carburetors like, solex and zenith are belongs to this class.

2.5. Constant Vacuum Carburetor –

In this type, the air and fuel flow areas are being varied as per the demand on the engine, while the vacuum is maintained to be always same.

It is also called as Variable Choke Carburetor, and automobile carburetors like, SU and Carter carburetors belong to this class.

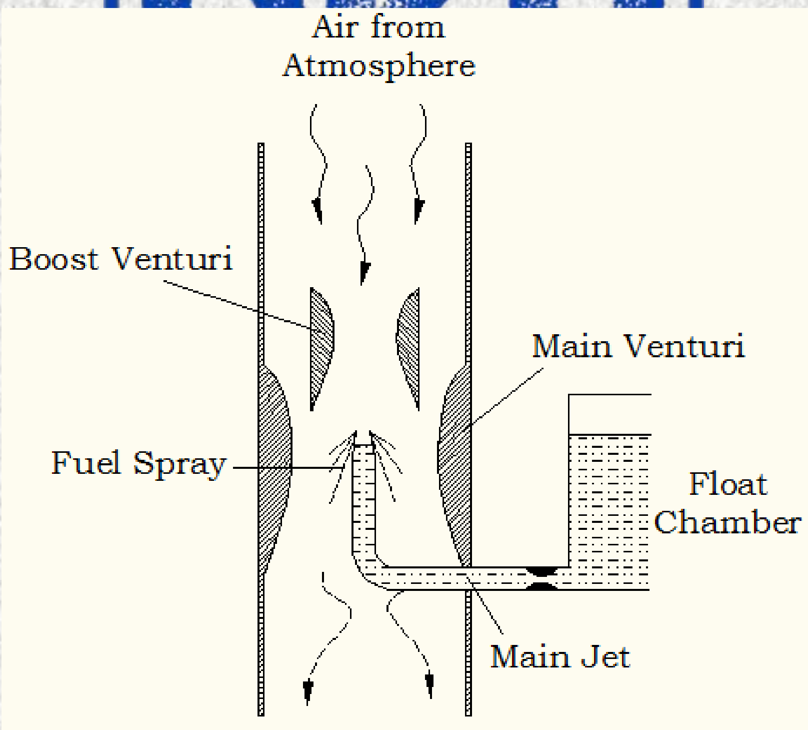
2.6. Multi-Venturi Carburetor –

This system uses double or triple venturi. The boost venturi is located concentrically within the main venturi. The discharge edge of the boost venture is located at the throat of the main venture.

The boost venturi is positioned upstream of the throat of the larger main venturi. Only a fraction of the total air flows through the boost venturi.

The pressure at the boost venture exit equals the pressure at the main venture throat. The fuel nozzle is located at the throat of the boost venturi.

There are double and triple venturi type carburetors used in automobiles, figure shown below is the double ventrui carburetor.



2.7. Dual Barrel Carburetor –

This type consists of two barrels in a carburetor, and each contains a fuel jet, venturi tube, an idling system, choke and throttle valve. The float chamber and accelerating pump are common to both the barrels.

Passenger car with six or more cylinders, are provided with dual carburetors, each venturi supplies the air-fuel mixture to half the cylinders.

The advantages of dual carburetor over single barrel carburetors are as follows,

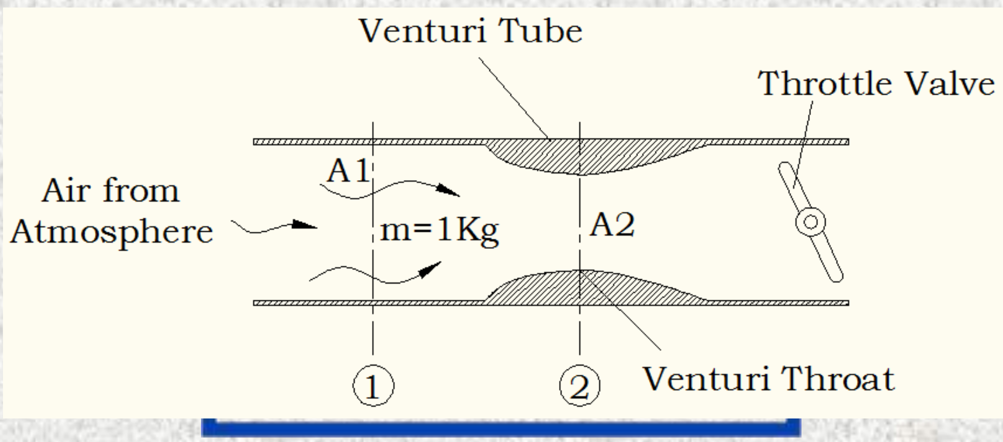
1. It supplies a charge of the mixture to the cylinders which is uniform in quality.
2. Volumetric efficiency is higher in case of a dual carburetor.
3. It is compact in the design.

3. Air-Fuel Ratio of a Simple Carburetor : Air Assumed Incompressible -

The air-fuel ratio is the ratio of mass of air to the mass of fuel in air-fuel mixture prepared for combustion.

$$A : F = \frac{\dot{m}_{air}}{\dot{m}_{fuel}}$$

a. Calculating actual mass flow rate of air through venturi ($\dot{m}_{air}$) –



Applying steady flow energy equation (SFEE) at inlet section-1 and throat section-2,

$$q + h_1 + \frac{{C_1}^2}{2} + g\,Z_1 = w + h_2 + \frac{{C_2}^2}{2} + g\,Z_2$$

Where, $q = heat\ transfer,$

$w = work\ transfer$

$C = velocity\ of\ flow$

$Z = datum\ height$

$h = enthalpy = u + P\,v$

$u = Internal\ energy = C_v\,T$

As heat transfer and work transfer is zero and venturi tube is horizontal, put, $q = w = 0$ and $Z_1 = Z_2$, above equation becomes,

$$h_1 + \frac{{C_1}^2}{2} = h_2 + \frac{{C_2}^2}{2}$$

$$(u_1 + P_1\, v_1) + \frac{{C_1}^2}{2} = (u_2 + P_2\, v_2) + \frac{{C_2}^2}{2}$$

As air flow is assumed incompressible, therefore, $T_1 = T_2$ and thus internal energy $u_1 = u_2$, and specific volume $v = \frac{1}{\rho}$

$$\frac{P_1}{\rho_1} + \frac{{C_1}^2}{2} = \frac{P_2}{\rho_2} + \frac{{C_2}^2}{2}$$

Also, the change in density is negligible for small change in pressure, density of air, $\rho_{air} = \rho_1 = \rho_2$

$$\frac{P_1}{\rho_{air}} + \frac{{C_1}^2}{2} = \frac{P_2}{\rho_{air}} + \frac{{C_2}^2}{2}$$

$$\frac{{C_2}^2 - {C_1}^2}{2} = \frac{P_1 - P_2}{\rho_{air}} \quad \ldots\ldots .eq.1$$

Applying continuity equation applied at section 1 and 2,

$$\rho_1\ A_1\ C_1 = \rho_2\ A_2\ C_2 \ldots\ldots \rho_1 = \rho_2\ \ for\ incompressible\ air$$

$$A_1\ C_1 = A_2\ C_2$$

$$C_1 = \frac{A_2}{A_1}\ C_2 \ \ldots\ldots Put\ in\ eq.1$$

$$\frac{{C_2}^2 - \left(\frac{A_2}{A_1}\ C_2\right)^2}{2} = \frac{P_1 - P_2}{\rho_{air}}$$

$${C_2}^2 - \left(\frac{A_2}{A_1}\right)^2 {C_2}^2 = 2\left(\frac{P_1 - P_2}{\rho_{air}}\right)$$

$${C_2}^2\left[1 - \left(\frac{A_2}{A_1}\right)^2\right] = \frac{2\,(P_1 - P_2)}{\rho_{air}}$$

$${C_2}^2 = \frac{2\,(P_1 - P_2)}{\rho_{air}\left[1 - \left(\frac{A_2}{A_1}\right)^2\right]}$$

$$C_2 = \sqrt{\frac{2\,(P_1 - P_2)}{\rho_{air}\left[1 - \left(\frac{A_2}{A_1}\right)^2\right]}}$$

$$C_2 = \sqrt{\frac{2\,(\Delta P)}{\rho_{air}\left[1 - \left(\frac{A_2}{A_1}\right)^2\right]}} \quad \ldots\ldots . \Delta P\ is\ drop\ in\ pressure$$

Thus, theoretical mass flow rate of air is,

$$(\dot{m}_{air})_{th} = \rho_{air}\ A_2\ C_2$$

$$(\dot{m}_{air})_{th} = \rho_{air}\ A_2 \sqrt{\frac{2\,(\Delta P)}{\rho_{air}\left[1-\left(\frac{A_2}{A_1}\right)^2\right]}}$$

$$(\dot{m}_{air})_{th} = A_2 \sqrt{\frac{2\,(\Delta P)\,\rho_{air}}{\left[1-\left(\frac{A_2}{A_1}\right)^2\right]}}$$

Coefficient of Discharge for air is the ratio of actual mass flow rate the theoretical mass flow rate.

$$Cd_{air} = \frac{\dot{m}_{air}}{(\dot{m}_{air})_{th}}$$

$$\dot{m}_{air} = Cd_{air}\ (\dot{m}_{air})_{th}$$

$$\dot{m}_{air} = Cd_{air}\ A_2 \sqrt{\frac{2\,(\Delta P)\,\rho_{air}}{\left[1-\left(\frac{A_2}{A_1}\right)^2\right]}}$$

As compared to inlet, throat area is very small, neglecting term $\left(\frac{A_2}{A_1}\right)$, the term,

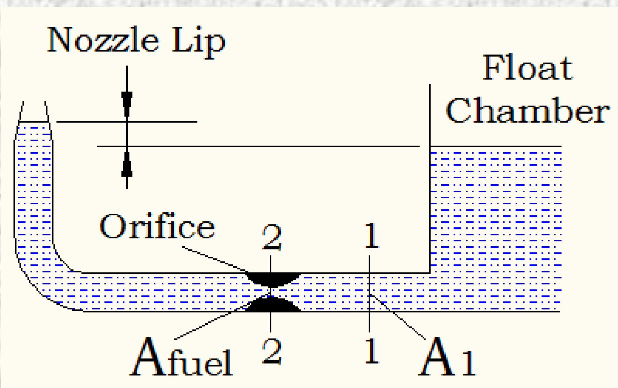
$1-\left(\frac{A_2}{A_1}\right)^2$ is called as approach factor.

Considering $A_2 = A_{air}$

$$\boldsymbol{\dot{m}_{air} = Cd_{air}\ A_{air}\ \sqrt{2\ \rho_{air}\,(\Delta P)} \ldots\ldots\ actual\ mass\ flow\ rate\ of\ air}$$

b. Calculating mass flow rate of fuel through orifice $(\dot{m}_{fuel})$ –

Applying Bernoulli's Equation to fuel flow through an orifice of cross sectional area A_{fuel} as shown,



$$\frac{{C_1}^2}{2} + \frac{P_1}{\rho_1} = \frac{{C_2}^2}{2} + \frac{P_2}{\rho_2} + h\,g$$

Where , C = velocity of fuel flow in m/sec.
P = pressure in N/m^2
ρ= density in Kg/m^3
h = nozzle lip in m

Neglecting C1 as it is very less as compared to C2,

$$\frac{P_1}{\rho_1} = \frac{{C_2}^2}{2} + \frac{P_2}{\rho_2} + h\,g$$

$$\frac{{C_2}^2}{2} = \frac{P_1}{\rho_1} - \frac{P_2}{\rho_2} - h\,g \quad \ldots\ldots \quad \rho_1 = \rho_2 = \rho_{fuel}$$

$$\frac{{C_2}^2}{2} = \frac{P_1}{\rho_{fuel}} - \frac{P_2}{\rho_{fuel}} - h\,g$$

$$\frac{{C_2}^2}{2} = \frac{P_1 - P_2}{\rho_{fuel}} - h\,g$$

$$\frac{{C_2}^2}{2} = \frac{(\Delta P)}{\rho_{fuel}} - h\,g$$

$${C_2}^2 = 2\,\frac{(\Delta P)}{\rho_{fuel}} - h\,g$$

$$C_2 = \sqrt{2\,\frac{(\Delta P)}{\rho_{fuel}} - h\,g}$$

Actual mass flow rate of fuel is,

$$\dot{m}_{fuel} = Cd_{fuel}\,\rho_{fuel}\,A_{fuel}\,C_2$$

$$\dot{m}_{fuel} = Cd_{fuel}\,\rho_{fuel}\,A_{fuel}\,\sqrt{2\,\frac{(\Delta P)}{\rho_{fuel}} - h\,g}$$

$$\boldsymbol{\dot{m}_{fuel} = Cd_{fuel}\;A_{fuel}\;\sqrt{2\;\rho_{fuel}\,(\Delta P - h\,g\,\rho_{fuel}\,)}}$$

This is the mass flow rate of fuel, Where, Cd_{fuel} = coefficient of discharge of fuel orifice

Thus the air fuel ratio is,

$$A:F = \frac{\dot{m}_{air}}{\dot{m}_{fuel}}$$

$$A:F = \frac{Cd_{air}\;A_{air}\;\sqrt{2\;\rho_{air}\,(\Delta P)}}{Cd_{fuel}\;A_{fuel}\;\sqrt{2\;\rho_{fuel}\,(\Delta P - h\,g\,\rho_{fuel}\,)}}$$

$$A : F = \frac{Cd_{air}}{Cd_{fuel}} \frac{A_{air}}{A_{fuel}} \sqrt{\frac{2\,\rho_{air}\ \Delta P}{2\ \rho_{fuel}\,(\Delta P - h\,g\,\rho_{fuel})}}$$

If nozzle lip is neglected then,

$$A : F = \frac{Cd_{air}}{Cd_{fuel}} \frac{A_{air}}{A_{fuel}} \sqrt{\frac{\rho_{air}}{\rho_{fuel}}}$$

This is the air fuel ratio when air considered as incompressible.

Prob . 1 – A simple carburetor has the venture of throat diameter of 8cm and the coefficient of discharge is 0.94. The fuel orifice has the diameter of 0.5cm and its coefficient of discharge of 0.7. Find air fuel ratio if the pressure drop amount to 0.14 bars when,

a) Nozzle lip is neglected
b) Nozzle lip is taken into account and it is equal to 0.5cm.

Assume the density of fuel as 780 Kg/m³, density of air as 1.293 Kg/m³ and approach factor as 1. SPPU : Dec.-12, 8-Marks

Ans. – Throat dia., $d_{air} = 8cm = 0.08m$, $C_{d\_air} = 0.94$, Fuel Orifice dia., $d_{fuel} = 0.5cm = 0.005m$, $C_{d\_fuel} = 0.7$, $\Delta P = 0.14\ bar = 1.4\ x\ 10^4\ N/m^2$, $\rho_{fuel} = 780\ Kg/m^3$, $\rho_{air} = 1.293\ Kg/m^3$, Approach factor, $\left[1 - \left(\frac{A_2}{A_1}\right)^2\right] = 1$, Nozzle lip, $h = 0.5cm = 0.005m$

a) Air-Fuel Ratio when nozzle lip is neglected -

We know, air fuel ratio when nozzle lip is neglected and approach factor is 1,

$$A : F = \frac{Cd_{air}}{Cd_{fuel}} \frac{A_{air}}{A_{fuel}} \sqrt{\frac{\rho_{air}}{\rho_{fuel}}}$$

$$= \frac{Cd_{air}}{Cd_{fuel}}\ x\ \frac{\frac{\pi}{4}\,(d_{air})^2}{\frac{\pi}{4}\,(d_{fuel})^2}\ x\ \sqrt{\frac{\rho_{air}}{\rho_{fuel}}}$$

$$= \frac{0.94}{0.7}\ x\ \frac{\frac{\pi}{4}\,(0.08)^2}{\frac{\pi}{4}\,(0.005)^2}\ x\ \sqrt{\frac{1.293}{780}}$$

$$\boldsymbol{A : F = \frac{13.9}{1} \ldots\ldots Ans.}$$

b) Air-Fuel Ratio when nozzle lip is considered -

We know, air fuel ratio when nozzle lip is 0.5cm and approach factor is 1,

$$A:F = \frac{Cd_{air}}{Cd_{fuel}} \frac{A_{air}}{A_{fuel}} \sqrt{\frac{2\ \rho_{air}\ \Delta P}{2\ \rho_{fuel}\ (\Delta P - h\ g\ \rho_{fuel})}}$$

$$= \frac{Cd_{air}}{Cd_{fuel}}\ x\ \frac{\frac{\pi}{4}\ (d_{air})^2}{\frac{\pi}{4}\ (d_{fuel})^2}\ x\ \sqrt{\frac{2\ \rho_{air}\ \Delta P}{2\ \rho_{fuel}\ (\Delta P - h\ g\ \rho_{fuel})}}$$

$$= \frac{0.94}{0.7}\ x\ \frac{\frac{\pi}{4}\ (0.08)^2}{\frac{\pi}{4}\ (0.005)^2}\ x\ \sqrt{\frac{2\ x\ 1.293\ x\ 1.4\ x\ 10^4}{2\ x\ 780\ x\ [(1.4\ x\ 10^4) - (0.005\ x\ 9.81\ x\ 780)]}}$$

$$\boldsymbol{A:F = \frac{14.02}{1}\ \ldots..Ans.}$$

Prob . 2 – The diameter of main jet of a simple carburetor is 0.2 cm and the pressure drop across the venture is equivalent to 12 cm of water and the coefficient of discharge is 0.68. Density of petrol is 760 Kg/m³. Find the mass flow rate of fuel into the carburetor. $Note:\ (10.33\ m\ of\ water\ column = \ 1.013\ x\ 10^5\ N/m^2)$ SPPU : Dec.-12, 8-Marks

Ans. – $d_{fuel} = 0.2cm = 0.002m,\ \Delta P = 12cm\ of\ water = 0.12\ m\ of\ water,\ C_{d\_fuel} = 0.68,\ \rho_{fuel} = 760\ Kg/m^3$

We know,

$$10.33\ m\ of\ water\ column = \ 1.013\ x\ 10^5\ N/m^2$$

$$0.12\ m\ of\ water\ column = 0.12\ x\ \frac{1.013\ x\ 10^5}{10.33} = 1176.8\ N/m^2$$

Thus pressure drop across the venture is,

$$\Delta P = 1176.8\ N/m^2$$

Now, the mass flow rate of fuel is,

$$\dot{m}_{fuel} = \ Cd_{fuel}\ \ A_{fuel}\ \sqrt{2\ \rho_{fuel}\ (\Delta P - h\ g\ \rho_{fuel}\ )}$$

$$= \ Cd_{fuel}\ x\ \frac{\pi}{4}\ (d_{fuel})^2\ x\ \sqrt{2\ \rho_{fuel}\ (\Delta P - h\ g\ \rho_{fuel}\ )}$$

Neglecting nozzle lip as it is not given, $h\ g\ \rho_{fuel} = 0$,

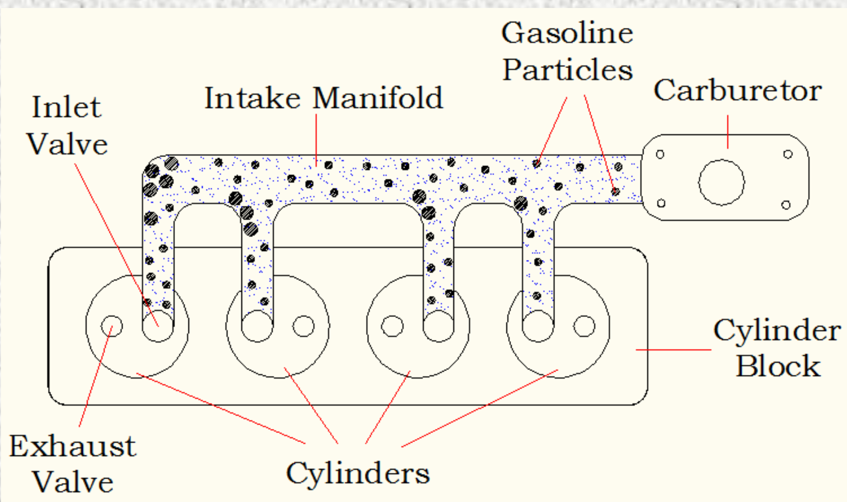
$$= \ Cd_{fuel}\ x\ \frac{\pi}{4}\ (d_{fuel})^2\ x\ \sqrt{2\ \rho_{fuel}\ \ \Delta P}$$

$$= \ 0.68\ \ x\ \frac{\pi}{4}\ (0.002)^2\ x\ \sqrt{2\ x\ 760\ x\ 1176.8}$$

$$\boldsymbol{\dot{m}_{fuel} = \ 0.0028\ Kg/sec\ \ \ldots..Ans.}$$

4. Drawbacks of Carburetor System :

In modern SI engines carburetors are completely replaced by Gasoline Injection System in spite of development of efficient carburetors which are cheap and reliable.



In carburetor engine, uniformity of mixture strength is difficult to realize in each cylinder of a multi-cylinder engine. As shown in figure, air-fuel mixture from carburetor is supplied to cylinders but the cylinder farthest from carburetor, receives rich mixture as gasoline moves to the end of the manifold and accumulates there and when valve opens enrich mixture supplied to the cylinder. Whereas the nearest mixture supplied with the leanest mixture as there is no waiting time for air-fuel mixture. This problem is called as misdistribution.

Thus the following are the inherent drawbacks of the carburetors,

1. Low volumetric efficiency – due to restriction of mixture flow across the venturi and other metering elements in air passage.
2. Non supply of exact air fuel (A/F) ratio at all loads.
3. Distribution of mixture flow is non-uniform to cylinder in case of multi-cylinder engine – due to resistance to mixture flow in unequal length of intake manifolds.
4. Economy of fuel is affected during idling and low load running of the engines.
5. Possibility of back firing at low speed particularly in multi-cylinder engines.
6. Exhaust emissions are high.
7. Problem of ice formation at low temperatures.

5. Electronic fuel injection system :

The Electronic Fuel Injection System (EFI) use engine sensors, a computer, and solenoid operated fuel injectors to meter and inject the right amount of fuel into the engine cylinders. Thus EFI system uses electrical and electronic devices to monitor and control engine operation.

The computer (ECU-Electronic Control Unit) receives electrical signals in the form of current or voltage from various sensors. It then used the stored data to operate the injectors, ignition system and other engine related devices.

The fuel injectors in EFI system is a solenoid operated fuel valves, when it is not energized spring pressure makes the injector to remain closed and no fuel will enter the engine. When computer sends the signal through the injector coil, the magnetic field attracts the injector armature and fuel inject into the intake manifold.

The injector pulse width decided by computer controls the period for which injector is energized and kept open. Under full load condition, computer sense a wide open throttle, high intake manifold pressure, and high inlet air flow, then ECU will increase the injector pulse width to enrich the mixture which results to produced higher power.

Under low load conditions, the ECU shorten the pulse width which kept injector in open position for shorter time, thus air-fuel mixture become leaner and gives better fuel economy.

EFI system has a cold start injector which sprays fuel into the center of the engine intake manifold when the engine is cold. The cold start injector ensures easy engine startup in very cold weather.

5.1. Merits of EFI System –

Followings are the merits of EFI system over the conventional carburetor,

1. Improvement in volumetric efficiency due to comparatively less resistance in the intake manifold which will cause less pressure losses.
2. Manifold wetting is eliminated due to the fuel being injected into or closed to the cylinder and need not flow through the manifold.
3. Atomization of fuel is independent of cranking speed and therefore starting will be easier.
4. Better atomization and vaporization will made the engine less knock prone.
5. Formation of ice on the throttle plate is eliminated.
6. Distribution of fuel being independent of vaporization, less volatile fuel can be used.
7. Variation of air-fuel ratio is almost negligible even when the vehicle takes different positions like turning, moving on gradients, uneven roads etc.
8. Position of the injection unit is not so critical and thereby the height of the engine can be less.

5.2. Demerits of EFI System –

Followings are the demerits of EFI system as compared to conventional carburetor,

1. EFI system has high maintenance cost.
2. Difficulty in servicing.

3. There are number of sensors, hence possibility of malfunction of some sensors.
4. Overall cost of the system is very high.

5.3. Components of Electronic Fuel Injection System : SPPU : May-18, 6-Marks

The objective of the fuel injection system is to meter, atomize and uniformly distribute the fuel throughout the air mass in the cylinder. At the same time it must maintained required air-fuel ratio as per the load and speed condition. To achieve this a numbers of components are required as mentioned below,

1. Electronic Control Unit (ECU) – it control ignition timing and quality of fuel to be injected.
2. Element and Controlling Devices –
   a. Pumping Element – it moves the fuel from the fuel tank to the injectors.
   b. Metering Element – it measures and supplies the fuel at the rate demanded by load and speed conditions of the engine.
   c. Mixing Element – it atomizes the fuel and mixes it with air to form a homogenous mixture.
   d. Metering Control – it adjusts the rate of metering in accordance with load and speed of the engine.
   e. Mixture Control – it adjust fuel-air ratio as demanded by the load and speed.
   f. Distributing Element – it divides the metered fuel equally among the cylinders.
   g. Timing Control – it fixes the start and stop of the fuel-air ratio mixing process.
   h. Ambient Control – it compensates for changes in temperature and pressure of either air or fuel that may affect the various elements of the system.
3. Sensors –
   a. Exhaust Gas / Oxygen Sensor – it senses the amount of oxygen in the engine exhaust and calculates air-fuel ratio. Sensor output voltage changes in proportion to air-fuel ratio.
   b. Engine Temperature Sensor – it senses the temperature of the engine coolant, and from this data the computer adjusts the mixture strength to rich side for cold starting.
   c. Air Flow Sensor – it monitors mass or volume of air flowing into the intake manifold for adjusting the quantity of fuel.
   d. Air Inlet Temperature Sensor – it checks the temperature of the ambient air entering the engine for fine tuning the mixture strength.

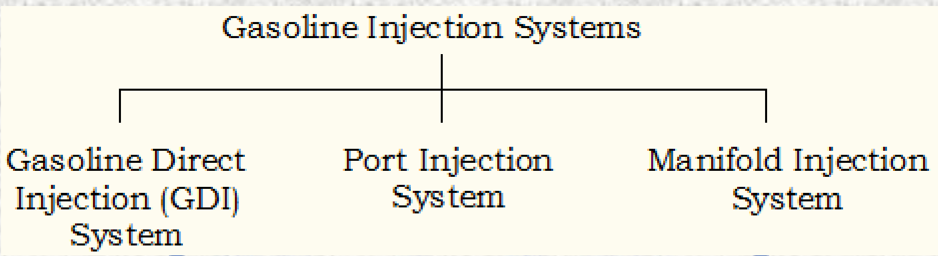
e. Throttle Position Sensor – it senses the movement of the throttle plate so that the mixture flow can be adjusted for engine speed and acceleration.

f. Manifold Pressure Sensor – it monitors vacuum in the engine intake manifold so that the mixture strength can be adjusted with changes in engine load.

g. Camshaft Position Sensor – it senses rotation of engine camshaft / crankshaft for speed and timing of injection.

h. Knock Sensor – microphone type sensor that detects ping or pre-ignition noise so that the ignition timing can be retarded.

6. Types of Gasoline Injection System : (on basis of position of injection)



6.1. Gasoline Direct Injection (GDI) System –

In GDI system injectors are located such a way that gasoline injected directly into the cylinder. The system used ECU and solenoid operated fuel injectors to meter the fuel.

Advantages of GDI system -

a. Improves volumetric efficiency of the engine.

b. Improves atomization and vaporization of fuel and it is independent of reduce gap spacing speed.

c. Ease of cold starting and low load running.

d. Specific fuel consumption is reduced.

e. Variation is A:F ratio is reduced.

f. Exhaust emissions are reduced.

g. Gives better performance on gradients.

Disadvantages of GDI system -

a. High initial cost.

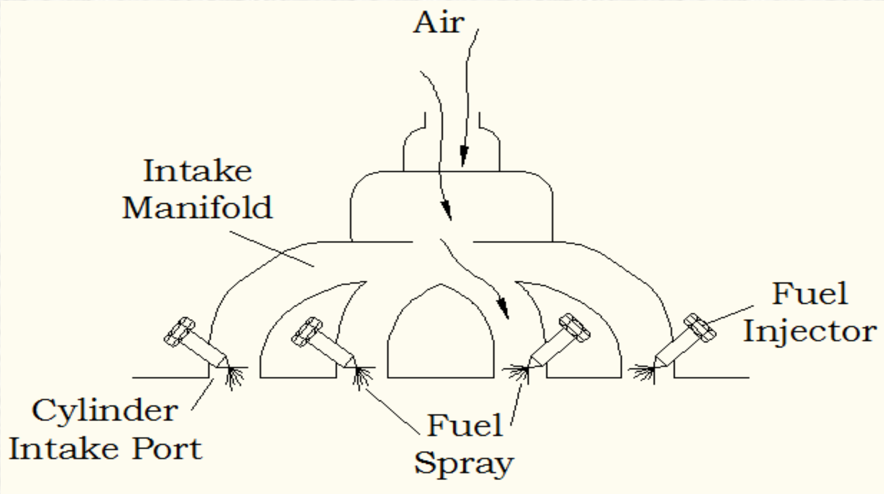
b. Higher maintenance cost.

c. Injector choking is possible.

6.2. Port Injection System –

In port injection system, fuel is injector is placed on the side of the intake manifold near the intake port. The injector sprays gasoline into the air, inside the intake manifold. The gasoline mixes with the air then passes through the intake valve and enters into the cylinder.

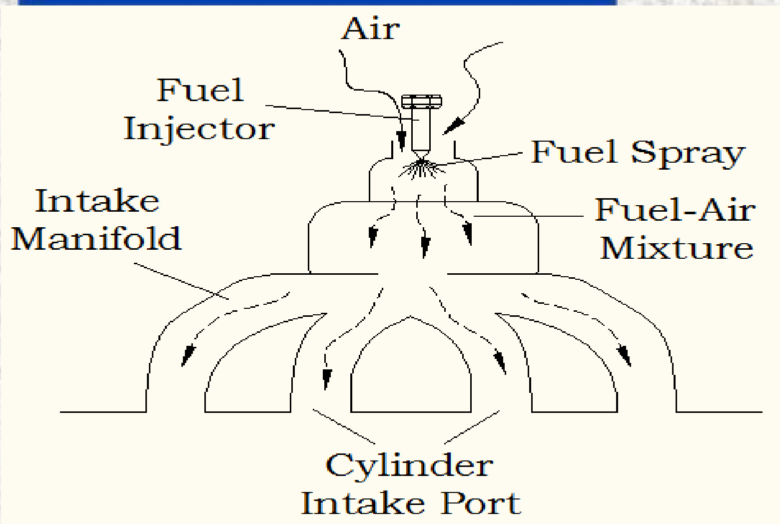
Every cylinder is provided with an injector in its intake manifold. If there are four cylinders, there will be four injectors.



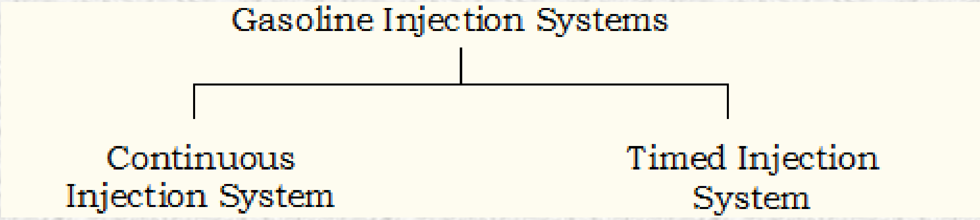
6.3. Manifold Injection System –

In manifold injection system, fuel is injector is placed in throttle body with the throttle valve controlling the amount of air entering the intake manifold.

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.



7. Types of Gasoline Injection System : (on basis of time of injection)



7.1. Continuous Injection System –

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 ECU system depending upon the load and speed.

Advantages of Continuous Injection system ;

a. It increases the atomization of fuel.

b. It provides uniform A:F ratio to all the cylinders..

c. The volumetric efficiency is high due to cooling effect of the charge caused by the evaporation of the injected fuel.

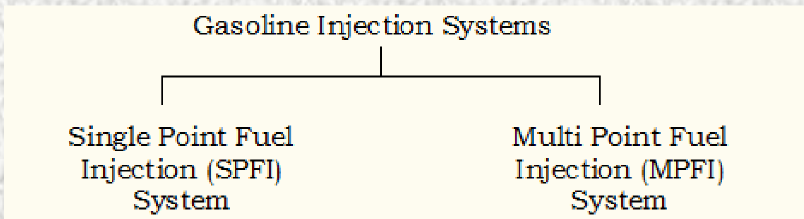
7.2. Timed Injection System –

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 pressure 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. However, it was noticed that direct injection caused oil dilution in the frequent warm up phases if the car is used for daily transportation.

8. Types of Gasoline Injection System : (on basis of no. of location of injection)



8.1. Single Point Fuel Injection (SPFI) System –

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. This is also called as throttle body injection system.

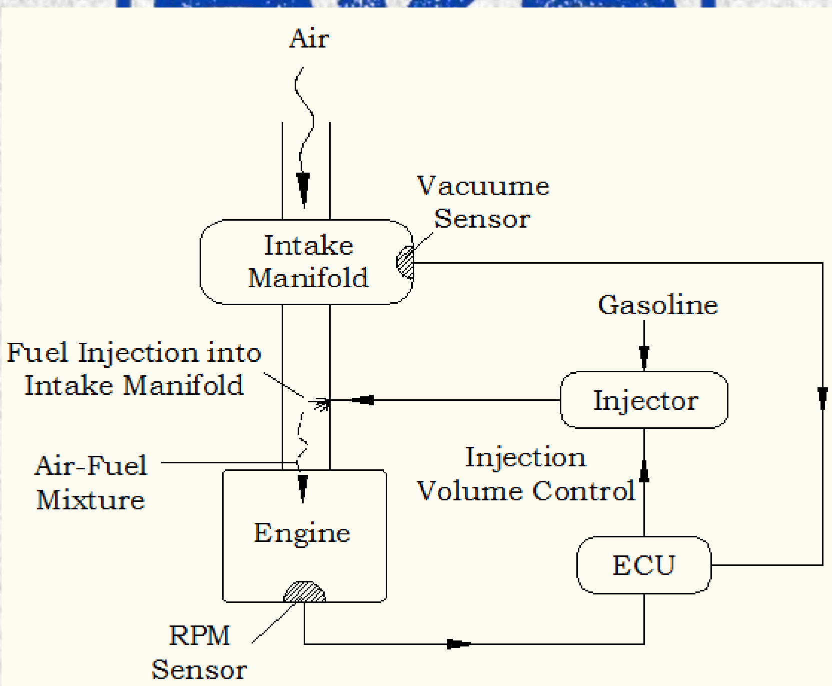
8.2. Multi Point Fuel Injection (MPFI) System – SPPU : May-18, 6-Marks

In this system, fuel is injected in more than one location. This is more common and is often called port injection system. This Multi Point Fuel Injection (MPFI) system is to supply a proper ratio of gasoline and air to the cylinders.

MPFI system uses electric fuel pump to spray fuel into engine intake manifold. It provide correct Air-Fuel ratio at all operating conditions of the engine.

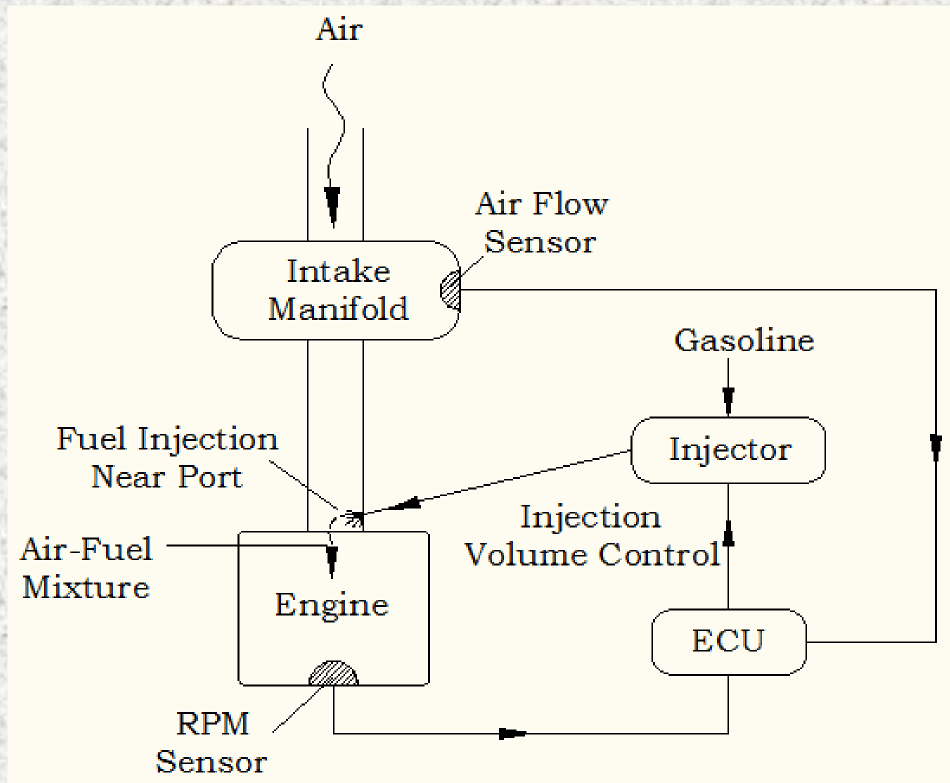
8.2.1. D-MPFI System :

The D-MPFI system is the manifold fuel injection system. The vacuum in the intake manifold is sensed first with the density of air inducted. 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.



8.2.2. L-MPFI System :

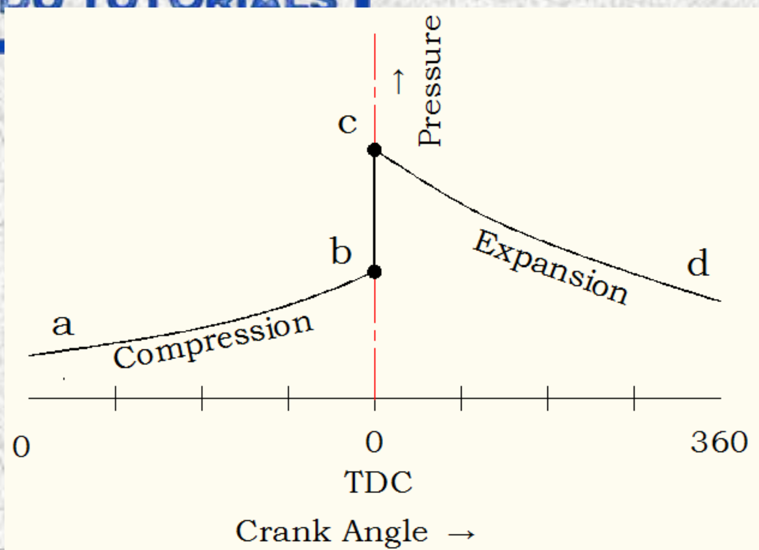
The L-MPFI system is a port fuel injection system. The fuel metering is controlled by the engine speed and the amount of air that actually enters the engine. This is called air-mass metering or air-flow metering. As air enters into the intake manifold, the air flow sensor measures the amount of air and sends information to the ECU. Speed sensors sends information of engine rpm and the ECU processes the information 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.



9. Stages of Combustion in SI Engines : SPPU : Dec.-17, May-16, Dec.-15, 6-Marks

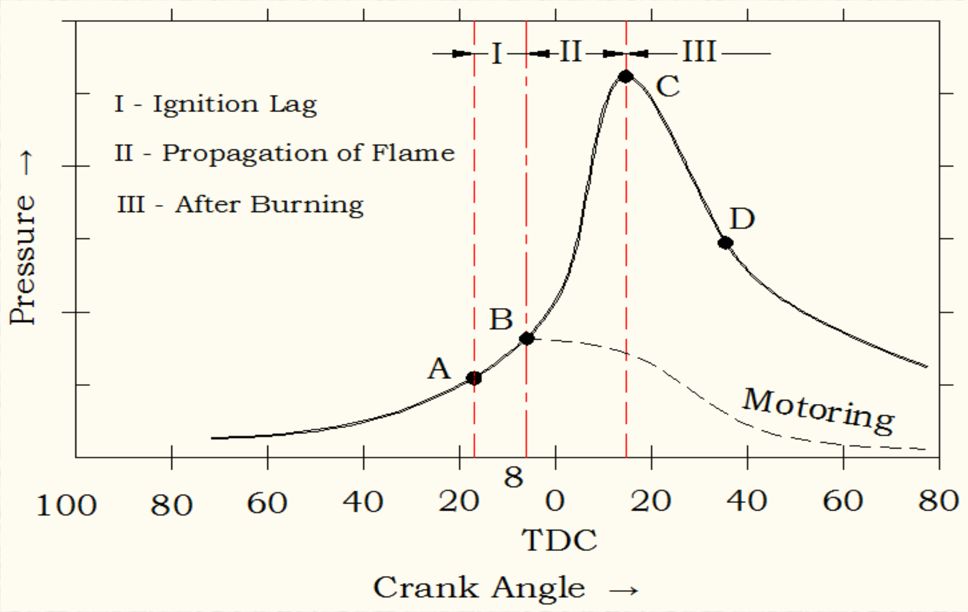
9.1. Ideal Stages of Combustion in SI Engine -

An Ideal or theoretical pressure crank angle diagram, during the process of compression (process a-b) piston moves toward TDC and compresses the entrapped charge till the point–b. At TDC the spark is ignited and combustion take place instantaneously and the entire pressure rises during the combustion take place at constant volume i.e. at TDC, shown by the process b-c. The peak pressure attend at the point–c, and then the expansion of gases take place from point–c to point-d.



9.2. Actual Stages of Combustion in SI Engine - SPPU : May-15, Dec.-14, 6-Marks

In an ideal 4-stroke engine, the entire pressure rise during combustion takes place at TDC at constant volume, but in actual engine this does not happen. The processes of combustion and pressure variation is shown as below,



In figure above, point A shows the spark ignition (18° bTDC), point B is the point at which the starting of pressure rise can be detected (8° bTDC) and point C shows the attainment of peak pressure (16° aTDC).

Thus A-B represents the first stage of combustion – Ignition Lag, B-C represents the second stage of combustion – Propagation of Flame and C-D represents third stage of combustion – After Burning.

Stage I - Ignition Lag (A-B) –

A-B is known as ignition lag or preparation phase in which growth and development of a self propagating nucleus of flame takes place. This is a chemical process depending upon temperature, pressure, type of fuel and residual exhaust gas.

Stage II – Propagation of Flame (B-C) –

B-C is known as flame propagation phase, it is concerned with the spread of the flame throughout the combustion chamber. The starting point is the first measurable rise of pressure is seen on the indicator diagram, where the line of combustion separated from compression line i.e. motoring curve at point B.

During the second stage the flame propagates practically at a constant velocity. The rate of heat release depends on the turbulence and the reaction rate which depends on the mixture composition. And the rate of pressure rise is proportional to rate of heat released.

The maximum/peak pressure is observed after TDC, shown by point C.

Stage III – After Burning (C-D) –

C-D is known as the after burning phase, it is started at a point of peak pressure (point C). Since the expansion stroke starts before this stage, the piston moving away from the TDC, there can be no pressure rise during this stage.

The flame velocity decreases during this stage, the rate of combustion becomes low due to lower flame velocity and reduced flame from surface.

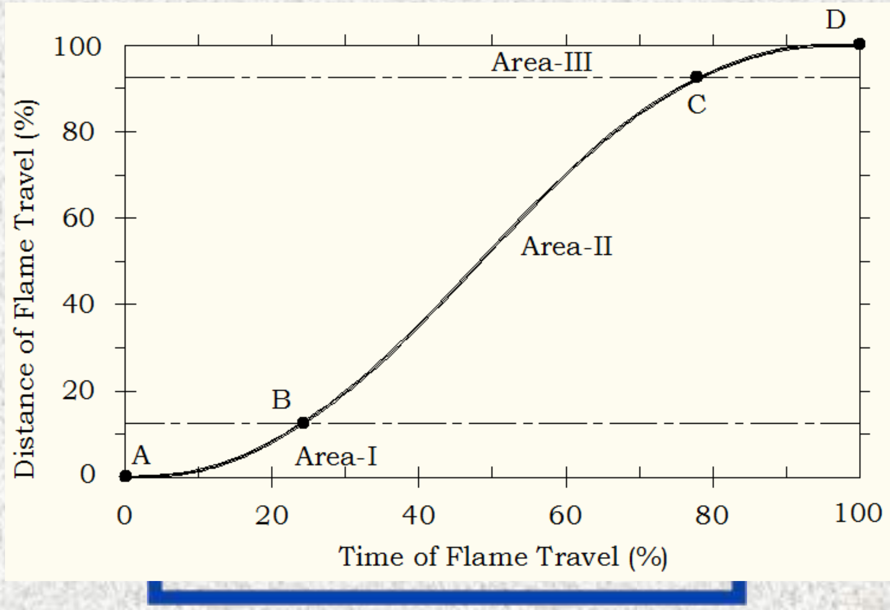
10. Flame Propagation :

Flame propagation mainly depends on two factors, *Reaction Rate* and *Transposition Rate*.

The reaction rate is the result of a chemical combination process in which the flame eats its way into the unburned charge.

The transposition rate is due to the physical movement of the flame front relative to the cylinder wall and is also the result of the pressure differential between the burning gases and the unburnt gases in the combustion chamber.

In area-I (A-B), the flame front progresses relatively slowly due to a low transposition rate and low turbulence. The transposition of the flame front is small as there is a comparatively small mass of charge burned at the start. Also the low reaction rate resulting in a slow advance of the flame, this is due to quiescent layer of gas (the zone of A/F mixture gas near spark plug closed to cylinder wall which have lack of turbulence).



In area-II (B-C), where flame front leaves the quiescent zone and proceeds into more turbulent areas and it consumed a greater mass of mixture, it progresses rapidly. The transposition rate and reaction rate is high all the time, due to high turbulence and greater mass of charge availability for flame propagation.

In area-III (C-D), the volume of unburned charge is very much less towards the end of flame travel and so the transposition rate again becomes negligible therby reducing the flame speed. The reaction rate also reduced as flame again entering the end charge zone of low turbulence.

11. Rate of Pressure Rise :

The rate of pressure during the combustion process has a considerable influence on the peak pressure obtained and consequently it affects the power output and the smooth running of the engine.

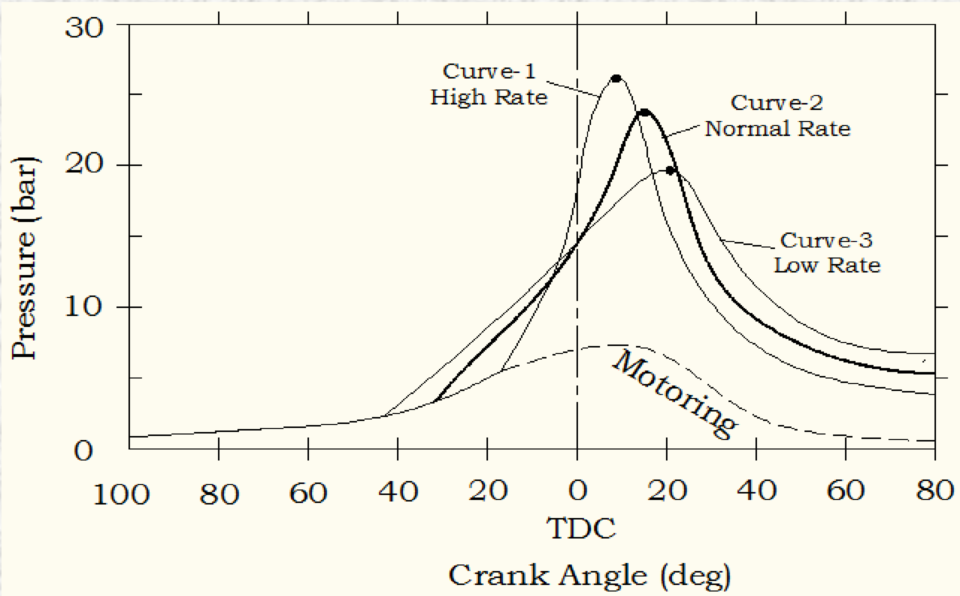
The rate of pressure rise depends on the rate at which the mass of mixture burns in the cylinder and the ignition timing.

In the given figure of pressure rise against crank angle, Curve-1 represents the high rate of combustion results in higher rate of pressure rise producing higher peak pressure at a point closer to TDC.

This high rate of pressure rise causes rough running of the engine due to vibrations and jerks produced on the crankshaft. Also it creates the undesirable situation of abnormal combustion called detonation.

Curve-3 represent a low rate of combustion which takes longer time for the completion of combustion which required initiation of burning at early point on the compression stroke.

Curve-2 represents normal rate of pressure rise, which is a compromise between curve-1 and curve-3, this results in the peak pressure being reasonably close to the beginning of the power stroke. This is accomplished by designing and operating the engine in such a manner that approximately one half of the maximum pressure is reached by the time the piston reaches TDC.



12. Abnormal Combustion : SPPU : May-16, 6-Marks

In normal combustion the flame initiated by the spark travels across the combustion chamber in a fairly uniform manner. Under certain operating conditions the combustion deviates from its normal course leading to loss of performance and possible damage to the engine. This type of combustion may be termed as an abnormal combustion or knocking combustion. The consequences of this abnormal combustion process are the loss of power, recurring pre-ignition and mechanical damage to the engine.

12.1. The Phenomenon of Detonation in SI Engine :

SPPU : May-17, May-14, 6-Marks

In SI engine combustion is initiated at spark plug and the flame spreads across the combustion chamber and consumed the combustible mixture. During combustion of mixture the flame front which separates the fresh mixture from the burnt.

In the normal combustion the flame travels across the combustion chamber from A towards D. The advancing flame front compresses the end charge BDB' as the burnt gases tries to expand and exert pressure to end charge. This raises the pressure and temperature of end charge. Also the heat transfer from burnt gases to fresh end charge raise the temperature. Inspite of all these if end charge does not autoignite and get consumed by the flame front. Then there will be no any knock and the combustion is known as normal combustion.

Figure below shows cross –section of combustion chamber with flame front advancing from the spark plug to end charge.

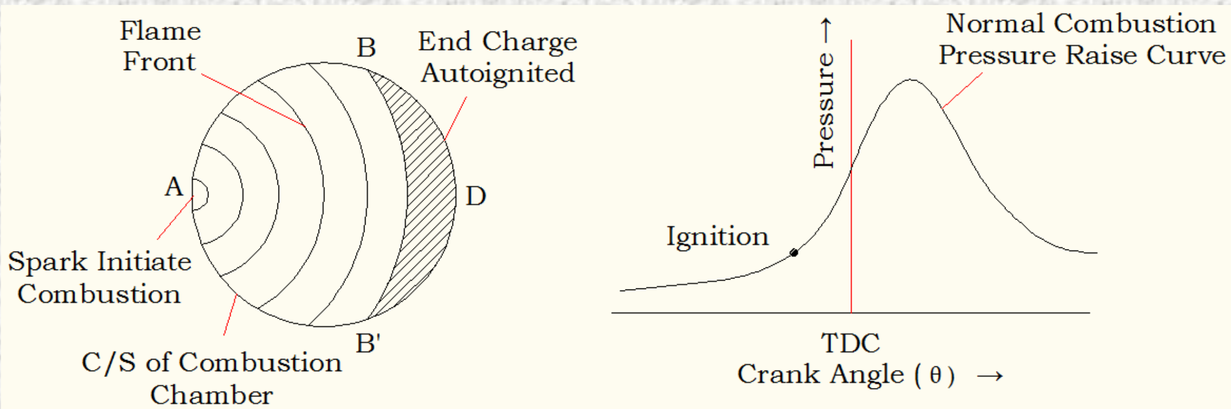
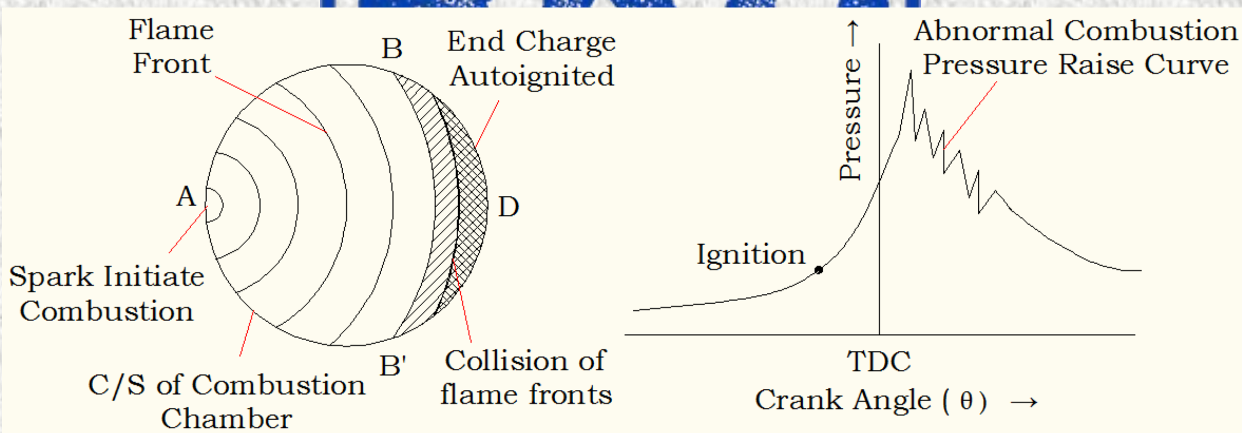


Figure below shows abnormal combustion, which means the end charge BDB' reaches to self ignition temperature and get auto-ignited before the flame front reach to end charge.



When main flame front from spark plug travel and reaches to level BB' the fresh charge ahead get compressed and reached to the temperarure where it get autoignited and new flame front travel in opposite direction. When theses two flame front collides a pressure pulse is generated. The gas in the chamber is subjected to compression and rarefaction along the pressure pulse until pressure equilibrium is restored. This disturbance forces the walls of the combustion chamber to vibrate and produces the audible sound in the engine known as knock.

13. Effect of Engine Variables on Detonation :

SPPU : Dec.-17, May-14, 6-Marks

Detonation / Knock

Density Factor

1. Compression Ratio
2. Mass of Inducted Charge
3. Inlet Temp. of Mixture
4. Temp. of Combustion Chamber Wall
5. Retarding the Spark Timing
6. Power Output of Engine

Composition Factor

1. Fuel-Air Ratio
2. Octane Value of Fuel

Time Factor

1. Turbulence
2. Engine Speed
3. Flame Travel Distance
4. Engine Size
5. Combustion Chamber Shape
6. Location of Spark Plug

13.1. Effect of Density Factor on Knock :

13.1.1. Compression Ratio -

Increase in compression ratio increase the tendency of engine knock. Increase in compression ratio increases the pressure and temperature of the gases at the end of the compression stroke. This decreases the ignition lag of the end fresh charge with increasing the auto-ignition possibility of end charge. Also the overall increase in density of the charge due to higher compression ratio increase the pre-flame reactions in the end charge.

13.1.2. Mass of Inducted Charge -

Increase in the inducted charge mass increases the tendency of engine knock. With the increase in the inducted charge mass, both the density and the temperature of the charge increases at the time of ignition. Which may promote the end charge for the auto-ignition.

13.1.3. Inlet Temperature of the Mixture -

Increase in the inlet temperature of the mixture, increases the tendency of engine knock. As the inlet temperature of mixture increase it makes the compression temperature higher thereby increasing the tendency of knocking.

13.1.4. Combustion Chamber Wall Temperature -

Increase in the combustion chamber wall temperature, the tendency of engine knock increases. As the overheating or hot spot of the combustion chamber act as an additional source for ignition and thus knocking may increases.

13.1.5. Retarding/Advancing the Spark Timing -

Retarding the spark timing, reduces the tendency of engine knock, as retarding spark timing from optimized timing, the peak pressure are reached further down on the power stroke and thus the lower the magnitude which reduce the knocking.

13.1.6. Power Output / Engine Output -

Increase the power output increases the tendency of engine knock. As increase in power output, the temperature of the cylinder and the combustion chamber walls and also the pressure of the charge increases, thereby increasing the end charge temperature. This increases the tendency of engine knock.

13.2. Effect of Composition Factor on Knock :

13.2.1. Fuel-Air Ratio -

Fuel-Air ratio affect the flame temperature and the reaction time. Flame speed is higher at the little richer mixture than stochiometic mixture, this minimize the time for complete combustion, thus minimize the reaction time for auto ignition. This result in lowering the tendency of engine knock.

But for leaner and richer mixture reduces the flame speed and thus sufficient time is available for the end charge reaction to get auto ignited. Thus increases the engine tendency to knock.

13.2.2. Octane Value of the Fuel -

Increase in octane rating of fuel decreases the tendency of engine to knock. A higher octane number means the higher self-ignition temperature of the fuel which reduce the tendency of knocking.

13.3. Effect of Time Factor on Knock :

13.3.1. Turbulence -

Increases in turbulence decrease the tendency of engine to knock. Increasing turbulence increases the flame speed and reduces the time available for the end charge to attain auto-ignition conditions thereby decreasing the knock tendency of the engine.

13.3.2. Engine Speed -

An increase in engine speed increase the turbulence of the mixture and thus result in increase in flame speed and reduces the tendency of engine knock.

13.3.3. Flame Travel Distance -

The tendency of engine knock is reduced by shortening the time for flame front to travel and reach to end charge. Engine size and spark plug position are the factors governed the flame travel distance.

13.3.4. Engine Size -

The flame requires a longer time to travel across the combustion chamber of a large engine. Which may gives sufficiency time for the end charge to reach the auto-ignition. Thus larger size engine has a greater tendency for engine knocking

13.3.5. Shape / Compactness of Combustion Chamber -

The more compact combustion chamber, lesser the flame travel distance and combustion time, hence lesser the tendency to engine knock. Also shape of the combustion chamber should promote the turbulence so that flame speed will increase and knocking tendency will decreases.

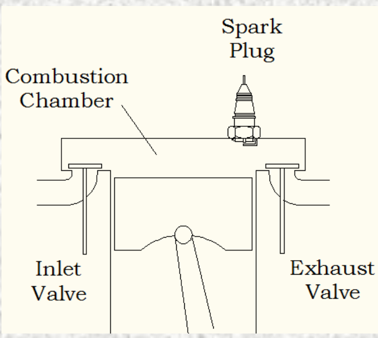
13.3.6. Location of Spark Plug -

Spark plug is centrally located in the combustion chamber, to minimize the flame travel distance. This result in minimum knocking tendency. There for more than two spark plug may used for the large size engine.

13.4. Types of Combustion Chambers for SI Engines : SPPU : May-17, 6-Marks

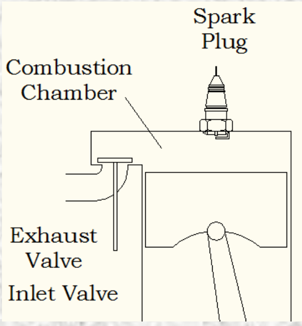
13.4.1. T-Head Combustion Chamber :

The T-head combustion chambers were used in the early stage of engine development. It have two valves on either side of the cylinder and thus required two cam shafts to operate them. The distance across the combustion chamber is very long, hence knocking tendency is high in this type of engine.



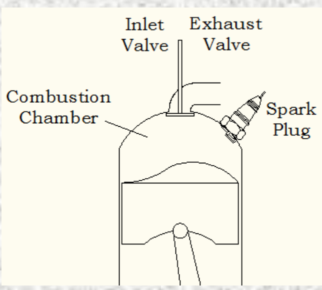
13.4.2. L-Head Combustion Chamber :

The T-head combustion chamber is modified into L-head type, which have two valve on the same side of cylinder and operated by a single camshaft. The air flow has to take two right angle turns to enter the cylinder. This causes a loss of velocity head and a loss in turbulence level resulting in a slow combustion.



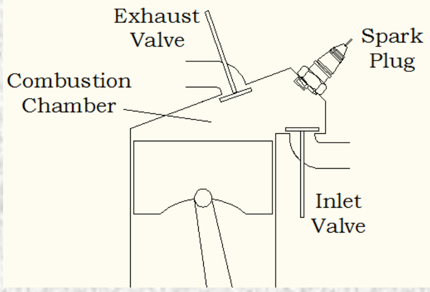
13.4.3. I-Head Combustion Chamber :

The I-head type is also called overhead valve combustion chamber in which both the valve are located on the cylinder head. This arrangement have less surface to volume ratio and therefore less heat loss, less flame travel length and hence greater freedom from knock. Also it have higher volumetric efficiency from larger valves lifts.



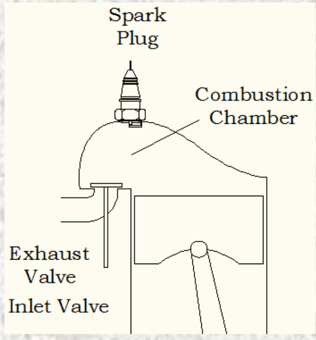
13.4.4. F-Head Combustion Chamber :

The F-head type arrangement is the compromise between L-head and I-head type combustion chambers. In this type one valve is in the cylinder head and the other in the cylinder block. This required two separate cam shaft as both valve needed to be operated separately.



13.4.5. Recardo Turbulent Combustion Chamber :

Recardo combustion chamber overcomes the disadvantage of L-head combustion chamber. This provides a turbulent head. In this main body of the combustion chamber is concentrated over the valves leaving a slightly restricted passage communicating with the cylinder thereby creating additional turbulence during the compression stroke. This arrangement reduce the flame travel and thus knocking tendency of engine. In this type removal of cylinder head without disturbing valve gears etc. is possible and also it make easy to lubricate the valve mechanism.



14. Rating of Fuels in SI Engines

SI engine fuels are rated for their antiknock quality. It is called as Octane Number, which represents the ability of fuel to resist knocking.

To determine of antiknock value of SI engine fuel, it is compared with a mixture of two reference fuels, Iso-Octane (C_8H_{18}) which assigned a 100 octane number and N-Heptanes (C_7H_{16}) which assigned a zero octane number.

The octane number of a fuel is defined as the percentage (by volume) of iso-octane in a mixture of iso-ocatne and n-heptanes, which exactly matches the knocking intensity of the given fuel in a standard engine under standard operating conditions.

14.1. Determination of Octane Number :

Octane number determination is carried out in three steps as follow,

Step – 1 : Standard Knock : Variable compression engine is run at specified condition with definite compression ratio and a definite blend of reference fuels (i.e. blend mixture of iso-octane and n-heptane). The intensity of knock at these standard conditions is called standard knock. The knock meter is adjusted to give a particular reading under these conditions.

Step – 2 : Test Fuel Run : The test fuel is now used and air-fuel ratio is adjusted to give maximum knock intensity. The compression ratio of the engine is gradually changed until the knock meter reading shows the readings of standard knock. Now this compression ratio is fixed.

Step – 3 : Reference Fuel Run : Now known blend of reference fuel (i.e. mixture of iso-octane and n-heptane) is used in engine, and check the knock readings. The blend of reference fuel which gives a knock meter readings to the standard knock, will match the knocking characteristics of test fuel. The percentage of the iso-octane in the mixture of reference fuel blend is the octane number of the test fuel.

14.2. Performance Number :

The performance Number (PN) is defined as the ratio of Knock Limited indicated mean effective pressure (KL_{imep}) of the test fuel to the iso-octane.

Where, Performance Number (PN) is calculated as,

$$PN = \frac{KL_{imep}\ of\ Test\ Fuel}{KL_{imep} of\ Iso - Octane}$$

In the reference fuel, iso-octane is given octane number 100. The addition of certain compounds (tetraethyl lead) to iso-octane produces fuels of greater antiknock quality above 100 octane number.

Further each octane number at the higher range of the octane scale will produce greater antiknock effect compared to the same unit at the lower end of the scale. (e.g. octane number increase form 92 to 93 produce greater antiknock effect than a same unit increase from 32 to 33 octane number).

Because of this non-linear variation, a new scale was derived which expresses the approximate relative engine performance and the units of this scale are known as the Performance Number (PN). Octane Number (ON) above 100 can be computed by,

$$Octane\ Number\ (ON > 100) = 100 + \frac{PN - 100}{3}$$

14.3. Highest Useful Compression Ratio (HUCR) :

Highest Useful Compression Ratio (HUCR) is define as the highest compression ratio at which the fuel can be used in an engine without any detonation/knock, at the specified standard test engine under standard operating conditions.

Some of the following fuels are tested for HUCR on Recardo E6 engine,

Sr. No.	Test Fuel	HUCR
1	n-heptane	3.75
2	Iso-Octane	10.96
3	Benzene	14.6
4	Toluene	15.0

15. Additives / Dopes for SI Engines :

To enhance the anti-knock characteristics of the fuel, certain compounds are added, these compounds are known as additives / dopes.

These additives / dopes must remain stable and in liquid form at atmospheric conditions and should vaporize in intake manifolds with out any harmful deposition on engine parts.

Most common additive used is Tetra Ethyl Lead [Pb $(C_2\,H_5)_4$], called as TEL. Lead is the heavy metal and get deposited on spark plug and cause the fouling of spark plug and it deposition on exhaust valve causes corrosion, to avoid this TEL added with Ethelene Dibromide [$C_2\,H_4\,Br_2$] to make a volatile mixture, and commonly called as Ethyl Fluid.

One percent of ethyl fluid by volume, it brings down the appreciable knocking tendency in the engine and increases the octane number rating of fuels.

There are other metallic dopes such as, $(CO)_6$, $Fe\,(CO_3)_2$ and non-metallic dopes like methyl aniline $(C_7\,H_9\,N)$, toluidine $(C_7\,H_{11}\,N)$, aniline $(C_5\,H_7\,N)$ etc.

Exercise

1. Define Carburetion and explain the principal of carburetion.
2. List out the requirement of a good carburetor.
3. What are different air fuel ratio used in an engine.
4. Explain in short, *i.* Rich mixture, *ii.* Stoichiometric mixture, *iii.* Lean mixture.
5. Explain the working of simple carburetor.
6. Derive the expression for air fuel ratio of a simple carburetor.
7. Describe the essential parts of a modern carburetor.
8. Explain with neat sketch,
    a. Main metering system
    b. Idling system
    c. Economizer system
    d. Acceleration pump system
    e. Choke system.
9. Explain the basic types of carburetors.
10. What are the drawback of simple carburetors.
11. Explain the MPFI system for gasoline engine with the help of neat sketch.
12. Briefly explain the stages of combustion of SI engine.
13. Explain the various factors influence the flame speed.

14. What is meant by abnormal combustion ?
15. Explain the phenomenon of detonation in SI engine.
16. Explain the various types of combustion chambers used in SI engine.
17. Discus the effect of the following variable on flame propagation,
    a. Fuel-Air ratio
    b. Compression ratio
    c. Engine load
    d. Turbulence
18. Write short note on Pre-ignition and Auto-ignition.
19. Write short note on HUCR.
20. What is ignition lag.
21. What is Octane Number and how it is determine for the fuel.
22. What is dopes used for SI engine
23. Explain the requirement of a good combustion chamber of SI engine.
24. Draw the neat sketch of the following combustion chambers used in SI engines,
    a. T-headed combustion chamber
    b. F-headed combustion chamber

25. A simple jet carburetor is required to supply 5 Kg of air and 0.5 Kg of fuel per minute. The fuel specific gravity is 0.75. the air is initially at 1 bar and 300 K. Calculate the throat diameter of the choke for a flow velocity of 100 m/sec. Velocity coefficient is 0.8 of that of the choke, calculate orifice diameter assuming, $C_{d\_fuel} = 0.6$ and $\gamma = 1.4$. (***Ans.*** $\boldsymbol{d_{fuel} = 2.34\ mm}$)
26. A 4-stroke petrol engine has a swept volume of 2 liters and is running at 400 rpm. The fuel-air ratio is 1 : 14 and the volumetric efficiency at this speed is 75%. The venture throat diameter of the carburetor fitted to the engine is 3cm. Determine the air velocity at the throat if the discharge coefficient for air is 0.9. The ambient pressure and temperature are 1 bar and 20 °C respectively. Find the diameter of the fuel jet if the specific gravity of the fuel is 0.76. Assume the coefficient for fuel flow is 0.6 and pressure drops to 0.96 bar at the throat. (***Ans.*** $\boldsymbol{C_2 = 82\ m/sec, d_{fuel} = 1.95\ mm}$)

27. A simple jet carburetor is required to supply 0.1 Kg of air per second and 0.45 Kg/min of fuel of density 740 Kg/m^3. The air is initially at 1.013 bar and 300 K.

    a. Calculate the throat diameter of the choke for a flow velocity of 92 m/sec. and velocity coefficient is 0.8.

    b. If the pressure drop across the fuel metering orifice is 0.75 of that at the choke, calculate the orifice diameter. Assume coefficient of discharge is 0.6. (***Ans.*** $\boldsymbol{d_{air} = 31.3\ mm,\ d_{fuel} = 2.34\ mm}$)

28. A petrol engine consumes 10 Kg of petrol per hour. Fuel air ratio is 0.06 : 1. Coefficient of discharge for the venture is 0.8 and that for main petrol jet0.75. Choke diameter is 23 mm. Top of the jet is 6 mm above the petrol level in the float chamber. Calculate size of fuel jet of a simple carburetor. Take specific gravity of petrol as 0.75, atmospheric pressure 1 bar and air temperature 25 ºC. (***Ans.*** $\boldsymbol{d_{fuel} = 0.366\ mm}$)

Prof. Rahul Krishnaji Bawane
Pimpri Chinchwad College of Engineering & Research, Ravet, Pune.

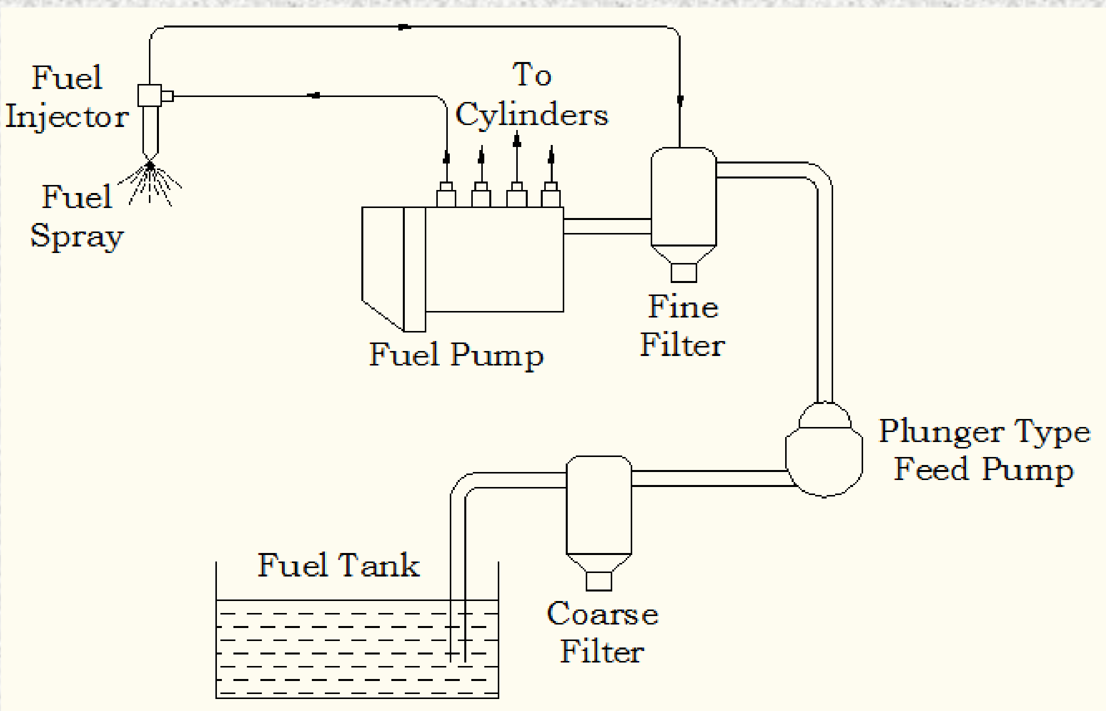
Unit – III : Compression Ignition (CI) Engines

Fuel supply system, types of fuel pump, injector and distribution system, Combustion in compression ignition engines, stages of combustion, factors affecting combustion, Phenomenon of knocking in CI engine. Effect of knocking, Methods of knock control, Types of combustion chambers, rating of fuels in CI engines. Dopes & Additives, Comparison of knocking in SI & CI engines.

1. Fuel Supply System :

The fuel injection system is the most important part of the working of CI engines. The engine performance is greatly dependents on the effectiveness of the fuel injection system.

The injection system has to perform the important duty of initiating and controlling the combustion process of CI engines.



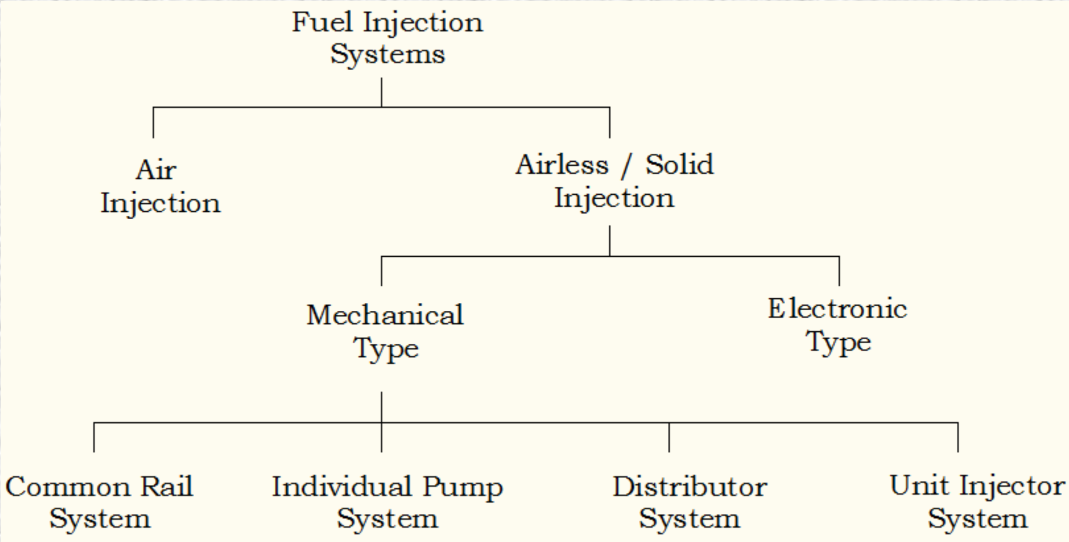
1.1. Requirement of Fuel Injection System -

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

1. Accurate metering of the fuel injected per cycle – the quantity of the fuel metered should vary to meet changing speed and load requirements of the engine.
2. Proper control of rate of injection – to achieved desired heat release patter during combustion.
3. Accurate injection timing in the cycle – to ensure the maximum power output with clean burning and fuel economy.
4. Proper atomization of fuel and spray patter – to ensure very find droplet spray for rapid vaporization and mixing of fuel and air.

5. Uniform distribution of fuel droplets in the combustion chamber – to avoid rich mixture zone and abnormal combustion.
6. No lag during beginning and end of injection – to eliminate dribbling of fuel droplets into the cylinder.

2. Classification of Fuel Injection System :



2.1 Air Injection System -

In this system, fuel is forced into the cylinder by means of compressed air. It needs multistage compressor to supply compressed air at about 70 bar pressure and the fuel pump to draw the desired fuel from fuel tank. The mixture of compressed air and fuel is injected hence called as air injection system.

Advantages of Air Injection System –

1. It provides good atomization of fuel.
2. Heavy viscous fuels which are cheap can be used.
3. Fuel pump needs to develop only small pressure as injection assisted by high pressure compressed air.

Disadvantages of Air Injection System –

1. It cannot be used for portable engine due to requirement of air compressor.
2. The compressor run from engine power, thus neat power output and mechanical efficiency get reduced.
3. Due to multistage compressor, the unit becomes bulky and expensive.
4. It needed separate maintenance of air compressor.

2.2 Airless / Solid Injection System -

In this system, fuel in liquid form is injected directly into the combustion chamber without any assistance of compressed air. Hence this system is called as airless or solid (not a mixture of air and fuel) injection system.

When the fuel is injected into combustion chamber at pressure about 70 bars helps it to atomized and form a fine spray which get vaporized by the high temperature of cylinder compressed air, and form air-fuel mixture.

This system comprise mainly following components,

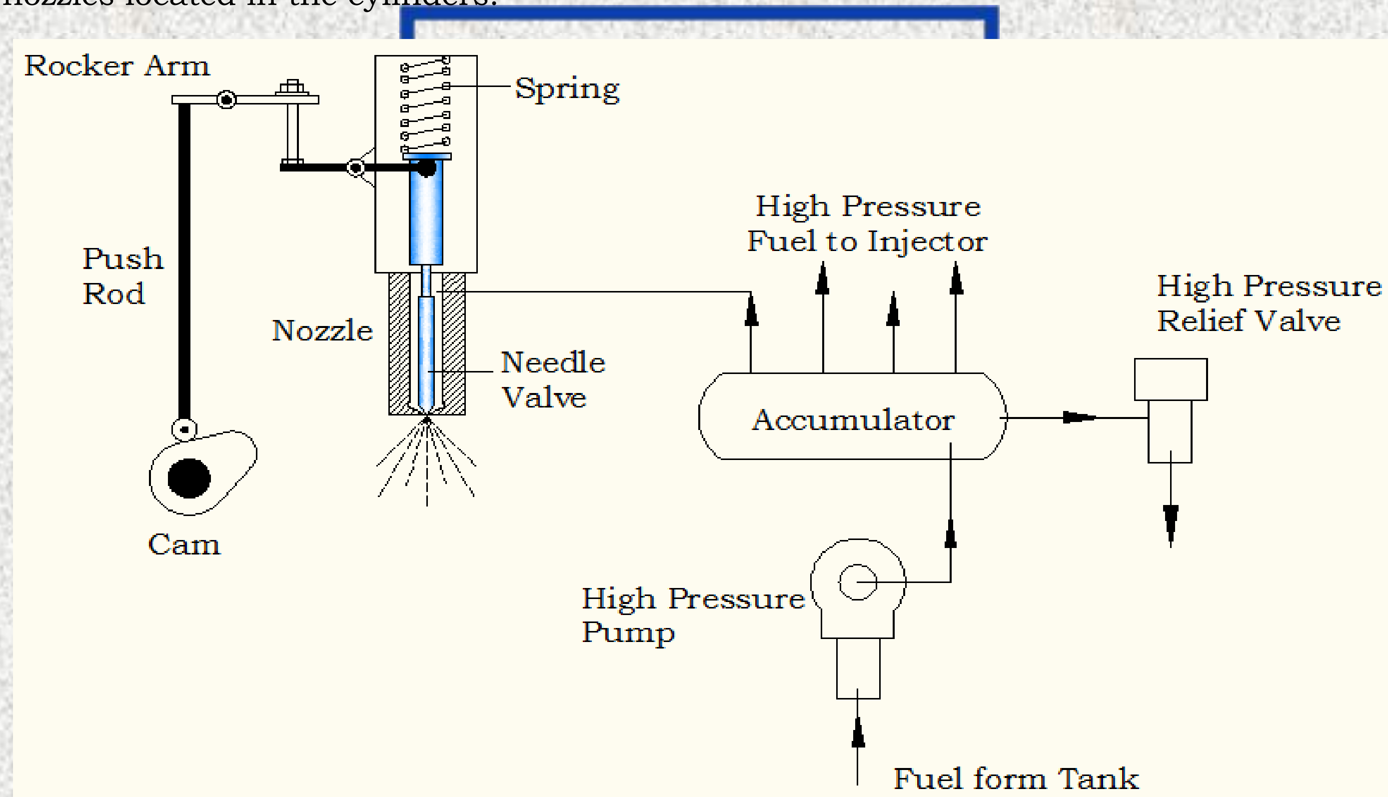
1. Fuel tank
2. Fuel filters – to prevent dust and abrasive particles from entering the pump and injectors to minimize the wear and tear of the components.
3. Fuel feed pump – to supply fuel from the main fuel tank to the injection system.
4. Injection pump – to meter and pressurize the fuel for injection.
5. Governor – to ensure that the amount of fuel injected is in accordance with variation in load.
6. Injector – to take the fuel from the pump and distribute it in the combustion chamber by atomizing it into fine droplets.

2.3. Mechanical Injection Systems –

Mechanical injection system have the following types,

2.3.1. Common Rail (CRDI-common rail direct injection) system –

In the common rail system a high pressure (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 push rod and rocker arm arrangement open valve and allow fuel to enter the respective cylinder through the nozzle. The pressure in the fuel header is maintained higher which enable fuel to get injected and spread in the combustion chamber.

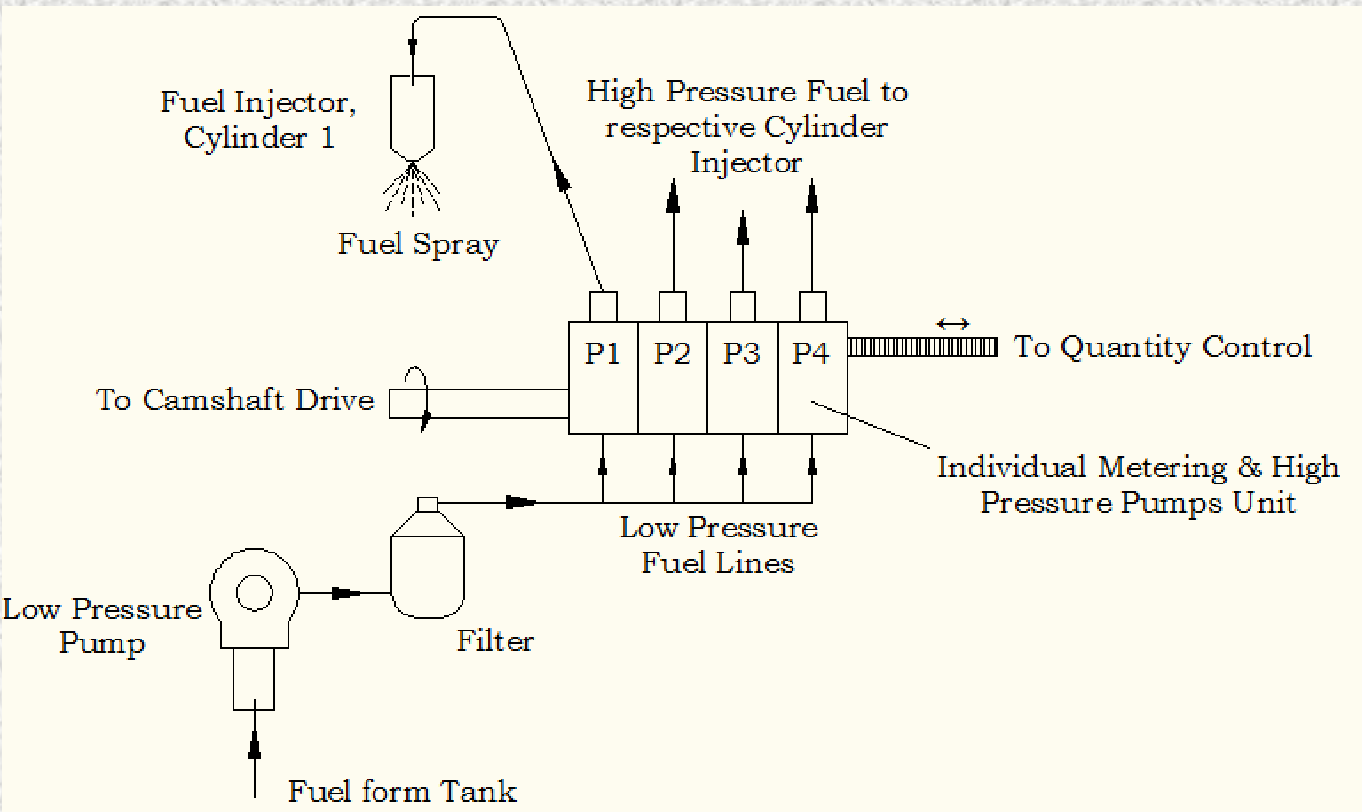
The amount of fuel injected can be controlled by controlling the length of period of push rod stroke.

2.3.2. Individual Pump Injection system –

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. Fuel from tank is supplied to low pressure (LP) pump through which raises pressure to about 2.5bar and supplied to the fuel to individual high pressure (HP) pump of respective cylinder.

HP pumps increase the pressure to about 100 bar and above and meter the amount of fuel to be injected by the effective stroke of the plunger.

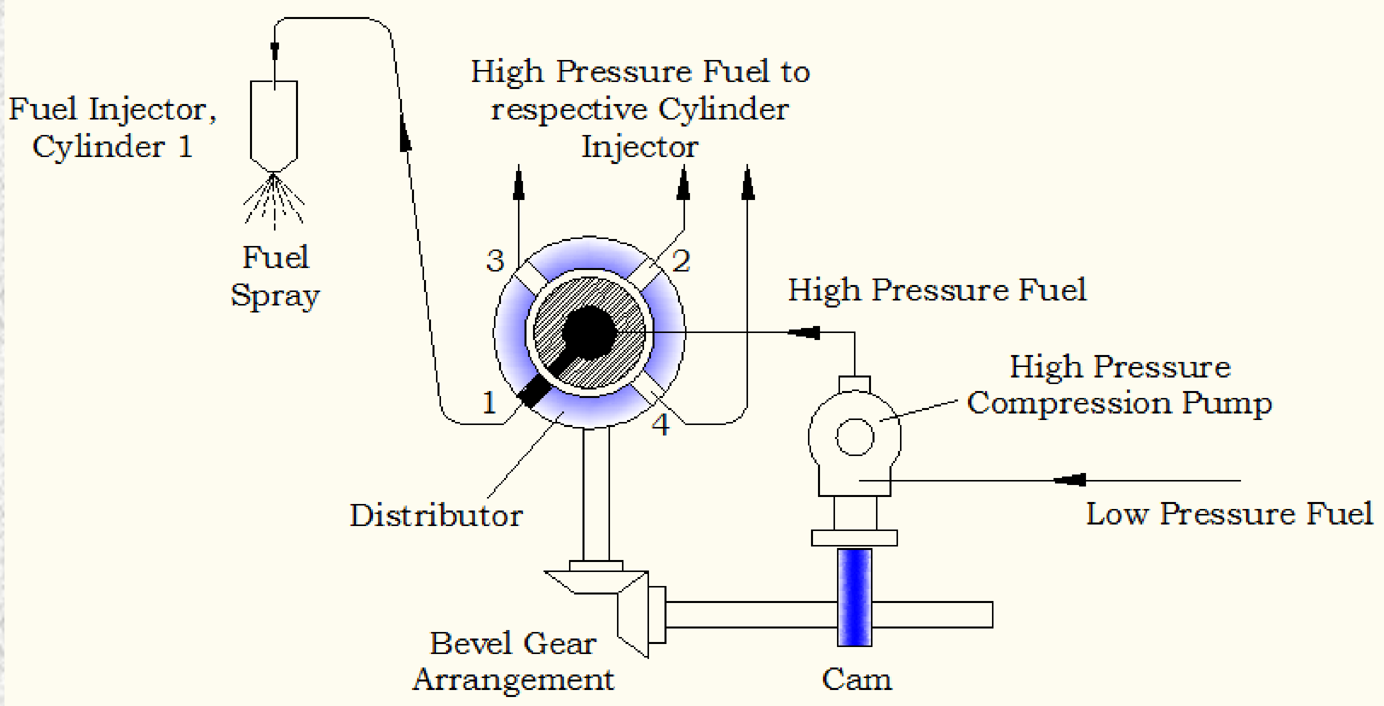
The HP pumps are placed close to the cylinder. The high pressure pump plunger is actuated by a cam and produces the fuel pressure necessary to open the injector valve at the correct time.



2.3.3. Distributor Injection system –

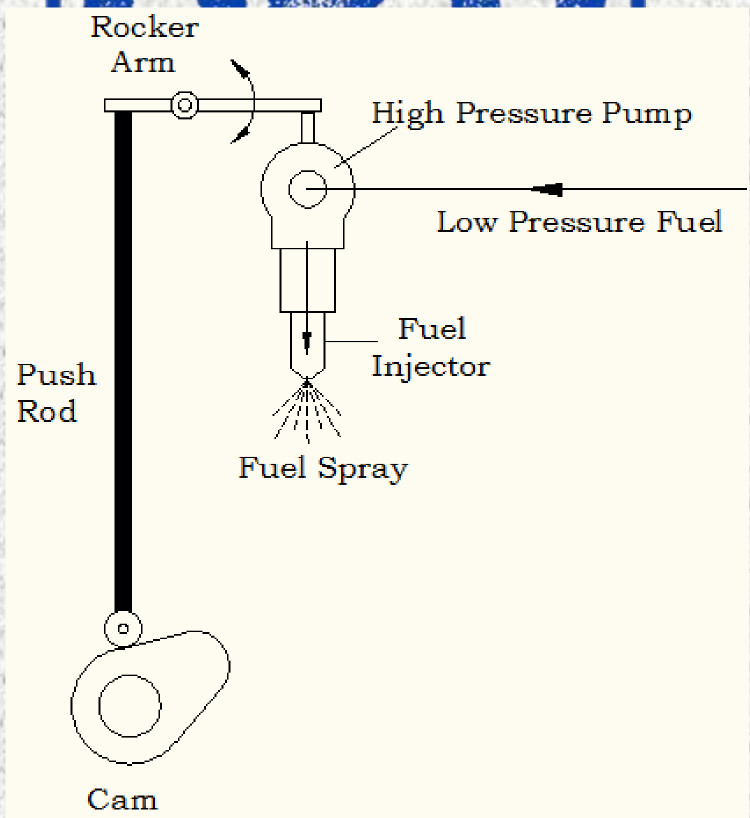
In this system the pump which pressurizes the fuel also meters and times it. The fuel pump after metering the required amount of fuel supplies it to a rotating distributor at the correct time for supply to each cylinder.

The number of injection strokes per cycle for the pump is equal to the number of cylinders. There is one metering element in each pump, a uniform distribution is ensured also cost of the fuel injection system is less than that of individual pump system.



2.3.4. Unit Injector system –

In the unit injector system the pump and the injector nozzle are combined in one housing and integrated as one unit.



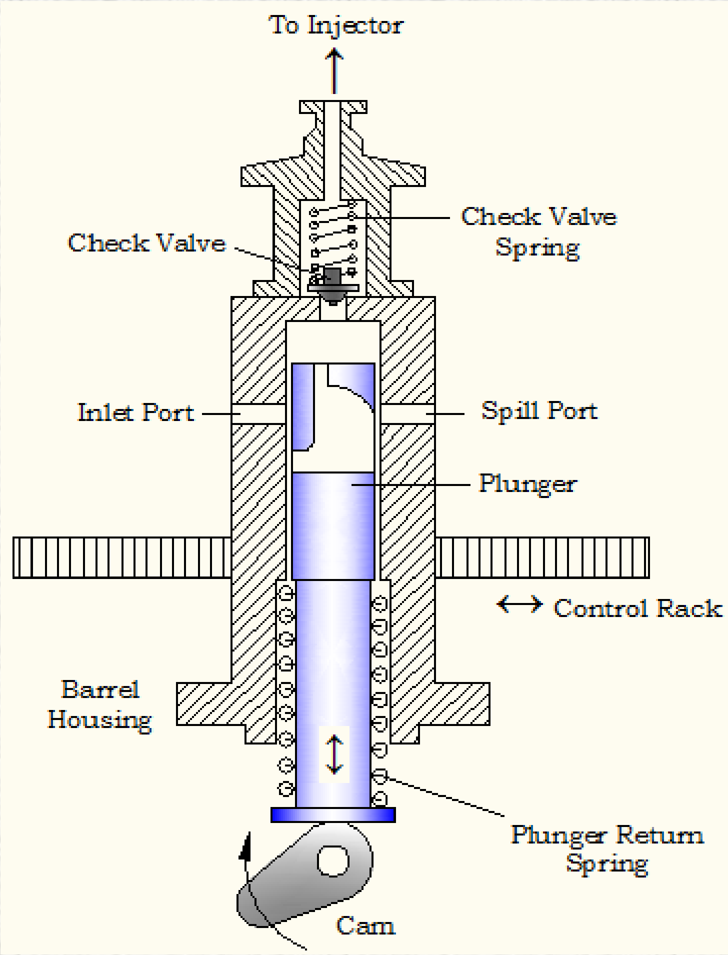
Each cylinder is provided with one of these unit injectors. Fuel is brought up to the injector by a low pressure pump where at the proper time a rocker arm actuates the plunger and thus injects the fuel into the cylinder. The amount of fuel injected is regulated by the effective stroke of the plunger.

3. Types of Fuel Pump :

The main function of fuel pump is to deliver accurately metered quantity of fuel under high pressure in the range of 100bar to 200 bar, at the correct instant to the injector.

There are following two types of fuel pumps,

3.1.1.Working of Jerk Pump (Bosch Fuel Pump) -



Bosch fuel pump is a jerk type pump which consist of a barrel in which a plunger reciprocates when driven by a camshaft. The plunger has a constant stroke and is single acting. The pump barrel has two radially opposite holes, one is inlet port and other is spill port.

During the delivery stroke a cam raises the plunger up against the plunger return spring. The plunger has vertical groove extended from top face of plunger to helix.

When plunger at bottom of stroke barrel get filled by fuel through inlet port and up stroke of plunger when closes port then entrapped fuel get compressed by further plunger movement. The high pressure fuel then moves out from check valve by lifting it against spring and fuel is delivered to injector.

As soon as helix groove open spill port the pressure get released and the check valve spring force to stop the delivery.

A rack and pinion arrangement is provided to rotate the plunger, thus the quantity of fuel delivered per stroke is regulated.

4. Fuel Injector :

Fuel injector atomize the fuel into very fine droplets it increase the surface are of the fuel droplets resulting in better mixing which ensure complete combustion. Atomization is done by forcing the fuel through a small orifice of nozzle under high pressure.

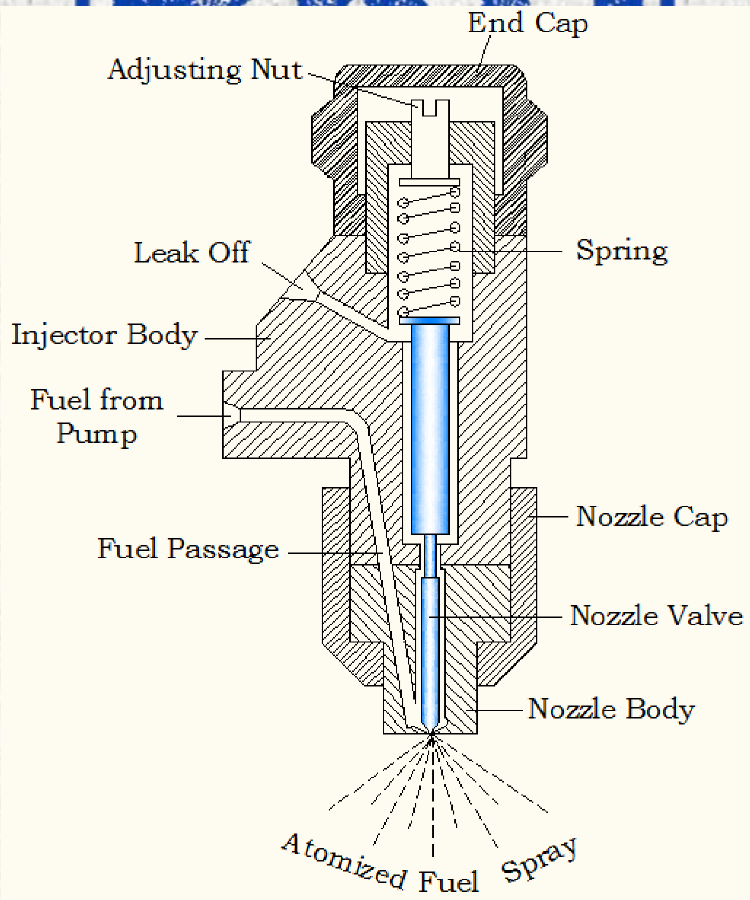
The fuel injector consist of following parts,

1. Needed valve
2. Compression spring
3. Nozzle
4. Injector body

The fuel from fuel pump is fed to the nozzle mouth through fuel passage, the fuel pressure acts on the nozzle valve which lifts it against the spring and allows the fuel to enter into the combustion chamber in the form of atomized spray though orifice.

One the fuel from delivery pump gets exhausted, the spring tension pushes the nozzle valve back on its seat. To provide the lubrication between nozzle valve and its guide a small quantity of fuel is allowed to leak through the clearance and then drained back to the fuel tank though leak off port.

The adjusting screw provided on the top of spring house is used to adjust the spring tension and hence the valve opening pressure.



5. Injector Nozzle :

Nozzle is that part of an injector through which the liquid fuel is sprayed into the combustion chamber.

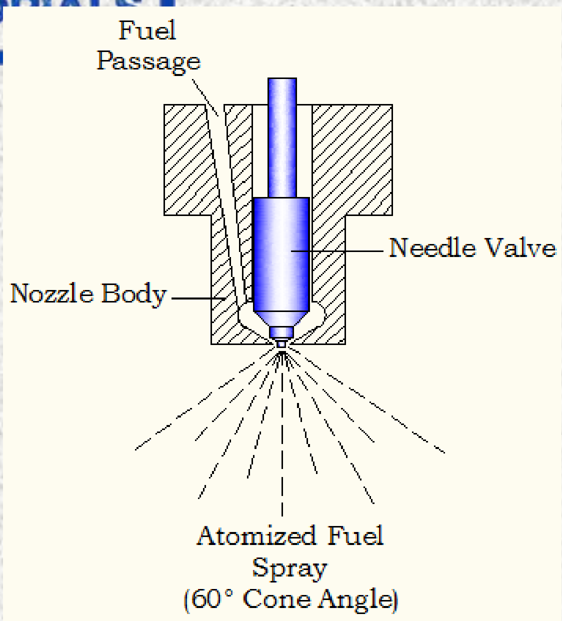
A good nozzle should fulfill the following functions,

1. **Atomization –** this is the first phase in obtaining proper mixing of the fuel and air in the combustion chamber.
2. **Distribution of fuel –** distribution of fuel to the required areas within the combustion chamber for complete combustion. Following factors are affect the distribution,
    a. **Injection Pressure –** higher the injection pressure better the dispersion and penetration of the fuel into all the desired locations in combustion chamber.
    b. **Density of air in the cylinder –** if the density of compressed air in the combustion chamber is high then the resistance to the movement of the droplets is higher and dispersion of the fuel is better.
    c. **Physical properties of fuel –** the properties like self ignition temperature, vapour pressure, viscosity etc. play an important role in the distribution of fuel.
3. **Prevention of impingement on walls –** this is necessary because fuel striking the walls decomposes and produces carbon deposits, this causes smoky exhaust as well as increase in fuel consumption.
4. **Mixing –** mixing the fuel and air in case of non-turbulent type of combustion chamber should be taken care by nozzle.

5.1. Types of Nozzle :

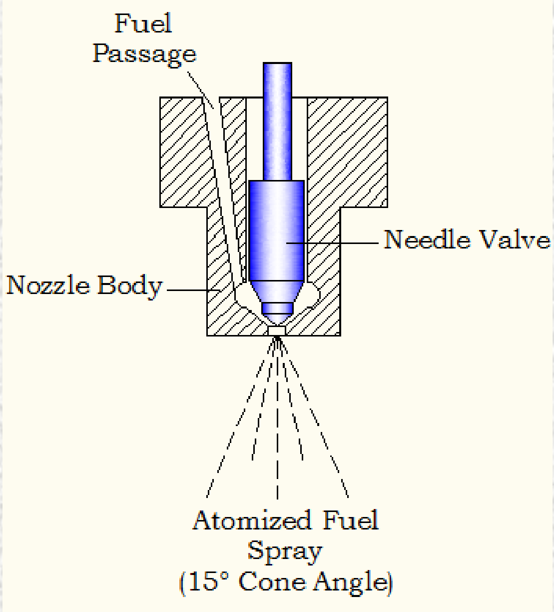
5.1.1. Pintle Nozzle –

In this type, stem of the nozzle valve is extended to form a pin or pintle which protrudes through the mouth of the nozzle. The size and shape of the pintle can be varied according to the requirement. It provides a spray operating at low injection pressure of 8-10 MPa. The spray cone angle is generally 60°. Advantage of this nozzle is that it avoids weak injection and dribbling. Also it prevents the carbon deposition of nozzle hole.



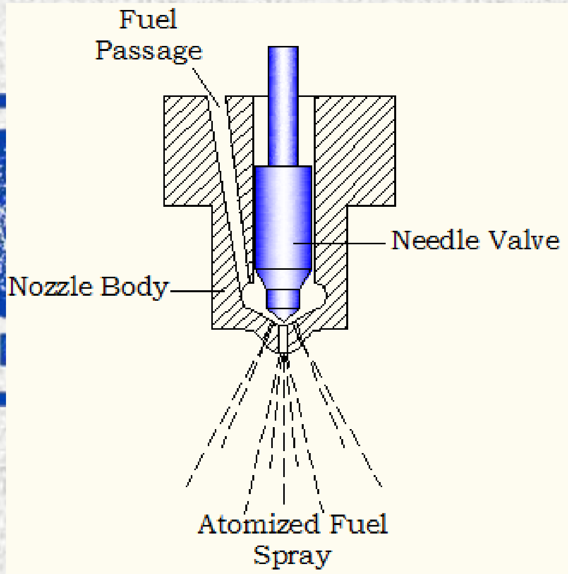
5.1.2. Single Hole Nozzle –

In this type, at centre of the nozzle body there is a single hole which is closed by the nozzle valve. The size of the hole is usually of the order of 0.2 mm. injection pressure is of order of 8-10 Mpa, and spray cone angle is about 15°. Disadvantage of this type is that they tend to dribble. Also their spray angle is too narrow to facilitate good mixing unless higher velocities are used.



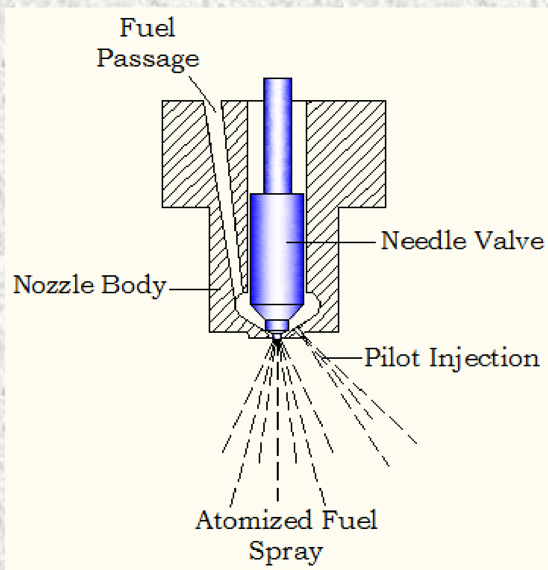
Multi Hole Nozzle –

This type consist of a number of holes bored in the tip of the nozzle. The number of holes varies from 4 to 18 and the size from 35 to 200 μm. The hole angle may be from 20° upwards. These nozzles operate at high injection pressures of the order of 18 MPa. This type has the advantage of distribute the fuel properly even with lower air motion available in open combustion chambers.



5.1.3. Pintaux Nozzle –

It is the pintle nozzle which has an auxiliary hole drilled in the nozzle body. 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. Main advantage of the nozzle is better cold starting performance. This has drawback of poorer injection characteristics than multi-hole nozzle.



6. Quantity of Fuel and Size of Nozzle Orifice :

Fuel jet velocity at the exit of orifice –

$$V_f = C_{d\_fuel} \sqrt{\frac{2\left(P_{inj} - P_{cyld}\right)}{\rho_{fuel}}} \quad in\ m/sec$$

where as, … C_{d_fuel} = coefficient of discharge of fuel from orifice

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

P_{cyld} = charge pressure inside the cylinder in N/m^2

ρ_{fuel} = fuel density in Kg/m^3

Time for one injection –

$$t_1 = \left(\frac{\theta_{Inj\_crank}}{360}\ x\ \frac{60}{N}\right) \quad in\ sec.$$

where as, … θ_{inj_carnk} = Injection crank angle in degree

N = engine speed in rpm

Volume of fuel injected per second –

Q_{fuel_vol} = (area of all orifice) x V_f x t_1 x (no. of injection per sec. for one orifice)

where as, … area of all orifice $= \frac{\pi}{4} d_o^2\ x\ n_o$ in m^2

n_o = no. of orifice

d_o = diameter of orifice

no. of injection per sec for one orifice $= \frac{N_i}{60}$

$N_i = \frac{rpm}{2}$ for 4 – stroke engine

N_i = rpm for 2 – stroke engine

Fuel consumption per cycle –

$$m_{f\_cycle} = \frac{BSFC\ x\ BP}{Cycle/hr} \quad in\ kg/cycle$$

where as, … cycle/hr $= \frac{rpm}{2}$ for 4 – stroke engine

cycle/hr = rpm for 2 – stroke engine

Fuel supply per cylinder per sec –

$$m_{fuel\_1} = \frac{m_{fuel}}{t} = C_{d\_fuel}\, A_{orifice} \sqrt{2\left(P_{inj} - P_{cyld}\right)\ \rho_{fuel}}$$

Prob. 1 – A six cylinder, four stroke diesel engine develops 125 KW at 3000 rpm. Its brake specific fuel consumption is 200 gm/KW hr. calculate the quantity of fuel to be injected per cycle per cylinder. Specific gravity of the fuel may be taken as 0.85.

Ans : No. of Cylinders, $K = 6$, 4-Stroke, $Power\ Output, BP = 125\ KW$, $N = 3000\ rpm$, $bsfc = 200\ gm/KW\ \ hr = 0.2\ Kg/KW\ \ hr$, $SG_{fuel} = 0.85$.

We know,

$$SG_{fuel} = \frac{\rho_{fuel}}{\rho_{water}}$$

$$0.85 = \frac{\rho_{fuel}}{1000}$$

$$\rho_{fuel} = 850\ Kg/m^3$$

We know, fuel consumption per hour,

$$m_{fuel} = bsfc\ x\ BP$$

$$m_{fuel} = 0.2\ x\ 125$$

$$\boldsymbol{m_{fuel} = 25\ Kg/hr}$$

Now the fuel consumption per cylinder is,

$$m_{fuel\_cyld} = \frac{m_{fuel}}{K}$$

$$m_{fuel\_cyld} = \frac{25}{6}$$

$$\boldsymbol{m_{fuel\_cyld} = 4.17\ Kg/hr\ per\ cylinder}$$

Considering fuel consumption per cycle per cylinder is,

$$m_{fuel\_cyld\_cycle} = \frac{m_{fuel\_cyld}}{N/2}$$

$$m_{fuel\_cyld\_cycle} = \frac{4.17/60}{3000/2}$$

$$m_{fuel\_cyld\_cycle} = 4.63\ x\ 10^{-5}\ \ Kg/cycle\ \ per\ cylinder$$

$$\boldsymbol{m_{fuel\_cyld\_cycle} = 4.63\ x\ 10^{-2}\ \ gm/cycle\ per\ cylinder}$$

Prof. Rahul Krishnaji Bawane
Pimpri Chinchwad College of Engineering & Research, Ravet, Pune.

We know, mass of fuel injected per cylinder per cycle can be calculated as,

$$m_{fuel\_cyld\_cycle} = V_{fuel\_cyld\_cycle}\ x\ \rho_{fuel}$$

$$V_{fuel\_cyld\_cycle} = \frac{m_{fuel\_cyld\_cycle}}{\rho_{fuel}}$$

$$V_{fuel\_cyld\_cycle} = \frac{4.63\ x\ 10^{-5}}{850}$$

$$V_{fuel\_cyld\_cycle} = 0.00000005457\ m^3/cycle\ per\ cylinder$$

$$\boldsymbol{Volume\ of\ Fuel\ injected\ = 0.05457\ cc/cycle\ per\ cylinder\ \ \ \ldots..Ans.}$$

Prob. 2 – Calculate the diameter of the fuel orifice of a four stroke engine which develops 25 KW per cylinder at 2500 rpm. The specific fuel consumption is 0.3 kg/ KW hr fuel with 30 °API (American petroleum institute gravity). The fuel is injected at a pressure of 150 bar over a crank travel of 25°. The pressure in the combustion chamber is 40 bar. Coefficient of velocity is 0.875 and specific gravity is given

$$SG = \frac{141.5}{131.5 + °API}$$

Ans. : BP = 25 KW per cylinder, N = 2500 rpm, bsfc = 0.3 Kg/KW hr, °API = 30, P_{inj} = 150 bar, crank travel = 25°, P_{cyld} = 40 bar, C_d = 0.875.

We know, duration of fuel injection is,

$$Duration\ Injection\ in\ sec. = t_{inj} = \frac{crank\ travel}{360\ x\ \frac{N}{60}}$$

$$t_{inj} = \frac{25}{360\ x\ \frac{2500}{60}}$$

$$t_{inj} = 1.667\ x\ 10^{-3}\ sec.$$

Given that the SG of fuel is,

$$SG = \frac{141.5}{131.5 + °API}$$

$$SG = \frac{141.5}{131.5 + 30}$$

$$SG = 0.8762$$

Thus density of fuel is,

$$SG = \frac{\rho_{fuel}}{\rho_{water}}$$

$$0.8762 = \frac{\rho_{fuel}}{1000}$$

$\rho_{fuel} = 876.2\ Kg/m^3$

Now the velocity of fuel injected is,

$$V_{inj} = C_d \sqrt{\frac{2\,(P_{inj} - P_{cyld})}{\rho_{fuel}}}$$

$$V_{inj} = 0.875\ x\ \sqrt{\frac{2\,(150\ x\ 10^5 - 40\ x\ 10^5)}{876.2}}$$

$$V_{inj} = 138.65\ m/sec.$$

Volume of fuel injected per cycle is,

$$Vol_{fuel\_cylce} = \frac{bsfc\ x\ BP}{\frac{N}{2}\ x\ \rho_{fuel}}$$

$$Vol_{fuel\_cylce} = \frac{(0.3/60)\ x\ 25}{\frac{2500}{2}\ x\ 876.2}$$

$$Vol_{fuel\_cylce} = 0.114\ x\ 10^{-6}\ \ m^3/cycle$$

Now calculating the nozzle orifice area is,

$$A_o = \frac{Vol_{fuel\_cylce}}{V_{inj}\ x\ t_{inj}}$$

$$A_o = \frac{0.114\ x\ 10^{-6}}{138.65\ x\ 1.667\ x\ 10^{-3}}$$

$$A_o = 0.4932\ x\ 10^{-6}\ m^2$$

Thus diameter of orifice is,

$$A_o = \frac{\pi}{4}\ (d_o)^2$$

$$0.4932\ x\ 10^{-6} = \frac{\pi}{4}\ (d_o)^2$$

$$\boldsymbol{d_o = 0.792\ x\ 10^{-3}\ m\ = 0.792\ mm\ \ \ ....Ans.}$$

7. Combustion in Compression Ignition (CI) Engines :

In the compression ignition (CI) engines, only air is compressed though a high compression ratio during compression stroke, raising its pressure and temperature to a high value. Fuel is injected by the injector into this highly compressed air in the combustion chamber. The fuel jet disintegrates into a core of fuel surrounded by a spray envelope of air and fuel particles, which is created by the atomization and vaporization of fuel.

The turbulence of air resulting in better mixing of fuel and air. Evaporation of liquid fuel droplets starts by absorbing the latent heat of vaporization from the surrounding air. As soon as this fuel vapor and air mixture within combustible range reach the level of auto-ignition temperature ignition starts. Thus in CI engine there must a certain delay period before the ignition starts.

The fuel air mixture is heterogeneous, this is because of the fact that the fuel droplets cannot be injected and distributed uniformly throughout the combustion space. In combustion chamber orderly and controlled movement is imparted to the air and the fuel so that a continuous flow of fresh air is given to each burning droplet, this air motion is termed as air swirl which is produced by the shape of the combustion chamber.

7.1. Stages of Combustion in CI Engines : SPPU : May -18, 6-Marks

The combustion of a CI engine is considered to be taking place in four stages,

Stage I : Ignition Delay – SPPU : May -17, 6-Marks

Ignition delay period is defined as the period of inactivity between the time when first droplet of fuel hits the hot compressed air in combustion chamber and the time it starts through the actual burning.

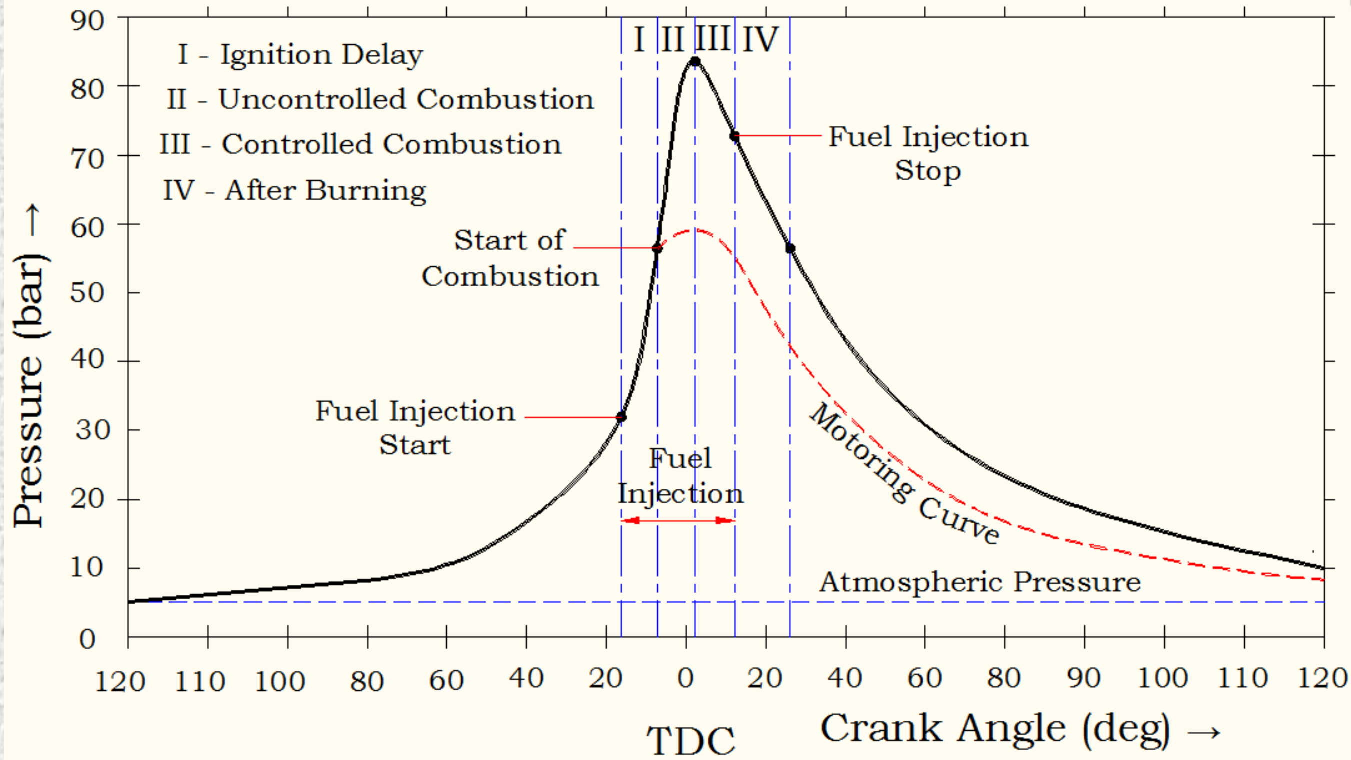
On pressure–crank angle diagram, it is shown between the fuel injection start to the start of combustion, this ignition delay is divided into two parts as physical delay and chemical delay.

***Physical Delay** – the delay period between the time of fuel injection starts to the attainment of chemical reaction conditions.*

In this, the fuel is atomized, vaporized and form a mixture with air and then temperature raised to its self ignition temperature.

***Chemical Delay** – the period where chemical reaction start slowly and then accelerate until ignition takes place.*

Chemical delay is larger than physical delay. In CI engine, the ignition delay is shorter than the duration of fuel injection.



Stage II : Uncontrolled Combustion / Rapid Combustion Period –

The period of rapid combustion / uncontrolled combustion is shown on pressure-crank angle diagram (indicator diagram), which starts from the start of combustion (ie end of ignition delay) and it ended at the peak pressure point.

During the delay period a considerably amount of fuel accumulates in the combustion chamber and is properly atomize and vaporized. When the actual ignition starts, the fuel burns at an extremely rapid rate. This phase is called the period of uncontrolled combustion.

The rate of heat release during this period is maximum. The longer the delay period the more rapid and higher is the pressure rise since more fuel would have accumulated in the combustion chamber.

Stage III : Controlled Combustion –

At the end of uncontrolled combustion, the temperature and pressure are so high that the fuel droplets injected during the uncontrolled combustion burn almost and now any further pressure rise can be controlled by the fuel injection rate.

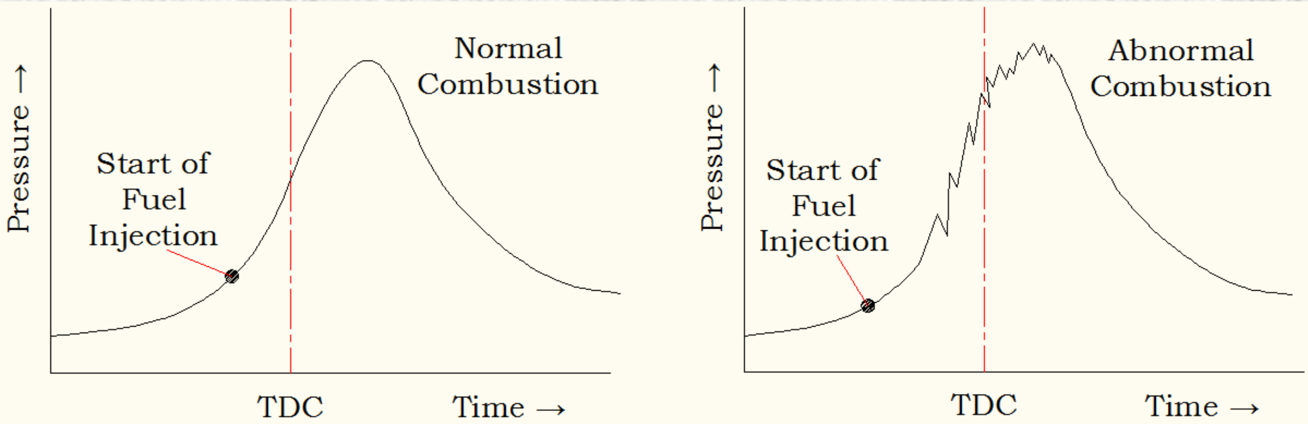
Controlled combustion starts from the end of uncontrolled combustion and ends at the point of maximum cycle temperature.

Stage IV : After Burning –

Combustion does not cease 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 period starts from the point of maximum cycle temperature and continues over a part of the expansion stroke.

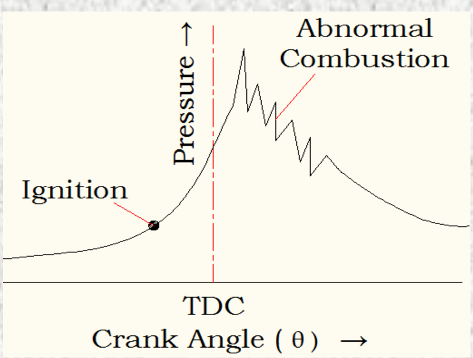
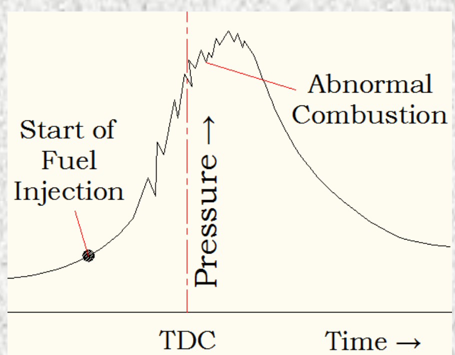
8. Abnormal Combustion / Knock in CI Engine : SPPU : May -17, 6-Marks

In CI engine the injection of fuel takes place for a certain interval of time. In shorter delay period, the first droplets of the fuel being injected will commence actual burning phase in a relatively short time after injection and small amount of the fuel will be accumulated in the chamber when the actual burning commences. The mass burning of the mixture will produce a smooth pressure rise and the combustion will be normal.



If ignition delay is longer the actual burning of the first droplets is delayed and greater quantities of fuel droplet are accumulated in the combustion chamber. When the actual burning starts, the ignition of the large amount of accumulated fuel causes a violent and instantaneous rise of pressure. Under such conditions extreme pressure differentials are produced and violent gas vibrations known as detonation or knock occurs.

8.1. Comparing Knock in SI and CI Engine : SPPU : May-17, May-14, 6-Marks

Description	SI Engines	CI Engines
Indicator Diagram		
Auto-Ignition Location	Detonation occurs due to end charge auto-ignition, away from the spark plug.	Detonation occurs due to first portion of mixture auto-ignited, near the spray enveloped.

Timing	Detonation occurs near the end of combustion.	Detonation occurs beginning of the combustion.
Intensity of Knocking	Intensity of knocking is high due to homogeneous mixture in SI engines.	Intensity of knocking is low due to heterogeneous mixture in CI engines.
Noise Level	Knocking noise is very high as compared to normal working noise of engine, hence distinguishing by human ear is possible.	Knocking noise is nearer to normal working noise of engine, hence distinguishing by human ear not possible
Factors Tending Knocking	Short Delay Period, Low Self Ignition Temp. (SIT) of fuel causing knocking.	High Delay Period, High Self Ignition Temp. (SIT) of fuel causing knocking.
Compression Ratio	High Compression Ratio make engine knock prone.	Low Compression Ratio make engine knock prone.
Inlet Pressure & Temperature	High inlet pressure & temperature causes engine to knock.	Low inlet pressure & temperature causes engine to knock.

8.2. Effect of Knocking :

Noise –

As intensity of detonation increases, the sound intensity increases and its is harmful to engine parts.

Mechanical Damage –

The high pressure shock waves may damage piston and increase rate of wear and erosion of piston.

Increase Heat Transfer –

The temperature of detonating engine cylinder is higher than non-detonating engine which causes increase heat transfer.

Decrease in Power Output and Efficiency –

The thermal efficiency and power output by detonating engine is less than non-detonating engine, due to abnormal combustion.

8.3. Methods of Knock Control :

To reduce the knock, the CI engine must design to reduce delay period.

1. The delay period can also be reduced by reducing the degree of turbulence as it will reduce heat loss. However, it will increase the combustion period and thus reduce torque and thermal efficiency.
2. The delay angle is reduced, cetane number is increased, by adding chemical dopes, called ignition accelerators. The chemical dopes increase the preflame reactions and reduce the flash point. These chemicals increase the cetane rating of diesel fuel by auto-ignition at lower temperatures.
3. There would be high rate of pressure rise and high maximum pressure in the uncontrolled combustion if large amount of fuel collects in the delay period. It can be reduced by arranging the injectors so that only a small amount of fuel is injected at first.

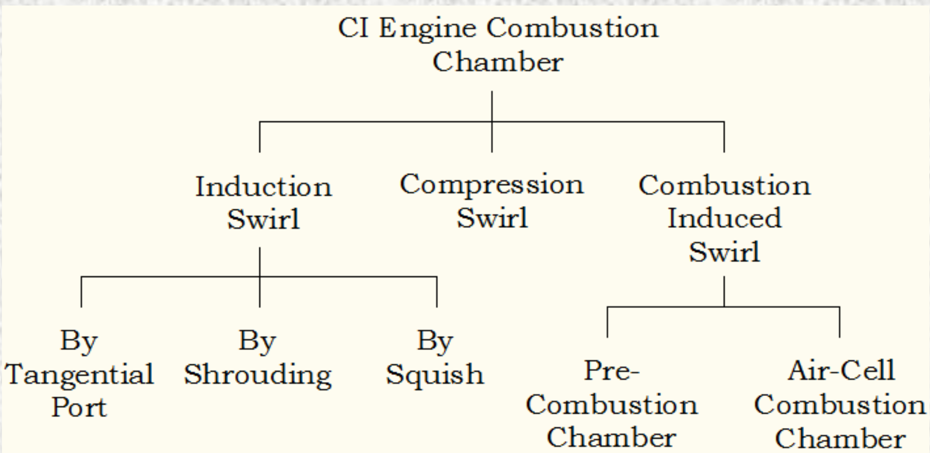
9. Types of Combustion Chambers in CI Engines : SPPU : May-16, Dec.-15 6-Marks

Objectives of Combustion Chamber Design in CI Engine - SPPU : May-14, 6-Marks

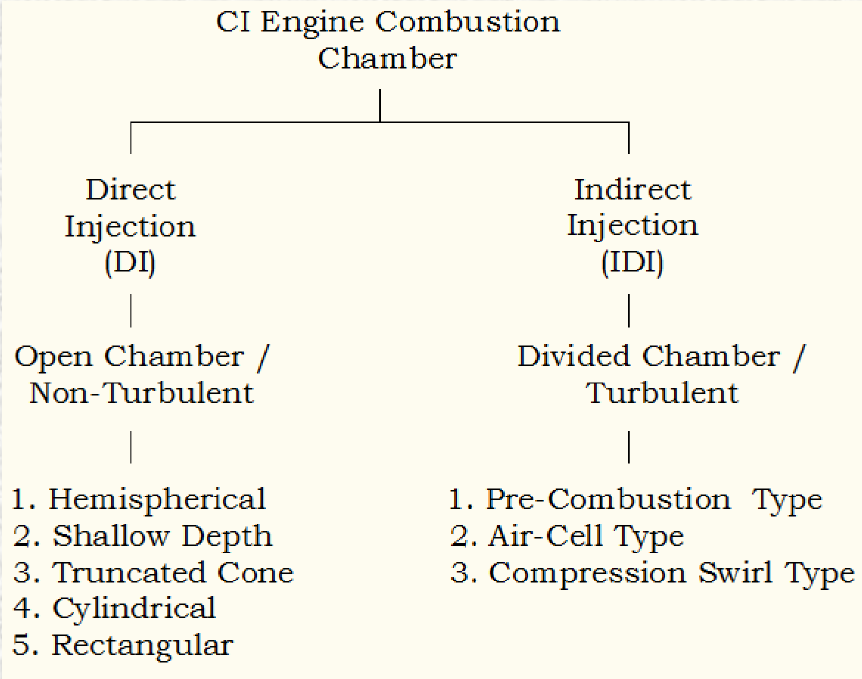
The mixing of fuel and air is of great importance in CI engine to obtain the required power, efficiency and short delay period for smooth operation. To fulfill these, following are the main objectives of combustion chamber design in CI engines.

1. Fuel to be atomized into fine droplet spray.
2. Forming proper air-fuel mixture .
3. There must be turbulence to enhance chemical reaction and mixing.
4. Complete combustion to obtain peak pressure.
5. Low surface to volume ratio to reduce heat losses and to achieve high thermal efficiency.
6. Short delay period for smooth, knock free operation of engine.

According to Air Swirl –



According to Injection Location / Turbulence –



9.1. Air Swirl :

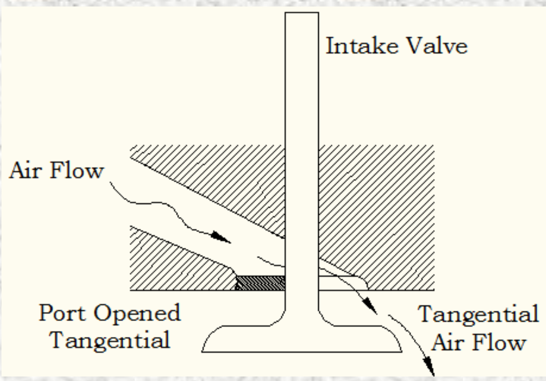
Air movement generated to form a better air-fuel mixture in CI engine is known as air swirl.

9.1.1. Induction Swirl –

In this method, the flow of air is directed in desired direction through a port tangential to the piston or by shrouding the intake valve head or by squish during the compression stroke. This method is used in case of open chamber / direct injection type combustion chambers.

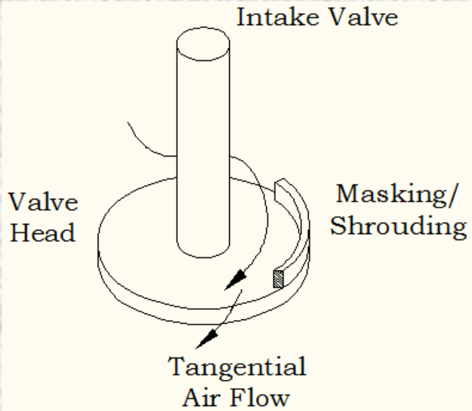
a) Tangential Port –

In this type the port opening to the combustion chamber is made tangential so that the air entering through this port moves tangential to the combustion chamber.



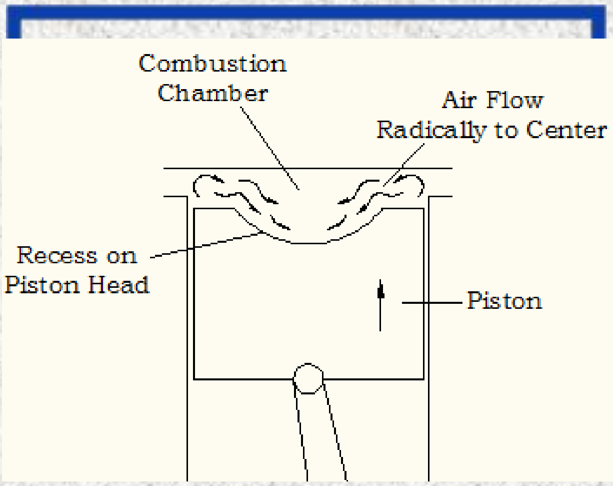
b) Shrouding Intake Valve –

In this type the intake valve is provided with masking / shrouding on its head so that air entered is directed to move around the periphery of masked part. The angle of mask commonly used form 90° to 140°.



c) Squish during Compression Stroke -

In this type, the flow of air from periphery to the centre of the cylinder into the combustion recess by squeezing it out from between the piston and cylinder head towards the end of compression stroke.



Advantages of Induction Swirl –

1. It allows high excess air to lower the combustion chamber temperature with low turbulence and low heat losses, these enhance the thermal efficiency of the engine to approach the ideal air cycle efficiency.
2. In open combustion chamber the intensity of swirl is low, thus heat loss to walls is less, resulting in easier cold starting.
3. Swirl is generated during induction, no additional work is supplied to the engine.

Disadvantages of Induction Swirl –

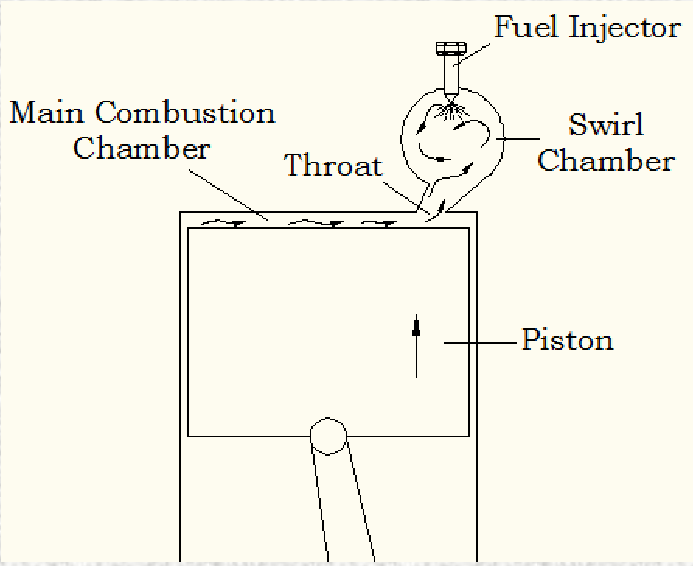
1. This swirl is weak in intensity, thus required multi-orifice nozzle with high injection pressure.
2. Small nozzle openings are frequently clogged and change fuel spray patter, thus needed regular cleaning which increases maintenance cost.

3. Use of shrouded valves lowers the volumetric efficiency.
4. Swirl is not proportional to speed and hence efficiency is not maintained over a wide range in a variable speed engine.

9.1.2. Compression Swirl - SPPU : May -18, 6-Marks

In this method swirl is generated into a divided chamber known as swirl chamber, this generates a very strong swirl. It has a spherical shaped swirl chamber connected to the main combustion chamber by throat and maximum possible air is transferred to it during compression stroke.

The fuel is injected into the swirl chamber and the ignition and bulk of the combustion takes place therein. Increase in the pressure in the swirl chamber causes a rush of hot gases and partially burned fuel outwards into the main combustion chamber through throat, resulting into excessive turbulence and better combustion.



Advantages of Compression Swirl –

1. Due to strong swirl a single orifice injector with low injection pressure is required.
2. Due to strong swirl there is a greater utilization of air. Therefore for a given power, the size of the engine is smaller as compared to open chamber induction swirl engine.
3. In this injector is located towards one side of the cylinder, hence there is freedom to use larger valves with a free entry.
4. The swirl generated is proportional to the speed, thus they are suitable for variable speed operation.
5. The swirl chamber produces smoother engine operation because the small swirl chamber absorbs the initial shock of peak pressure and saves the piston from extreme pressure variations.

Disadvantages of Compression Swirl –

1. The work done during compression is considerable and there is a corresponding loss during expansion also, which result in lowering mechanical efficiency.
2. Because of strong swirl, there is greater heat loss to the combustion chamber walls.
3. More energy is wasted in the exhaust gases which may lead to decreased exhaust valve life and other problems in exhaust manifold.
4. The cylinder construction is more expensive than in the case of open chambers.

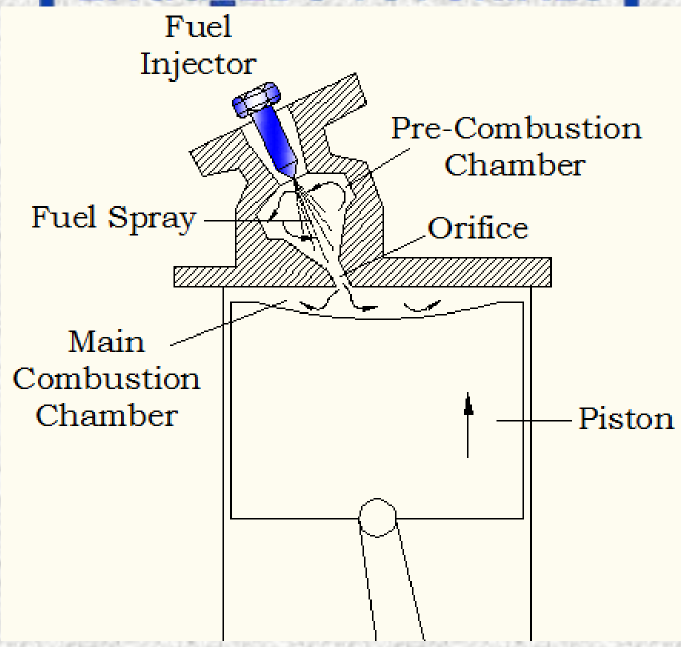
9.1.3. Combustion Induced Swirl -

In this method a small pressure rise is obtained by combustion of fuel and air in a separate cell and then it is forced through a small hole at high velocity to create swirl turbulence.

This method is employed in pre-combustion and air-cell combustion chamber designs.

a) Pre-Combustion Chamber Type -

It consists of pre-combustion chamber connected to the main chamber through a number of very small holes. Pre-combustion chamber is of 20-30% of the total clearance volume. The fuel is injected into pre-combustion chamber in such a manner that bulk of it reaches the orifice separating the two chambers.



The combustion is initiated in the pre-combustion chamber and the resulting pressure rise forces the flaming droplets together with some air and their combustion products to rush out at high velocity through the small holes, thus both creating strong secondary turbulence and distributing them throughout the air in the main combustion chamber where bulk of combustion takes place and about 80% energy is released in main combustion chamber.

Advantages of Pre-Combustion Chamber Type –

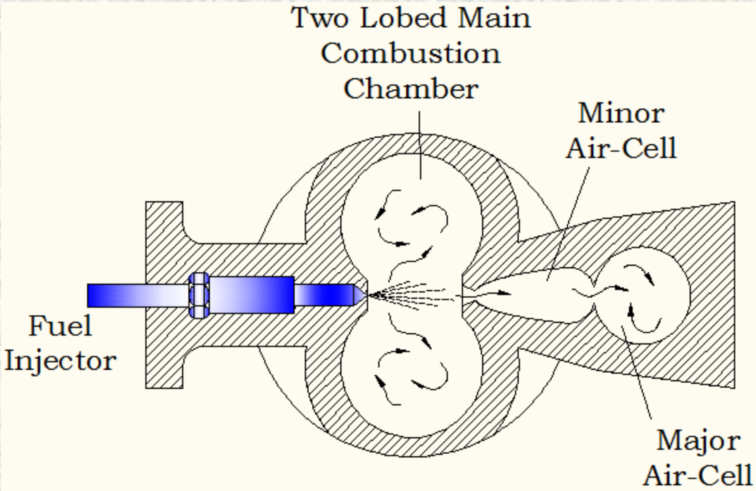
1. It has multi-fuel capability without any modification, because the temperature of pre-combustion chamber is nearly constant at 2000°C and the difference between optimum injection timing between petrol and diesel fuel is only 2°.
2. It uses single hole pintle type of nozzle. .
3. Low ignition quality fuels are required.
4. The initial shock of combustion is limited to pre-combustion chamber only.

Disadvantages of Pre-Combustion Chamber Type –

1. It has high heat losses due to high velocity and smaller size of valve due to central location of pre-combustion chamber.
2. Thermal efficiency is low.
3. Cold starting is difficult.

b) Air-Cell Chamber Type

In this clearance volume is divided into two parts, one in the main cylinder and the other called the energy cell. The energy cell is divided into two parts, major and minor which are separated from each other and from the main chamber by narrow orifices.



A pintle type of nozzle injects the fuel across the main combustion chamber. During the compression the pressure in the main chamber is higher than that inside the energy cell due to restricted passage area between the two. At TDC the difference in pressure will be high and air will be forced at high velocity through the opening into the energy cell and this moment the fuel injection also begins. The combustion starts initially in the main chamber where the temperature is comparatively higher but the rate of burning is very slow due to absence of any air motion.

In energy cell the fuel is well mixed with air and high pressure is developed due to heat release and the hot burning gases blow out through the small passage into the main combustion chamber. This high velocity jet produces swirling motion in the main combustion chamber and thereby thoroughly mixes the fuel with air resulting in complete combustion.

Advantages of Air-Cell Chamber Type –

1. It required less injection pressure.
2. The direction of spraying is not very important.
3. Smooth running of engine.
4. This can be used for high speed engines.

Disadvantages of Air-Cell Chamber Type –

1. It has poor cold starting performance requiring heater plugs.
2. Specific fuel consumption is high because there is a loss of pressure due to air motion through the duct and heat loss due to large heat transfer area.
3. It has low thermal efficiency.

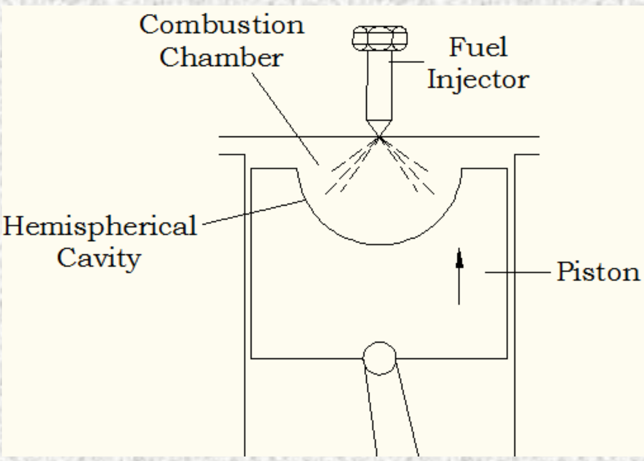
10. Open Chamber / Direct Injection / Non-Turbulent Combustion Chambers

SPPU : May-16, Dec.-15 6-Marks

An open type 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 may designs as follows,

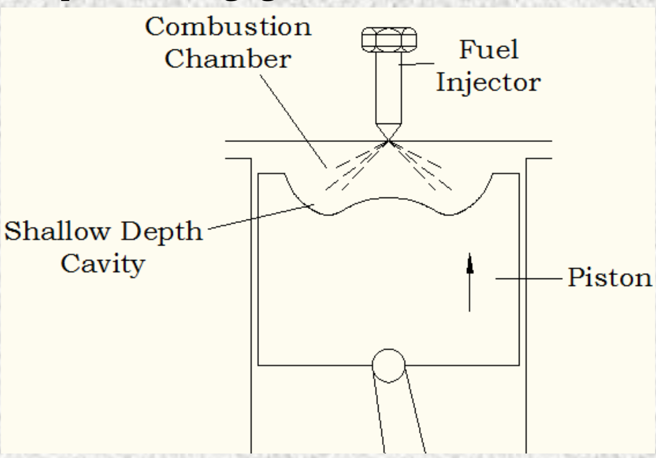
10.1. Hemispherical Type Chamber

This chamber has a centrally hemispherical cavity as shown, and it gives small squish only.



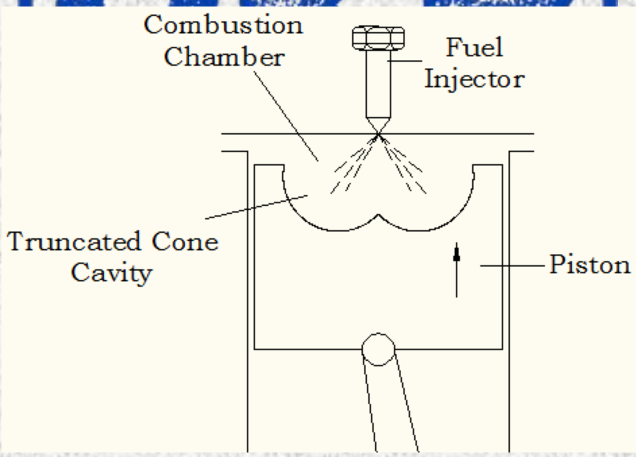
10.2. Shallow Depth Type Chamber

In this, the depth of the cavity provided in the piston is quite small. This chambers are usually adopted or the large engines running at low speeds. Since the cavity diameter is very large the squish is negligible.



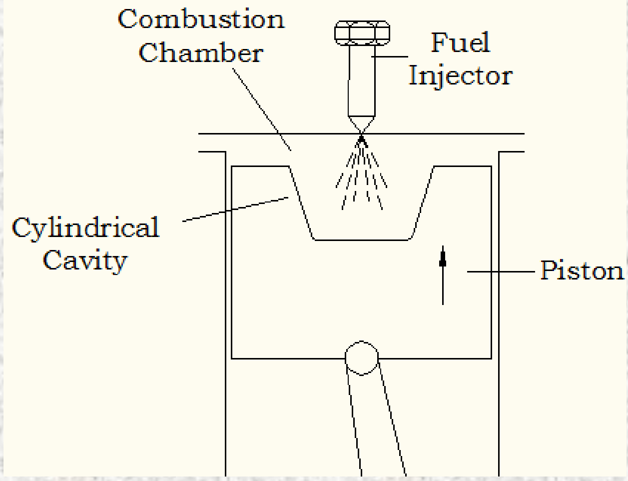
10.3. Toroidal / Truncated Cone Type Chamber

In this, air swirl is obtained by shrouding the inlet valves and produces powerful squish with air movement. The cone angle of spray for this type of chamber is 150° to 160°.



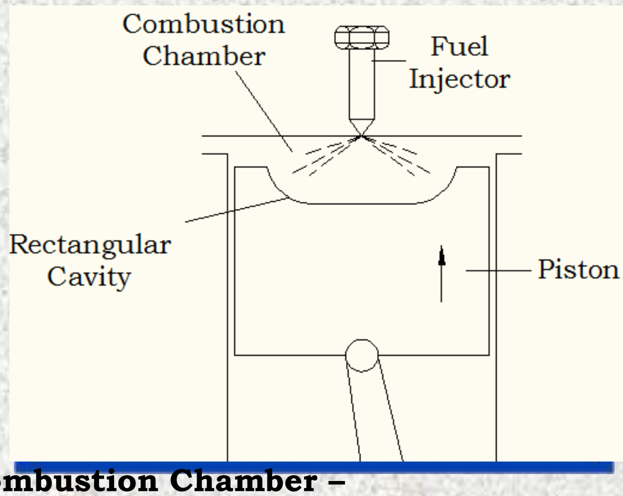
10.4. Cylindrical Type Chamber

It provided better squish, the desired level of squish can be obtained by varying depth to diameter ratio of the cavity. Swirl is produced by shrouding inlet valve.



10.5. Rectangular Type Chamber

It have a rectangular shape cavity, with larger diameter and comparatively small depth. This chambers have lower squish.



Advantages of Open Combustion Chamber –

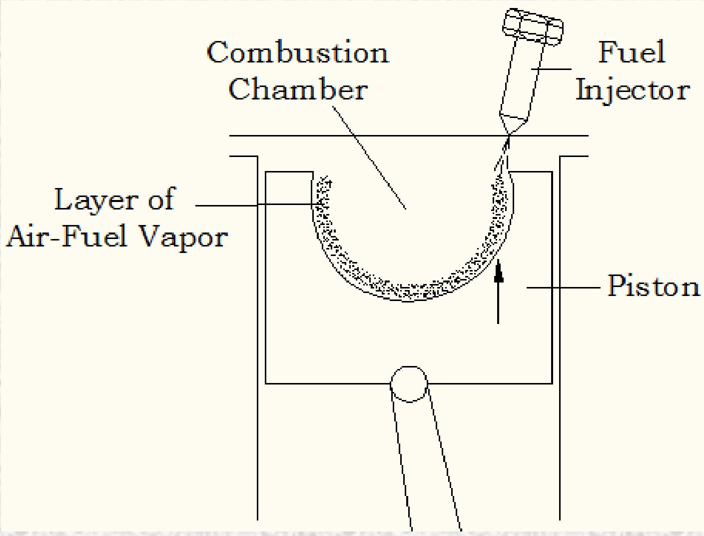
1. Heat loss to cylinder wall is small due to less turbulence in combustion chamber, it give easy starting.
2. It is simple in construction.
3. Thermal efficiency is high.
4. Less costly fuel with longer delay period can be used.

Disadvantages of Open Combustion Chamber –

1. Supply of excess air required for better utilization of fuel but this results into lower maximum temperature, thus power output suffers.
2. High injection pressure are needed.
3. Higher maintenance cost, as small opening for injecting fuel given frequent clogging of orifice.

10.6. MAN Open Combustion Chambers

The MAN combustion chamber is a special type of open combustion chamber, which was developed in West Germany. This is of spherical shape situated in the piston crown. Due to the presence of strong swirling action of air, a thin liquid film of fuel is formed over the combustion space, this eliminates the formation of fine mist.



The principles employed in the design of this chamber are,

- To prevent the initial formation of heterogeneous fuel-air mixture so that the reaction rate is normal.
- To prevent access of oxygen to the fuel to eliminate formation of peroxide.
- To assist early formation of slowly reacting hydrocarbon compounds.

For this type of chamber, it is claimed that combustion takes place in a homogeneous gas phase, thermal decomposition is prevented and there is no soot formation. The pressure rise is at low rate, thus ease of starting.

Advantages of MAN Open Combustion Chamber –

1. Smooth running of engine with high thermal efficiency.
2. Can use poor ignition quality fuels.
3. Weight to power ratio is low.
4. High volumetric efficiency.
5. Low smoke levels in exhaust.
6. Easy cold starting.

Disadvantages of MAN Open Combustion Chamber –

1. Engine at no load and low load gives diesel odour in exhaust.
2. At no load and low load emission is with smoke and high level of hydrocarbons.

11. Rating of Fuels in CI Engines :

Fuel used in CI engine, i.e. diesel is rated by the Cetane Number. The fuel cetane ($C_{16}H_{34}$) is has good ignition qualities and it is assigned a rating of 100 cetane number.

Whereas, alpha-methyl-naphthalene ($C_{10}H_7CH_3$) has poor ignition quality and it is assigned zero cetane number.

Cetane Number is defined as, the percentage by volume of cetane in a mixture of cetane 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 the tendency to diesel knock.

11.1. Diesel Index :

An alternative method of rating quality of diesel is called Diesel Index,

$$Diesel\ Index = \frac{Aniline\ Point\ (°F) x\ API\ gravity\ at\ 15°C}{100}$$

Where, ...

Aniline Point : *it represents the lowest temperature at which the diesel fuel is completely miscible with an equal volume of aniline.*

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

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

11.2. Dopes & Additives :

Dopes and additives are used to raise the cetane number of the fuels in the range of 50 – 60 for high speed engines.

Such additives serve to reduce the self ignition temperature of fuel by acting as a local ignition point.

Sr. No.	Dope / Additives	Increase in Cetane Number on adding of 5% by weight
1	Amyl Nitrate	13.5
2	Amyl Nitrite	9
3	Ethyl Thionitite	10

Exercise

1. What are the requirement of a fuel injection system in CI engines ?

2. Classify the diesel injection system and explain any one in brief.
3. Differentiate between air injection and airless injection system used in CI engine
4. Draw the neat sketch of different types of fuel injector nozzle nozzle used in CI engine.
5. Explain the working principle of jerk fuel injection system.
6. Explain the working of Bosch fuel injector pump with neat sketch.
7. Explain the working of distributor type fuel injection pump with neat sketch.
8. Explain with neat sketch the common rail fuel injection system.
9. Draw a neat diagram of pressure – crank diagram.
10. Factor which leads to increased possibility of detonation in SI engine tend to reduce knocking in CI engine. – discus.
11. Explain the concept of ignition delay period in CI engine.
12. What are the different stages of combustion in CI engines ?
13. What are the significance of controlling delay period in CI engines ?
14. What are the effect of following engine variables on diesel knock ?
15. Discuss the phenomenon of normal and abnormal combustion in CI engines.
16. Compare between the SI engine and CI engine about abnormal combustion.
17. Enlist the requirement of good combustion chamber for CI engines.
18. Write note on turbulence in CI engine combustion chamber.
19. Explain with neat sketch the induction swirl combustion chamber.
20. Enlist the combustion chambers used in CI engines.
21. Explain the air-cell chamber and list its advantages and disadvantages.
22. Explain the divided combustion chamber using compression swirl.
23. Explain with neat sketch any four direct injection combustion chambers used in CI engines.
24. Give the classification of combustion chamber used in CI engines.
25. Write short note on MAN combustion chamber.
26. Define cetane number of a fuel used in CI engines.

27. What do you understand by diesel index.

28. What are additives ? List out the various additives used for CI engine fuel.

29. A four cylinder, four stroke diesel engine develops a power of 180 KW at 1500 rpm. The BSFC is 0.2 kg/KW hr. At the beginning of injection pressure is 30bar and the maximum cylinder pressure is 50 bar. The injection is expected to be at 200 bar and maximum pressure at the injector is set to be about 500 bar. Assuming the followings,

    C_d for injector = 0.7
    SG of fuel = 0.875
    Atmospheric pressure = 1 bar
    Effective pressure difference = Avg. pressure difference over injection period

    Determine the total orifice area required per injector if the injection takes place over 15° crank angle. (Ans. $m_{fuel\_cycle} = 2\,x\,10^{-4}\,Kg,\ \ t_{inj} = 1.667\,x\,10^{-3}\,sec.\ \ V_{inj} = 186.33\ m/sec.\ \ Vol_{fuel\_cycle} = 0.2286\,x\,10^{-6}\ m^3/cycle,\ \ \boldsymbol{A_o = 0.736\,x\,10^{-6}\,m^2}$.

Unit – IV : Testing of IC Engines

Syllabus :

Objective of Testing, Various performance parameters for IC engine – Indicated Power, Brake Power, Friction Power, SFC, AF ratio etc. Methods to determine various performance parameters, characteristic curves, heat balance sheet.
Supercharging – Supercharging and Turbo-charging methods and their limitations.

1. Testing of IC Engines :

Testing of an engine is used to measure the various parameter intended to quantify the engine performance level in terms of power output, efficiency, fuel consumption, emission etc.

There are following objectives of the engine tastings,

Testing of an engine is carried out to get the correct information regarding the performance of the engine, which data then used to judge whether an engine working under-performance or not.

Testing of an engine also have an objective to justify the rating of an engine and the guaranteed specific fuel consumption and power output.

It also used for validation of the engine performance with the design specifications.

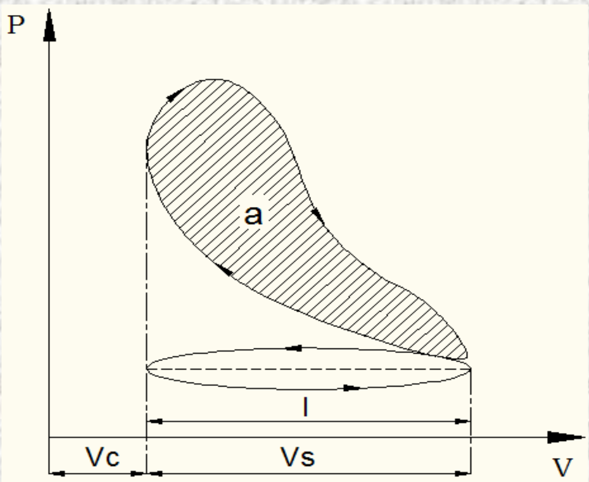
2. Performance Parameter :

Engine performance is an indication of the degree of success with which it is doing its assigned job. This degree of success is compared on the basis of following parameters,

2.1. Indicated Mean Effective Pressure (P_{ime}) –

It is define as, the constant pressure which if acted over the full length of the stroke would produce the same amount of work done by the piston as it is actually produced by the engine.

It is calculated from the indicator diagram as follow,



Prof. Rahul Krishnaji Bawane
Pimpri Chinchwad College of Engineering & Research, Ravet, Pune.

Let ,

a = Net area of indicator diagram in cm^3
l = Length of indicator diagram in cm
K = Spring Constant in $N/cm^2/cm$

$$P_{ime} = Mean\ Height\ of\ Daigram\ x\ Sproing\ Constant$$

$$\boldsymbol{P_{ime} = \frac{a}{l}\ x\ K \quad in\ N/cm^2}$$

2.2. Indicated Power (IP) –

It is define as, the power developed within the engine cylinder, and it is denoted by IP.

Indicated Power (IP) of engine at a particular speed can be calculated with the help of indicator (P-v) diagram.

Let ,

P_{ime} = Indicated Mean Effective Pressure in N/cm^2
A = Cross sectional area of piston in cm^2
d = Diameter of piston in cm
L = Length of stroke in cm
N = Speed of engine in rpm
n = Number of power stroke per minute
= N *for 2-stroke engine*
= $N/2$ *for 4-stroke engine*

$$IP = Workdone\ per\ cycle\ x\ No.of\ power\ stroke\ per\ min.$$

$$IP = (Force\ on\ piston\ x\ Stroke\ length\ )\ x\ No.of\ power\ stroke\ per\ min.$$

$$IP = (P_m x\ A\ x\ L\ )\ x\ n$$

$$IP = P_m\ A\ \ L\ \ n \quad \text{in Nm/min.}$$

$$IP = \frac{P_m\ L\ A\ \ n}{60} \quad in\ Nm/\sec\ \ (watt)$$

$$\boldsymbol{Indicated\ Power\ (IP) = \frac{P_m\ L\ A\ \ n}{60000} \quad in\ KW\ per\ cylinder}$$

2.3. Brake Power (BP) –

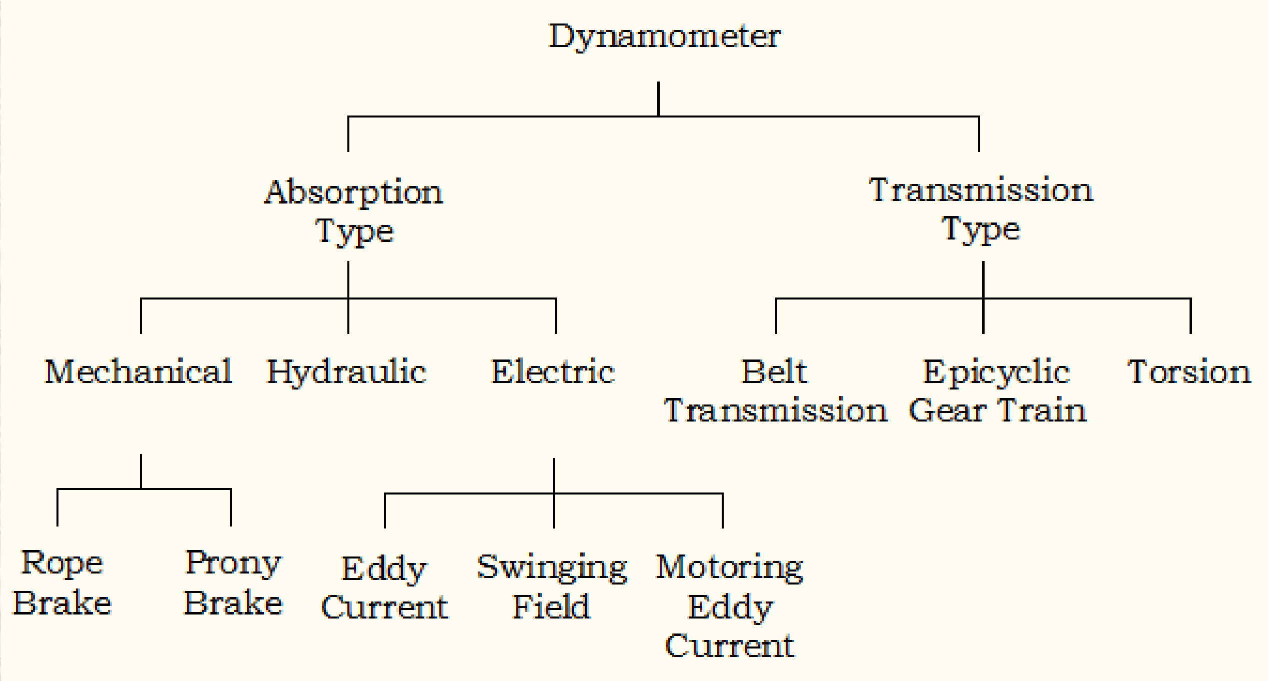
The Brake Power is define as the useful power available at the output shaft (crank shaft) of an engine, and it is denoted by BP.

The brake power (BP) is always less than indicated power (IP) because of the following power losses,

- Pumping loss due to induction and exhaust.
- Mechanical losses / friction power losses in the moving parts.
- Resistance of air to flywheel rotation.

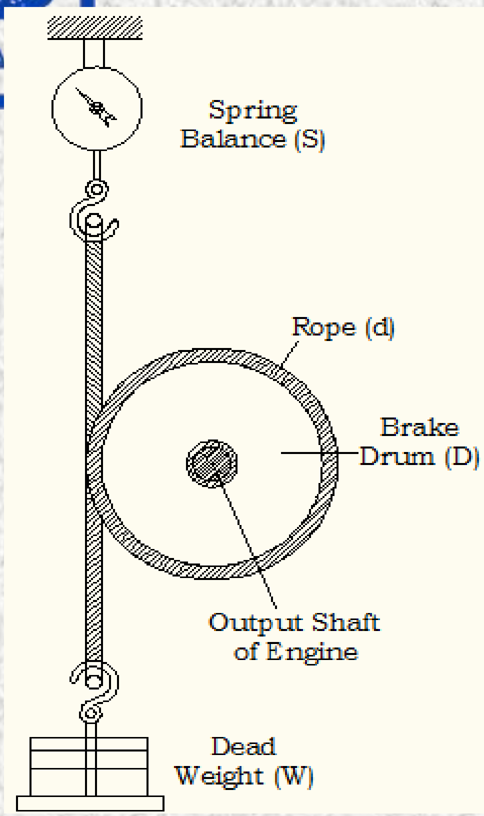
- Power losses to drive the auxiliaries like fuel pump, lubrication oil pump, water pump, governor, dynamo etc.

Brake Power can be calculated with the help of following methods,



2.3.1. Rope Brake Dynamometer –

The rope brake dynamometer consists of a number of turns of rope wound around the rotating drum attached to the output shaft. One side of the rope is connected to a spring balance and other to a loading device.



The power absorbed is due to friction between the rope and the drum.

Let , W = Dead weight in N

S = Spring balance reading in N

N = Engine speed in rpm

D = Diameter of brake drum in m

d = Diameter of rope in m

R = Effective radius of brake drum in m

$= \dfrac{D+d}{2}$

N = Engine speed in rpm

$$Brake\ Load\ (Net\ Load) = (W - S)$$

$$Brake\ Torque\ (T) = (W - S)x\ R$$

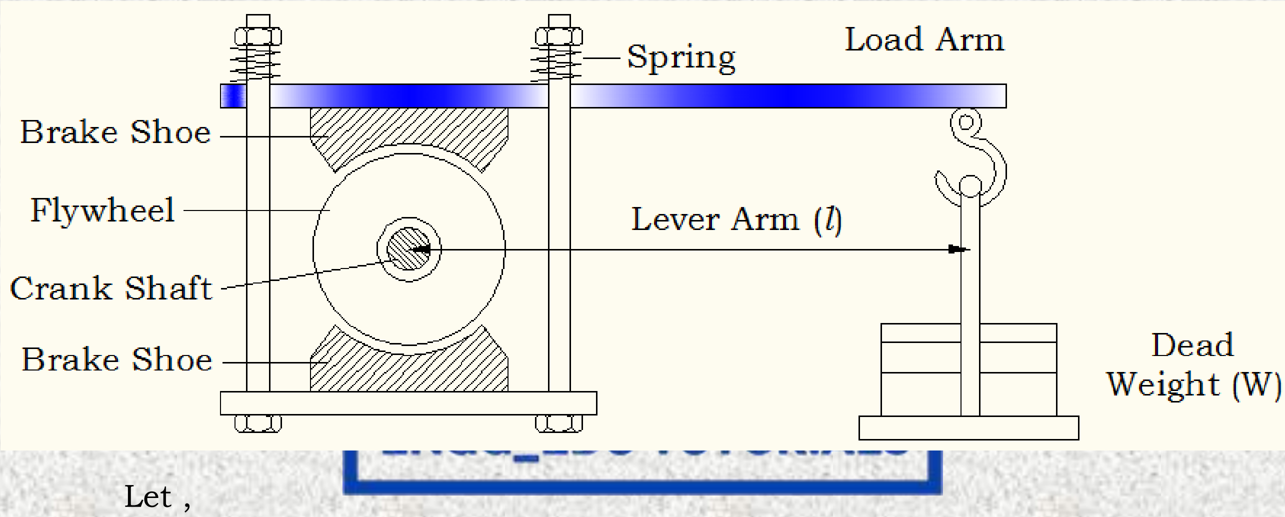
$$Brake\ Power\ (BP) = (W - S)R\ x\ \frac{(2\ \pi\ N)}{60}\ \ in\ watt$$

$$Brake\ Power\ (BP) = T\ x\ \frac{(2\ \pi\ N)}{60}\ \ in\ watt$$

$$\boldsymbol{Brake\ Power\ (BP) = \frac{2\ \pi\ N\ T}{60000}\ \ in\ KW}$$

2.3.2. Prony Brake Dynamometer –

The prony brake consists of a frame with two shoes gripping the flywheel. The pressure of the brake shoes on the flywheel can be varied by the spring loaded using nuts on the top of the frame. The wooden block when pressed into contact with the rotating drum opposes the engine torque and the power is dissipated in overcoming frictional resistance. The power absorbed is converted into heat and hence it required cooling.



Let ,

W = Dead weight in N
l = Length of lever arm in m
N = Engine speed in rpm

$$Brake\ Torque\ (T) = W\ x\ l\ \ in\ Nm$$

$$Brake\ Power\ (BP) = (W\ x\ l)\ x\ \frac{(2\ \pi\ N)}{60}\ \ in\ watt$$

$$Brake\ Power\ (BP) = T\ x\ \frac{(2\ \pi\ N)}{60}\ \ in\ watt$$

$$\boldsymbol{Brake\ Power\ (BP) = \frac{2\ \pi\ N\ T}{60000}\ \ in\ KW}$$

2.3.3. Eddy Current Dynamometer –

It consists of a stator on which a number of electromagnets are fitted and a rotor disc made of copper or steel which is coupled to engine output shaft.

When rotor rotates eddy currents are produced in the stator due to magnetic flux setup by the passage of field current in the electromagnets. These eddy currents oppose the rotor motion, thus loading the engine. The torque is measured with the help of a moment arm and the load. This dynamometer also required cooling as eddy currents are dissipated in producing heat.

Advantages of Eddy Current Dynamometer are,

1. It can be used to measure high power output at all speed, thus suitable for automobile and aircraft engines.
2. Overall size is less as compared to other dynamometer.
3. Eddy current development is smooth, hence the torque developed is smooth and continuous under all operating conditions.
4. These can be made in all sizes for measurement of power.

2.3.4. Swinging Field Dynamometer –

It consists a DC shunt motor, the field coils tends to rotate due to the magnetic drag. This dynamometer can be switch over from motor mode to generator mode.

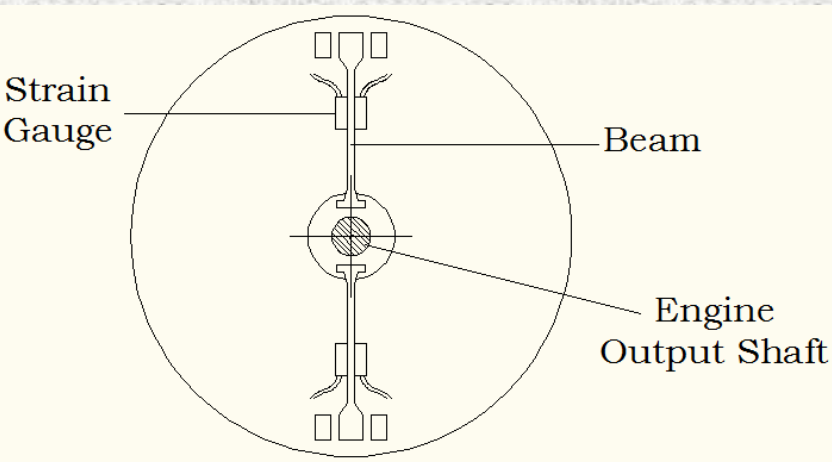
When used as motoring device an external source of DC voltage is needed to drive the motor. And when used as power absorbing device (DC generator) it converts mechanical energy into electric energy which is dissipated in an external resistor. The load is controlled by changing the field current.

This type of dynamometer used to measure brake power of high speed engines.

2.3.5. Transmission (Mechanical Type) Dynamometer –

Transmission dynamometer is also called Torque-meter, it consist of a set of strain gauges fixed on the rotating shaft and the torque is measured by the angular deformation of the shaft which is indicated as strain of the strain gauge.

Usually a four arm bridge is used to reduce the effect of temperature to minimum and the gauges are arranged in pairs such that the effect of axial or transverse load on the strain gauges is avoided.

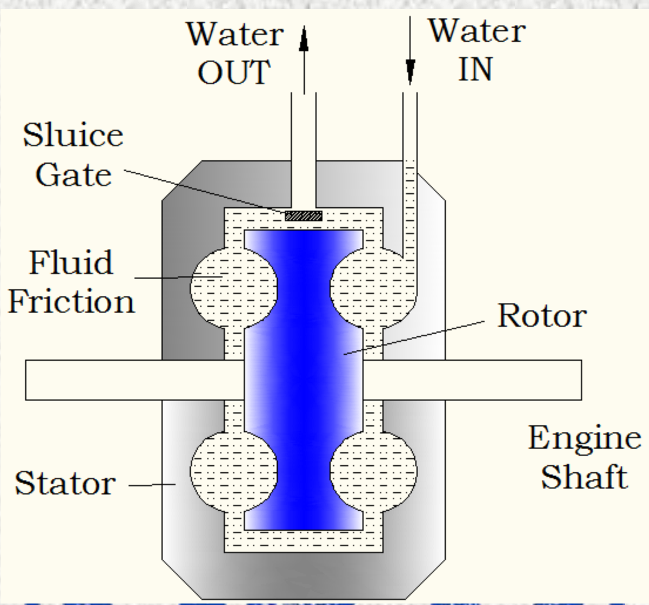


These are very accurate and used where continuous transmission of load is necessary. These are used mainly in automatic units.

2.3.6. Hydraulic Dynamometer –

It works on the principle of dissipating the power in fluid friction. It consists of an inner rotating member or impeller coupled to the output shaft of the engine. This impeller rotates in a casing filled with some hydraulic fluid.

The outer casing tends to revolve with the impeller due to centrifugal force developed, but it is resisted by a torque arm supporting the balance weight.



The output can be controlled by regulating the sluice gate which can be moved in and out to partially or wholly obstruct the flow of water between impeller and the casing.

2.4. Friction Power (FP) –

The friction power (FP) is the engine internal power losses, mainly due to pumping losses and friction losses. Pumping Losses – during the inlet and exhaust stroke piston moves against the gaseous pressure opposite to its movement, thus to overcome pressure resistance some power is losses called as pumping losses.

Friction Losses – during the motion between two sliding or rotating parts some power is lost to overcome friction between them, also to drive engine accessories like water pump, ignition unit etc. engine power is used, all these power losses are called as friction losses.

Thus the power available at output shaft of engine is always less than that of generated and the difference is equal to the friction power.

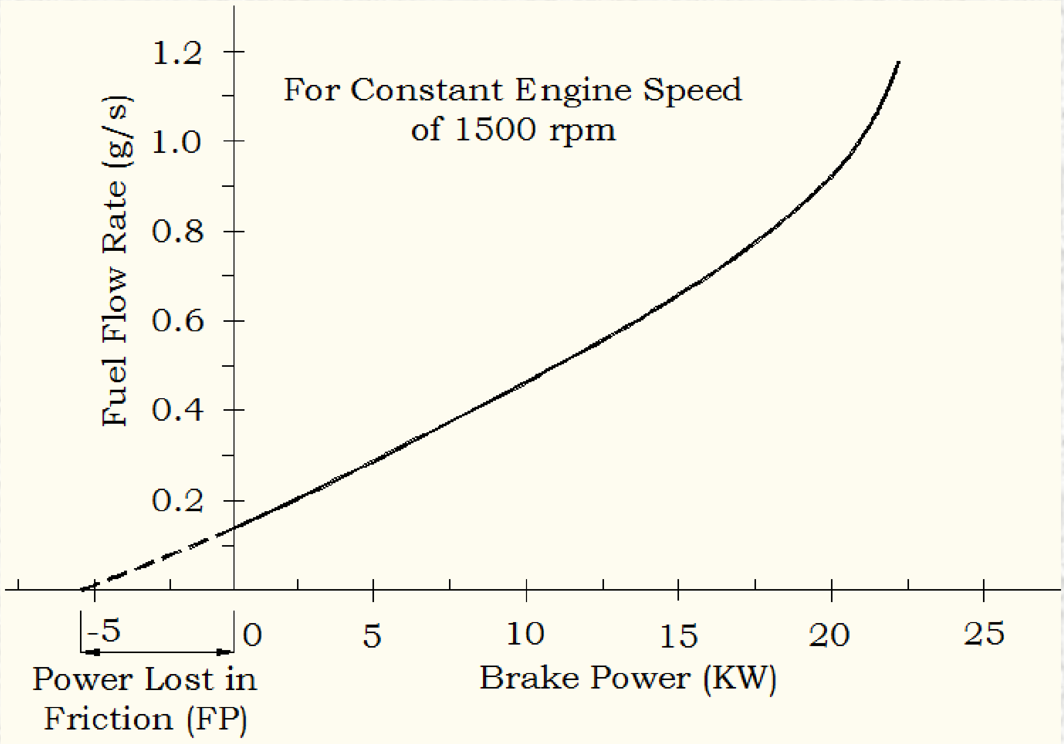
$$Friction\ Power = Indicated\ Power - Brake\ Power$$

$$\boldsymbol{FP = IP - BP}$$

The following methods are used to determine the friction power (FP).

2.4.1. Willan's Line Method – SPPU : Dec.-18, 7-Marks

In this method, a graph is prepared between fuel consumption on Y-axis and brake power on X-axis at constant speed. And fuel consumption Vs. brake power curve is extrapolated on the negative axis of the brake power. The intercept of the negative axis is taken as friction power of the engine at that speed. The extrapolated part is shown by dotted line.



From the graph, at BP = 0, the engine consumes some fuel, this means that the energy supplied by fuel is spent in overcoming friction.

Drawbacks of this methods,

- This method holds good mainly for CI engines
- The long distance is to be extrapolated from data obtained between 5 – 40% load towards the zero line of fuel input.
- This method gives approximate friction power due to the errors involved in extrapolation.

2.4.2. Morse Test Method –

This method can be used only on multi-cylinder engines. In this each cylinder is cut off in turn and measures the brake power. SI engine cylinders cut off by shorting respective spark plug and CI engine cylinders are cut off by cutting off the fuel supply to respective injector.

Consider a four cylinder engine coupled to a dynamometer. Through out the test engine is run at constant speed (i.e. N rpm), and it is assumed that friction losses are the same whether the cylinder is working or motoring.

Let , measurement of brake power,

BP = Brake Power when all cylinder working in KW

$BP1$ = Brake Power when cylinder 1 is cut off (i.e. cyld. 2,3 & 4 working)

$BP2$ = Brake Power when cylinder 2 is cut off (i.e. cyld. 1,3 & 4 working)

$BP3$ = Brake Power when cylinder 3 is cut off (i.e. cyld. 1,2 & 4 working)

$BP4$ = Brake Power when cylinder 4 is cut off (i.e. cyld. 1,2 & 3 working)

Now the measurement of indicated power,

$IP1$ = Indicated Power when cylinder 1 is cut off (i.e. cyld. 2,3 & 4 working)

$$IP1 = BP - BP1$$

$IP2$ = Indicated Power when cylinder 2 is cut off (i.e. cyld. 1,3 & 4 working)

$$IP2 = BP - BP2$$

$IP3$ = Indicated Power when cylinder 3 is cut off (i.e. cyld. 1,2 & 4 working)

$$IP3 = BP - BP3$$

$IP4$ = Indicated Power when cylinder 4 is cut off (i.e. cyld. 1,2 & 3 working)

$$IP4 = BP - BP4$$

Thus, total indicated power of the engine is,

$$\boldsymbol{IP = IP1 + IP2 + IP3 + IP4}$$

Calculation for Friction Power is,

Total Friction Power = Total Indicated Power – Total Brake Power

$$\boldsymbol{FP = IP - BP}$$

2.4.3. Motoring Test Method –

In this method, engine is coupled with swinging field type dynamometer. The test engine is operated at the rated speed by its own power and allow to reach a steady state condition. During this power generated is absorbed by the swing field dynamometer.

When engine reach to steady state condition, the ignition is then cut off and dynamometer is converted to run as motor to crank the engine at the same speed which it was previously operating. The power supplied by the dynamometer to the engine is measured which is equal to the friction power of engine at that speed.

Following factors affect the real data in this method,

- There is always a engine temperature difference when run by its own power and when it is motor by dynamometer.
- Due lower temperature during motoring, the clearance between the piston rings and wall is more which reduces the friction.
- Piston subjected to much higher pressure in firing engine as compared to motored engine.
- There is a difference in back pressure of fired engine and motored engine.

2.5. Mechanical Efficiency (η_{mech}) –

It is define as *the ratio of the brake power to the indicated power.* It is denoted as η_{mech}

$$\eta_{mech} = \frac{BP}{IP}$$

2.6. Thermal Efficiency (η_{th}) –

Thermal efficiency of engine is the ratio of the power produced to the heat supplied due to combustion of fuel.

Indicated Thermal Efficiency (η_{ith}), *it the ratio of indicated power (IP) to the heat supplied.*

$$\eta_{ith} = \frac{IP}{m_f \ x \ CV}$$

Brake Thermal Efficiency (η_{bth}), *it the ratio of brake power (BP) to the heat supplied.*

$$\eta_{bth} = \frac{BP}{m_f \ x \ CV}$$

Where,

m_f = mass of fuel supplied

CV = Calorific Value of fuel

2.7. Specific Output –

The specific output of the engine is defined as the brake power per unit of piston displacement.

$$Specific\ Output = \frac{BP}{A\ x\ L}$$

Where,

A = Area of piston

L = Stroke of piston

2.8. Relative Efficiency (η_{rel})–

Relative efficiency is defined as the ratio of the brake thermal efficiency to the air-standard efficiency.

$$\eta_{rel} = \frac{\eta_{bth}}{\eta_{ait-std}}$$

Where, $\eta_{ait-std}$ SI Engine = $1 - \frac{1}{(r)^{\gamma-1}}$

$\eta_{ait-std}$ CI Engine = $1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{{r_c}^{\gamma}-1}{\gamma\,(r_c-1)}\right]$

r = Compression Ratio

r_c = Cut off Ratio

γ = 1.4

2.9. Volumetric Efficiency (η_{vol}) –

Volumetric efficiency (η_{vol}) is defined as the ratio of actual volume of charge inducted during suction stroke to the swept volume.

It also defined as, *the actual mass of charge inducted during suction stroke to the mass of charge corresponding to swept volume of the engine at atmospheric pressure and temperature.*

$$\eta_{vol} = \frac{m_{air-act}}{m_{air-th}}$$

Where,

$m_{air-act}$ = from air consumption measuring device

m_{air-th} = for two stroke engine = $\rho_{air} V_{swept} N$

= for four stroke engine = $\rho_{air} V_{swept} \frac{N}{2}$

2.10. Specific Fuel Consumption (SFC)–

Specific Fuel Consumption (SFC) is defined as the amount of fuel required to be supplied to an engine to develop 1KW power per hour.

Brake Specific Fuel Consumption (BSFC) is the amount of fuel required to be supplied to an engine to develop 1KW brake power per hour.

$$BSFC = \frac{m_f}{BP} \, x \, 3600 \quad in \, Kg \, / \, KW \, h$$

Indicated Specific Fuel Consumption (ISFC) is the amount of fuel required to be supplied to an engine to develop 1KW brake power per hour.

$$ISFC = \frac{m_f}{IP} \, x \, 3600 \quad in \, Kg \, / \, KW \, h$$

Where,

m_f = fuel consumed in Kg

3. Methods of improving engine performance –

The engine performance can be improved by increasing the energy input to the engine or by improving the conversion efficiency of the engine.

The following methods can be used to improve the engine performance,

1. By increasing mass flow rate of mixture
2. By using fuel of higher calorific value
3. By using fuel with higher octane rating in SI engine and cetane rating in CI engine
4. By supercharging the engine
5. By increasing engine speed
6. By improving engine volumetric efficiency by reducing pressure losses in intake manifolds and reducing the mixture flow restrictions
7. By using higher compression ratios
8. By using fuel additives, exhaust gas recirculation, positive crankcase ventilation etc.
9. By reducing heat losses.

4. Heat Balance Sheet –

Heat energy is supplied to the engine by the combustion of fuel. Only a part of this heat energy is converted into useful work at the engine crankshaft and the remainder is lost.

The accounting of the energy supplied, the energy losses and the useful energy output is expressed in the tabulated sheet known as heat balance sheet.

To draw a heat balance sheet for an internal combustion engine a complete test must be made on the engine while running at constant speed. The various terms used in heat balance sheet are as follows,

Heat supplied $(Q_{supplied})$ **–** This count the total heat supplied by the fuel on combustion, and this is calculated as,

$$Q_{supplied} = m_f \; x \; CV \qquad in\; KJ/min$$

Where,

m_f = Mass of fuel consumed in Kg/min

CV = Calorific value of fuel in KJ/Kg

Heat Equivalent to Brake Power (Q_{bp}) **–** Heat equivalent to useful work can be measured by measuring the brake power of the engine.

$$Q_{bp} = BP \; x \; 60 \qquad in\; KJ/min$$

$$Q_{bp} = \frac{2\,\pi\,N\,T}{60\;x\;1000}\; x\; 60$$

Where,

$BP = \frac{2\,\pi\,N\,T}{60\;x\;1000}$ = brake power in KJ/min

Heat Rejected to Cooling Water (Q_{water}) **–** Heat part which is carried away by the cooling water is considered as heat rejected to cooling water, and it is calculated as,

$$Q_{water} = m_{water} x \; Cp_{water} \; x \; \Delta T_{water} \qquad in\; KJ/min$$

Where,

m_{water} = mass of cooling water in Kg/min

Cp_{water} = specific heat of water in KJ/Kg K

ΔT_{water} = $(T_{wo} - T_{wi})$ = cooling water temperature difference in °C

Heat carried away by exhaust gases (Q_{gas}) **–** Heat part which is carried away by the exhaust gases is considered as heat rejected from the engine, and it is calculated as,

$$Q_{gas} = m_{gas} \; x \; Cp_{gas} \; x \; (T_{gas} - T_{air}) \qquad in\; KJ/min$$

Where,

m_{gas} = mass of exhaust gas in Kg/min

Cp_{gas} = specific heat of exhaust gas in KJ/Kg K

T_{gas} = temperature of exhaust gas in °C

T_{air} = temperature of air at inlet in °C

Heat unaccounted $(Q_{unaccounted})$ **–** Heat part which is expended in heating engine parts, engine oil, heat losses by radiation and convection to surrounding, all are consider under the unaccounted heat loss, and it is calculated as,

$$\boldsymbol{Q_{unaccounted} = Q_{supplied} - (Q_{bp} + Q_{water} + Q_{gas})}$$

All these information if tabulated as the heat balance sheet as follow,

Heat Supplied	KJ/min	%	Heat Expenditure	KJ/min	%
Heat supplied by fuel on combustion $(Q_{supplied})$	---	100 %	1. Heat utilized in Brake Power (Q_{bp})	---	---
			2. Heat rejected to cooling water (Q_{water})	---	---
			3. Heat carried away by exhaust gases (Q_{gas})	---	---
			4. Unaccounted heat loss $(Q_{unaccounted})$	---	---
Total heat supplied	---	100%	Total heat expenditure	---	100%

Prob . 1 – A single cylinder engine running at 2000rpm develops a torque of 10Nm. The indicated power of the engine is 2.3 KW. Find the loss due to friction power as the percentage of brake power. Also calculate mechanical efficiency of the engine.

Ans. – No. of cylinder, K = 1, N = 2000rpm, T= 10Nm, IP = 2.3 KW.

Loss of friction power as % of brake power –

We know, the brake power is,

$$BP = \frac{2\,\pi\,N\,T}{60\,x\,1000}\,KW$$

$$BP = \frac{2\,\pi\,x\,2000\,x\,10}{60\,x\,1000}$$

$$\boldsymbol{BP = 2.0944\,KW}$$

The friction power is,

$FP = IP - BP$

$FP = 2.3 - 2.09$

$\mathbf{FP = 0.2056\ KW}$

Now, % loss of friction power as of BP is,

$$\%\text{ loss of friction power as of BP} = \frac{FP}{BP}$$

$$\%\text{ loss of friction power as of BP} = \frac{0.21}{2.09}\ x\ 100$$

$$\mathbf{\%\ loss\ of\ friction\ power\ as\ of\ BP = 9.817\ \%\ \ldots\ldots Ans.}$$

Prob . 3 – The following data were recorded in a test one hour duration on a single cylinder oil engine working on 4-stroke cycle,

Bore = 300mm
Stroke = 450 mm
Fuel used = 8.8 Kg
Calorific value of fuel = 41800 KJ/Kg
Average speed = 200 rpm
Mean effective pressure = 5.8 bar
Brake friction load = 1860 N
Quantity of cooling water = 650 Kg
Temperature rise = 22 °C
Diameter of brake wheel = 1.22 m

Calculate,

1. Mechanical efficiency
2. Brake thermal efficiency
3. Draw Heat Balance Sheet

Ans. – $Duration\ of\ trial = t = 1\ hr = 60\ min = 3600\ sec.$, $No.of\ cylinder = 1$, $No.of\ strokes = 4$, $bore = d = 300mm = 0.3m$, $stroke = L = 450mm = 0.45m$, $m_f = 8.8\ Kg\ in\ 1\ hr\ trial$, $CV = 41800\ KJ/Kg$, $N = 200\ rpm$, $Mean\ effective\ pressure = P_{me} = 5.8\ bar$, $Brake\ load = W = 1860\ N$, $Quantity\ of\ cooling\ water = m_w = 650\ Kg\ in\ 1\ hr\ trial$, $Temperature\ rise = \Delta T = 22\ °C$, $Diameter\ of\ brake\ wheel = D_b = 1.22\ m.$

1. Mechanical efficiency -

We know, indicated power is,

$$IP = \frac{P_{me}\ L\ \ A\ \ n}{60\ x\ 1000}$$

$$IP = \frac{P_{me}\ L\ \left(\frac{\pi}{4}\ d^2\right)\left(\frac{N}{2}\right)}{60\ x\ 1000}$$

$$IP = \frac{5.8\ x\ 10^5\ x\ 0.45\ x\ \left(\frac{\pi}{4}\ 0.3^2\right)\left(\frac{200}{2}\right)}{60\ x\ 1000}$$

$$\mathbf{IP = 30.75\ KW}$$

We know BP using brake drum dynamometer is,

$$BP = \frac{2\ \pi\ N\ W\ R_b}{60\ x\ 1000}$$

$$BP = \frac{2\ \pi\ x\ 200\ x\ 1860\ x\ 0.61}{60\ x\ 1000}$$

$$\boldsymbol{BP = 23.76\ KW}$$

Thus, mechanical efficiency is,

$$\eta_{mech} = \frac{BP}{IP}$$

$$\eta_{mech} = \frac{23.76}{30.75}$$

$$\boldsymbol{\eta_{mech} = 0.7727 = 77.27\%\ \ldots..Ans.1}$$

2. Brake thermal efficiency -

We know, brake thermal efficiency is,

$$\eta_{bth} = \frac{BP}{m_f\ CV}$$

$$\eta_{bth} = \frac{23.76}{\frac{8.8}{3600}\ x\ 41800}$$

$$\boldsymbol{\eta_{bth} = 0.2325 = 23.25\ \%\ \ldots..Ans.2}$$

3. Draw Heat Balance Sheet (on the basis of KW) –

Heat supplied –

$$Q_s = m_f\ CV$$

$$Q_s = \frac{8.8}{3600}\ x\ 41800$$

$$\boldsymbol{Q_s = 102.18\ KW}$$

Heat Equivalent to BP –

$$Q_{bp} = BP$$

$$Q_{bp} = 23.76$$

$$\boldsymbol{Q_{bp} = 23.76\ KW}$$

Heat carried out by cooling water -

$$Q_w = m_w\ x\ C_{p\_water}\ x\ \Delta T$$

$$Q_w = \frac{650}{3600}\ x\ 4.187\ x\ 22$$

$$\boldsymbol{Q_w = 16.63\ KW}$$

Unaccounted Heat (Heat carried out by exhaust, radiation) -

$Q_{un} = Q_s - (Q_{bp} + Q_w)$

$Q_{un} = 102.18 - (23.76 + 16.63)$

$\boldsymbol{Q_{un} = 61.79\ KW}$

Thus heat balance sheet on KW basis is,

Heat Supplied	KW	%	Heat Utilized	KW	%
$Q_s = m_f\ \ CV$	102.18	100%	Heat Equivalent to BP, $Q_{bp} = BP$	23.76	23.25%
			Heat carried out by cooling water, $Q_w = m_w\ x\ C_{p\_water}\ x\ \Delta T$	16.63	16.27%
			Unaccounted Heat, $Q_{un} = Q_s - (Q_{bp} + Q_w)$	61.79	60.47%
Heat Supplied	102.18	100%	Heat Utilized	102.18	100%

1. Super charging :

Supercharging is the method of increasing the power output of the engine without increasing its weight and size. This can be made possible by increasing the density of the charge supplied to engine cylinder, which ensures greater amount of the charge aspirated into the same stroke volume.

Thus supercharging can be defined as, *the admittance of the more charge into the engine cylinder than what the engine can take during the normal suction stroke.*

The device which is used to increase the pressure (density) of air or charge above the atmospheric pressure is called supercharger.

a. Factors which increase the power output by supercharging –

– **Increase in power due to increase in inlet pressure of the charge –**

The output of the supercharged engine is increased due to the increased density of the charge. This is obtained by increasing the inlet pressure of the charge by using compressor called as supercharger.

As charge pressure increased, the corresponding temperature of charge also increases which decreases the charge density, therefore it is cool in the after-cooler before it enters into the cylinder.

– **Increase in power due to the additional filling of the engine cylinder –**

The high pressure fresh charge in the engine cylinder, causes the compression of the residual gases present in the clearance volume. This facilitates the additional filling of the cylinder.

Thus the density as well as the volume of the charge increases. Hence the power output is increased.

– **Increase in power due to the positive gas exchange work –**

In the case of the naturally aspirated engine, the suction pressure is less than the exhaust pressure. So a negative work is done during the gas exchange process.

In the supercharged engine the inlet pressure is higher than the exhaust pressure, therefore a positive work is done during the gas exchange process. This is because, during the exhaust stroke the piston has to move only against the back pressure in both the cases whether the engine is naturally aspirated or supercharged, but during the suction stroke the pressure is higher in the supercharged engine, hence there is additional energy delivered to the piston and therefore more power is obtained.

b. Effect of Supercharging -

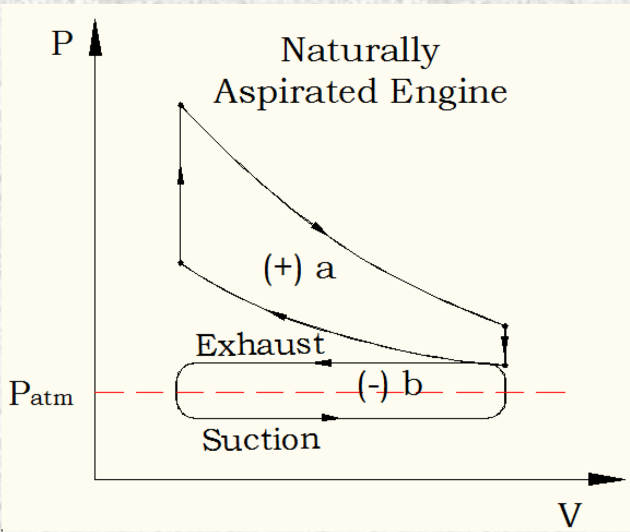
The induced charge by the supercharger during suction helps in better mixing of fuel and air during its compression stroke due to the turbulent effect created by the supercharger and the increased temperature helps in vaporization of fuel.

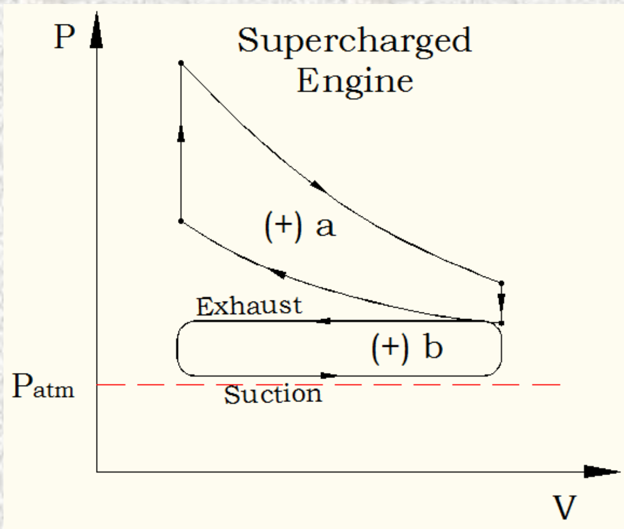
Thus the power output of the supercharged engine is increased without increasing engine weight and size.

The increase in temperature of charge, reduces the charge density and it is may tend to detonation in SI engine.

The supercharger is driven from the engine power itself, hence it reduces the power output available.

The p-v diagram for the naturally aspirated and supercharged engines are shown below,





Mean effective pressure for naturally aspirated engine is,

$$P_{me} = \left[\frac{area\ (a) - \ area\ (b)}{l}\right] \times Spring\ Cosntant$$

Mean effective pressure for supercharged engine is,

$$P_{me} = \left[\frac{area\ (a) + \ area\ (b)}{l}\right] \times Spring\ Cosntant$$

Thus there is increased mean effective pressure with supercharged engine, which result into increase in power output of the engine.

c. Objectives of Supercharging -

There are following objective of supercharging an engines,

1. To increase the power output of the engine by increasing the density of charge at intake.
2. To reduce the weight to power ratio. It is very useful in case of aircraft, racing cars and marine applications.
3. To overcome the loss of power at high altitudes either in case of static engines or in case of aircraft applications. The loss of power of an engine is estimated to be 1% per 100 m of altitude.
4. To reduce the bulk of the engine where weight and space are important consideration like in case of locomotives and marine engines.

d. Effects of Supercharging -

- **Power Output :** The power output of a supercharged engine is higher than its naturally aspirated engine, due to increased in mean effective pressure.

- **Fuel consumption :** In SI engines, the use of a lower compression ratio, increased heat losses due to higher values of specific heats and dissociating losses at higher temperatures, all result in lowering the thermal efficiency and in producing higher brake specific fuel consumption for supercharged engines.
  In CI engines fuel consumption is less than that of naturally aspirated engines due to better fuel distribution, improved combustion and increased mechanical efficiency.

- **Volumetric Efficiency :** Supercharging increases volumetric efficiency slightly, since the residual gases in the clearance are compressed by the inducted charge which is at a pressure higher than the residual exhaust gas pressure.

- **Mechanical Efficiency :** Mechanical efficiency of the supercharged engine is increases, inspite of an Increase in intake pressure, increases gas load, hence large bearing areas and heavier components are needed. Thus frictional losses are increased. But the friction losses is less than the power gained by supercharging.

2. Limits of Supercharging -

Supercharging of an IC engine is limited to certain limit, due to following reasons,

1. Increased maximum pressure in the cylinder tend to increase detonation tendency in case of SI engines.
2. Increase in intake pressure increases the maximum pressure attained in the cylinder. This increases the weight of the cylinder since the engine has to be designed to withstand higher pressure.
3. Higher peak pressure increases the friction losses, and bearing loads.

4. Excessive supercharging may result into higher mean cylinder wall temperatures and it may cause the melting of piston top and pre-ignition problems.
5. Higher temperature will lead to higher exhaust gas temperature, it causes overheating of exhaust valves.

a. Supercharging limits of SI engines –

There are following reasons due to which SI engines are not supercharged,

1. The compression ratio in SI engines is limited from knock and detonation considerations for a given octane rating of fuel to be used.
2. Knocking tendency in SI engines increases with the increase in pressure temperature, compression ratio and density of charge and the mixture strength.
3. The engine supercharged, the increased induction pressure will increase the peak pressure and temperature which may lead to increase in detonation.
4. Otherwise supercharged engine, the compression ratio needs to reduced, but it reduces the power output and thermal efficiency and increase the bsfc.

SI engines are supercharged in the following cases only,

1. For compensation of reduced pressure at high altitudes in case of static engines and aero-plane engines.
2. For increased power requirement at the time of take off in aero engines on the expense of high fuel consumption. But it is required for short duration of time.

b. Supercharging limits of CI engines –

1. In CI engines, increase in induction pressures due to supercharging help to decrease the knocking tendency, and improve combustion characteristics.
2. Supercharging in these engines improves power output, thermal efficiency and it can use inferior fuels with low cetane rating.
3. In CI engines, supercharging is limited due to peak pressure and temperatures, thermal stresses developed, mean temperature of cylinder wall temperatures and loads on bearings etc.
4. Also supercharging limited due to fuel economy in CI engines.

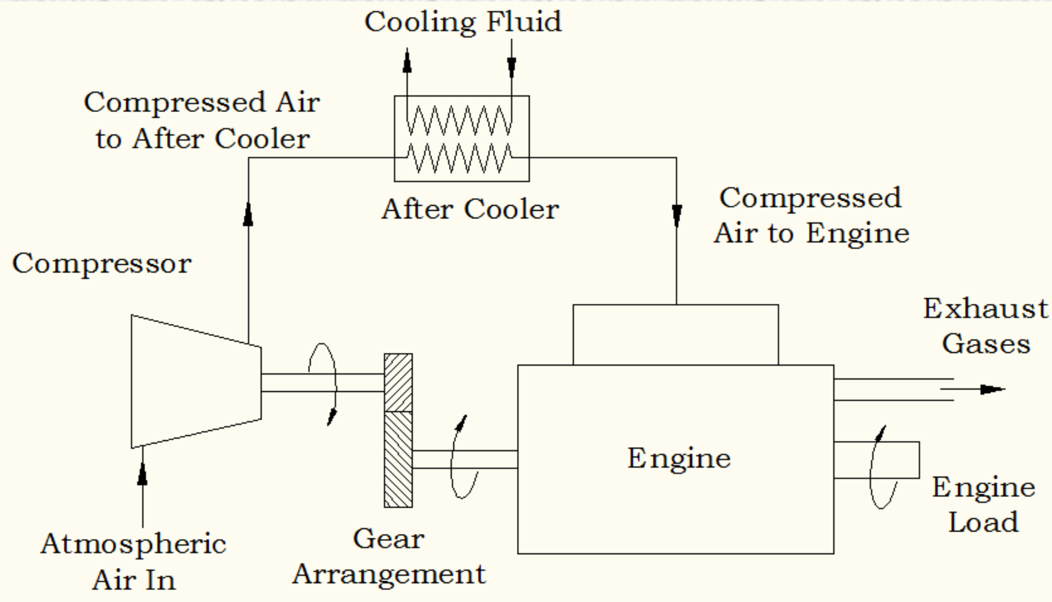
3. Methods of Supercharging -

The following arrangements are used to supercharge engines,

a. Mechanical Supercharging (Gear Driven Supercharger) –

In this method, a compressor coupled to the engine with step up gearing to increase the rotational speed of compressor. In this a certain percentage of engine output is utilized to drive the compressor. Compressor uses atmospheric air and compressed it and supplied to engine through after cooler. After cooler through which the compressed cool air supplied to engine. This will further increase the density of the intake air.

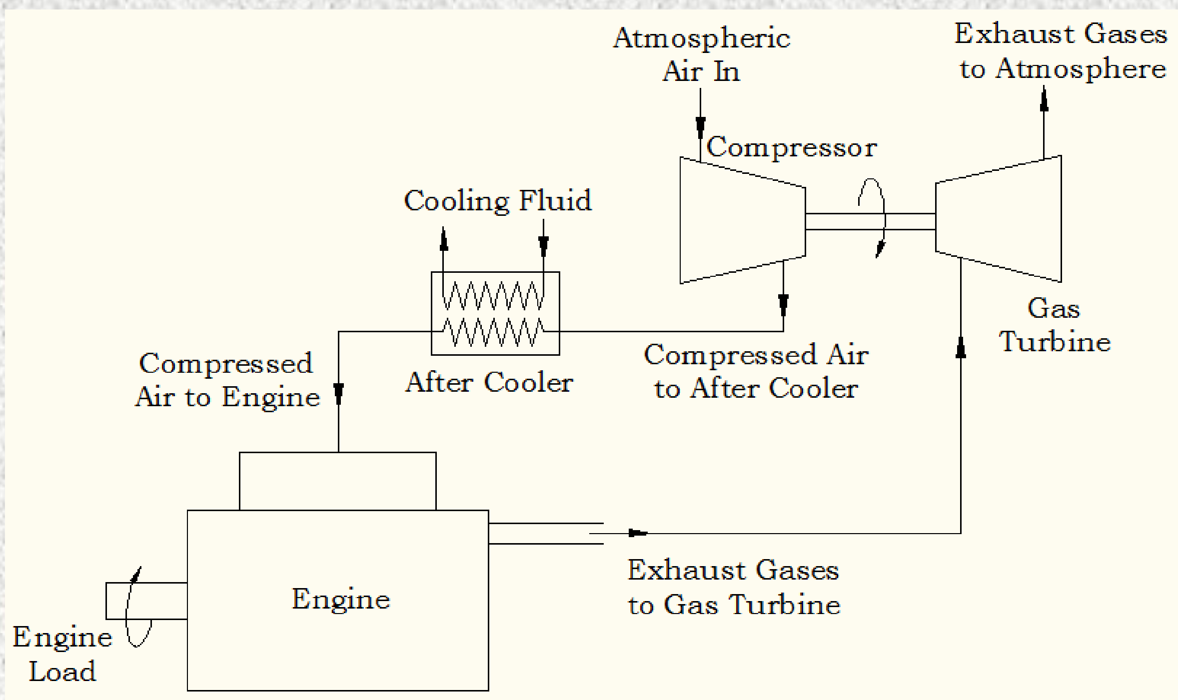
The net output increase due to supercharging is obtained by subtracting the power used for compressor from the engine gross output.



b. Turbo-charged (Gas Turbine Driven) Supercharger –

In this arrangement the engine so equipped are said to be turbo-supercharged. The exhaust energy of the engine is used to drive the gas turbine which is coupled to a compressor.

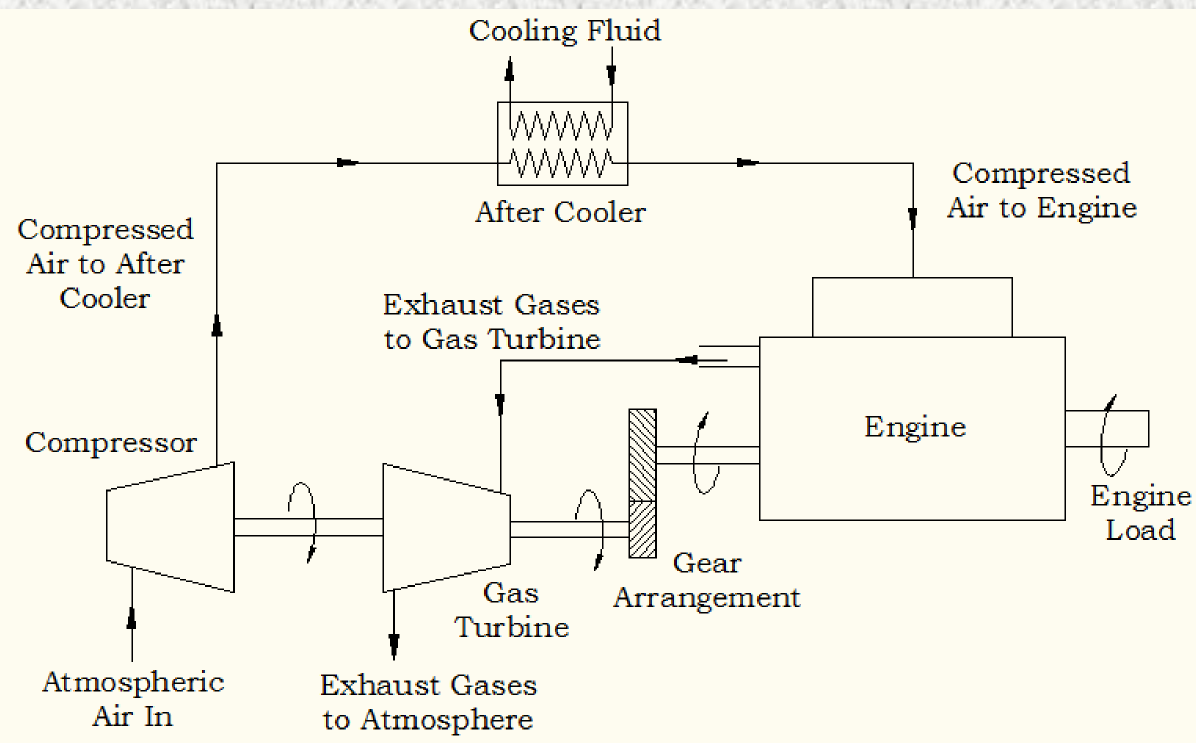
There is no mechanical coupling of compressor or gas turbine with the engine. The hot exhaust gases are supplied to gas turbine where its energy is utilized to drive gas turbine. The mechanical work output of gas turbine is supplied to compressor, which inhaled atmospheric air and compressed it and supplied to engine through after cooler. After cooler through which the compressed cool air supplied to engine. This will further increase the density of the intake air.



c. Engine Driven, Compressor & Turbine Supercharger –

In this, if the turbine output is insufficient to run the compressor particularly at part loads, the engine power takes care of the load of compressor. Engine runs the compressor through the common shaft passing through the gas turbine.

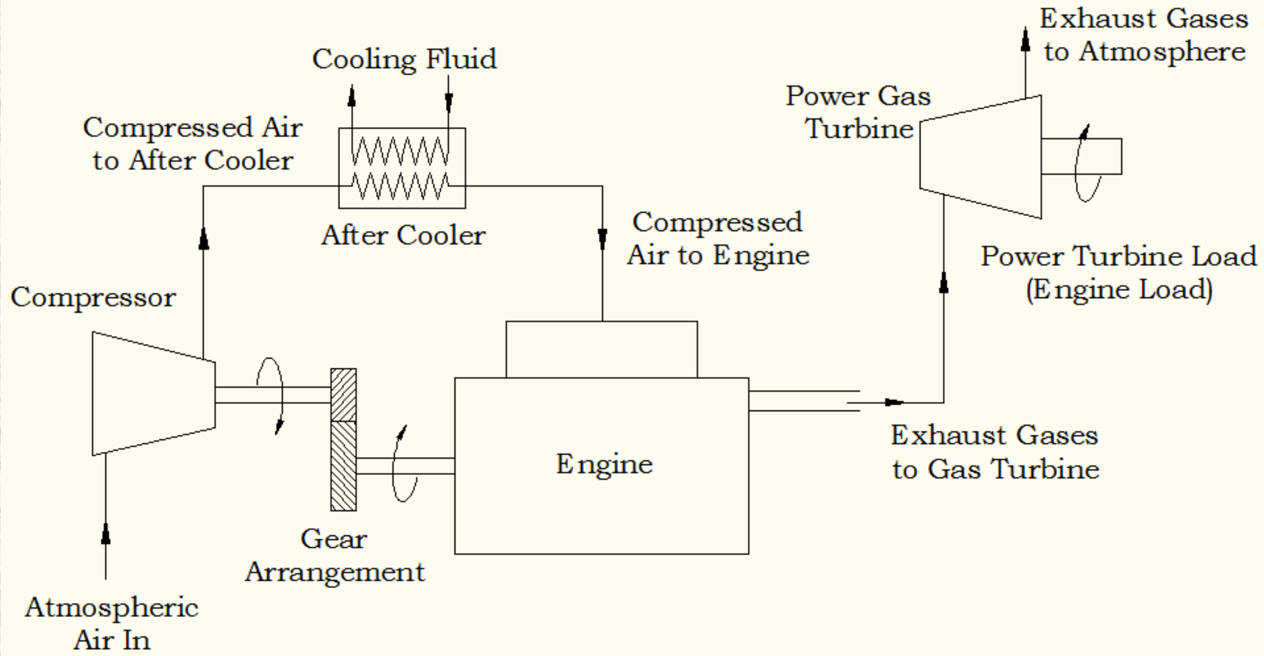
When gas turbine producing the sufficient power then it supplied power to run the compressor and any addition power of turbine can be fed to the engine.



d. Free Piston Engine or Gear Driven Supercharger –

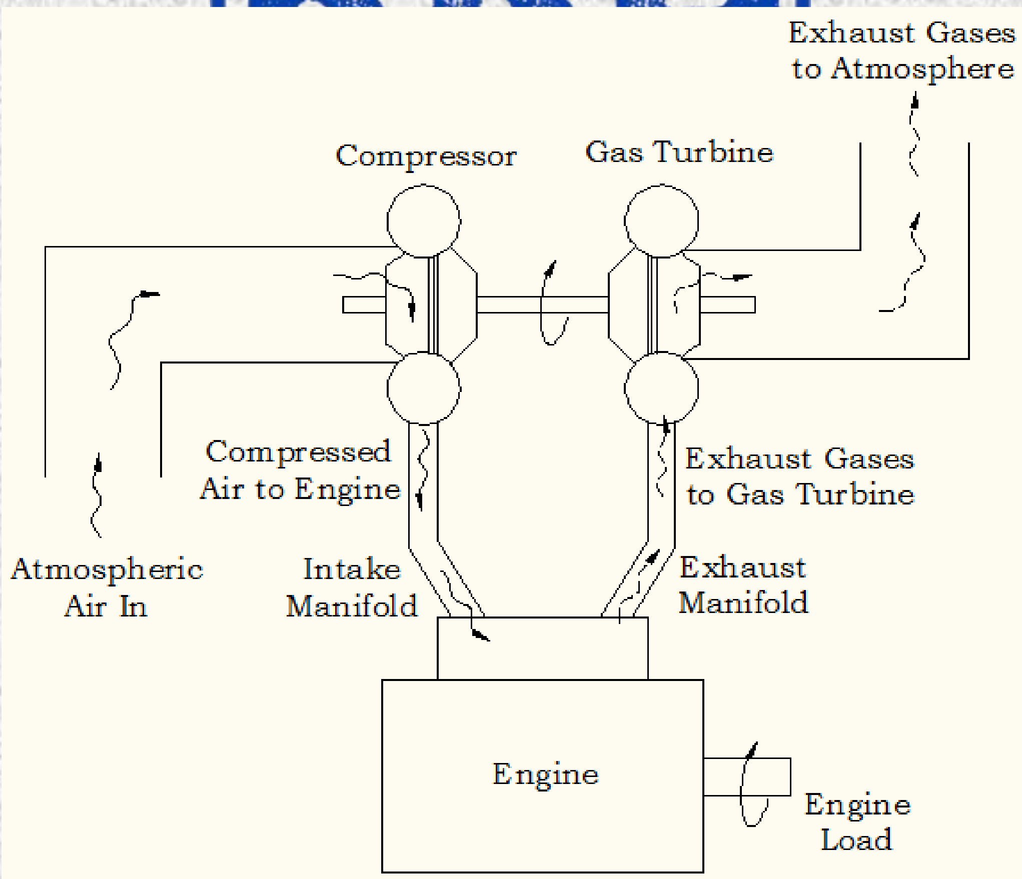
In this arrangement, engine drive only the compressor. Air from the compressor supplied to engine through after cooler. Engine supercharging increases the cycle pressure and temperature. The exhaust gas are supplied to power gas turbine, which utilized energy of exhaust gas and converted to mechanical work output. The ultimate load is taken care by this power gas turbine output.

Hence may time it is called as free piston engine.



4. Turbo-charging -

The major parts of a turbocharger are, turbine wheel, turbine housing, turbocharger compressor wheel, compressor housing and bearing housing.



During the engine operation, hot exhaust gases blow out through the exhaust valve opening into the exhaust manifold. The exhaust manifold and the connecting tubing route these gases into turbine housing. As the gases pass through the turbine housing, they strike on the fins or blades on the turbine wheel.

When engine load is high enough, there is enough gas flow and this makes the turbine wheel to spin rapidly. The turbine wheel is connected to the compressor wheel by the turbo-shaft.

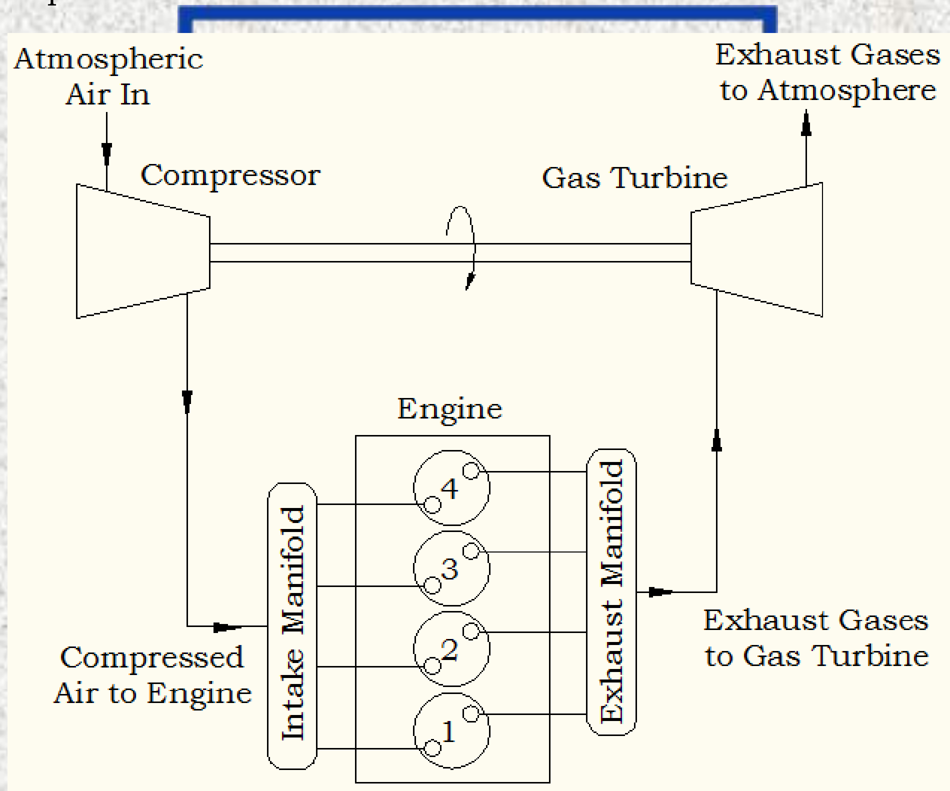
As such the compressor runs with the turbine which inhaled atmospheric air and supplied the compressed air to the engine.

5. Methods of Turbo-charging -

There are following methods of turbocharging,

a. Constant Pressure Turbo-charging –

In this method the exhaust of all the cylinders are released to common exhaust manifolds from where it supplied to run the gas turbine. It could be seen that the exhaust pressure of all the cylinder is constant and above the atmospheric pressure. The exhaust manifold is made big in size to absorb any pressure pulsations. The system is very efficient in operation.



Limitations of constant pressure turbo-charging –

- To maintain constant pressure and to avoid any pressure pulsations, it is necessary to use large diameter exhaust pipe. It increases the space requirement of the engine.
- Response of the system to change in load is poor because the acceleration of turbine occurs slowly when the load on the engine is suddenly increased reason

is that due to increased load the corresponding exhaust energy is not sufficient to accelerate the turbine.

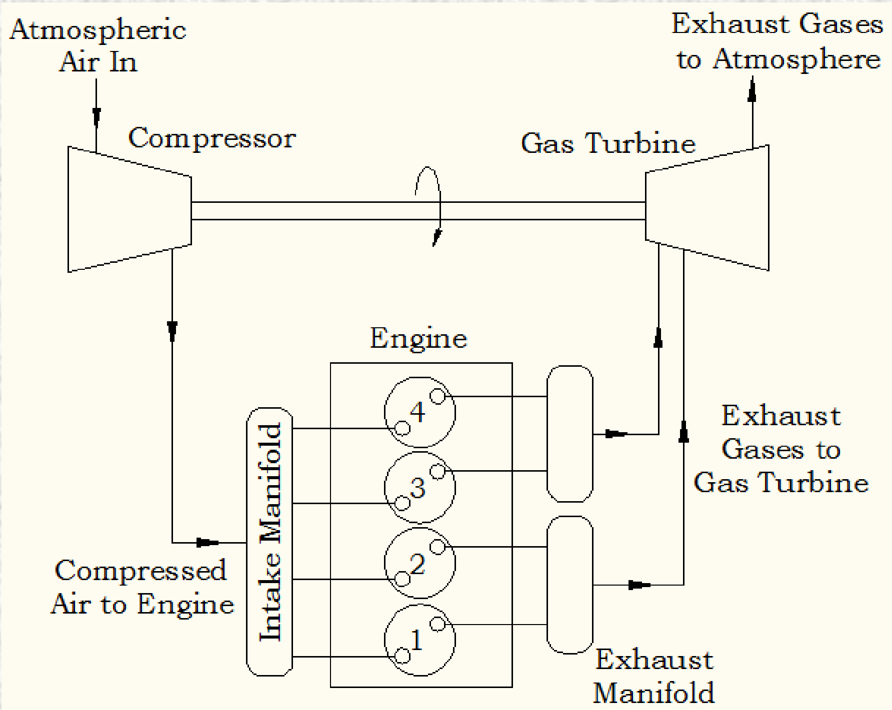
- For efficient running of turbine higher pressure ratio are necessary. It implies that large pressure drop must occurs during scavenging. It makes the process of scavenging bit difficult.
- This system is not suitable for two stroke engines since the exhaust energy converted by turbine is not sufficient to run the compressor alone.

b. Pulse (Buchi Type) Turbo-charging –

The main objective of this system is to utilize the kinetic energy of blow down gases to drive the gas turbine without much increase in exhaust pressure. To achieve this, the exhaust lines are grouped together to receive the exhaust pulses as soon as the exhaust valve opens.

Then these gases are gathered and passed from the narrow exhaust pipes directly to the gas turbine by the shortest route.

Separate exhaust pipes are used so that exhaust process of various cylinders do not interface with another.



Limitations of pulse (Buchi Type) turbo-charging –

- With high turbine pressure ratio, the recovery of energy is poor. Pressure ratio for turbine is limited too.
- Engine with large number of cylinders requires complicated intake and exhaust pipe arrangement.
- Poor turbine efficiency is obtained in case of one or two cylinders.
- Scavenging process is disturbed if the waves have to travel through long exhaust pipes to turbine.

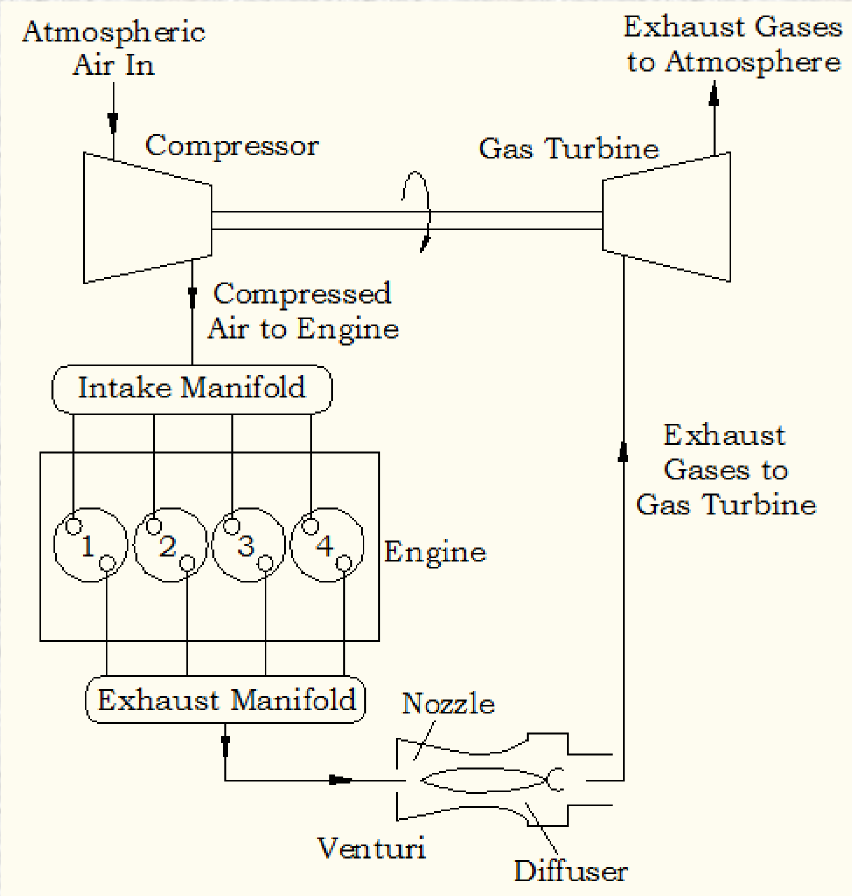
c. Pulse Converter Turbo-charging –

Pulse converter turbocharging system has the advantages of both constant pressure and pulse turbo charging system and avoids drawbacks of both.

In this system, the exhaust of engine cylinder from exhaust manifolds is joined in a venturi section. The pressure pulse is converted into kinetic energy in the nozzle section of venturi. It creates a suction effect in the exhaust line and helps in scavenging process of the engine.

In the diffuser part of the venturi the pressure gradually increase and high pressure exhaust is supplied to the turbine.

This system is suitable and efficient even at part load conditions for low pressure ratio turbocharging.



Exercise

1. What is dynamometer? Give the classification of dynamometer.
2. Write short note on Hydraulic Dynamometer.
3. Explain with neat sketch rope brake dynamometer used in testing of IC engine.
4. With the help of neat sketch explain the working of Prony type dynamometer.
5. Explain the working of mechanical type of transmission dynamometer.

6. Discuses the principle and working of an Eddy Current dynamometer.
7. What is Willan's line method ?
8. Write short note on Heat Balance Sheet.
9. Define the Indicated Power, Brake Power, Friction Power.
10. Explain the motoring test to measure the friction power.
11. Explain the Morse test during performance test of IC engine.
12. Discuss the volumetric method of fuel measurement.
13. Explain the working of orifice flow meter for measurement of fuel consumption.
14. With neat sketch explain the air flow meter.
15. Define the volumetric efficiency, mechanical efficiency, thermal efficiency.
16. Discuss the various losses in IC engines.
17. An engine working on Otto cycle, having compression ratio = 6, brake power = 14.8 KW, suction pressure = $1\,x\,10^5\ N/m^2$, and suction temperature = 27 °C, relative efficiency = 68%, mechanical efficiency = 78%. Calculate the air standard and thermal efficiency of an engine. **(Ans. $\eta_{air\_std} = 51.16\%$, $\eta_{ith} = 34.79\%$, $\eta_{bth} = 27.13\%$).**
18. A single cylinder, four stroke CI engine has the following data, bore = 10cm, stroke = 11cm, brake torque = 28 Nm, speed = 1600 rpm, imep = 6.5 bar. Calculate the brake power, indicated power, friction power, and mechanical efficiency. **(Ans. $BP = 4.69\ KW$, $IP = 7.49\ KW$, $FP = 2.79\ KW$, $\eta_{mech} = 62.6\%$).**
19. Morse test conducted on four cylinder, four stroke engine of 75mm bore and 100mm stroke. Brake Power : when all cylinder are working is 31.2KW, when cylinder-1 cut off is 22.2KW, when cylinder-2 cut off is 22.06KW, when cylinder-3 cut off is 21.76KW and when cylinder-4 cut off is 21.32KW. Clearance volume is 100cc and Calorific Value of fuel used is 43600 KJ/Kg.
   Calculate, indicated power, friction power, mechanical efficiency and air standard efficiency. **(Ans. $IP = 37.46\ KW$, $FP = 6.26\ KW$, $\eta_{mech} = 83.28\ \%$, $\eta_{air\_std} = 49.0\%$).**
20. An engine uses 0.28 Kg/KW hr of fuel based on brake power of calorific value of 45000 KJ/Kg. Its mechanical efficiency is 80%. Compression ratio is 5.6. Find brake thermal efficiency, indicated thermal efficiency, air standard efficiency and relative efficiency. Assume $\gamma = 1.4$. **(Ans. $\eta_{bth} = 28.57\ \%$, $\eta_{ith} = 35.71\ \%$ $\eta_{air\_std} = 49.8\ \%$, $\eta_{rel} = 71.7\ \%$).**
21. A six cylinder and four stroke engine has a bore diameter of 100mm and stroke length of 120mm, working on compression ratio of 8. The relative efficiency is 60% and indicated fuel consumption is 0.25 Kg/KW hr. Calculate the calorific value of fuel used. In case the engine has a indicated mean effective pressure of 8 bar at 3000 rpm. Also find the corresponding fuel consumption. **(Ans. $CV = 42502.9\ KJ/Kg$, $IP = 113.09\ KW$ $m_f = 28.27\ Kg/hr$).**
22. The following data obtained during a trial on two-stroke engine, bore = 200mm, stroke = 250mm, imep = 4.5 bar, fuel consumption = 7 K/hr, calorific value = 43600

Kj/Kg, speed = 3 revolution per second. Calculate, indicated power and indicated thermal efficiency. ***(Ans.*** $\boldsymbol{IP = 10.6\ KW,\ \eta_{ith} = 12.51\ \%}$**).**

23. What is supercharging ?
24. Differentiate between the supercharger and turbocharger.
25. Why supercharging is not preferred for SI engine?
26. Explain with neat sketch constant pressure turbocharger.
27. List the various methods of turbo-charging, explain briefly the pulse turbo-charging.
28. What is turbo-charging? Explain the thermodynamic cycle with turbo-charging.
29. What do we mean by supercharging? Differentiate between supercharged and non-supercharged engines.
30. Supercharging is preferred in diesel engine than petrol engine, why?

RKB
ENGG_EDU TUTORIALS

Unit – V : IC Engine Systems

Syllabus :

IC Engine Systems : *Cooling System, Lubrication System, Ignition System, Governing system, Starting System*
IC Engine Emissions and Control : *Air pollution due to IC engine and its effect, Emissions from petrol/gas and diesel engines, Sources of emissions, Euro norms, Bharat stage norms, Emission control methods for SI and CI engines*

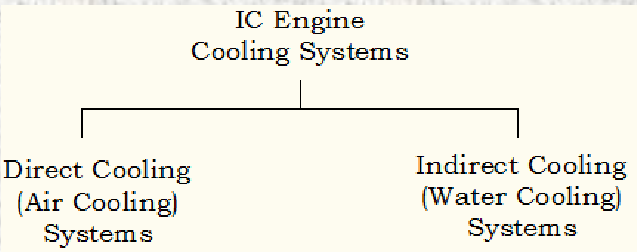
1. Cooling Systems : SPPU : Dec.-15, 6-Marks

Internal combustion engines, the process of converting thermal energy to mechanical energy, high temperatures are produced in the cylinders of the engine as a result of the combustion process. A large portion of the heat from the gases of combustion is transferred to the cylinder head, walls, piston, and valves etc. unless this excess heat is carried out and these parts adequately cooled, the engine will be damaged.

A cooling system must be provided not only to prevent damage to the engine parts due to overheating, but also to maintain certain temperature limits in order to obtain maximum performance for m the engine.

Thus adequate cooling is a fundamental requirement associated with reciprocating internal combustion engines. Hence, a cooling system is needed to keep the engine from not getting so hot as to cause problems and yet to permit it to run hot enough to ensure maximum efficiency of the engine.

There are following types of IC engine cooling systems,

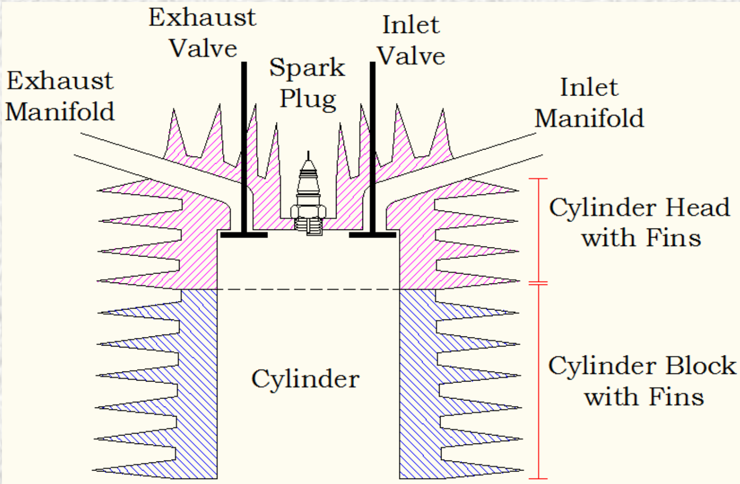


1.1 Direct Cooling / Air Cooling Systems –

In this type of IC engine cooling, heat from cylinder walls is directly transferred to surrounding air. In air cooling system, air is used as a cooling medium and it is used for small capacity engines. The heat transfer coefficient for air cooling is very low, which can be increased by using the forced flow of air over the engine surface and also by increasing the heat transfer rate from the cylinder surface with the help of fins provided.

In case of two wheelers and aircrafts, the high air velocities resulting from their motion is taken to advantage while in case of multi-cylinder air cooled engines used for cars the air velocities are increased with the help of a fan driven by the engine crank shaft.

The heat is gradually dissipated from the root of the fin upto its tip and to the surroundings air, due to this a temperature gradient exists along its length from the root to the tip. At the tip of the fin the heat transfer capacity becomes minimum due to least temperature difference, hence it is found that the rectangular fins are less efficient and heavier compared to trapezoidal or triangular fins.



The cooling fins are either cast integral with the cylinder and cylinder heat or they are fixed to the cylinder block separately. The number of the fins used are 2 to 3 per cm and the height of the fins are normally 2 cm to 5 cm. and spacing is limited to 2.5 mm.

Advantages of Air Cooling Systems –

1. Cooling medium is air which is free of cost available.
2. Engine becomes light weight and simple in design as there is no water jackets, radiator, cooling medium pumps and piping arrangements.
3. Thus the air cooling system is less costly and installation of the system is easy.
4. The problems like leakage of coolant, loose connection of piping's arrangement etc. are absent.
5. No antifreeze solution required.
6. It is almost maintenance free cooling system.

Disadvantages of Air Cooling Systems –

1. Heat transfer rate is less due to low heat transfer coefficient of air. Thus this system can used for low capacity engines.
2. Cooling is not uniform. It may cause the distortion of cylinder.
3. Cylinder wall temperatures are high.
4. If fan is used to improve heat transfer rate to lower the cylinder wall temperature, 5 to 10 % of power is lost to run the fan.
5. Specific fuel consumption is high.
6. It limits the use of compression ratio.
7. Its use is limited to scooters and motor cycles.

1.2 Indirect Cooling / Water Cooling Systems – SPPU : May-17, 6-Marks

In case of indirect cooling system, the heat from the cylinder walls is transferred to surrounding air through water.

Water cooled engines the cylinder and the cylinder heat are enclosed in a water jacket. This water jackets is connected to a radiator. Water is caused to flow in the jacket where it cools the engine, then it gives up this heat to air in the radiator and is again circulated in the water jacket.

For ease of cold starting many types and anti-freeze solutions are added to the water. Commonly used anti-freeze materials are, kerosene, wood alcohol, denatured alcohol, glycerin, sugar solution, calcium or magnesium chloride, ethylene glycol and propylene glycol.

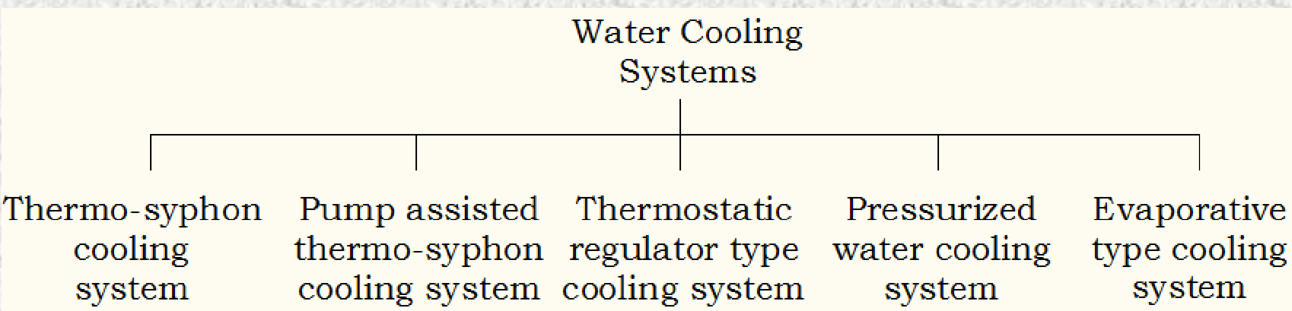
Advantages of Water Cooling Systems –

1. Cooling system is compact in size.
2. Heat transfer rate is high, thus it is used for heavy duty engines.
3. More even cooling is achieved.
4. Volumetric efficiency of the engine is high.
5. The engine may locate anywhere conveniently as cooling not depends on air motion.

Disadvantages of Water Cooling Systems –

1. Water cooling systems, required radiator, water pump and other hoses which increases overall weight of the system.
2. Engine performance related to climatic condition, starting is difficult in cold whether condition and high altitude.
3. It needed anti-freeze solution in cold condition.
4. Scale formation in water jackets reduces the heat transfer rate and thus cooling efficiency affected.
5. Cost and maintenance of the system is high.
6. Cooling system fails if adequate water level not maintained.

2. Types of water cooling system -

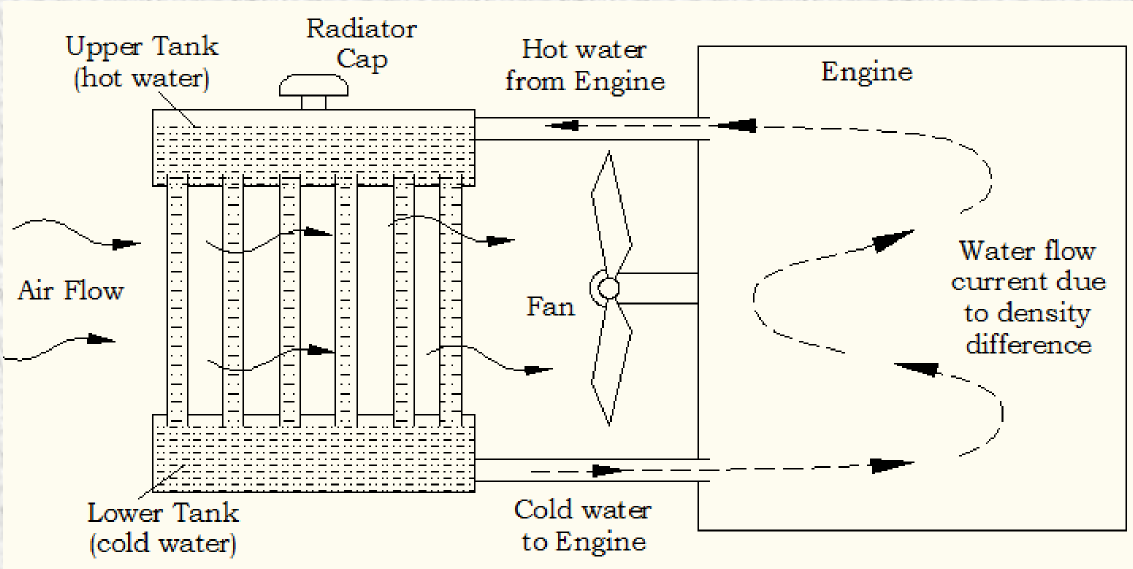


2.1 Thermo-syphon type water cooling systems –

In thermo-syphon type water cooling systems, the force required to circulate the water through the system is the difference in pressure head due to hot and cold water.

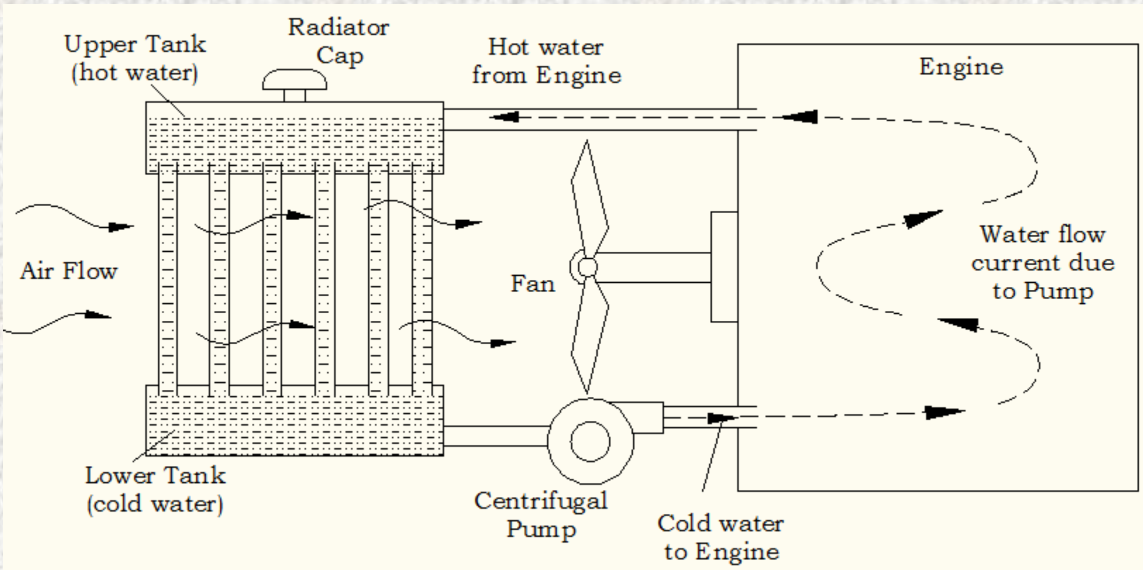
In this system, water in cooling water jackets surrounding to engine get heated by absorbing heat from the engine, and this hot water rises due to lower density and supplied to radiator at top. A radiator having upper and lower tanks connected with thin pipes attached to fins. Thus hot water during it pass cooled down due to air blow and collected into lower tank. From the lower tank of radiator water is supplied to cooling water jacket inlet. And the flow continue.

The heat removal rate / capacity is depends on the temperature difference of water in cooling jacket and engine wall temperature. And the water flow is obtained due to the density difference of hot and cold water.



2.2 Pump Assisted Thermo-Syphon water cooling systems -

In this the improved thermo-syphon system in which, the water is circulated through jackets around the parts of the engine to be cooled, and is kept in motion by a centrifugal pump which is driven by the engine.

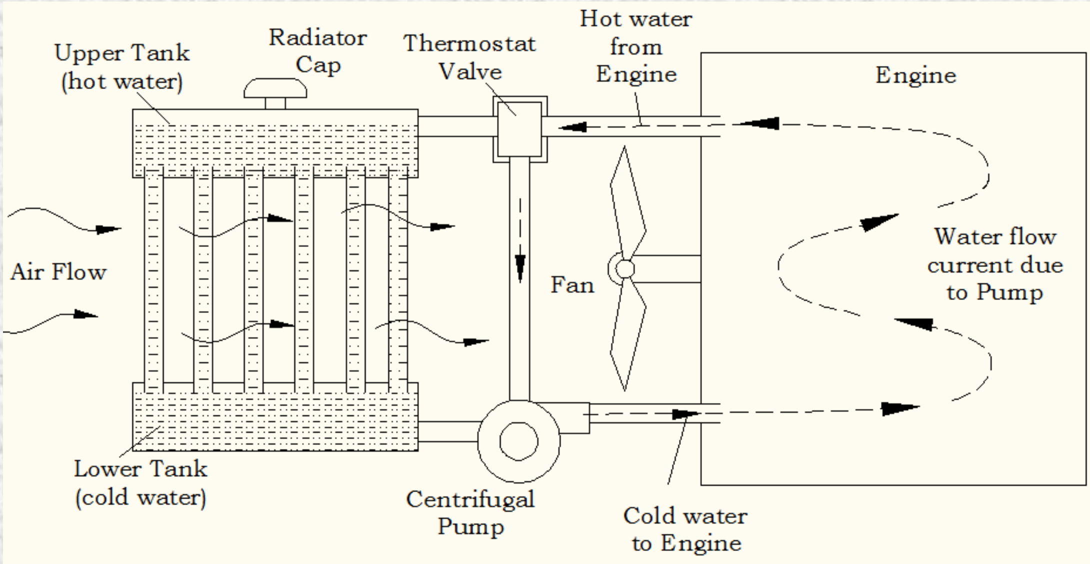


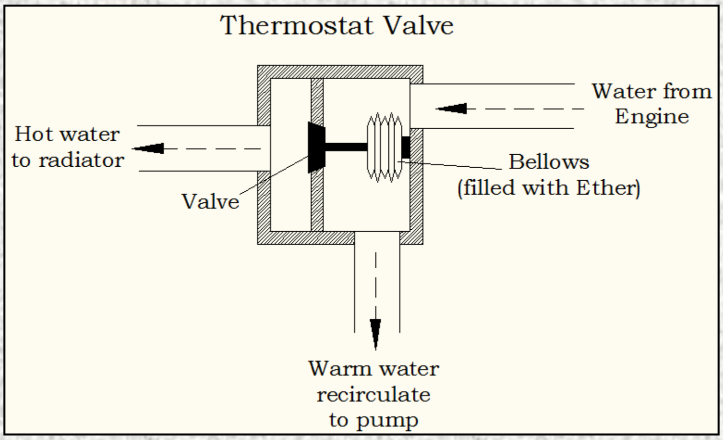
A pump is mounted on the engine and driven by the crankshaft with a fan belt. The water is passed through the radiator where it is cooled by air drawn through the radiator by a fan and by the air draft due to the forward motion of the vehicle.

The major advantage of this system is effective and positive cooling of all the parts of the engine. It can easity take the overload as the engine speed increases, water circulation is also increased and same effective cooling can be maintained by this system.

2.3 Thermostatic Regulator type water cooling systems – SPPU : Dec.-15, May-14, 6-Marks

In the pump assisted thermo-syphon system, as the pump starts, it will start the circulation of cooling water, and this circulating water continuously absorb the heat from engine and carried always due to which engine takes longer time to get warm up to desired working temperature which reduce the efficiency of the engine and engine output. To overcome this difficulty a thermostat is incorporated in the cooling system as shown,





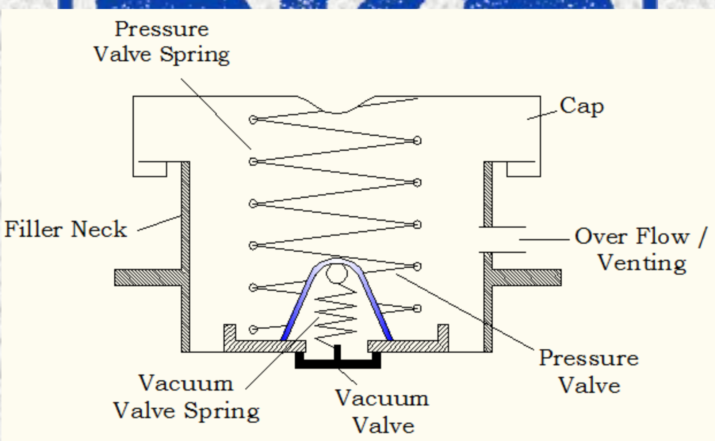
A thermostat consists of a valve attached to a bellows containing volatile liquid such as ether. Heating of bellow by the water around it causes vaporization of the liquid, which in turn expand the bellows and the valve get opened and water flow to radiator upper tank. If the water temperature is not reach to the working temperature of the engine, the ether not expanded enough to open the flow to radiator, so the outcome warm water flow re-circulated to engine, hence engine warm up time get reduced and soon engine reach to working condition, thus power loss or efficiency reduced time is minimized.

2.4 Pressurized water cooling systems –

The boiling temperature of water increases with the increase in pressure, this fact is used to cool the heavy duty engines, and the cooling system is known as pressurized water cooling systems.

In this type of cooling system, the pressure type radiator cap fits over the radiator. The cap contains two valves, the pressure valve and the vacuum valve. The pressure valve consists of a valve held against a valve seat by a calibrated spring. The spring holds the valve closed so that pressure is produced in the cooling system. If the pressure rises above that for which the system is designed, the pressure valve is raised off its seat. This relieves the excessive pressure.

The vacuum valve prevent the formation of a vacuum in the cooling system when the engine is stopped and begins to cool. If the vacuum forms, the atmospheric pressure from the outside causes the small vacuum valve to open and admit air into the radiator. Without a vacuum valve, the pressure within a radiator might drop so low that atmospheric pressure would collapse the radiator.

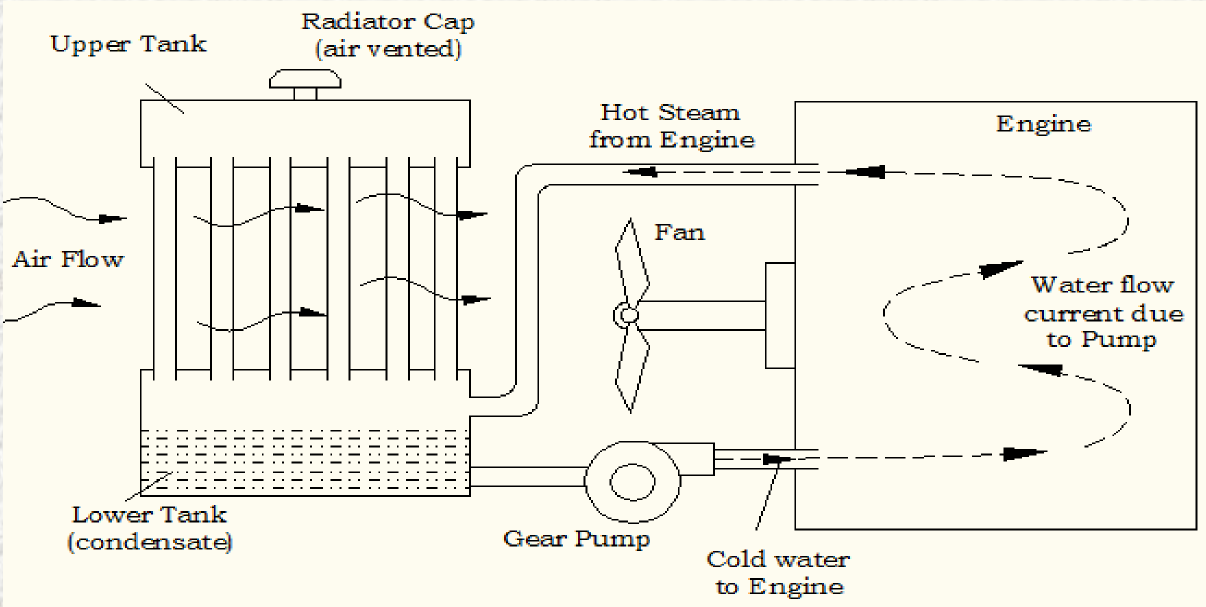


2.5 Evaporative water cooling systems –

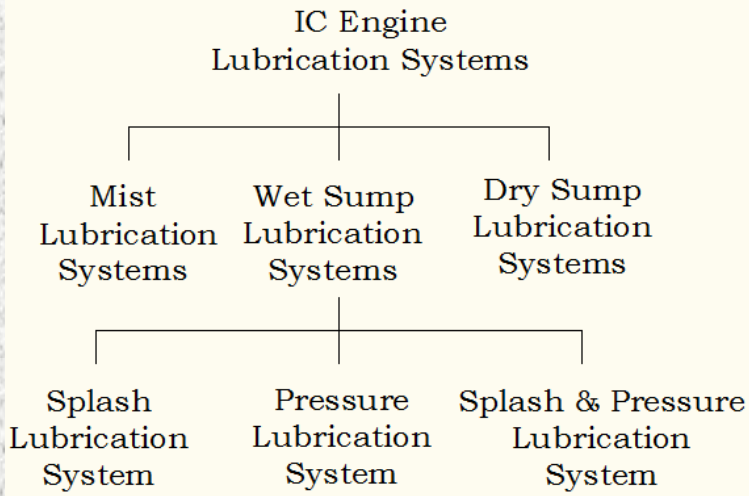
When the water vaporized, it absorbs the latent heat of vaporization from the metal surfaces to be cooled. The quantity of heat thus removed is much more and it is also possible to use a much smaller quantity of cooling water and a smaller radiator.

In this system a positive displacement type gear pump is used, to return the water from the condensed steam from the lower tank of the radiator to the engine water jackets. The water jacket is maintained under pressure and allows only the steam to leave the jacket by means of a throttle. The steam leaving the jacket is condensed in a radiator and returned to the engine jacket by the gear pump.

This system with relatively complex controls has a well designed steam cooling system, and using the high octane fuels it is possible to operate an engine at a temperature higher than that possible with the normal cooling system, and with a possible gain in thermal efficiency.



3. Lubrication Systems :



3.1 Mist Lubrication System –

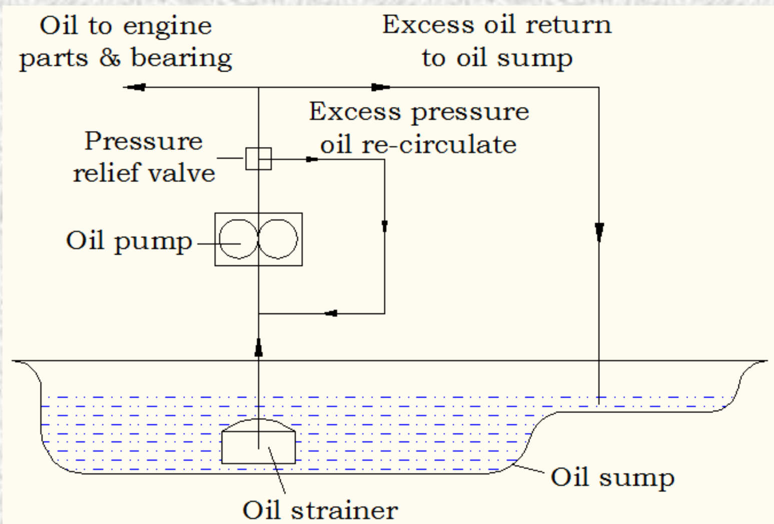
This system is used where crankcase lubrication is not suitable. The mist lubrication system is most commonly used in two stroke engines, the lubricating oil is mixed with the fuel itself. The fuel with oil is inducted through the carburetor, fuel is vaporized and the oil in the form of mist goes via the crankcase into the cylinder. The oil which strikes the crankcase wall lubricates the main and connecting rod bearings, and the rest of the oil lubricates the piston, piston rigs and cylinder.

The advantage of this system is its simplicity and low cost as it does not required any additional lubricating system parts like, pump, filters, piping or storage etc.

In this, the excess oil burned with fuel and deposits on piston and exhaust port which affect engine efficiency, also causes heavy exhaust smoke.

3.2 Wet Sump Lubrication System –

In wet sump lubrication system, the bottom of the crankcase contains an oil sump form which the lubricating oil is pumped to various engine components by a pump. After lubricating these parts, the oil flows back to the sump by gravity. Again it is picked up by a pump and re-circulated through the engine lubricating system.



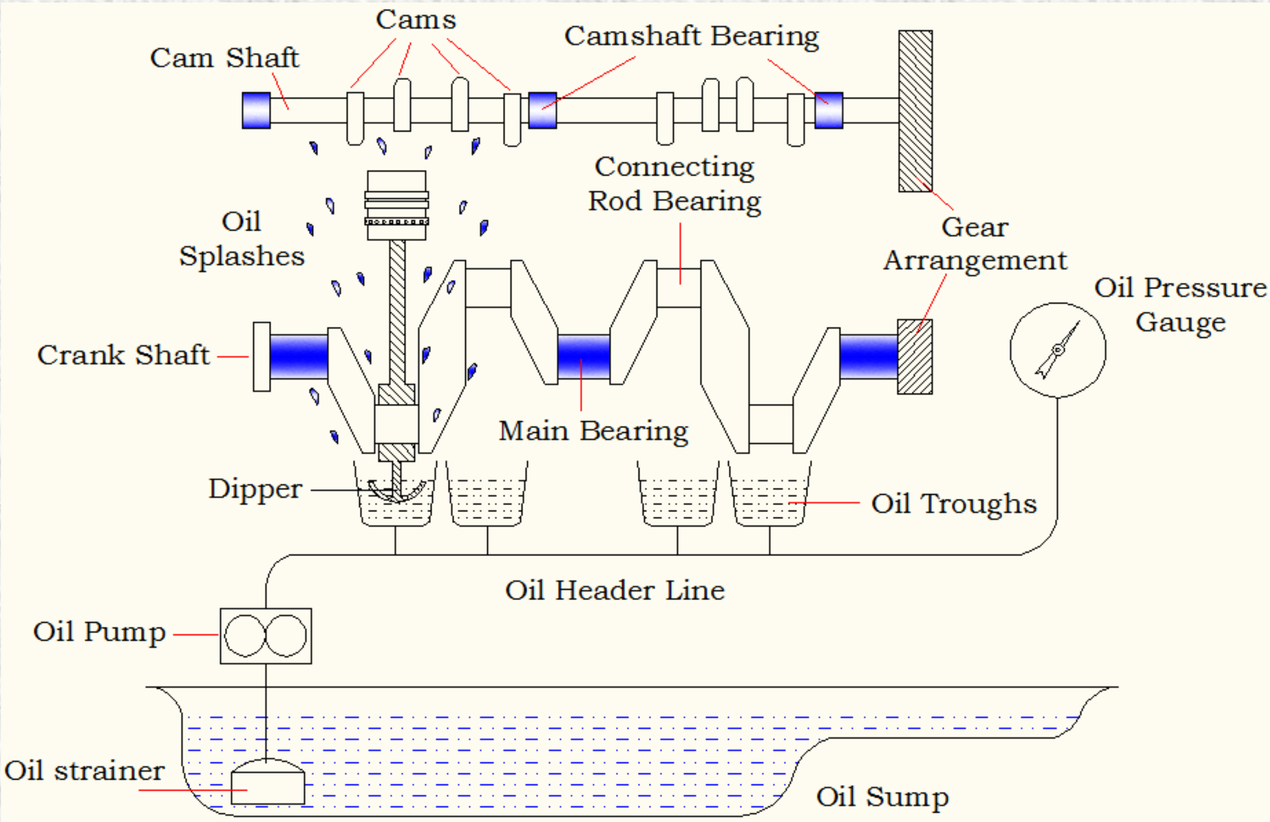
In this type, the oil sump is always having lubricating oil, hence called as wet sump lubrication system.

3.2.1 Splash Lubrication System –

This lubrication system is used in light duty engines like lawn mowers, golf carts etc.

The lubricating oil is pumped by a low pressure oil pump from oil sump to small toughs placed under each connecting rod. The big end of the connected rods are provided with the dipper which dips into the oil trough when come to bottom and splashed it when moves up. The oil is splashed to all engine parts like main bearings, camshaft bearings, lower portion of the cylinder walls, piston, cams etc. The excess oil returns into the sump by gravity.

The lubrication is done due to splash of oil thus it does not required high pressure oil pump and tubing.



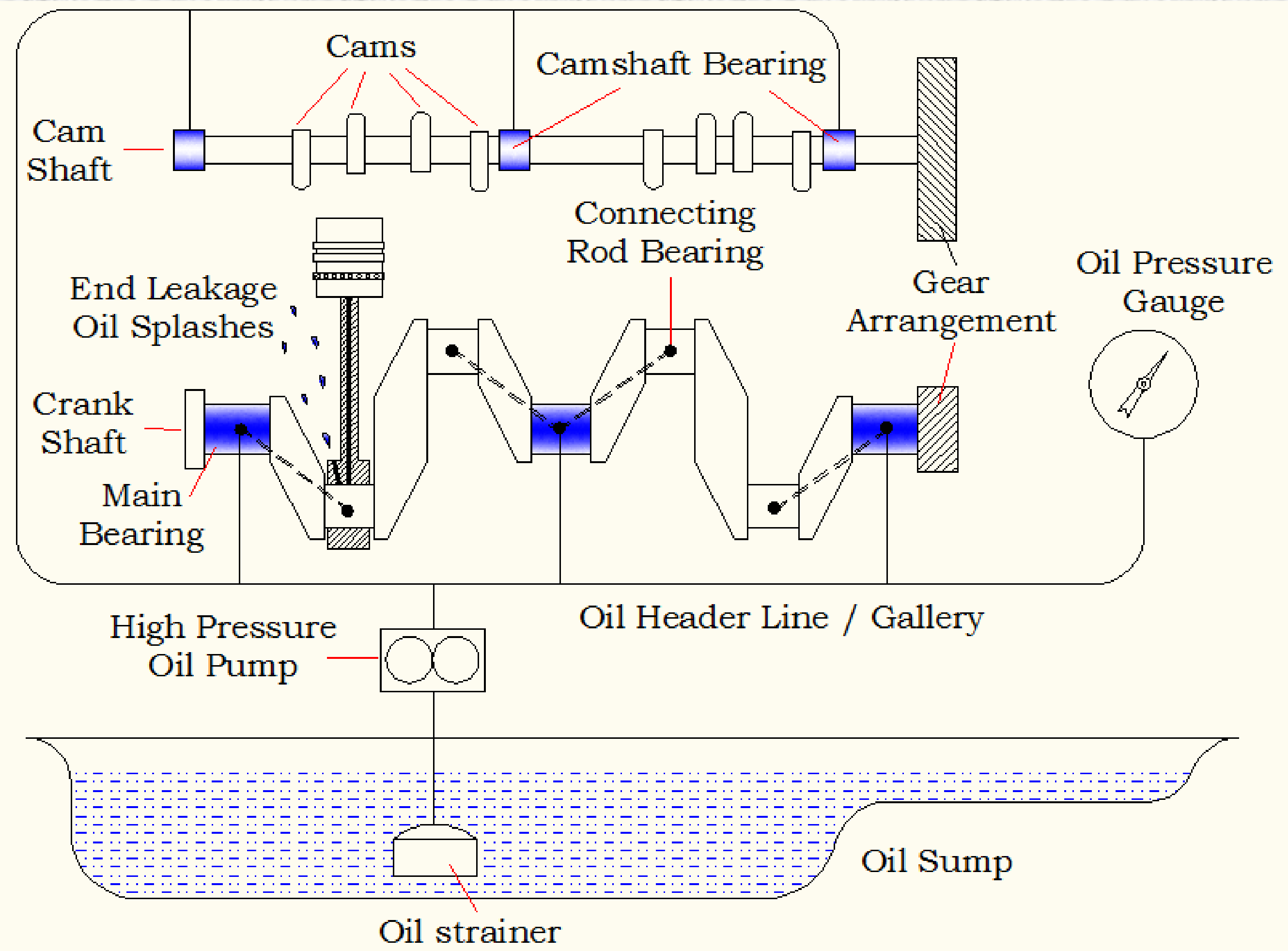
3.2.2 Pressure Lubrication System – SPPU : May-17, May-16, Dec.-15, 7-Marks

All modern cars and buses engines are lubricated by pressure lubrication system.

In this system, crankshaft having internal drill passage for lubrication, the high pressure pump delivered oil at 2 to 5 bar to the main distributor gallery, through a pressure relief valve. The pressure relief valve keeps the delivery pressure constant, excess pressure oil released and re-circulate back to pump.

The oil supplied to distributor gallery to crankshaft bearings and then through internal oil passage to crank pins etc. the oil further moves through connecting rod and lubricate piston pins , camshaft bearing and cams also lubricating with separate oil lines.

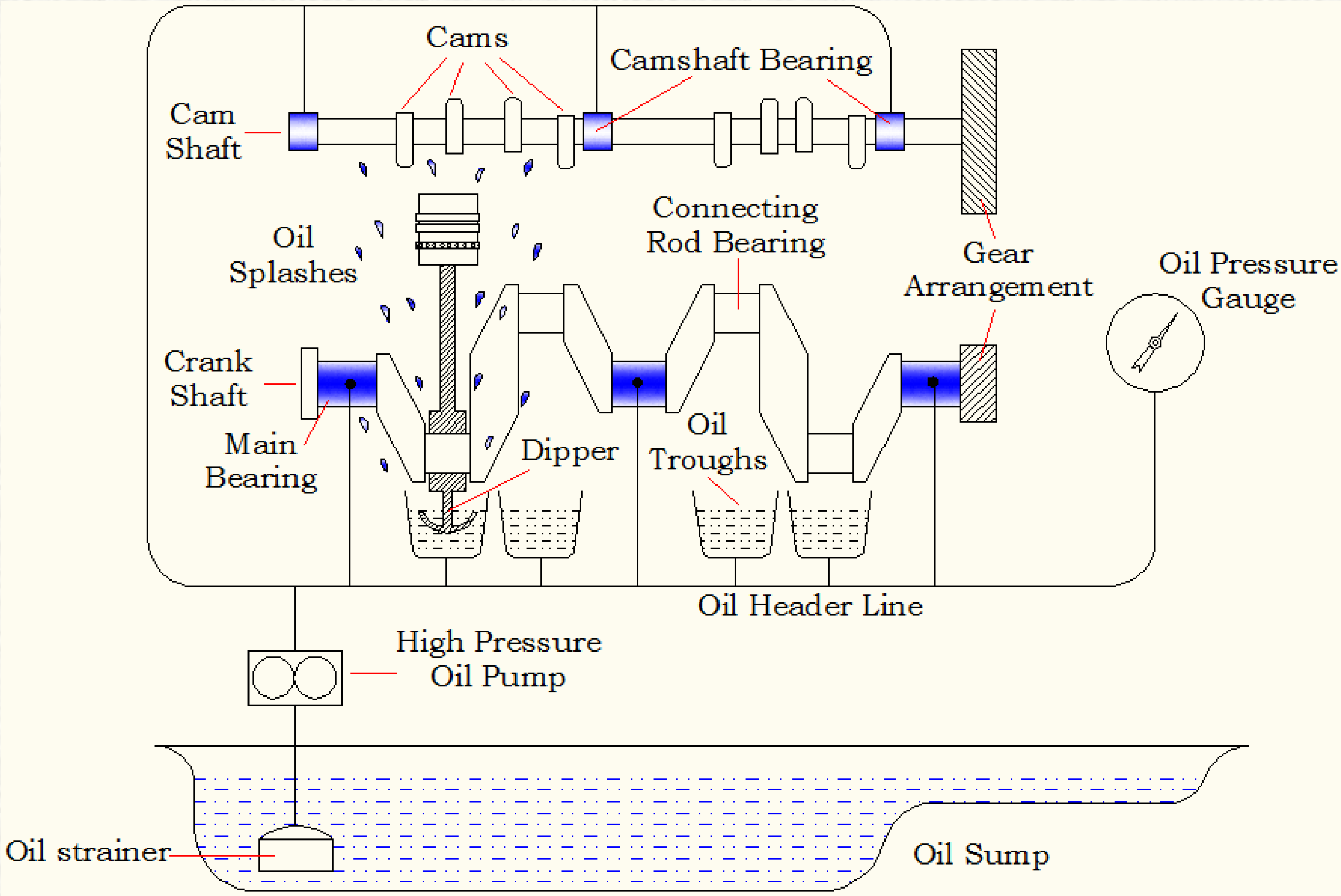
The cylinder walls, rocker arm tappet rollers, piston rings and cylinder walls are lubricated by oil spray from around the piston pins and the main and connecting rod bearings.



3.2.3 Splash and Pressure Lubrication System – SPPU : Dec.-17, 6-Marks

This lubrication system is used for medium speed stationary engines, it is a combination of the splash lubrication and pressure lubrication systems.

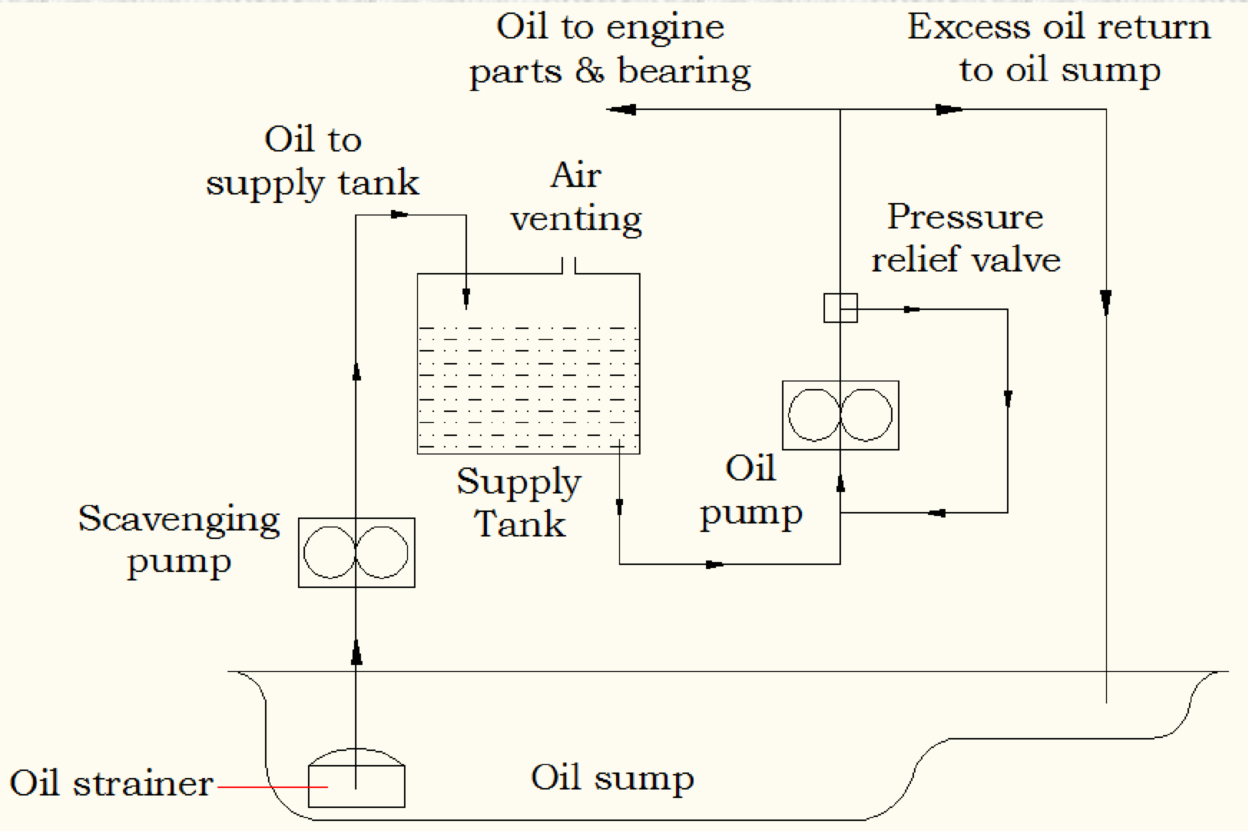
It is more simple and less expensive to install than the complete pressure lubrication system. In this system, oil is drawn from the sump by means of pump and then delivered to the crankshaft main bearings and camshaft bearings. The big end bearings of the connecting rods are lubricated by the splash system using dippers and through slots cut in the lower ends of the connecting rods. The other parts of the engine are lubricated by splash or spray of oil thrown up by dippers.



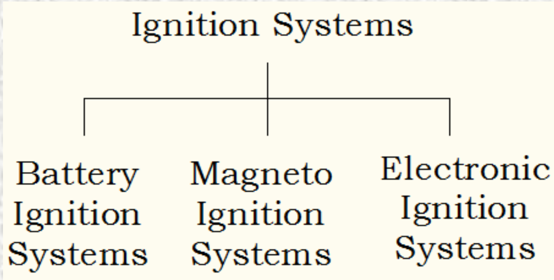
3.3 Dry Sump Lubrication System – SPPU : May-15, 6-Marks

In this system, the supply of lubricating oil is carried in an external tank. An oil pump draws oil from the supply tank and circulates it under pressure to the various bearings of the engine. Oil dripping from the cylinders and bearings into the sump is removed by a scavenging pump which is fed back to the supply tank.

The capacity of scavenging pump is always greater than the oil pump this prevents oil from accumulating in the oil sump, so it remains dry, hence called as dry sump lubrication system.


Oil to engine parts & bearing
Excess oil return to oil sump
Oil to supply tank
Air venting
Pressure relief valve
Supply Tank
Oil pump
Scavenging pump
Oil strainer
Oil sump

4. Ignition Systems :



4.1 Battery Ignition System –
SPPU : Dec.-17, May-17, May-16, May-15, Dec.-15, 6-Marks

Most of the modern spark-ignition engine use battery ignition system. The required components of the system are, battery of 6 volt or 12 volt, induction coil, contact breaker, condenser, distributor and spark plug.

Battery is connected to the primary of the induction coil through starting switch, other end of the primary coil is connected to the breaker and through it to the ground. When breaker contact points are closed, as one terminal of the battery is grounded, the circuit is closed by passing the current from the battery - starting switch - primary coil - contact breaker - ground and back to the battery.

Thus the primary circuit is closed, a current is flow through the primary coil and magnetize core of the coil. The emf is induced in the secondary but this is not sufficient to produce a spark at the spark plug.

When primary circuit is opend due to separation of contact breaker points, the magnetic field collapses, the emf induced in the secondary coil, which is directly proportional to the rate at which the magnetic field collapse. A capacitor is connected across the contact breaker helps to collapse the field very rapidly by absorbing part of the energy of the magnetic field which is thrown back into the primary winding and produces a very high voltage in the secondary. This emf in the secondary coil is sufficient to ignite the charge is supply to respective spark plug with the help of distributor.

Ballast resistor is made of iron wire, it having property that its electrical resistance increases very rapidly if a certain temperature is exceed. When engine runs for a long time at low speed then induction coil get overheated, thus as ballast coil exceed its temperature, its resistance increases very rapidly and the primary circuit current reduces to safe value. In cold condition ballast resistance not come in to play hence more current flow in the primary circuit.

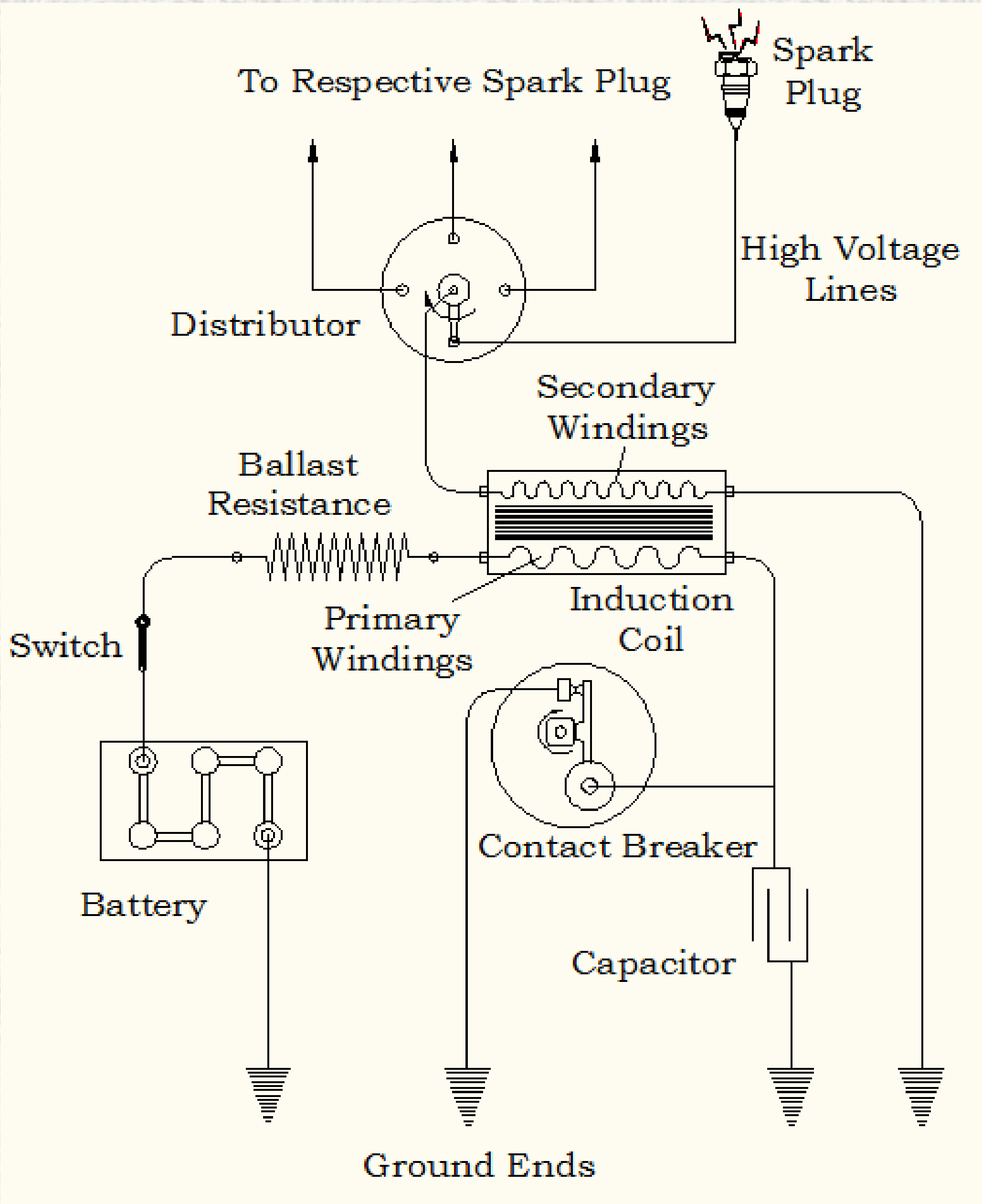
Advantages of Battery Ignition Systems –

1. Its initial cost is low compared with magneto.
2. It provides better sparks at low speeds of the engine during starting and idling.
3. The maintenance cost is negligible except the battery.

4. The spark efficiency remains unaffected by advance and retard positions of the timing control mechanism.
5. The simplicity of the distributor drive.

Disadvantages of Battery Ignition Systems –

1. The engine con not be started if the battery runs down.
2. The weight of the battery ignition system is greater than magneto.
3. The wiring involved in the coil ignition is more complicated than that used in magneto, hence it more likelihood of defects occurring in the system.
4. The sparking voltage drops with increasing speed of the engine.

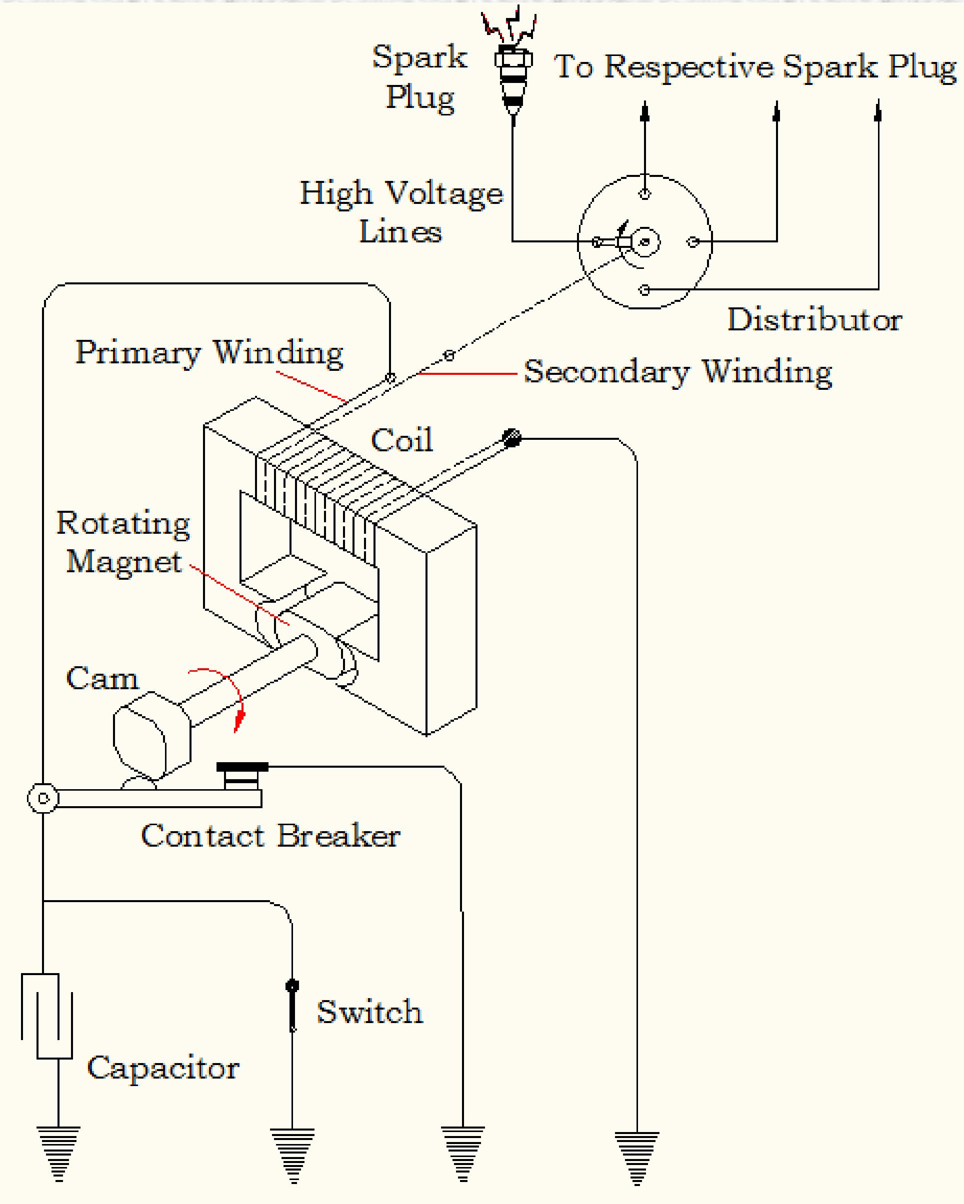


4.2 Magneto Ignition System – SPPU : May–18, 7-Marks

Magneto ignition system, has its own electric generator to provide the necessary energy for the system., and thus it does not required battery, also it replaces ignition coil.

The figure shows rotating magnet type magneto ignition system, in this a rotating magnet attached to the cam which controls the opening and closing of the contact breaker. The primary and secondary winding are kept stationary, one end of both are grounded and other end of primary winding is grounded through the contact breaker, a capacity is attached in parallel to the contact breaker.

The rotating magnet is rotated by engine, thus it producing a voltage in primary winding, the circuit is completed through contact breaker, during this capacitor get charged. A current is flow through the primary coil and magnetize core. The emf is induced in the secondary but this is not sufficient to produce a spark at the spark plug.



When primary circuit is opend due to separation of contact breaker points, the magnetic field collapses, the emf induced in the secondary coil, which is directly proportional to the rate at which the magnetic field collapse. A capacitor is connected across the contact breaker helps to collapse the field very rapidly by absorbing part of the energy of the magnetic field which is thrown back into the primary winding and produces a very high voltage in the secondary. This emf in the secondary coil is sufficient to ignite the charge is supply to respective spark plug with the help of distributor.

Advantages of Magneto Ignition Systems –

1. It does not required battery and ignition coil, hence corresponding cost is reduced.

2. With the increase in speed the current generated also increased, thus it gives strong spark, therefore it is used in racing cars and aero-plane.
3. It having light weight, occupies less space and less maintenance, thus it is favored in two wheelers.
4. It is more reliable as the complicated ignition coil and battery run down problems are not there.

Disadvantages of Magneto Ignition Systems –

1. Since wiring carry high voltage current, there is a strong possibility of leakage which may cause.
2. This system required extensive shielding to prevent leakage of high voltage current.
3. At low speed, it develops poor quality of spark at the time of starting, thus some time separate battery is needed for starting.

4.3 Comparison between Battery Ignition and Magneto Ignition system – SPPU : May-14, 6-Marks

Battery Ignition System	Magneto Ignition System
It needed external power source of 6 or 12 volt battery and ignition coil.	It produces its own power thus does not required external battery, also replace ignition coil.
Current for primary circuit is obtained from the battery.	Current for primary circuit is generated by the magneto.
In this system, intensity of spark is inversely proportional to the engine speed, hence not preferred in high speed engines.	In this system, intensity of spark is directly proportional to the engine speed, hence preferred in high speed engines.
This system is less reliable as compared to magneto ignition system.	This is more reliable as compared to battery ignition system.
This system is heavy in weight, also occupies more space, hence not preferred in two wheelers.	This system is light in weight, and occupies less space, hence preferred in two wheelers.
It needed excessive maintenance of battery, thus maintenance cost is more.	It does not have battery, thus maintenance cost is less.
It is used in low and medium speed vehicles like cars, buses, trucks etc.	It is used in high speed vehicles like racing cars, aero-planes etc.

5. Governing Systems : SPPU : May-16, 6-Marks

Governing system of the engine is to keep it operating at a certain speed irrespective of the load conditions. If the engine load decreases, the speed of engine will increases (unless reduction done in fuel supply). As engine speed increases the centrifugal force on the rotating weights of the governor also increases and moves up the control sleeve, this motion transfer to the fuel metering mechanism to reduce the fuel supply, thereby the speed of an engine brought to the rated value.

On the other side, if the engine load increases, it slow down the speed (unless increase in fuel supply) as a result of fuel supply is not sufficient according to the increased engine load. As the engine speed decreases, the centrifugal force on the rotating weights on the governor will also decreases and it moves down the control sleeve, this motion transfer to the fuel metering mechanism to increase the fuel supply, thereby the speed of an engine brought to the rated value.

Thus the function of governor is to control the fluctuation of speed of the engine within the certain prescribed limit according to the variation of engine load from no load condition to maximum load condition.

1.1 Types of Governing / Methods of Governing : SPPU : May–18, 7-Marks

There are following types of IC engine governing used as,

1. Hit and Miss Governing – In this method fuel supply is completely cutting off for one or more cycles, to bring down the speed on the engine to rated value.
2. Quality Governing – in this method, by varying the supply of fuel to the cylinder per cycle the quality of the air-fuel mixture (rich mixture to lean mixture) is changes according to the engine requirement.
3. Quantity Governing – in this method, the supply of air as well as fuel are varied keeping the ratio of air-fuel constant, thus the quantity of the mixture supplied is varied to keep engine speed within the rated limit.
4. Combination Governing – in this type, the quality and quantity methods are combined to control the engine speed within the rated limit.

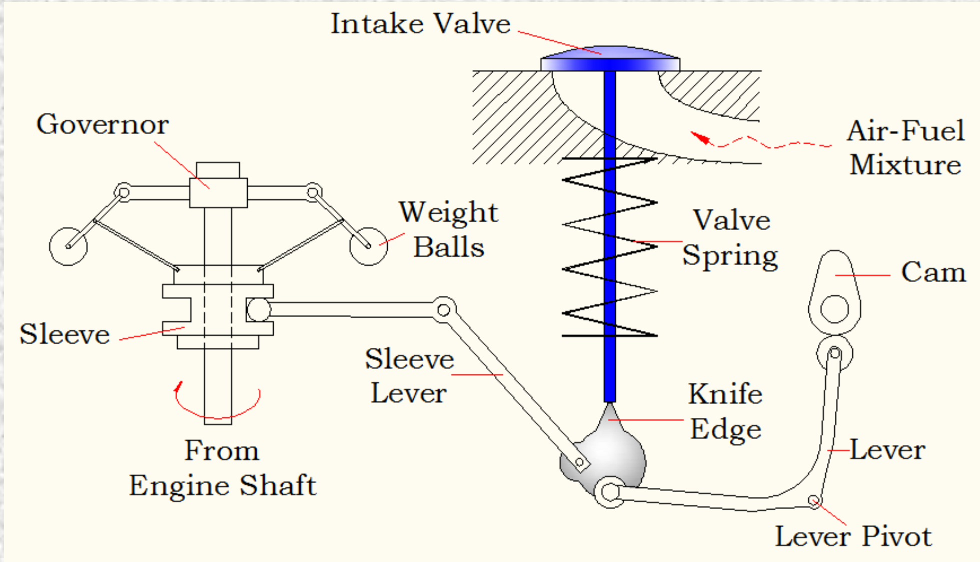
5.1.1. Hit and Miss Governing :

In the hit and miss governing system, when engine speed exceed the rated value then, some cylinder explosion / power stroke is skipped thus the overall power developed is reduced so that engine speed brought back to its rated speed limit and once it bring back to rated limits the skipped cylinder made active again.

SI engines - as shown in figure, when engine working in rated speed then the governor weight balls does not experience much centrifugal force and they can not pull up sleeve above the excess speed limit.

When engine speed increases and extends the rated speed, this causes the weight balls to fly out by greater centrifugal force to that extend which lifted sleeve above the excess speed limit. At this point the sleeve lever pull out the knife edge from the valve such that even cam actuating knife edge with the help of lever but it is out of the range to push the valve to supply the air-fuel ratio. Thus without charge this particular cylinder become inactive and misses the power. Overall power developed is reduced and speed bring down.

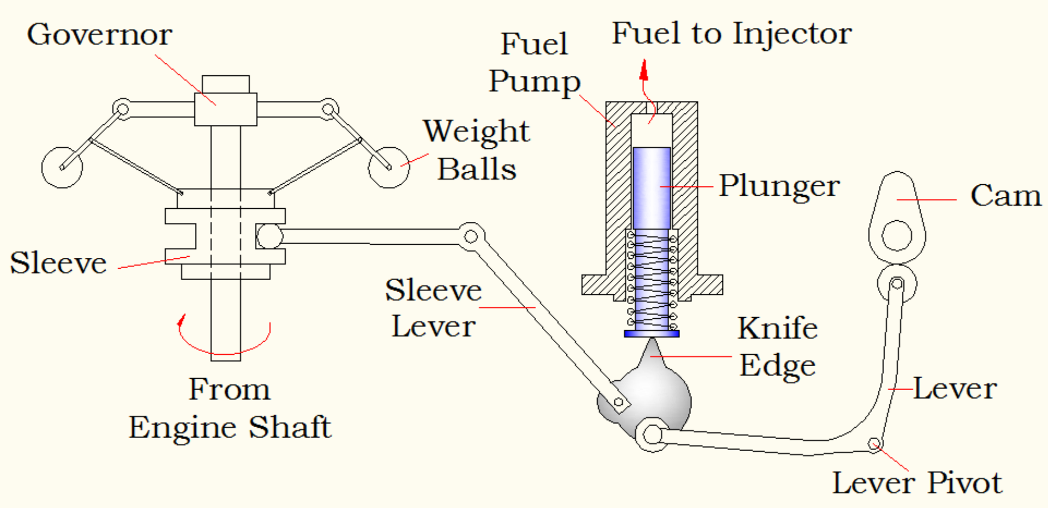
Once engine speed reach to the rated speed level the fly balls come down due to lesser centrifugal force this bring back the knife edge in the reach of valve and that cylinder again hit the power i.e. it become active by continuing supply of charge again.



CI engines - as shown in figure, the governing mechanism almost remain same only the difference is, the knife edge actuation is given to the fuel pump plunger. when engine working in rated speed then the governor weight balls does not experience much centrifugal force and they cannot pull up sleeve above the excess speed limit.

When engine speed increases and extends the rated speed, these causes the weight balls to fly out by greater centrifugal force to that extend which lifted sleeve above the excess speed limit. At this point the sleeve lever pull out the knife edge from the plunger such that even cam actuating knife edge with the help of lever but it is out of the range to push the plunger to supply the fuel. Thus without fuel this particular cylinder become inactive and misses the power. Overall power developed is reduced and speed bring down.

Once engine speed reach to the rated speed level the fly balls come down due to lesser centrifugal force this bring back the knife edge in the reach of valve and that cylinder again hit the power i.e. it become active by continuing supply of fuel again.



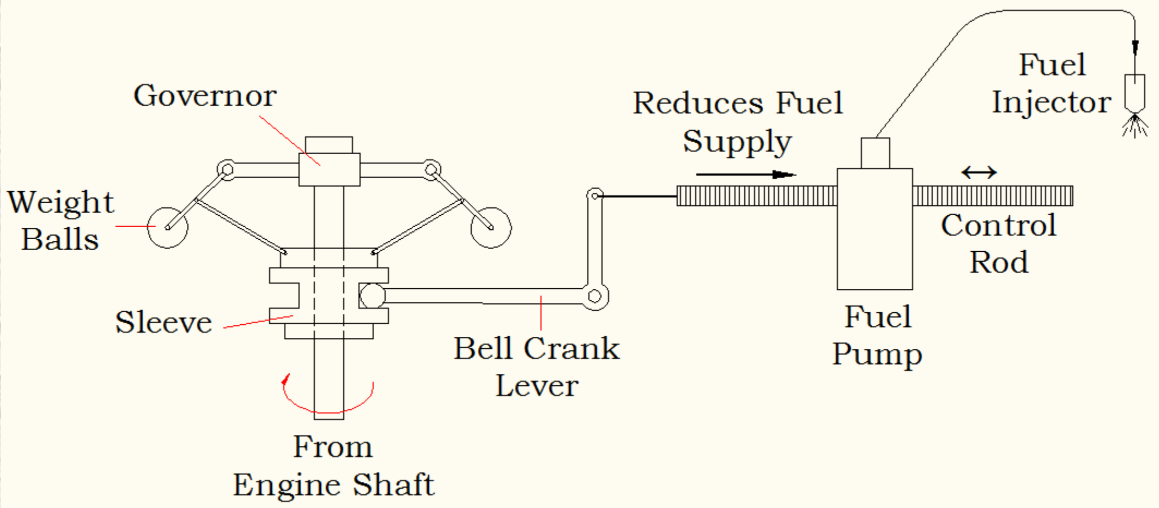
5.1.2. Quality Governing :

Quality governing uses the method to control the fuel supply to cylinder while air supply kept constant, the example of this type is Hartwell Governor. Thus the air-fuel mixture quality is controlled to keep the engine speed with in the rated speed limit, hence this method is called as Quality Governing Method.

As shown in figure given below, the governor receive the rotational power from crankshaft, thus when engine rotates the governor spindle along with weight balls are rotated. The centrifugal force acting on the weight balls accordingly it may fly outward or inward. Engine running within the normal rated speed, the centrifugal force acting on governor weight balls are not enough to lift the sleeve beyond the excess speed limit, thus fuel supply to cylinder does not affect.

When engine speed exceed the rated speed, the centrifugal force acting on weight balls causes it to fly out to such level that sleeve lifted up and this motion transfer to the control rod through bell crank lever and the control rod movement then reduces the plunger stroke in turn reduces the fuel supply. The reduced fuel supply result in lean mixture and production of less power bring speed back to rated speed limit.

Once engine speed reach to the rated speed level the weight balls come down due to lesser centrifugal force this bring back the control rod to its normal working range so that the cylinder now start receiving regular fuel supply again.



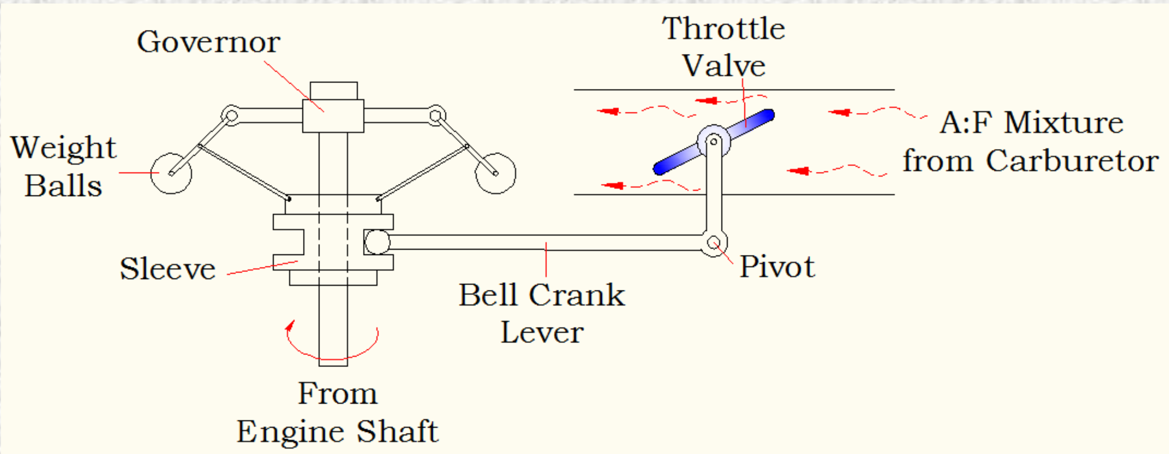
5.1.3.Quantity Governing :

Quantity governing may be accomplished by varying the quantity of air-fuel mixture entering the cylinder, while the quality of air-fuel mixture keeps constant. It is applied to petrol engines by having a throttle valve in the pipe leading from the carburetor to the engine cylinder.

In this method, a throttle valve is connected to the governor with the help of bell crank lever, when engine speed exceed the rated limit, then the weight balls fly out on account of the higher centrifugal force and this causes lift of the sleeve to such level where bell crank lever actuate the throttle valve and rotate it to closing position. Thus the total quantity of the air-fuel supplied to engine reduces and thus intern engine speed reduces.

Once engine speed reach to the rated speed level the weight balls come down due to lesser centrifugal force this bring back the throttle valve to its normal working range so that the cylinder now start receiving regular air-fuel supply again.

This type of quantity governors are used in static petrol and gas engines, in case of automobiles the throttle valve is either hand operated or accelerator pedal operated.



5.1.4. Combination Governing -

The governing of an engine may be obtained by combining two or more of the above methods. For instance, quality or quantity governing at high loads has been successfully combined with hit and miss governing at low loads. Also quality governing at high loads is used with quantity governing at low loads. The latter system is economical and gives close governing.

6. Starting Systems :

Engine starting from its initial cold state condition required externally to supply some power to move the piston inside the cylinder to initiate the suction stroke, compression stroke and when the charge is able to ignite then engine actually started repeating the stroke and cycle continuous at its own.

The following methods are used to start the engine from its cold dead state,

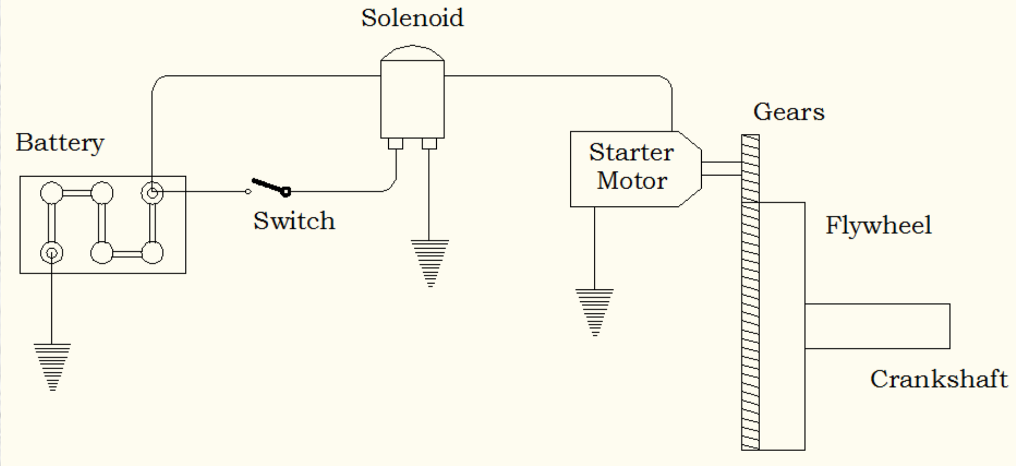
6.1. Hand or Kick Starting -

Hand starting is used in old days of the evolution of the engine starting, in this type a crank is provided which engaged to the crankshaft and rotated with the help of hand till engine ignition started. Even in some stationary or small engine a rope is wounded on the crankshaft pulley and pulled out which gives the motion to the crankshaft to move piston inside the cylinder.

Kick starting is used in small SI engine or motor cycles, in this kick motion is cause the crank to rotate which intern moves piston inside the cylinder to complete the required suction, compression processes before ignition of the charge.

6.2. Electrical Starting - SPPU : May-14, 6-Marks

Electrical starting consists of the battery, starting switch, starting motor. The starting motor shaft gear engaged with the teeth wheel of flywheel. When starting switch is On a battery supplies the energy to the starting motor. The starting motor are series wound motors designed to operate on large currents at low voltages, and thus it supplied the torque to the flywheel. The flywheel is mounted on the crankshaft thus, in tern power is supplied to the piston to move inside the cylinder till the engine start after completing necessary suction, compression, ignition and exhaust processes and the cycle is repeated by its own.



These system are used in all moderate and heavy engines, especially in all automobiles. It become very easy to start the engine, but the drawback of the system is, it become inoperative if battery goes on discharge.

6.3. Compressed Air Starting -

In case of multi-cylinder internal combustion engines, especially in diesel engine, all cylinders are cut-off except for one cylinder. Now the engine power developed is used to run the other cylinder which suck air and compressed, thus all those are working as air compressor and compressed air stored into a receiver.

When engine needed to start from dead condition, the stored compressed air is supplied to some cylinder which are cut-off, thus those cylinder use to crank the engine and the cylinder where the fuel supply is not cut-off, the power is generated and once engine started, all cylinder are made active and thus engine get started.

6.4. Auxiliary Petrol Starting -

In auxiliary petrol starting system, instead of using a motor and battery as in case of electrical starting, a small capacity petrol engine is used to crank the diesel engine. The petrol engine is coupled to diesel engines through reduction gears, the petrol engine is usually started by cranking. When diesel engine picks up adequate speed and it starts firing, the drive disengages the petrol engine and it is subsequently shut down.

6.5. Hot Bulb Ignition Starting -

In this system a chamber of bulb shape is attached to the cylinder head and it is unjacketed for cooling. It is heated by a blow lamp before staring the engine and the fuel is injected into the hot combustion chamber at the end of the compression stroke and ignition takes place partly due to heat of the compressed charge of air and partly due to heat of the hot bulb. The blow lamp is removed after the engine takes up its speed. The ignition then goes on due to the combined effect of compression heat and the heat retained by the combustion chamber from the previous cycle.

This method of ignition is used in semi-Diesel engines, where heavy oils can be successful dealt with. The system is also known as surface ignition or hot combustion chamber ignition.

7. IC Engine Emissions : SPPU : Dec.-17, May-17, 7-Marks

7.1. Air pollution due to IC engines –

Air pollution can be defined as addition to our atmosphere of any material which will have a deleterious effect on life upon our planet. The atmospheric air is made up of 78% of nitrogen and 21% of oxygen by volume and remaining other gases and particulates like argon, carbon-dioxide, dust particles etc.

Air is required for breathing and contents of pure air as such are healthy. However, certain toxic gases and substances which are emitted by various machinery, plants, equipments etc. and mixed with atmosphere air are very harmful to human beings, animals and plants. These undesirable gases and substances in the air are called air pollutants.

The main pollutants contributes by automobiles are carbon monoxide, unburned hydrocarbons, oxides of nitrogen, and lead and other particulate emissions. The pollutants from one car do not amount too much, but if we consider the very large number of cars, which number rising very rapidly, then the pollutants amount becomes millions of tones.

IC engines generates undesirable emissions during the combustion process. The emissions exhausted into the surroundings pollute the atmosphere and causes the problems, like global warming, acid rain, smog, odours, respiratory and other health hazards. The major causes of these emissions are non-stoichiometric combustion, dissociation of nitrogen, and impurities in the fuel and air.

7.2. Air pollutants and their harmful effects –

Carbon Monoxide (CO) – It is produced by incomplete combustion of fuel, generally caused by insufficient oxygen. It causes headache, nausea and breathing problems.

This gas has strong affinity to combine with hemoglobin in the blood and reduces its oxygen carrying capacity to body tissues.

Hydrocarbons (HC) – Hydrocarbons is unburnt fuel escaping to atmosphere. The major source of HC emissions are the engines of automobiles, aircrafts, ships, locomotives and oil refineries.

This gas causes irritation in respiratory system.

Oxides of nitrogen (NOx) – There are several oxides of nitrogen's like, NO, NO_2, N_2O, N_2O_3 etc. commonly called as oxides of nitrogen (NOx). It is produced when N_2 and O_2 combine at higher temperature above 1100°C.

These gases irritates the eyes, nose and throat and it causes coughing, headache and damage to lungs. NO_2 is most poisonous with penetrating odour which can destroy lung's tissues.

Soot – Soot are solid particles of pure carbon. They get suspended in the air and are breathed in by humans and animals. Their size is few microns. The maximum amount of soot is produced at full load by diesel engines.

Soot is very dangerous to health as it produces lung cancer.

Aldehydes – Aldehydes are formed in diesel engines mostly at idling speed due to cold flames.

This cause unpleasant smell due to pungent, odour from diesel fuel and causes irritation to eyes and nose.

SO_2, H_2S and Lead – Toxic gas SO2 and bad smell of gas H2s are formed on burning of sulphur present in petroleum products.

Poisonous lead compounds are also released to atmosphere if TEL and TML are added in petrol as an additive or dopes.

Particulate Matters – Particulates are ash, carbon and liquid like sulphuric acid vapors. These are heavily produced on burning of diesel fuel. Size of particulate matters is generally less than 100 μm and they remain suspended in atmospheric air for long time.

Particulate matters breathed in respiratory system causes bronchitis and cardiac diseases.

7.3. Effect of air pollution on Human Health and Environment :

Respiratory and Heart Problems – Air pollution causes various respiratory diseases and heart problems, these results in increase in the death toll. There have been instances where several respiratory and heart conditions have developed, children are observed to contract pneumonia and asthma due to increased exposure to air pollutants.

Depletion of Ozone Layer – Air pollution is responsible for the depletion of the ozone layer, ozone is present in the stratosphere and protects humans from dreadful ultraviolet rays. This ozone layer has been depleting due to chlorofluorocarbons and hydro-chlorofluorocarbons in the air. Due to this thinning of the ozone layer harmful ray getting through and causing eye and skin related problems and also it affect the agricultural crops and food chain system.

Acid Rain – The suspended air pollutant particles and gases like nitrous oxides and sulphur oxides get converted into harmful form of nitric acid or sulphuric acid. These acids causes acid rain and have a very large potential to harm the natural resources, all living things, crops etc. Thus acid rain affect the living as well as non-living substances, erodes the buildings, bridges materials, buses, trains, it kills plants and destroy agricultural crops and other living creatures.

Eutrophication – It is a condition which is caused when there is existence of high amount of nitrogen present in pollutants enters the water bodies and leads to the development of algae. The development of algae hampers the conditions where fishes can live leading to the death of marine plants and also the various animal species existing in these water bodies. The presence of the algae is a sign of low oxygen level in the water as the algae uses pretty much all of the available oxygen in the water body.

Effects on Wildlife – Air pollution not just affect the human beings but animal also affect by it. Toxic chemicals that exist in the atmosphere due to pollution, force the wildlife to migrate to new places and search for more adaptable habitats. Pollutants that deposit in the water bodies and seas adversely affect the marine and aquatic animals as well.

Global Warming - Increased air pollution is directly related to the cause of global warming which is increasing the temperature worldwide. The rising sea levels, thermal expansion of sea water, melting ice bergs, displacement of living beings from their habitats and even loss of habitats with serious signals of impending disasters.

7.4. Air pollution due SI engine :

Emission of pollutants from SI engine through exhaust gases –

When the sufficient amount of oxygen is available then there will be an ideal combustion take place and all carbon particles are converted into CO2 and all hydrogen converted into water vapors, but in actual this is not happened and always having some other productions of combustions present in the exhaust gases like,

Carbon Monoxide (CO) emission – Its presence shows that there was deficiency of oxygen which results in incomplete combustion causes some carbon get converted to CO2 and some converted to CO. Also some time due to dissociation of CO2 at high temperature take place which added CO in exhaust gases.

Hydro-Carbon (HC) emission – Unburnt HC emissions in the exhaust gases are because of the incomplete combustion, also combustion chamber geometry and engine operating parameters also influence the HC emissions. In addition to these, there are other reasons behind the HC emissions are as, crevice volumes and flow in crevices, leakage past the exhaust valve, valve overlap, deposits on walls, oil on combustion chamber walls etc.

Oxides of Nitrogen (NOx) emission – Oxides of nitrogen (NOx) are produced at very high temperature and presence of excess oxygen. Oxygen and Nitrogen of atmospheric air combine inside the combustion chamber when temperature exceeds 1000 °C. At low temperature during expansion and exhaust stokes the nitric oxides is dissociated in N_2 and O_2, this reduces NOx emissions considerably.

Emission of pollutants from SI engine through evaporation of gasoline –

When the gasoline evaporated and this entered into the atmosphere, the pollution thus added is known as emission of pollutants due to evaporation of gasoline.

Fuel tank evaporation emissions – Gasoline (HC) entered into the atmosphere from tank, most commonly when filling the gasoline into it, also some part is evaporated through air vent of the tank cap. This leakages or evaporation increases considerably in the summer seasons due to higher temperature.

Carburetor evaporation emission – In carburetor there is a air vent provided to float chamber through which gasoline evaporated part get escape and entered into the atmosphere. Also from the overflow port of the carburetor some evaporation leak off into the atmosphere.

Emission of pollutants from SI engine through crankcase blowby –

When the piston rigs fails to seal the combustion pressure pass through it, i.e. when cylinder wall wear out and charge pass through it and enter into the crankcase is known as blowby. Unburnt gasoline thus enter into crankcase from where entered into atmosphere through air venting to crankcase.

7.5. Air pollution due CI engine : SPPU : May-17, May-15, 7-Marks

The major components of air pollution from CI engine are same as in the SI engines, but the concentration of pollutant are varies as,

Carbon Monoxide (CO) emission – Its emissions is mainly due to deficiency of oxygen during combustion and also due to dissociation of CO2. But in CI engine the concentration of CO emission is at high temperature take place which added CO in exhaust gases lesser than that of the SI engine.

Hydro-Carbon (HC) emission – CI engines HC emission are 1/5 of that of the SI engine but having higher soot particle emissions, as it operates with overall lean air-fuel ratio. Also the components in diesel fuel have higher molecular weights than gasoline, this results higher boiling and condensing temperatures. Therefore, soot formation in more in CI engines.

Oxides of Nitrogen (NOx) emission – Oxides of nitrogen (NOx) emission level in CI engine is higher than that of SI engine. These are produced at very high temperature and presence of excess oxygen. Both these conditions are most favorably satisfied in CI engines, as due to higher compression ratio the overall pressure and temperature is always higher in CI engine and excess oxygen is always available surrounded to fuel due to heterogeneous mixture of air-fuel.

Aldehydes and other emission – Aldehydes are more pronounced in diesel engines. Aldehydes in diesel fuel has pungent odor. H_2S and SO_2 gases are formed on burning of sulphur present in the fuel. Also ash dust, H_2SO_4 etc. are also produced in the CI engine exhaust emissions.

Smoke and Particulate emission – Smoke means the visible product of the combustion which is due to poor combustion of fuel which mostly contain the CO, Soot and HC emissions. The color of smoke is white mainly due to liquid droplets of lubricating and fuel oil and its appears in exhaust under cold starting, idling and low load condition run, also when piston rings are wear out, the smoke appearance become white. Black smoke in exhaust gases is due to incomplete combustion of fuel, the blackness of smoke is increases with increase in load on the engine.

8. Emission Norms / Standards :

Emission standards are the legal requirements governing air pollutants released into the atmosphere. Emission standards set quantitative limits on the permissible amount of specific air pollutants that may be released from specific sources over specific timeframes. They are generally designed to achieve air quality standards and to protect human life.

9.1 Euro norms :

Emission norms are prescribed CO, HC and NOx levels set by the government which a vehicle would emit when running on roads. All the manufacturers need to implement the same for vehicles being manufactured from the date of implementation.

EURO norms refer to the permissible emission levels from both petrol and diesel vehicles, which have been implemented in Europe and European Countries.

Euro Norms	Operational Year	Vehicle Types
EURO-I	1993	Passenger Cars
EURO-II	1996	Passenger Cars
EURO-III	2000	Any Vehicles
EURO-IV	2005	Any Vehicles
EURO-V	2008	Light Passenger And Commercial Vehicles
EURO-VI	2015	Light Passenger And Commercial Vehicles

Specification of Euro Norms for Petrol Engine Vehicles,

Euro Norms	CO (gm/KW-hr)	HC (gm/KW-hr)	HC+NOx (gm/KW-hr)	NOx (gm/KW-hr)
EURO-I	2.72	----	0.97	----
EURO-II	2.2	----	0.5	----
EURO-III	1.3	0.2	----	0.15
EURO-IV	1.0	0.1	----	0.08

Specification of Euro Norms for Diesel Engine Vehicles,

Euro Norms	CO (gm/KW-hr)	HC (gm/KW-hr)	NOx (gm/KW-hr)	PM (gm/KW-hr)	Smoke (m^{-1})
EURO-I	4.5	1.1	8.0	0.36	----
EURO-II	4.0	1.1	7.0	0.25	----
EURO-III	2.1	0.66	5.0	0.15	0.8
EURO-IV	1.5	0.46	3.5	0.02	0.5
EURO-V	1.5	0.46	2.0	0.02	0.5
EURO-VI	1.5	0.13	0.4	0.01	----

9.2 Indian (Bharat) Stage Norms :

SPPU : May-17, May-16, 7-Marks

Bharat stage emission standards (BSES) are emission standards instituted by the Government of India to regulate the output of air pollutants from internal combustion engines and Spark-ignition engines equipment, including motor vehicles. The standards and the timeline for implementation are set by the Central Pollution Control Board under the Ministry of Environment & Forests and climate change.

The standards, based on European regulations were first introduced in 2000. Progressively stringent norms have been rolled out since then. All new vehicles manufactured after the implementation of the norms have to be compliant with the regulations. Since October 2010, Bharat Stage (BS) III norms have been enforced across the country. In 13 major cities, Bharat Stage IV emission norms have been in place since April 2010 and it has been enforced for entire country since April 2017. In 2016, the Indian government announced that the country would skip the BS-V norms altogether and adopt BS-VI norms by 2020.

Indian / Bharat Stage Norms for four wheeler vehicles,

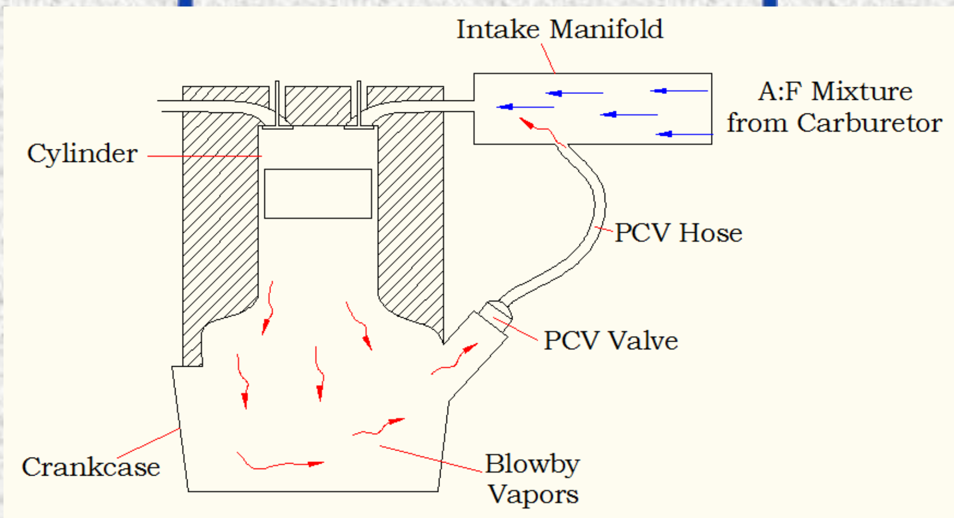
Indian Norms	Reference of Euro Norms	Operational Year	Region
India 2000	Euro-I	2000	Nationwide
Bharat Stage - II	Euro-II	2005	Nationwide
Bharat Stage - III	Euro-III	2010	Nationwide
Bharat Stage - IV	Euro-IV	2017	Nationwide
Bharat Stage - V	Euro-V	--- To be skipped ---	
Bharat Stage - VI	Euro-VI	2018	Delhi and National Capital Region
		2020	Nationwide (Proposed)

10 Emission Control Methods for SI and CI Engines : SPPU : Dec.-17, 7-Marks

The main objective of the emission control methods is to minimize the release of pollutants to atmosphere form IC engines. There are two approaches as, change or modification in engine design – to reduce or control those parameter which promote the production of pollutants like compression ratio, combustion chamber, valve timing, fuel supply system etc. and other treatment of exhaust gases – conversion of harmful air pollutants into un-harmful substances before they are released into the atmosphere.

10.1 Positive Crankcase Ventilation (PCV) System – SPPU : May–18, 6-Marks

It is a system that was developed to remove harmful vapors from the engine and to prevent those vapors from being expelled into the atmosphere. The PCV system does this by using manifold vacuum to draw vapors from the crankcase into the intake manifold.



Positive crankcase ventilation involves recycling these gases through a PCV valve to the intake manifold, from where pumped back into the cylinders for another shot at combustion. When the engine is idling the air pressure in the intake manifold is lower than the air pressure in the crankcase, and it's this lower pressure that sucks the blow-by gases through the PCV valve and back into the intake. When the engine speeds up, the air pressure in the intake manifold increases and the suction slows down, reducing the amount of blow-by gas recycled to the cylinders. Since the whole point of positive crankcase ventilation is to keep these gases out of the crankcase, the PCV valve is designed to close off when this happens and block the backflow of gases.

10.2 Catalytic Converter System –

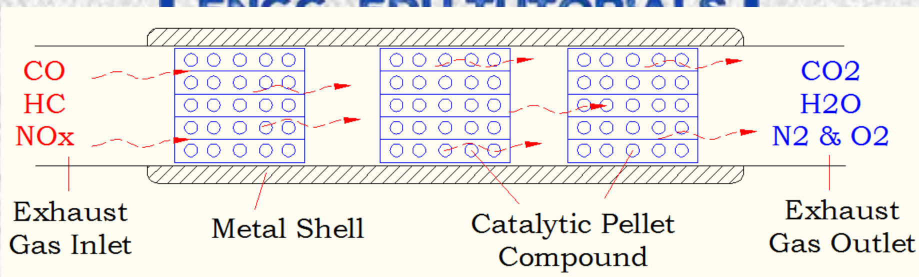
A catalytic converter is an exhaust emission control device that converts toxic gases and pollutants in exhaust gas from an internal combustion engine into less-toxic pollutants by catalyzing a redox reaction (an oxidation and a reduction reaction). Catalytic converters are usually used with internal combustion engines fueled by either gasoline or diesel.

Inside the converter, the gases flow through a dense honeycomb structure made from a ceramic and coated with the catalysts. The honeycomb structure means the gases touch a bigger area of catalyst at once, so they are converted more quickly and efficiently.

Typically, there are two different catalysts in a catalytic converter:

One of them tackles nitrogen oxide pollution using a chemical process called reduction (removing oxygen). This breaks up nitrogen oxides into nitrogen and oxygen gases (which are harmless, because they already exist in the air around us).

The other catalyst works by an opposite chemical process called oxidation (adding oxygen) and turns carbon monoxide into carbon dioxide. Another oxidation reaction turns unburned hydrocarbons in the exhaust into carbon dioxide and water.

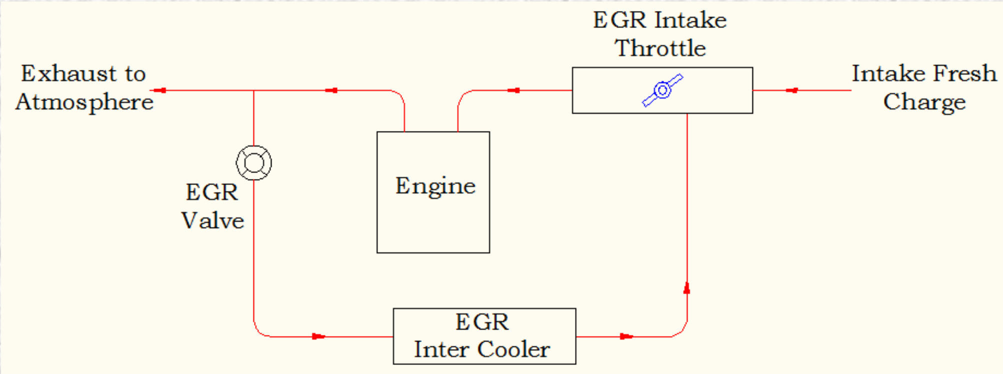


In effect, three different chemical reactions are going on at the same time. That's why we talk about three-way catalytic converters. (Some, less-effective converters carry out only the second two (oxidation) reactions, so they're called two-way catalytic converters.) After the catalyst has done its job, what emerges from the exhaust is mostly nitrogen, oxygen, carbon dioxide, and water (in the form of steam).

10.3 Exhaust Gas Recirculation (EGR) System –

SPPU : May-17, May-15, Dec.-15, May-14, 6-Marks

Exhaust Gas Recirculation (EGR) is used as a nitrogen oxide (NOx) emissions reduction technique in petrol/gasoline and diesel engines. EGR works by re-circulating a portion of exhaust gas back to the engine cylinders. This dilutes the O2 in the incoming air stream and provides gases inert to combustion to act as absorbents of combustion heat to reduce peak in-cylinder temperatures. NOx is produced in high temperature mixtures of atmospheric nitrogen and oxygen that occur in the combustion cylinder, and this usually occurs at cylinder peak pressure.



A properly operating EGR can theoretically increase the efficiency of gasoline engines via several mechanisms as,

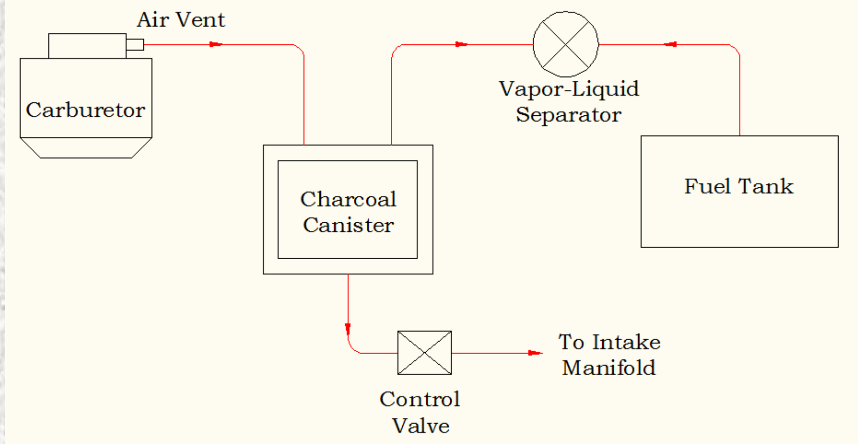
Reduced throttling losses - The addition of inert exhaust gas into the intake system means that for a given power output, the throttle plate must be opened further, resulting in increased inlet manifold pressure and reduced throttling losses.

Reduced heat rejection - Lowered peak combustion temperatures not only reduces NOx formation, it also reduces the loss of thermal energy to combustion chamber surfaces, leaving more available for conversion to mechanical work during the expansion stroke.

Reduced chemical dissociation - The lower peak temperatures result in more of the released energy remaining as sensible energy near TDC, rather than being bound up (early in the expansion stroke) in the dissociation of combustion products. This effect is minor compared to the first two.

10.4 Evaporative Emission Control System –

In evaporative emission control system, vapors from the fuel tank passed through a vapor-liquid separator, where liquid gasoline is separated and sent back to fuel tank. Residual vapors are passed to charcoal canister. Similarly some vapors from carburetor float chamber through its air vent are also supplied to charcoal canister. Charcoal canister absorbs the fuel vapors and stored them. Vapors laden air from both the fuel tank and the carburetor passes through the canister. The HC are left in the canister due to the process of adsorption and air leaves which get sucked into intake manifold.

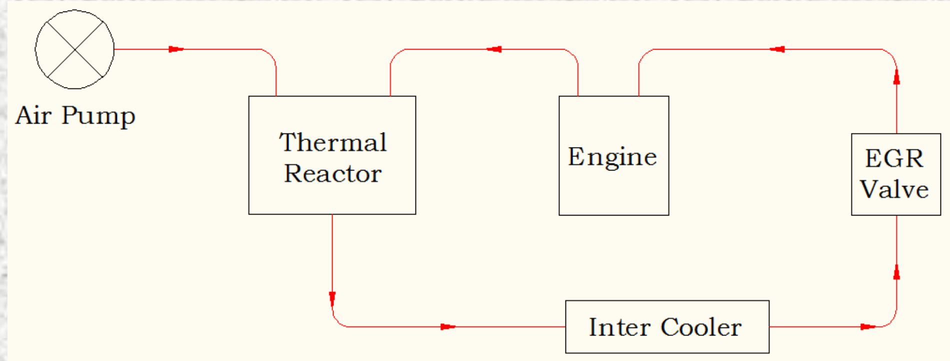


In this way, evaporative emission especially HC get arrested and clean air supplied to either engine or escape to atmosphere.

10.5 Thermal Reactor Package System –

The emission control methods discussed above either reducing NOx and tends to increase HC and CO emission and vice versa. The thermal reactor package system is the method which can reduce all the emissions NOx, HC and CO to the desirable level.

Thermal reactor is made of high nickel steel so as to withstand high temperatures to which it is exposed continuously. It consists of a heating reactor chamber in the exhaust system which provides sufficient residence time for oxidation of HC and CO to get converted into CO2 and water vapors. Thermal reactor consists of two enlarged exhaust manifolds which allows burning of HC and CO in excess amount of O2. The additional air allowed assures complete combustion, some part of this exhaust gas is re-circulated by the EGR system which reduces the formation of NOx.



Exercise

1. Explain the necessity of cooling system of an engine.
2. Classify the cooling system and explain any one with neat sketch.
3. What is evaporating cooling system.

4. What are the advantages and disadvantages of air cooling system?
5. What are the advantages and disadvantages of liquid cooling system?
6. Compare between air cooling and water cooling systems.
7. What is pressurized cooling system?
8. Differentiate between wet sump and dry sump lubrication system.
9. With neat sketch explain any one wet sump lubrication system.
10. Explain with neat sketch dry sump lubrication system.
11. Explain with neat sketch Magneto Ignition system.
12. What are different types of ignition system used in IC engine.
13. List out the various components of battery ignition system and explain its working in brief.
14. Explain the function of distributor in ignition system.
15. Explain the electronic ignition system.
16. Enlist the various types of governing systems used in IC engine.
17. With neat sketch explain the air flow meter.
18. Explain Hit and Miss governor used in Diesel engine with neat sketch.
19. Explain the function of the governing systems.
20. Explain with neat sketch quantity governing system.
21. Explain with neat sketch quality governing system.
22. Explain with neat sketch the electrical starting system.
23. Write short note on starting systems used in IC engines.
24. Explain mechanical governor with the help of neat sketch.
25. What are the different pollutants emissions from CI and SI engines.
26. Write short note on Euro Norms.
27. What are the sources of HC emissions.
28. Write short note on NOx emissions.
29. Write short note on EGR / or charge dilution system.

30. Explain Bharat Stage Norms.

31. Explain in brief use of catalytic converter to reduce the harmful emission.

32. What is crankcase ventilation?

33. Discuss the effect of air pollutants on human health.

34. What is air pollutants? Explain their harmful effects on human.

35. Enlist the Bharat stage emission norms.

36. Write short note on emission control methods.

37. How the thermal reactor package work to reduce the emission.

38. What are the various sources of air pollutants.

Unit VI : Positive Displacement Compressors (Reciprocating and Rotary)

Syllabus :

Reciprocating Compressor : *Single stage compressor – computation of work done, isothermal efficiency, effect of clearance volume, volumetric efficiency, Free air delivery, Theoretical and actual indicator diagram, Multi-staging of compressor, Computation of work done, Volumetric efficiency, Condition for maximum efficiency, Inter-cooling and after cooling, Capacity control of compressors*

Rotary Compressor : *Introduction, vane compressors, roots blower, screw compressor.* ***(Numerical treatment on Reciprocating compressor single stage and multistage only)***

1. Air Compressor :

The compressor is a device that convert mechanical work into potential energy in terms of pressure stored in compressed air. It is a work absorbing device.

It can be defined as, A compressor is a device which consumed mechanical energy and stored as potential energy in terms of increased pressure of the working fluid.

When the working fluid is an air then the compressor is known as air compressor.

2. Classification of Air Compressor :

2.1 According to the pressure ratio of compressor – the air compressor can also be classified according to the pressure ratio as,

Fan – the device which give the pressure ratio of 1.1 are classied as fan.

Blower – the device which give the pressure ratio of 1.1 to 2.5 are classied as blower.

Compressors – the device which give the pressure ratio above 2.5 are classied as compressor.

2.2 According to the principle of operation – the air compressor can be classified according to the workign principle of compression as,

Positive Displacement Compressor – these are work by forcing air into a chamber whose volume is decreased to compres the air.

Non-Positive Displacement Compressor – these are work on centrifugal action, the rotating component imparts its kinetic energy to the air which is eventually converted into pressure enerngy.

2.3 According to the relative motion of the components – the air compressor can be classified according to the working principle as,

Reciprocating Air Compressor – these are further classified as, Signle-Stage reciprocating air compressor and Multi-Stage reciprocating air compressor.

Rotary Air Compressor – these are further classified as, Rotary Screw air compressor, Rotary Vane air compressor, Scroll Compressor, Turbo Compressor, Axial Compressor.

2.4 According to the no. of stages of compression – the air compressor can be classified according to the no. of stages of compression of working fluid from initial entry pressure to final delivery pressure as,

Signle-Stage Compressor – in this working fluid is compressored in a single cylinder and in one single compression process from initial entry pressure to final delivery pressure.

Multi-Stage Compressor – in this working fluid is compressored in more than one cylinder and in more that one compression process from initial entry pressure to final delivery pressure.

2.5 According to the discharge pressure – the air compressor can be classified according to the pressure delivered as,

Low Pressure Air Compressors (LPACs) – the air compressor which have a discharge of 150 psi (i.e. 10 bar) or less.

Medium Pressure Air Compressors (MPACs) – the air compressor which have a discharge of 151 psi to 1000 psi (i.e. inbetween 10 bar to 70 bar).

High Pressure Air Compressors (HPACs) – the air compressor which have a discharge above 1000 psi (i.e. more than 70 bar).

2.6 According to the capacity of compressor – the air compressor can also be classified according to the delivery flow rate capacity of working fluid as,

Low Capacity Compressors – the air compressor which have a volume flow rate capacity upto 10 m^3/min or less.

Medium Capacity Compressors – the air compressor which have a volume flow rate capacity inbetween 10 m^3/min to 300 m^3/min.

High Capacity Compressors – the air compressor which have a volume flow rate capacity more than 300 m^3/min.

3. Applications of Compressed Air :

In industry, compressed air is so widely used that it is often regarded as the fourth utility, after electricity, natural gas and water.

Compressed air is used for many purposes as,

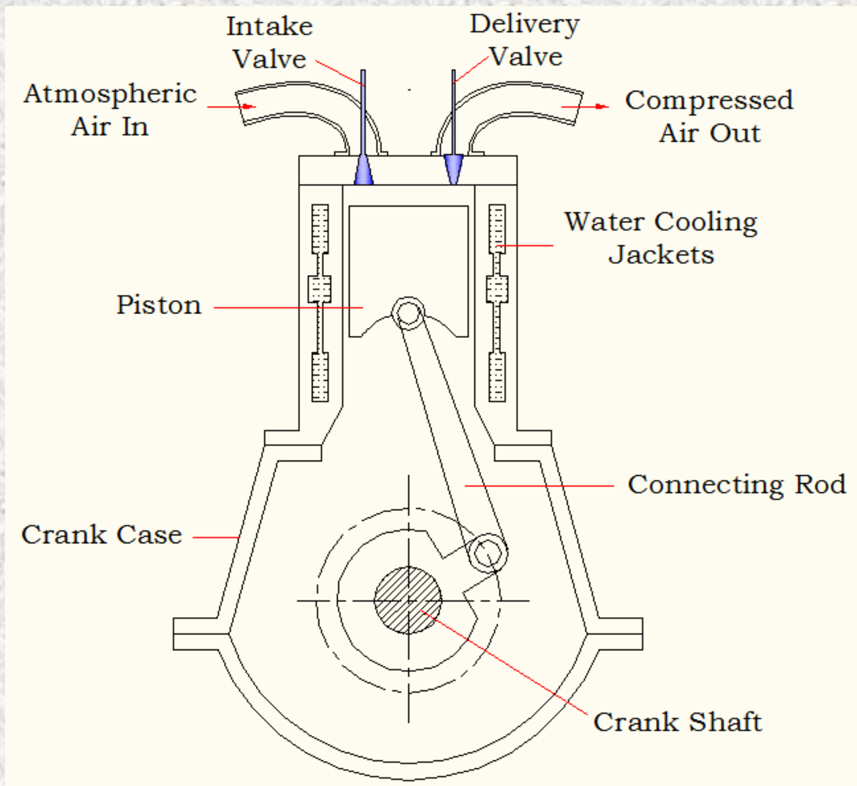
- Pneumatics
  - Pneumatic post - using capsules to move paper and small goods through tubes.
  - Air tools
  - HVAC control systems
- Vehicle propulsion
- Energy storage (compressed air energy storage)
- Air brakes
  - Railway braking systems
  - Road vehicle braking systems
- Underwater diving, for breathing and to inflate buoyancy devices
- Refrigeration using a vortex tube
- Air-start systems in engines
- Ammunition propulsion
  - Air guns
  - Air-soft equipment
  - Paintball equipment
- Cleaning dust and small debris in tiny spaces
- Sandblasting in machine shops
- Injection molding
- Food and beverage capping and fermentation
- Compressed air from Lysefjorden/Preikestolen (Norway) is being sold in cans, mostly to China.

4. Reciprocating Air Compressed :

The reciprocating compressor having basic working principal similar to IC engine, only the difference is that there is no any power stroke and exhaust stroke. In compressor only two stroke are take place one is suction in which fressh atmospheric air is taken in and other is compression in which compressed air is taken out to receiver.

4.1 Working of Reciprocating Air Compressed :

The conventinal reciprocating air compressor is shown in the figure below, it consists of a piston which reciprocates in a cylinder with help of crankshaft which receive power from external source. When piston moves down from extream top position, the negative pressure developed inside the cylinder causing intake valve to get open and atmospheric air get sucked inside the cylinder. This is continue till piston reach to bottom expream position, i.e. pressure inside the cylinder equals to outside atmospheric pressure. When piston moves upward inside pressure rises and intake valve get closed. Now the entrapped air get compressed continuously as piston moves upword. Then pressure exceed to set delivery pressure then delivery valve opens and compressed air delivered to receiver, this continues till piston reach to extream top position and cycle repeated.



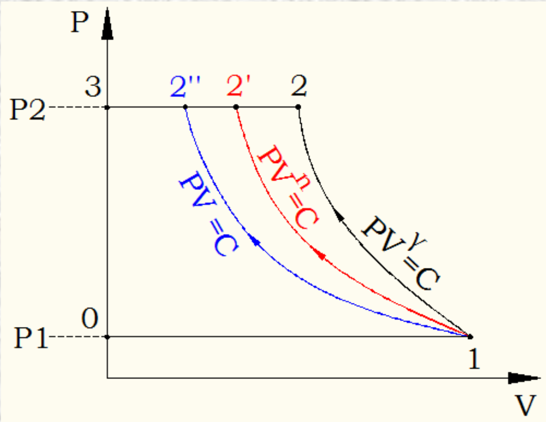
4.2 Assumptions :

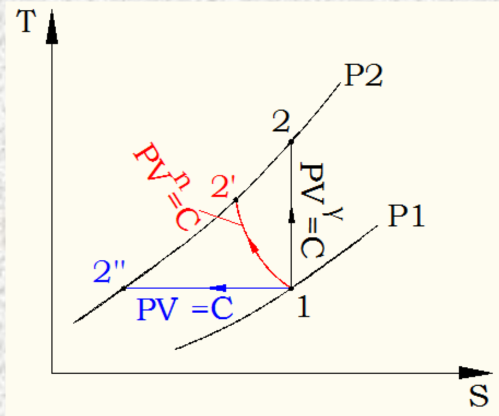
Following assumptions are made in considering the reciprocating air compressor operations,

- The working fluid 'Air' is a behave like a perfect gas
- There is no any clearance volume.
- Friction losses are negligible, so consider as zero power loss in friction.
- There are no wire-drawing effects in the valve or pipe line.
- There is not heat addition or rejection from cylinder, so cylinder is well insulated.

5. Single Stage Air Compressor : SPPU : May – 15, 6-Marks

Single stage air compressor compressed working fluid in a single stage upto delivery pressure. The pressure ratio for single stage is limited to 5, as increasing pressure ratio the construction to handle that much pressure by single cylinder become heavier and bulky. Also compressing air to higher pressure ratio in single stage causes consumption of more power.





As shown in PV and TS diagram, in single stage reciprocating air compressor, air is sucked inside the cylinder at atmospheric pressure P1, process 0-1, at point-1 cylinder is completely filled with air, this entrapped air then compressed to delivery pressure P2 by following isentropic compression $(P\,V^\gamma = C)$, polytrophic compression $(P\,V^n = C)$ or isothermal compression $(P\,V = C)$. Then delivered to receiver at delivery pressure P2, process 2-3.

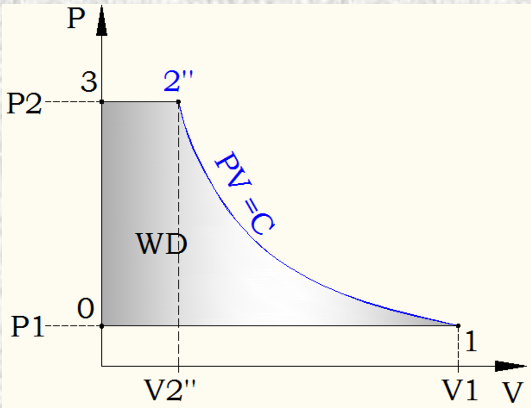
The graph area under the curve is representing the actual work consumed by the compressor, thus from PV diagram, it is clear that when compression follows the isothermal compression process it consumed minimum work (Area under 0-1-2”-3) and when compression is isentropic then it took maximum work (Area under 0-1-2-3). Where as the practically possible process, i.e. polytrophic process consumed work (Area under 0-1-2’-3) which is in between that of isentropic and isothermal compression.

5.1. Isothermal Compression Work Done :

Work done on the air when compressed from P1 pressure to P2 pressure, if the compression follows isothermal process (1-2”) is,

$$Isothermal\ Work\ Done\ =\ Area\ under\ the\ curve\ 0 - 1 - 2'' - 3$$

$Net\ Work\ Done = Work\ Done\ during\ compression\ (1 - 2'') +\ Work\ Done\ during\ Delivery\ (2'' - 3) -\ Work\ Done\ during\ suctiontion\ (0 - 1)$



$$Isothermal\ Work\ Done\ =\ Area\ under\ the\ curve\ 0-1-2''-3$$

$$Isothermal\ Work\ Done = P1\,V1\ \ln\left(\frac{V1}{V2''}\right) + (P2\,V2'') - (P1\,V1)$$

$$\ldots\ldots\ldots for\ process\ 1-2,\qquad \frac{P1\,V1}{T1} = \frac{P2\,V2''}{T2}$$

$$P1\,V1 = P2\,V2''\ \ldots\ldots as\ isothermal\ process\ T1 = T2$$

$$= P1\,V1\ \ln\left(\frac{V1}{V2''}\right) + (P1\,V1) - (P1\,V1)$$

$$= P1\,V1\ \ln\left(\frac{V1}{V2''}\right)\ \ldots\ldots P1\,V1 = P2\,V2''\ \Rightarrow \frac{V1}{V2''} = \frac{P2}{P1}$$

$$= P1\,V1\ \ln\left(\frac{P2}{P1}\right)\ \ldots\ldots P1\,V1 = m\,R\,T1$$

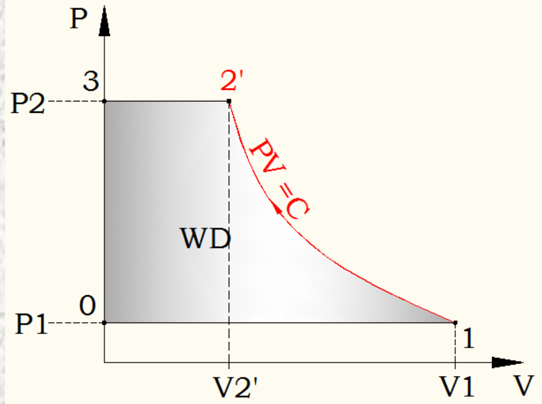
$$\boldsymbol{Isothermal\ Work\ Done = m\ R\ T1\ \ln\left(\frac{P2}{P1}\right)}$$

5.2. Polytropic Compression Work Done :

Work done on the air when compressed from P1 pressure to P2 pressure, if the compression follows polytropic process (1-2') is,

$$Polytropic\ Work\ Done\ =\ Area\ under\ the\ curve\ 0-1-2'-3$$

$Net\ Work\ Done = Work\ Done\ during\ compression\ (1-2') + Work\ Done\ during\ Delivery\ (2'-3) - Work\ Done\ during\ suctiontion\ (0-1)$



$$Polytropic\ Work\ Done = Area\ under\ the\ curve\ 0 - 1 - 2' - 3$$

$$Polytropic\ Work\ Done = \frac{P2\ V2' - P1\,V1}{n-1} + P2\ V2' - P1\,V1$$

$$= \frac{(P2\ V2' - P1\,V1) + (n-1)\,(P2\ V2' - P1\,V1)}{n-1}$$

$$= \frac{1 + (n-1)}{n-1}(P2\ V2' - P1\,V1)$$

$$= \frac{n}{n-1}(P2\ V2' - P1\,V1) \ \ldots..P1\,V1 = m\,R\,T1\ \&\ P2\,V2' = m\,R\,T2$$

$$= \frac{n}{n-1}(m\,R\,T2 - m\,R\,T1)$$

$$\boldsymbol{Polytropic\ Work\ Done = \frac{n}{n-1}\,(m\,R\,T1)\left[\frac{T2}{T1} - 1\right]}$$

$$\ldots\ldots\ldots.for\ process\ 1 - 2', \quad \frac{P1\,V1}{T1} = \frac{P2\,V2'}{T2}$$

$$\frac{T2}{T1} = \left(\frac{P2}{P1}\right)\left(\frac{V2'}{V1}\right)$$

$$for\ Polytropic\ process, P1\ \ V1^{n} = P2\ \ V2'^{n} \Rightarrow \left(\frac{V2'}{V1}\right) = \left(\frac{P1}{P2}\right)^{\frac{1}{n}} = \left(\frac{P2}{P1}\right)^{-\frac{1}{n}}$$

$$Thus\ put, \frac{T2}{T1} = \left(\frac{P2}{P1}\right)\left(\frac{P2}{P1}\right)^{-\frac{1}{n}} = \left(\frac{P2}{P1}\right)^{\frac{n-1}{n}}$$

$$\boldsymbol{Polytropic\ Work\ Done = \frac{n}{n-1}\,(m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]}$$

$$\ldots..P1\,V1 = m\,R\,T1$$

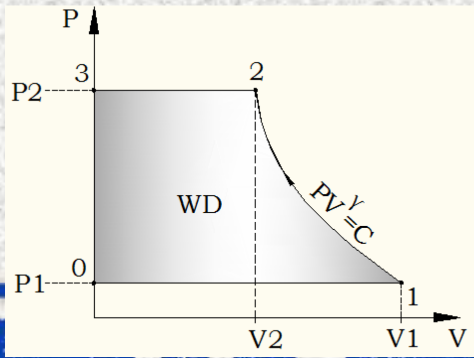
$$\textbf{Polytropic Work Done} = \frac{n}{n-1}(P1\,V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]$$

5.3. Isentropic Compression Work Done :

Work done on the air when compressed from P1 pressure to P2 pressure, if the compression follows isentropic process (1-2) is,

$$Isentropic\ Work\ Done = Area\ under\ the\ curve\ 0-1-2-3$$

$$Net\ Work\ Done = Work\ Done\ during\ compression\ (1-2) + Work\ Done\ during\ Delivery\ (2-3) - Work\ Done\ during\ suctiontion\ (0-1)$$



$$Isentropic\ Work\ Done = Area\ under\ the\ curve\ 0-1-2-3$$

$$Isentropic\ Work\ Done = \frac{P2\ V2 - P1\,V1}{\gamma - 1} + P2\ V2 - P1\,V1$$

$$= \frac{(P2\ V2 - P1\,V1) + (\gamma-1)(P2\ V2 - P1\,V1)}{\gamma - 1}$$

$$= \frac{1 + (\gamma - 1)}{\gamma - 1}(P2\ V2 - P1\,V1)$$

$$= \frac{\gamma}{\gamma - 1}(P2\ V2 - P1\,V1)\ \ldots..P1\,V1 = m\,R\,T1\ \&\ P2\,V2 = m\,R\,T2$$

$$= \frac{\gamma}{\gamma - 1}(m\,R\,T2 - m\,R\,T1)$$

$$\textbf{Isentropic Work Done} = \frac{\gamma}{\gamma - 1}(m\,R\,T1)\left[\frac{T2}{T1} - 1\right]$$

$$\ldots\ldots..\ldots.for\ process\ 1-2, \qquad \frac{P1\,V1}{T1} = \frac{P2\,V2}{T2}$$

$$\frac{T2}{T1} = \left(\frac{P2}{P1}\right)\left(\frac{V2}{V1}\right)$$

$$for\ Isentropic\ process, P1\ \ V1^{\gamma} = P2\ \ V2^{\gamma} \Rightarrow \left(\frac{V2}{V1}\right) = \left(\frac{P1}{P2}\right)^{\frac{1}{\gamma}} = \left(\frac{P2}{P1}\right)^{-\frac{1}{\gamma}}$$

$$Thus\ put, \frac{T2}{T1} = \left(\frac{P2}{P1}\right)\left(\frac{P2}{P1}\right)^{-\frac{1}{\gamma}} = \left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\boldsymbol{Isentropic\ Work\ Done = \frac{\gamma}{\gamma-1}\ (m\ R\ T1)\left[\left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}$$

$$\ldots..P1\ V1 = m\ R\ T1$$

$$\boldsymbol{Isentropic\ Work\ Done = \frac{\gamma}{\gamma-1}\ (P1\ V1)\left[\left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}$$

5.4. Free Air Delivery (FAD) : SPPU : May – 17, 6-Marks

It is the volume of air delivered under the conditions of temperature and pressure existing at the compressor intake i.e. volume of air delivered at surrounding air temperature and pressure.

In the absence of any given free air conditions, these are generally takes as 1.01325 bar and 15 °C.

Thus, free air delivery (FAD) represents the rate of volume of surrounding air which is sucked by the compressor and delivered at discharge pressure.

5.5. Isothermal Efficiency : SPPU : May – 17, 6-Marks

It is defined as the ratio of isothermal work done to the actual work done (i.e. Polytropic work done).

Isothermal work done or isothermal power is calculated from the theoretical indicator diagram drawn on the basis of an assumption that the compression is isothermal, and actual work done or actual power is determine from the actual indicator diagram taken during the test on the compressor.

Mathematically,

$$\eta_{iso} = \frac{P1\ V1\ \ \ln\left(\frac{P2}{P1}\right)}{\frac{n}{n-1}\ (P1\ V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]}$$

5.6. Indicated Power (IP) :

The indicated power is also called as air power, which is required to drive the compressor is given by the equation as,

$$Indicated\ Power = \frac{W \quad n}{60000}\ KW$$

$$\ldots\ldots\quad W = Work\ Done\ and$$

$$n = No.of\ Stroke\ per\ min.\ completed\ by\ compressor.$$

$$n = 2N\ if\ compressor\ is\ double\ acting.$$

$$n = N\ if\ compressor\ is\ single\ acting$$

$$N = speed\ of\ compressor\ in\ rpm$$

5.7. Mechanical Efficiency :

It is defined as the ratio of air power i.e. indicated power to the shaft power i.e. brake power.

Indicated power is determine from the actual indicator diagram taken during the test on the compressor, and brake power is the power delivered to the shaft of the compressor or the power required to drive the compressor.

Mathematically,

$$\eta_{mech} = \frac{Indicated\ Power}{Brake\ Power}$$

5.8. Polytropic Efficiency :

It is define as the ratio of polytropic work done to the actual work input.

The actual work input may be taken as isentropic work input in case of the actual work input is not given.

Mathematically,

$$\eta_{mech} = \frac{\frac{n}{n-1}\,(P1\,V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]}{\frac{\gamma}{\gamma-1}\,(P1\,V1)\left[\left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}$$

6. Methods of Achieving Isothermal Compression -
SPPU : May-15, May–14, 6-Marks

From the PV diagrams, it is observed that work required for isothermal compression is lesser than other polytropic and isentropic compression. To achieve isothermal compression the speed of the compressor should be low which means time taken for the isothermal compression process would increase.

The following methods are the practical means of achieving isothermal compression as,

Water Spray Injection – some of the heat was removed by injecting water into the cylinder towards the end of the compression stroke.

Suitable Cylinder Dimensions – cooling can be done more effectively if surface area is increased by using cylinder of large diameter and shorter stroke length.

Water Jacketing – water kept circulating around the compressor and air is cooled.

Inter Stage Cooling – the compression is divided into two or more stages and in-between two stages cooling is provided.

Prob . 1 – A single stage reciprocating air compressor intake air at the rate of 1 m^3/min. at 1.013 bar and 15 °C. Then compressed air is delivers at 7 bar according to the law $PV^{1.35}$ = constant. Calculate power required for the compression.

Ans. – Single Stage Compressor, V1=1 m^3/min., P1 = 1.013 bar = $1.013x10^5$ N/m^2, T1 =15 °C, P2 = 7 bar = $7x10^5$ N/m^2, $PV^{1.35}$ = constant, n = 1.35.

Power required for the compression –

We know, power required for compression is equal to the work done on the air during compression, thus,

$$P = WD_{poly} = \frac{n}{n-1}(P1\,V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$P = WD_{poly} = \frac{1.35}{1.35-1}\left(1.013x10^5\ x\ \frac{1}{60}\right)\left[\left(\frac{7}{1.013}\right)^{\frac{1.35-1}{1.35}} - 1\right]$$

$$P = WD_{poly} = 4236.88\ Watt$$

$$\boldsymbol{P = 4.237\ KW\ \ldots.Ans.}$$

Prob . 2 – A single stage, single cylinder reciprocating air compressor delivers air at 6 bar. The rate of air taken in during suction is 12 Kg/min. at 1.013 bar and 27 °C. the compression take place with the index of 1.25.

Calculate,

i) Work required for delivering 1 Kg of air.
ii) Actual power required to run the compressor if η_{mech} = 80%.

Ans. – Single Stage, Single Cylinder Compressor, $\dot{m}$=12 Kg/min., P1 = 1.013 bar = $1.013x10^5$ N/m^2, T1 =27 °C = 300 K, P2 = 6 bar = $6x10^5$ N/m^2, n = 1.25.

i) Work required for delivering 1 Kg of air –

We know, compression index n = 1.25 indicates the compression process is polytropic, thus,

$$WD_{poly} = \frac{n}{n-1}\,(P1\,V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] \quad \ldots.. \; P1\,V1 = m\,R\,T1$$

$$WD_{poly} = \frac{n}{n-1}\,(m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] \quad \ldots. \; delivery\ of\ air, \quad m = 1\,Kg$$

$$WD_{poly} = \frac{1.25}{1.25-1}\,(1\,x\,0.287\,x\,300)\left[\left(\frac{6}{1.013}\right)^{\frac{1.25-1}{1.25}} - 1\right]$$

$$\boldsymbol{WD_{poly} = 183.94\,KJ/Kg\ of\ air\ \ldots..Ans.1}$$

ii) Actual power required to run the compressor if η_{mech} = 80% –

We know, actual power means the brake power, and indicated power also known as air power which required to drive the compressor, thus,

$$IP = \dot{m}\,x\,WD_{poly}$$

$$IP = \frac{12}{60}\,x\,183.94$$

$$\boldsymbol{IP = 36.788\,KW}$$

Now the mechanical efficiency is,

$$\eta_{mech} = \frac{IP}{BP}$$

$$0.88 = \frac{36.788}{BP}$$

$$\boldsymbol{BP = 41.805\,KW \ldots\ldots Ans.2}$$

Prob . 3 – A single cylinder air compressor delivers 9 Kg of air per minute. The air is compressed from 1 bar and 27 °C to 7 bar. The compression process follows the law $PV^{1.25} = C$.

Find,

i) Work Done
ii) Brake Power required if mechanical efficiency is 85%.

SPPU : Dec.–15, 7-Marks

Ans. – Single Stage, Single Cylinder Compressor, $\dot{m}$=9 Kg/min., P1 = 1 bar = $1x10^5$ N/m^2, T1 =27 °C = 300 K, P2 = 7 bar = $7x10^5$ N/m^2, n = 1.25.

i) Work Done –

We know, compression is polytropic,

$$WD_{poly} = \frac{n}{n-1}\,(P1\,V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] \quad \ldots\ldots\ P1\,\dot{V}1 = \dot{m}\,R\,T1$$

$$WD_{poly} = \frac{n}{n-1}\,(\dot{m}\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$WD_{poly} = \frac{1.25}{1.25-1}\left(\frac{9}{60}\ x\ 287\ x\ 300\right)\left[\left(\frac{7}{1}\right)^{\frac{1.25-1}{1.25}} - 1\right]$$

$$\boldsymbol{WD_{poly} = 30723.05\ Watt = 30.72\ KW\ \ \ldots\ldots Ans.1}$$

iii) Brake Power required if (η_{mech} = 85%) –

We know, the indicated power is the polytropic work done,

$$IP = WD_{poly}$$

$$IP = 30.72$$

$$\boldsymbol{IP = 30.72\ KW}$$

Now the mechanical efficiency is,

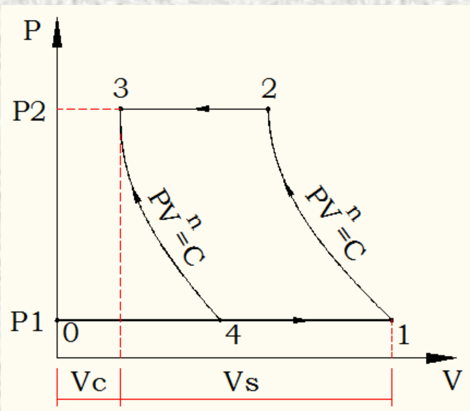
$$\eta_{mech} = \frac{IP}{BP}$$

$$0.85 = \frac{30.72}{BP}$$

$$\boldsymbol{BP = 36.14\ KW\ \ldots\ldots Ans.2}$$

7. Effect of Clearance in Compressor

The clearance in the cylinder ensures the mechanical freedom to moving parts like valves and piston without much wear and tear. Also provides the necessary space for valve operation movement.



The figure shows PV diagram of the single cylinder single acting and single stage reciprocating compressor with the clearance.

The intake valve is open at point-0 and air at atmospheric pressure P1 is taken in cylinder, this air suction continue till point-1. After the suction when intake valve get closed the entrapped air is compressed till the pressure reaches to delivery pressure P2, shown by process 1-2. The delivery valve get opened at point-2 and air delivery is started from point-2 and ends at point-3.

Thus, the some small quantity of air Vc at pressure P2 remaining in the dead space at the end of delivery. In next cycle first compressed air at P2 get expanded till the pressure reduced to atmospheric, shown by expansion of air by process 3-4. Once inside cylinder pressure reach to P1 at point-4, then after further movement of piston create a negative pressure inside the cylinder and the intake valve get opened and now actual outside atmospheric air is sucked into the cylinder.

Therefore the volume of air sucked inside the cylinder when clearance is provided is (V1-V4) which is lesser than the stroke volume (V1-V3), this means handling capacity of the compressor is reduced due to clearance provided. Due to this, in compressor the clearance volume is kept as small as possible.

7.1 Volumetric Efficiency of Compressor at Suction Conditions - SPPU : May–17, 6-Marks

The volumetric efficiency of a compressor is the ratio of actual mass or volume of air delivered to the mass or volume corresponding to the displacement of piston.

OR

The volumetric efficiency of a compressor is the ratio of free air delivered (FAD, i.e. rate of volume of atmospheric air which is actually sucked by the compressor and delivered at discharge pressure.) to the stroke volume.

Mathematically,

$$\eta_{vol} = \frac{V_{FAD}}{V_s}$$

$$\eta_{vol} = \frac{V_1 - V_4}{V_s} \quad \ldots\ldots V_1 = V_s + V_c$$

$$\eta_{vol} = \frac{(V_s + V_c) - V_4}{V_s} \quad \ldots\ldots Eq.1$$

For polytropic air expansion process 3-4,

$$P_3\, V_3{}^n = P_4\, V_4{}^n$$

$$\ldots\ldots P_3 = P_2 = delivery\ pressure, and\ V_3 = V_c = clearance\ volume$$

$$\ldots\ldots P_4 = P_1 = intake\ pressure$$

$$P_2\, V_c{}^n = P_1\, V_4{}^n$$

$$\left(\frac{V_4}{V_c}\right)^n = \left(\frac{P_2}{P_1}\right)$$

$$\left(\frac{V_4}{V_c}\right) = \left(\frac{P_2}{P_1}\right)^{1/n}$$

$$V_4 = \left(\frac{P_2}{P_1}\right)^{1/n} V_c \quad \ldots\ldots put\ in\ Eq.1$$

$$\eta_{vol} = \frac{(V_s + V_c) - \left(\frac{P_2}{P_1}\right)^{1/n} V_c}{V_s}$$

$$\eta_{vol} = 1 + \frac{V_c}{V_s} - \left(\frac{P_2}{P_1}\right)^{1/n} \frac{V_c}{V_s} \quad \ldots\ldots Clearance\ Ratio, C = \frac{V_c}{V_s} \quad and\ Pressure\ Ratio, R_p = \frac{P_2}{P_1}$$

$$\eta_{vol} = 1 + C - (R_p)^{1/n} C$$

$$\boldsymbol{\eta_{vol} = 1 + C - C\,(R_p)^{1/n}}$$

It is the measure of handling capacity of the compressor, which is reduced with increased in clearance. But there is no effect on the work done for the air delivered.

7.2 Volumetric Efficiency of Compressor at Ambient Conditions

The ambient conditions pressure (P_0) is assumed to be 1 bar and ambient condition temperature (T_0) is assumed to be 15°C.

Let the V_0 be the volume of air delivered at ambient pressure and temperature.

We know, mass of air at ambient condition is,

$$m = \frac{P_0\, V_0}{R\, T_0}$$

And from PV diagram, mass of air sucked is,

$$m = \frac{P_1\,(V_1 - V_4)}{R\, T_1}$$

Since the mass of air sucked remains constant,

$$m = \frac{P_0\, V_0}{R\, T_0} = \frac{P_1\,(V_1 - V_4)}{R\, T_1}$$

$$V_0 = \frac{P_1\ T_0}{P_0\ T_1}\,(V_1 - V_4)$$

The volumetric efficiency at ambient conditions is,

$$\eta_{vol\_o} = \frac{V_0}{V_s}$$

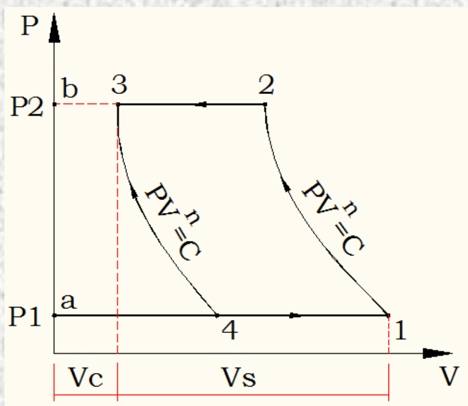
$$\eta_{vol\_o} = \frac{\frac{P_1\ T_0}{P_0\ T_1}\,(V_1 - V_4)}{V_s}$$

$$\eta_{vol\_o} = \frac{P_1\ T_0}{P_0\ T_1}\left[\frac{(V_1 - V_4)}{V_s}\right] \quad \ldots\ldots \frac{V_1 - V_4}{V_s} = 1 + C - C\,(R_p)^{1/n}$$

$$\boldsymbol{\eta_{vol\_o} = \frac{P_1\ T_0}{P_0\ T_1}\left[1 + C - C\,(R_p)^{1/n}\right]}$$

7.3 Work Done on Air by Compressor with clearance :

Considering polytropic compression process take place with the polytropic compression index 'n' and compressed air from intake pressure P1 to delivery pressure P2 in single stage with clearance in compressor cylinder.



$$WD_{poly}/cycle = \text{Area under curve } 1-2-3-4$$

$$WD_{poly}/cycle = A\,(a-1-2-b) - A\,(a-4-3-b)$$

$$WD_{poly}/cycle = WD_{poly} - WD_{poly}$$

$$WD_{poly}/cycle = \frac{n}{n-1}(P1\,V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] - \frac{n}{n-1}(P4\,V4)\left[\left(\frac{P3}{P4}\right)^{\frac{n-1}{n}} - 1\right]$$

$$put \;\ldots\ldots P4 = P1, \qquad P3 = P2$$

$$WD_{poly}/cycle = \frac{n}{n-1}(P1\,V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] - \frac{n}{n-1}(P1\,V4)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\boldsymbol{WD_{poly}/cycle = \frac{n}{n-1}\,P1(V1-V4)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]}$$

Similarly,

For Isentropic compression, the work done is,

$$\boldsymbol{WD_{isen}/cycle = \frac{\gamma}{\gamma-1}\,P1(V1-V4)\left[\left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}$$

For Isothermal compression, the work done is,

$$\boldsymbol{WD_{iso}/cycle = P1(V1-V4)\;\ln\left(\frac{P2}{P1}\right)}$$

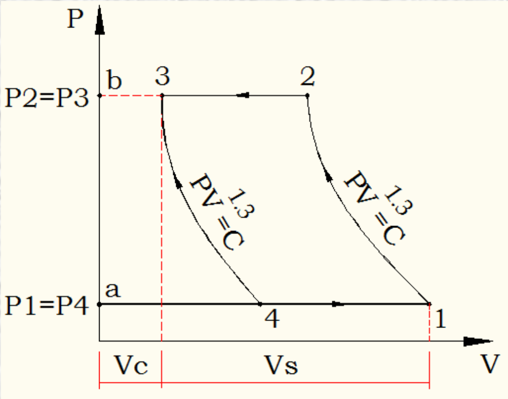
Prob . 8 – A single stage, single cylinder and single acting reciprocating compressor, the actual volume of air sucked inside the cylinder during intake is 11.5 m^3/min. Intake pressure and temperature are 1.013 bar and 27 °C respectively. The delivery pressure 920 KPa. Take clearance 5.5 % of the stroke, compressor run at the speed of 410 rpm, L/D ratio equal to 1.25 and compression index is 1.3.

Calculate,

i) Volumetric Efficiency
ii) Cylinder Dimensions

iii) Indicated Power

Ans. – Single Stage, Single Cylinder, Single Acting Reciprocating Compressor, $\dot{V} = (V1 - V4) = 11.5\ m^3/min$, P1 = 1.013 bar = $1.013x10^5$ N/m^2, T1 =27 °C = 300K, P2 = 920 KPa = 9.2 bar = $9.2x10^5$ N/m^2, $V_c = 5.5\ \%\ V_s$, N = 410 rpm, L/D = 1.25, n = 1.3.



i) Volumetric Efficiency –

We know,

$$Clearance\ Volume\ Ratio, C = \frac{V_c}{V_s}\ \ldots\ldots\ldots\ldots\ V_c = 0.055\ V_s$$

$$\boldsymbol{C = \frac{V_c}{V_s} = 0.055}$$

And pressure ratio is,

$$R_p = \frac{P2}{P1}$$

$$R_p = \frac{9.2}{1.013}$$

$$\boldsymbol{R_p = 9.0819}$$

The volumetric efficiency when clearance is consider,

$$\eta_{vol} = 1 + C - C\left(R_p\right)^{1/n}$$

$$\eta_{vol} = 1 + 0.055 - 0.055\ (9.0819)^{1/1.3}$$

$$\boldsymbol{\eta_{vol} = 0.7548 = 75.48\ \%\ \ \ldots\ldots Ans.1}$$

ii) Cylinder Dimensions -

We know, volume flow rate is,

$$\dot{V} = Stroke\ Volume\ x\ Speed$$

$$\dot{V} = \left(\frac{\pi}{4}\, D^2\, L\right) x\, N \ldots\ldots L/D = 1.25 \;\Rightarrow L = 1.25\, D$$

$$\dot{V} = \left(\frac{\pi}{4}\, D^2\, x\, 1.25\, D\right) x\, N$$

$$11.5 = \left(\frac{\pi}{4}\, D^2\, x\, 1.25\, D\right) x\, 410$$

$$\boldsymbol{D = 0.3057\, m \quad \ldots..Ans.2}$$

From L/D ratio,

$$L/D = 1.25$$

$$L = 1.25\, D$$

$$L = 1.25\, x\, 0.3057$$

$$\boldsymbol{L = 0.3821\, m \quad \ldots..Ans.2}$$

iii) Indicated Power –

We know, indicated power is a polytropic work done as,

$$IP = WD_{poly} = \frac{n}{n-1}\, P1(V1 - V4)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$IP = \frac{1.3}{1.3-1}\, x\, 1.013\, x\, 10^5 \left(\frac{11.5}{60}\right)\left[\left(\frac{9.2}{1.013}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$\boldsymbol{IP = 55854.5\, Watt = 55.85\, KW \quad \ldots.Ans.3}$$

Prob . 9 – A single stage, single cylinder and single acting reciprocating air compressor, deliver 0.75 Kg/min. at a pressure of 7 bar. The intake pressure and temperature are 1.013 bar and 27°C. The cylinder bore and stroke are, 100mm and 150mm respectively. The clearance volume is 3.5% of the stroke volume. Assume polytropic compression with index of 1.3.

Calculate,

i) Volumetric Efficiency
ii) Power required to drive the compressor if mechanical efficiency is 82%
iii) Speed of the compressor per min.

Ans. – Single Stage, Single Cylinder, Single Acting Reciprocating Compressor, $\dot{m} =$ 0.75 Kg/min, P2 = 7 bar = 7x10^5 N/m^2, P1 = 1.013 bar = 1.013x10^5 N/m^2, T1 =27 °C = 300K, D=100mm, L=150mm, $V_c = 3.5\,\%\, V_s$, n = 1.3.

i) Volumetric Efficiency –

We know the volumetric efficiency when clearance is consider,

$$\eta_{vol} = 1 + C - C\,(R_p)^{1/n}$$

$$\ldots..C = \frac{V_c}{V_s} = 3.5\% = 0.035 \quad and \quad R_p = \frac{P2}{P1}$$

$$\eta_{vol} = 1 + C - C\left(\frac{P2}{P1}\right)^{1/n}$$

$$\eta_{vol} = 1 + 0.035 - 0.035\left(\frac{7}{1.013}\right)^{1/1.3}$$

$$\boldsymbol{\eta_{vol} = 0.8802 = 88.02\ \%} \quad \ldots\ldots \boldsymbol{Ans.1}$$

ii) Power required to drive the compressor if mechanical efficiency is 82% –

We know, Indicated Power i.e. Polytropic work done is,

$$IP = WD_{poly} = \frac{n}{n-1}\ P1\ V1\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] \quad \ldots\ldots P1\,V1 = \dot{m}\,R\,T1$$

$$IP = WD_{poly} = \frac{n}{n-1}\ \dot{m}\,R\,T1\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] \quad \ldots.R\ for\ air = 287\ J/Kg - K$$

$$IP = \frac{1.3}{1.3-1}\ x\ \frac{0.75}{60}\ x\ 287\ x\ 300\ x\left[\left(\frac{7}{1.013}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$\boldsymbol{IP = 2621.82\ Watt = 2.62\ KW}$$

Now, the power supplied to drive the compressor, i.e. BP is,

$$\eta_{mech} = \frac{IP}{BP}$$

$$0.82 = \frac{2.62}{BP}$$

$$\boldsymbol{BP = 3.195\ KW} \quad \ldots \boldsymbol{Ans.2}$$

iii) Speed of the compressor per min. –

We know, the rate of actual volume intake per min. is,

$$P1\,\dot{V}1 = \dot{m}\,R\,T1$$

$$1.013\ x\ 10^5\ \ \dot{V}1 = 0.75\ x\ 287\ x\ 300$$

$$\boldsymbol{Actual\ Air\ Intake, \dot{V}1 = 0.6375\ m^3/min}$$

Now, rate of theoretical volume intake per min. is,

$$\dot{V}_s = \frac{\pi}{4} D^2 \, L \, N$$

$$\dot{V}_s = \frac{\pi}{4} 0.1^2 \, x \, 0.15 \, x \, N$$

$$\textbf{Theoretical Air Intake}, \dot{V}_s = (1.178 \, x \, 10^{-3} \, N) \, m^3/min$$

We know, Volumetric efficiency is,

$$\eta_{vol} = \frac{Actual\ Air\ Intake}{Theoretical\ Air\ Intake}$$

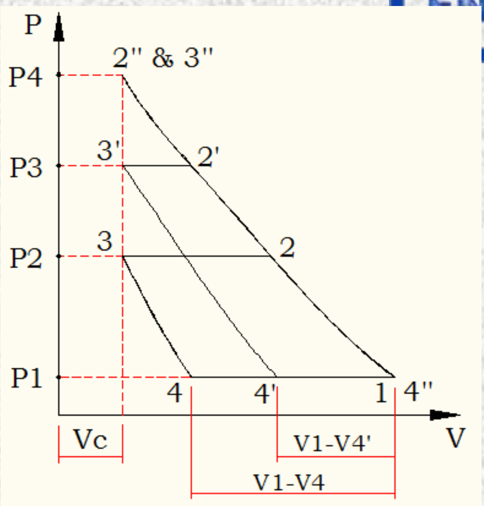
$$\eta_{vol} = \frac{\dot{V}1}{\dot{V}_s}$$

$$0.8802 = \frac{0.6375}{(1.178 \, x \, 10^{-3} \, x \, N)}$$

$$\boldsymbol{N = 614.83 \, rpm \quad \ldots . Ans.3}$$

8. Limitations of Single Stage Compression :

Volumetric efficiency when clearance is considered is given by $\eta_{vol} = 1 + C - C\,(R_p)^{1/n}$. It is observed that, the η_{vol} of a reciprocating compressor is depends on the clearance ratio 'C', pressure ratio 'R_p' and index of expansion.



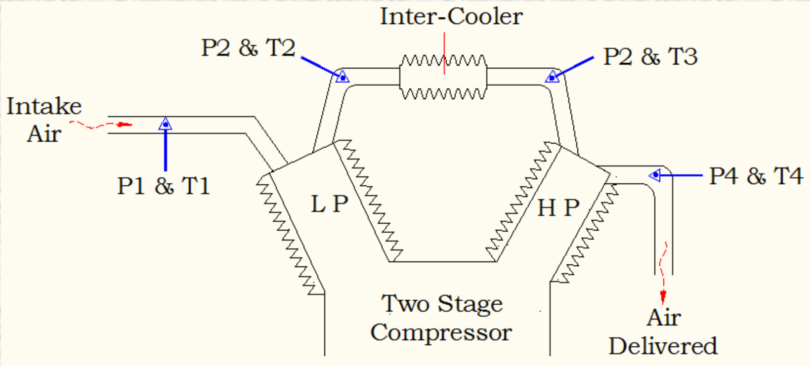
From the figure, it is clear that, volume of air taken in is compressed which causes decrease in volume and increase in corresponding delivery pressure. For the same clearance volume Vc and fixed intake pressure P1, if delivery pressure is P2 the residual compressed air in clearance volume expanded to point-4, thus the effective volume of air taken into the compressor is (V1-V4). When delivery pressure increases to level P3, then residual compressed air expanded to point-4', thus effective volume of air taken into the compressor is (V1-V4') which is lesser than the first one.

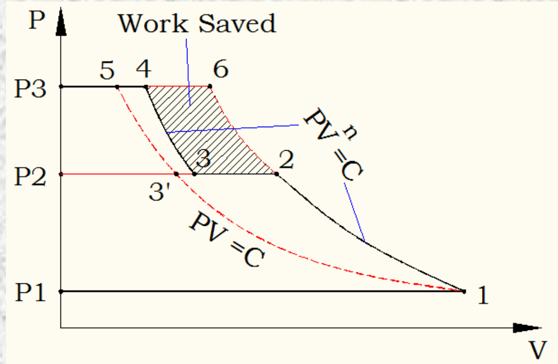
On increasing delivery pressure further, at some point the compression ends at the clearance volume i.e. all air compressed to clearance volume and there is no air delivery. When these entrapped air expands it follows the same path and reach to point-4", and thus there will be only compression and re-expansion of air again and again and no air delivery.

Therefore the maximum pressure ratio attainable with a single stage reciprocating compressor is limited by the clearance volume.

9. Multi Stage Compression with inter-cooling :

The arrangement of two stage compressor with inter cooler and its PV diagram are shown in following figure,





In two stage compressor with inter-cooler, the air is first taken into low pressure (LP) cylinder. After compression to some desired intermediate pressure P2, the air from the LP cylinder at condition 2 is passed through inter-cooler. The condition of air leaving the inter cooler is shown by the point where its temperature is reduced from T2 to T3 (temperature T3 < T2). It is possible to cool the air in the inter-cooler to initial intake temperature T1, by properly regulating the supply of cooling water I the inter-cooler. Finally air is compressed from condition 3 to condition 4 in high pressure (HP) cylinder and is discharge to receiver at final delivery pressure P3. If the same pressure ratio is achieved by single stage then the compression process will be 1-2-6, but doing inter-cooling the some work is saved shown by shredded area. If the inter-cooling is perfect and temperature achieved will be intake temperature then the curve passing through these points is shown by dotted curve line is a isothermal compression.

9.1. Advantages of Multi Stage Compressor with Inter-Cooling :

SPPU : Dec.-17, May – 16, May – 15, Dec. -15, 6-Marks

- The work done per kg of air can be reduced by introducing an inter-cooler in between two stages of compression, as compared to single stage compression for the same delivery pressure.
- Better mechanical balance can be achieved with multi-stage compressors.
- The pressure and hence temperature range in each state is reduced.
- Loss of air due to leakage is less.
- Higher volumetric efficiency can be achieved as the pressure of each stage is less than the overall pressure ratio, as the volumetric efficiency is also a function of a pressure ratio.
- Effective lubrication is possible due to lower temperature range.

– High pressure cylinder is designed to withstand high pressure where as the Low pressure cylinder is designed withstand low pressure. This reduces the cost of the compressor.

9.2. Disadvantages of Multi Stage Compressor with Inter-Cooling :

– Design and Construction is more complicated than that of single stage compressor.

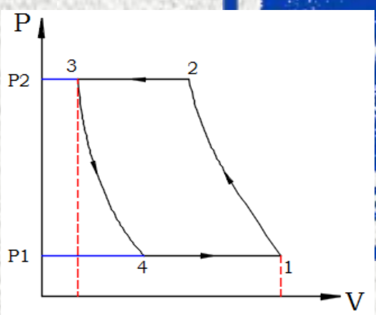
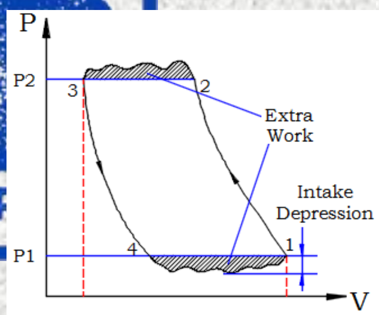
– The initial cost is more.

– Not suitable for low pressure ratio or less delivery pressure.

– It needed additional component inter-cooler and its accessories like water pump, hose system, filters etc.

– The additional accessories like water pump consumed power, thus overall power requirement is more.

10. Differentiate between Theoretical and Actual Indicator Diagram for Reciprocating Compressor : SPPU : May -17, 6-Marks

Theoretical Indicator Diagram	Actual Indicator Diagram
 	
The intake valve opens at point-4, and thus it is assumed at point-4 cylinder experience the negative pressure instantly.	The intake valve not opened at point-4, but when the pressure inside the cylinder is slightly below the atmospheric pressure.
Valve inertia factor not considered.	Valve inertia is play the role in delay in opening
Intake of air is assumed at constant atmospheric pressure throughout the suction process.	Intake of air is actually started at pressure below the atmospheric pressure and ends when pressure equal to atmospheric. The pressure difference is called as intake depression.
The intake is at the atmospheric pressure, the suction curve is a smooth and horizontal.	The abrupt opening of the valve causes the formation of eddies and valve bounce which result in wavy curve during suction process 4-1.

Compression is assumed to continue till the point-2, and delivery valve is open instantly and delivery starts, so no further pressure rise take place.	Compression continues beyond the delivery line pressure and then delivery valve opens.
The delivery process curve is smooth and horizontal and take place at constant delivery pressure.	The delivery process curve is wavy, due to valve flutter and inertial effect.
There is work done is equals to that of required for the compression process only, not extra work is done during suction and delivery.	The shaded area represent the extra work of compression required. It is because of the delay in opening and closing of valves, flutter and inertial effect of valves.

11. Ideal intermediate pressure with perfect inter-cooling of two stage compression : SPPU : May -18, 6-Marks

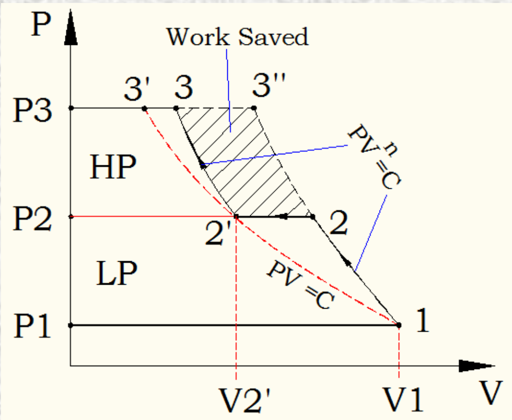


Figure shows the PV diagram of two stage compressor with perfect inter-cooling. In stage one, low pressure (LP) cylinder air is intake at pressure P1 and temperature T1, then it is compressed polytropically to pressure P2, process shown by 1-2. The exits of the LP cylinder at point-2 with pressure P2 and temperature T2 is enter into the inter-cooler where its temperature bring back to intake temperature, i.e. $T2' = T1$, thus there is a perfect inter-cooling. Now at point-2', the air at pressure P2 and temperature T1 is enter into high pressure (HP) cylinder where it is again compressed polytropically to delivery pressure P3, the process shown by 2'-3.

If the compression take place in single stage, then the curve will be 1-2-3", but using two stage with inter-cooling the compression curve is 1-2-2'-3, thus the shaded area is the work saved duet to multi-stage compression.

If the temperature is kept constant then the compression will follow the curve 1-2'-3' which is isothermal compression which result in minimum work done which is practically not possible.

The total work done is,

$$WD_{poly} = WD_{stage-1} + WD_{stage-2}$$

$$WD_{poly} = \frac{n}{n-1}\ P1\ V1 \left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1}\ P2\ V2' \left[\left(\frac{P3}{P2}\right)^{\frac{n-1}{n}} - 1\right]$$

$$WD_{poly} = \frac{n}{n-1}\ (m\,R\,T1\,) \left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1}\ (m\,R\,T2') \left[\left(\frac{P3}{P2}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\ldots\ldots\ Perfect\ inter-cooling, \qquad T2' = T1$$

$$WD_{poly} = \frac{n}{n-1}\ (m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1}\ (m\,R\,T1)\left[\left(\frac{P3}{P2}\right)^{\frac{n-1}{n}} - 1\right]$$

$$WD_{poly} = \frac{n}{n-1}\ (m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1 + \left(\frac{P3}{P2}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\boldsymbol{WD_{poly} = \frac{n}{n-1}\ (m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} + \left(\frac{P3}{P2}\right)^{\frac{n-1}{n}} - 2\right]}$$

Now, if the intake pressure P1 and delivery pressure P3 are fixed, then optimum intermediate pressure for minimum work of compression in both LP and HP cylinder can be calculated by differentiating WD with respect to pressure P2 and equal to zero as,

For minimum work done condition,

$$\frac{d}{d\,(P2)}\left(WD_{poly}\right) = 0$$

$$\frac{d}{d\,(P2)}\left\{\frac{n}{n-1}\ (m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} + \left(\frac{P3}{P2}\right)^{\frac{n-1}{n}} - 2\right]\right\} = 0$$

$$\ldots Put\ \frac{n-1}{n} = x,\quad \frac{n}{n-1} = \frac{1}{x}$$

$$\frac{d}{d\,(P2)}\left\{\frac{1}{x}\ (m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{x} + \left(\frac{P3}{P2}\right)^{x} - 2\right]\right\} = 0$$

$$\frac{1}{x}\ (m\,R\,T1)\ \frac{d}{d\,(P2)}\left[(P2)^x (P1)^{-x} + (P3)^x (P2)^{-x} - 2\right] = 0$$

$$\frac{d}{d\,(P2)}\left[(P2)^x\ (P1)^{-x} + (P3)^x\ (P2)^{-x} - 2\right] = 0$$

$$(P1)^{-x}\frac{d}{d\,(P2)}(P2)^x + (P3)^x\ \frac{d}{d\,(P2)}(P2)^{-x} - \frac{d}{d\,(P2)}(2) = 0$$

$$(P1)^{-x}\ x\,.\ (P2)^{x-1} + (P3)^x\ .-x\,.(P2)^{-x-1} - 0 = 0$$

$$\left[(P1)^{-x}\ x\,.\ (P2)^{x-1}\right] - \left[(P3)^x\ .x\,.(P2)^{-(x+1)}\right] = 0$$

$$\left[(P1)^{-x}\ x\,.\ (P2)^{x-1}\right] = \left[(P3)^x\ .x\,.(P2)^{-(x+1)}\right]$$

$$\frac{(P2)^{x-1}}{(P2)^{-(x+1)}} = \frac{(P3)^x}{(P1)^{-x}}$$

$$(P2)^{x-1}\ (P2)^{x+1} = (P3)^x\ (P1)^x$$

$$(P2)^{x-1+x+1} = (P3)^x\,(P1)^x$$

$$(P2)^{2x} = (P3)^x\,(P1)^x$$

$$(P2)^2 = (P3)\,(P1)$$

$$P2^2 = P3\;P1 \qquad Also \qquad P2\;P2 = P3\;P1$$

$$\boldsymbol{P2 = \sqrt{P1\;P3} \qquad Also \qquad \frac{P2}{P1} = \frac{P3}{P2}}$$

Therefore, the ideal intermediate pressure, with which the minimum work of compression is required is the geometric mean of the intake pressure (P1) and delivery pressure (P2) of two stage compression. This is also known as the condition for maximum efficiency.

Thus, with perfect inter-cooling and ideal intermediate pressure P2, the work of two stage compression will be,

$$WD_{poly} = \frac{n}{n-1}\,(m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} + \left(\frac{P3}{P2}\right)^{\frac{n-1}{n}} - 2\right] \;\ldots\ldots\; put,\quad \frac{P2}{P1} = \frac{P3}{P2}$$

$$WD_{poly} = \frac{n}{n-1}\,(m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} + \left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 2\right]$$

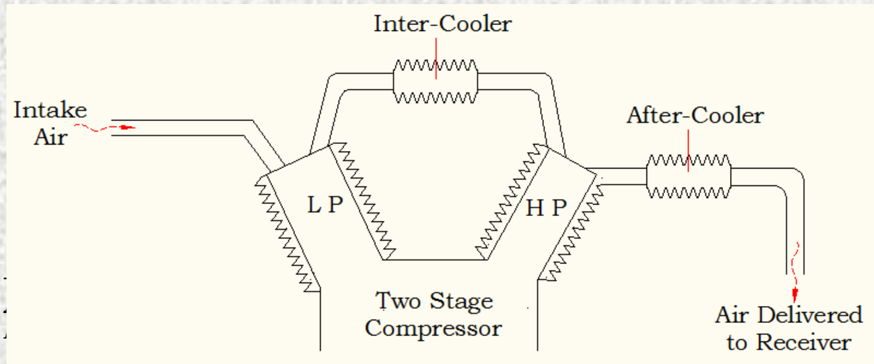
$$WD_{poly} = \frac{n}{n-1}\,(m\,R\,T1)\left[2\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 2\right]$$

$$\boldsymbol{WD_{poly} = 2\,\frac{n}{n-1}\,(m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]}$$

Similarly, for K-stage compression with perfect inter-cooling and ideal intermediate pressure is,

$$\boldsymbol{WD_{poly} = K\,\frac{n}{n-1}\,(m\,R\,T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]} \;\ldots.K = No.of\;compression\;stages.$$

12. After Cooling of Compression :



After compression, high pressure air is supplied to the receiver for storage. But with increase in pressure there is rise in temperature too, which causes increased in specific volume of compressed air, thus required large size of receiver.

If the compressed air is cooled before the storage in receiver it will result in decreased in specific volume thus needed small size receiver.

This objective is completed by incorporating a heat exchanger after compression, this is called as after-cooler. It is provided in between the compressor outlet and receiver inlet, the compressed high pressure and high temperature air is cooled by circulating water into the heat exchanger, thus compressed air is cooled almost upto atmospheric temperature.

Thus after-cooler reduces the specific volume of compressed air supplied to the receiver which help in reducing the size of the receiver.

13. Capacity control of compressors :

The compressors are not run continuously all the time at their maximum rated capacity, so it become essential to have some means to control the amount of air supplied by the compressor in order to meet the required demand.

Thus there are following three methods are used to controlled the air delivery of the compressor as,

13.1. Throttle Control Method –

In the throttle control method, the opening of the intake valve is controlled by the pressure in the receiver.

When receiver pressure is lesser, the intake valve is opened to its maximum possibility, but as the receiver pressure build up the opening of intake valve decreases accordingly. Thus, receiver pressure exceeds the normal pressure, the intake valve get partly closed and the quantity of air sucked by the compressor is reduced.

The limitation of this method is that the temperature of air delivered may rise to dangerous value due to high pressure ratio.

13.2. Clearance Control Method –

In the clearance control method, the clearance volume pockets are provided in side the cylinder. When the receiver pressure is lesser then these clearance volume pocket remain inactive and dose not taking part of compression.

When receiver pressure exceeds the normal value, these clearance volume pockets come into operation, by which these clearance volume pockets are brought in communication with cylinder volume with the help of automatically operated valves.

Thus, increase in volume reduces the volumetric efficiency and reduces the rate of volume of air intake, compressed and delivered to receiver.

13.3. Blow-off Control Method –

In the blow-off control method, a pressure blow-off valve is provided in the high pressure (HP) cylinder. When the pressure in the receiver exceed the normal value due to decrease in the demand, the high pressure air is released to the atmosphere by this blow-off valve.

As soon as the receiver pressure falls the blow-off valve automatically get closed and compressor run normally.

Prob . 11 – A two stage air compressor with perfect inter-cooling takes in air at 1 bar and 27 °C. The law of compression in both the stages is $PV^{1.3}$ = constant. The compressed air is delivered at 9 bar from the HP cylinder to an air receiver.

Calculate per Kg of air,

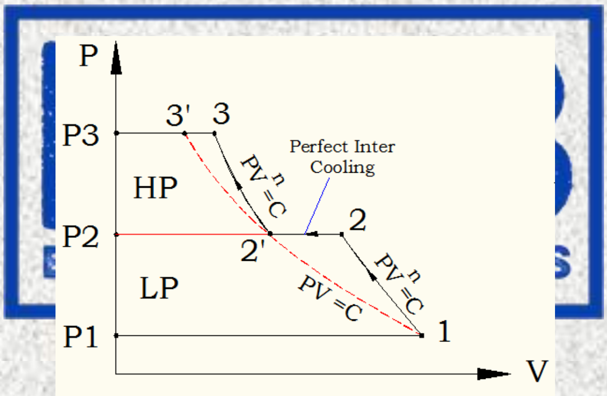
i) Minimum work of compression

ii) Heat rejected in intercooler

iii) Work required for single stage compression to the same deliver pressure.

SPPU : May – 18, 7-Marks

Ans. – Two Stage Reciprocating Compressor, Perfect Inter-cooling, K = No. of stages = 2, P1 = 1 bar = $1x10^5$ N/m², T1 = 27 °C = 300 K, $PV^{1.3}$ = constant, P3 = 9 bar = $9x10^5$ N/m².



i) Minimum work of compression –

Considering ideal intermediate pressure (P2) and perfect inter-cooling,

$$P2 = \sqrt{P1\ P3}$$

$$P2 = \sqrt{1\ x\ 9}$$

$$\boldsymbol{P2 = 3\ bar = 3\ x\ 10^5\ N/m^2}$$

We know, the minimum work of two stage compression is,

$$WD_{poly} = K\ \frac{n}{n-1}\ (m\ R\ T1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$WD_{poly} = 2\ x\ \frac{1.3}{1.3-1}\ (m\ x\ 287\ x\ 300)\left[\left(\frac{3}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$\frac{WD_{poly}}{m} = WD_{poly\_per\ Kg} = 2\ x\ \frac{1.3}{1.3-1}\ (287\ x\ 300)\left[\left(\frac{3}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$\boldsymbol{WD_{poly\_per\ Kg} = 215324\ J\ per\ Kg\ of\ air = 215.324\ KJ\ per\ Kg\ of\ air\quad ....Ans.1}$$

ii) Heat rejected in intercooler –

We know, in polytropic compression process 1-2, the pressure ratio in terms of temperature is,

$$\frac{P2}{P1} = \left(\frac{T2}{T1}\right)^{\frac{n}{n-1}}$$

$$\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} = \left(\frac{T2}{T1}\right)$$

$$T2 = T1\ x\ \left(\frac{P2}{P1}\right)^{\frac{n-1}{n}}$$

$$T2 = 300\ x\ \left(\frac{3}{1}\right)^{\frac{1.3-1}{1.3}}$$

$$\boldsymbol{T2 = 386.57\ K}$$

With the perfect inter-cooling, the temperature of air after inter-cooling brought to initial intake temperature, thus,

$$\boldsymbol{T2' = T1 = 300\ K}$$

Heat rejected per Kg of air to the inter-cooler is,

$$Q_{rej} = m\ \ C_{p\_air}\ \ (T2 - T2')$$

$$\frac{Q_{rej}}{m} = Q_{rej\_per\ Kg} = C_{p\_air}\ \ (T2 - T1)\quad .....Assume, C_{p\_air} = 1\ \ KJ/KgK$$

$$Q_{rej\_per\ Kg} = 1\ \ (386.57 - 300)$$

$$\boldsymbol{Q_{rej\_per\ Kg} = 86.57\ KJ\ per\ Kg\ of\ Air\quad ....Ans.2}$$

iii) Work required for single stage compression to the same deliver pressure –

If the compression take place in single stage to the same delivery pressure, then the work required is,

$$WD_{poly} = \frac{n}{n-1}\ \ (m\ R\ T1\ )\left[\left(\frac{P3}{P1}\right)^{\frac{n-1}{n}} - 1\right]$$

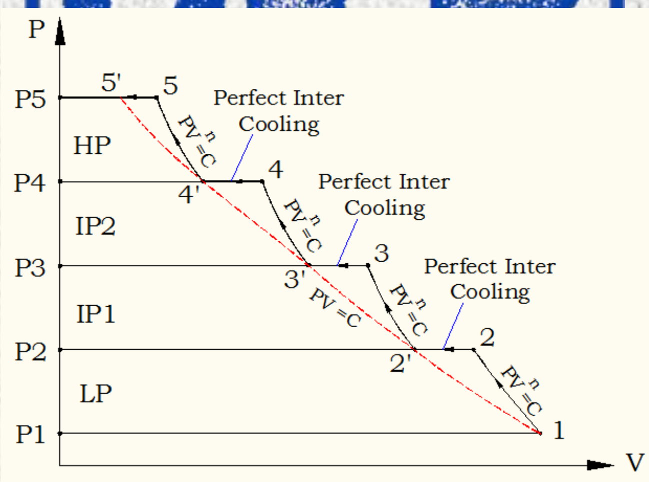
$$WD_{poly} = \frac{1.3}{1.3-1} \ (m\ x\ 287\ x\ 300)\left[\left(\frac{9}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$\frac{WD_{poly}}{m} = WD_{poly\_per\ Kg} = \frac{1.3}{1.3-1} \ (287\ x\ 300)\left[\left(\frac{9}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$\boldsymbol{WD_{poly\_per\ Kg} = 246391\ J\ per\ Kg\ of\ air = 246.391\ KJ\ per\ Kg\ of\ air \quad ....Ans.3}$$

Prob . 12 – A reciprocating air compressor has four stage compressions with 2 m³/min of air being delivered at 150 bar when initial pressure and temperature are 1 bar and 27 °C respectively. Compression occur polytropically following polytropic index of 1.25 in four stages with perfect inter-cooling between stages. For the optimum inter-cooling conditions determine the intermediate pressures and the work required for driving compressor. SPPU : Dec.–17, May-17, 7-Marks

Ans. – Four Stage Reciprocating Compressor, $\dot{V}1 = 2\ m^3/min.$, P5 = 150 bar = $150x10^5$ N/m², P1 = 1 bar = $1x10^5$ N/m², T1 = 27 °C = 300 K, n = 1.25, Perfect Inter-cooling, K = No. of stages = 4, $PV^{1.25}$ = constant.



i) Intermediate Pressures when perfect inter-cooling and ideal intermediate pressures conditions –

We know, for compression in no. of stages with ideal intermediate pressures with perfect inter-cooling is,

$$y = \sqrt[K]{\frac{P_{delivery}}{P_{intake}}} = \frac{P2}{P1} = \frac{P3}{P2} = \frac{P4}{P3} = \frac{P5}{P4}$$

$$....\ K = no.of\ stages\ of\ compression,$$

$$P1 = intake\ pressure,$$

$P5 = delivery\ pressure$

$$y = \sqrt[4]{\frac{150}{1}} = \frac{P2}{P1} = \frac{P3}{P2} = \frac{P4}{P3} = \frac{P5}{P4}$$

$$y = \sqrt[4]{\frac{150}{1}}$$

$$\mathbf{y = 3.4996}$$

$$y = \frac{P2}{P1}$$

$$3.4996 = \frac{P2}{1}$$

$$\mathbf{P2 = 3.4996\ bar}$$

$$y = \frac{P3}{P2}$$

$$3.4996 = \frac{P3}{P2}$$

$$3.4996 = \frac{P3}{3.4996}$$

$$\mathbf{P3 = 12.2472\ bar}$$

$$y = \frac{P4}{P3}$$

$$3.4996 = \frac{P4}{P3}$$

$$3.4996 = \frac{P4}{12.2472}$$

$$\mathbf{P4 = 42.8603\ bar}$$

ii) Work required for driving compressor –

We know, work done for K no. of stages when perfect inter-cooling and ideal intermediate pressure is maintained,

$$WD_{poly} = K\ \frac{n}{n-1}\ (P1\ V1)\left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$WD_{poly} = 4\ x\ \frac{1.25}{1.25-1}\ \left(1x10^5\ x\ \frac{2}{60}\right)\left[\left(\frac{3.4996}{1}\right)^{\frac{1.25-1}{1.25}} - 1\right]$$

$$\mathbf{WD_{poly} = 18980.38\ Watt = 18.98\ KW}$$

14. Rotary Compressor :

Rotary compressors are the devices which develop pressure and have a rotor as their primary element which actually rotates in the casing, as compared to reciprocating compressor in which piston slides inside the cylinder.

These compressor are used for supplying large volume of air upto 3000 m^3/min. but at a very low pressure upto 10 bar.

The compressor of air follows the law $PV^n = constant$, but the index of compression may be as high as 1.7 if no cooling device are used, if the inter-cooler used in between two stages then value of index of compression can be reduced which approximates adiabatic compression. The compressor running speeds are upto 40000 rpm.

14.1. Classification of Rotary Compressor :

Rotary Compressor

- Positive Displacement Compressor
  1. Roots Blower
  2. Vane Type Blower
  3. Screw Compressor
- Non-Positive Displacement Compressor
  1. Radial or Centrifugal
  2. Axial Flow

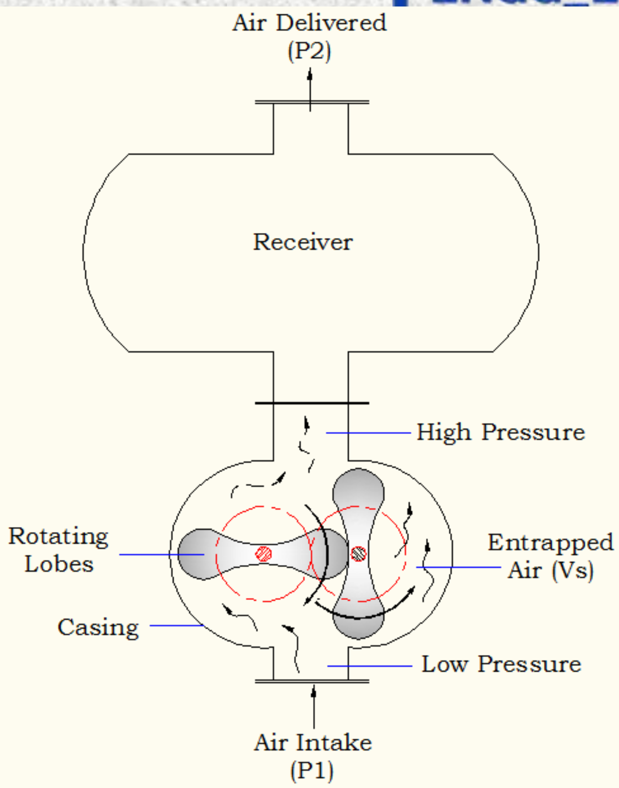
14.2. Positive Displacement Compressor :

In positive displacement rotary compressor, the air is trapped in between two sets of engaging surfaces and change in pressure is either by the back flow of air or both by squeezing action and back flow of air.

The single stage blowers have pressure ratio upto 2.5 bar with the air handling capacity ranges from 0.5 m^3/min. to 1500 m^3/min.

- **Root Blower (Lobe Type) Compressor -** SPPU : May-18, Dec.-15, May-14, 6-Marks

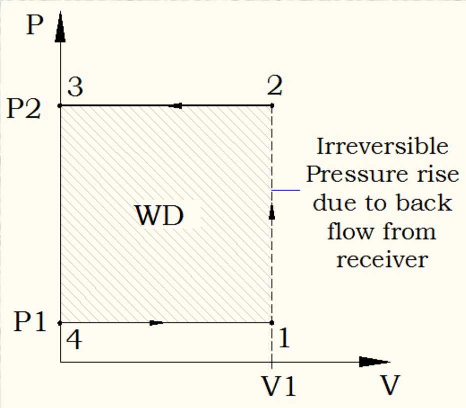
It consist of two rotors rotating in opposite directions. The lobes of the rotors are of epicycloids, hypocycloid or involutes profiles because this ensures correct mating.



The high pressure side for all angular positions of the rotors a small clearance is provided between the rotors and the cylinder surface to reduce wear. The leakage through this clearance increases with increasing pressure ratio and reduces the efficiency of the compressor.

The volume of air V_s at atmospheric pressure P1 is entrapped between the left hand rotor and casing the volume of the air once entrapped does not dreases from entry to exit and therefore pressure is not developed till the exit port is uncovered. As the exit port opns some high pressure air from receiver will rush back and mix with the air volume 'Vs' irreversibly until receiver pressure P2 if the volume of receiver is assumed large. The air is then transferred to the receiver. This happens 4 times in one revolution in case of two lobed rotor and 6 times in case of 3 lobed rotor.

The root blower has a pressure 1 : 2 and capacity around 8.5 m3/min. with speed of 250 rpm.

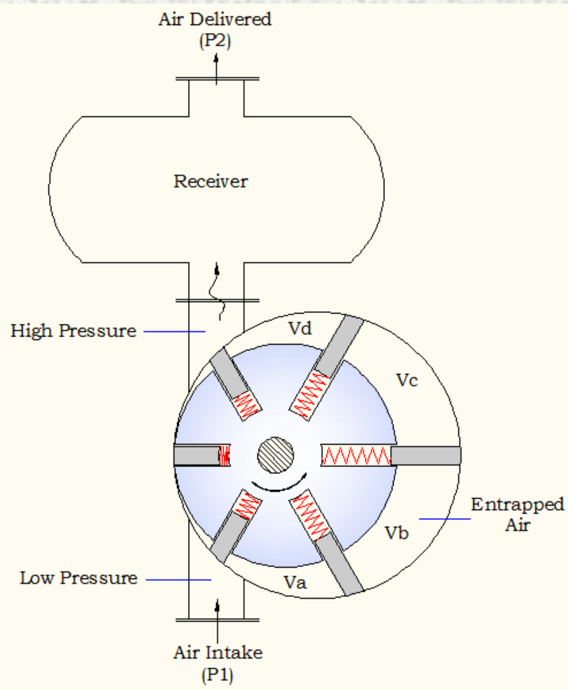


The work done per revolution of rotor is,

$$WD = (P2 - P1)\,Vs$$

- **Vane Type Blower Compressor -**

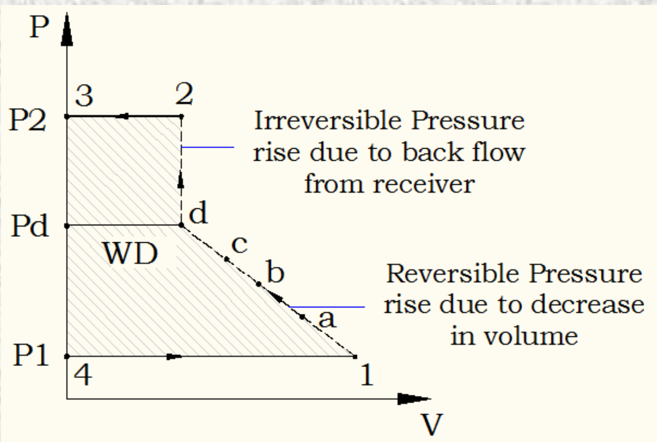
Vane type blower consists of a rotor located eccentrically in a cylindrical outer casing.



The vane type blower compressor require less work compared to root blower for the same capacity and pressure rise. The capacity of air deliver upto 350 m3/min. at pressure ratio 4 and speed at rotation is about 250 rpm.

The rotor carries a set of spring loaded vanes in the slots of the rotor. The volume of air V1 at atmospheric pressure P1 is entrapped between two vanes as in root blower.

At the rotation proceeds, the entrapped air is first compressed irreversibly from condition 1 to d as compression takes place due to the decrease in volume provided for the entrapped air. Then the air is compressed from the pressure Pd to P2 irreversibly. This irreversible compression is just similar to the compression of root blower.



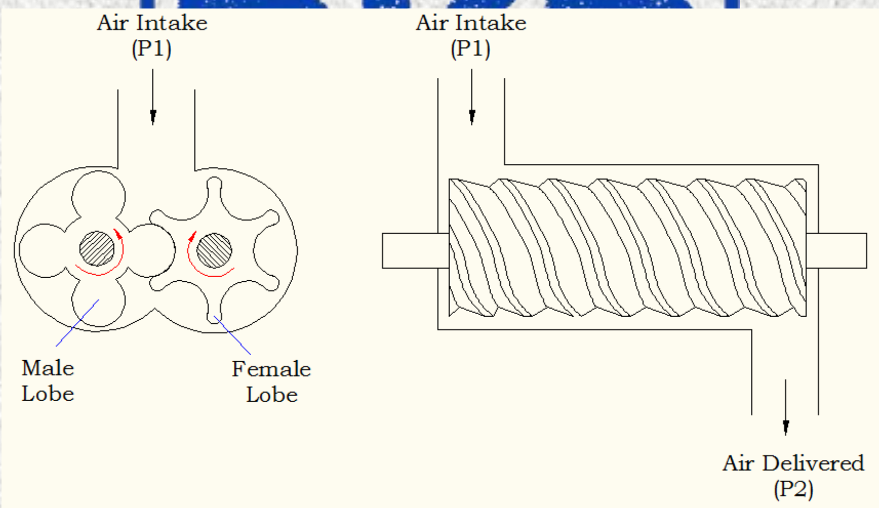
- **Screw Type Compressor -**

The screw type compressor consist of two helical grooved rotors which are mesh with each other in a housing. The electric motor drives a male rotor shaft and female follows it but in the some designs the metallic cylindrical rotor.

Suction – voids are created on both male and female side of rotor and gas is drawn in through the inlet port when the rotor is in unmesh condition. Due to continuous rotation of rotor inter-lobe space increase in size, and gas flow continuously inside the compressor, just before to the point at which the inter-lobe space leaves the inlet port, the entire length of the inter-lobe space completely filled with gas.

Compression – during forward motion of the rotor screw the male rotor lobe mesh with another female inter-lobe. The air is trapped between the space from suction end and it moves forward in axial direction causing compression by decreasing the volume of trapped air in inter-lobe space and hence pressure is gradually increased to the discharge end.

Discharge – as air reaches to discharge end and discharge port is uncovered, the compressed air is discharges by further meshing of lobe and inter-lobe space. Due to overlap cycle of suction and discharge it gives smooth and continuous flow of air without creating noise.



14.3. Non-Positive Displacement (Steady Flow) Compressor :

In non-positive displacement or steady flow compressor, the fluid is not positively contained within specified boundaries, but is in a continuous steady flow through out the machine. There is a transfer of energy from the machine rotor to the fluid and the rise in pressure is primary due to dynamic effects. Thus these compressor also called as Dynamic Compressor and Blade Compressor.

- **Centrifugal Compressor -**

The centrifugal type compressor are used to supply large quantities of air but at a lower pressure ratio. It consist of a rotating impeller, diffuser and casing. It can produce the pressure ratio upto 4 : 1 per impeller and with number of impellers pressure ratio can reach upto 10 : 1. These compressor can be adapted for air flows ranging from 15 m^3/min. to 1200 m^3/min.

- **Axial Flow Compressor -**

In axial flow compressor, air essentially flows parallel to the axis. It consist of a number of rotating blades rows fixed on a rotating drum and stator blades rows fixed on casing. Each stage consists of one moving row of blades and one fixed row of blades. This type of compressor can give a pressure ratio of 1.2 to 1.3 per stage. A pressure ratio of 10 : 1 can be obtained by using multi-stage. The number of stages used vary from 4 to 16. These compressor can be adapted for air flows ranging from 1200 m^3/min. to 42000 m^3/min.

15. Comparison between Reciprocating and Positive Displacement type Rotary Compressor : SPPU : May-17, Dec.-17, May–16, 6-Marks

Positive Displacement Reciprocating Compressor	Positive Displacement Rotary Compressor
It can deliver large pressure ratio upto 5 in single stage and upto 300 in multi-stage compressor.	It can deliver pressure ratio upto 2.5, in some multi-stage application upto 10.
Compression of air takes place with the help of piston and cylinder arrangement with reciprocating motion of piston.	Compression takes place due to rotary motion of blades or rotors.
These are used to deliver low volume flow rate upto 30 m^3/min. but at high pressure ratio.	These are used to deliver large volume flow rate upto 1500 m^3/min. but at low pressure ratio.
The pressure rise is due to variation in volume of cylinder by means of moving boundary formed by piston.	The pressure rise is due to either by back flow of air as in case of root blower or due to decrease in volume and back flow as in case of vane blowers.
It has more number of moving parts, more wear and tear, more lubrication and maintenance required.	It has less number of moving parts, less wear and tear therefore less lubrication and maintenance is required.
More starting torque is required.	Less starting torque required and step-less capacity control.
Mechanical efficiency is low as compared to rotary compressor.	Mechanical efficiency is high as compared to rotary compressor.
These have suction and delivery valves.	These have inlet and outlet ports.

It gives intermittent delivery of air. (if it is not a type of double acting).	It gives continuous delivery of air.
It have relatively high vibrations, thus overall installation cost is more.	It have relatively low vibrations, thus overall installation cost is less.
It required receiver compulsory.	It does not required receiver compulsory.
These are run at low speeds.	These are runs at high speeds.
These are either air cooled or water cooled to reduced the work of compression.	These are not required any type of cooling.

Exercise

1. Derive an expression for volumetric efficiency of a reciprocating compressor.
2. Define, i) Isothermal Efficiency, ii) Volumetric Efficiency.
3. Explain vane compressor with a neat sketch.
4. Derive an expression for intermediate pressure for two stage compression with perfect inter-cooling.
5. Explain with a neat sketch, any one capacity control method used for reciprocating compressors.
6. Explain actual indicator diagram of a single stage reciprocating compressor.
7. Explain the factors which affect volumetric efficiency of a reciprocating air compressor.
8. Discuss various methods of increasing isothermal efficiency of the compressor.
9. Explain the function of inter-cooler and after-cooler in case of reciprocating air compressor.
10. Explain with neat sketch any one of rotary compressor.
11. Explain root blower compressor with neat sketch.
12. Define, i) Isothermal Efficiency, ii) Free Air Delivery.
13. Define, i) Pressure Ratio, ii) Clearance Volume Ratio.

14. What are the advantages of multi-staging in reciprocating air compressor.

15. Differentiate between reciprocating and rotary compressor.

16. Prove the volumetric efficiency of an air compressor is given by $\eta_v = 1 + C - C\left(R_p\right)^{1/n}$.

17. Explain the screw type rotary compressor.

18. Derive the expression for minimum work of compression in two stage air compressor.

19. Derive the size of cylinder for a single acting single stage compressor consuming 35 KW. Also calculate mean effective pressure. Intake conditions are 1 bar and 15 °C and polytropic index is 1.3, speed is 100 rpm and mean piston speed is 152 m/min. delivery pressure is 6 bar. ***Ans. [L = 0.76 m, D=0.397 m, Pm = 2.2 bar]***

20. A single cylinder reciprocating compressor has a bore of 120mm and a stroke of 150 mm and is driving at a speed of 1200 rpm. It is compressing CO2 form a pressure of 120 KPa and temperature of 25 °C to a temperature of 215 °C. Assuming polytropic compression index as 1.3, no clearance and 100 % volumetric efficiency, Take R = 0.189 KJ/KgK. Calculate, i) pressure ratio, ii) indicated power, iii) mass flow rate. ***Ans. [R_p = 8.47, IP = 11.25 KW, m = 4.33 Kg/min.]***

21. A single stage, single acting reciprocating air compressor delivers 0.7 Kg of air per min. at 6 bar. The suction temperature and pressure are 25 °C and 1 bar. The bore and stoke of the compressor are 100 mm and 150 mm respectively. The clearance is 3% of swept volume. Assuming index of compression and expansion to be 1.3. Find, i) volumetric efficiency of the compressor, ii) power supplied to drive the compressor if mechanical efficiency is 85%, iii) speed of the compressor. ***Ans. [η_v = 91 %, P = 2.6 KW, N = 463 rpm]***

22. An ideal reciprocating compressor has a displacement volume of 14 liters and clearance volume of 0.7 liters. It receivers air at 1 bar and displaces at 5 bar. The compression is polytropic with index 1.3 and expansion is isentrpic with index 1.4. Determine the net cycle work. ***Ans. [WD = 2413.5 Nm]***

23. Two stage compressor is used to compress air from 1 bar to 16 bar. The compression is as per the law $PV^{1.25}$=C. the temperature of air at inlet of compressor is 300K. Neglecting the clearance and assuming perfect inter-cooling.

Find out the indicated power in KW to deliver 5 m^3/min. air measured at inlet conditions and find intermediate pressure also. ***Ans. [$P_{intermedicate}$ = 4 bar, IP = 26.63 KW]***

24. A single acting, two stage air compressor with perfect inter-cooling delivers 15 m3/min of air at 25 bar pressure. The suction conditions are at 1 bar and 15 °C. the compression follows the law $PV^{1.25}$ = Constant. Calculate, i) indicated power, ii) FAD, iii) isothermal efficiency. ***Ans. [IP = 94.93 KW, FAD = 15.32 m3/min., η_{iso} = 84.7%]***

RKB
ENGG_EDU TUTORIALS