Unit 1

INTERNAL COMBUSTION ENGINES (7ME5-11)

SYLLABUS

1	**Introduction:** Objective, scope and outcome of the course.
2	**History of IC engines:** Nomenclature, Classification & Comparison, SI & CI, 4stroke- 2 stroke, First Law analysis, Energy Balance. Fuel air cycles, Actual cycles.
3	**Testing & Performance:** Performance parameters, Measurement of operating parameters e.g. speed, fuel & air consumption, Powers,IHP, BHP, FHP, Efficiencies Thermal, Mechanical, Volumetric, Emission Measurement, Indian & International standards of Testing, Emission.

CHAPTER 1

INTRODUCTION

This chapter introduces and defines the internal combustion engine. It list ways of classification of engines and terminology used in engine. Descriptions are given of engine components and basics of four stroke and two stroke cycle for both spark ignition engine and compression ignition engine.

1.1 Engine

At one time, the primary source of power for the work was man's muscles. Later animals were trained to help and afterwards the wind and running stream were yoked. The great step was taken when man learned the art of energy conversion.

An engine is a device which transforms one form of energy into another form of energy. While transforming energy from one form to another, an important role plays by the efficiency of conversion. Most of engines convert thermal energy into mechanical energy.

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

Heat engines may be classified into two categories:

(i) External combustion engines.

(ii) Internal combustion engines.

1.1.1 External Combustion engines (E.C. engines)

In this engine combustion of fuel takes place outside the cylinder as in case of steam engine. In steam engine the heat of combustion is employed to generate steam which is used to move a piston in a cylinder. The examples of external combustion engine are hot air engines, steam turbines and closed gas turbines. External combustion engines are generally used for driving locomotives, ships, generation of electric power etc.

1.1.2 Internal combustion engines (I.C. engines)

The internal combustion engine is a heat engine that converts chemical energy of a fuel into mechanical energy, made available on a rotating shaft. Chemical energy of fuel is first converted to thermal energy by means of combustion of fuel with oxygen from the air occurs within the cylinder of the engine. This thermal energy raises the pressure and temperature of the gases within the engine and high pressure gas then expands against the mechanical mechanism of the engine. This expansion is converted by mechanical linkage of engine to a crankshaft. The crankshaft (output of engine) is connected to a transmission or power train to transmit the rotating mechanical work to the desired use. This will often be the propulsion of a vehicle for engine(I.e. locomotive, auto mobile, marine or airplanes) other application include stationary engine drive generation or pumps and portable engines for things like saw, chain and lawn movers. The internal combustion engines group includes engines employing mixtures of combustible gases and air known as gas engines, those using lighter liquid fuel spirit known as petrol engine and those using heavier liquid known as oil compression ignition or diesel engine.

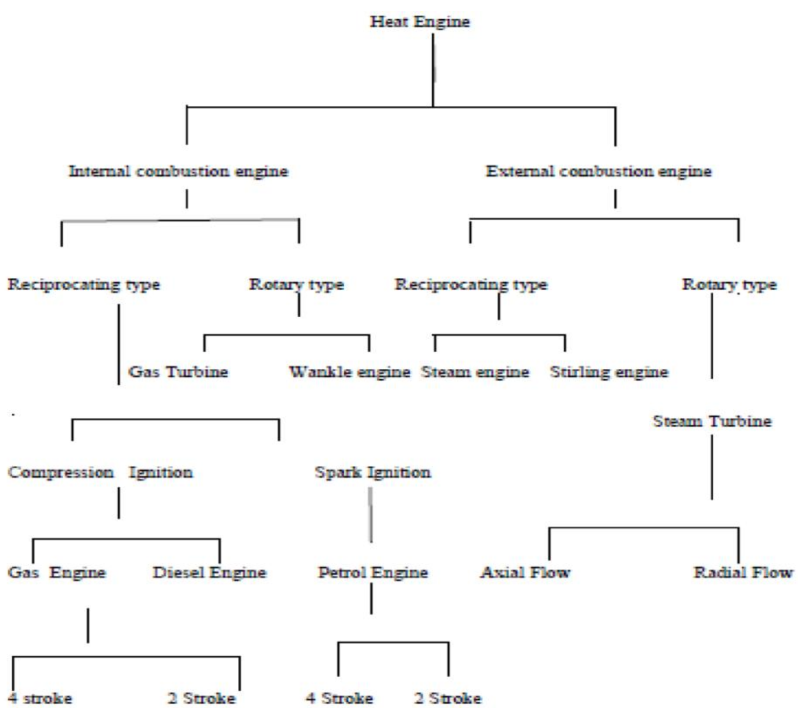


Fig. 1.1: Classification of heat engine

1.1.3 Advantages of reciprocating internal combustion engines over external combustion engines

(i) Mechanical simplicity
(ii) Overall efficiency is high
(iii) Weight to power ratio is generally low
(iv) Initial cost low
(v) Easy starting from cold condition
(vi) Less space is required due to compact unit

1.1.4 Advantages of External combustion engines over internal combustion engines

(i) Starting torque is high
(ii) Cheaper fuels can be used. Even solid fuel can be used.
(iii) It is possible to have flexibility in arrangement due to external combustion of fuel
(iv) External combustion engines are self-starting with the working fluid, where as in case of internal combustion engines, some additional equipments are used.

1.2 Historic Atmospheric engines

Most of very earliest internal combustion engines of the 17th & 18th centuries can be classified as atmospheric engines. There were large engines with a single cylinder and piston. Combustion was initiated in the open cylinder using various fuels which were available. Same early steam engines also were atmospheric engines. The open cylinder was filled with hot steam instead of combustion. The end was then closed and steam was allowed to cool and condense.

In 1859, the discovery of crude oil in Pennsylvania finally made available the development of reliable fuels which could be used in these new engines. Up to this time the lack of good fuels was drawback in engine development. Improved hydro carbon products began to appear as early as the 1860 and gasoline, lubricating oils and internal combustion engine evolved together.

1.2.1 Development of I.C. Engines

During the second half of the 19th century, many different styles of internal combustion engines were built and tested. The first fairly practical engine was invented by J.J.E Lenoir which appeared in 1860. Several hundred of these engines were built during next decade with power up to 4.5 KW and up to 5% mechanical efficiency.

The Otto Langen engine was first introduced in 1867 with efficiency improved to about 11%. This was a type of atmospheric engine. Otto was given credit when his proto type engine was built in 1876, although many people were working on four stroke cycle engine. In the 1880, the internal combustion engine first appeared in Automobile. Also the two stroke cycle engine became practical and was manufactured in large number in this decade. Rudolf Diesel, by 1892 had perfected his compression ignition engine after year of development work which included the use of solid fuel in his early experimental engines into basically the same diesel known today. Early diesel engines were noisy, slow, large single cylinder engines. Multi cylinder compression ignition engines were not made small enough to be used with automobile and trucks until 1920. Wankle's first rotary engine was tested at NSV, Germany in 1957. The practical stirling engines in small number are being produced since 1965.

1.3 Classification of I.C. Engines

Internal combustion engines may be classified as given below:-

1. According to cycle of operations
   - (i) Four Stroke cycle engines
   - (ii) Two Stroke cycle engines
2. According to cycle of combustion
   - (i) Otto cycle
   - (ii) Diesel cycle
3. According to method of ignition
   - (i) Spark ignition
   - (ii) Compression ignition
4. According to method of cooling
   - (i) Air cooled engine
   - (ii) Water cooled engine
5. According to method of governing
   - (i) Hit and miss governing engine

(ii) Quality governing engine

(iii) Quantity governing engine

6. According to speed of engine

   (i) Low speed engine

   (ii) Medium Speed engine

   (iii) High speed engine

7. According to valve arrangement

   (i) Over head valve engine

   (ii) L-head type engine

   (iii) T-head type engine

   (iv) F-head type engine

8. According to their uses

   (i) Stationary engine

   (ii) Marine engine

   (iii) Portable engine

   (iv) Automobile engine

   (v) Aero engine etc.

9. According to number of cylinder

   (i) Single cylinder engine

   (ii) Multi cylinder engine

10. According to fuel employed

   (i) Oil engine

   (ii) Gas engine

   (iii) LPG engine

   (iv) Petrol engine

   (v) Kerosene engine

   (vi) Dual fuel engine

11. Method of fuel input

   (i) Carburetor

   (ii) Fuel injector

12. According to arrangement of cylinder

(i) Single cylinder

(ii) In line or Straight engine

(iii) V-engine

(iv) Opposed cylinder engine

(v) W-engine

(vi) Opposed piston engine

(vii) Radial engine

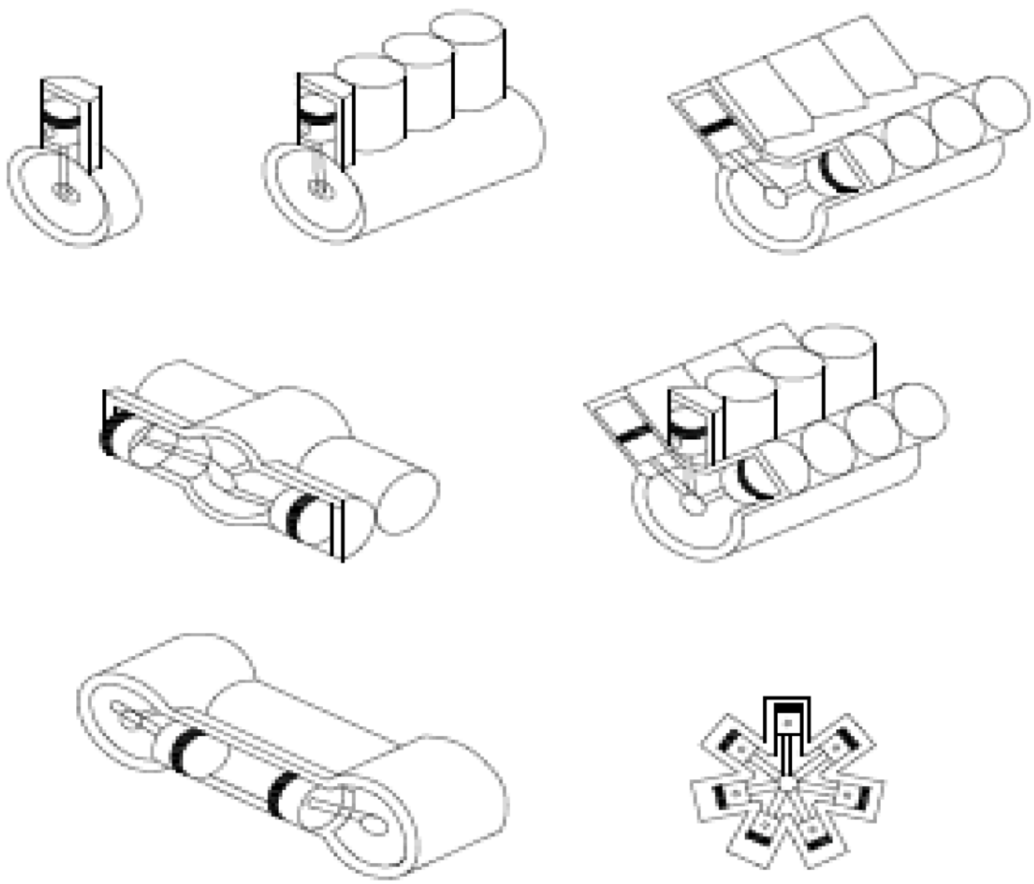


Fig. 1.2: Engine Classification by Cylinder Arrangement

(a) Single Cylinder (b) In-line, or straight (c) V engine (d) Opposed cylinder (e) W engine (f) Opposed piston (g) Radial (mark)

1.4 Application of I.C. Engine

Internal combustion engines are generally used for:

(i) Road vehicles (e.g. cars, buses, scooters, motorcycles etc.)

(ii) Locomotives

(iii) Aircraft

(iv) Pumping sets

(v) Scrapers, Power shovels, bull-dozer etc.

(vi) Several industrial application

Applications of various engines are listed below:

(i) Small Four stroke Petrol engine

These engines are primarily used in automobiles.

These are also used in pumping sets and mobile electric generator sets.

(ii) Four Stroke diesel engines

The four stroke diesel engine is manufactured in diameter ranging from 50 mm to 600 mm with speeds ranging from 100 to 4400 rpm, the power delivered per cylinder varying from 1 to 1000 KW.

Diesel engine is used for the following

(a) Pumping sets

(b) Construction machinery

(c) Air compressors

(d) Drilling jigs

(e) Tractors

(f) Jeeps

(g) Cars and taxies

(h) Electric generator plant

(i) Diesel electric locomotives

(j) Boats and ships

(iii) Small two-strokes petrol engine

These engines are employed where simplicity and the low cost of the prime mover are primary consideration. The 50 c.c. engines develop maximum brake power of about 1.5 KW at 5000 rpm and used in moped. The 100 c.c. engine develops maximum brake power of 3 KW at 5000 rpm and is used in scooters.

The 150 c.c. engine develops maximum brake power of about 5 KW at 5000 rpm and is used in motorcycles.

The 250 c.c. engine develops maximum brake power of 9 KW at 4500 rpm and is used in motorcycles.

(iv) Two Stroke diesel engines

These engines having very high power and are usually used for ship. Propulsion and have bores above 60 cm inflow with exhaust valves or loop scavenged.

1.5. Working Cycles

Most of Internal combustion engines operate on either a four stroke cycle or two stroke cycles. These basic cycles are standard for all engines with only slight variations.

An internal combustion engine can operate on any of following cycles

(a) Constant Volume or Otto cycle
(b) Constant Pressure or Diesel cycle
(c) Dual combustion cycle

(a) Constant Volume or Otto Cycle:

Heat is supplied at constant volume so it is called constant volume cycle. Petrol, gas and oil engine work on this cycle. The proper mixing of petrol and air take place in the carburetor in petrol engine. The proportionate mixture is drawn into the cylinder during the suction stroke. Air and gas are mixed outside the engine cylinder in gas engine. In light oil engine the fuel is converted to vapours by a vaporiser which received heat from the exhaust gases of engine.

(b) Constant Pressure or Diesel cycle

Air is drawn in the engine during suction stroke in this cycle during compression stroke, air gets compress and pressure and temperature of air are increased. A metered quantity of fuel is injected in the form of fine sprays by a fuel injector just before the end of compression stroke. Due to high pressure and temperature of the air the fuel is ignited and work obtained by the hot gases.

(c) Dual Combustion cycle

Heat is added partly at constant volume and partly at constant pressure so it is called dual cycle. Air is drawn in the cylinder during suction stroke. The air is compressed in hot combustion chamber during compression stroke. The heat of compressed air and combustion chamber ignited the fuel. The fuel is injected at the end of compression stroke and continued until the point of cut off is reached. The burning of fuel takes place at constant volume first and continues to burn at constant pressure during the first part of expansion.

1.5.1 Four Stroke S.I. Engine

1. First Stroke

   Induction or Intake stroke: During this stroke piston moves from top dead centre (IDC) to bottom dead centre (BDC) with the intake valve open and exhaust valve closed. This increases volume in the combustion chamber, which in turn creates a vacuum. The air enters into the cylinder due to pressure difference through the intake system from atmospheric pressure on the outside to the vacuum on the inside.

2. Second Stroke

   Compression Stroke - In this stroke intake valve closes and the piston travels back to TDC with all valves closed. This compresses the air fuel mixture. Due to compression, pressure and temperature both are increased. At the end of compression stroke, the spark plug is ignited and combustion is initiated.

3. Combustion

   Combustion of air fuel mixture occurs in a very short but finite length of time with the piston near TDC. It starts near the end of compression stroke slightly before TDC and lasts into the power stroke slightly after TDC. Due to combustion temperature is increased in the cylinder and raises the pressure in cylinder to a high peak value.

4. Third Stroke

   Expansion Stroke or Power Stroke - In this stroke both valves are closed. The high pressure created by combustion pushes the piston away from TDC. As the piston moves from TDC to BDC, cylinder volume is increased. This is the stroke which produces the work output of the engine cycle.

5. Exhaust blow down

At the end of power stroke the exhaust valve is opened and exhaust blow down occurs. In the cylinder, pressure and temperature are still high relative to the surrounding and a pressure difference is open to atmospheric pressure due to pressure difference most of the hot gases (exhaust gas) to be pushed out of the cylinder through exhaust system when the piston is near BDC. Opening the exhaust valve before BDC reduces the work obtained during the power stroke but is required because of the finite time needed for exhaust blow down.

6. Fourth Stroke

Exhaust Stroke - when piston reaches BDC, exhaust blow down is completed, but the cylinder is still full of exhaust gases. The piston travels from BDC to TDC in the exhaust stroke. This pushes most of the exhaust gases out of the cylinder into the exhaust system at about atmospheric pressure. At the end of exhaust stroke before TDC, the intake valve starts to open, so that it is fully opened by TDC when the new intake stroke starts the next cycle. Exhaust valve finally is fully closed some time after TDC. When both the valves are opened for some time it is called valve overlap.

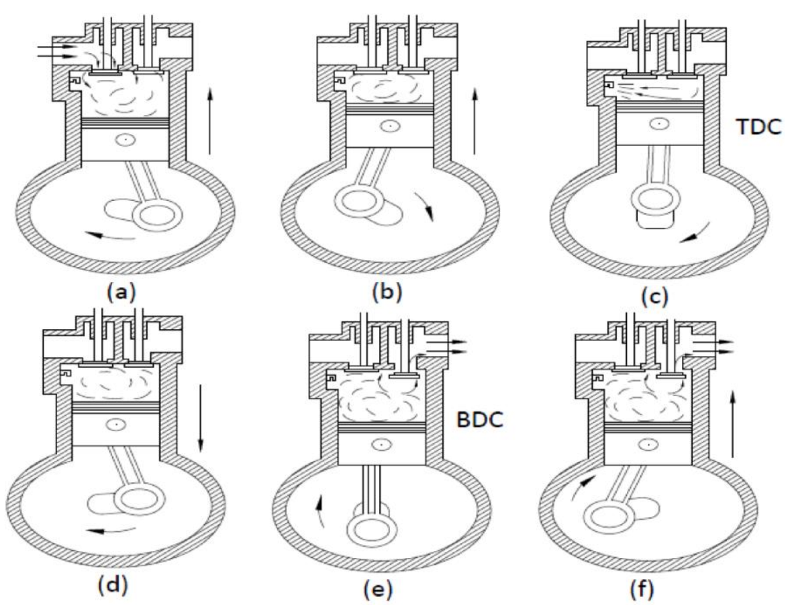


Fig. 1.3: Four- stroke SI engine operating cycle

(a) Intake stroke. Ingress of air-fuel as piston moves from TDC to BDC. (b) Compression stroke. Piston moves from BDC to TDC. Spark ignition occurs near end of compression stroke. (c) Combustion at almost constant volume near TDC. (d) Power or expansion stroke. High cylinder pressure pushes piston from TDC towards BDC. (e) Exhaust blow down when exhaust valve opens near end of expansion stoke. (f) Exhaust stroke.

1.5.2 Four Stoke C.I. Engine

1. First Stroke

   Intake Stroke: In this stroke the piston travels from TDC to BDC with the intake valve opened and exhaust valve closed. This creates an increasing volume in combustion chamber. Due to the differential of pressure difference from atmospheric, air is induced into the cylinder. In this stroke only air is induced in the cylinder, no fuel is added to the incoming air.

2. Second Stroke

   Compression Stroke: In this stroke only air is compressed and compression is to tend higher pressure and temperature. At the end of compression stroke fuel is injected directly into the combustion chamber, where it mixes with very hot air. Due to high pressure and temperature, fuel is evaporated and self ignited and combustion occurs.

3. Combustion

   Combustion is fully developed and continues at about constant pressure until fuel injection is completed and the piston moves from TDC to BDC.

4. Third Stroke

   Power Stroke: At the end of the combustion piston travels from TDC to BDC. Power is produced due to pressure and temperature drop.

5. Exhaust Blow down

   At the end of power stroke exhaust valve is opened and exhaust blow down occurs. At this point, pressure and temperature are high relative to surrounding. Due to this pressure difference hot gases pushed out of the cylinder through exhaust system when the piston is near BDC. This exhaust gases carry away a high amount of enthalpy, which lowers the cycle thermal efficiency.

6. Fourth Stroke

Exhaust Stroke: When piston reaches BDC, exhaust blow down is completed. Exhaust valve is opened and piston moves from BDC To TDC in this stroke. This pushes most of remaining exhaust gases out of cylinder. At the end of exhaust stroke before TDC, the intake valve starts to open, so that it is fully open by TDC when new intake stroke starts the heat cycle.

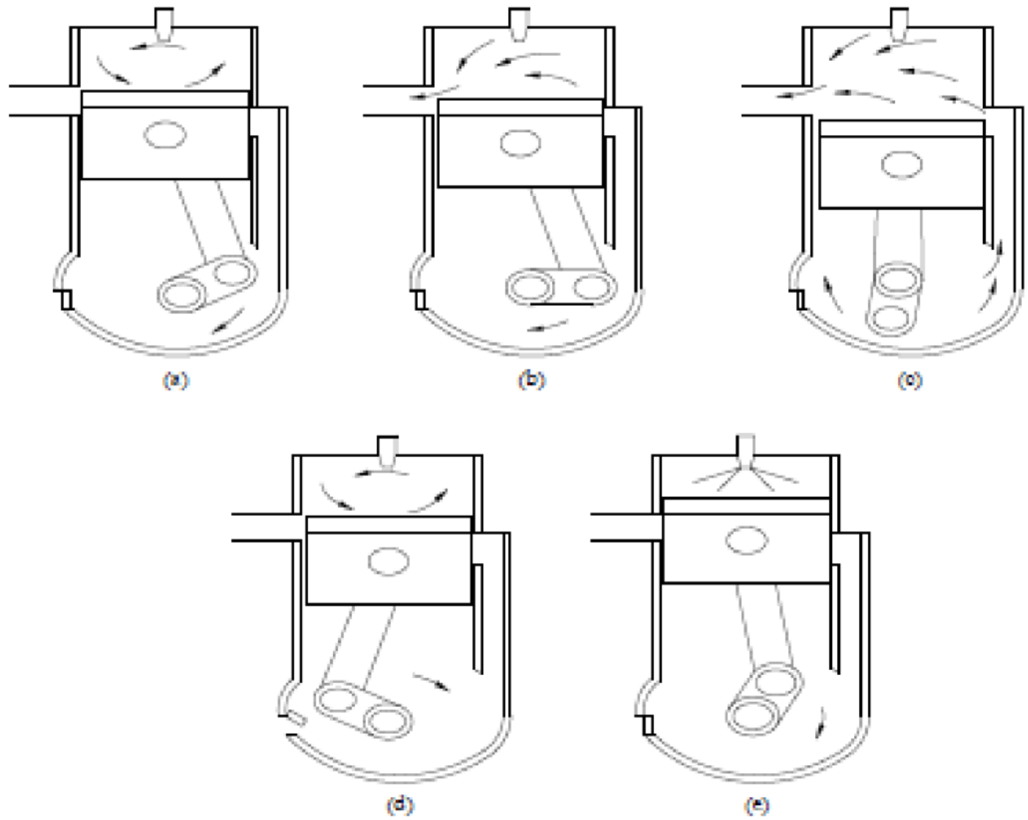


Fig. 1.4: Two–stroke SI engine operating cycle with crankcase compression

(a) Power or expansion stroke. High cylinder pressure pushes piston from TDC towards BDC with all ports closed. Air in crankcase is compressed by downward motion of piston. (b) Exhaust blow down when exhaust port opens near end of power stroke. (c) Cylinder scavenging when intake port opens and air fuel is forced into cylinder under pressure. Intake mixture pushes some of the remaining exhaust out the open exhaust port. Scavenging lasts until piston passes BDC and closes intake and exhaust ports. (d) Compression stroke. Piston moves from BDC to TDC with all ports closed intake air fills crankcase. Spark ignition occurs near end of compression stoke. (e) Combustion at almost constant volume near TDC.

1.6 Two Stroke Cycle Engine

In 1878, A British engineer Dugald clerk introduce a cycle which could be completed in two strokes of piston rather than four strokes. The engines using this cycle were called two stroke cycle engines. Instead of valves, ports are used in two stroke engine.

1.6.1 Two Stroke S.I Engine

1. First Stroke:

Expansion or Power Stroke: Very high pressure created by the combustion process forces the piston down in the power stroke. As the piston travels towards BDC, the expanding volume of combustion chamber causes decrease in pressure and temperature .

2. Exhaust blow down

The exhaust valve opens 75 BDC and blow down occur. As the piston approaches BDC, a slot in the side of the cylinder which is uncovered.

3. Intake and Scavenging

At about 50 before BDC, the intake slot on the side of the cylinder is opened and intake air-fuel enters under pressure. Fuel is mixed to the air with either a carburetor or fuel injection. Mixture pushes much of remaining exhaust gases out the open exhaust valve and fills the cylinder with a combustible air fuel mixture. The piston passes BDC and quickly covers the intake port and then the exhaust part. The higher pressure at which air enters the cylinder is established in one or two ways.

4. Second Stroke

Compression Stroke: The piston travels towards TDC and compresses the air fuel mixture to a higher pressure and temperature with all valves is closed. At the end of the compression stroke, the spark plug is ignited by the time the piston gets to TDC; combustion occurs the next engine cycle begins.

1.6.2 Two Stroke CI Engine cycle

In a two stroke Diesel cycle engine all the operations are the same as in the spark ignition engine with the differences: firstly in this case, only air is admitted into cylinder instead of air fuel mixture and secondly fuel injector is used and combustion is initiated by self-ignition.

1.7 Comparison of Four Stroke and Two Stoke engine

Table 1.1: Comparison of Four Stroke and Two Stoke engine

S. No.	Four Stroke engine	Two Stroke Engine
1	The cycle is completed in four stroke of piston and two revolution of the crankshaft. One power stoke is obtained in every two revolution.	The cycle is completed in two strokes of the piston or in one revolution of the crankshaft. One power stroke is obtained in each revolution of crankshaft.
2	Turning movement is not so uniform and hence heavier fly wheel is needed.	More uniform turning movement and hence lighter fly wheel is needed.
3	Because of one power stoke for two revolutions, power produced for same size of engine is small or for the same power heavy and bulky engine.	Because of one power stroke for one revolution, power produced for same size of engine is more or for the same power, the engine is light.
4	Lesser cooling and lubrication requirement because of one power stroke in two revolutions.	Greater cooling and lubrication requirement because of one power stroke in one revolution.
5	Lesser rate of wear and tear	Great rate of wear and tear.
6	Four Stroke engine contains valve and valve mechanism.	Two stroke engines contains points (some engines are fitted with exhaust valve)
7	Initial cost is high, because of	Due to light weight and absence of

	heavy weight and complication of valve mechanism.	valve mechanism, cheaper in initial cost.
8	Volumetric efficiency higher due to more time of induction.	Due to lesser time of induction, volumetric efficiency low.
9	Higher thermal efficiency.	Lower thermal efficiency.
10	Four stroke engine used where efficiency is important such as cars, buses, trucks, tractors, airplane, power generations etc.	Two stroke petrol engines used in very small size engine such as lawn mover, scooters, motor cycles two stroke diesel engines used in ship propulsion.

1.8 Engine component

Some major components of engine with their functions & materials process of production are described below.

(i) **Cylinder block**

The cylinder block is the main unit in engine. It produces support & covering to most of major components of the engine. The casing of such cylinder engine is cast as a single unit and is known as cylinder block. The top segment is called cylinder head. The cylinder head & cylinder block have passage for water coming from water jackets in case of water cooled engine and have fine outside the engine casing in the care of air cooled engine. The cylinder head & cylinder block are quality fitted to each once by a number of bolts and have head gas kit in between for the air sealing. In the bottom portion of the cylinder there is a sump of lubricating oil two portions is known as crank case. The inner surface of cylinder block under cylinder head is machined finished accurately to poison size is called bore of force. The cylinder is made of cast iron & steel the same reason of damping vibration & substring components and is made by casting.

(ii) **Cylinder**

As per the name it is a cylinder vessel or a space cylindrical in shape in which piston move reciprocally. The valuing volume created in the cylinder during the operation of engine is filled with working fluid. The only fluid in the form of mixture of air & atomized fuel in case of petrol engine; atomized fuel in case of diesel engine. The cylinder is made in side in the cylinder block by the process of casting is forming forward by the machining process for the material used are cast iron & aluminum alloy.

(iii) Piston

It is a cylinder component that fits into cylinder. It forms the moving boundary of combustion system. Normally made up of cast iron, cast steel & aluminum alloy. The aluminum alloy has the advantage of higher thermal conductivity & lower specific gravity.

(iv) Piston Rings

The piston rings are housed in the circumferential grooves provided on the outer surface of the piston. It gives gastight fitting between the piston & cylinder and prevents the package of gases at high pressure. They are made of special grill cast iron. The material detains its classic property at very high temp. The upper piston rings are called compression rings & the lower piston rings are called the oiling rings.

(v) Combustion chamber

It is the space formed in upper part of cylinder, due to enclosure or piston top & cylinder load. Combustion of fuel & released thermal energy in the chamber, build up pressure in upper part of cylinder.

(vi) Intake manifold

Path from which air-fuel mixture is taken up to inlet valve.

(vii) Exhaust manifold

Passage through which burnt gases from combustion chamber escapes into atmosphere. Passage starts with exhaust valves & ends as since.

(viii) Valves

Combustion chamber contains two valves at the cylinder head. One is input valve, which contribute the admission of the charge into the petrol engine & Air into diesel engine during suction stroke of engine. Second one is exhaust valve used to discharge exhaust gases into atmosphere.

Here we can't commit about the size of the valve, but one valve is bigger than the other.

* If we use input valve bigger in size, we can get air-fuel mixture or air fast in the cylinder & due to small size of exhaust valve the built gases can exhaust at fast pail into atmosphere.

* If we use input valve smaller in size, we can get mixture or air at high pressure in cylinder, provides us the better & uniform combustion. Due to good mixing & motion, here exhaust valve is large in size results into less time blockage of burnt gases, hence no red hot zone.

(ix) Spark plug

Function of spark plug is to make mixture attain the self-ignition temperature. It is only used in petrol engine & generally mounted on the cylinder head.

(x) Connecting Rod

This rod connects the crankshaft with piston & transmits the reciprocating motion of the piston to crank shaft as rotary motion. Rod is designed as I –cross section for sustaining bending forces one end of the rod is of small cross section & is connected to piston with "gudgeon" pin & another end is wide connected to crank shaft by crank pin. Made by forging using steel.

(xi) Crank shaft

Both crank & crankshaft are steel forged & machined to a smooth finish. Crankshaft is supported by main bearing & has a heavy wheel, called flywheel, to even out the fluctuation of torque , the power required for any useful purpose is taken from crank shaft only. The crankshaft is the backbone of the engine. It is made up of alloy steel & SG iron & made by forging.

(xii) Gudgeon pin

This pin connects the connecting rod with the piston. It is made up of steel by forging.

(xiii) Camshaft

The function of cam shaft is to operate the intake & exhaust valves through its associated parts. These parts are pushing tools, locker arm, valve springs & toppers. This shaft also provides the direction to ignition system. The cam shaft is driven positively from crank shaft at half the speed of crankshaft.

(xiv) Cams

These are the essential part of engine designed for a desired profile to give much needed motion to the valve through the follower. It is designed in such a way to open the valves at the correct timing to keep them open for the necessary duration.

(xv) Fly wheel

It is a wheel mounted on the crankshaft which store excess energy during the power stroke & return that energy during the other strokes. The net torque impacted to the crankshaft during one complete cycle of operation of the engine fluctuates causing a change in the other velocity of shaft.

(xvi) Push rod & Rocker arm

Push rod & Rocker arm are used to transmit the motion of the cam to the valves. These links together are known as valve gear.

(xvii) Water Jacket

The water or cooling agents are circulated in this jacket.

(xviii) Bed plate

The lower portion of the crank case is known as bed-plate. These bed plates are held by the bed-bolts to concentrate foundation.

(xix) Governor

It is run by a drive from the crankshaft. The function of the governor is to regulate the charge in the case of SI engine and amount of fuel in the case of CI engine to maintain the speed of the engine constant, when the load requirement features.

(xx) Carburetor

The function of carburetor is to supply the uniform air-fuel to the cylinder of a petrol engine through intake manifold. The amount of mixture entering the cylinder is controlled by a throttle valve.

(xxi) Fuel system

a. Petrol Engine

In petrol engine purpose of fuel system is to supply fuel inside the cylinder for burning. Here fuel pump lifts petrol from the fuel tank & supplies it to carburetor. Air form surrounding also reaches the carburetor via the air cleaner. Liquid petrol atomizes & mixes with air with in carburetor via the air cleaner. Liquid petrol atomizes & mixes with air with in carburetor, then goes to cylinder for burning.

b. Diesel Engine

In diesel engine a fuel injection pump drives diesel fuel to the injector, & then diesel in vapors form is injected into the cylinder by nozzles. Diesel and air do not mix before entering into the cylinder as same in the case of petrol engine.

(xxii) Lubrication System

Lubrication system helps in reducing the friction between any two moving parts whether in rotary motion or any other. For this purpose, the lubrication oil is pumped by a geared pump or by other means and reaches all joints & external regions through passage made of it.

(xxiii) Ignition System

Function of this system is to ignite the fuel with in cylinder. When the ignition switch is made “on” current flows form battery of magnate to the ignition oil which raises low voltage to a very light valve.

1.9 TWO STROKE ENGINE

A two stroke engine is one which completes its cycle of operation in one revolution of the crankshaft or in two stroke of the piston. In this engine the functions of the intake and exhaust process of the four-stroke are taken care of by the incoming fresh charge which is compressed either in the crankcase or by means of a separate blower while the engine piston is near the bottom dead center. The engine piston needs only to compress the fresh charge and expand the products of combustion. Since a two stroke engine will have twice as many cycle per minute as four-stroke engine operating at the same speed and with the same number of number of cylinder, theoretically it will develop twice the power when operating at the same mean effective pressure. As with the four stroke engine the power output of this engine also depends upon the number of kilogram of air per minute available for combustion.

Two stroke engine is most simple in construction as there are very few moving parts and but more difficult to analyze because of overlapping the inlet and exhaust. As there are only two strokes for performing four basic operations, the overlapping processes cannot be avoided. During the downward stroke, expansion is carried out during part of the stroke and during remaining part of the stroke, the exhaust and charging are carried out simultaneously, this process is also continued during the upward motion of the remaining part of the stroke, and the compression is carried out.

1.9.1 Working principle of two strokes sparks Ignition Engine

Figure shows various components their location and at different strokes occurring in 2-storke there are three ports in it.

i) Inlet Port ii) Transfer Port iii) Exhaust Port

Valve and value operating mechanism are not used in this engine. A deflector type piston is employed, which during its reciprocating motion allows opening and closing of port. Various operational stages in its working are the following.

1) Suction stroke, when inlet port is open and mixture of atomized petrol + air enters into crank case in case of Spark Ignition Engine and Air in case of Compression Ignition Engine.

2) Compression stroke, when transfer port and exhaust port are closed, the piston is near the top of cylinder.
3) Power stroke, when sparking cause expansion of charge the piston is moving down towards BDC.
4) Exhaust stroke, when exhaust port is open. At this junction, the transfer poet is also open and a fresh charge is being supplied with into the cylinder. Due to deflector shaped piston the charge goes upward and not to the E.P. however, escape of some un-burnt fuel cannot be ruled out.

The location of Exhaust Port is made a little higher then transfer port 1 to 2 mm. high this is done so with the intention that the fresh charge does not go out of exhaust port without burning.

However this does not provide a full proof arrangement and the loss of fresh charge is major deficiency in 2-stroke engines

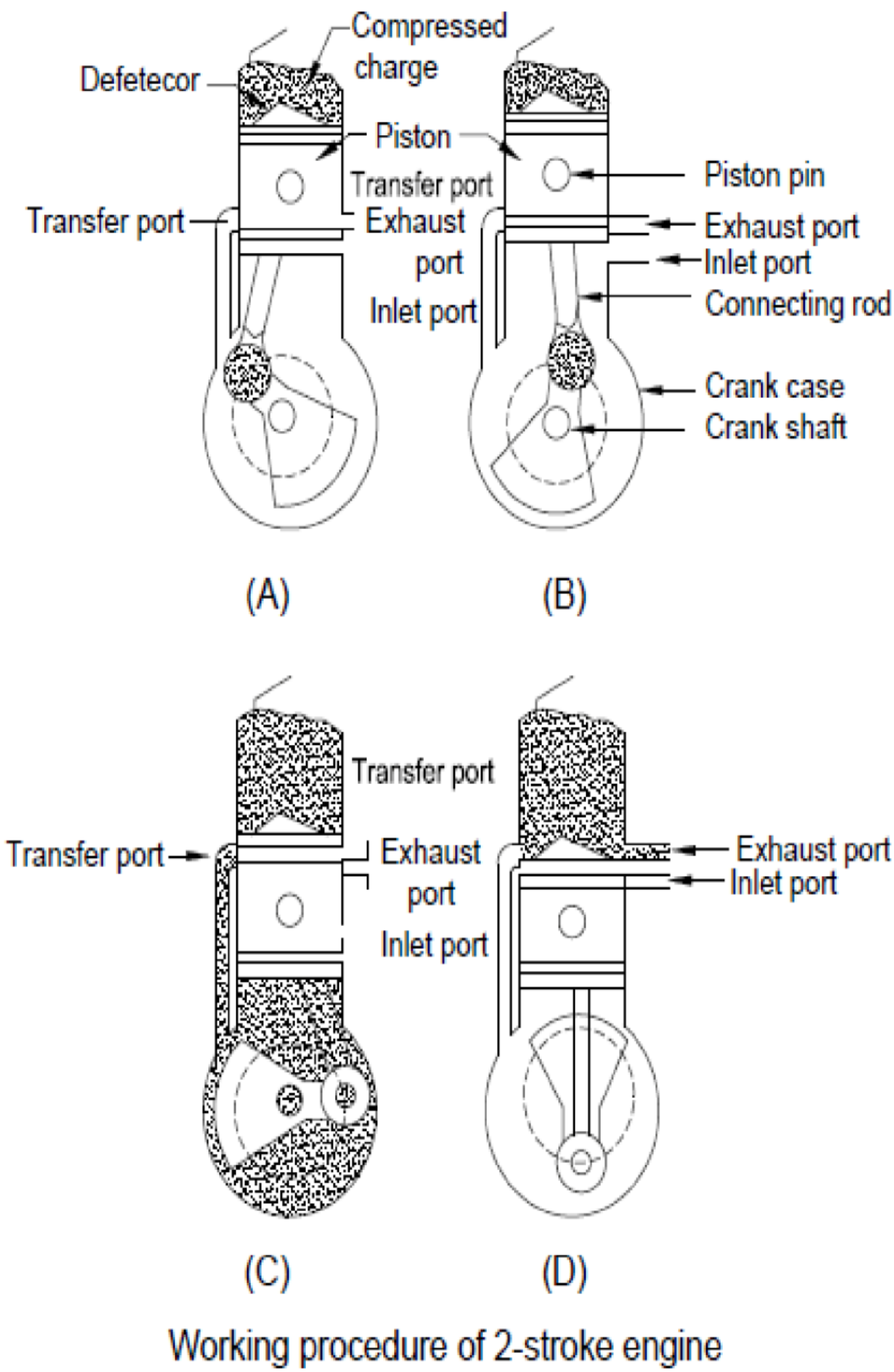
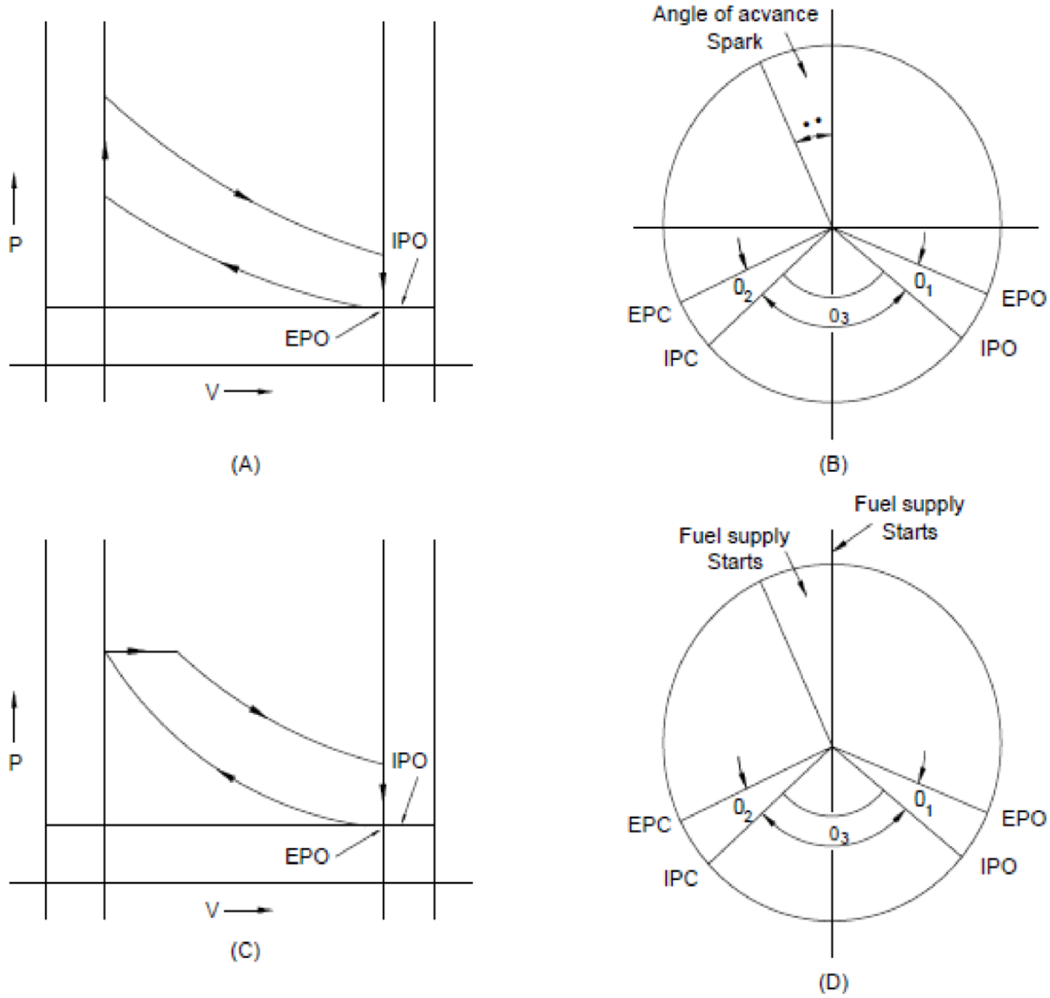


Fig. 1.4: Working procedure of 2-stroke engine



A , B - Part Hming diagram for SI Engine
C , D - Part Hming diagram for CI Engine

Fig. 1.5: Port Timing Diagrams

As shown in figure, the overlapping of the port is represented by the angle θ_3 the value of θ_3 for diesel engine is higher than the value of petrol engine as there is no risk of losing the fuel as only air is supplied to the engine during the charging whereas, the mixture is supplied in the petrol engine so the overlap angle (θ_3) should be as small as possible to avoid the loss of fuel as some mixture is carried with the exhaust.

All cases are taken in designing of the engine that the loss of mixture should be as minimum as possible. Equal care is also taken to avoid the loss of air in diesel engine because otherwise, the exhaust gases will be removed effectively and completely but the amount of air left inside the engine cylinder will be less and power output of the engine is considerably reduced.

1.9.2 Types of two stroke engine

Two stroke engines can be categorized in two types depending on scavenging methods.

1) Crank case scavenged engine
2) Separately scavenged engine

1.9.2.1 Crank Case Scavenged Engine

In this engine, the charge is compressed in the crankcase by the underside of the piston during the expansion stroke.

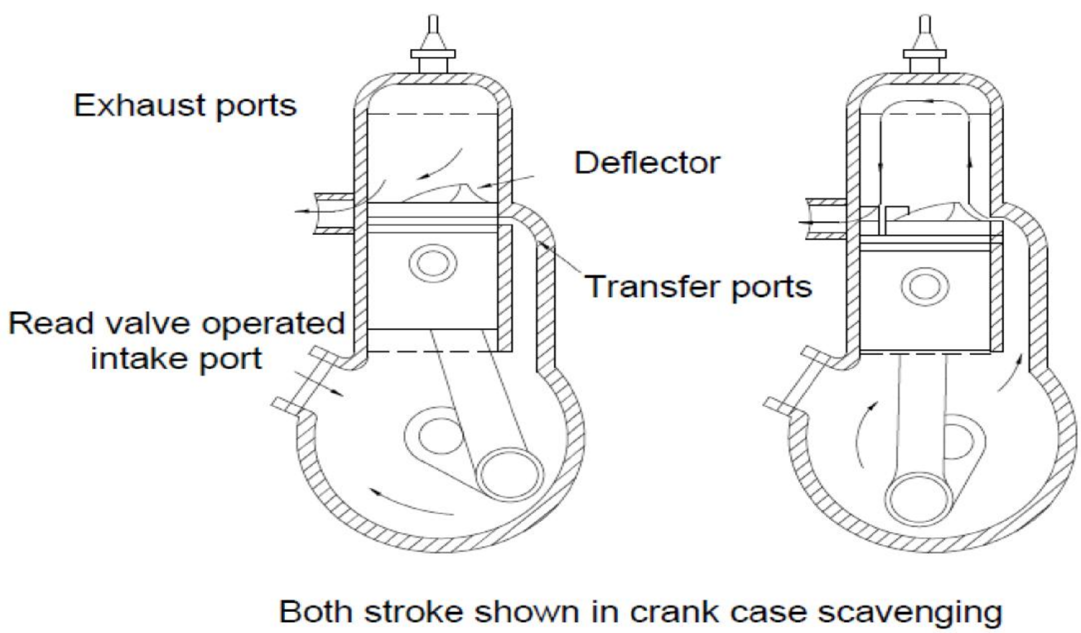


Fig. 1.6: Both strokes in crankcase scavenging

The compressed charge passes through the transfer port into engine cylinder flushing the products of combustion the process is called scavenging & this type of engines are called the crank case scavenged engines.

Firstly exhaust port opened up & all the combustion product escape into atmosphere resulting the cylinder pressure drop to atmosphere when the piston goes down this downward motion of piston also un cover the transfer port which will let the fresh mixture from the crankcase to the engine cylinder.

The port & top of piston is aligned in a way that will refer fresh change toward the cylinder head & this will result in force escape of combustion products. Also it will minimize the flow of fresh air or mixture through outlet port.

The top of piston which is projected outward is called crown or deflector. Again when piston start moving from BDC to TDC turn output & transfer port will be closed & input port opened, it help fresh mixture to enter in crank case. Also the fresh mixture which went into cylinder in previous stroke will get compressed & heated up. Ignition & expansion take place in the usual manner & cycle is repeated. Due to the flow restriction in the input reed valve & the transfer ports the engine gets changed with less than one cylinder displacement volume.

1.9.2.2 Separately Scavenged Engine

In this engine a blower is attached to the inlet port of engine. Here the Air-fuel mixture is supplied at a slightly higher pressure. When the piston start moving from top to Bottom Dead center. The exhaust port opened up at exactly 60^0 before BDC & air starts escaping. After 10^0 when the pressure goes significantly down, the input port got opened which results in input of fresh mixture. Here the inlet ports are designed in such a way that incoming air get divested toward the cylinder head on the inlet side & moves down on exhaust side forming a loop before reaching the exhaust. Piston deflators are not used as they are heavy & tried to become over heated at high output.

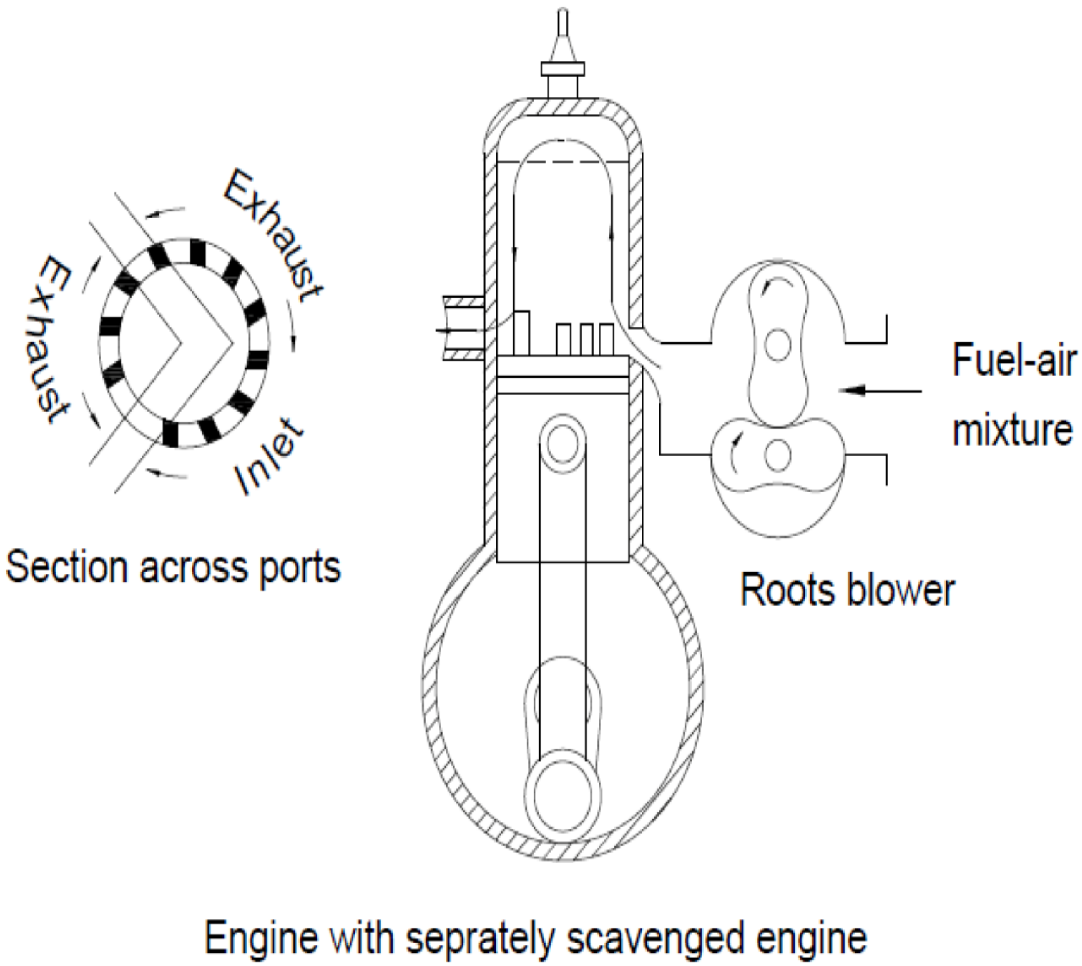


Fig. 1.7: Engine with separately scavenged engine

1.9.3 Terminologies

In absence of a terminology that is commonly accepted or at least commonly understands, it is necessary to explain and define the lines used in connection with two stroke cylinder. In this part a list of terms used will be explained. The term can be easily understood from the diagram showing the charging process of the engine. The flow of gases though a two stroke cycle engine is represented in fig. the hatched areas represents fresh air or mixture and the cross hatched areas represents combustion gases. The width of channels represents the quality of the gases expended by volume.

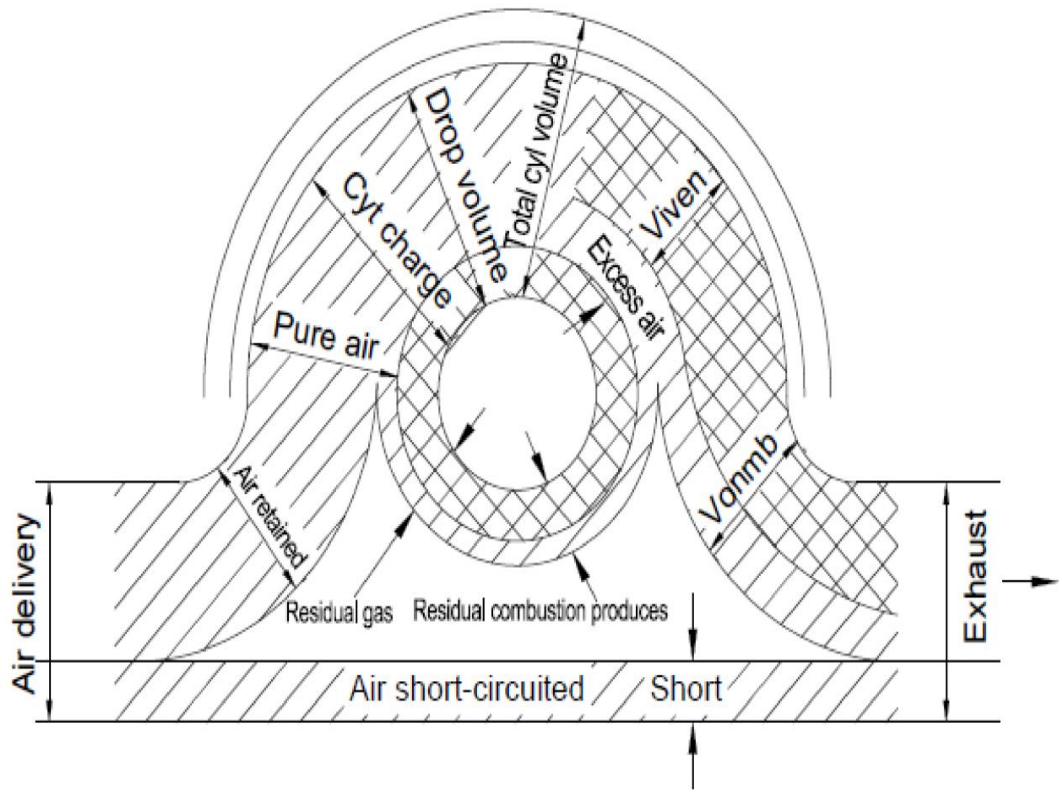


Fig. 1.8: Representation of air movement in blower

1. **Delivery Ratio (R_{del})**

In two stroke engines we normally use separate blower to supply the A/F mixture into cylinder or it could be done by the piston of the engine. The air delivered V_{del} could be more or less than the swept volume of the engine, it mainly depends on the relative capacity of the blower. This delivery ratio is defined as the ratio of volume of air delivered to the reference volume.

$$R_{del} = V_{del} / V_{ref}$$

Here according to our needs the reference volume could be total effective cylinder volume displacement volume or effective displacement volume. It is always preferred to use total effective cylinder volume, as it is only the quality of charge in the remaining total cylinder volume at exhaust port closure that enter into the combustion.

The delivery ratio could also be defined on mass basis, that is ratio between fresh air mass into cylinder to reference mass.

Hence delivery ratio is a measure of the air supplied to the cylinder relative to the cylinder connects. Delivery ratio usually varies between 1.2 and 1.5 except for crankcase scavenged engines where it is less than even one.

2. **Trapping efficiency**

Mixture delivery could be divided into two part firstly the mixture existed through the exhaust part without combustion, usually called short circuiting air, V_{short},

Second one the mixture which remains in the engine & participate in combustion, called retained mixture V_{ret}.

Hence term trapping efficiency is the ratio of mixture retained to that of the short circulating mixture & used to calculate the ability of engine to retained the mixture.

$$\eta_{trap} = V_{ref} / V_{short}$$

Here short circulating air is calculated by (1- η_{trap}) trapping efficiency is mainly controlled by the geometry of the port & overlap time.

3. **Relative cylinder charge**

Some volume of residual gases (V_{res}) remains in the engine even after exhausts of burnt gases & constitutes the charge (V_{ch}) in addition with mixture retained. Hence

$V_{cn} = V_{ref} + V_{res}$

Relative cylinder change is the ratio of volume of change in the cylinder to the reference volume.

$$C_{ref} = V_{ch} / V_{ref}$$

$$= (V_{ref} + V_{res}) / V_{ref}$$

It is a measure of ability of filling the cylinder irrespective of compositions the relative cylinder charge may be either more or less than unity depending upon the scavenging pressure and the port height.

4. Scavenging efficiency

Successful exhaust of the combustion charge from the previous cycle from the cylinder is the measure of the purity of charge shape of combustion chamber & type of scavenging are the continue of the same. The representative determination of purity is quite difficult then the experimental determination. As experimentation determination only need the analysis of gas sample through Orsat apparatus or gas analyses.

The scavenging efficiency is the measure of the success in clearing the cylinder of residual gases from the previous cycle. It indicates to what extent the residual gases in the cylinder are replaced with fresh air. If it is equal to unity, it means that all the gases existing in the cylinder at the beginning of scavenging have been swept off.

5. Charging efficiency

The power output of the engine is entirely depends on the amount of fresh air in cylinder the charging efficiency is the ratio of volume of retained air with the reference volume.

$\eta_{ch} = V_{ret} / V_{ref}$

It is measure of ability of filling cylinder with fresh mixture. Hence

$\eta_{ch} = R_{del} \ \eta_{trap.}$

It is most important aspect as entire power production of engine is entirely depends on the fresh mixture in the cylinder that can be burned and produce heat.

6. Pressure loss coefficient

The pressure loss coefficient is ratio between main upstream and downstream pressures during scavenging period and expresses the loss of pressure to which the scavenging air is subjected to when it crosses the cylinder.

1.9.4 Two stroke air capacity

Nothing is lost through exhaust port in four stroke engine as it tries to retain all one charge enters in the intake stroke, excepts which the large valve overlapping happens. Hence a air meter could be installed on the inlet value to measure air inlet in the cylinder & the engine output will be directly proportional to measured air.

But as the exhaust port is open in two strokes at the time of air coming into cylinder, so this make preserving the all the mixture in the cylinder impossible the amount of charge enters the cylinder will led to short circulating air. Hence power of two strokes not depends on the air entering the cylinder but it depends on the returned mixture in the cylinder. The charge or mixture retained in the cylinder is called the air capacity of two stroke engine.

1.9.5 Theoretical Scavenging Processes

Scavenging ability is a measure of "how much mixture is retained in the engine". Perfect scavenging, perfect mixing & compute short circuiting are the three theoretical scavenging processes.

a.) Perfect Scavenging

Ideally the combustion products & fresh mixture should not be get mixture in both manner mass as well as heat transfer during scavenging into by blower, it pusher the combustion product in the cylinder out through exhaust port at the opposite cud. It is assumed that there is no mixing of fresh charge & combustion products & the flow through the exhaust port is considered to be combustion product only. However when sufficient fuel oil has entered to fill the entire cylinder volume the flow abruptly changes form one of products to one of oil. This ideal process would represent perfect scavenging without any short calculating loss.

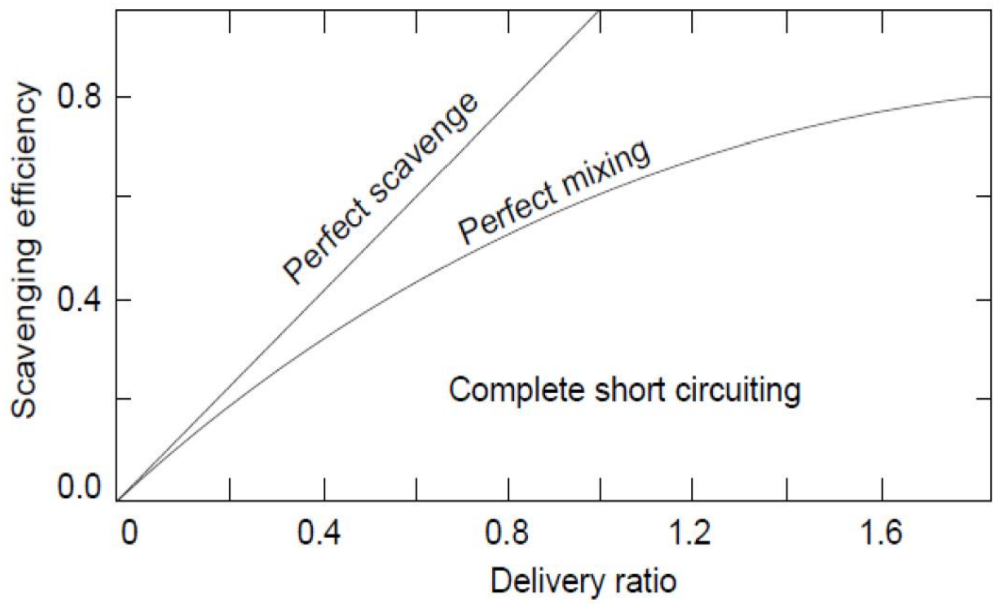


Fig. 1.9: Scavenging efficiency v/s Delivery ratio

b.) Perfect Mixing

The perfect mixing is the scavenging process in which when the fresh charge come in it mixes with the combustion products already in the cylinder thoroughly & a part of this mixture passes by exhaust port at a rate equal to the flour rate of incoming fresh mixture. This homogeneous mixture initially consist the product of combustion, and then gradually changes to fresh mixture only with the flow.

For the case of perfect mixing the scavenging efficiencies can be represented by following equation.

$\eta_{sc} = 1\text{-}e^{-Rdel}$

Where

η_{sc} - Scavenging efficiency

R_{del} - Delivery ratio

c.) Short Circuiting

This type of scavenging is highly unacceptable as the fresh mixture coming from the inlet port run directly out from the exhaust port without removing the combustion product this is dead low & must be avoided.

1.9.6 Actual Scavenging process

The actual scavenging process is neither one of perfect scavenging more of perfect mixing. It properly consists partially of perfect scavenging mixing & short circulating. Fig shows the variation of delivery ratio and trapping efficiency with crank angle for three different scavenging modes i.e. perfect scavenging, perfect mixing & intermediate scavenging.

1.9.7 Delivery ratio & trapping efficiency variation for a crankcase scavenged engine

The scavenging parameters for the intermediate scavenging is shown in fig. this represent the actual scavenging process.

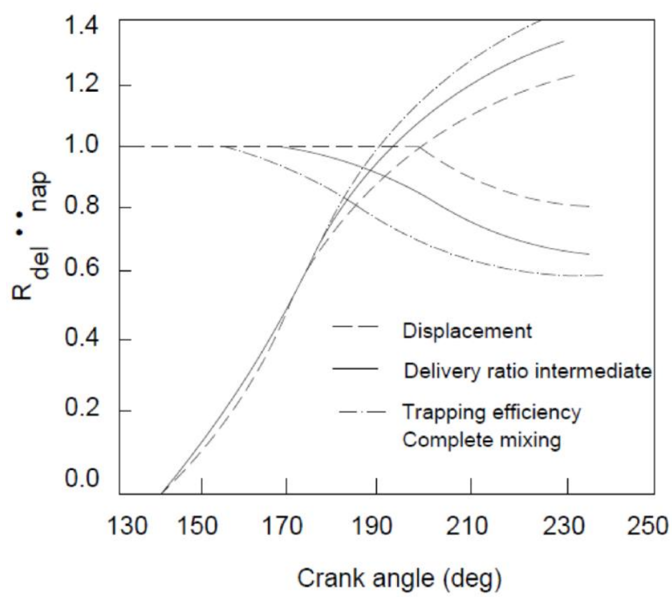


Fig. 1.10: Displacement v/s Crank angle

It can be seen from this figure that a curtain amount of combustion products is initially pushed out the cylinder without being diluted by fresh air. Gradually mixing and short circuiting cause the out.

Flowing products to be diluted by more & more fresh air until ultimately the situation is the same as for perfect mixing the first phase of scavenging process is a perfect scavenging process which then gradually changes into a compute mixing process.

1.9.8 Scavenging Process

At the end of the expansion stroke, the combustion chamber of a two stroke engine is left is because, unlike four-stroke engines, there is no exhaust stroke available to clear the cylinder. The process of cleaving the cylinder, after the expansion stroke is known as scavenging. This must be computed in very short duration available between the end of expansion stroke and start of charging process.

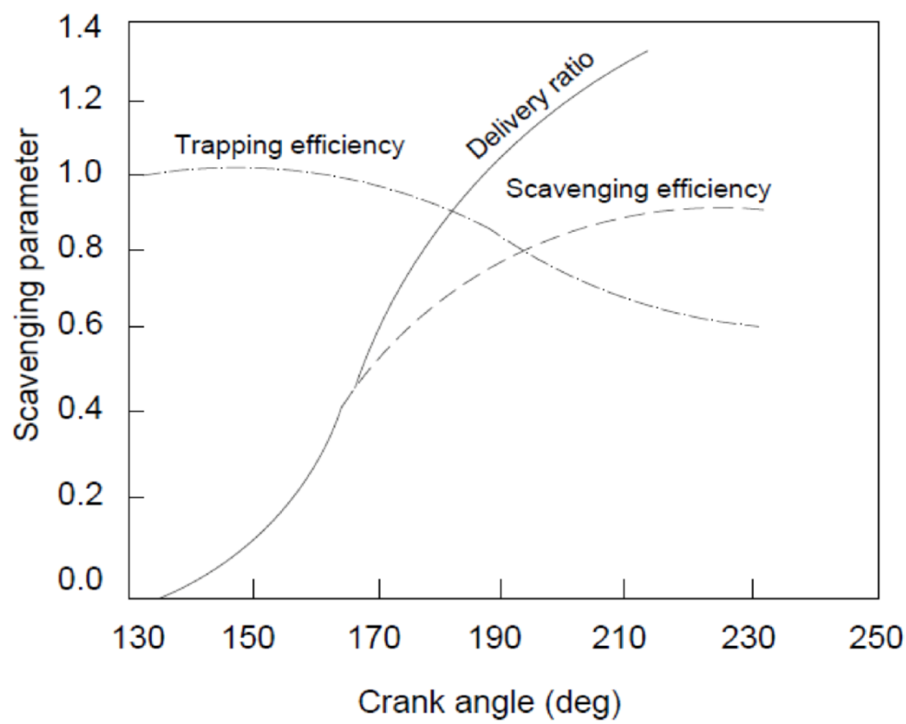


Fig. 1.11: Scavenging parameter v/s Crank angle

The efficiency of a two stroke engine depends to a great degree on the effectiveness of scavenging process, since bad scavenging gives a low mean indicated pressure and hence, results in a high weight & high lost per brake horse power for engine. Not only that the lubricating oil becomes more contaminated so that its lubricating qualities are reduced and result in increased wear of piston to higher mean temperature and grater heat stresses on the cylinder walls.

This it goes without saying that every improvement in the scavenging leads to improvement in engine and its efficiency in several direction and hence, a detailed study of scavenging process and different scavenging system in worthwhile.

1.9.8.1 Types of Scavenging process

1. Cross flow

This method was used for early crank case compression two stoke, as used for small motorcycle. In this method, the transfer port (or inlet port for engine cylinder.) and exhaust port are situated on the opposite sides of engine cylinder, so that the brunt gases would be pushed out by the cross flow. The piston crown was usually shaped with a raised rib as a deflector piston, so that the fresh change was intended to move upward into a vertical loop, then downward with the exhaust gas.

In practice, the gas flow failed to follow the idealized pattern the rib of the deflator piston also gave a poor shape for the combustion chamber, with long flame path & excessive surface area this method of scavenging has now been almost entirely replaced by loop scavenging. Although for piston ported two stroke engines, cross flow scavenging is now very common in for stroke engines, where their inlet and exhaust values head.

2. Backflow or loop

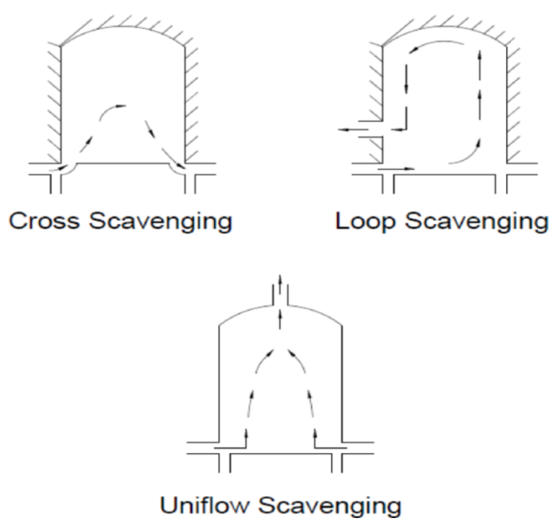


Fig. 1.12: Types of scavenging

Rather than the flow loop being vertical the gases are encouraged to move in two horizontal loops. In this method the inlet and outlet ports are situated on the same side of the engine cylinder the fresh charge, while entering into the engine cylinder forms a loop and pusher out the burnt gases.

3. Uni flow

In this method when fresh charge enters into the engine from one side or sometime from both side & pushers the exhaust product from the top of cylinders. In uniflow scavenging the fresh charge & exhaust product flows in the same direction. The uniflow method of scavenging has been widely used for two stroke diesel such as Detroit diesel series 53, 71, 92,110 and also in large ship propulsion engines.

1.9.9 Advantages of two stroke engine

1.) Here the cycle is computed in 2stroek as there is a working stoke for each revolution, this the power developed by engine in each cycle is double then the four stroke engine.

2.) The work required is quite less approx half flat of four stroke engine because of less friction

3.) Higher flywheel is required, as it has more uniform turning moment on crankshaft because it has working stroke in each revolution.

4.) Higher power to weight ratio.

5.) Lesser space required.

6.) Low maintenance.

1.9.10 Disadvantages of two stroke engine

1.) It is less efficient at high speeds because of reduced volumetric efficiency.

2.) High fuel consumption & reduced thermal efficiency.

3.) Less effective compression.

4.) High consumption of lubricating oil.

5.) With heavy loads, high heating & with light loading running of engine is not smooth because increased dilution of charge.

Illustrative Example

Example: - Obtain the engine dimensions of a two cylinder two stroke i.c. engine from following data.

Engine speed = 4000 rpm, Volumetric efficiency = .77, Mechanical efficiency = .75, Fuel consumption = 10 L/H, SP graving = .73, enthalpy of fuel= 10,500 kcal/kg. , A/F ratio = 18:1, Piston speed = 600 m/min, indicated mean effective pressure=5.

Find also the brake power output and thermal efficiency. Assume standard pressure & temp.

Solution: - Fuel consumption = 10*.73=7.3hg/hl.

Air supply/hr. = 7.3*18 = 131.4 Kg

Total charge = 131.4 + 7.3 = 138.7 Kg

Mass of charge at standard temperature & pressure corresponding to swept volume.

ɱ = Total charge / η_v = 138.7/.77= 179 Kg.

Now $P_{u} =$ ɱRT per cylinder

$1.013 * 10^5 * v = 179/2 * 267.2*298$

V = 75.5 ɱ^3/hr

Volume swept by piston per stroke.

$V_s \quad = \frac{75.5*10^6}{60*4000}$

$= 314$ cc

Piston speed = length of stroke x 2 x 4000

$\therefore \quad L = \frac{600}{2*4000} x10^2$ cm.

$= 7.5$ cm

Swept volume $\frac{\pi}{u} d^2 L = 314$

$\therefore \; d = \sqrt{\frac{4*314}{\pi*7.5}} = 7.3$ cm

HP = $\frac{2*5*314*4000}{450000}$ = 27.9 metric HP

∴ BHP = 27.9*.75 = 20.9 metric HP

Thermal efficiency = $\frac{20.9*736*3600}{10*.73.10500*4187}$

= .173 or 17.3%

1.10 ENGINE CYCLES & TESTING

A cycle is defined as a repeated series of operations occurring in the certain order. The cycle may be of imaginary actual or perfect engine. The former is called actual cycle and the latter is called ideal cycle. The cycle experienced in the cylinder of an I. C. engine is very complex. Air (in C. I. engine) or air-fuel mixture (in S I. engine) is included and mixed with the slight amount of exhaust residual remaining from previous cycle. This mixture is then compressed and combusted, due to that changing the composition to exhaust products consisting of CO_2, H_2O, and $N_2$with many other losses components. Then after an expansion process, the exhaust valve is opened and this gas mixture is expelled to surroundings. It is an open cycle with changing composition, a difficult system to analyse. The real cycle is approximated with an ideal air standard cycle which differs from the actual by following.

1. The gas mixture in the cylinder is considered as air for the cycle and properties values of air are used for analysis. When the gas composition is mostly CO_2, H_2O and N_2using air properties does not create large errors in the analysis. Air will be toward as an ideal gas with constant specific heat.
2. The open cycle is changed into a closed cycle by assuring that the exhaust gases are fed back into intake system. This works with air standard cycle as both intake and exhaust gases are air.
3. The combustion process is replaced with a heat addition.
4. The open exhaust process is replaced with a closed system heat rejection process.

   The analysis of all air standard cycle are based on following assumptions.

1. The gas in the engine cylinder is a prefect gas.
2. The compression and expansion processes are adiabatic and they take place without friction.

3. The physical constants of the gas in the cylinder are same as those of air at moderate temperature (C_p=1.005 kj/kgk, C_v=0.718 kj/kgk)
4. The cycle is treated closed with the same air always remaining in the cylinder to repeat the cycle.
5. No chemical reaction takes place in the cylinder.

1.10.1 CARNOT CYCLE

Carnot cycle has the highest possible efficiency and consists of four following operations:

a) Isothermal expansion c) Isothermal compression

b) Adiabatic expansion d) Adiabatic compression

Following are the four stages of Carnot cycle:

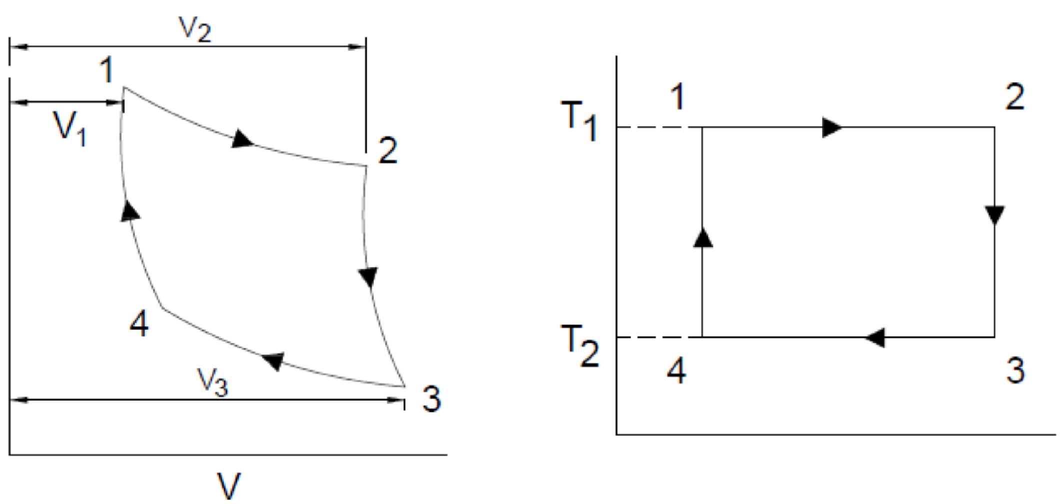


Fig. 1.13 Carnot Cycle

(a) Isothermal expansion (1-2) :- Line 1-2 represents the isothermal expansion which takes place at constant temperature $T_{1,}$ when source of heat H is supplied to the end of cylinder. Heat supplied is indicated by

R T $\log_e$ r, where r is expansion ratio.

(b) Adiabatic expansion (2-3):- Line 2-3 represents the application of non-conducting cover to the end of the cylinder. Temperature falls from T_1 to T_2 by adiabatic expansion.

(c) Isothermal compression (3-4) :- Line 3-4 represents the isothermal compression which takes place when sump is applied to the end of cylinder that is rejected in this process whose value is given by $RT_2 \log_e r$, where r is compression ratio.

(d) Adiabatic compression:-Line 4-1represents repeated application of non conducting cover and adiabatic. Compression due to which temperature increases from T_2 to T_1.

It may be noted that ratio of expansion during isothermal 1-2 and ratio of compression during isothermal 3-4 must be equal to get a closed cycle.

According to law of conservation of energy

Heat supplied = work done + Heat rejected

Work done = Heat supplied- Heat rejected

$$= RT_1 \log_e r - RT_2 \log_e r$$

Efficiency of cycle = $\frac{Work\ done}{Heat\ supplied} = \frac{R\ loge(r)\ (T1-T2)}{R\ T1\ loge\ r} = \frac{T1-T2}{T1}$

It is quite obvious that if T_2 temperature decreases, efficiency increases and it becomes 100% if T_2 absolute zero, which is impossible to attain. It is not possible to produce and engine that should work on Carnot's cycle as it would work simultaneously on isothermal and adiabatic.

Example : An engine works on Carnot cycle between 400^0 C and 40^0 C produce 150 KJ of work. Determine (i) engine thermal efficiency (ii) Heat added

Solution:- T_1=400+273=673 K

T_2=40+273=313 k

(i) Engine thermal efficiency = $\frac{673-313}{673}$=0.535

(ii) Heat added = $\frac{Work\ done}{Heat\ supplied}$=

$$0.535 = \frac{150}{Heat\ supplied}$$

Heat added= $\frac{150}{0.535}$=280.37 kj

For isothermal process

$P_1V_1=P_2V_2$

$P_2=\frac{P1V1}{V2}=\frac{20}{1.5}$=13.33 bar

For isotropic process

$P_2V_2^{\gamma}=P_3V_3^{\gamma}$

$P_3=P_{2\times}(\frac{V2}{V3})^{\gamma}$= 13.33 $(\frac{V1}{V4})^{\gamma}$

$=13.33\times(\frac{1}{6})^{1.4}$

P_3= 1.07 bar

Hence

P_1=20 bar $T_1=T_2$=673k

P_2=13.33 bar

P_3=1.07 bar $T_3=T_4$=328.3k

P_4=1.62 bar

(ii) Change in entropy

$S_2-S_1=m\ R\ \log_e(\frac{V2}{V1})=\frac{P1V1}{T1}\log_e(\frac{V2}{V1})$

$=\frac{20\times10^5\times0.20}{10^3\times273}\log_e(1.5)$

$=1\text{-}46\ \log_e(1.5)$

=0.25 kj/kgK

(iii) Mean effective pressure of cycle

$P_m=\frac{Work\ done\ Per\ cycle}{Stroke\ Volume}$

$\frac{V3}{V1}=6\times1.5=9$

Stroke Volume $V_5=V_3\text{-}V1=9\ V_1\text{-}V_1=8V_1=8\times0.20$

$= 1.6\ m^3$

$P_m = \frac{(Qs-Qr)}{V5} \times 5$ $\{Qs = T1(S2 - S1) = 673 \times 0.25 = 168.25kj$

$T_m = \frac{Qs-Qr\alpha}{V5}$

Example : In a Carnot cycle, the maximum pressure and temperature are limited to 20 bar and 400˚. The ratio of isentropic compression is 6 and isothermal expansion is 1.5. Assuming the volume of the air at the beginning of isothermal expansion as 0.20 m^3determine:-

1) The temperature and pressure
2) Change in entropy during isothermal expansion
3) Mean effective pressure of the cycle
4) Mean thermal efficiency of the cycle
5) The theoretical power if there are 200 working cycles per minutes.

Solution:-=

Maximum pressure P_1=20 bar

Maximum temperature $T_1 = T_2$=400+273=673 k

Ratio of isentropic compression $\frac{V4}{V1}$=6

Ratio of isothermal expansion $\frac{V2}{V1} = 1.5$

1) Temperature and pressure in the cycle at main points :-

$\frac{T1}{T4} = (\frac{V4}{V2})^{\gamma-1} = (6)^{1.4-1} = (6)^{(.4)} = 2.05$

$T_4 = \frac{T1}{2.05} = \frac{673}{2.05} = 328.3\ k = T_3$

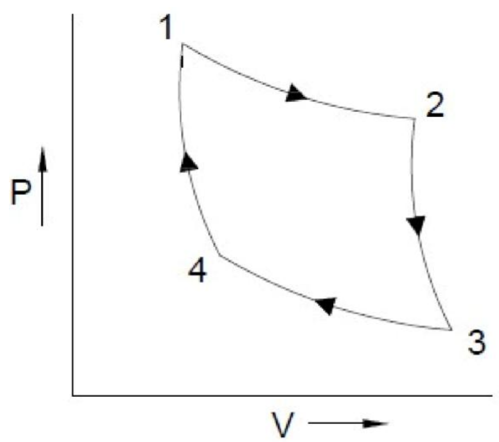
$\frac{P1}{P4}$=$(\frac{V4}{V1})^{\gamma}$=$(6)^{1.4}$ =12.29

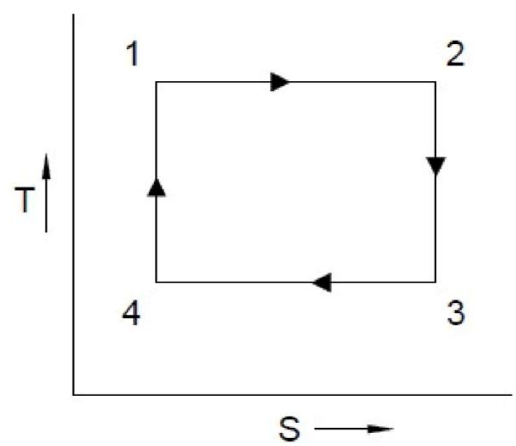
P_4 =$\frac{P1}{12.29}$=$\frac{20}{12.29}$=1.62 bar

P_m=$\frac{Qs-Qr}{V5}$ $\{Qr = T4(S3 - S4)\ = 328.3 \times$
$0.25\ = 82.05\ k$

= $\frac{(168.25-82.05)\times 10^3}{1.6\times 105}$

= 0.538 bar





(iv) Mean thermal efficiency

Heat supplied Q_s=$P_1V_1\log_e\frac{V2}{V1}$

= $T_1(S_2 - S_1)$

$= 673 \times 0.25$

= 168.25 KJ

Heat rejected = Q_r= $P_4V_1\ \log_e(\frac{V3}{V4})$

$= T_4(S_3\text{-}S_2)$

Q_r=328.3×0.25

= 82.05 K J

Efficiency $\delta = \frac{(Qs-Qr)}{Qs} = 1-\frac{Qr}{Qs}$

$= 1- \frac{82.05}{168.25}$

= 1- 0.48 = 0.52

= 52%

(v) Power of engine

Power of engine working on this cycle is given by

P= (16825-82.05)×(200/60)

= 287.33 K W

Example : A reversible engine converts one sixth of the heat input work. When the temperature of the sink is reduced by 80˚ C, its efficiency is double. Find the temperature of the source and sink.

Solution: -

Let T_1 = Temperature of source

T_2 = Temperature of sink

$$\frac{T\,1-T\,2}{T\,1}=\frac{1}{6}$$

$6\,T_1\text{-}6\,T_2 = T_1$

$5\,T_1 = 6\,T_2 =$ or$T_1 = 1.2\,T_2$

$$\frac{T1[(T2-(80+273)]}{T1}=\frac{1}{3}$$

$3\ T_1 - 3\ T_2 + 1059 = T_1$

$2\ T_1 = 3\ T_2 = 1059$

$2 \times (1.2\ T_2) = 3\ T_2 - 1059$

$2.4\ T_2 = 3\ T_2 - 1059$

$0.6\ T_2 = 1059$

$T_2 = \frac{1059}{0.6} = 1765\ k$

$T_2 = 1492^\circ C$

$T_1 = 1.2\ T_2 = 2118\ k$

$= 1845^\circ C$ Ans.

1.10.2 OTTO CYCLE OR CONSTANT VOLUME CYCLE

This cycle was concerned by Otto, so it is called Otto cycle. Petrol, gas and many types of engine work on this cycle. Theoretical P-V and T-S diagram is shown below: -

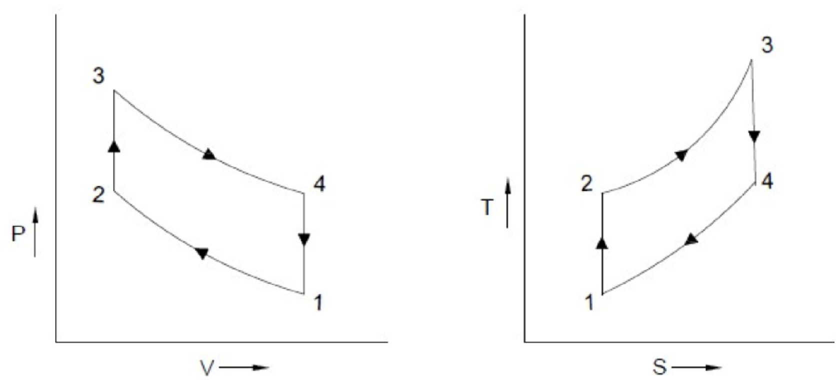


Fig. 1.14 Otto Cycle

The point 1 represents that cylinder is full of air with volume V_1, pressure P_1 and Temperature T_1,

Line 1-2 shows the adiabatic compression of air

In this process P_1, V_1, and T_1 changes to P_2, V_2 and T_2 respectively.

Line 2-3 represents the supply of heat to the air at constant volume so that $P_2$and T_3 respectively.

Line 3-4 shows the adiabatic expansion of air , during this process P_3, V_3 and T_3 change to P_4, V_4 and T_4respectively.

Line 4-1 represents the heat rejection at constant volume till original state reaches.

Consider 1 kg. of air as working substance.

Heat supplied at constant volume = C_v $(T_3\text{-}T_2)$

Heat rejection at constant volume = C_v $(T_4\text{-}T_2)$

Work done = Heat supplied – Heat rejected

$= C_v(T_3\text{-}T_2)\text{-}\ C_v\ (T_4\text{-}T_2)$

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

$= \frac{Cv\ (T3-T2)-\ Cv\ (T4-T2)}{Cv\ (T3-T2)}$

$= 1 - \frac{T4-T3}{T3-T2}$

Let Compression ratio $r_c = \frac{V1}{V2} = \frac{V4}{V3} = r$

$r_c = r_e = r$

$\frac{T2}{T1} = (\frac{V1}{V2})^{\gamma-1}$

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

$\frac{T3}{T4} = (\frac{V4}{V3})^{\gamma-1}$

T_3

$ή_{otto} = 1 - \frac{T_4\ -T_1}{T_3\ -T_2}$

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

$$= 1 - \frac{T4 - T1}{(\pi_c)^{\gamma-1}(T4 - T1\,)}$$

$$ή_{otto} = 1 - \frac{1}{r^{\gamma-1}}$$

This is the air standard efficiency of Otto cycle.

The network done per kg. in the Otto cycle can also the expressed in terms of P, V, if P is in bar, then work d

$$W = \left(\frac{P_3V_3 - P_1V_4}{\gamma-1} - \frac{P_2V_2 - P_1V_1}{\gamma-1}\right) \times 10^2 \text{ K J}$$

$$\frac{P_3}{P_4} = r^{\gamma\text{-}1} = \frac{P_2}{P_1}$$

$$\frac{P_3}{P_2} = \frac{P_4}{P_1} = r_p \qquad (r_p \text{:- Pressure ratio})$$

$$V_1 = rV_2 = V_4 = r\,V_3$$

$$W = \frac{1}{\gamma-1}\left[P_4V_4\left(\frac{P_3V_3}{P_4V_4} - 1\right) - P_1V_1\left(\frac{P_2V_2}{P_1V_1} - 1\right)\right]$$

$$= \frac{1}{\gamma-1}\left[P_4V_4\left(\frac{P_3}{P_4r} - 1\right) - P_1V_1\left(\frac{P_2}{P_1r} - 1\right)\right]$$

$$= \frac{V_1}{\gamma-1}\left[P_4((r)^{\gamma-1} - 1) - P_1((r)^{\gamma-1} - 1)\right]$$

$$= \frac{V_1}{\gamma-1}\left[((r)^{\gamma-1} - 1)(P_4 - P_1)\right]$$

$$= \frac{P_1V_1}{\gamma-1}\left[((r)^{\gamma-1} - 1)(r_p - 1)\right]$$

Mean effective pressure P_m

$$P_m = \left[P_4V_4\left(\frac{P_3V_3 - P_4V_4}{\gamma-1} - \frac{P_2V_2 - P_1V_1}{\gamma-1}\right) - (V_1 - V_2)\right] \text{ bar}$$

$$P_m = \left[\frac{\frac{P_1V_1}{\gamma - 1}(r^{\gamma-1} - 1)(r - 1)}{V_1 - V_2}\right]$$

$$= \frac{\frac{P_1V_1}{\gamma-1}[(r^{\gamma-1} - 1)(r_p - 1)]}{V_1\left(\frac{r-1}{r}\right)}$$

$$P_m = \frac{P_1 r[(r^{\gamma-1} - 1)(r_p - 1)]}{(\gamma-1)(r-1)}$$

Example : An engine of 300 mm bore and 400 mm stroke works in Otto cycle. The clearance volume is 0.01650 m^3. The initial pressure and temperature are 1 bar and 50° C. If the maximum pressure is limited to 25 bar find the following: -

- **(i)** Air standard efficiency
- **(ii)** Mean effective pressure

Solution:-

Bore diameters of engine = D = 300 mm = 0.30 m

Stroke of engine = L= 400 mm = 0.40 m

Clearance volume = V_c = 0.01650 m^3

Initial pressure = P_1= 1 bar

Initial temperature = T_1= 50 + 273 = 323 k

Maximum pressure = P_3 =25 bar

Swept volume = V_5= T_1/4 D^2 L = T_1/ 4 × $(0.3)^2$×(0.40)

V_5= 0.113

Compression ratio r = $\frac{V_s + V_c}{V_c} = \frac{0.113+0.01650}{0.01650}$

Air Standard efficiency

$$\acute{\eta}_{otto} = 1 - \frac{1}{r^{\gamma-1}} = 1 = \frac{1}{(8)^{1.4-1}}$$

$= 0.5650\ \pi = 56.5\ \%$

Mean effective pressure

$T_1V_1^{\ \gamma} = T_2V_2^{\ \gamma}$

$$T_2 = P_2\left(\frac{V_1}{V_2}\right)^{\gamma} = 1 \times (8)^{1.4} = 18.38\ bar$$

$$r_p = \frac{P_3}{P_2} = \frac{25}{18.38} = 1.36$$

Mean effective pressure

$$P_m = \frac{P_1 r[(r-1)(r_p - 1)]}{(\gamma-1)(\pi-1)}$$

$$P_m = \frac{1\times8[(8^{1.4-1}-1)(1.36-1)]}{(1.4-1)(8-1)}$$

$$= \frac{8[(2.297-1)(0.36)]}{(0.4)(7)}$$

$$P_m = 1.334\ bar$$

Hence mean effective pressure = 1.334 bar

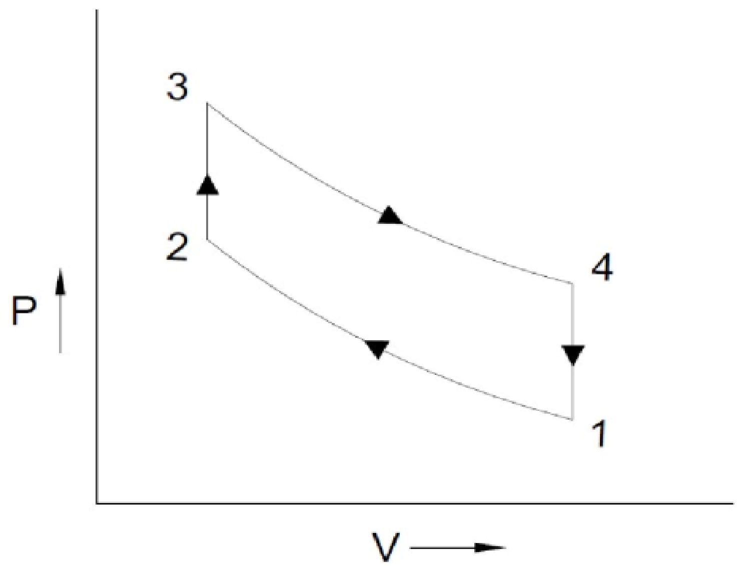
Example : The minimum pressure and temperature in an Otto cycle are 100 kpa and 27˚ C. The amount of heat added to the air per cycle is 1800 k j / k g. Determine.

(1) Pressure and temperatures at all points of air air standard Otto cycle.
(2) Calculate the specific work and thermal thermal efficiency of cycle for a compression ratio of 8.1.

Solution: - P_1 = 100 kpa = 10_5 N/m^2 or 1 bar,

T_1 = 27+ 273 =300 k , Heat added = 1800 k s/ kg

π=8.1, C_v= 0.72 ks / kg, γ=1.4



Consider 1 kg of air

(i) Pressures and temperatures at all points

$\frac{T_1}{T_2} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = r^{\gamma-1} = (8)^{1.4-1} = 2.297$

$T_2 = 300 \times 2.297 = 689.1$ k

$P_1 V_1{}^{\gamma} = P_2 V_2{}^{\gamma}$

$\frac{T_1}{T_2} = \left(\frac{V_1}{V_2}\right)^{\gamma} = (8)^{1.4} = 18.379$

$P_2 = 1 \times 18.379 = 18.379$ bar

Heatadded during the process

$C_v(T_3 - T_2) = 1800$

$0.72(T_3 - 689.1) = 1800$

$T_3 = \frac{1800}{0.72} + 689.1$

$T_3 = 3189.1\ k = 2916.1\,^{\circ}C$

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

$P_3 = \frac{P_2 T_3}{T_2} = \frac{18.379 \times 3189.1}{689.1}$

$P_3 = 85.05$ bar

Adiabatic expansion process

$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = r^{\gamma-1} = (8)^{1.4-1} = 2.297$

$T_4 = \frac{T_3}{2.297} = \frac{3189.1}{2.297} = 1388.3$ k

$P_3 V_3^{\gamma} = P_4 V_4^{\gamma} = P_4 = P_3 \left(\frac{V_3}{V_4}\right)^{\gamma} = 85.05 = \left(\frac{1}{8}\right)^{1.4}$

$P_4 = 4.62$ bar

(ii) Specific work = Heat added – Heat rejected

$= C_V(T_3 - T_2) - C_V(T_4 - T_1) = C_V[(T_3 - T_3)(T_4 - T_1)]$

$= 0.72[(3189.1-689.1)-(1388.3-300)]$

$= 0.72[(2500-1088.3)=1016.4$ kJ/kg

Thermal efficiency

$ή_{th} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{(8)^{1.4-1}}$

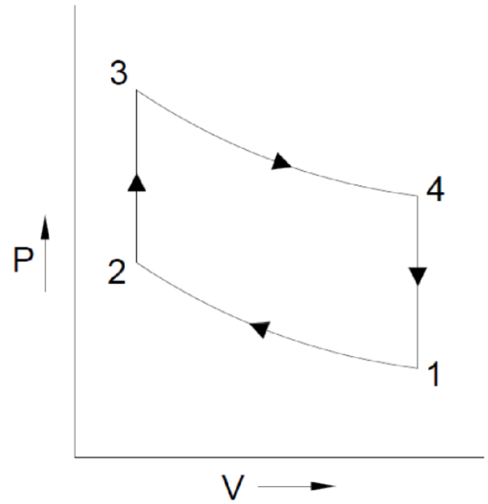
$= 0.5647 = 56.47\%$

Example : An air standard otto cycle has a volumetric compression ratio of 6, the lowest cycle pressure of o.1MPa and temperature limits of 27˚ C and 1500˚ C. Determine the temperature and pressure after is entropic expansion also determine by what percentage cycle has been improved.

Solution: $r = \frac{V_1}{V_2} = \frac{V_4}{V_3}$ $P_1 = 0.1$ MPa = 1 bar

$T_1 = 27+273 = 300$ k $T_3 = 1500+273 = 1773˚$ C

$\gamma = 1.4$



For compression process 1-2

$P_1V_1^{\gamma} = P_2V_2^{\gamma}$

$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{\gamma} = 1(6)^{1.4} = 12.3\ bar$

$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1} = 300 \times (6)^{0.4}$

$T_2 = 614.4\ k$

For constant volume Process- 2-3

$\frac{P_2}{T_2} = \frac{P_2}{T_2} = 12.3 \times \frac{1773}{614.4}$

$P_3 = 35.5$ bar

For expansion process – 3-4

$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = (6)^{1.4} = 2.048$

$T_4 = \frac{T_3}{2.048} = \frac{1773}{2.048} = 865.7\ k$

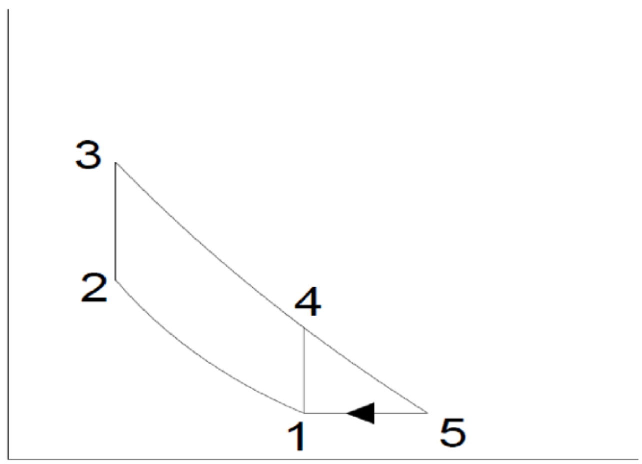
$P_3V_3^{\gamma} = P_4V_4^{\gamma}$

$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{\gamma}$

$P_4 = 35.5\left(\frac{1}{6}\right)^{1.4}$

P_4= 2.88 bar

Process required to complete the cycle is the constant pressure scavenging. The cycle is called Atkinson cycle.



Percentage improvement in efficiency

$\acute{\eta}_{otto} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{(6)^{1.4}-1} = 0.5716$

$= 51.16\%$

$\acute{\eta}_{atkinson} = \frac{Work\ done}{Heat\ supplied}$

$= \frac{Heat\ supplied\ - Heat\ rejected}{Heat\ supplied}$

$= \frac{Cv\ (T_3 - T_2) - Cp\ (T_5 - T_1)}{Cv\ (T_3 - T_2)}$

$= 1 - \frac{Cp\ (T_5 - T_1))}{Cv\ (T_3 - T_2)}$

$\acute{\eta}_{atkinson} = 1 - \frac{\gamma\,(T_5 - T_1)}{Cv\ (T_3 - T_2)}$

$\frac{T_5}{T_3}=\left(\frac{P_5}{P_3}\right)^{\gamma-\frac{1}{\gamma}}$

$T_5= 1773\left(\frac{1}{35.65}\right)^{\frac{1.4-1}{1.4}} = 599.4\ k$

$\acute{\eta}_{atkinson=}\ 1-\frac{1.4(599.4-300)}{(1773-614.4)}$

$= 0.638 = 63.8\ \%$

Improvement in efficiency

$= 63.8 – 51.16 = 12.64\ \%$ Ans.

Example : An engine working on Otto cycle has a volume of 0.50 m^3, pressure 1 bar and temperature 40˚ C at the beginning of compression stroke. At the end of compression stroke the pressure is 12 bar. 200 k j of heat is added at constant volume. Determine following: -

1) Pressure, temperature and volume
2) Percentage clearance
3) Efficiency
4) Mean efficiency pressure
5) Ideal power development by the engine if the number of working cycle per minutes 210.

Solution: -

$V_1= 0.50\ m^3$

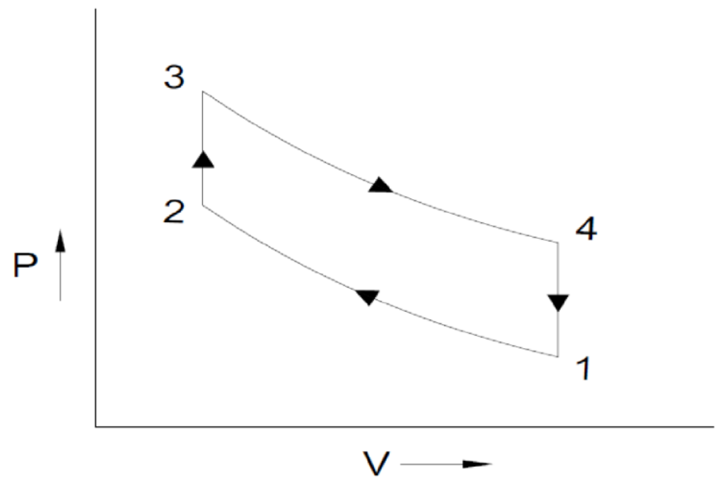
Initial pressure $P_1= 1$ bar

Initial temperature $T_1 = 40+273 = 313$ k

Pressure at the end of compression = 12 bar

Heat added at constant volume = 200 k j

Number of working cycle/ min = 210



i) Pressure, temperature and volume for adiabatic compression

$P_1V_1^{\gamma} = P_2V_2^{\gamma}$

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

$r = \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} = \left(\frac{12}{1}\right)^{\frac{1}{1.4}} = (12)^{0.714}$

$r = 5.89 \cong 5.9$

$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (r)^{\gamma-1} = (5.9)^{0.4} = 2.03$

$T_2 = T_1 \times 2.03 = 313 \times 2.03$

$T_2 = 636.6\ k$

$V_2 = \frac{P_1V_1 \times T_2}{T_1 \times P_2} = \frac{1 \times 0.50 \times 636.6}{313 \times 12}$

$V_2 = 0.084\ m^3$

Heat supplied during 2-3 Process

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

$m = \frac{P_1V_1}{RT_1} = \frac{1 \times 10^5 \times 0.50}{287 \times 313}$

$m = 0.55$ k g

$Q_s = m\, C_v(T_3 – T_2)$

$200 = 0.55 \times 0.71(T_3 – 636.6)$

$T_3 = \frac{200}{0.55 \times 0.71} + 636.6$

$T_3 = 1148.76\ k$

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

$P_3 = \frac{P_2}{T_2} \times 1148.76$

$P_3 = 21.65\ bar$

$V_3 = V_2 = 0.084\ m^3$

For isentropic process 2-4

$P_1 V_1^{\gamma} = P_2 V_2^{\gamma}$

$P_4 = P_3 \left(\frac{V_3}{V_4}\right)^{\gamma} = P_3 \times \left(\frac{1}{r}\right)^{\gamma}$

$= 21.65 \times \left(\frac{1}{5.9}\right)^{1.4}$

$P_4 = 1.80$ bar

$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma - 1} = \left(\frac{1}{r}\right)^{\gamma - 1} = \left(\frac{1}{5.9}\right)^{1.4 - 4}$

$\frac{T_4}{T_3} = 0.49$

$T_4 = 0.49 \times 1148.76 = 564.8$ k

$T_4 = 564.8$ k

$V_4 = V_1 = 0.50\ m^3$

ii) Percentage clearance

$$\frac{V_c}{V_s} = \frac{V_2}{V_1 - V_2} \times 100 = \quad \frac{0.084}{0.50 - 0.084} \times 100$$

$$\frac{V_c}{V_s} = 20.19\% \text{ Ans.}$$

iii) Efficiency

Heat rejection per cycle is given by

$Q_r = mc_v(T_4 – T_1)$

$= 0.55 \times 0.71\ (564.8 - 313)$

$Q_r = 98.3$ k j

Air Standard efficiency of two cycles

$$ή_{otto} = \frac{(Qs - Qr)}{Qs} = \frac{200 - 98.3}{200}$$

$=0.5085$

$=50.8\ \%$

$$ή_{otto} = 1 - \frac{1}{(5.9)^{0.4}} = 1 - \frac{1}{2.03} = 50.8\ \%$$

iv) Mean effective pressure

$$P_m = \frac{Work\ \ done}{Swept\ \ Volume} = \frac{(Qs - Qr)}{V_1 - V_2}$$

$P_m = 2.44$ bar

v) Power developed

P_m = Work done per second × No. of cycle per second

$= (200 - 98.3) \times \frac{210}{60}$

$P_m = 355.95$ k w

$= 356$ k w Ans.

1.10.3 CONSTANT PRESSURE OR DIRECT CYCLE

Diesel cycle was introduced by Dr. R. Diesel in 1997. In this cycle heat is supplied at constant pressure instead of constant volume as supplied in Otto cycle. P-V and T-S diagram of this cycle shows in following diagram.

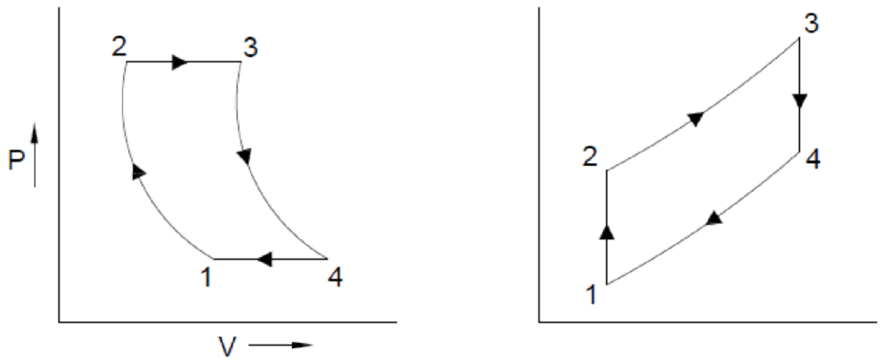


Fig. 1.15 Constant pressure cycle

Point 1 represents that the cylinder is full of air. P_1, V_1 and T_1 be the corresponding pressure, volume and temperature at that point. The process 1-2 represents the adiabatic process, in which piston compresses the air adiabatically till the values become P_2, V_2 and T_2(i.e. PVγ=Constant) 2-3 process represents the heat is added at constant pressure. During this addition of heat volume increases from $V_2$to V_3 and temperature $T_2$to T_3 corresponding to point. 3-4 process represents the adiabatic expansion process, in which pressure, volume, temperature value because P_4, V_4, T_4 respectively.

Heat is rejected at constant volume till to point 1, when it returns to its original stante when consider 1 k g of air:

Heat supplied at constant pressure = $C_p(T_3 - T_2)$

Work done = Heat supplied – Heat rejected

$= C_p(T_3 - T_2) – C_v(T_4 - T_1)$

$$ή_{diesel} = \frac{Work\ done}{Swept\ Volume}$$

$$= \frac{C_p\,(T_3 - T_2) - C_v(T_4 - T_1)}{C_p\,(T_3 - T_2)}$$

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

ή$_{diesel}$= $1 - \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)}$

Let compression ratio r = $\frac{V_1}{V_2}$

And cut off ratio ρ = $\frac{V_3}{V_2}$

During constant pressure 2-3

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

$T_3 = \rho\, T_2 = \rho\, T_1 r^{\gamma - 1}$

During adiabatic expansion -3-4

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

$$= \left(\frac{V_4}{V_2} \times \frac{V_2}{V_3}\right)^{\gamma - 1} \qquad (V_4 = V_1)$$

$$= \left(\frac{V_1}{V_2} \times \frac{V_2}{V_3}\right)^{\gamma - 1}$$

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

$$T_4 = \frac{T_3}{\left(\frac{r}{\rho}\right)^{\gamma - 1}} = \frac{ST_1(r)^{\gamma - 1}}{\left(\frac{r}{\rho}\right)^{\gamma - 1}} = T_1 \rho^{\gamma}$$

ή$_{diesel}$= $1 - \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)}$

$= 1 - \frac{(T_1 \rho^{\gamma} - T_1)}{\gamma[(\rho T_1 (r)^{\gamma - 1} - T_1 r^{\gamma - 1})]}$

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

$\acute{\eta}_{diesel} = 1 - \frac{1}{r^{\gamma-1}.\ \gamma}\left[\left(\frac{\rho^{\gamma-1}}{(\rho-1)}\right)\quad\right]$

The network can be expressed for diesel cycle as follows

$$W = P_2(V_3 - V_4) + \frac{P_3V_3 - P_4V_4}{\gamma-1} - \frac{P_2V_2 - P_1V_1}{\gamma-1}$$

$$= P_2(\rho.V_2 - V_2) + \frac{P_3\rho.V_2 - P_4rV_2}{\gamma-1} - \frac{P_2V_2 - P_1rV_1}{\gamma-1}$$

$$\left\{\frac{V_3}{V_2} = \rho \ and\ \frac{V_1}{V_2} = r\right\}$$

$$W = P_2V_2(\rho-1) + \frac{P_3\rho.V_2 - P_4rV_2}{\gamma-1} - \frac{P_2V_2 - P_1rV_1}{\gamma-1}$$

$$= \frac{V_2[P_2(\rho-1)(\gamma-1) + P_3\rho - P_4r - (P_2 - P_1r)]}{\gamma-1}$$

$$= \frac{V_2\left[P_2(\rho-1)(\gamma-1) + P_3\left(\rho - \frac{P_4r}{P_3}\right) - P_2\left(1 - \frac{P_1r}{P_2}\right)\right]}{\gamma-1}$$

$$= \frac{P_2V_2\left[(\rho-1)(\gamma-1) + \rho - \rho^{\gamma}.r^{1-\gamma} - \left(1 - r^{1-\gamma}\right)\right]}{\gamma-1}$$

$$\left\{\frac{P_4}{P_3} = \left(\frac{V_3}{V_4}\right)^{\gamma} = \left(\frac{\rho}{r}\right)^{\gamma} = \rho r^{\gamma}\right\}$$

$$= \frac{P_2.V_1.r^{1-\gamma}[(\delta-1)(\gamma-1) + \delta - \delta^{\gamma}.r^{1-\gamma} - (1 - r^{1-\gamma})]}{\gamma-1}$$

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

$$= \frac{P_1.V_1.r^{1-\gamma}\left[\gamma(\rho-1) - r^{1-\gamma} - (\rho^{\gamma}-1)\right]}{\gamma-1}$$

Mean effective pressure P_m

$$P_m = \frac{P_1.r^{\gamma}\left[\gamma(\rho-1) - r^{1-\gamma}(\rho^{\gamma}-1)\right]}{(\gamma-1)(r-1)}$$

Example : The stroke and cylinder diameter of a compression ignition engine are 300 mm 200 mm respectively. If clearance volume is 0.0005 m^3 and fuel injection intake for 5 percent of stroke, determine the efficiency of engine.

Solution: - Stroke length = L= 300 mm = 0.3 m

Diameter of cylinder = D = 200 mm = 0.2 m

Clearance volume V_2=0.0005 m^3

Swept volume $V_S = \frac{\pi}{4} D^2 L = \frac{\pi}{4} \times (0.2)^2 \times (0.3)$

$V_S = 0.0094$

Total volume = 0.0094 + 0.0005

= 0.0099 m^3

Cut off = $V_3 = V_2 + \frac{5}{100} V_S$

$= 0.0005 + \frac{5}{100} \times 0.00094$

= 0.0005+0.00045

= 0.00095 m^3

Cut off ratio $\rho = \frac{V_3}{V_2} = \frac{0.00095}{0.0005}$

$\rho = 1.9$

Compression ratio $= \frac{V_1}{V_2} = \frac{0.00095}{0.0005} = 19.8$

$\acute{\eta}_{diesel} = 1 - \frac{1}{\gamma r^{\gamma-1}} \left[\frac{\rho^{\gamma}-1}{\rho-1}\right]$

$= 1 - \frac{1}{1.4(19.8)^{0.4}} \left[\frac{(1.9)^{1.4}-1}{1.9-1}\right]$

$= 1 - (0.21)(1.61) = 1 - 0.35$

η_{diesel}= 0.65=65%

Example :- An engine with 150 mm cylinder diameters and 250 mm stroke works on diesel cycle. Initial pressure and temperature of air used are 1 bar and 27° C. Cut off is 10% of stroke. Determine if r=14.

(a) Pressure and temperature at all points
(b) Air standard efficiency
(c) Mean efficiency pressure
(d) Power of engine rpm 380.

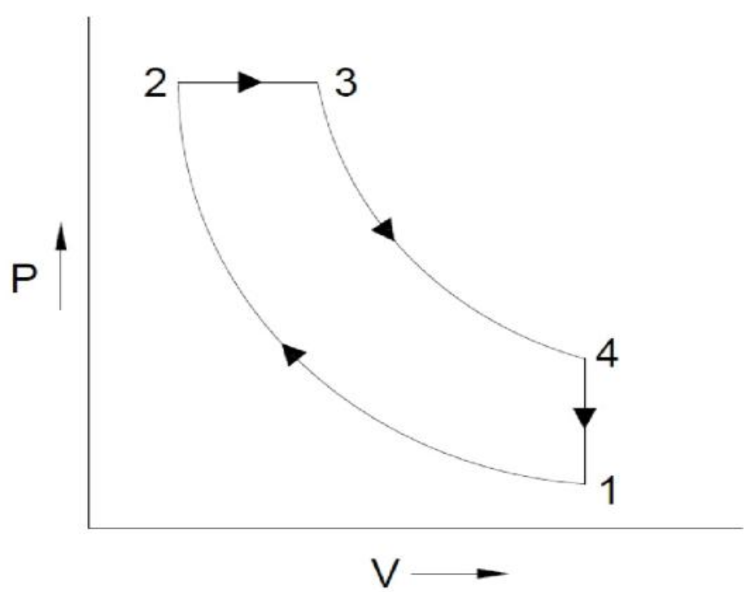
Solution: - Cylinder diameterD =150 mm

Stroke Length L =250 mm

Initial pressure P_1= 1 bar

Initial temperature T_1= 27+273 = 300 k

Cut off = $\frac{10}{100}.V$



(a) Pressure and temperature at all points

$V_S = \frac{\pi}{4} D^2 L = \frac{\pi}{4} \times (0.15)^2 \times (0.25)$

$= 0.0044\ m^3$

$V_1 = V_S + V_c = V_S + \frac{V_s}{\pi - 1}$

$V_1 = V_S\left(\frac{\pi}{\pi - 1}\right)$

$= 0.0044 \times \frac{14}{13}$

$V_1 = 0.0047\ m^3$

Mass of air

$P_1V_1 = mRT_1$

$m = \frac{P_1V_1}{RT_1} = \frac{1 \times (10)^5 \times 0.0047}{287 \times 300}$

$m = 0.0055$ kg/cycle

$P_1V_1^{\gamma} = P_2V_2^{\gamma}$

$P_2 = P_1 r^{\gamma} = 1 \times (14)^{0.4}$

$= 40.23$ bar

$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1} = (14)^{1.4-1}$

$T_2 = T_1(r)^{\gamma-1} = 300 \times (14)^{1.4}$

$T_2 = 862$ k

$V_2 = V_c = \frac{V_s}{r-1} = \frac{0.0044}{13}$

$V_2 = 0.00033\ m^3$

$P_2 = P_3 = 40.23$

$= \frac{\rho - 1}{r - 1}$

$\frac{10}{100} = \frac{\rho - 1}{r - 1}$

$\rho = 1.3 + 1 = 2.3$

$V_3 = \rho V_2 = 2.3 \times 0.00033$

$V_3 = 0.00076\ m^3$

$\frac{V_3}{T_1}=\frac{V_2}{T_2}$

$T_3=T_2\times\frac{V_3}{V_2}=862\times\frac{0.00076}{0.00033}$

T_3=1985 k

$P_2V_3\gamma = P_4V_4^{\gamma}$

$P_4=P_3\left(\frac{V_3}{V_4}\right)^{\gamma}$ $\left\{\frac{V_4}{V_3}=\frac{\pi}{\rho}=\frac{14}{2.3}=6.08\right\}$

$P_4=\frac{40.23}{(6.08)^{1.4}}=\frac{40.23}{12.53}$=3.2 bar

P_4=3.2 bar

$\frac{T_4}{T_3}=\left(\frac{V_3}{V_4}\right)^{\gamma-1}=\left(\frac{1}{6.08}\right)^{\gamma}=0.48$

$T_4=T_3\times 0.48=1385\times0.48=952$ k

T_4= 952 k

$T_4=T_1$= 0.0047

(b) Air standard efficiency: -

$ή_{diesel}=1-\frac{1}{\gamma r^{\gamma-1}}\left[\frac{\gamma-1}{\delta-1}\right]$

$=1-\frac{1}{1.4(14)^{0.4}}\left[\frac{(2.3)^{1.4}-1}{2.3-1}\right]$

= 1 – 0.248×01.69

= 1-0.42 = 0.58

$ή_{diesel}$= 58%

(c) Mean effective pressure P_m

$P_m=\frac{P_1.r^{\gamma}\left[\gamma(\rho-1)-r^{1-\gamma}(\rho^{\gamma}-1)\right]}{(\gamma-1)(r-1)\cdot}$

$=\frac{1(14)^{1.4}\left[1.4(2.3-1)-(14)^{1.4}\left((2.3)^{1.4}-1\right)\right]}{(1.4-1)(14-1)\cdot}$

$=\frac{40.23[1.82-0.34\times 2.20]}{13\times 0.4}=\frac{42.99}{5.2}=8.26$

P_m= 8.26 bar

(d) Power = work done per cycle = $P_m V_S = \frac{8.26\times(10)^5\times 0.0044}{(10)^3}$

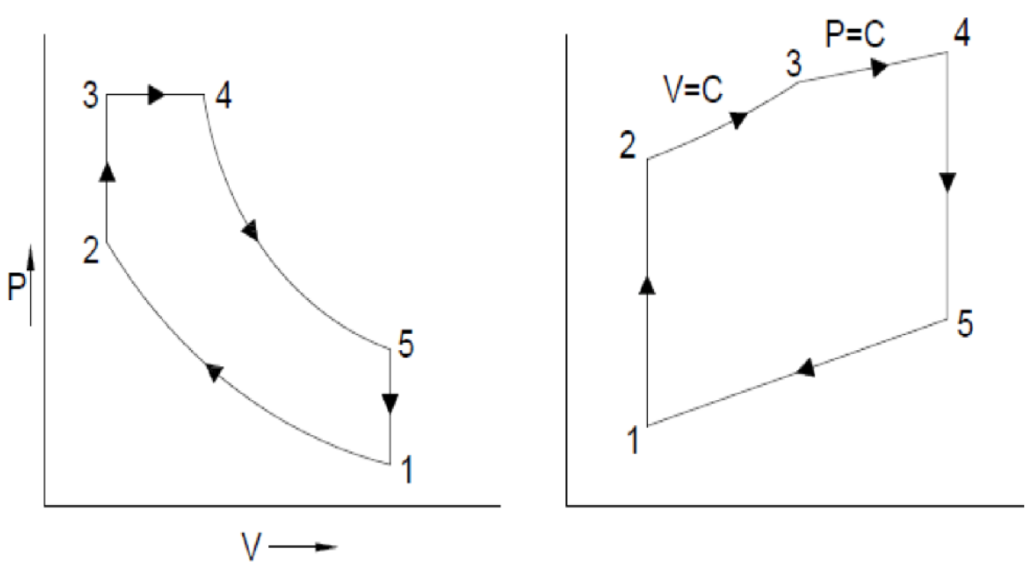
= 3.64 k j/ cycle

Work done per cycle = $3.64\times\frac{380}{60}=23\ k\ w$

Power = 23 k w

1.10.4 DUAL COMBUSTION CYCLE

Dual combustion cycle is a combination of otto and Diesel cycles, also called the limited pressure or mixture cycle. In this cycle partly heat is added at mixed cycle. In this cycle partly at constant pressure so that more time is available to fuel for combustion.



Fi. 1.16 DUAL COMBUSTION CYCLE

The dual combustion cycle consists of following operations:-

1) 1-2 Adiabatic compression
2) 2-3 Partly heat is added at constant volume partly
3) 3-4 Heat is added at constant pressure
4) 4-5 Adiabatic expansion
5) 5-1 Rejection of heat at constant volume.

Total heat supplied = Heat supplied at constant volume + Heat supplied at constant pressure

$Qs.=C_v(T_3-T_2) + C_p(T_4-T_3)$

Heat rejected = $Q_R=C_V(T_5-T_1)$

Work done = Heat supplied – Heat rejected

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

$$ή_{dual}= \frac{Work\ done}{Heat\ supplied}$$

$$= \frac{Cv(T3-T2)+Cp(T4-T3)-Cv(T5-T1)}{Cv(T3-T2)+Cp(T4-T3)}=$$

$$= 1-\frac{Cv(T5-T1)}{Cv(T3-T2)+Cp(T4-T3)}$$

$$ή_{dual}= 1-\frac{T5-T1}{(T3-T2)+\gamma(T4-T3)} \qquad \{\frac{Cp}{Cv}=\gamma\}$$

Adiabatic Compression:-

$$\frac{T2}{T1}=(\frac{V1}{V2})^{\gamma-1}= r^{\gamma-1} \qquad \{\frac{V1}{V2}=r\}$$

Compression Ratio

During Process 2-3

$$\frac{P3}{T3}=\frac{P3}{T2}$$

$$\frac{T3}{T2}=\frac{P3}{P2}=\beta$$ β=Pressure Ratio

During Process 3-4

$$\frac{V3}{T3}=\frac{V4}{T4}$$

$$\frac{T4}{T3}=\frac{T4}{V3}=\rho$$ δ=Cut off Ratio

$$T_4=\rho T_3$$

During adiabatic expansion- 4-5

$$\frac{T4}{T5}=\left(\frac{V5}{V4}\right)^{\gamma\text{-}1}$$ $V_5=V_1$

$$=\left(\frac{V1}{V2}\right)^{\gamma\text{-}1}$$

$$=\left(\frac{V1}{V2}\times\frac{V2}{V3}\times\frac{V3}{V4}\right)^{\gamma\text{-}1}$$

$$\frac{T4}{T5}=\left(\frac{r}{\rho}\right)_{\gamma\text{-}1}$$

$$\frac{\rho T}{T5}=\left(\frac{r}{\rho}\right)_{\gamma\text{-}1}$$

$$T_5=\delta.T_3\left(\frac{r}{\rho}\right)^{\gamma\text{-}1}$$

$$\frac{T2}{T1}=r^{\gamma\text{-}1}$$

$$T_1=\frac{T3}{\beta}.r^{\gamma\text{-}1}$$

$$\acute{\eta}_{dual}=1-\frac{T5-T1}{(T3-T2)+\gamma(T4-T3)}$$

$$=1-\frac{\left[S.T_3\left(\frac{r}{\pi}\right)^{\gamma}-\frac{T_3}{\beta}.\frac{1}{r^{1-\gamma}}\right]}{\left[\left(T_3-\frac{T_3}{\beta}+\gamma(\rho T_3-T_3)\right)\right]}$$

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

$$= 1- \frac{1}{r^{1-\gamma}}.\frac{(\beta\rho^{\gamma}-1)}{[(\beta-1)+\beta\gamma(\rho-1)]}$$

Work done is given by

$$W= P_3(V_4 - V_3)+ \frac{P_4V_4-P_5V_5}{\gamma-1} - \frac{P_2V_2-P_1V_1}{\gamma-1}$$

$$= P_3V_3(\boldsymbol{\delta}-1)+ \frac{(P_3\rho.V_3-P_5\pi V_3)(P_2V_3-P_1\pi V_3)}{\gamma-1}$$

$$= \frac{P_3V_3(\rho-1)(\gamma-1)+P_4V_3\left[\left(\rho-\frac{P_5\pi}{P_4}\right)-P_2V_3\left(1-\frac{P_1\pi}{P_2}\right)\right]}{\gamma-1}$$

$$\frac{P_3}{P_4} = \left(\frac{V_3}{V_4}\right)^{\gamma} = \left(\frac{\delta}{r}\right)^{\gamma} \; and \quad \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} = r^{\gamma}$$

$P_3 = P_4$, $V_2=V_3$, $V_5=V_1$

$$W=\frac{V_3\left[P_3(\rho-1)(\gamma-1)+P_3\left(\rho-\rho^{\gamma}.r^{1-\gamma}\right)-P_2\left(1-r^{1-\gamma}\right)\right]}{\gamma-1}$$

$$=\frac{P_2V_2\left[\beta(\delta-1)(\gamma-1)+\beta\left(\rho-\rho^{\gamma}.r^{1-\gamma}\right)-P_2\left(1-r^{1-\gamma}\right)\right]}{\gamma-1}$$

$$= \frac{P_1(r)^{\gamma}\frac{V1}{\pi}\left[\beta\gamma\ (\rho-1)+(\beta-1)-\pi^{1-\gamma}\left(\beta\rho^{\gamma}-1\right)\right]}{\gamma-1}$$

$$= \frac{P_1V_1(r)^{\gamma-1}\left[\beta\gamma\ (\rho-1)+(\beta-1)-r^{1-\gamma}\left(\beta\rho^{\gamma}-1\right)\right]}{\gamma-1}=$$

Mean effective pressure is given by

$$P_m=\frac{W}{V_1-V_2} = \frac{W}{V_1\left(\frac{r-1}{r}\right)}$$

$$= \frac{P_1V_1\left[(r)^{\gamma-1}\beta\gamma\ (\delta-1)+(\beta-1)-r^{1-\gamma}\left(\beta\rho^{\gamma}-1\right)\right]}{(\gamma-1)V_1\left(\frac{r-1}{r}\right)}$$

$$P_m = \frac{P_1 r^{\gamma}[\beta(\rho-1)+(\beta-1)-r^{1-\gamma}(\beta\rho^{\gamma}-1)]}{(\gamma-1)(r-1)}$$

Example : A diesel engine working on a dual combustion cycle has a stroke volume 0.0090 m^3 and compression ratio 15:1. The fuel has a calorific value of 43800 kj/kg . At the end of suction, the air is at 1bar and 100^0c. The maximum pressure in the cycle is 60 bar and air fuel ratio is 20:1. Find the thermal efficiency. C_p=1kj/kgk and c_v= 0.71 kj/ kgk.

Solution:-Initial Temperature T_1=100+273= 373

Initial pressure P_1=1 bar

Maximum pressure P_3= P_4=60 bar

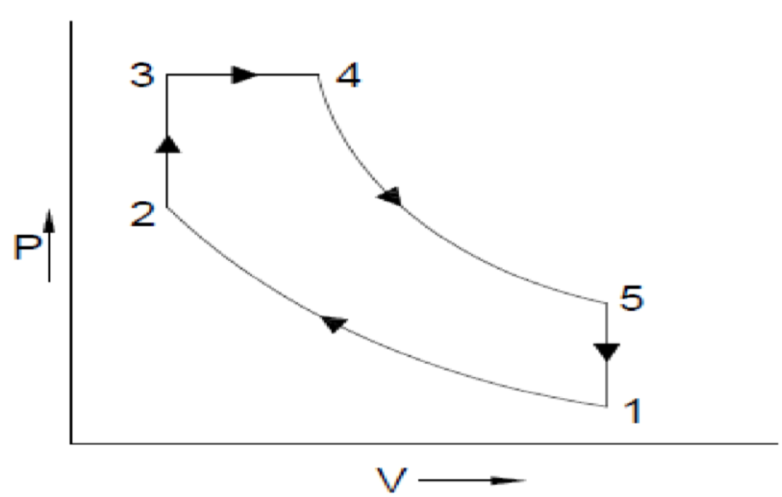
Stroke volume V_S= 0.0090 m^3

Air fuel ratio = 20:1

Compression ratio r =15:1

Calorific value C=43800 kj/kg

C_P= 1 kj/kgkC_v=0.71 kj/kgk



Thermal efficiency:

$V_S = V_1 - V_2 = 0.0090\ m^3$

$r = \frac{V_1}{V_2} = 15, = V_1 = 15V_2$

$K_1 v_2 = 0.0090$

$K_1 V_2 = 0.0090$

$V_2 = \frac{0.0090}{k1} = 0.00064\ m^3$

For process 1-2 (adiabatic)

$P_1 v_1^y = p_2 v_2^y$

$P_2 = p_1 (\frac{v1}{v2})^y = 1 \times (15)^{1.4}$

$P_2 = 45.5$ bar

$\frac{T2}{T1} = (\frac{v1}{v2})^{y-1} = \pi^{y-1} = (15)^{1.4-1} = 3.04$

$T_2 = T_1 \times 3.04 = 373 \times 3.04 = 1134$ k

$T_2 = 1134$ k or 861^0c

For process 2-3 (cemtantvalume)

$\frac{P2}{T2} = \frac{P3}{T3}$

$T_3 = T_2 \times \frac{P3}{P2} = 1134 \times \frac{60}{45.5}$

$T_3 = 1495$ K $= 1222^0$c

$P_1 v_1 = mRT_1$

$M = \frac{P1v1}{RT1} = \frac{1 \times 10^5 \times 0.0096}{287 \times 373}$

M = 0.0089 kg (air)

Heat added at constant volume (2-3)

$Q_s = mc_v(T_3 - T_2)$

0.0089× 0.71 (1495 – 1134)

Q_s = 2.281 kj

Amount of fuel added during process 2-5

$$= \frac{2.281}{43800} = 0.000052 \text{ kg}$$

AS air Fuel ratio is 20:1

Total amount of fuel added $= \frac{0.0089}{20}$

M_1 = 0.00044 kg

Quantity of fuel added during process = 3-4

= 0.00044 – 0.000052

= 0.000388 kg

Heat added at constant pressure b-4

= 0.000388× 43800

= 16. 99 ks

$Q_s = m\, c_p\, (T_4 - T_3)$

$16.99 = (0.0089=0.00044)\, c_p(T_4 - T_3)$

$16.99 = (0.0093) \times 1\, (T_4 - 1495)$

$T_3 = \frac{16.99}{0.0093} + 1495$

$T_3 = 3321\ k = 3048^0c$

$$\frac{V3}{T3} = \frac{V4}{T4}$$

$V_{4\,=}\quad \frac{V3\ T4}{T3} = \frac{00064 \times 3321}{1495}$

$V_4 = 0.0014\ m^3$

$\frac{T4}{T5} = (\frac{v5}{v4})^{y-1} = (\frac{0.0096}{0.0014})^{1.4-1} = 2.16$

$T_5 = \frac{T4}{2.16} = \frac{3321}{2.16} = 1537\ k$

Heat rejected during constant volume process

$Q_s = mq\ (T_1 - T_1)$

$= (0.0084+0.00044) \times 0.71\ (1537 - 373)$

$= 7.89$ kj/kg

Work done $= Q_5 - Qr$

$= 16.99 - 7.89 = 9.1$ kj/kg

$\acute{\eta} t\ h = \frac{work \quad done}{Heat \quad Supplied} = \frac{9.1}{16.99} = 0.535$

$\acute{\eta} t\ h = 53.5\ \%$

1.10.5 BRAYTON CYCLE

This cycle is a constant pressure cycle for a perfect gas. It is also called joule cycle. An ideal gas turbine plant would perform the processes that make up a Bray ton cycle.

The various processes are as follows:

1-2 :- The air is compressed isentropic ally from the lower pressure p_1 to higher pressure p_2 Temperature increasing from T_1 to T_2 .

2-3 :- Volume is increased from $V_2$to V_3 when heat blows into the system and temperature from $T_2$to T_3while pressure remains constant

3-4 :- The air is is isentropic ally ercpandedformp$_2$ to p_1 and temperature decreases form T_3 to T_4 .

4-1 :- Heat is rejected form the system as volume decreases from V_4 to V_1 and temperature form T_4 to T_1

$$= \frac{work \quad done}{Heat \quad reveived}$$

$$= \frac{mc\ p(T3 - T2) - mc\ p(T4 - T1)}{m\,c\,p\,(\ T3\ T2)}$$

$$= 1 - \frac{T4 - T1}{T3 - T2}$$

Now form isentropic expansion

$$\frac{T_2}{T_1} = \left(\frac{P_1}{P_2}\right)^{\gamma - \frac{1}{\gamma}}$$

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

$$\frac{T_3}{T_4} = \left(\frac{P_1}{P_2}\right)^{\gamma - \frac{1}{\gamma}} \text{or } T_3 = T_4 (r_p)^{\frac{\gamma - 1}{\gamma}}$$

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

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

Pressure ratio for maximum work is a function of the limiting temperature ratio.

Work output during the cycle

Heat received/cycle- Heat rejected / cycle

$= mC_p(T_3-T_2)-mC_p(T_4-T_1)$

$= mC_p(T_3-T_4)- mC_p(T_2-T_1)$

$= mC_pT_3\left(1-\frac{T_4}{T_3}\right)-\left(\frac{T_2}{T_1}-1\right)$

When consider a given turbine, the minimum temperature $T_1$and maximum temperature $T_3$are prescribed. T_1being the temperature of the atmosphere and T_3 the maximum temperature which the metals of turbine would with stand. Consider the specific heat at constant pressure C_p to be constant. Then

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

$$Z=\frac{\gamma-1}{\gamma}$$

When have, work output/cycle

$$W= K\left[T_3\left(1-\frac{1}{r_p{}^z}\right)-T_1(r_p{}^z-1)\right]$$

When differentiating

$$\frac{\alpha w}{\alpha \pi_p}=K\left[T_3\times\frac{Z}{r_p(z+1)}-T.Zr^{(z-1)}\right]=0$$

$$\frac{ZT_3}{r_p(z+1)}=T_1Z(r_p)^{(z-1)}$$

$$r_p{}^{2z}=\frac{T_3}{T_1}$$

$$r_p=\left(\frac{T_3}{T_1}\right)^{\frac{1}{2Z}} \qquad .\text{-}. \qquad r_p=\left(\frac{T_3}{T_1}\right)^{\frac{1}{2/\gamma-1}}$$

Thus the pressure ration for maximum work is a function of the limiting temperature ratio.

Work ratio:-

It is defined as the ratio of net work output to the work done by the turbine

Work ratio = $\frac{W_T-W_c}{W_T}$

$$= \frac{mC_p(T_3-T_4)-mC_p(T_2-T_1)}{mC_p(T_3-T_4)}$$

$$= 1 - \frac{T_2-T_1}{T_3-T_4}$$

$$= 1 - \frac{T_1}{T_3}\left[\frac{(r_p)^{\frac{\gamma-1}{\gamma}} - 1}{1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}}\right]$$

Work ratio = 1 - $\frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{\gamma}}$

1.10.6 STIRLING CYCLE

This cycle consists of two isothermal and two constant volume processes. Following processes are shown in fig: -

Process 1-2 is the isothermal compression with heat rejection Q_L to the surroundings at temperature T_L process 3-4 is the isothermal expansion with heat addition Q_H form a source at temperature T_H .

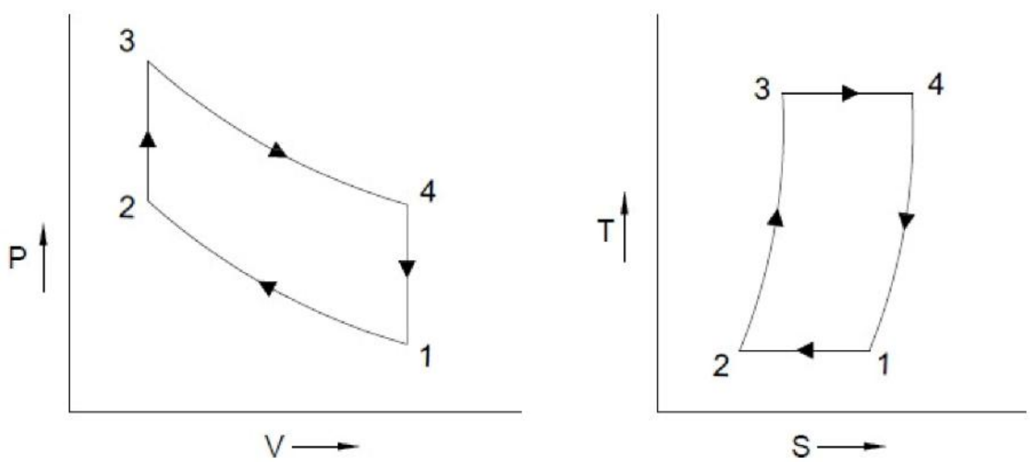


Fig. 1.17 STIRLING CYCLE

Heat rejected :Q_{12} = W_{1-2} = R T_1 log $\frac{V_1}{V_2}$ (compression)

$Q_{23} = C_v\ (T_H - T_L)$

$W_{23} = 0$

Heat supplied $Q_{34} = W_{3\text{-}4} = R\ T_H \log \frac{V_4}{V_3}$

$= \quad R\ T_H \log \frac{V_1}{V_2}$ *(expansion)*

$Q_{4\text{-}1} = -\ C_v(\ T_L - T_H)$ *or* $C_v\ (\ T_H - T_L\ )$

$W_{4\text{-}1} = 0$

The efficiency of stirling cycle is less than that of cannot cycle due to heat transfer of constant volume processes. It wever, if a regenerative arrangement is used such as $Q_{4\text{-}1} = Q_{23}$

$$\delta = \frac{R\ TH \log \frac{V1}{V2} - R\ TL \log \frac{V1}{V2}}{R\ TH \log \frac{V1}{V2}}$$

$$\delta = \frac{T_{H} - T_{L}}{T_{H}}$$

Which means the regenerative stirling cycle has same efficiency as the Carnot Cycle.

1.10.7 ATKINSON CYCLE

Atkinson cycle consists of two adiabatic a constant volume and a constant pressure process. This cycle consists the following processes

(a) 1-2 Heat rejection at constant pressure
(b) 2-3 Adiabatic compression
(c) 3-4 Heat addition at constant volume
(d) 4-1 Adiabatic expansion.

Processes are shown in p-v- diagram

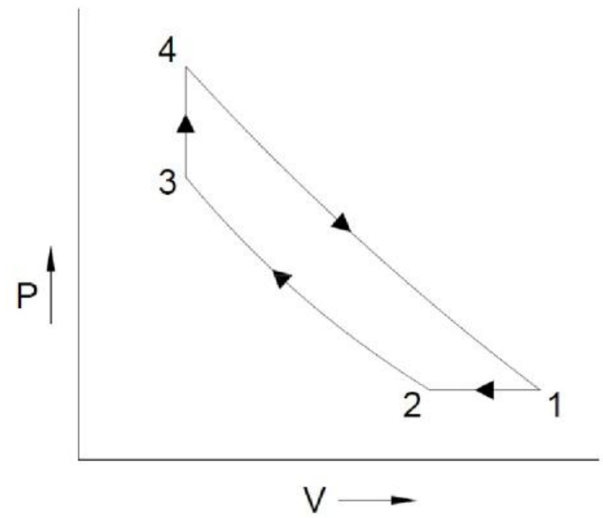


Fig. 1.17 Atkinson cycle

For 1 Kg of air v →

Compression ratio = $\frac{V_2}{V_3}$ =

Expansion ratio = $\frac{V_1}{V_4}$= π

Heat supplied at constant volume = C_v (T_4- T_3)

Heat rejected = C_v (T_1- T_2)

Work done = Heat supplied – Heat rejected

= C_v (T_4- T_3) = C_v (T_1- T_2)

$$\eta=\frac{work \quad done}{Heat \quad reveived} = \frac{C_V(T_4-T_3)-C_P(T_1-T_2)}{C_v(T_4-T_3)}$$

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

During adiabatic compression 2-3

$$\frac{T_3}{T_2}=\left(\frac{V_2}{V_3}\right)^{\gamma-1}=(\alpha)^{\gamma-1}$$

$$T_3 = T_2 = (\alpha)^{\gamma-1}$$

During constant pressure

$$\frac{V_1}{T_1} = \frac{V_2}{T_2}$$

$$\frac{T_1}{T_2} = \frac{V_2}{V_1} = \frac{\alpha}{\pi}$$

During adiabatic expansion 4-1

$$\frac{T_4}{T_1} = \left(\frac{V_1}{V_4}\right)^{\gamma-1} = (r)^{\gamma-1}$$

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

$$= \frac{\alpha T_4}{(r)^{\gamma}}$$

$T_2 = \frac{\alpha T_4}{(r)^{\gamma}} . (\alpha)^{\gamma-1} = \left(\frac{\alpha}{r}\right)^{\gamma} T_4$

$\eta = 1 - \gamma \left\{\frac{\frac{T_4}{(r)^{\gamma-1}} - \frac{\alpha T_4}{(r)^{\gamma}}}{T_4 - \left(\frac{\alpha}{r}\right)^{\gamma} . T_4}\right\} = 1 - \gamma \left(\frac{r-\alpha}{(r)^{\gamma} - (\alpha)^{\gamma}}\right)$

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

1.10.8 ERICSSON CYCLE

This cycle was invented by Ericsson, so it is called Ericsson cycle.

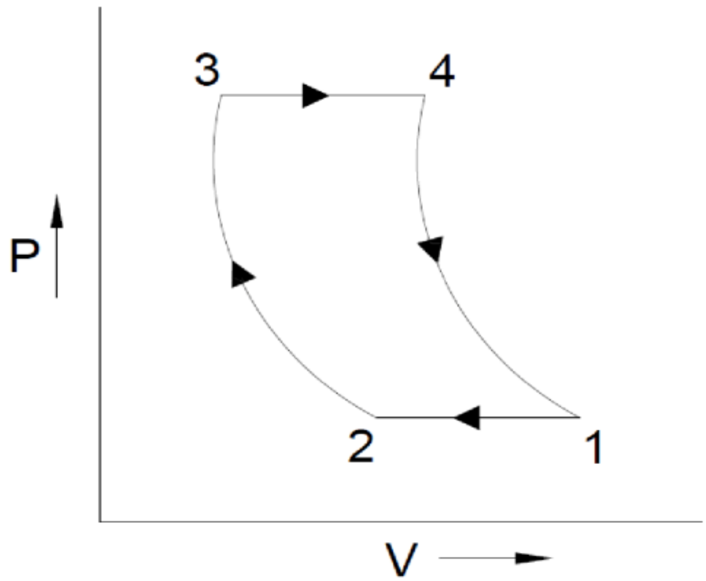


Fig. 1.18 Ericsson Cycle

This cycle consists following processes

(a) 1-2:- Heat rejection at constant pressure

(b) 2-3:- Isothermal compression

(c) 3-4:- Heat addition at constant pressure

(d) 4-1:- Isothermal expansion

For 1 kg of air

$$\text{Volume ratio } \pi = \frac{V_2}{V_3} = \frac{V_1}{V_4}$$

Heat supplied to air from an external source = R T_1 loge π

Heat rejected by air = R T_2 loge π

Work done = Heat supplied – Heat rejected

= R T_1 loge π - R T_2 loge π

$$\eta = \frac{work \quad done}{Heat \quad supplied} = \frac{R\ loge \quad r\ (T_{1-T_2})}{R\ T_{1\ loge} \quad r} = \frac{T_{1-T_2}}{T_1}$$

1.10.9 Testing

The aim of the development engineer is to reduce. The capital cost and running cool of engine. The parameters of engines are so different in nature that it is almost physically impossible to take care of all parameter during engine design. Therefore, to improve the performance of engine, it is necessary to conduct. The different test on the engine. The types and nature of the test to be conducted will depend upon a different number of factors. The performance of the engine as given specification of manufacturer Parameters:-

Performance of engine is an indication of the degree of success with which it does its assigned job .conversion of chemical energy into thermal energy and converted into useful mechanical work. The basic performance parameters are discussed below:-

(a) Power and Mechanical efficiency
(b) Mean effective pressure and torque
(c) Specific output
(d) Volumetric efficiency
(e) Fuel air ratio
(f) Thermal efficiency and heat balance
(g) Specific weight
(h) Specific fuel consumption
(i) Exhaust smoke and other emission

9.9.1 Power and Mechanical efficiency

(a) Indicated power (I. P.)

The total power developed by combustion of fuel in the combustion chamber is called indicated power.

I_p=(P_mLANn 10)/6 KW

Where P_m = Indicated mean effective pressure (bar)

n = Number of cylinders

L = Length of stroke (in meter)

A = Area of piston (m^2)

N = rpm. (Speed)

(b) Brake power (B. P.)

The power developed by an engine at the output shaft is called the brake power.

BP= $\frac{2\pi NT}{60\times1000} kw$

Where T= Torque in N-m

N = Speed in rpm

The difference between Indicated power and Brake power is called Friction power F.P.

$$F.P. = I.P. – B.P.$$

The ratio of B. P. to I. P. is called mechanical efficiency.

$$\text{Mechanical efficiency } \delta_{mesh} = \frac{B.P.}{I.P.}$$

1.10.10 Measurement of I. C. engine

To find out the performance of an engine following basic measurements are usually undertaken: -

(a) Speed
(b) Air consumption
(c) Fuel consumption
(d) Smoke density
(e) Exhaust gas analysis
(f) Brake Power
(g) Indicated power and friction power
(h) Heat going to cooling water
(i) Heat going to exhaust

(a) Speed measurement

The speed may be measurement by

(i) Revolution counters

(ii) Electrical tachometer

(iii) Mechanical tachometer

(b) Air consumption measurement

The air consumption can be measured by following methods

(i) Air box method

(ii) Viscous flow air method

Viscous flow air meter

Viscous flow air meter is another design of air meter. With the air box the flow is proportional to the square root the pressure difference across the orifice.

With the Adcock meter the air flow through a form of honey comb so that flow is viscous. The resistance of the element is directly proportional to the air velocity and is measured by means of an inclined manometer. Felt pads are fitted in the manometer connections to clamp out fluctuations.

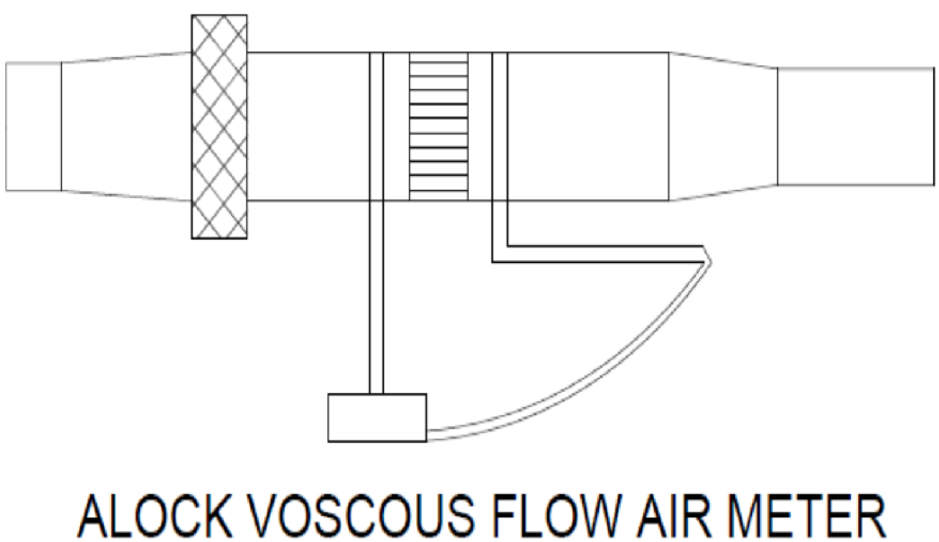


Fig. 1.19 ALOCK VOSCOUS FLOW METER

(c) Fuel Measurement

The fuel consumed by an engine can be measured by the following methods:

(i)Fuel flow method

(ii)Gravimetric method

(iii)Continuous flow meter

Air box method

It consists of air box chamber fitted with sharp edged is located away from the suction connection to the engine. There is a pressure depression due to the suction of engine in the air box or chamber. Which causes the flow through the orifice. The volume of chamber should be sufficiently large for obtaining a steady flow compared with the swept volume of cylinder. A water manometer is used to measure the pressure different causing the flow through the orifice.

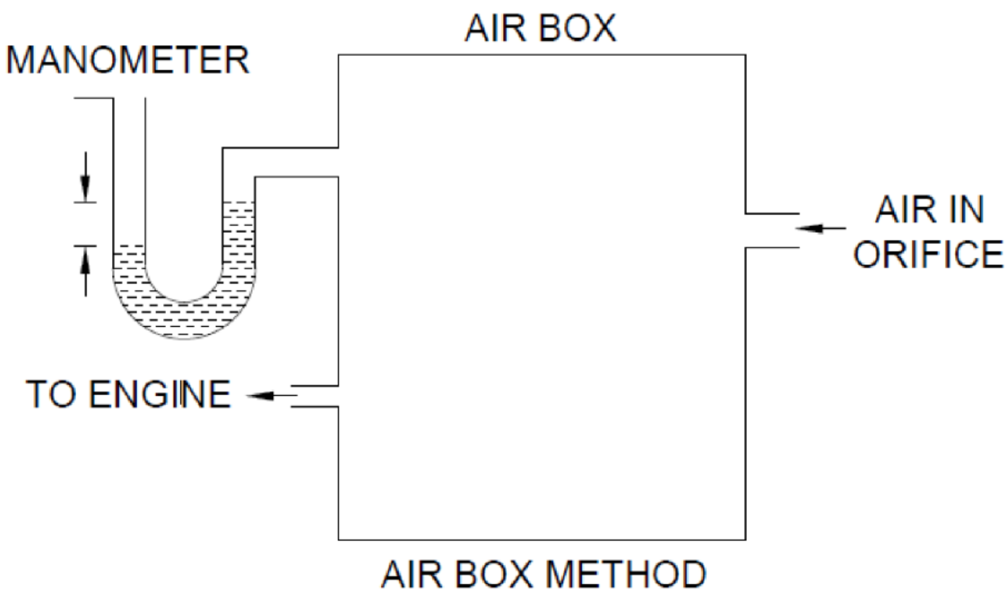


Fig. 1.20 Air box method

The volume of air passing through orifice

$$V_a = C_a X A C_a = C_d A \sqrt{2g \frac{10 h_w}{S_a}}$$

$$= k_1 \, A \, C_d \sqrt{\frac{h_w}{S_a} \frac{M^3}{S}}$$

$$= 840 \, A \, C_d \sqrt{\frac{h_w}{S_a} \frac{M^3}{min}}$$

Thermal efficiency and heat balance

Thermal efficiency is the ratio of indicated work done to energy supplied by the fuel.

$$\eta_{th} = \frac{work\ done}{m_b \times C.V.}$$

m_b= Mass of fuel used in kg/sec.

C.V. = Calorific value of fuel

Based on indicated power, the indicated thermal efficiency

$$\eta_{th}\ (I) = \frac{B.P.}{m_b \times C.V.}$$

And based on Brake power, Brake thermal efficiency

$$\eta_{th}\ (B) = \frac{B.P.}{m_b \times C.V.}$$

Heat balance sheet indicates the performance of an engine.

The quantity of fuel used in a given time and its calorific value, the amount, inlet and outlet temperature of cooling water and the weight of exhaust gases are recorded for I. C. engine. Heat indifferent items are found after calculating I. P. and B. P. as follows: -

(i) Heat absorbed in I. P.

Heat equivalent of I. P. (per minute) = I. P. × 60 k s

(ii) Heat taken away by cooling water

Heat taken away by cooling water = $m_w \times C_w(t_2 - t_1)$

$M_{w=}$ Mass of cooling water per minute

T_1= Initial temperature of cooling water

T_1= Final temperature of cooling water

C_w= Specific heat of water

(b) Mean effective pressure and torque

In is defined as hypothetical pressure which is thought to be acting on the piston throughout the power stroke. If mean effective pressure is based on Indicated power, it is called indicated mean effective pressure (Imep) and if based Brake power, it is called brake mean effective pressure (Bmep).

Friction mean effective pressure can be defined as

$$F_{mep}=I_{mep}-B_{mep}$$

Since the engine power is dependent on its size and speed, therefore it is not possible to compare engine on the basis of either power or torque.

(c)Specific output: -

Specific output is defined as the brake output pre unit of piston displacement and is given by

Specific output = $\frac{B.P.}{A \times L}$

(d) Volumetric efficiency

Volumetric efficiency is defined as the ratio of actual volume of the charge induced during suction stroke to the swept volume of piston.

(e) Fuel-Air ratio

It is the ratio of mass of fuel to the mass of air in the fuel air mixture.

The ratio of actual fuel-air ratio to that of stoichiometric fuel air ratio required to burn the fuel supplied is called Relative fuel air ratio.

Absorption dynamometer can be classified as

Prony brake, Rope brake

Hydraulic brake, Fan brake

Electrical brake dynamometer.

(a) Transmission dynamometer

These types of dynamometers are used where continuous transmission of load is necessary. These are mainly used in automatic units. There are also called torque meters.

(b) Rope brake dynamometers: -

It this type of dynamometer, rope is wound around the circumference of the brake wheel. To one end of rope is attached a spring balance and other end carries the load (w). The speed of engine is measured by tachometer.

Work/revolution = torque angle turn per revolution

$$= (w\text{-}s) \times \frac{D+d}{2} \times 2\pi$$

$$= (w - s)\,(D + d) \times \pi$$

Work/done /min $= (w - s)\pi(D + d)\pi$

Work done/min $= \frac{(w-s)\pi(D+d)N}{60}$

(c) Exhaust smoke measurement

The following smoke meters are used

(i)Bosch Smoke meter

(ii)PHS Smoke meter

(iii)Hat ridge Smoke meter

(d)Exhaust emission measurement

Substances which are emitted to the atmosphere from any opening down stream of the exhaust part of the engine are termed as exhaust emission. Some of the more commonly used instruments for measuring exhaust components are given below: -

(1) Flame ionisation detector
(2) Spectroscopic analysers
(3) Gas chromatography

(d) B.P. Measurement

The Brake power of engine can be determined by a brake of some kind applied to the brake pulley of engine. The arrangement is known as dynamometer and dynamometer are classified into following two types:

(i) Absorption (ii) Transmission

i) Absorption Dynamometers: - Those dynamometer absorb the power to be measured by friction is called absorption dynamometer. The power absorbed in friction is finally discipitated in from of heat energy. The pressure is increased when the piston reaches B. D. C. and starts back toward the TDC.

The work produced during 6-7 (Intake process) is cancelled by 7-6 (Exhaust process)

Compression ratio $=\frac{V_1}{V_2}$

Larger compression ration = $\frac{V_1}{V_2} = \frac{V_4}{V_3}$

a) Miller cycle engines are usually supercharged or turbo charged with peak intake fold pressured of

150-200 kph

b) Miller cycle engines have a higher thermal efficiency due to absence of pump work.

A greater not indicated work per cycle is obtained as a result of the shorted compression Stroke, combined with the longer expansion stroke.

(iii) Heat taken away by exhaust gases

Heat taken away by exhaust gases

$= m_e \times C_{pg}(t_e - t_r)$

m_e= Mass of exhaust gases (kg/mm)

C_{pg}= Mean specific heat at constant pressure

t_e= Temperature of exhaust gases

t_r= Room temperature

(g) Specific weight

Specific weight is defined as the weight of engine in kg for each B.P. developed. It is an indication of the engine bulk.

(h) Specific fuel consumption

Specific fuel consumption is the mass of fuel consumption per kw developed per hour, and is a criterion of economical power production.

$$\text{s.f.c.} = \frac{mf.}{b.p.} = \text{kg/kwh.}$$

(i) Exhaust smoke and other emission

Smoke is an indication of incomplete combustion. Exhaust emission have of late become a matter of grave concern and with the enforcement of legislation on air pollution in many countries, it has become necessary to view as performance parameter.

1.10.11 MILLER CYCLE

The miller cycle is a modern modification of the Atkinson cycle and has an expansion ratio greater than the compression ratio, which is accomplished, however in much a different way.

A Miller cycle engine uses unique valve's timing to obtain the same desired results, where as a complicated mechanical linkage system of some kind is required for an engine designed to operate on the Atkinson cycle.

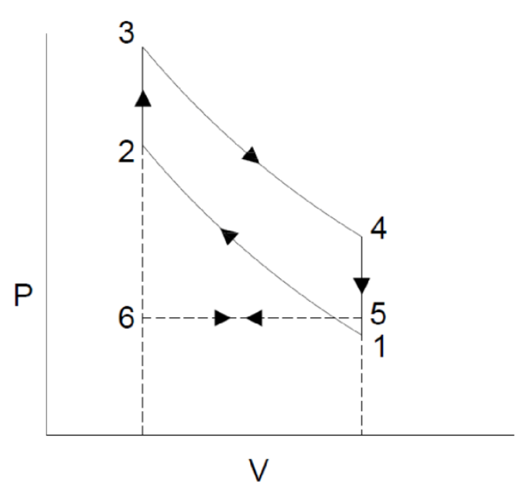


Fig. 1.21 Miller cycle

Air intake is un throttled in a miller cycle.

The quantity of air ingested into each cycle then controlled by closing the intake value at the proper time, long before B. D. C.

During the latter part of intake, when piston moves towards B. D. C. the cylinder pressure is reduced.

$$\text{B. P.} = \frac{(w-s)\pi(D+d)N}{60\times1000}\text{K W}$$

Where

W=Weight at the end of rope (N)

S=Spring balance (N)

N= Engine Speed (rpm)

D= Diameter of brake wheel (m)

D= Diameter of rope

(D+d)= Effective diameters of brake wheel

$$\text{B.P.} = \frac{(w-s)\pi(D+d)N}{60\times1000} \quad \text{if d is neglected}$$

$$= \left(\frac{T\times2\pi N}{60\times1000}\ K\ W\right)$$

1.11 ENGINE EMISSION & CONTROL

Internal combustion engines generate undesirable emissions during the combustion process the emission exhausted into the surrounding pollutes the atmosphere & causes the following problems.

1.) Global warming
2.) Acid rain
3.) Smog
4.) Odors
5.) Respiratory & other health hazards.

The major causes of these emissions are non stoichiometric combustion, dissociation of nitrogen and impurities in the fuel and air the emissions of concern are: unborn hydrocarbon (HC), oxides of carbon (Cox), oxides of nitrogen (NOx), oxides of sulpher (Sox), and solid carbon particles.

1.11.1 Air pollution due to IC engines

During the late 1940s, air pollution as a problem was first recognized in the Los Angeles basin in California. Two cause of this were the large population density and the natural weather conditions of the area. Smoke and other pollutants from many factories and automobiles combined with fog that was common in this ocean area and smog resulted. During the 1950s, the smog problem increases along with the increase in population density and automobile density. At this stage it was released that the automobile was one of the major contributors to the problem. By the 1960s emission standards were begins to be enforced in California. During the next decade, emission standard were adopted the rest of the United States and in Europe & Japan. By making engines more fuel efficient and with the use of exhaust after treatment, emissions per vehicle of HC, CO & NOx were reduced by about 95% during the 1970s & 1980s. Lead, one of the major air populates, was phased out as a fuel additive during the 1980s. More fuel-efficient engines were developed, and by the 1990s the average automobile consumed less than half the fuel used in 1970. However, during this time the number of automobiles greatly increased, resulting in no overall decrease in fuel usage.

Further reduction of emissions will be much more difficult and costly. A world population grows, emission standards have become more stringent out of necessity the strictest laws are generally initiated in California with the rest of the US & world following.

1.11.2 Engine emissions

1. Exhaust emissions
2. Non-exhaust emissions

1.11.2.1 Exhaust emissions

1. Un burnt hydrocarbon (HC)
2. Oxides of carbon (CO & CO_2)
3. Oxides of nitrogen (NO & NO_2)
4. Oxides of Sulphur (SO& SO_3)
5. Particulates
6. Soot & smoke

First four are common to both SI & CI engines and the last two are mainly for CI engines the main non-exhaust emission is the un burn hydrocarbon from fuel tank & crankcase blow by. fig shows the variation of HC, CO & NOx emissions as a function of equivalence ratio for an SI engines. It is clearly seen that all the three emissions area string functions of equivalence ratio.

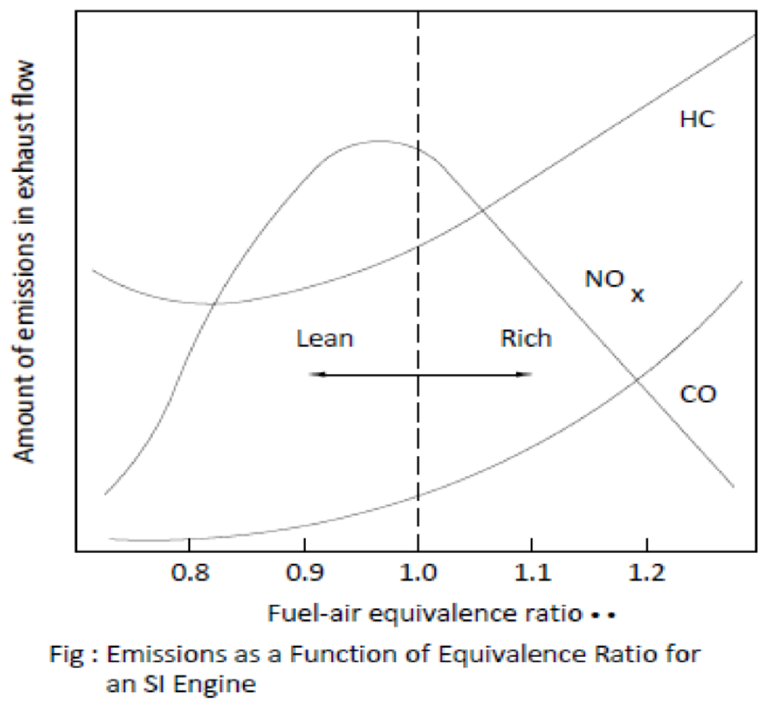


Fig. 1.22: Emission as a function of equivalence ratio for SI engine

We can notice form the fig. 11.1 that a rich mixture does not have enough oxygen to react with all the carbon & hydrogen, and both HC & CO emissions increase. For ϕ<.8, HC emissions also increase due to poor combustion & misfire the generation of nitrogen oxide emissions is a function of the combustion temperature, highest near stoichiometric conditions when temperature are at the peak value maximum NOx emission occur at slightly lean conditions, where combustion temperature is high and there is an less of oxygen to react with the nitrogen.

Fig. 1.23 shows a qualitative picture of HC, CO & NOx emissions with respect to equivalence ratio, ϕ for a four-stroke SI Diesel engine. As it can be seen HC will decrease slightly with increase in ϕ due to higher cylinder temperature making it easier to burn up any over-mixed (very lean) or under-mixed (rich) fuel-air mixture. At high loads, however HC may increase again if the amount of fuel in regions too rich to burn during primary combustion process CO emissions will be very low at all equivalence ratio since excess air is always available NOx emissions will steadily increase as ϕ increase due to increasing fraction of cylinder contents being burnt gases close to stoichiometric during combustion and also due to higher peak temperature & pressures.

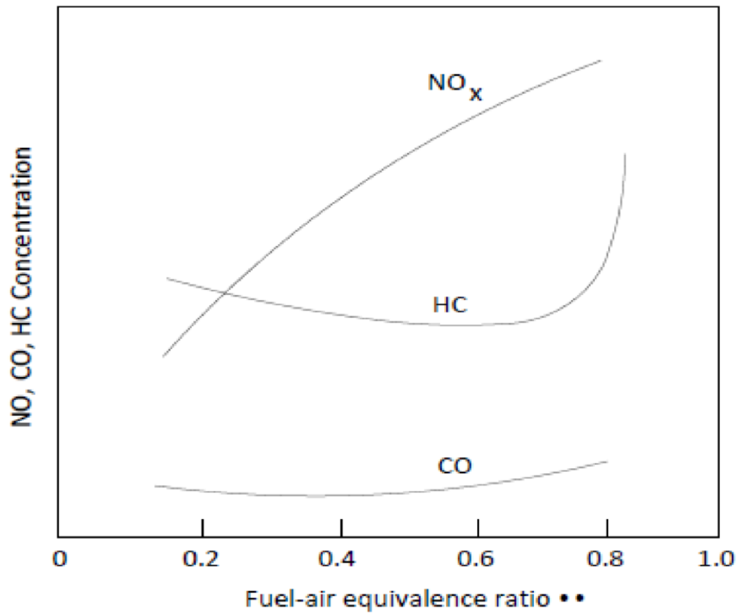


Fig : Emissions as a Function of Equivalence Ratio for a CI Engine

Fig. 1.23: Emission as a function of equivalence ratio for a CI engine

Exhaust gases leaving the combustion chamber of an SI engines contain up to 6000 ppm of hydrocarbon components, the equivalent of 1-1.5% of th3e fuel. About 40% of this is unburned components of the fuel the other 60% consists of partially reacted components that were not present in the original fuel. These consists of small non equilibrium molecules, which are formed when large fuel molecules break up (thermal cracking) during the combustion reaction. It is often convenient to treat these molecules as if they contained one carbon atom, as CH_1.

Hydro carbon emissions will be different for each gasoline blend, depending on the original fuel components. Combustion chamber geometry and engine operating parameter also inference the HC component spectrum.

When hydro carbon emissions get into the atmosphere, they act as irritants & odorants. Some are carcinogenic. All components except CH_4 react with atmosphere gases to from photochemical smog.

1.11.3 Hydrocarbon emission

Fig. shows the variation of HC emission levels with respect to equivalence ratio for an SI engine. It is evident that it is a strong function of air-fuel ratio with a fuel rich mixture there is not enough oxygen to react with all the carbon, restating in high level of HC and CO in the exhaust products. This is particularly true during stating when the air-fuel mixture is properly made very rich. It is also true to a liner extent during rapid acceleration under load. If air-fuel ratio is too lean poorer combustion occurs, again resulting in HC emissions, the extreme of poor combustion for a cycle is total misfire. This occurs more often as air-fuel ratio is made leaner. One misfire out of 1000 cycles gives exhaust emissions of 1gm/kg of fuel used.

The causes for hydro carbon emission from SI engine are:

1. Incomplete combustion
2. Crevice volumes and flow in crevices
3. Leakage past the exhaust value.
4. Valve overlap
5. Deposits on walls
6. Oil on combustion chamber walls

1.11.3.1 Incomplete combustion

Even when the fuel and air entering an engine are at the ideal stoichiometric conditions, perfect combustion does not occur and some HC ends up in the exhaust. There are several reasons for this. Complete and homogeneous mixing of fuel and air is almost impossible; the incomplete combustion is due to:

1. Improper mixing: Due to incomplete mixing of the air and fuel some fuel particles do not find oxygen to react with, this causes HC emissions.
2. Flame quenching: As the flame goes very close to the walls it get quenched at the walls leavening a small volume of unreached air-fuel mixture. The thickness of this unburned

layer is of the order of 100 microns. However, it may be noted that some of this mixture near the wall that does not originally get burned as the flame front passes will burn later in the combustion process due to additional mixing, swirl and turbulence.

Another reason for flame quenching is the expansion of gases, which occurs during combustion and power stroke. As the piston moves down from TDC to BDC during power stroke, expansion of the gases lowers both pressure and temperature within the cylinder this makes combustion slow and finally quenches the frame somewhere late in the expansion stroke. This leave some fuel particles interacted and cause HC emission.

High exhaust gas contamination cause poor combustion and which in turn cause quenching during expansion. This is experienced at low load & idle conditions. High levels of EGR will also cause quenching of frame &will result in HC emissions.

It has been found that HC emissions can be reduced by incorporating an additional spark plug at appropriate locations in the engine combustion chamber. By starting combustion at two points, the frame travel distance & total reaction time are both reduced, and less expansion quenching will result.

1.11.3.2 Crevice volumes & flow in crevices

The volumes between the piston, piston rings & cylinder wall are shown here those crevices consists of a series of volumes (numbered 1 to 5) connected by flow restrictions such as the ring side clearance & ring gap. The geometry changes as each ring moves up & down in its ring grooves, sealing either at the top or bottom ring surface, the gas flow pressure distribution & ring motion are therefore coupled.

During the compression stroke & early part of the combustion process, air & fuel are compressed into the crevice volume of the combustion chamber at high pressure. As much as 3.5% of the fuel in the chamber cab be forced into this crevice volume. Later in the cycle during the expansion stroke, pressure, and reverse blow by occur. Fuel & air, flow back into the combustion chamber, where most of the mixture is consumed in the frame. However before the last elements of reverse blow by occurs flame will be quenched & unreacted fuel particles come out in the exhaust. Location of the spark plug relative to the top compression ring gap will affect the amount of HC in engine exhaust. Further the spark plug is from the ring gap, the greater is the HC in the exhaust this is because more fuel will be forced into the gap before the frame front passes.

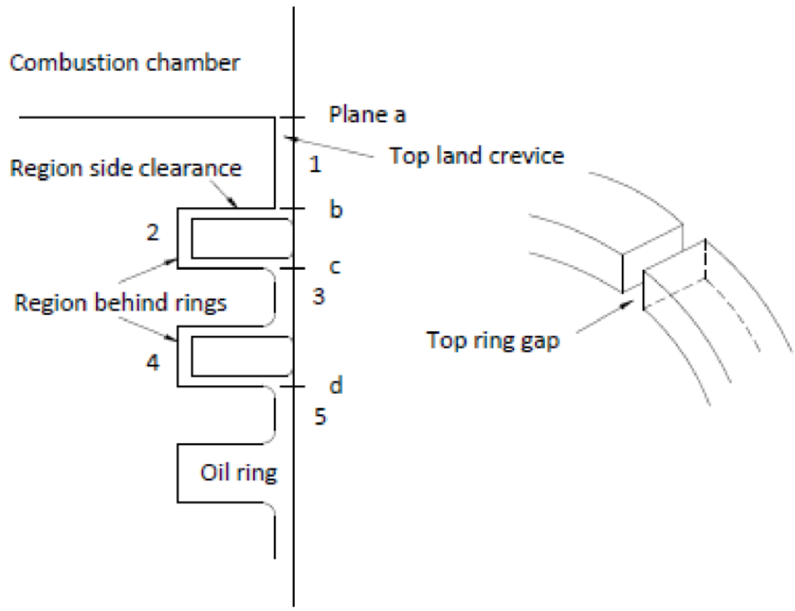


Fig: Schematic of Piston and Ring Assembly in Automotive Spark-Ignition Engine Showing Various Crevice Volume

Fig. 1.24: Schematic of piston and ring assembly in automotive Spark-ignition engine showing various Crevice volumes

Crevice volume around the piston rings is greatest when the engine is cold, due to the differences in thermal expansion of the various materials up to 80% of all HC emissions can come from this source.

1.11.3.3 Leakage past the exhaust valve

As pressure increase during compression and combustion, some amount of air-fuel mixture is forced into the crevice volume around the edges of the exhaust value & between the value and value seat. A small amount even leaks past the valve into the exhaust manifold. When the exhaust valve opens, the air-fuel mixture which is still in this crevice volume gets carried into the exhaust manifold. This cause a momentary increase in HC concentration at the start of blow down process.

1.11.3.4 Value overlap

Value overlap is a must to obtain satisfactory performance from the engine. During value overlap, both the exhaust and intake value are open, simultaneously creating a path where the fresh air-fuel mixture can flow directly into the exhaust. A well designed engine minimizes this flow, but a small amount of fresh fuel-air mixture escape is inevitable. The worst condition for this is at idle ϕ low speed, when the overlap in terms of time is the largest.

11.3.5 Deposits on walls

Gas particles including those of fuel vapor, are absorbed by the deposits on the walls of the combustion chamber. The amount of absorption is a function of gas pressure the maximum absorption occurs during compression & combustion. Later in cycle, when the exhaust value opens & cylinder pressure gets, reduced, absorption capacity of the deposits material become lower. Gas particles are desorbed back into the cylinder. These particles inducing some HC, comes out form the cylinder during the exhaust stroke. This problem is greater in engines with higher compression ratio due to the higher pressure these engines generate. More gas absorption occurs as pressure goes up. Clean combustion chamber walls with minimum deposits will reduce HC emission in the exhaust. Most gasoline blend includes additives to reduce deposits buildup in engines.

Older engines will typically have a greater amount of wall deposit buildup. These increase HC emissions correspondingly. This is due to age & to fewer swirls that was generally found in earlier engine design. High swirl helps to keep wall deposits to a minimum when unleaded gasoline is use. HC emissions form wall deposits becomes more severe. When leaded gasoline is burned the lead compounds make the walls harder & less porous to gas absorption.

1.11.3.6 Oil on combustion chamber walls

A very thin layer of oil gets deposited on the cylinder walls to provide lubrication between the walls and the moving piston. During the intake & compression strokes, the incoming air and fuel comes in contact with this oil film. In the same way as wall deposits, this oil film absorbs & desorbs gas particles, depending on gas pressure.

During compression & combustion, when cylinder pressure is high, gas particles including fuel vapors, are absorbed into the oil film when pressure is later reduced during expansion & blow down, the absorption capability of the oil is reduced & fuel particles are desorbed back into the cylinder. Some of this fuel ends up in exhaust.

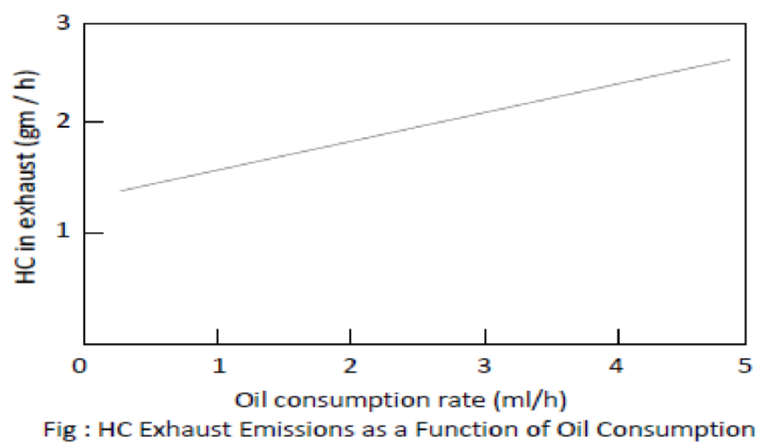


Fig 1.25: HC exhaust emission as a function of oil consumption

As an engine ages, the clearance between piston rings & cylinder walls become greater & a thicker film of oil is left on the walls. Some of this oil film is scraped off the walls during the compression stroke & gets burned during combustion. Oil is a high molecule weight hydrocarbon compound that does not burn as readily as gasoline. Some of it comes out as HC emission. This happens at a very slow rate with a new engine but increase with engine age & wear.

Fig shows how HC emissions go up as oil consumption increases, oil consumption increases as the piston rings & cylinder walls wear. In older engines, oil being burned in the combustion chamber is a major source of HC emissions. In addition to oil consumption going up as piston rings wear, blow by & reverse blow by also increases. The increase in HC emissions is therefore both form combustion of oil & from the added crevice volume flow.

Often as an engine ages, due to wear, clearance between the pistons and cylinder walls increases this increase oil consumption contributes to an increase in HC emissions in three ways.

1. There is added crevice volume.
2. There is added absorption desorption of fuel in the thicker oil film on cylinder walls, and
3. There is more oil burned in the combustion process.

1.11.4 Carbon monoxide (CO) emission

Carbon monoxide is a colorless and orderless but a poisonous gas. It is generated in an engine when it is operated with a fuel-rich equivalence ratio, when there is not enough oxygen to convert all carbon to CO_2, some fuel does not get burned and some carbon ends up as CO. typically the exhaust of an SI engine will be about .2 to 5% carbon monoxide not only is CO considered an undesirable emissions but it also represents lost chemicalenergy. CO is a fuel that can be combusted to supply additional thermal energy.

$$CO + \frac{1}{2} O_2 \qquad CO_2 + heat$$

Maximum CO is generated when an engine runs rich. Rich mixture is required during starting or when accelerating under load. Even when the intake air-fuel mixture is stoichiometric or lean, some CO will be generated in the engine. Poor mixing, local rich regions and incomplete combustion will also be the source for CO emissions.

A well designed SI engine operating under ideal conditions can have an exhaust mole fraction of CO as low as .001. CI engines that operate overall lean generally have very low CO emissions.

1.11.5 Oxides of Nitrogen (NOx)

Exhaust gases of an engine can have up to 2000 ppm of oxides of nitrogen. most of this will be nitrogen oxides (NO), with a small amount of nitrogen dioxides (NO_2). There will also be traces of other nitrogen-oxygen combinations. These are all grouped together NOx.

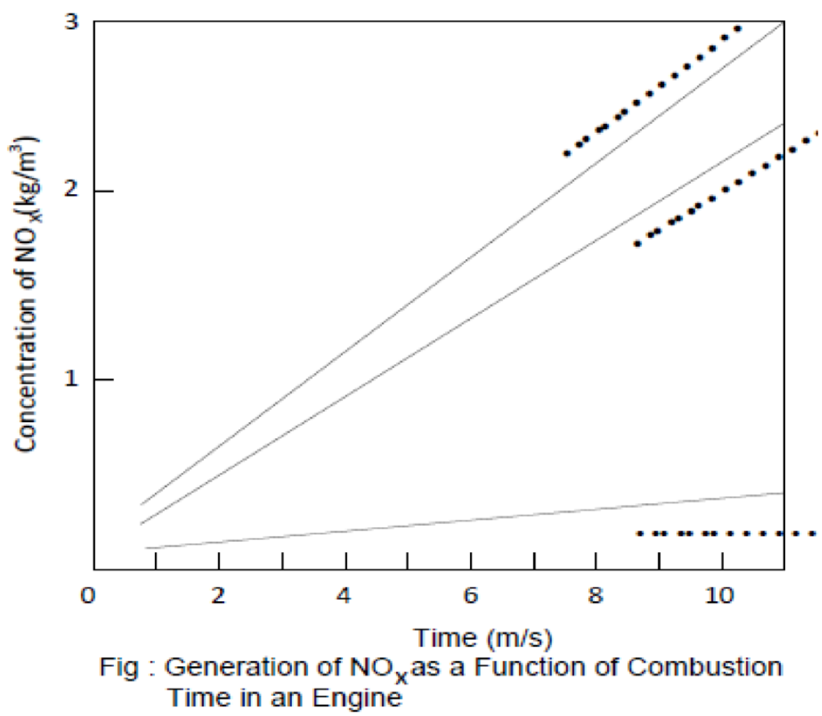


Fig 1.26: Generation of NO_x as a function of combustion time in an engine

NOx is created mostly from nitrogen in the air. Nitrogen can also be found in fuel blends further fuel may contain trace amount of NH_3, NC & HCN but this would contribute only to a minor degree there are a number of possible reaction that form no these include but are not limited to

$O+N_2$ $\quad$ $NO+N$

$N+O_2$ $\quad$ $NO+O$

$N+OH$ $\quad$ $NO+H$

NO, in turn, can further react to form NO_2 by various means, including

$NO+H_2O$ $\quad$ NO_2+H_2

$NO+O_2$ $\quad$ $NO_2+ O$

At low temperature, atmospheric nitrogen exists as stable diatomic modules. At low temperature very little NOx is created.

In addition to temperature the formation of NOx, depends on pressure and air-fuel ratio. Combustion duration plays a significant role in NOx formation within the cylinder. fig. shows the NOx versus time relationship & supports the fact that NOx is reduced in modern engines with fast-burn combustion chamber. The amount of NOx generated also depends on the location of spark plug within the combustion chamber. The highest concentration is formed around the spark plug where the highest temperature occurs.

Also NOx is one of the primary causes of photo chemical smog, which has become a major problem in many large cities of the world. Smog is formed by the photo chemical reaction of auto mobile exhaust & atmospheric air in the presence of sunlight. NO_2 decomposes into NO & monatomic oxygen.

NO_2+ energy form NO + O+ Smog sunlight

Monatomic oxygen is highly relative & initiate a number of different reactions, one of which is the formation of ozone

$$O+O_2 \qquad O_3$$

1.11.6 Particulates

Soot particles are clusters of solid carbon spheres these spheres have diameters from 9 nm to 90 nm. ($1nm=10^{-9}m$). But most of them are within the range of 15-30nm. The spheres are solid carbon with HC & traces of other components absorbed on the surface. A single soot particle may contain up to 5000 carbon spheres.

Carbon spheres are generated in the combustion chamber in the fuel-rich zones where there is not enough oxygen to convert all carbon to CO_2:

$$Cx\ Hy + 2O_2 \qquad a\ CO_2 + Bh_2O + c\ Co + dC(s)$$

Then as turbulence and mass motion continue to mix the components in the combustion chamber, most of these carbon particles find sufficient oxygen to further react and are converted to CO_2:

$$C(s) + O_2 \qquad CO_2$$

More than 90 to 95% of carbon particles originally generated within an engine are thus converted to CO_2 & never comes out as carbon particles.

Up to about 25% of the carbon in soot comes from lubrication oil components which vaporize & then react during combustion. The rest comes from the fuel & amounts to .02-0.5% of the fuel.

At light loads, cylinder temperature are reduced & can drop to as low as 200.C during final expansion & exhaust blow-down under these conditions, so f can be as high as 50% of the total mass of soot.

Particulates generation can be reduced by engine design & control of operation conditions, but quite often this will create other adverse results.

Aldehydes

A major emission problem when alcohol fuel is used is the generation of aldehydes, an eye and respiratory irritant.

This is a product of incomplete combustion and would be a major problem if as much alcohol fuels were used as presently as gasoline.

1.11.7 Sulpher

Many fuels used in CI engines contain small amounts of sulpher. When exhaust in the form of SO_2 & SO_3 (SOx) they contribute to the acid rain problem of the world. Untraded gasoline generally contains 150-500 ppm sulpher by weight. Some diesel fuels contains up to 5500 ppm by weight, but in the developed countries this is restricted by low to a tenth of this value or less at high temperature, surphur combines with hydrogen to form H_2S and with oxygen to form SO_2.

$$H_2+S \qquad H_2S$$

$$O_2+S \qquad SO_2$$

Engine exhaust can contain up to 20 ppm of SO_2. SO2 then combines with oxygen in the air to form SO_3

$$2SO_2+O_2 \qquad 2SO_3$$

These combine with water vapour in the atmosphere to form sulpheric acid (H_2SO_4) & sulphurous acid (H_2SO_3), which are ingredients in acid rain

$$SO_3+H_2O \qquad H_2SO_4$$

$$SO_2+H_2O \qquad H_2SO_3$$

1.11.8 Lead

Lead was a major gasoline additive form its introduction in 1920s to when it was phased out in the 1980s the additive TEL (tetraethyl lead) was effectively used to increase gasoline octane number, which allowed higher compression ratio's & more efficient engines. However, the resulting lead in the engine exhaust was a highly poisonous pollutant.

1.11.9 Phosphorus

Small amounts of phosphorus are emitted in engine exhaust these come from impurities in fuel blends & lubricating oil.

1.11.10 Gasoline engine emission control

An emission control programmer aims at reducing the counteraction of CO, HC & NOx in the exhaust the main approaches which have been used for this purpose are:

1. Engine design modification
2. Treatment of exhaust gas
3. Fuel modification

1.11.10.1 Engine design modification

The engine modification approach for improving the exhaust emission in aimed at following:

1. Use of leaner air-fuel ratio:-
   The carburetor may be modified to provide relatively lean and stable air-fuel mixtures. during idling and reuse operation with this modification idle speed needs to be increased to prevent starting & rough idle associated with leaner fuel-

air ratios. fuel distribution is improved by better manifold design, inlet air heating, raising of coolant temperature & use of electronic fuel injection system.

2. Retarding ignition timing:-

   Retarding ignition timing allows increased time for fuel burning. The controls are designed to retard the spark timing at idle while providing normal spark advance during acceleration & cruising. Retarding the spark reduces NOx emission by decreasing the maximum temperatures. It also reduces HC emission by causing higher exhaust temperature. However, retarding the ignition timing results in greater cooling requirement & there is some loss in power & fuel economy.

3. Modification of Combustion Chamber configuration to reduce quenches areas:

   Modification of combustion chamber using attempts to avoid flame quenching zones where combustion might otherwise be incomplete & resulting in high HC emission. This includes reducing surface to volume ratio, reduced squish area, reduced deal space around piston rings and reduced distance of the top piston ring from the top of the piston.

4. Lower compression ratio:-

   The lower compression ratio reduces the quenching effect by reducing the quenching area, thus reducing HC. Lower compression ratio also reduces NOx emissions due to lower maximum temperature. However, reducing the compression ratio result in some loss in power and fuel economy. But there is advantage of reduced octane number which will make it easier to phase the lead out of petrol i.e. use of unleaded gasoline.

5. Reduced value overlap:-

   Increased valve overlap allows some mixture to escape directly & increase emission level this can be controlled by reducing value overlapping.

6. Alternative in induction system:-

   The supply of designed air-fuel ratio to all cylinder under all operating conditions can be affected by alterations in induction system which include inlet air heaters, use of carburetors which have closer carburetion tolerances & special type of carburetors e.g. high velocity carburetors or multi-choke carburetors. This also includes the fuel injection in manifold.

1.11.10.2 Exhaust gas oxidation

Exhaust gas form the engine manifold is treated to reduce HC/CO emissions. A number of device have been listed these are.

1. After burner:-
   An addition of an after-burner to the exhaust system can completely burn the partially burned HC in the exhaust gases. After burns have not been successful in curbing the emission due to the difficulty in sustaining the combustion during low HC emissions of high heat losses over a large area.
2. Exhaust manifold reactor:-
   All these reactors work on the fact that if air is mixed into the high temperature HC, they will react to complete the oxidation of the HC. This is actually development of the after-burner concept. The changes in design of the after burner to minimize the heat loss have led to the treatment of exhaust gases just after the manifold and the need of providing sufficient time for oxidation & mixing through different shapes have been evolved. In an earlier type of reactor developed by all-pout, the entry of exhaust gases was radial & the air flow peripheral.
3. Catalytic convertor:-
   It consist of two separate elements one for Nox and other for HC/CO emission the secondary air is injected ahead of the first elements the flow in the convector is a real.

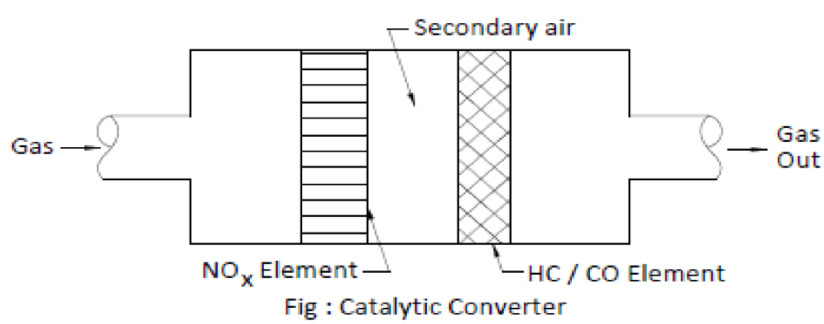


Fig 1.27 Catalytic Converter

1.11.10.3 Exhaust emission control by fuel variation

It is a well established fact the fuel-air ratios leaner then than stoichiometric results in almost insignificant amount of CO & reduces HC with reduced specific fuel consumption. Hence the ability of a fuel to burn in mixture leaner than stoichiometric ratio is a rough indication of its potential emission reducing characteristics & reduced fuel consumption.

Both methane & steam reformed hexane are very attractive fuels form air pollution point of view but for what of technological program we are unable to use them at present.

1.12 Total emission control package

With the pollution control laws becoming stringent there has arisen a danger of piston engine being outdated by steam & electric cars in spite of its better fuel consumption.

Automobile companies have developed two systems after a long and detailed experimental study of various possible systems these are:

1. Thermal reactor package
2. Catalytic convertor package

There are three basic methods of emission control using this approach:

1. Thermal reactors which rely on homogeneous oxidation to control CO & HC.
2. Oxidation catalytic for CO & HC.
3. Dual catalyst systems, which incorporate in series a reduction catalyst for NOx & an oxidation catalyst for CO & HC where NOx control is needed with the first two methods, exhaust gas recirculation (EGR) is added to the system.

1.12.1 Thermal reactor package

A thermal reactor is a chamber in the exhaust system designed to provide sufficient reside time to allow appreciable homogenous oxidation of CO to HC to occur. In order to improve CO conversion efficiency, the exhaust temperature is increased by spark retard. This, however results in fuel economy loss the schematic layout of this system.

1.12.2 Catalytic convertor packag

This principle of catalytic convertor package is to control the emission levels of various pollutants by charging the chemical characteristics of the exhaust gases.

In contrast to thermal reactors, efficient catalytic oxidation catalysts can control CO & HC emissions almost completely at temperature equivalently to normal exhaust gas temperature thus the fuel economy lose necessary to increase the exhaust temperature is avoided.

Catalyst materials such as platinum and palladium are applied to a ceramic support which has been treated with an aluminum oxide wash coat. This results in an extremely porous structure providing a large surface area to stimulate the combination of oxygen with HC and CO. This oxidation process converts most of these compounds to water vapour and carbon-dioxide.

The main advantage of the catalytic convector over the thermal reactor is that the former allows a partial decoupling of emission control from engine operation in that the conversion efficiencies for HC & CO are very high at normal exhaust temperature.

1.12.3 Other emission control devices

1. Direct air injection:- if compressed air is introduced into the combustion chamber in addition to air-fuel charge from the carburetor, better combustion and hence reduced hydrocarbon & carbon monoxide emission will result. This will also give a tremendous power boost with some saving in fuel. But extra equipment in the form of air compressor and air values will raise the cost very much. Also exhaust gas reregulation will still be needed to curb NOx emissions.
2. Ammonia injection: - as a fuel, ammonia does not hold much promise, but if used as in exhaust additive, it can give excellent control for NOx emission. Ammonia and nitric oxide interact to form nitrogen and water.

   For an effective utilization of ammonia injection, the exhaust gas temperature has to be kept within strict limits and the injection device has to be put sufficiently down to bring the gas temperature to 165c. This also demands a very close tolerance in air-fuel ratio supplied by the carburetor the present carburetor are incapable to this & it might be necessary to adopt electronic injection system to keep close control over fuel-air ratio.

3. Electronic injection:- It is possible to develop an electronic injection system with sensors for air temperature, manifold pressure & speed which will precisely regulate the fuel supply giving only such air fuel ratio as will give no hydrocarbon or carbon monoxide emission.

   Since the injection can be affected in individual intake port, the problem of fuel distribution among various cylinders will automatically be avoided.

   The emissions on deceleration can be completely removed by shutting off the fuel supply when the throttle is closed. But this system will still not be able to control the NOx emission. Combination of electronic injection and ammonia as an exhaust additive has a relative future.

1.13 SI engine emissions

SI engine emissions are divided into three categories as exhaust emission, evaporative emission & crank case emissions.

The major constituents which contribute to air pollution are CO, NOx & HC coming from SI engine exhaust.

The percentage of different constitutes coming out from the above three mentioned sources are:

1. Hydro carbon 15 to 25% (form fuel tank & carburetor)
2. Hydro carbon 50 to 60%
   Carbon Monoxide 10%
   Nitrous oxide 10%
   (Form exhausts emission)

The other constituents includes SO_2 & lead compounds. The petrol rarely contains sulpher therefore; SO_2 is not a pollutant form S.I. engine exhaust. Petrol contains lead in small percentages but its effect is more serious on human health therefore Delhi Govt. has restricted the use of petrol with lead.

The processes by which pollutants form with in the cylinder in conventional SI engine are illustrated qualitatively. The figure shows the formation of pollutants during 4-strokes of the cycle. NO forms throughout the high temperature burned gases behind the flame through chemical reactions. NO formation rate increase with an increase in gas temperature. As the burned gases cool, during expansion stroke, the reactions involving NO freeze and leave no concentrations far in excess of level corresponding to equilibrium temperature at exhaust conditions.

CO also forms during combustion process with lean A:F mixtures. But in high temperature products even with lean mixtures there is sufficient CO in exhaust become of dissociation of CO_2 later in expansion stroke, the CO oxidation process also freezes as the gas temperature falls.

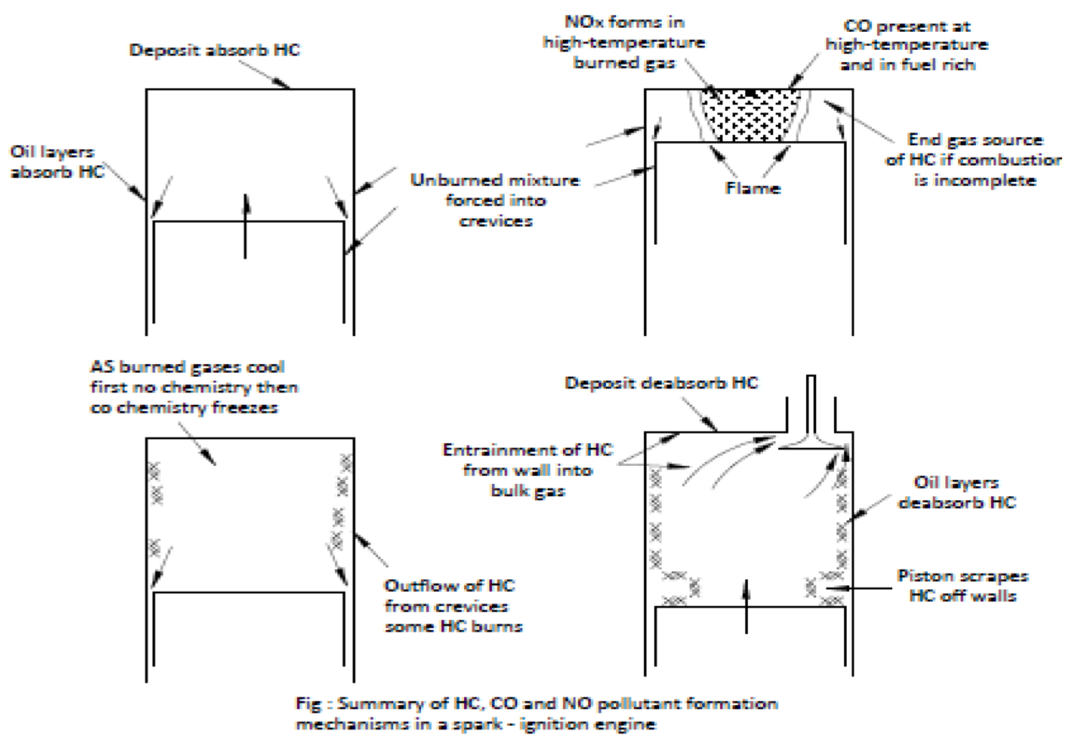


Fig. 1.28: Summary of HC, CO and NO pollutant formation mechanism in a spark ignition engine

The burned hydrocarbon emission comes from different sources. During compression and combustion, the increasing cylinder pressure forces some of the gases in the cylinder into crevices connected to combustion chamber, the volumes between the piston rings & cylinder wall are largest of these. Most of this gas entering into crevices is unburned. Air-fuel mixture escaped from primary combustion zone. This is because the flame cannot enter into these narrow crevices. this gas which leaves these crevices later in the expansion & exhaust presses, is one source of unburned carbon emission. Another possible source is the combustion chamber walls. A quench layer containing unburned & partially burned A:F mixture is left at the wall when the flame dies as it approaches the wall. This unburned HC in this layer burns rapidly if the combustion chamber walls are clean. The next source of HC is the layer of lubricate oil on cylinder wall, and piston which absorbs HC before and after combustion.

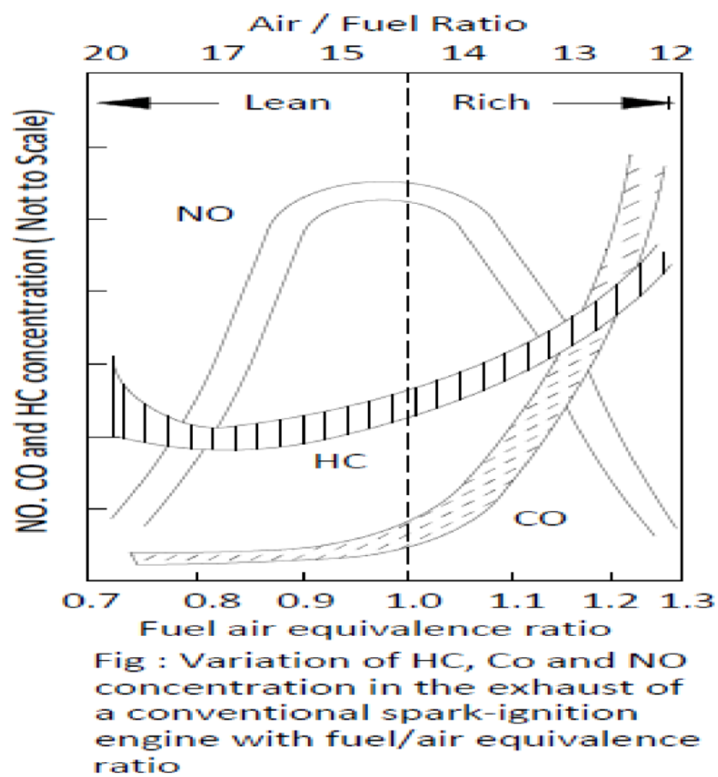


Fig. 1.29: Variation of HC, CO and NO concentration in the exhaust of a conventional spark ignition engine with fuel/air equivalence ratio

One of the most important variables in determining SI engine emission is the fuel air equivalence ratio (ϕ). Fig shows how NOx, CO & HC emissions vary with this parameter. The SI engines are always operated at stoichiometric or slightly rich mixture. It is deal form the figure that leaner mixtures give lower emissions until the combustion quality becomes poor.

At the starting of the engine, very rich mixture is supplied as vaporization is very slow thus, until the engine warms up and this enrichment is stopped, CO & HC emissions are high. At part load conditions, lean mixture can be used which will reduce HC & CO emission & modulate NOx emissions. Use of regarded exhaust to dilute the engine intake mixture lowers the NOx level but Detroites combustion quality. Exhaust gas recirculation (EGR) method is used with stoichiometric mixtures in many engines to reduce emission.

The source of pollution are mainly three as mentioned earlier, the engine exhaust, (CO, NOx, HC), the crank case breather (HC) and direct evaporation of petrol from carburetor & fuel tank particularly in hot weather (HC).

1.14 Exhaust emission and factors affecting the emission

If the combustion is complete, the exhaust will contain only CO_2 & H_2O vapour. But when rich mixture is supplied, there is no sufficient O_2 for complete combustion & part of the carbon converts to CO. whereas, lean mixtures, form NO as excess O_2 is available to react with N_2 particularly at high temperature the different constituents which are exhausted from SI engine & different factors which will affect the percentages of different constituents are:

1. Hydrocarbon:- the unburned hydrocarbon emission is the direct result of incomplete combustion. The emission amount of hydrocarbon is closely related to design variables as indention system & combustion chamber design and operating variables as A:F ratio, speed, load & mode of operation as idling, running or accelerating the design of combustion chamber is important as fuel air mixture when comes in contact with the walls get quenched & do not burn. This unburned quenched mixture is forced out of the combustion chamber during the exhaust. Because of high local concentration of hydrocarbon leads to exhaust unburned hydrocarbons.

The following factors affect HC emissions

a. Surface / volume ratio
b. Wall quenching
c. Incomplete combustion
d. Spark plug timing
e. Effect of compression ratio.

2. Carbon monoxide (CO):- carbon monoxide, being an intermediate product of combustion, remains in the exhaust if the oxidation of CO & CO_2 is not complete.

   Theoretically, it can be said that the petrol engine exhaust can be made free form CO by operating it A:F , A:F ratio = 15. However, some CO is always present in the exhaust even at lean mixture & can be as high as 1%.

   The percentage of CO increases during engine idling but decrease with speed. CO-emission is high during idling & reach maximum during declaration they are lowest during acceleration & at steady speed.

3. Oxides of Nitrogen: - oxides of N_2 generally occur mainly in the form of NO & NO_2. These are generally formed at high temperature. Hence higher temperature & availability of free O_2 are the main two reasons for the formation of NO & NO_2.

   The factors affecting the formation of NOx are A: F ratio, RPM & angle of advance.

1.15 Evaporative emission

There are two sources of evaporative emissions, the fuel tank & carburetor. The main factor governing the tank emissions are fuel volatility & ambient temperature but the tank design & location can also influence the emission as location affects the temperature.

Carburetor emissions may be divided into categories as running losses & parking losses. Although most internally vented carburetor have an external vent which open at idle throttle position. Carburetor losses are significant only during hot condition when the vehicle is in operation the fuel volatility also affects the carburetor emissions.

1.16 Crank case Emission

It consists of engine blow by-gases & crank-case lubricant fumes. From pollution point of view, blow by gases are most important. The blow-by is the phenomenon of leakage past the piston from the cylinder to the crankcase because of pressure difference. The blow by of HC emissions are about 20% of the total HC emission from the engine. This is further increases to 30% if the piston-rings are worn.

1.17 Lead Emission

Lead emission come only from SI engines. The lead is present in the fuel as lead tetraethyl or tetraethyl to control the self ignition tendency of fuel-air mixture that is responsible for knock.

It would be impossible to eliminate lead completely form all patrols immediately, because a large number of existing engines rely upon the lubrication provided by a lead film to however a very small lead content would suffice for that purpose.

Pollution due to Diesel Emission:-

Exhaust emission from diesel engines generally contain the same constituents as the gasoline engines, but the level of the constituents vary widely from a low to a high value. The range of concentration of different constituents in a typical diesel exhaust, may be as follows.

- CO from almost 0% to 2%
- HC from a few ppm to about 1000 rpm
- NO_x from about 100 ppm to about 2000 ppm
- RCHO from a few ppm to about 100 ppm

Factors affecting the exhaust emission in diesel engines.

Type of engine: - whether scavenged or naturally aspirated or turbocharged, 2-stroke type or 4-stroke type etc.

1. Mode of operation:- weather operated at full load or part load, under idling or acceleration conditions .
2. Engine operating variables.
3. Engine maintainace.

1.18 Effect of engine operating variable on exhaust emissions

The effect of diesel engine variables on exhaust emissions is almost the same as that of petrol engine. However, there are some differences due to different combustion chamber designs of diesel engines eg.

1. NO_X produced in a pre-combustion chamber engine is less than that produced by a direct-injection engine. It is because of lower peak temperature in latter case.
2. NO_X concentration is lower at high A/F ratio. It is because of the reason that the additional fuel tends to cool the charge, therefore the peak temperature gets lowered.
3. The concentration of NO_X increase linearly with increase in power output of the engine.
4. The NO_X emission is higher in a 4-stroke high speed naturally aspirated engine than a 4-stroke turbocharged engine.

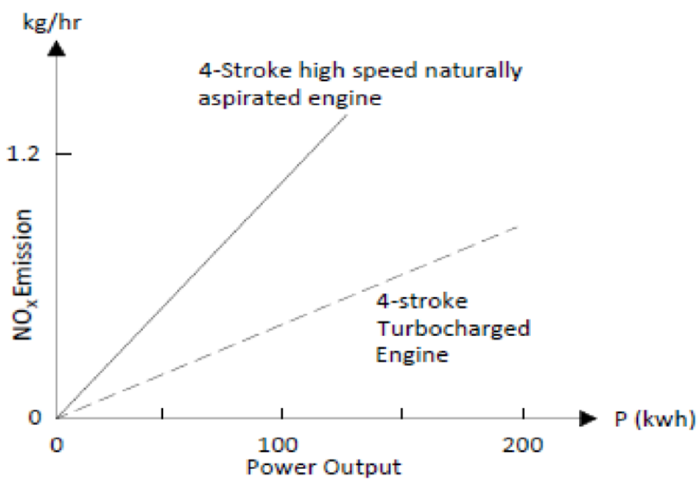


Fig : Effect of Engine power output on NO_xemission in different engines

Fig. 1.30: Effect of engine power output on NO_x emission in different engines

1.19 Diesel soot and smoke, their formation & control

Smoke is a product of combustion which forms due to incomplete combustion of fuel. It is visible constituents of exhaust emission. It is a gaseous phase consisting of very small particles of partially burnt fuel called "Soot". This soot completely burns in presence of oxygen, if available during combustion cycle. However if sufficient oxygen is not available, it goes out in atmosphere with exhaust. It the quality of soot going with exhaust is substantial, it become visible and the exhaust appears as “smoke”.

1.19.1 ype of smoke

1. Blue-White Smoke: - blue white smoke is formed owing to suspension of intermediate products of combustion with liquid dropouts of lubricating oil. It emits as exhaust when the engine starts from a cold state or when more than sufficient quality of lubricating oil flows past the worn piston rings.
2. Black smoke: - black smoke is caused due to suspension of carbon particle in exhaust gases. Darkening effect in it is caused due to bigger soot particles. The amount of black smoke increase with increase in engine load & depletion of air.

1.19.2 Factor affecting formation of smoke

1. Quality of fuel
   - A. More volatile fuels produce less smoke.
   - B. Heavier fuels produce more smoke.
   - C. High volatility fuel produces less “white smoke”.
   - D. High cetane number fuel produces less “white smoke”.
   - E. Cetane number has almost no effect on “black smoke”.
   - F. The fuel of about 45 cetane number is optimum to produce “whit smoke” within acceptable limits.
2. Load on the engine
   - A. Overloaded engine produces “black smoke”.
   - B. Intensity of smoke increases with rise in load from “no load” to “full load” condition. Initially it rises slowly but in higher load range the rise is at much faster rate. It is because of less availability of oxygen.
3. A/F ratio
   - A. A richer mixture produces more smoke. It is because the amount of oxygen available is less.
   - B. The level of smoke increase with increase in A/F ratio.
   - C. The smoke density decreases with increase in F/A ratio for turbo charged engine, where as it increase for naturally aspirated engine.

4. Type of engine

A. Turbo charged engines have lower smoke level than the naturally aspirated engines at higher loads. It is because in the turbo charged engines ample of oxygen available even at full load.

5. Engine speed

A. The smoke density is too much at extreme engine speeds viz low & high.

B. At low engine speed, it is too much because of “charge heating”. But at high engine speed, it is too much because of wire drawing at inlet value.

6. Fuel injection system

A. The smoke level substantially increases due to following occurrences.

B. Inadequate injection

C. Excess penetration

D. Injection for a longer duration

E. Improper atomization.

6. Altitude

A. Engines running at higher altitude have greater smoke level.

7. Maintenance

A. Properly maintained engines have a lower smoke level.

B. Engines having worn piston rings have a higher smoke level. It is because the lubricating oil also burns with the fuel.

1.19.3 Mechanism of soot & smoke formation

The basic structure of soot is similar to that of graphite which has hexagonal formation.

Mechanism I-the mechanism of soot formation is normally attributed with following reactions.

$$2CO \qquad CO_2 + C$$

It suggests that the formation of carbon monoxide is catalyzed by carbon; therefore they build up rapidly & polymerize.

Mechanism-2

Another mechanism of soot formation suggest that the heavy ended hydrocarbons decompose into smaller basic units of C_2 & C_3 radials, which polymerize to form a C_6 ring type polymer. A typical reaction give below illustrates this mechanism.

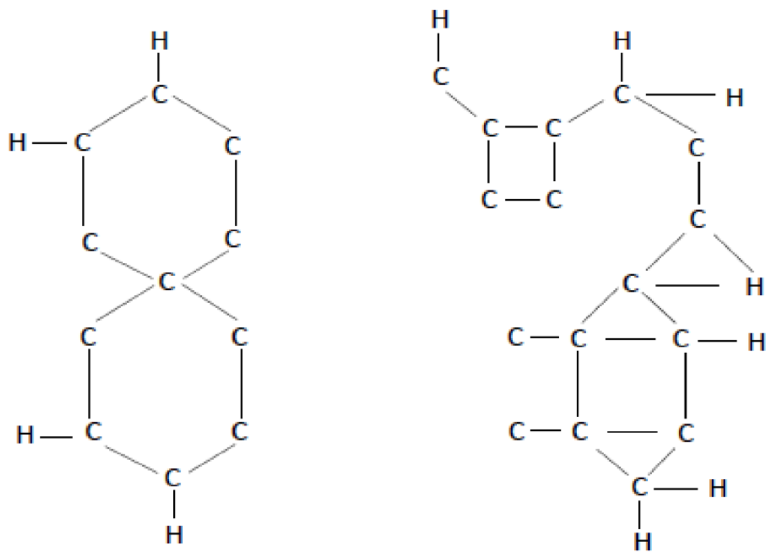


Fig : Structure of soot

Fig. 11.10: Structure of soot

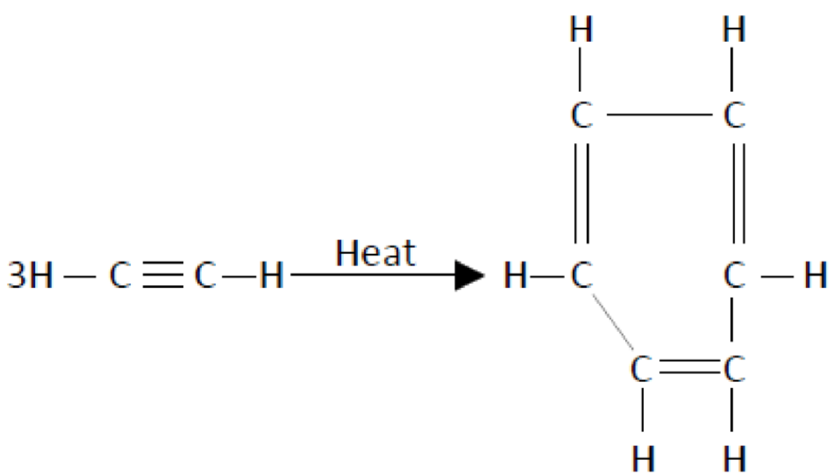


Fig. 1.31: Mechanism of Soot formation

1.19.4 Characteristics of soot

Some special characteristics are

1. It has a free valiancy.
2. It has agglomeration property.
3. It can be absorbed in metal surfaces.
4. It is a powerful absorption agent.

It is very difficult to remove it from the metal surface on which it sticks. The chemical composition of dry soot can be expressed as follows.

- $CH_{0.27}\ O_{0.22}\ N_{0.01}$ at idling.
- $CH_{0.21}\ O_{0.15}\ N_{0.01}$ at about 40 kmph.

1.20 Diesel odors and its control

Diesel exhaust has a pungent odors. Member's aldehydl family present in exhaust is supposed to be the source for such odors. The diesel odors cause irritation to human eyes & nose because of the presence of aldehyde up to about 30 ppm in it.

1.20.1 Mechanism of odors existence

Following phenomenon is supposed to be the cause of odor in diesel exhaust.

1. Products of partial oxidation on account of very lean mixture used during idling or part load operation of the engine.
2. There are certain regions in which the fuel, O_2 and lean mixture are outside the flammability limits. This results in creations of odors product due to incomplete combustion.

3. Effect of quenching on chemical reaction that takes place during combustion, also result in odors formation in diesel exhaust.

1.20.2 Factors affecting the production of odors

The odors production depends on the following factors

1. A/F ratio: - a very lean mixture results in odorless diesel exhaust. A correlation between F/A ratio and odors in diesel exhaust.
2. Engine operation mode:-
   a.) Maximum order produce during acceleration form idling.
   b.) Minimum odors products during medium speed or part load running of the engine.
3. Type of engine: - intensity of odors remains almost the same in all types of engines.
4. Fuel consumption:- It has no effect on odors intensity of exhaust.

1.20.3 Control of odors

Odor can be controlled by adopting the following means.

1.) Use of catalytic control system
2.) Use of catalytic container.
3.) Use of odors suppressive additives.