ANNAMACHARYA UNIVERSITY

EXCELLENCE IN EDUCATION; SERVICE TO SOCIETY
(ESTD UNDER AP PRIVATE UNIVERSITIES (ESTABLISHMENT AND REGULATION) ACT, 2016)
RAJAMPET-516126:A.P; INDIA

DEPARTMENT OF MECHANICAL ENGINEERING

LECTURE NOTES

COURSE TITLE	**: I.C. Engines & Gas Turbines**
COURSE CODE	**: 24AMEC52T**
REGULATION	**: AU-24**
YEAR & SEMESTER	**: III YEAR I SEM**
CREDITS	**: 3**
ACADEMIC YEAR	**: 2026-27**
PREPARED BY	**: Mr.B.Santosh Kumar**

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Title of the Course: I.C. Engines & Gas Turbines
Category: PC
Year III
Semester: I
Course Code: 24AMEC52T
Branch/es: Mechanical Engineering

Lecture Hours	Tutorial Hours	Practice Hours	Credits
3	0	0	3

Course Objectives:

1. To Gain knowledge on basic concepts of an I C Engine.
2. To Acquaint with concept of combustion in SI & CI engines.
3. To Familiarize about various methods for measurement of engine performance.
4. To Provide insight on Air compressors.
5. To Provide knowledge on Gas turbines.

Course Outcomes:

At the end of the course, the student will be able to

1. Summarize the concepts of IC engines.
2. Comprehend the combustion in SI & CI engines.
3. Calculate the performance of IC Engines.
4. Calculate the performance of Air compressors.
5. Calculate the performance of Gas Turbines.

Unit 1 Introduction to I.C. Engines 14

Classification - Working principles, Valve and Port Timing Diagrams, Engine systems – Fuel, Simple Carburetor, Fuel Injection System – Air Injection system, Solid Injection system and Electronic Injection system. Ignition –Battery ignition system and Magneto ignition system, Cooling – Air cooling (Cooling Fins) and liquid cooing system – Thermo syphon system and Forced Circulation system and Lubrication - Importance - Mist Lubrication System, Wet sump Lubrication system and Dry sump Lubrication system.
Actual Cycles and Their Analysis: Introduction, Comparison of Air Standard and Actual Cycles, Time Loss Factor, Heat Loss Factor, Exhaust Blow down - Loss due to Gas exchange process, Volumetric Efficiency. Loss due to Rubbing Friction, Actual and Fuel-Air Cycles of CI Engines.

Unit 2 Combustion in SI & CI Engines 11

Combustion in SI engine: Normal Combustion and Abnormal Combustion – Importance of Flame Speed and Effect of Engine Variables – Type of Abnormal Combustion, Pre-Ignition and Knocking (Explanation) – Fuel Requirements and Fuel Rating, Anti Knock Additives, Combustion Chambers.
Combustion in CI Engines: Stages of Combustion – Delay Period and Its Importance – Effect of Engine Variables – Diesel Knock– Combustion Chambers (DI And IDI), Fuel Requirements and Fuel Rating.

Unit 3 Testing and performance of IC engines. 12

Testing and Performance: Parameters of Performance - Measurement of Cylinder Pressure, Fuel Consumption, Air Intake, Exhaust Gas Composition, Brake Power – Determination of Frictional Losses and Indicated Power – Performance Test – Heat Balance Sheet and Chart**.**

Unit 4 Air Compressors 11

Reciprocating Compressors, Effect of Clearance volume in Compressors, Volumetric Efficiency, Single Stage and Multi Stage Compressors, Effect of Inter cooling and Pressure Drop in Multi - Stage Compressors,

Problems Related to Reciprocating Compressors, Working principles of Roots blower, Vane type Blower, Centrifugal Compressor - Axial Flow Compressors

Unit 5 Gas Turbines **12**

Gas Turbines: Simple gas turbine plant, Brayton cycle, closed cycle and open cycle for gas turbines, actual cycle. Methods to improve performance: regeneration, intercooling and reheating.

Introduction to jet propulsion: working principle of ramjet, turbojet, turboprop and pulse jet engines, Thrust, Thrust Power and Propulsion Efficiency

Prescribed Textbooks:

1. I.C. Engines - V. Ganesan- TMH- 4th edition -2017- ISBN-13. 978-1259006197
2. Thermal Engineering-Rajput-Lakshmi Publications -11th edition -2020- ISBN-13. 978-8131808047.

Reference Books:

1. Internal Combustion Engine Fundamentals- John B. Heywood- TMH-2018-ISBN: 9781260116106
2. Thermodynamics & Heat Engines- B. Yadav- Central Book Depot., Allahabad -2002-ISBN-13: 978-8185444031
3. Treatise On Heat Engineering in MKS And SI Units, 4/e, V P Vasandani, Metropolitan Book Co. (p) Ltd. ISBN-13: 978-8120003507

CO-PO Mapping:

Course Outcomes	Engineering Knowledge	Problem Analysis	Design/Development of solutions	Conduct investigations of complex problems	Engineering Tool usage	The Engineer and The World	Ethics	Individual and Collaborative team work	Communication	Project management and finance	Life-long learning	PSO1	PSO2
24AMEC52T.1	3	2	-	-	-	-	-	-	-	-	1	2	-
24AMEC52T.2	3	2	-	-	-	-	-	-	-	-	1	2	1
24AMEC52T.3	3	3	2	1	-	-	-	-	-	-	1	2	2
24AMEC52T.4	3	3	2	2	-	-	-	-	-	-	1	2	2
24AMEC52T.5	3	3	2	2	-	-	-	-	-	-	1	2	2

I.C Engines

Heat Engine:- Engine is a device which converts thermal energy into mechanical energy. (or) mechanical work.

Classification of Heat Engine:-

Heat Engines are classified into two types.

1. External combusion Engine
2. Internal combusion Engine

1) Rotary
2) Reciprocating

1) External Combustion Engines are those in which combustion takes place out side the engine. whereas in internal combustion Engines the combustion takes place within the engine.

eg:- steam Engine, steam power plant for E.C. Engines.

Automobiles, buses, car, Air crafts for I.C Engines.

Advantages of E.C Engines over I.C Engines:-

1. Starting torque is generally high.
2. Because of External combustion of fuel, cheaper fuels can be used.
3. Due to external combustion of fuel it is possible to have flexibulity in the arrangement.
4. These are self starting with the working fluid where as in case of I.C Engines additional equipment is required.

Advantages of I.C Engines over E.C Engines:-

1. overall efficiency is high.
2. Greater mechanical simplicity.
3. weight to power ratio is generally low.
4. low initial cost.

5. Easy starting even at Cold conditions.

6. Units are compact and their required very less space.

Classification of I.C Engines:-

1) According to the cycle of operation.

i) Two-Stroke cycle Engine

ii) Four-Stroke cycle Engine.

2) According to cycle of combustion.

i) Otto cycle Engine (At Const. vol. process)

ii) Diesel cycle Engine (At const. pressure. process)

iii) Dual combusion (or) semidiesel cycle (partially const. vol & const. pressure).

3) According to the arrangement of cylinders.

i) Horizontal Engine.

ii) Vertical Engine.

iii) V-type Engine

iv) Radial type Engines etc.

4) According to their ~~usage~~ uses.

i) Stationary Engine.

ii) portable Engine.

iii) marine Engine

iv) Automobile Engine

v) Aero-Engines etc

5) According to the fuel employed and method of fuel supply to the Engine cylinder.

i) Oil Engine.

ii) petrol (or) Diesel Engine

iii) Gas Engine

iv) Kerosene Engine → Carburettor

v) ~~carburetted~~, Hot bulb, Solid Injection and air Injection type Engines.

6.) According to the speed of the Engine.

i) Low speed Engine.

ii) medium speed Engine.

iii) High speed Engine.

7.) According to the method of Ignition.

i) Spark Ignition.

ii) Compression Ignition.

8.) According to the method of Cooling the cylinder

i) Air cooled Engine.

ii) Liquid cooled Engines (or) water cooled Engine.

9.) According to the method of governing

i) Hit and Miss governed Engine.

ii) Quality governed Engine.

iii) Quantity governed Engine.

10.) According to the valve arrangement.

i) Over head valve Engines.

ii) L-head type Engines.

iii) T-head type Engines

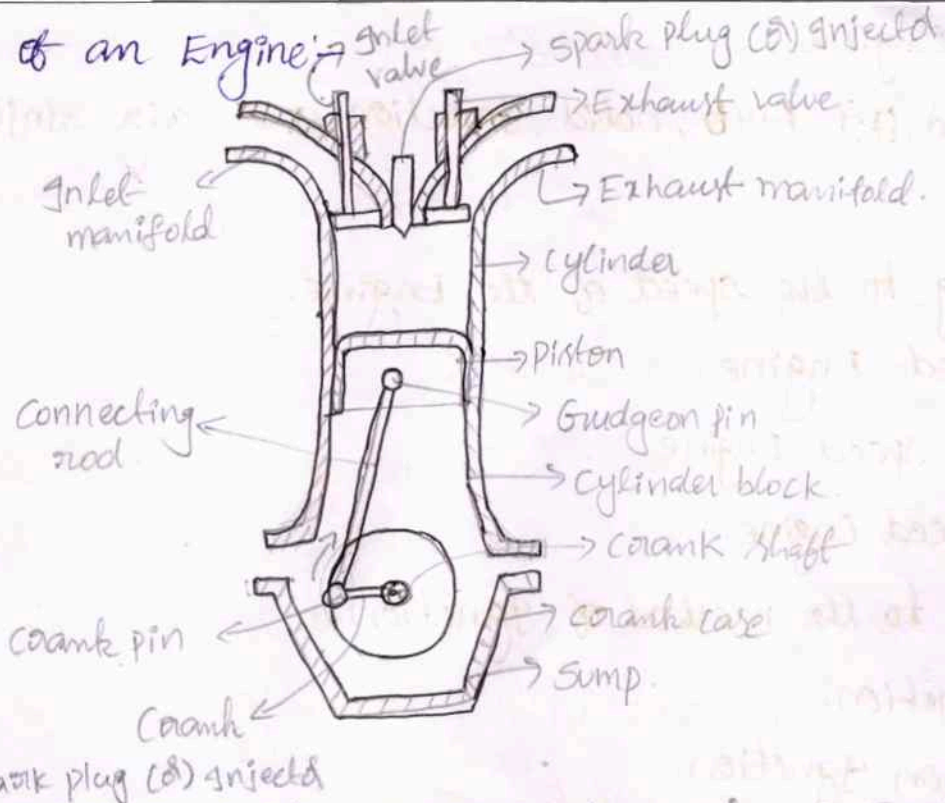
iv) F-head type Engines.

11. According to the number of cylinders. (Reciprocating or Rotary)

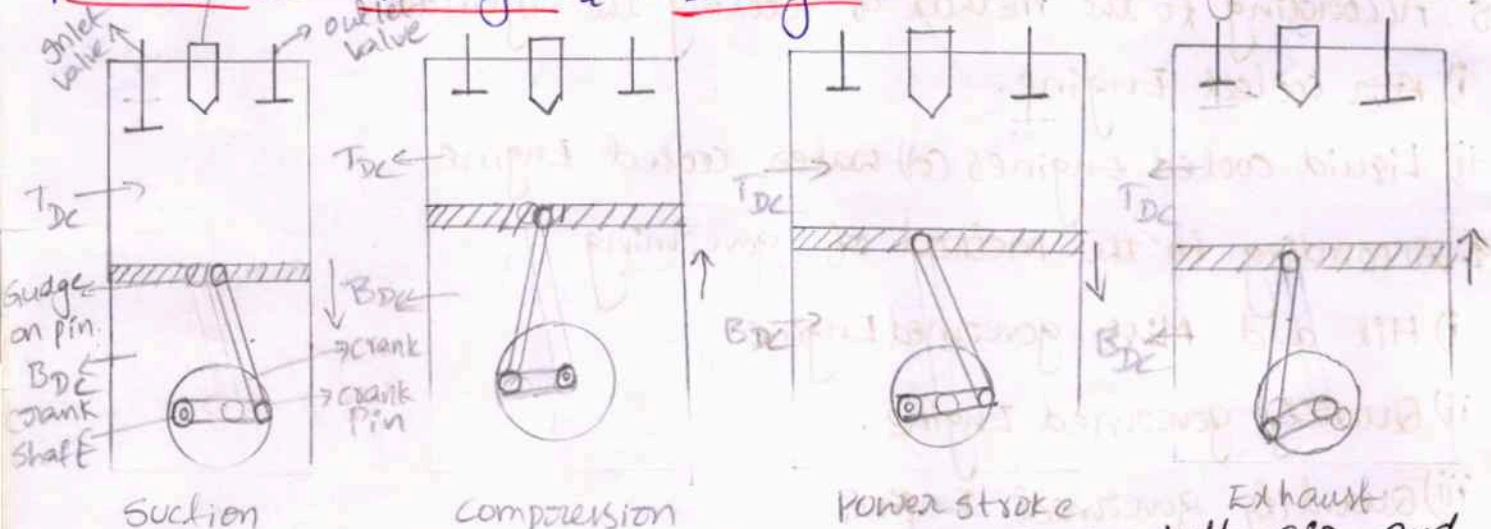
i) Single cylinder.

ii) multi cylinder (V-type, Inclined, opposed cylinders)

Components of an Engine:-



Four stroke S.I Engine & C.I Engine:- For S.I Engines:-



Suction process:- In the suction of S.I Engine both air and fuel gets mixed in the carburettor itself and the air fuel mixture get into the cylinder when the piston is moving from Top Dead centre (TDC) to Bottom Dead centre (BDC) where as in C.I Engine only air get into the cylinder.

Compression stroke (or) process:- In compression stroke both Inlet and exhaust valves gets closed and S.I Engines air fuel mixture is compressed when the piston is moving from BDC to TDC. where as in C.I Engine only air gets compressed.

power stroke:- In S.I Engines the compressed air fuel mixture is burned with the help of spark given by spark plug. where as in C.I Engines fuel is injected into the compressed air with the help of fuel injector and then combusion takes place. by that piston moves from T_{DC} to B_{DC}

Exhaust stroke:- In the Exhaust stroke both SI and C.I Engines, the valves (inlet and exhaust) are in closed position the piston moves from B_{DC} to T_{DC} while the piston is moving upword the burn gases in the combusion chamber escapes from the exhaust valve.

Two Stroke Gasoline Engines:-

→ low cost, simplicity & higher fuel consumption

→ 100 cc - 150 cc develops BP of about 5 KW at 5500 rpm

→ 250 cc " " " 10 KW at 5000 rpm.

2-Stroke Diesel engines:-

→ Very high power, used for ship propulsion

→ 400 - 900 mm bore (develops BP upto 37000 kW on single crank)

→ 12 cylinder 800 mm bore & 1550 mm stroke develops 20000 kW at 120 rpm

→ This speed allows the engine directly coupled to the propeller of ship without gear reducers.

4-Stroke Gasoline engines:-

→ BP of 30 - 60 KW at 4500 rpm

→ 6 - 8 cylinder → 185 KW

→ Buses & trucks 4000 cc, 6 cylinder → BP of 90 KW

→ Air crafts → 400 KW to 4000 KW

4-Stroke diesel engines:-

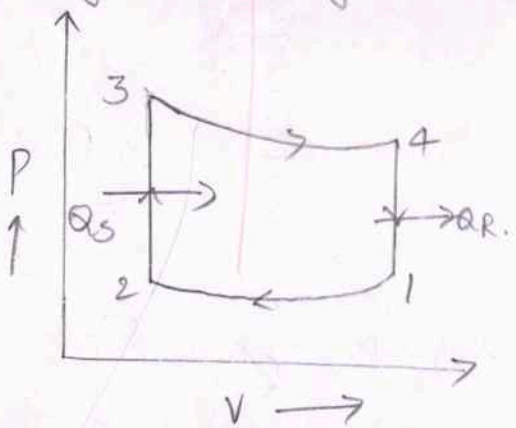
→ 50 mm - 1000 mm dia & speed 100 to 4500 rpm & BP of 1 to 35000 kW

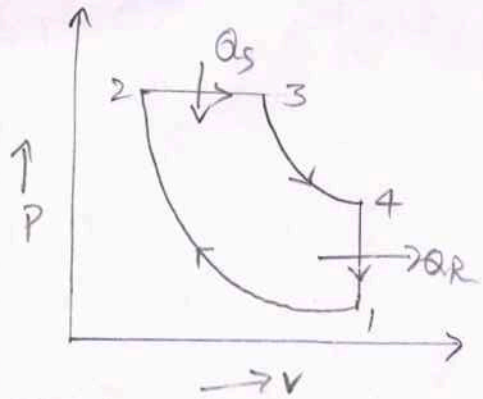
Comparision of S.I and C.I Engine.

Description.	S.I Engine.	C.I Engine.
1. Basic cycle.	Otto cycle (or) constant vol. cycle. (or) constant vol. heat addition process.	Diesel cycle (or) constant press process.
2. Fuel.	Gasoline, highly volatile (easily evaporate) fuel & self ignition temperature is very high.	Diesel oil, a non-volatile fuel and self ignition temperature is comparitively less (or) low.
3. Introduction of fuel.	A gaseous mixture of fuel & air is introduced during the suction stroke. The carburator and an ignition system are necessary.	Fuel is injected directly into the combustion chamber at high pressure at the end of compression stroke. a fuel pump and an injector are necessary.
4. Load control	Throttle controles the Quantity of fuel air mixture to control the load.	The quantity of fuel is regulated to control the Load. Air Quantity is not controlled.
5. Ignition.	Requires an ignition (or) system with spark plug in the combustion chamber primary voltage is provided either by battery (or) a magneto.	self ignetion occurs due to high temperature of air because of high compression ignetion system and spark ~~plug~~ plug are not necessary.
6. Compression Ratio.	6 to 10. upper limit is fixed by antiknock	16-20. upper limit is limited by weight

	Quality of the fuel.	Increase of the Engine.
7. Speed.	Due to light weight and also due to homogeneous combustion, they are high speed Engines.	→ Due to heavy weight and also due to heterogeneous combustion they are low speed Engines.
8. Thermal efficiency	Because of lower compression Ratio the maximum value of thermal efficiency that can be obtained is lower.	→ Because of higher compression ratio the maximum value of thermal efficiency that can be obtained is higher.
9. weight.	lighter. Due to comparatively lower peak pressures.	→ heavier. due to comparatively high peak pressures.

For ottocycle P-V Diagram;-

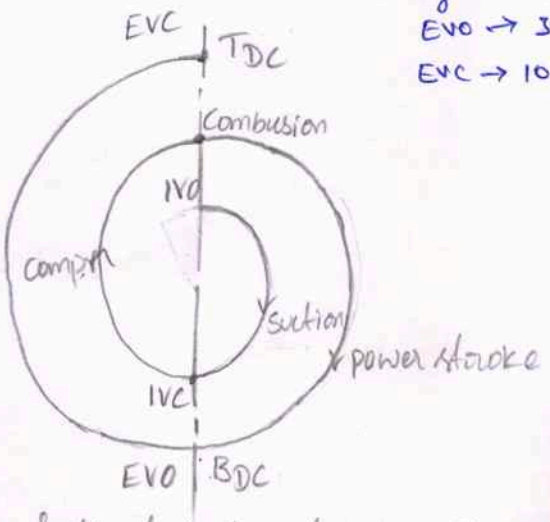




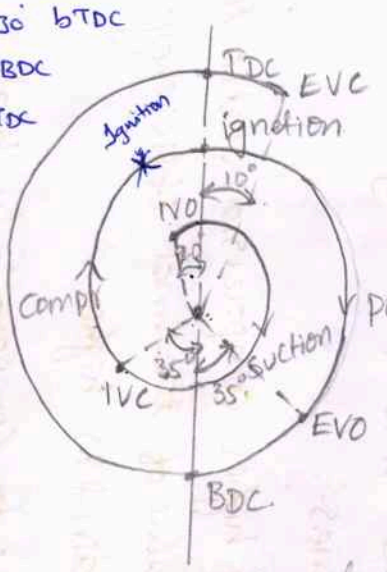
For Diesel cycle P-V Diagram.

Four Stroke S.I Engine :-

IVO → 10-20 b TDC
IVC → 30-40 a BDC
Ignition → 20-30 b TDC
EVO → 30-50 b BDC
EVC → 10-15 a TDC



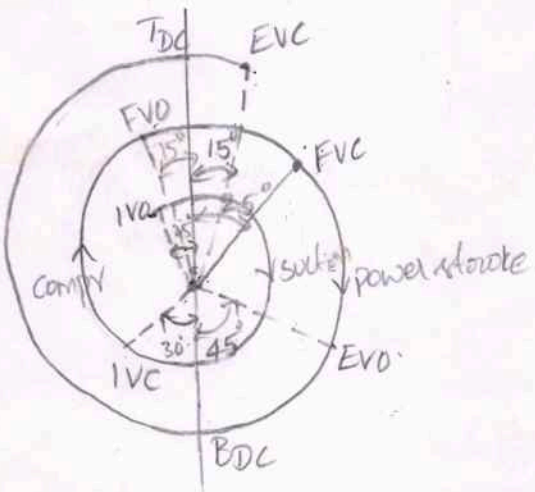
Theoritical Valve timing Diagram for S.I Engine



Experimental V.T.D fd S.I Engine

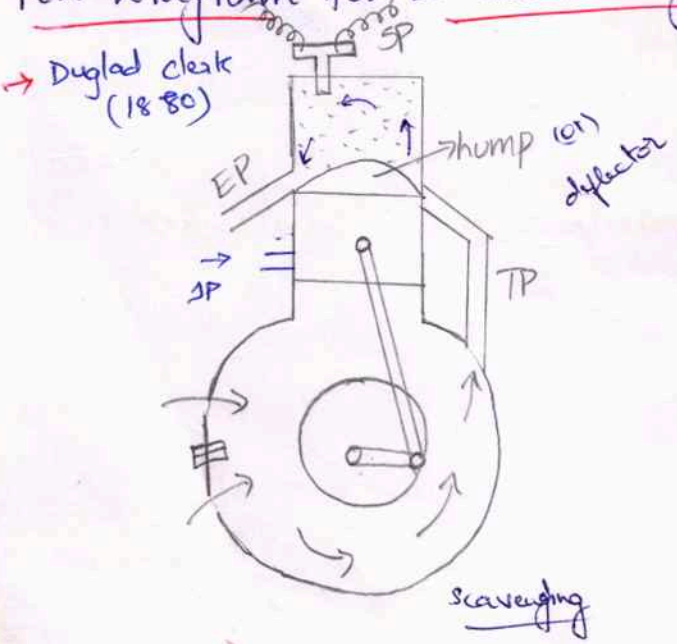
Actual Valve timing Diagram fd S.I Engine :-

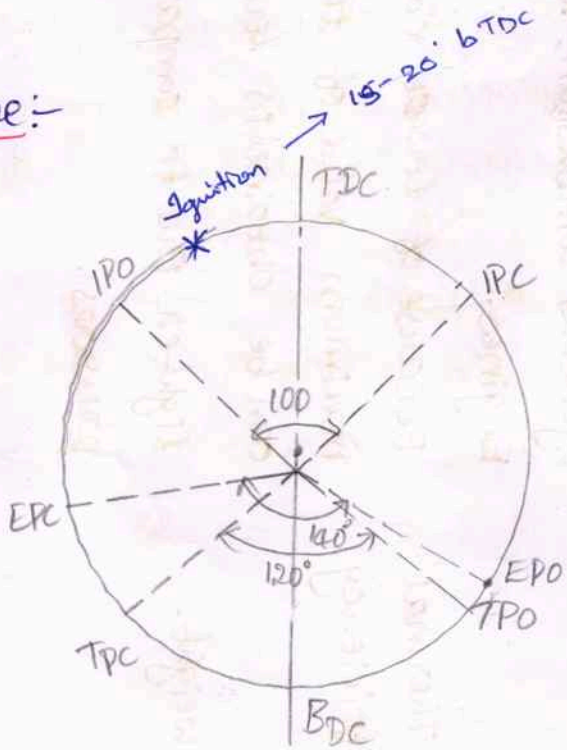
IVO → 10-20° b TDC
IVC → 25-40° a BDC
FVO → 10-15° b TDC
FVC → 15-20° a TDC
EVO → 39-50 b BDC
EVC → 10-15 a TDC



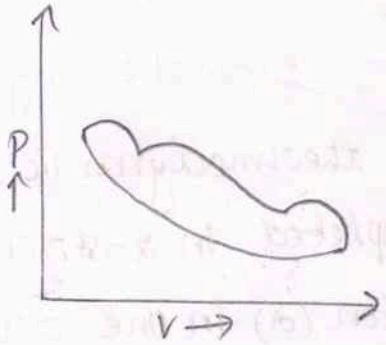
Port Diagram fd 2-Stroke Engine :-

→ Duglad clerk (1880)





Actual P-V diagram of 2-Stroke Engine :-



Nomeclature :-

1) Cylinder boore → d

2) Piston Area → A (Sq.cm)

3) Stroke → L (mm & cm)

(Distance b/w Dead centers is called stroke)

4) Stroke to boore ration → $\frac{l}{d}$ ratio.

$d > l$ it is called over square Engine.

$d < l$ it is under square Engine.

5) Dead centers → TDC & BDC.

6) clearence volume → V_c (If the volume of combusion chamber above the chamber when piston is at TDC is called clearence volume) (cm^3)

~~Cubic capacity~~

7) Displacement (or) swept volume → The volume swept by the piston when it is travelling from one Dead center to another is called Displacement volume → V_s. → $\frac{\pi d^2}{4} \times L$. (~~c.c~~).

8) Cubic capacity (or) Engine capacity :-

The displacement volume of cylinder multiply by no. of cylinders in an engine is called cubic capacity.

Cubic capacity $= V_s \times K$.

9) Compression Ration :- (or)

It is the ration of total cylinder volume when the piston is at BDC to the clearence volume.

$$r = \frac{V_s + V_c}{V_c} = 1 + \frac{V_s}{V_c}$$

Difference b/w 2-Stroke and 4-Stroke Engine.

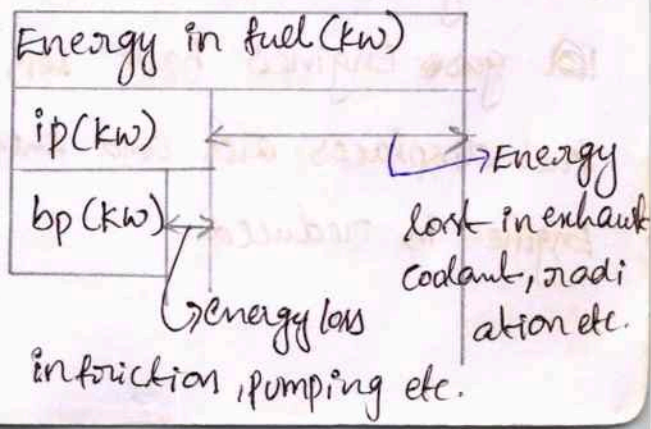
2-Stroke Engine	4-Stroke Engine.
1. The thermodynamic cycle is completed in 2-strokes of the piston (or) in one revolution of the crankshaft. Thus it is having one power stroke for one revolution of crank shaft.	1. The thermodynamic cycle is completed in 4-strokes of the piston (or) in two revolutions of the crankshaft. Thus it is having one power stroke for every two revolutions.
2. Because of the above turning moment is more uniform and hence a lighter fly wheel can be used.	2. Because of the above turning moment is not so uniform and hence a heavier fly wheel is needed.
3. Because of one power stroke for every revolution power produced for same size of the Engine is twice (or) for the same power the engine is lighter and more compact.	3. Because of one power stroke for two revolutions power produced for same size of Engine is less; (or) for the same power the Engine is heavier.
4. Higher rate of wear and Tear	4. Lower rate of wear and tear.
5. Greater cooling and lubrication is required.	5. Lesser cooling and lubrication is sufficient.
6. No valves are used, only ports are arranged.	6. Valves are arranged.
7. Because of absence of Valve actuating mechanism the Engine initial cost of the Engine is less.	7. Comparitively high initial cost.

8. Lower volumetric efficiency due to lesser time for mixture intake.	8. Higher volumetric efficiency due to higher time for mixture intake.
9. Thermal efficiency is low.	9. Thermal efficiency is high.
10. Part load efficiency is poor.	10. Part load efficiency is better.
11. Applications:- scotter, motor cycles, hand sprayers etc.	11. Applications:- car, bus, ~~are~~ aeroplanes etc.
12. A hump is provided at the top of the piston for escaping exhaust gases.	12. No such arrangement is provided.

Engine performance parameters:-

Indicated thermal efficiency (IT η_{ith})
Brake thermal efficiency (η_{bth})
Mechanical efficiency (η_m)
Volumetric efficiency (η_v)
Relative efficiency (or) efficiency Ratio (η_{rel})
Mean effective pressure (m.e.p (or) P_m)
Mean piston speed (S_p)
Specific fuel consumption (Sfc)
Fuel air (or) Air fuel ratio (A/F (or) F/A)
Calorific value of the fuel (CV)

Power entered into the engine cylinder is ip.
Power available at the end of crankshaft is Bp



1) Indicated Thermal efficiency :- It is the ratio of ~~energy in the~~ indicated power to the fuel ~~co~~ energy consuming.

$$\eta_{ith} = \frac{ip}{\text{fuel consumed}}$$

$$= \frac{ip}{\text{mass of the fuel/sec} \times \text{caldific value}}$$

2) Brake thermal efficiency :- It is the ratio of energy in brake power to the fuel energy consuming.

$$\eta_{bth} = \frac{bp}{\text{fuel consumed}}$$

$$= \frac{bp}{\text{mass of the fuel/s} \times \text{caldific value}}$$

3) Mechanical efficiency :-

It is defined as the ratio of breake power to indicated power.

$$\eta_m = \frac{bp}{ip} \quad (\text{or}) \quad \eta_m = \frac{\eta_{bth}}{\eta_{ith}}$$

$ip = bp + fp$

$fp = ip - bp$.

[fp = frictional power]

4) Volumetric efficiency :- volumeteric efficiency is defined as the ratio of actual volume flow rate of air into the intake system to the rate at which the volume is displaced by the system.

$$\eta_v = \frac{ma}{\text{ba}\,v_d}.$$

volumetric efficiency indicates the breathing ability of the engine. The normal range of volumetric efficiency at full throttling for S.I Engine is b/w 80-85%. whereas in C.I Engines 85-90%.. gas Engines have less volumetric efficiency since gaseous fuel displaces air and therefore the breathing capacity of the Engine is reduced.

5) Relative efficiency (or) Efficiency Ratio :- It is defined as the ratio of thermal efficiency of an actual cycle to that of an ideal cycle. The efficiency Ratio is verymuch useful criterian which indicates the degree of development of Engine.

$$\eta_{Rel} = \frac{\text{Actual } \eta_{th}}{\text{Air-standard } \eta_{th}}$$

6) Mean effective pressure :- It is the average pressure inside the cylinder of an I.C Engine, based on measured power output. For any particular Engine operating at given speed and power output there will be a specific Indicated mean effective pressure (IMEP) and a corresponding Brake mean effective pressure (BMEP).

$$i_p = \frac{P_{im} LANK}{60 \times 1000}$$

P_{imep} = which is thought to be acting on the piston throughout the power stroke.

where P_{im} → Indicated mean effective pressure in N/m^2.

L → length of the stroke in (~~mm~~) m

A → Area of the piston in m^2

N → Speed in RPM. ($N = N$ for 2-stroke

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

K → Number of cylinders.

It can also be defined as the ratio of area of the Indicated diagram to the length of the Indicated diagram.

where the length of the Indicated diagram is given by the difference b/w total volume and clearance volume.

$$P_{im} = \frac{\text{Area of Indicated Diagram (ID)}}{\text{length of ID}}$$

$$P_{im} = \frac{60000 i_p}{LANK}$$

7) Mean piston speed :- $\bar{S}_p = 2LN$.

8) Specific fuel consumption :- It is defined as the ratio of the fuel consumption per unit time to the unit power

$$\therefore SFC = \frac{\text{fuel consumed/sec}}{\text{Power}}$$

It is an important parameter that reflects the performance of the Engine. It is inversly proposional to thermal efficiency of the engine.

9) Fuel air (or) Air fuel ratio :- ~~It is~~ the ratio of actual fuel air ratio by ~~stachiomet~~ stoichiometric fuel ~~air~~ ratio. It is ~~denoted by ϕ~~ known as equalence air fuel ratio. (15:1)

10) calorific value :- It is the thermal energy released per unit Quantity of fuel.

Fuel Injection System In S.I Engines :- (or) Fuel Induction system in S.I engine :-

Carburation :- The procey of formation of combustible fuel air mixture by mixing the proper amount of fuel with air before admission to the engine cylinder is known as carburation.

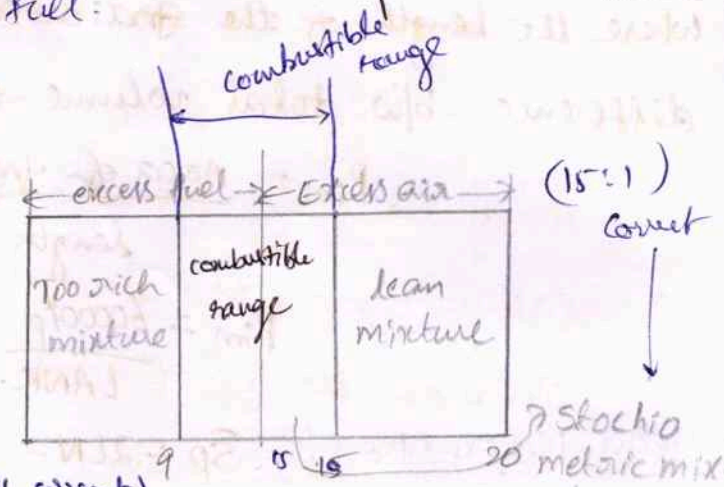
The device used for this process is known as carburettor.

Factors affecting the ~~carburetor~~ :-

1. Engine speed and load on the engine.
2. vaporization characteristics of fuel.
3. Temperature of incoming air
4. Design of the carburettor.

Carburation :-

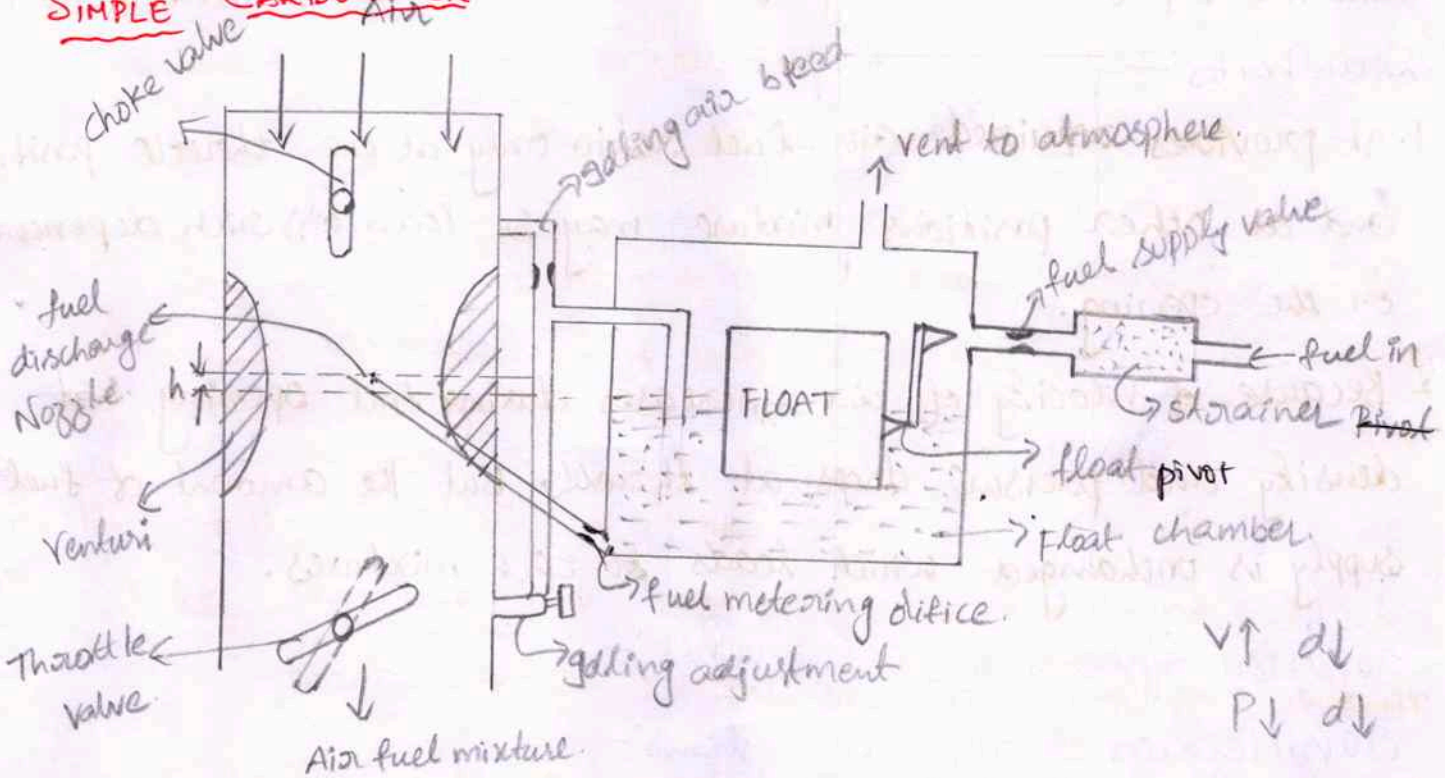
example
octane C_8H_{18}
15.12 : 1 (air)
15 : 1 → Stochio metric mixture



Air fuel mixtures :-

1. Rich mixture
2. poor d lean mixture
3. Stoichiometric mixture - (chemically correct)

SIMPLE CARBURETOR



Simple carburetor mainly consists of a float chamber, fuel discharge Nozzle and metering orifice, a venturi, a throttle valve and choke.

Float and needle valve system maintains a constant level of gasoline in the float chamber. Float chamber is vented to the atmosphere. During suction stroke air is drawn through the venturi. It is a tube of decreasing cross-section with a minimum area at the throat. it is also known as choke tube. As the air passes through the venturi the velocity increases and pressure drops. Because of differential pressure b/w the float chamber and the throat, known as carburetor depression, fuel is discharged into the air stream. The pressure at the throat at the fully opened throttle condition lies b/w 4-5 cm of mercury (cHg), below atmospheric pressure.

To avoid the over flow of fuel through the jet the level of the liquid in the float chamber is maintained at a level slight

below the tip of the discharge jet. The difference is marked as h.

Drawbacks:-

1. It provides required air-fuel ratio only at one throttle position and at other positions mixture may be lean (or) rich depending on the opening.
2. Because of velocity of air increases during full opening, density and pressure drops at throat but the amount of fuel supply is unchanged which leads to rich mixtures.

Injection systems in C.I Engines:-

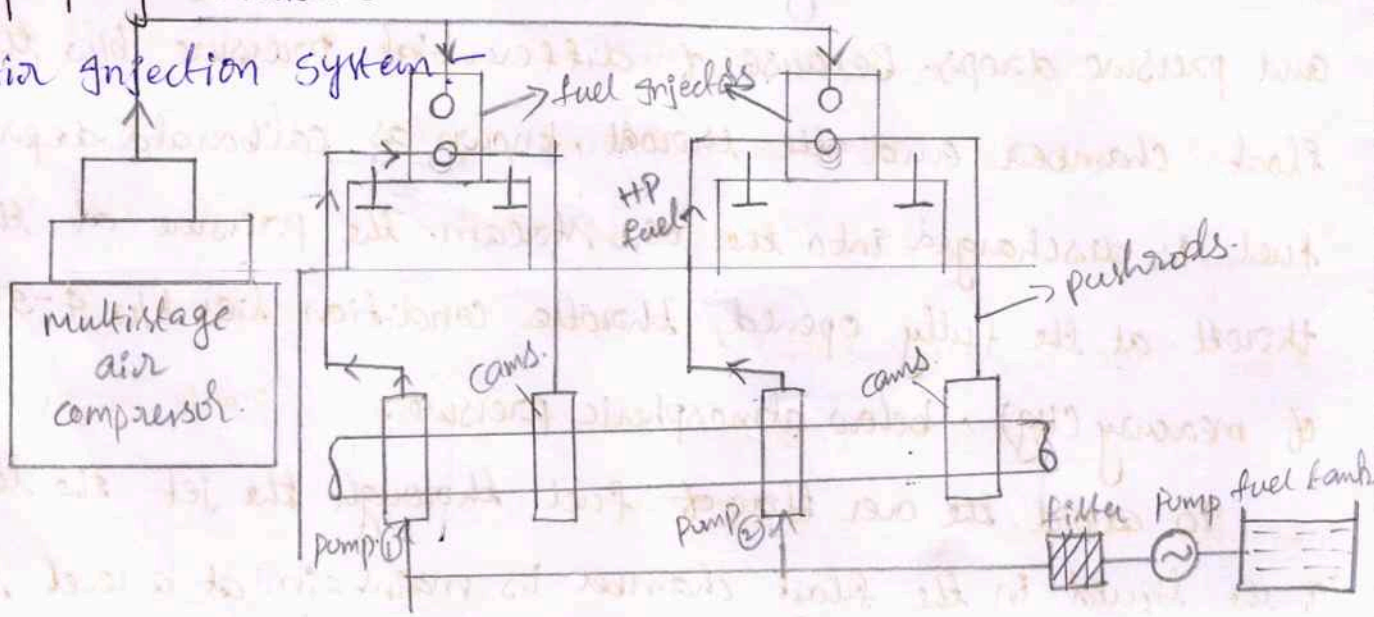
classification of injection systems:-

1. Air injection system.
2. Airless injection system (or) solid injection system.

Injection systems are designed to fulfill following requirements.

1. To meter or measure the correct quantity of the fuel to be injected.
2. To atomize the fuel into fine quantities.
3. Time the fuel injection.
4. Control the rate of fuel injection.
5. properly distribute the fuel in combustion chamber.

i) Air injection system:-



In Air injection system high pressure air is produced with the help of multistage compressor. The fuel is supplied to the fuel valve by means of camshaft. The fuel valve is opened by means of mechanical linkages operated by camshaft. The fuel valve is also connected to the high pressure line. (80 bar). When the valve is opened the air pushes the fuel thorough the fuel injector. the well automized fuel is sep. supplied to the combustion chamber.

Advantages:-

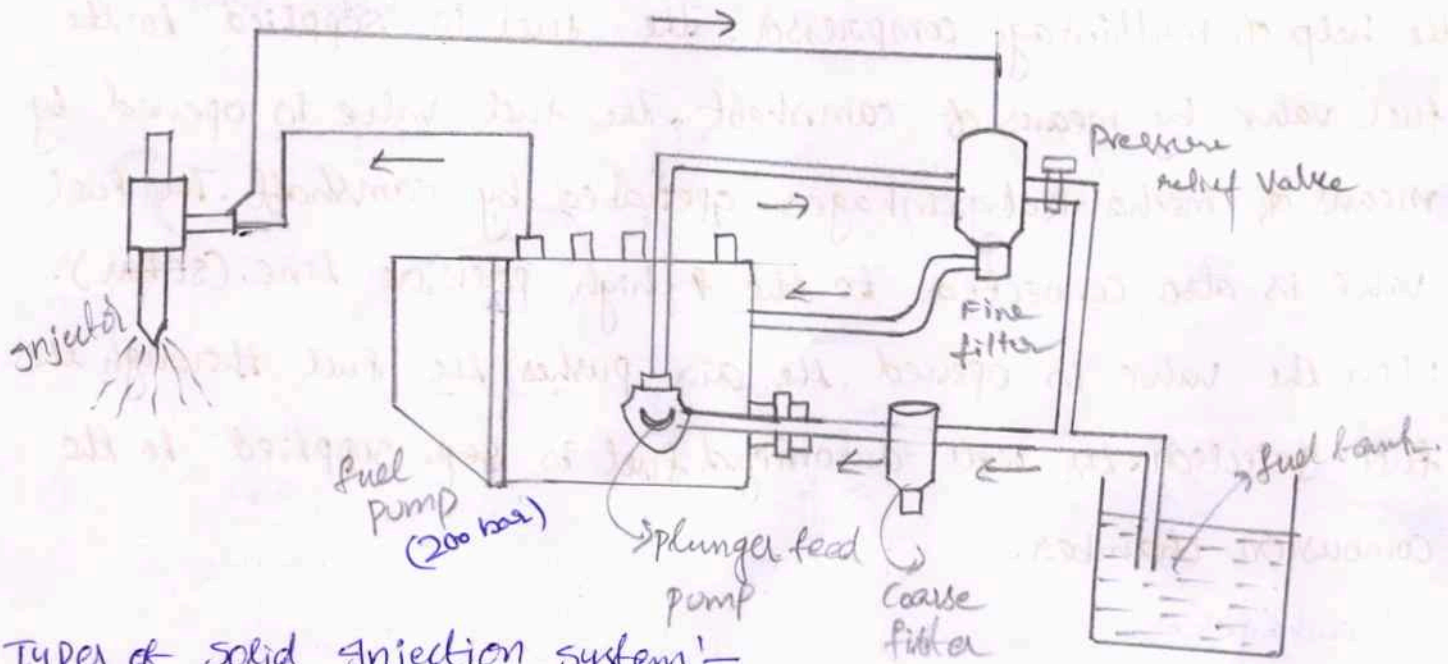
1. simple in working.
2. Good mixing of air and fuel.
3. heavy viscous fuels can also be injected.

Disadvantages:-

1. It requires multistage compressor. which consumes some amount of engine power.
2. weight and size of the system increases.
3. It totally depends on mechanical linkages.

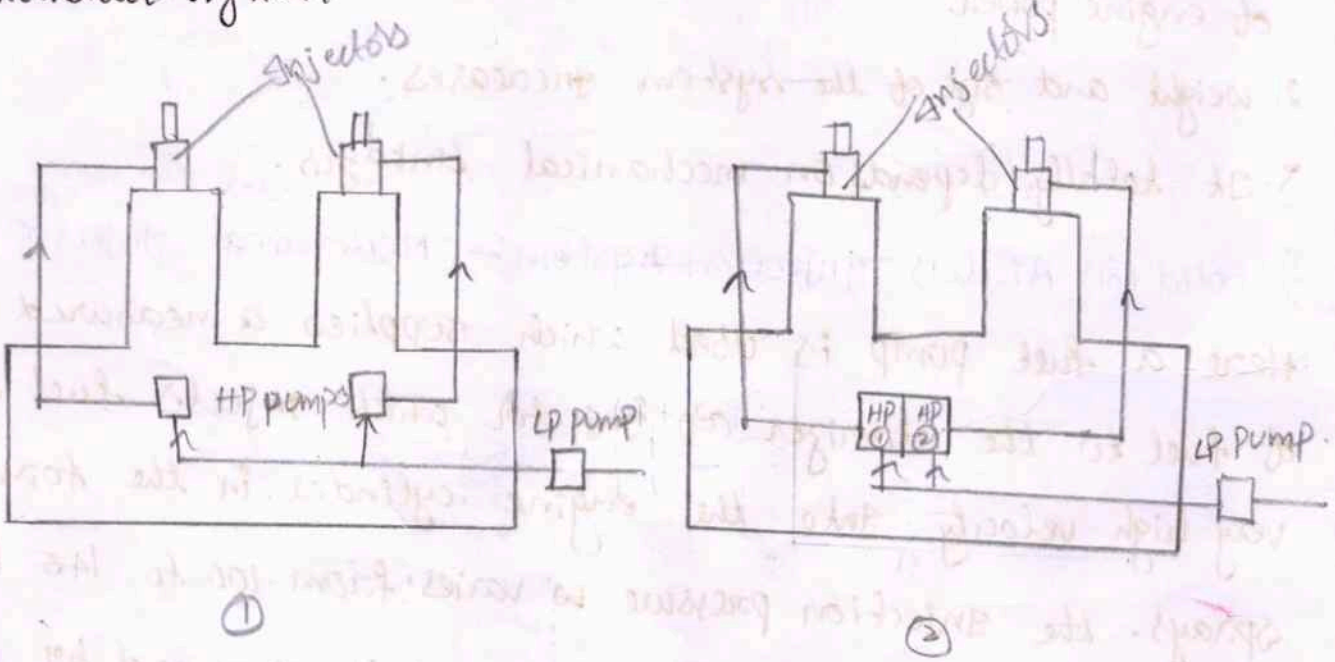
2) Solid (or) Airless injection system:- (Mechanical injectionsystem)

Here a fuel pump is used which supplies a measured Quantity of fuel to the atomizer (or) injector which injects fuel at a very high velocity into the engine cylinder. in the form of sprays. the injection pressure varies from 100 to 145 bar (even more in some cases) This pressure is produced by fuel pump. fuel injection system consists of i) fuel pump ii) fuel injector (or) atomizer and iii) filters.

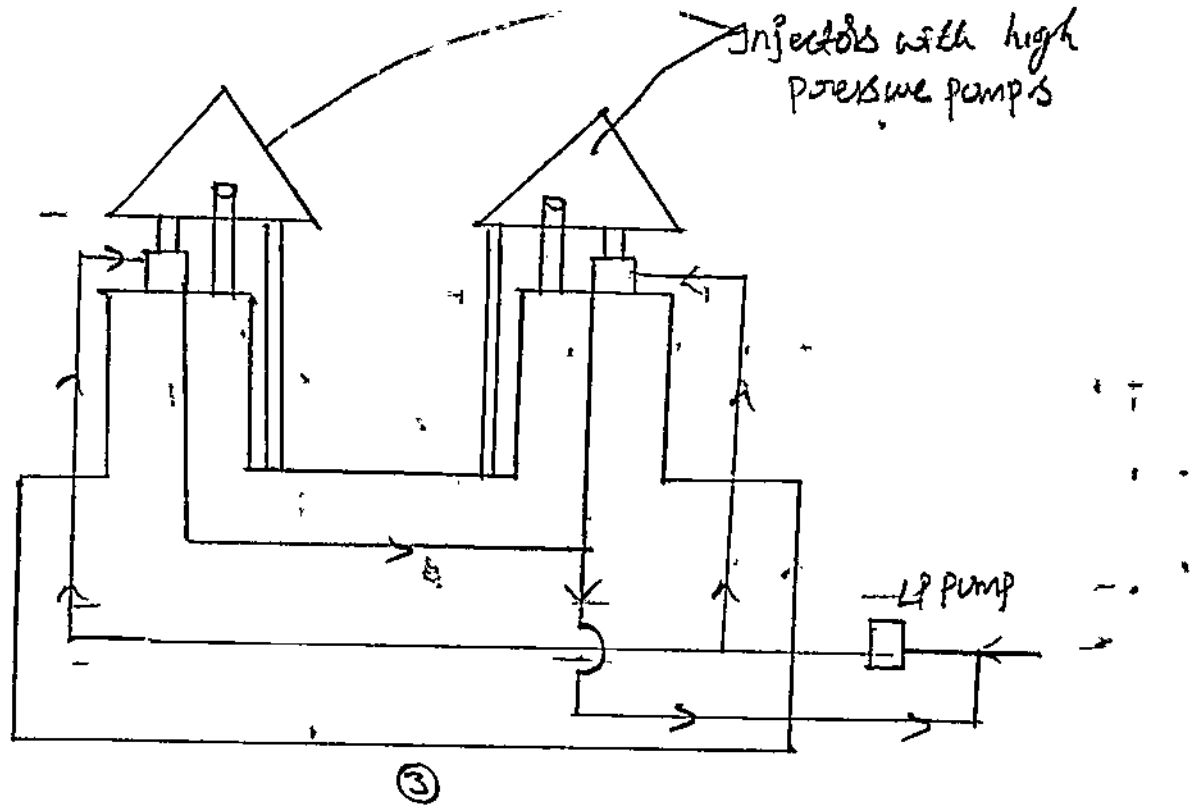


Types of solid injection system:-

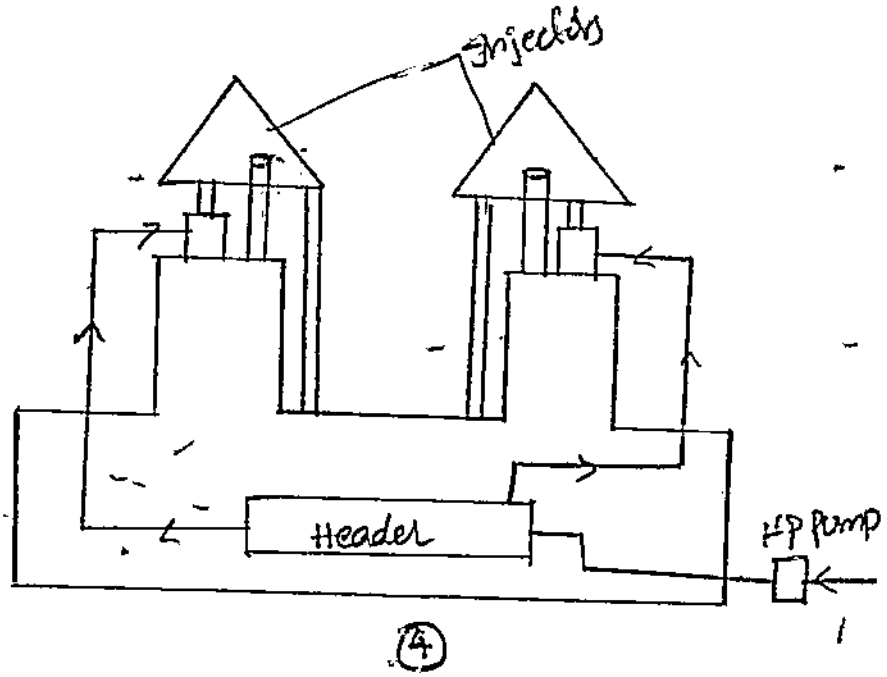
1. Individual pump and Nozzel system with separate pumps.
2. Individual pump and nozzel system with cluster.
3. Unit injector system.
4. common rail system.
5. distributor system.



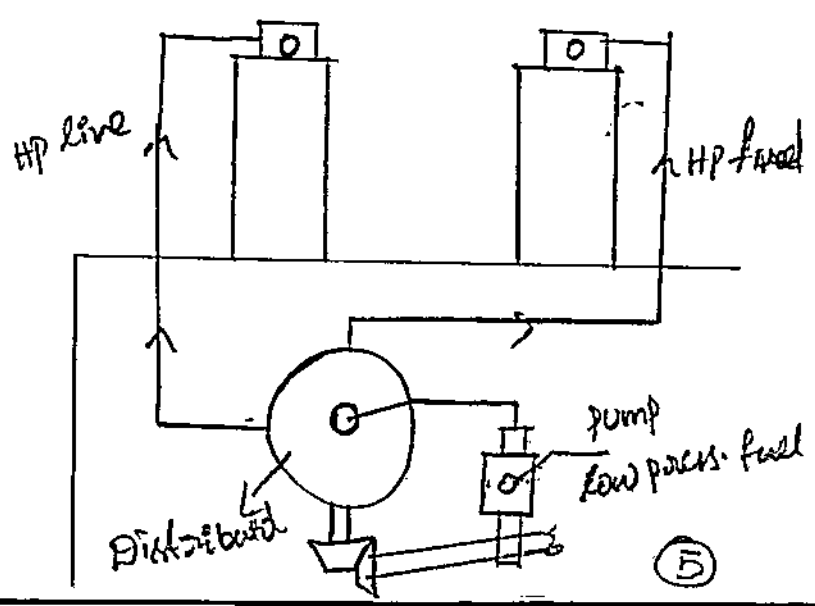
① ②


Injectors with high
pressure pumps
LP pump

③


Injectors
Header
HP pump

④


HP line
HP fuel
pump
low press. fuel
Distributor

⑤

Ignition system:-

Requirements of Ignition System:-

1. A source of electrical energy must be there.
2. A Means for stepping the voltage from the source to the very high potential required to produce a high tension arc across the spark plug gap.
3. A means for timing and distributing high-voltage.
4. Adjustment of the spark advanced with speed and load.

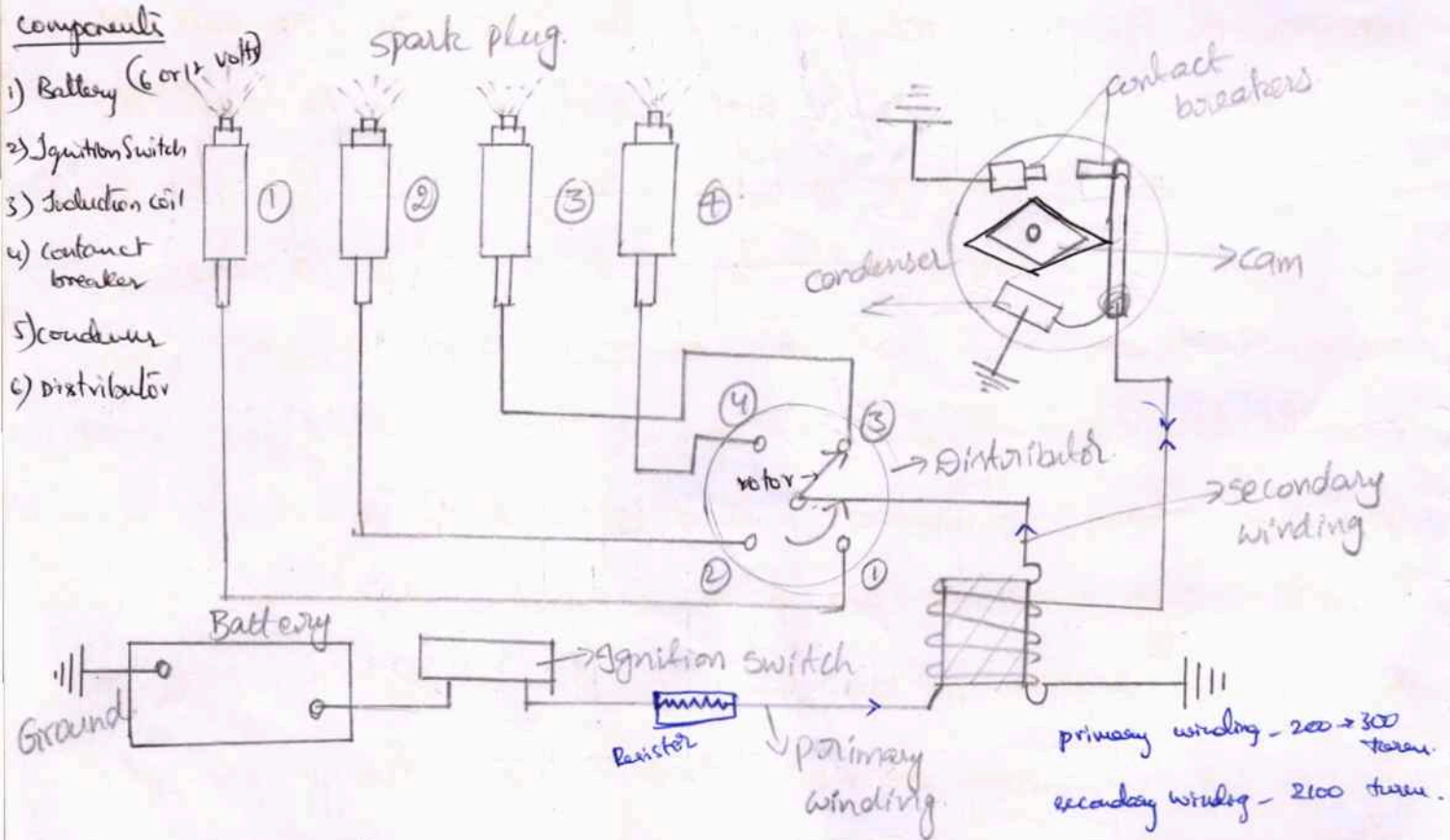
Types of Ignition system:-

i) Battery Ignition system.

ii) Magneto Ignition system.

(1) Battery Ignition system:-

Most of the modern S.I Engines use this Battery Ignition system.



One terminal of the battery is grounded to the frame of the engine and other is connected through the ignition switch. The other primary terminal is connected to one end of the contact

points of the circuit breaker and through closed points to the ground. The primary circuit of the ignition coil thus gets completed when contact points of the circuit breaker are together and switch is closed. The secondary terminal of the coil is connected to the central contact of the distributor and hence to the distributed rotor. secondary circuit consists of secondary windings of the coil, distributed and four spark plugs. The contact breaker is driven by a cam whose speed is half the engine speed (for four stroke engine) and breaks the primary circuit once for each cylinder during one complete cycle of engine.

The breaker points are held in contact by a spring except when forced apart by the lobes of the cam. To start, the ignition switch is made 'ON' and the engine is cranked, when the contacts touch, the current flows from battery through the switch, primary winding of induction coil to the circuit breaker points and the circuit is completed through the ground. A condenser connected across the terminals of the contact breaker points prevents the sparking at these points. The rotating cam breaks (or) opens the contacts immediately and breaking of this primary circuit brings about a change of magnetic field; due to which a very high voltage to the tune of 8000 to 12000 volts. is produced across the secondary terminals. Due to high voltage the spark jumps across the gaps in the spark plug and air fuel mixture is ignited in the cylinder.

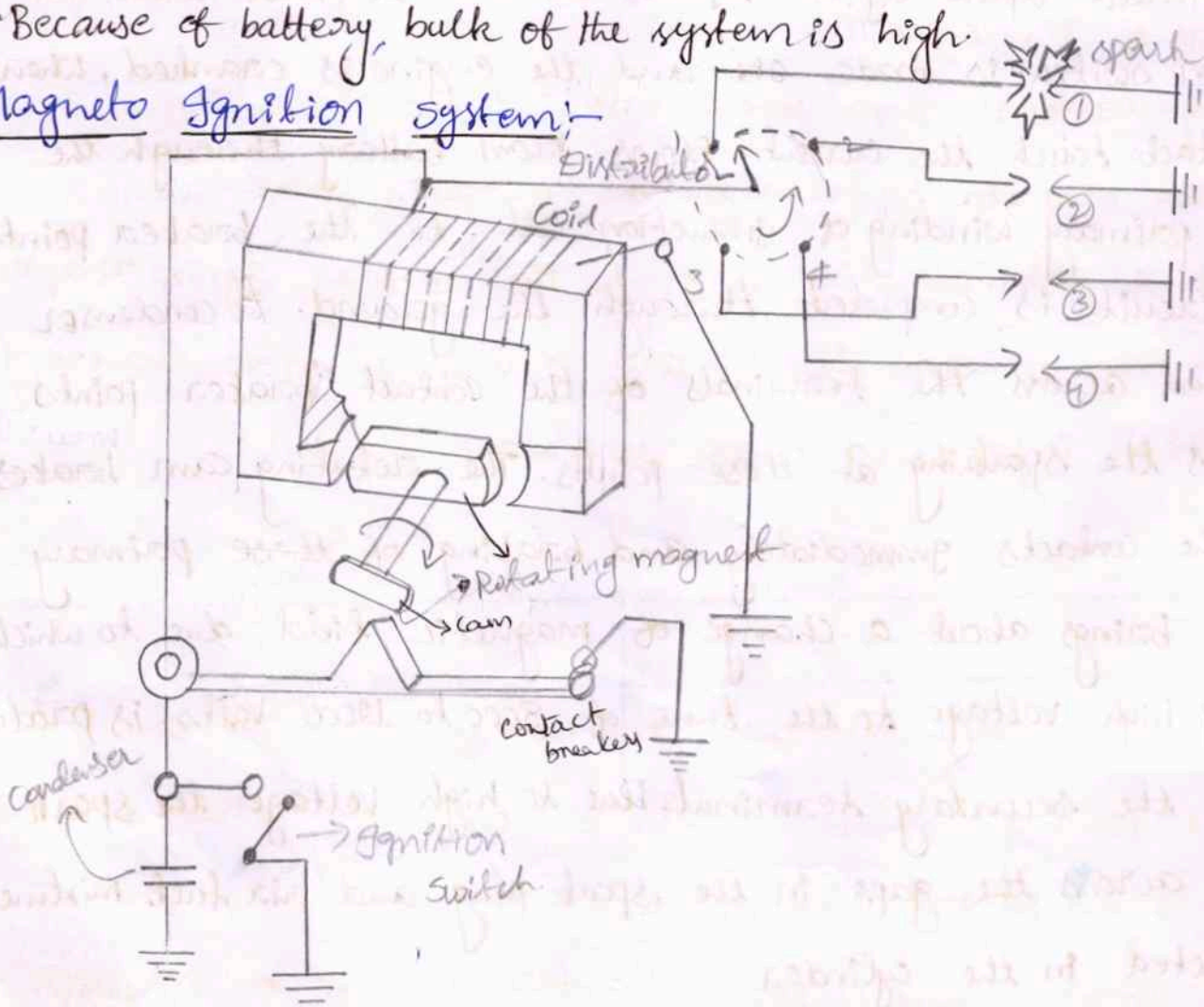
Advantages:-

1. It offers better sparks at low speeds, starting and for cranking purpose. The initial cost of the system is low.
2. It is a reliable system and periodical maintanence is not required except for battery.
3. The high speed engine ~~is dry~~ is usually simpler than magneto drive ~~firing~~ system.

Disadvantages:-

1. With increasing speed sparking voltage drops.
2. Incase battery runs down, the engine cannot be started as induction coil fails to operate.
3. Because of battery, bulk of the system is high.

Magneto Ignition System:-



It is a special type of ignition system with its own electric generator to provide necessary energy for the system. It is mounted on the engine & replaces all the components of coil ignition system except the spark plugs.

A magneto when rotated by the engine is capable of producing a very high voltage and does not need a battery as a source of external energy.

The high tension magneto ignition system incorporate the windings to generate the primary voltage as well as to step up the voltage and thus does not require another separate coil to Boost up the voltage required to operate the spark plugs.

These are of two types.

1. Rotating Armature type magneto ignition system
2. Rotating magneto type magneto ignition system.

The working principle is similar to the battery ignition system. With the help of cam the primary circuit flux is changed & high voltage is produced in the secondary circuit.

Differences b/w Battery ignition system & Magneto I.S:-

Battery	Magneto.
1. Battery is necessary. difficult to start the engine when battery is discharged.	1. No battery is needed. and there is no problem of battery discharge.
2. By Maintainence cost is high.	2. Maintainance cost is comparatively low.

3. current ~~from~~ for primary circuit is obtained from the battery.	3. The required electric current is generated by magneto
4. A good spark is available even at low speeds.	4. During starting Quality of spark is poor due to low speed.
5. Efficiency of the system decreases with the reduction in spark Intensity as engine speed rises.	5. Efficiency of the system improves as the engine speed rises due to high Intensity spark.
6. occupies more space	6. occupies less space.
7. commonly employed in cars, and light comercial vehicals.	7. Mainly used in racing cars and two wheelers.

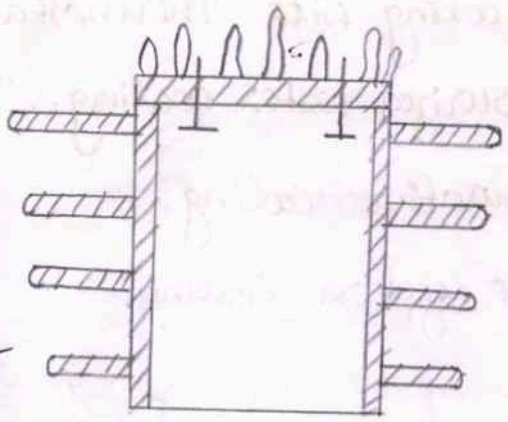
Cooling Systems:- (35-2750 C) temp range
Cooling system is necessary to remove unwanted heat from the cylinder so as to prevent. (cool to save)

i) Burning of lubrication oil & to minimize the wear of parts
ii) Seizure of the piston because of excessive expansion.
iii) over heating of spark plug and cylinder ~~valves~~ walls which lead to preignition.
iv) Excessive stores in bars due to unequal temperatures.
v) An excessive heat removal from the engine reduces engine power and increases the consumtion of fuel. and thus over cooling of the engine is also undesirable.

Types of Cooling systems:-

1. Air cooling.
2. liquid cooling.

i) Air cooling:-



Advantages:-

i) Engine design will be simpler. as no water jackets are required.

ii) Instalation and maintainence cost is very low.

iii) absence of cooling pipes, Radiator etc.

iv) The engine is not ~~fore~~ subjected to freezing troubles.

v) No danger of coolant leakage.

vi) The weight of the engine will be less.

Disadvantages:-

1. The air movement is noisy.

2. Non-uniform cooling.

3. Output of air cooled engine is less than the ~~equipment~~ water cooling.

4. Smaller useful compression ratio.

ii) Liquid cooling:- In this method the cylinder ~~walls~~ ~~valves~~ walls. and heads are provided with water jackets through which the cooling liquid is circulated. The heat is transferred from the cylinder walls to the liquid by conduction and convection. Thus the liquid becomes heated and is itself cooled by means of an air cooled radiator system. From the liquid heat gets transfer to the air with the radiators.

Methods of liquid cooling systems:-

1. Thermosyphon cooling

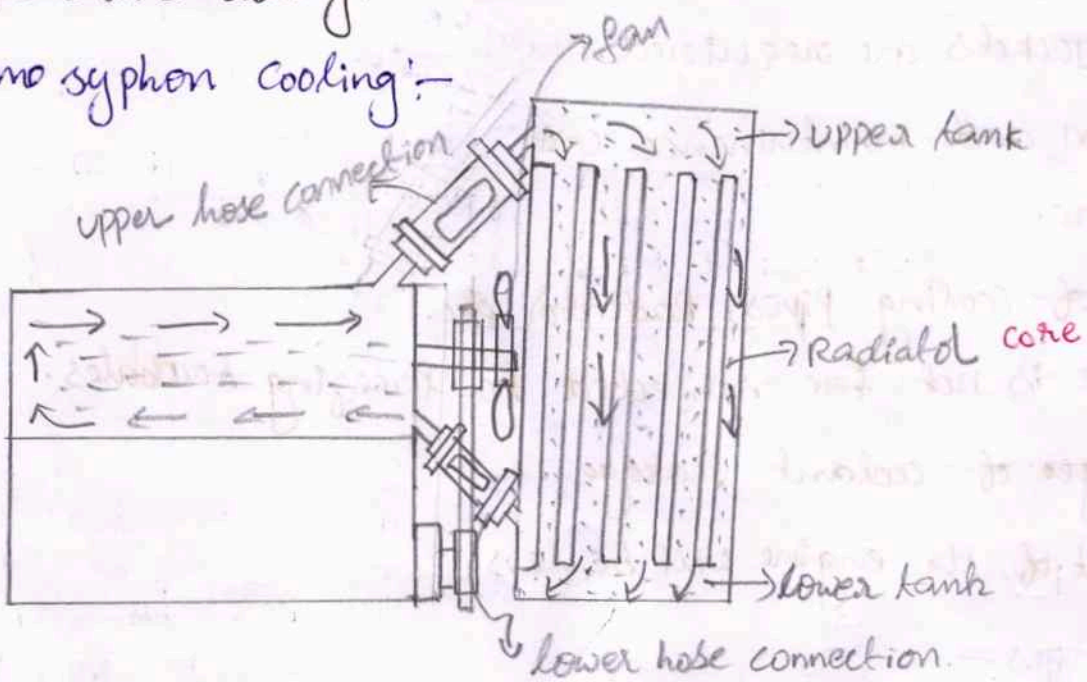
2. Forced (or) pump cooling.

3. ~~The~~ cooling with Thermostatic Regulator.

4. Pressurized water cooling.

5. Evapourative cooling.

6. Thermo syphon cooling:-



Advantages:-

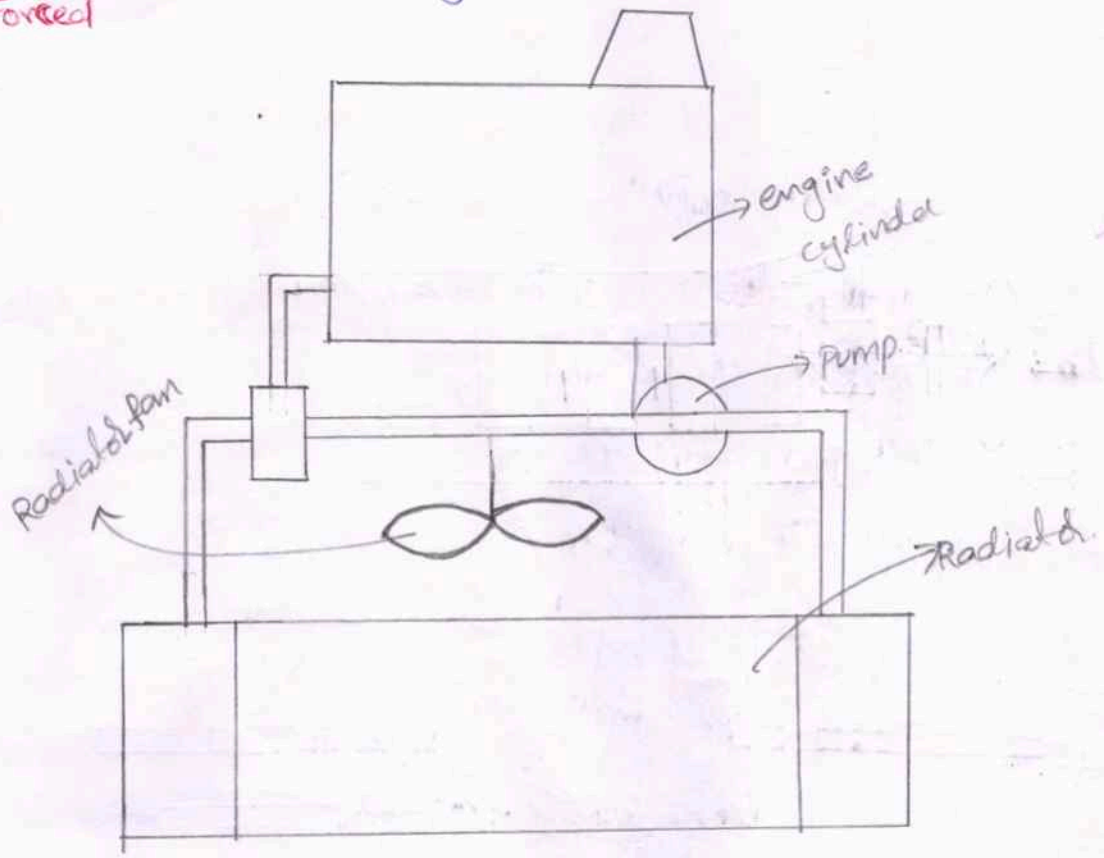
1. It is quite simple and automatic.
2. Working without any pump
3. If there is no leakage there is nothing to get out of order.

Disadvantages:-

1. cooling depends only on the temperature and is independent of engine speed.
2. The rate of circulation is low & in sufficient.
3. Circulation of water starts only after the engine has become hot enough to cause thermo shyphon cooling action.
4. It requires the radiator be above the engine for gravity flow of water to the engine.

ii) ~~Forced~~ (or) pump cooling :-

Forced



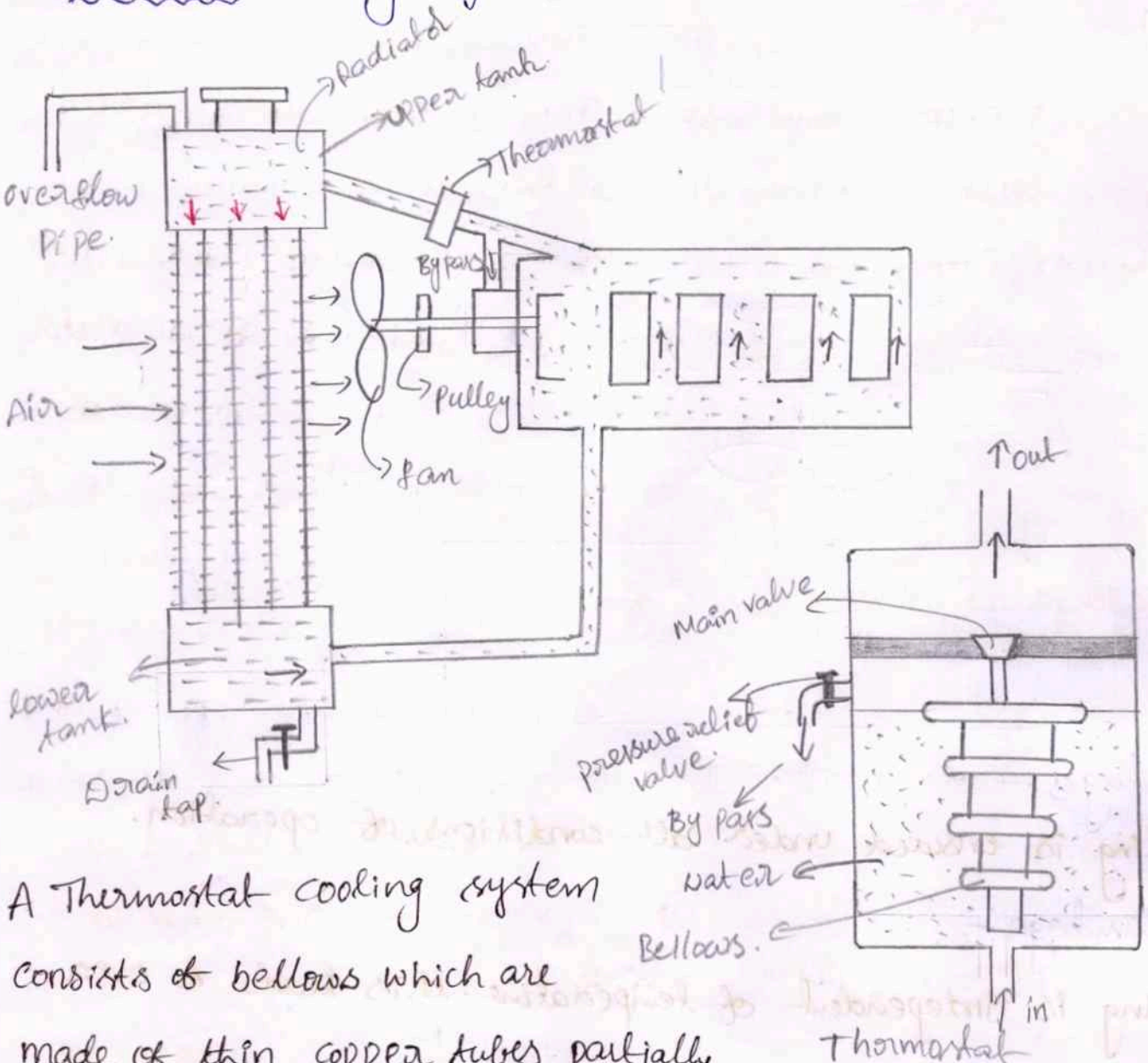
Advantages :-

1. cooling is ensured under all conditions of operation.

Disadvantages :-

1. cooling is independent of temperature. this leads to over cooling of the engine.
2. while moving up hill, the cooling requirement is increased because more fuel is burned. However the coolant circulation is reduced which may resulting in over heating. the engine.
3. As soon as the engine ~~start~~ stopped the cooling also ~~cease~~ Seizer. ceases. This is undesireable because cooling must continue till the temperatures are reduced to normal values.

iii) Thermostat cooling system:-



A Thermostat cooling system consists of bellows which are made of thin copper tubes partially filled with ether (or) methyl alcohol (volatile liquid like). The volatile liquid changes into vapour at the correct working temperature, thus creating enough pressure to expand the Bellows. The temperature at which thermostat operates is set by the manufactures & it cannot be altered. the movement of bellows opens the main valve in the ratio of temp. rise, increasing (or) restricting the flow of water from engine to the radiator. Hence when the normal temp. of the engine has been reached the valve opens and circulation of water comences.

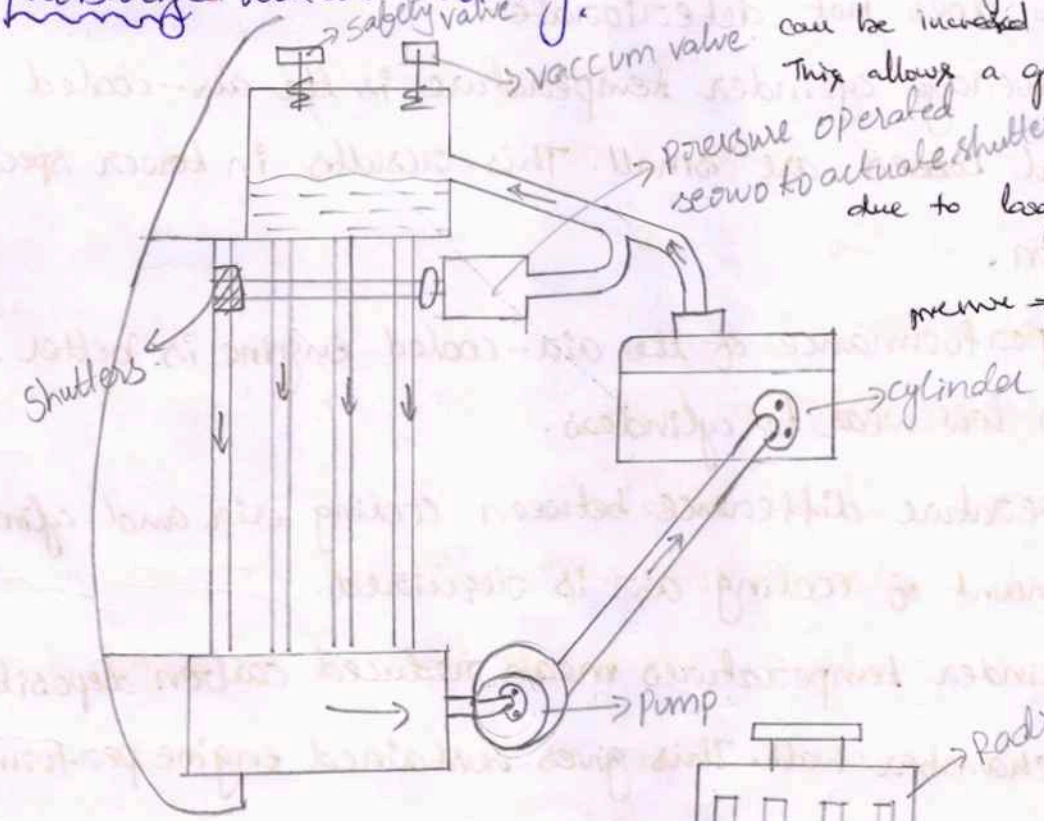
when the thermostat valve is not open and the engine is running, the water being pumped rises in pressure and causes the pressure relief valve to open. Thus the water completes its circulation through the By pass valve.

Another method of warming up the radiator water up to the normal temp. is by utilizing shutter in the radiator in order to restrict the incoming air through the radiator till the engine warms up. There after the shutter is open gradually so that the desired rate of cooling is achieved.

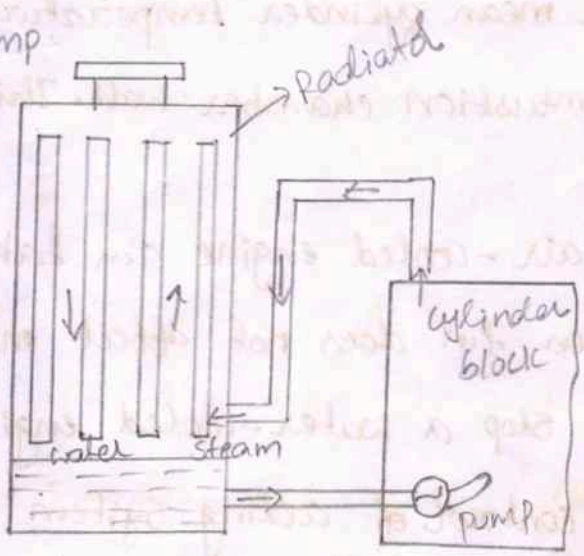
iv) pressurized water cooling:-

The boiling point of water (or) coolant can be increased by increasing its pressure. This allows a greater heat transfer to occur in the radiator due to large temp differential.

pressure → 1.5 to 2 bar	
Press	boiling temp
1 bar	100
2	121
5	153
10	180



pressurized water cooling



Steam (or) Vapour cooling

v) Evaporate cooling

Comparision of Air cooling and water cooling Systems:-

① Air cooling :-

Advantages :-

1. The direct transfer of heat from engine to air eliminates the use of water as a coolant. No water jacket, radiator and water pump are required. This may mean a reduction in weight by as much as 20%. The size of the engine is also small.
2. The engine design becomes much simpler.
3. The air-cooled engine is less sensitive to climatic conditions. No anti-freeze solution is needed. Due to greater temperature difference between the cooling air and cylinder, the cooling is hot weather does not deteriorate.
4. Due to high average cylinder temperature is the air-cooled engine thermal losses are small. This results in lower specific fuel consumption.
5. The warm-up performance of the air-cooled engine is better. This results in low wear to cylinders.
6. Since the temperature difference between cooling air and cylinder is more, less amount of cooling air is required.
7. Higher mean cylinder temperatures mean reduced carbon deposits on combustion chamber wall. This gives sustained engine performances.
8. An air-cooled engine can take up some degree of damage. A broken fin does not affect much while a hole in the radiator may stop a water-cooled engine.
9. The control of cooling system is much easier than in water-cooled engines.

Disadvantages:-

1. Due to the absence of the water passage the combustion noise is not attenuated. Rather, the air fan is an additional source of noise.
2. The volumetric efficiency of an air-cooled engine is lower due to higher cylinder head temperatures.
3. High specific output engines cannot be air-cooled due to the complex nature of the fins that are required.

ii) Water-Cooled Engine :-

Advantages:-

1. High specific output engines pose no problem with water cooling. The heat transfer coefficient of water is about 350 times that of air. This results in compact design.
2. Due to the high latent heat of water the water-cooling system allows greater amount of heat from any local hot spot. This acts as a useful safety valve for overheating troubles.
3. The water cooled engine can be installed anywhere in the vehicle.
4. The volumetric efficiency of water-cooled engines is higher than that of air-cooled engines.

Disadvantages:-

1. The need for a radiator and a pump increases the weight and the dimensions of the engine. Due to the presence of radiator the frontal area of the vehicle is increased resulting in greater air resistance.
2. Water-cooling system requires more maintenance. A slight leakage of the radiator may result in greater air resistance.

2. ~~water cooling system~~ breakdown of the engine.
3. The engine performance becomes more sensitive to climatic conditions. cold weather starting requires use of anti-freeze solutions which may, sometimes, result in deposits on the water side of the cylinder and in reduced heat transfer.
4. The warm-up performance is poor. This results in greater cylinder wear.
5. The power absorbed by the pump is slightly higher than that necessary for air-cooled engines.

(4-65) V Gowtham

Lubrication:-

~~Requis~~ purpose of lubrication:-

1. To reduce friction and wear b/w the parts having relative motion.
2. To cool the surface by carrying away heat generated due to friction
3. To seal the space adjoining the surfaces such as piston rings and cylinder ~~lining~~. liner.
4. To clean the surface by carrying away the carbon & metal particles caused by wear.
5. To absorb shock b/w bearings & other parts, consequently reduce noise

properties of lubrications:-

1. viscosity
2. Flash point (before fire point one flash is occured)
3. Fire point
4. Cloud point (when subjected to low temp oil changes to plastic or solid state).

5. pour point (ability to flow very easy) at low temp
6. oiliness. (spread everywhere & adhere to the surfaces).
7. corrosion. (should not corrode the parts)
8. Emulsification. (Mixing of more liquids becomes somewhat thick) (mixing with water)
9. physical stability.
10. Chemical stability.
11. ~~Ntt~~ Neutralization Number. (impurities not removed during Refining) It is a test to determine acidity or alkalinity of an oil)
12. adhesiveness.
13. Filmstrength.
14. Specific Gravity. (measure of density of oil) hydrometer is used to measure the gravity.

Types of Lubrication System:-

1) Wet Sump lubrication system
2) Dry Sump lubrication system.
3) Mist ~~Sump~~ lubrication system.

(1) Wet sump Lubrication System:-

These systems employ a large capacity oil sump at the base of crank chamber, from which the oil is drawn by a low pressure oil pump and delivered to various parts. oil ~~there~~ gradually returns back to the sump after serving the purpose.

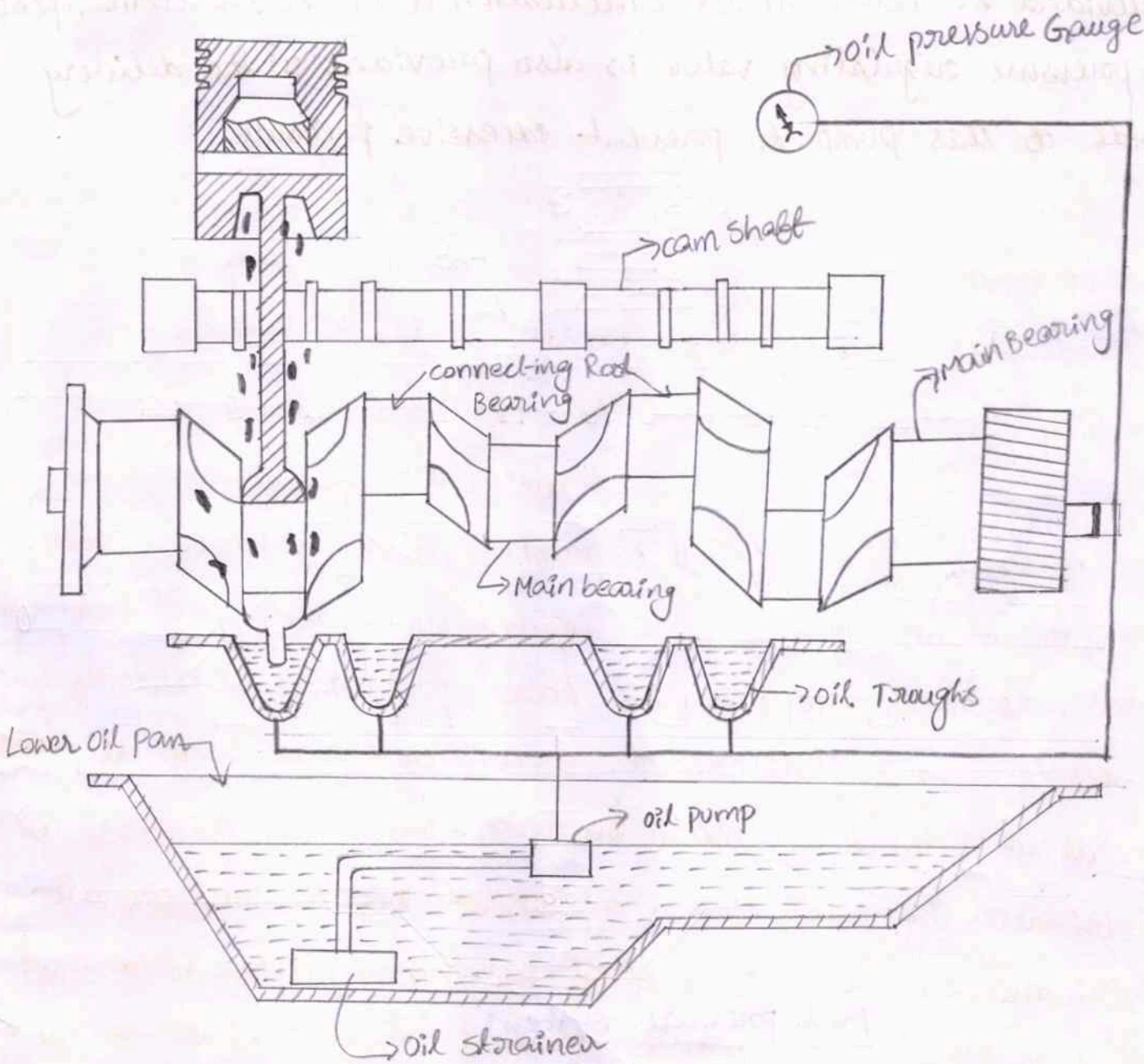
a) Splash System:- This system is used for some small four stroke stationary engines. In this case the caps on the big ends bearings of connecting rods are provided with scoops, which when the connecting rod is in the lowest position, just dip into oil troughs and thus direct the oil through holes in the caps to the big end bearings. Due to splash of oil it reaches the lower portion of the cylinder walls, crank shaft and other parts requiring

lubrication. Surplus oil eventually flows back to the oil sump. Oil level in the troughs is maintained by means of an oil-pump which takes oil from sump, through a filter.

Splash system is suitable for low and medium speed engines having moderate bearing load pressures. For high performance engines, which normally operate at high bearing pressures and rubbing speeds this system does not serve the purpose.

b) Semi-pressure system:- This method is a combination of splash and pressure systems. It incorporates the advantages of both. In this case main supply of oil is located in the base of crank chamber. Oil is drawn from the lower portion of the sump through a filter and is delivered by means of a gear pump at pressure of about 1 bar to the main bearings. The big end bearings are lubricated by means of a spray through nozzles. Thus oil also lubricates the cams, crank shaft bearings, cylinder walls and timing gears. An oil pressure gauge is provided to indicate satisfactory oil supply.

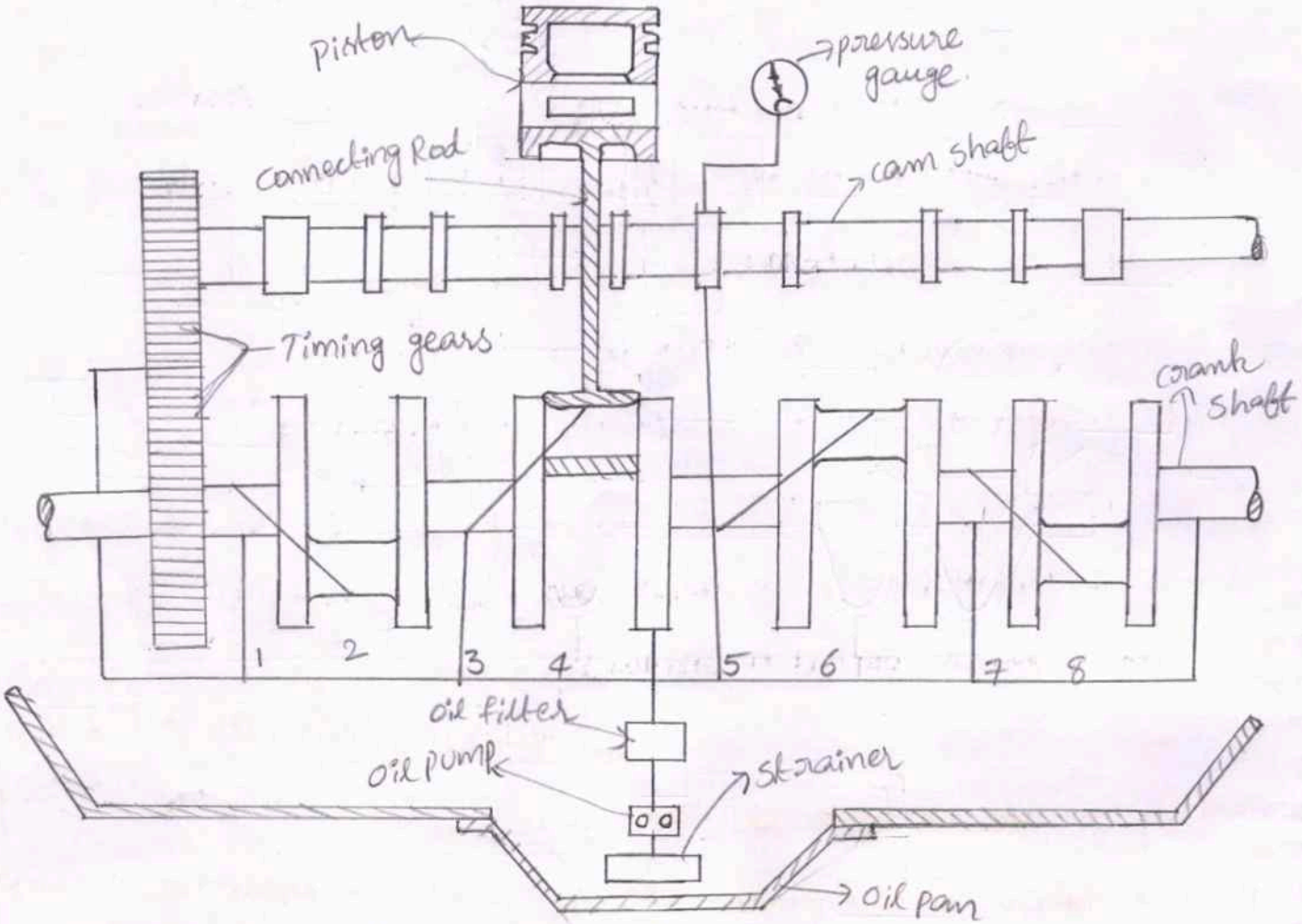
The system is less costly to install as compared to pressure system. It enables higher bearing loads and engine speeds to be employed as compared to splash system.



a) Splash system

c) Full pressure system:- In this system, oil from oil sump is pumped under pressure to the various parts requiring lubrication. The oil is drawn from the sump through filter and pumped by means of a gear pump. Oil is delivered by the pressure pump at pressure ranging from 1.5 to 4 bar. The oil under pressure is supplied to main bearings of crank shaft and cam shaft. Holes drilled through the main crank shafts bearing journals, communicate oil to the big end bearing and also small end bearings through hole drilled in connecting rods. A pressure gauge is

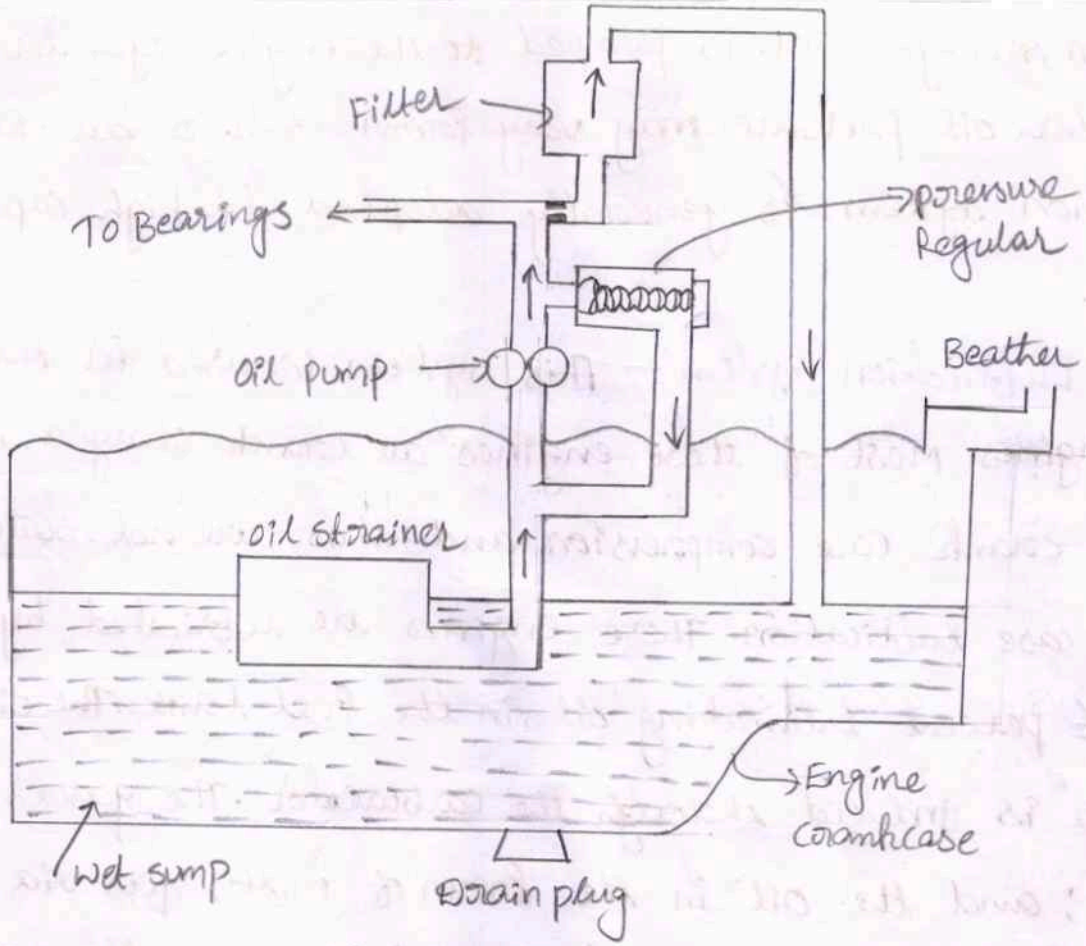
provided to confirm the circulation of oil to the various parts A pressure regulating valve is also provided on the delivery side of this pump to prevent excessive pressure.



Full pressure system

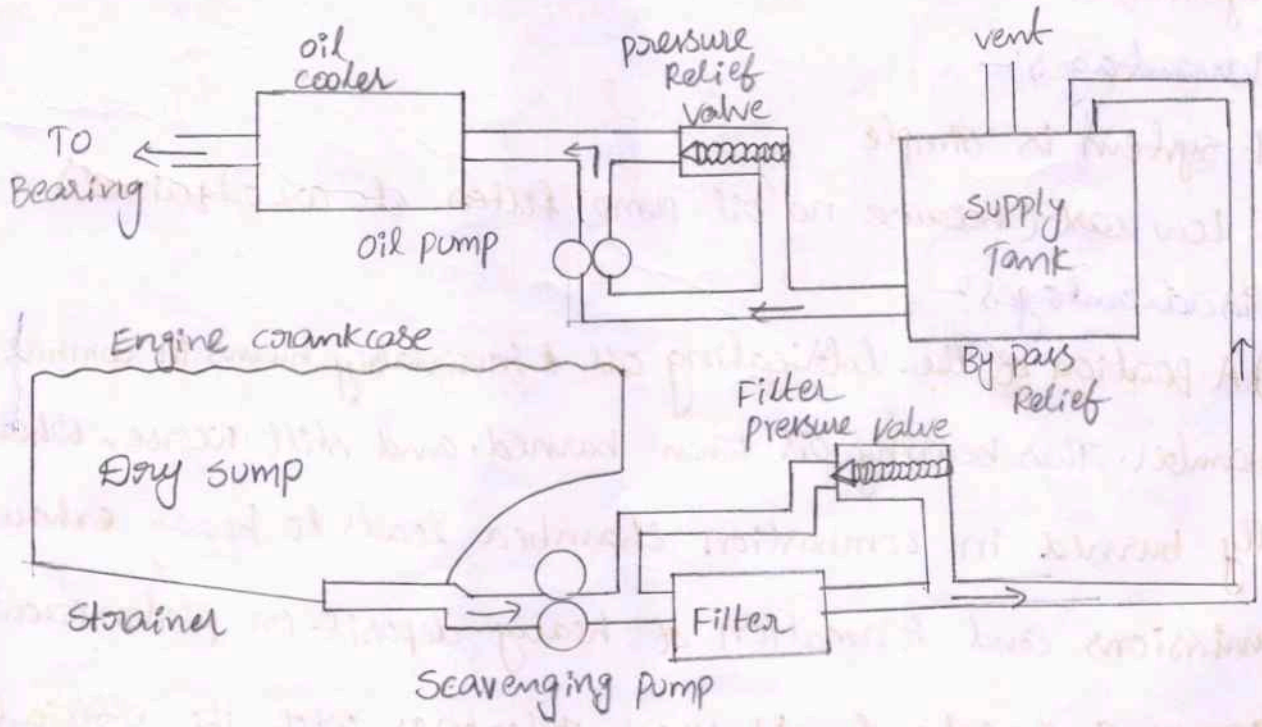
This system finds favour from most of the engine manufacturers as it allows high bearing pressure and Rubbing speeds.

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



Wet sump Lubrication system

2) Dry sump Lubrication system:- In this system, the oil from the sump is carried to a separate storage tank outside the engine cylinder block. The oil from sump is pumped by means of a sump pump through filters to the storage tank.



Dry sump Lubrication system

oil from storage tank is pumped to the engine cylinder through oil cooler. oil pressure may vary from 3 to 8 bar. Dry sump lubrication system is generally adopted for high capacity engines.

3) Mist Lubrication system:- This system is used for two Stroke cycle engines. Most of these engines are crank charged, i.e they employ crank case compression and thus, are not suitable for crank case lubrication. These engines are lubricated by adding 3 to 6 percent lubricating oil in the fuel tank. The oil and fuel mixture is induced through the carburetor. The gasoline is vaporised; and the oil in the foorm of mist, goes via crank case into the cylinder. The oil which impinges on the crankcase walls lubricates the main and connecting rod bearings, and rest of the oil which passes on the cylinder during charging and scavenging periods, lubricates the piston, piston rings and the cylinder.

Advantages:-

1. system is simple.
2. Low cost (because no oil pump filter etc. are required)

Disadvantages:-

1) A poation of the lubricating oil is invariably burns in combustion chamber. This bearing oil when burned, and still worse, when partially burned in combustion chamber leads to heavy exhaust emissions and formation of heavy deposit on piston crown, ring grooves and exhaust port which interferes with the efficient

engine operation.

2. One of the main functions of lubricating oil is the protection of anti-friction bearings etc. against corrosion. Since the oil comes in close contact with acidic vapours produced during the combustion process, it rapidly loses its anti-corrosion properties resulting in corrosion damage of bearings.

3. For effective lubrication oil and fuel must be thoroughly mixed. This requires either separate mixing prior to use or use of some additive to give the oil good mixing characteristics.

4. Due to higher exhaust temperature and less ~~effective~~ efficient scavening the crank case oil is diluted. In addition some lubricating oil burns in combusion chamber. This results in 5 to 15 percent higher lubricant consumption for two stroke engine of similar size.

5. Since there is no control over the lubricating oil, once introduced with fuel, most of the two stroke engines are over-oiled most of the time.

Actual cycles & their Analysis

Actual cycles have less efficiency than the ideal cycles

The major losses are due to,

i) Variation of specific heats with temperature.

ii) Dissociation of combustion products. (fuel & air do not completely combine at high temp (1600K) & leads to presence of CO, H_2, H & O_2 at equilibrium conditions)

iii) progressive combustion.

iv) Incomplete combustion of fuel.

v) Heat transfer into the walls of combustion chamber.

vi) Blow down at the end of exhaust process.

vii) gas exchange process.

comparision of Air standard cycles & Actual cycles:-

The actual cycles for I.C Engines differ from air standard cycles in many respects. These differences are mainly due to:

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

ii) The change in chemical composition of working substance.

iii) The variation of specific heats with temperature.

iv) The progressive combustion rather than the instantaneous combustion.

v) The heat transfer to and from the working medium.

vi) The substantial exhaust blow down loss. i.e loss of work in the expansion stroke due to early opening of the exhaust valve.

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

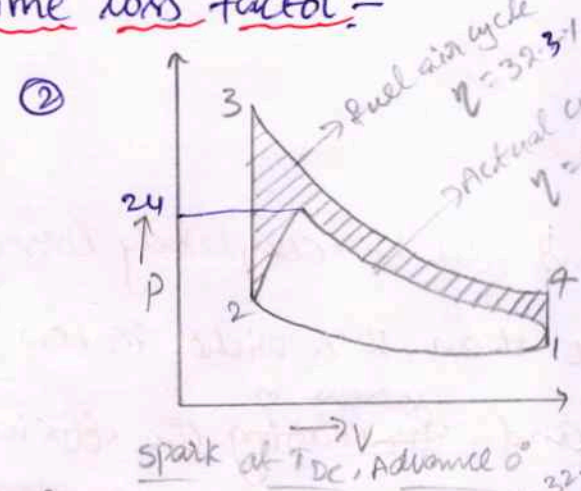
⇒ Out of these, major influence is ~~excesised~~ exercised by

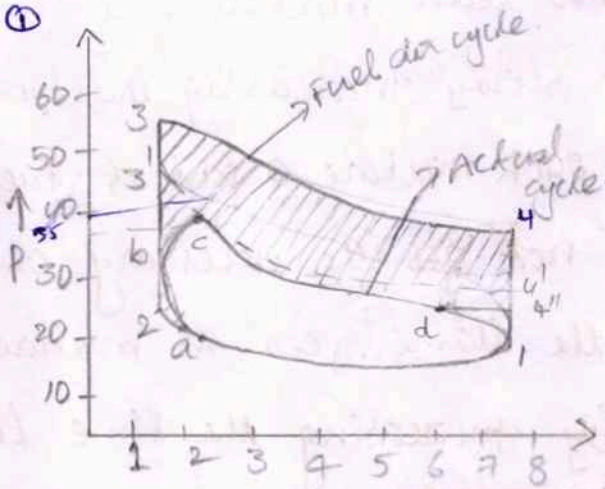
i) Time loss factor (i.e. Loss due to time required for mixing of fuel and air and also for combustion)

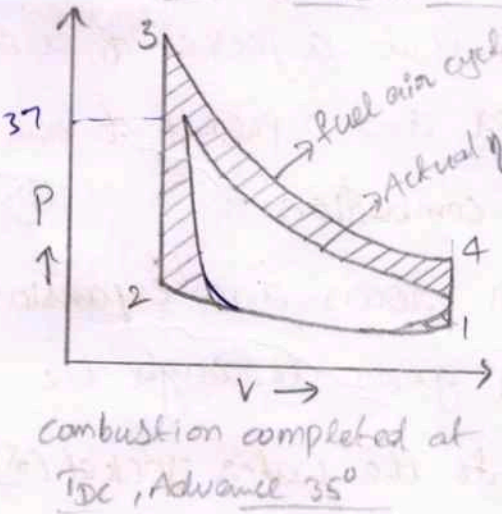
ii) Heat loss factor (i.e. loss of heat from gases to cylinder walls ~~value~~)

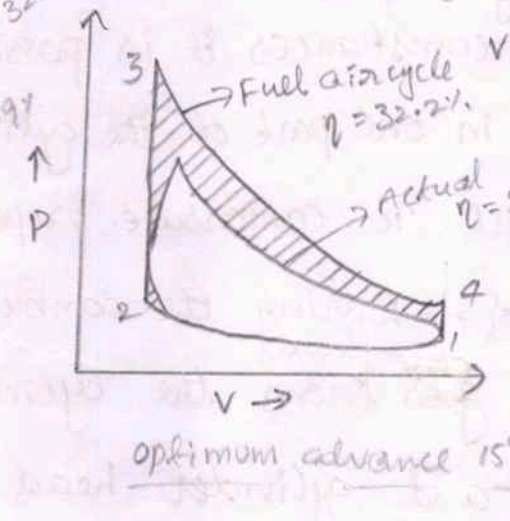
iii) Exhaust Blow down factor (loss of work in the expansion stroke due to early opening of the exhaust valve).

Time loss factor :-









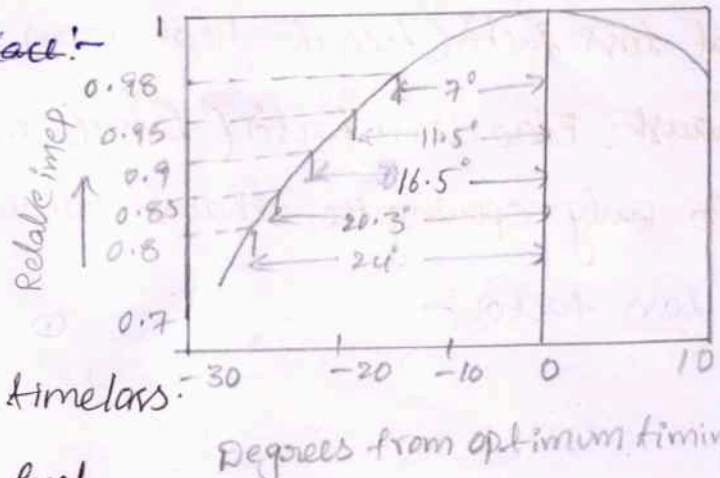
In air standard cycles heat addition is assumed to be an instantaneous process, but in actual cycles it is over a ~~definir~~ definite period of time. The time required for the combustion is such that under all circumstances some change in volume takes place while it is in progress. The crankshaft will usually turn about 30°-40° between the initiation of spark and end of

combustion process. There will be a time loss during this period and is called time loss factor.

In time loss factor spark advanced and air fuel mixture and clearence volume place key role.

Power ~~loss due to ignition place~~:-

Slightly linear mixture gives maximum efficiency. but too lean mixture will burn slowly increasing the time loss. In a rich mixture a part of fuel will not get the necessary oxygen and will be completely lossed. Also the flame speed in mixtures more than 10% richer is low thereby increasing the time losses and ~~increasing~~ decrease in the efficiency.



Under this circumstances it is possible that a pocket of excess oxygen is present in one part of the cylinder and a pocket of excess fuel in another part. To compensate proper combustion.

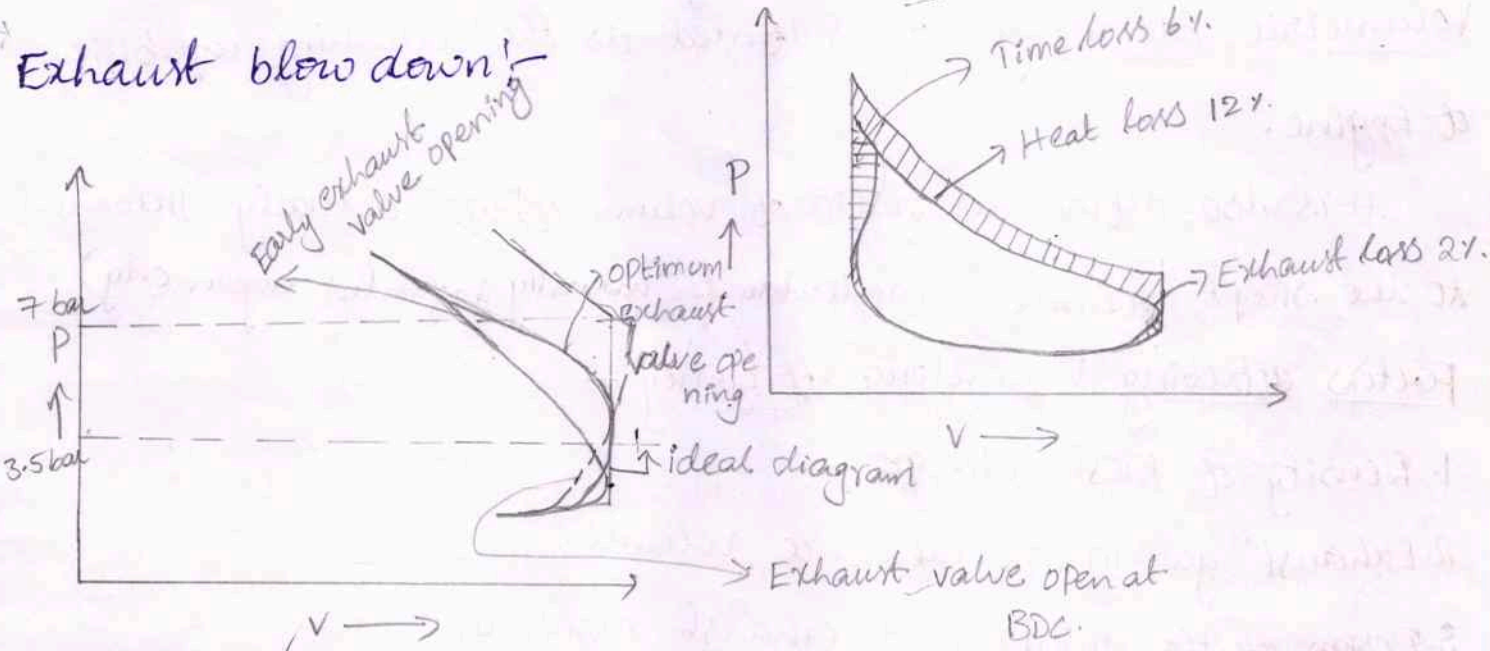
Heat Loss Factor:- During the combustion process and expansion stroke the heat ~~goes~~ flows from the cylinder gases through the cylinder walls and cylinder head into the water jacket (or) cooling fins. Some of the heat will be carried out by lubricant oil and coolants.

Heat loss during combustion will naturally have the maximum effect on the cycle efficiency while heat loss just before the end of expansion stroke can have very little effect because of its contribution to ~~the~~ the useful work is very less.

About 15% of total heat is lost during combustion and expansion. Of these, however, much is lost so late in the cycle to have contributed to useful work. If all the heat loss is recovered only about 20% of heat may appear as useful work.

Different losses are shown in the p-v diagram:

Exhaust blow down:-



The cylinder pressure at the end of exhaust stroke is about 7 bar depending upon the compression ratio employed. If the exhaust valve is opened at BDC, the piston has to do work against high cylinder pressures during the early part of the exhaust stroke. If the exhaust valve is opened too early a part of expansion stroke is lost. The best compromise is to open the exhaust valve 40 to 70° before BDC there by reducing the cylinder pressure to half way (say 3.5 bar) before the exhaust stroke begins.

Loss due to Gas Exchange Process:- The difference of work done in expelling the exhaust gases and the work done by the fresh charge during the suction stroke is called as pumping work. In other words loss due to the gas exchange process is due to the

pumping gas from lower inlet pressure to higher exhaust pressure the pumping loss increases at part throttle because throttling reduces suction pressure.

Pumping loss also increases with the speed. The gas exchange process affects the volumetric efficiency of the engine

Volumetric Efficiency :- It is defined as the breathing capability of Engine.

It is also defined as the ratio of volume of air actually inducted to the swept volume. (applicable for naturally aspirated engines only)

Factors affecting Volumetric efficiency :-

1. Density of fresh charge.
2. Exhaust gas in the clearence volume.
3. Design of the intake and exhaust manifolds.
4. The timing of intake and exhaust valves.

Losses Due to Rubbing friction :-

This losses are due to friction between piston and cylinder walls, friction in various bearings and the energy spent in operating cooling water pump, ignition systems and fans etc. The friction of piston rings increases with the engine speed. & also with increase in m.e.p.

The efficiency of the engine is maximum at full loads and decreases at part loads. It is because the percentage of direct heat loss, pumping work and rubbing friction losses increases at part loads. The approximate losses for Gasoline Engines of compression ratio 8:1 using a chemically correct mixture,

are given in the table, as percentage of fuel energy input.

Typical losses in a Gasoline Engine for $r = 8$

S.NO	item	Full load (%)	~~Half~~ part load (%)
(a)	Air standard cycle efficiency.	56.5	56.5
i)	Losses due to variation of specific heat and chemical equilibrium	13.0	13.0
ii)	Loss Due to progressive combustion	4	4
iii)	loss due to incomplete combustion.	3	3
iv)	Direct heat loss	4	5
v)	Exhaust blow down loss	0.5	0.5
vi)	pumping loss	0.5	1.5
vii)	Rubbing friction loss	3	6
(b)	Fuel air cycle efficiency = Air standard efficiency − losses (1) due to variation of specific heat and chemical equilibrium.	43.5	43.5
c)	Gross indicated thermal efficiency = point (b) − (2+3+4+5) = point 6 − point (2+3+4+5)	32	31
d)	Actual break thermal efficiency = indicated thermal efficiency − (point 6 + point 7)	28.5	23.5

Actual cycles & fuel-air cycles in CI Engine:-

In the diesel cycle losses are lessthan the otto cycle. The main loss is due to incomplete combustion and is the cause of main difference b/w fuel-air cycle and actual cycle of a diesel engine. In a fuel-air cycle the combustion is supposed to be completed at the end of constant pressure burning, whereas in actual practice after burning continueous upto half of the expansion stroke. The ratio b/w the actual efficiency and the fuel-air cycle efficiency is about 0.85 in the Diesel engines In the fuel-air cycle, when allowance is made for the presence of fuel and combustion products, varies reduction in the cycle efficiency. In actual cycles allowances are also made for the losses due to phenomenon such as heat transfer and finite combustion time. This reduces the cycle efficiency further.

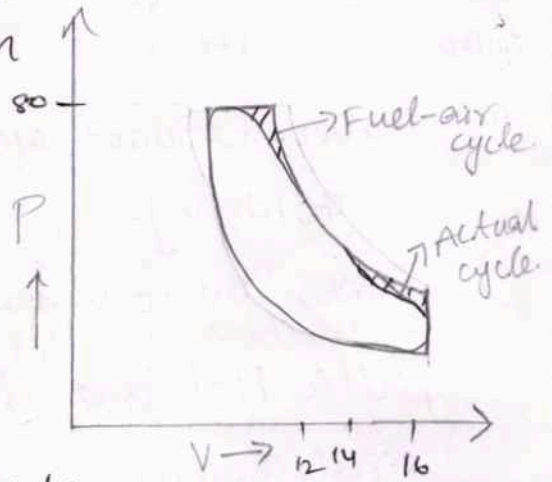


Fig: 2-Stroke CI engine

Combustion In S.I Engine

Combustion :- It is a chemical reaction in which certain elements of fuel like hydrogen and carbon combined with oxygen liberating heat energy and causing an increase in temperature of gases. Combustion takes place either in

i) Homogeneous mixture.

ii) Heterogeneous mixture.

Homogeneous Mixture :- 1. In S.I Engines nearly homogeneous mixture of air and fuel is formed in the carburator. (fuel and oxygen molecules are uniformly distributed)

2. Once fuel vapour & air mixture is ignited, the flame front appears and rapidly spreads through the mixture.

3. The flame front is a narrow zone separating the fresh mixture from the combustion chamber.

4. The flame propagation is caused by heat transfer and diffusion of burning fuel molecules from the combustion zone to the adjacent layers of unburnt mixture.

5. The velocity with which the flame front moves with respect to unburned mixture in the direction Normal to its surface is called Normal Flame Velocity.

6 In a homogeneous mixture with an equalence ratio ϕ (Ratio of actual fuel air ratio to stochiometric fuel-air ratio) close to one. the flame speed is of order of 40 cm/sec.

7. However in S.I Engines the flame speed is obtained when ϕ is in b/w 1.1 and 1.2 i.e when the mixture is slightly richer than stoichiometric value.

Heterogeneous Mixture:-

1. In Heterogeneous mixture the rate of combustion is determined by the velocity of mutual diffusion of fuel vapours and air and the rate of chemical reaction is of minor importance.
2. When the mixture is higher the combustion can takeplace in an overall linear mixtures since there are always local zones where ϕ value varies will be 1 and 1.2 corresponding to maximum rate of chemical reaction.
3. Ignition starts in this zone and flame produced helps to burn the fuel in the adjoining zones where the mixture is linear.
4. Similarly in the zones where the mixture is rich the combustion occurs because of high temperatures produced due to combustion initiated in the zones where ϕ is 1 to 1.2

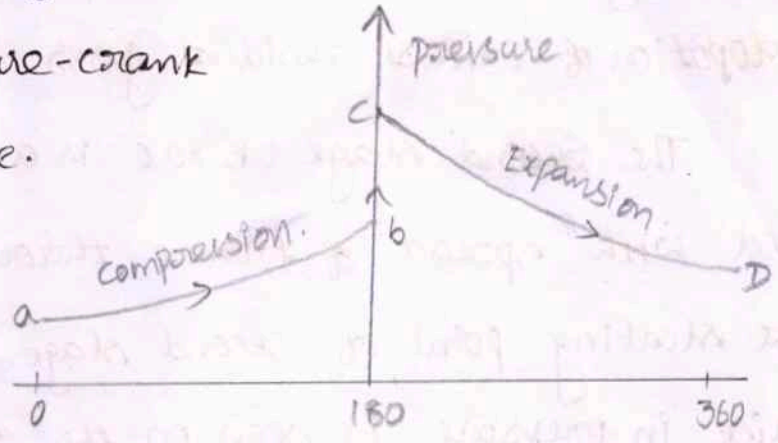
Combustion in S.I Engines:-

Under normal operating conditions combustion is initiated towards the end of compression stroke at the spark plug by an electric discharge. A turbulent flame develops following the ignition and propagates through this premixed charge of fuel and air, and also residual gases in the clearence volume untill it reaches the combustion chamber walls.

Stages of combusion in S.I Engines:-

A typical theotical pressure-crank angle diagram is shown above.

a-b → compression process
b-c → combusion process
c-d → Expansion process

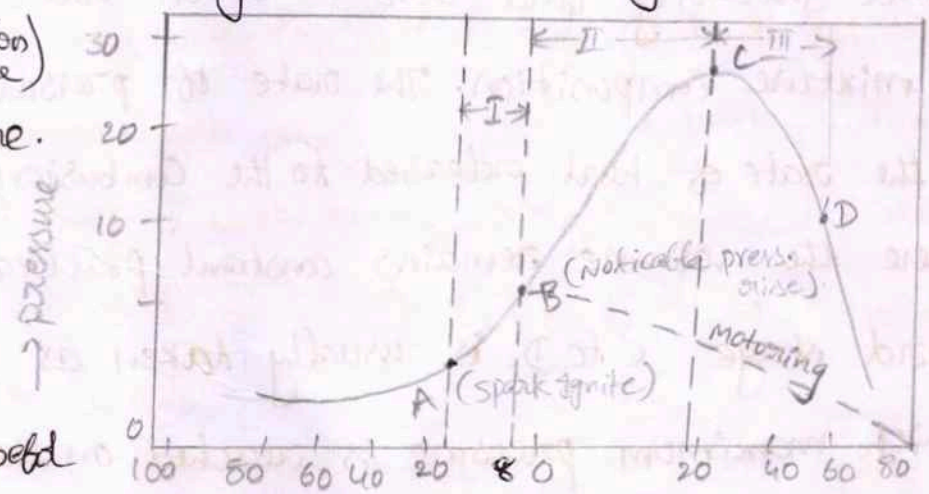


fd an ideal four stroke S.I Engine.

By the diagram we can observe that the entire pressure rise during combusion takeplace at constant volume i.e at TDC. However in actual engine this doesnot happen.

Sir Recardo known as father of Engine research descaibes the combusion process in S.I Engine as consisting of three stages.

1. Ignition Lag (preperation phase)
2. propogation of flame.
3. After burning.



i) 'A' is the point of parsage of spark (20° befd TDC). ii) 'B' is the point at which the begining of pressure rise can be detected. (8° before TDC). iii) 'C' is the point at peak pressures.

The first stage (A to B) represents ignition lag (d) preparation phase in which growth and development of a self propagating nucleus of flame takesplace. This is a chemical proces

depending upon pressure and temperature, Nature of the fuel and proportion of Exhaust residual gases.

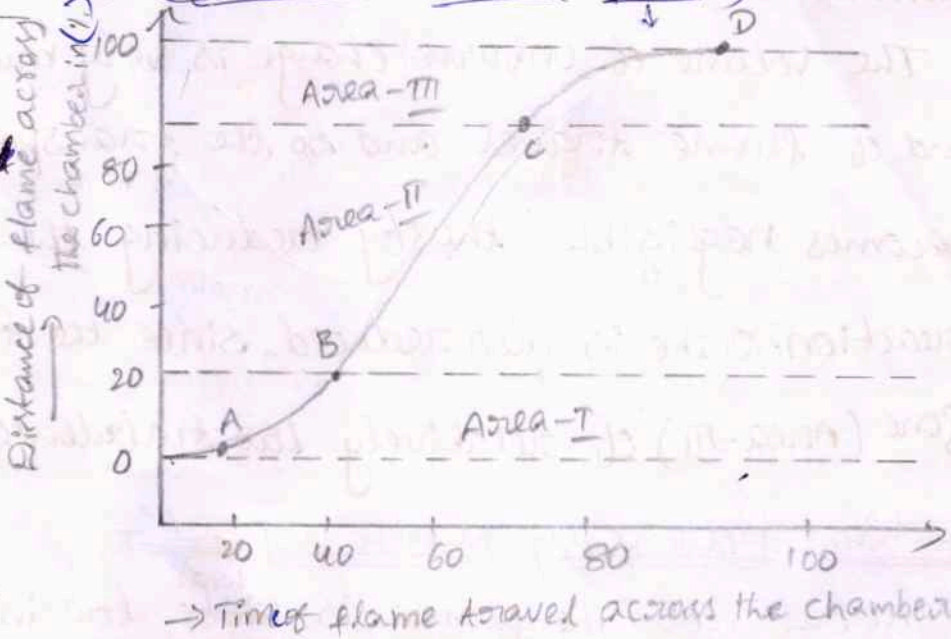
The second stage (B to C) is a physical one and it is concerned with spread of flame throughout the combusion chamber. The starting point of second stage is where the first measurable rise in pressure is seen on the indicated diagram. This can be seen from the deviation from the motoring curve. During this stage the flame propagates practically at a constant velocity. Heat transfer is low to the cylinder wall because only a small part of burning mixture comes in contact with the cylinder wall. The rate of heat release depends on the turbulence intensity and also reaction rate which is dependant on mixture composition. The rate of pressure rise is proposional to the rate of heat released to the combusion chamber walls where the volume remains constant practically. The starting of third stage (C to D) is usually taken as the instant at which maximum pressure is reached on the indicated diagram. The flame velocity decreases during this stage. The rate of combustion becomes low due to lower flame velocity and reduced flame front surface. Since the expansion stroke starts before this stage of combustion, with the piston moving away from TDC there can be no pressure rise during this stage.

Flame front propagation :- (Details of flame travel.)

The two important factors which determined the rate of movement of flame front across the combustion chamber are,

i) Reaction rate.

ii) Transposition rate.



The reaction rate is the result of a purely chemical combination process in which the flame eats its way into the unburnt charge The Transposition rate is due to the physical movement of the flame front relative to the cylinder wall. and is also the result of pressure differential b/w the burning gases and unburnt gases in the combustion chamber.

The above fig. shows the rate of flame propagation. In Area-I (A to B) the flame front progresses relatively slowly due to a low transposition rate and low turbulance. low reaction rate plays a dominate role resulting in a slow advance of flame. Also since the spark plug is to be necessarily located in a quiescent layer of gas that is close to the cylinder wall, the lack of turbulence reduces the reaction rate and hence the flame speed. As the flame front leaves the quiescent zone and proceds into more turbulent areas (Area-II) where it consumes a greater mass of mixture, it progresses more rapidly and at

constant rate.

The volume of unburned charge is very much less towards the end of flame travel and so the transposition rate again becomes negligible thereby reducing the flame speed. The reaction rate is also reduced, again since the flame is entering a zone (Area-III) of relatively low turbulence (C to D)

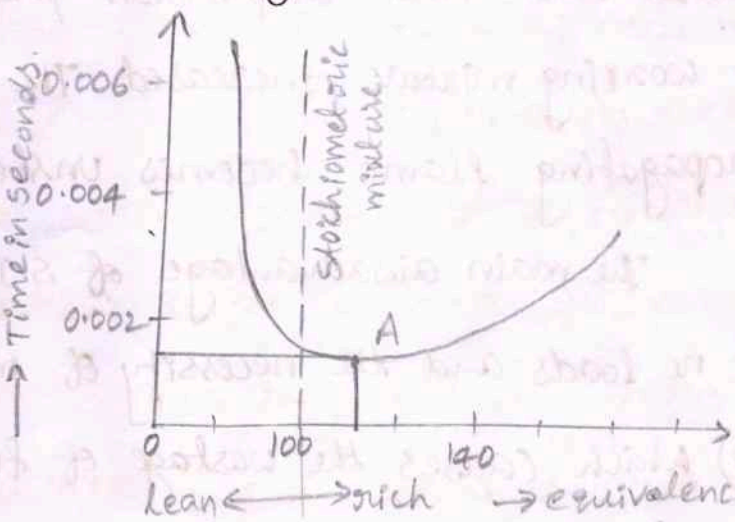
Factors Influencing Flame Speed :-

1. Turbulance :- The flame is ~~white~~ wide low in non-turbulant mixtures and it increases with increasing turbulance. which is mainly due to additional physical inter mingling of burning and unburn particles at the flame front. The turbulance in the incoming mixture is generated during the admission of fuel air mixture through comparitively narrow sections of intake pipe, valve openings etc. in the suction stroke. turbulance which is supposed to consists of many minute swirls appears to increase the rate of reaction and produce a higher flame speed than that made up of larger and fewer swirls. Suitable design of combustion chamber increases the turbulance during the compression stroke.

Generally turbulance increases to the heat flow to the cylinder walls. It also accelerate the chemical reaction by intake mixing of fuel and oxygen so that spark advance may be reduced. this helps in buring lean mixtures also. The increase of flame speed due to turbulance reduces the combustion duration. and hence minimizes the tendency of abnormal combustion. However

excessive turbulance may extinguish the flame resulting in rough and noisy operation of the engine.

Fuel-air ratio:- It has a significance and influence on flame speed. The highest flame velocity is obtained with somewhat richer mixture (point-A) as shown in the fig. which shows the effect of mixture strength on the rate of burning as indicated by the time taken for complete burning in a given engine. When the mixture is made leaner (or) richer the flame speed decreases. Less thermal energy is released in case of lean mixture resulting in lower flame temperature. very rich mixtures lead to incomplete combustion which results again in the release of less thermal energy.



Temperature and pressure:- Flame speed increases with an increase in intake Temperature and pressure. A higher initial pressure and Temperature may help to form a better homogeneous mixture which increases the flame speed. this is possible because of overall increasing density of the charge.

Compression Ratio:- A higher compression ratio increases the pressure and temperature of the working mixture which reduce the initial preparation phase of combustion and hence less ignition advance is needed. high pressure and high temperature of the

compressed mixture also speed up the second phase of combustion. Increased compression ratio reduces the clearence volume and therefore increases the density of the gases during burning. This increases the peak pressure and temperature and therefore total combustion duration is reduced. Thus the engines having higher compression ratios have higher flamespeeds.

Engine Output:- The cycle pressure is increased when the engine output is increased. With the increased throttle opening the cylinder gets filled to higher density. this results in increased flame speed. When the output is decreased by throttling the initial and final compression pressures decrease and dilution of working mixture increased. The smooth development of self propagating flame becomes unsteady and difficult.

The main disadvantage of S.I Engine are the poor combustion at no loads and the necessity of mixture enrichment (Φ b/w 1.2 to 1.3) which causes the wastage of fuel and discharge of unburn gases in the atmosphere.

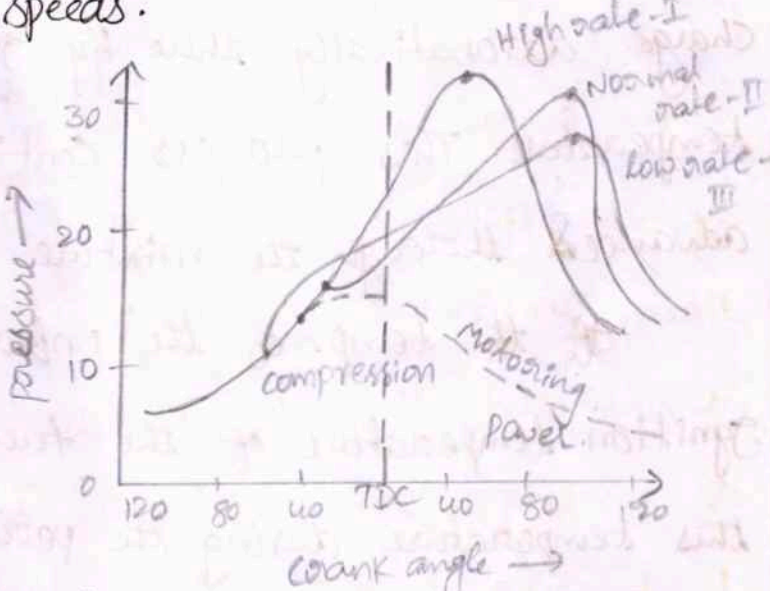
Engine speed:- The flame speed increases almost linearly with the engine speed since the increase in engine speed increases the turbulence inside the cylinder. The time required for the flame to transpose the combustion space would be half in the engine space is doubled. The crank angle required for the flame propagation during the entire phase of combustion will remain

nearly constant at all speeds.

Engine size:- The size of the engine does not have much effect on flame propagation. In large engines the time required for complete combustion is more because the flame has to travel a longer distance. this requires increased crank angle duration during the combustion. this is one of the reason why large sized engines are designed to operate at low speeds.

Eg:- Road roller.

Rate of pressure Rise:-



Abnormal combustion:-

In Normal combustion the flame initiated by the spark travels across the combustion chamber in a fairly uniform manner. and at certain operating conditions of the combustion deviates from its normal course leading to loss of performance. and possible damage to the engine. This type of combustion may be termed as an Abnormal combustion (or) Knocking knocking combustion. The consequence of this abnormal combustion process are, the loss of power, Reading preignition and mechanical damage to the engine.

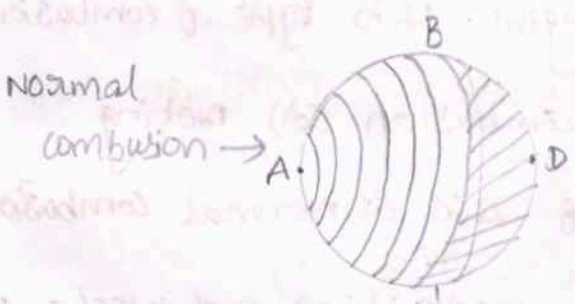
The phenomenon of knock in S.I Engine:-

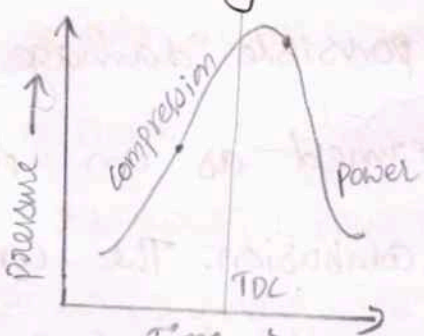
In S.I Engines combustion which is initially b/w spark plug electrodes, spreads across the combustion chamber. a definite flame

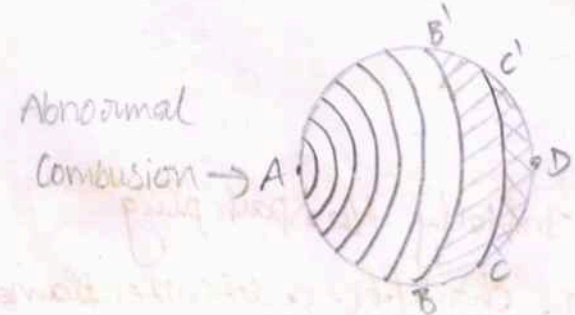
front which separates the flame mixture from the products of combustion travels from the spark plug to the other end of combustion chamber. Heat release due to combustion increases the temperature and pressure of the burned part of mixture above those of unburned mixture. An order added to effect pressure equalization the burned part of the mixture will expand and compress the unburned charge adiabatically there by increasing its pressure and temperature. This process continues as the flame front advances through the mixture. & pres & temp are increased further.

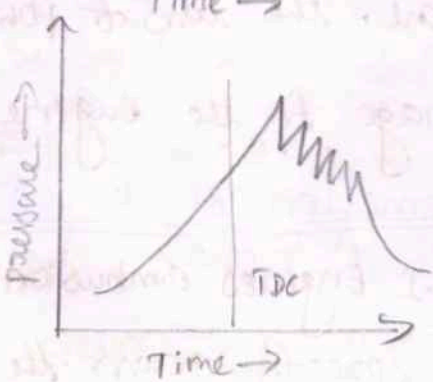
If the temp. of the unburned mixture exceeds the self ignition temperature of the fuel and remains at (or) above this temperature during the period of preflame reactions (ignition lag), auto ignition occurs at various pin point locations. This phenomenon is called as knocking.

The process of auto ignition leads to engine knock.









The process of flame front travels in the combustion chamber can be explained through the diagram.

Because of auto ignition another flame front start travel in opposite directions to the main flame front. When the two flame fronts collide a severe pressure pulse is generated. The gas in the combustion chamber is subjected to compression and "rare fraction" along the pressure pulse until the pressure equilibrium is restored. This disturbance can force close the walls of combustion chambers to vibrate at the same frequency as the gas.

It is to be noted that the onset of knocking is very much dependent on properties of the fuel.

1. If the unburn charge does not reach its auto ignition temp. there will be no knocking
2. If the initial phase that is ignition lag period, i.e., is longer than the time required for the flame front to burn through the unburned charge, there will be no knocking.

When auto ignition occurs two different types of vibration may be produced.

i) In one case a large amount of mixture may auto ignite giving rise to a high pressure throughout the combustion chamber and there will be a direct blow on the engine structure. A human ear can detect the resulting thudding noise. Sound and consequent noise from free vibrations

of the engine part.

ii) In second case large pressure differences may exist in the combustion chamber and the resulting gas vibrations can force the walls of chamber to vibrate at the same frequency as the gas.

Knock Limited parameters:-

1. Com knock limited compression ratio:- (KLCR)

The knock limited compression ratio is obtained by increasing the compression ratio on a variable compression ratio engine untill incipient knocking is observed. Any change in operating conditions such as air-fuel ratio (or) in the engine design that increases the KLCR is said to reduce the tendency towards knocking.

2. knock limited inlet pressure:-

The inlet pressure can be increased by opening the throttle (or) increasing supercharger delivery pressure untill incipient knock is observed. An increase in knock limited inlet pressure indicates reduction in the knocking tendency.

3. knock limited indicated mean effective pressure:-(KLIMEP)

An useful measure of knocking tendency called performance number has been developed from the consept of KLIMEP. This number is defined as the ratio of KLIMEP into the fuel to KLIMEP with ~~iso~~ Iso-octane when the inlet pressure is kept constant.

This performance number is related to octaine number and one of the advantage of this is that it can be applied to fuels whose knocking characteristics are superior to that of 100 octane ie., it exceeds the octain scale beyond 100.

Relative performance number → rpn

$$rpn = \frac{\text{Actual performance number}}{\text{performance no. corresponding to imep of 100}}$$

Effect of engine variables on knock:-

→ Temperature.
→ pressure.
→ density factd.
→ Time factd.
→ composition factd.

Density factd:-

→ compression ratio. ↑ Kn ↓
→ Mass of inducted charge ↓
→ Inlet temp. of mixture ↓
→ Temp. of combustion chamber walls. Hot spots should be avoided. Spark plug EV
→ Retarding the spark timing.
→ Power o/p of the engine ↓

1) compression ratio increases means pressure and temperature increases then density also increases. so knocking will be increased.

2) Mass of inducted charge:-

If mass increases density increases then pressure and temperature also increases. i.e knocking will be increases.

3) Inlet temp. of mixture:-

The temperature increases the knocking also increases.

4) Temp. of combustion chamber walls :-

It depends upon the spark plug position and Exhaust valve. The Temp. of the combustion chamber walls increases, density increases then knocking tendency increases. Hot spots should be avoided.

5) Retarding the spark timing :-

If Ignition is very near to TDC the knocking will decreases. and the Ignition is longer. ~~the TDC the pressure and temperature increases then knocking tendency increases.~~

6) Power o/p of the engine :-

The pressure and temperature in the combustion chamber increases the power o/p of the engine increases. then knocking tendency also increases. If the pressure of charge decreases, then knocking reduces

Time Factor :- flame speed, increasing Ignition lag & reducing time of exposure of unburnt gases for auto ignition.

1. Turbulance. ↑ knock ↓
2. Engine speed ↑ knock ↓
3. Flame travel distance

⇒ By shortening the time req. to travel (flame), then knocking reduces.

→ Engine size. ✓ (limited cylinder dia of 150 mm)
→ combustion chamber shape. ✓
→ Location of spark plug. (2 or more spark plugs can be used)

Composition Factor :-

1. Air fuel mixture → Flame speed is affected by A/F ratio.
2. octane value of fuel → A higher self ignition temp of fuel & a low preflame reactivity would reduce the tendency of knocking.

paraffin series ↑ knocking
naphthene series moderate "
aromatic series ↓ knock.

→ Usually compounds with more compact molecular structure are less prone to knock.

Increase in variable	Major effect on unburn reduce charge.	Action to be taken to knocking
1. Compression Ratio	increases temperature and pressure.	reduce. (2) ~~knocking to be increases~~.
2. Mass of charge inducted.	increases pressure.	reduce the mass of charge.
3. Inlet Temp.	increases temp. ~~so reduce~~	reduce the inlet temp.
4. Chamber wall temp.	increases temp.	decrease the temp. of combusion valve.
5. Spark advance.	increases pressure and temperature.	retard the spark advance.
6. Air-fuel ratio	increases pressure & temperature.	make very rich mixture.
7. Turbulance	decreases time factd.	Increase the turbulance.
8. Distance of flame travel.	increases time factd.	reduce the distance of flame travel.

*Combusion Chambers for S.I Engines:-

The main objectives & in the design of combustion chamber are,

1) Smooth Engine operation.
a) Reducing the possibility of knocking.
b) Moderate rate of pressure rise.

2) High power output and thermal efficiency.

i) A high degree of turbulance is needed to achieve high flame front velocity. turbulant is induced by inlet flow configuration

ii) ~~Turbo~~ tion of Squish. Squish can be induced in SI Engines by having a bowl in piston (or) with a dome shape cylinder head. Squish is the rapid radial movement of gas trapped in b/w the piston & cylinder head into the bowl (or) dome.

ii) High volumetric efficiency (that is more charge) during suction stroke results in an increased power output.

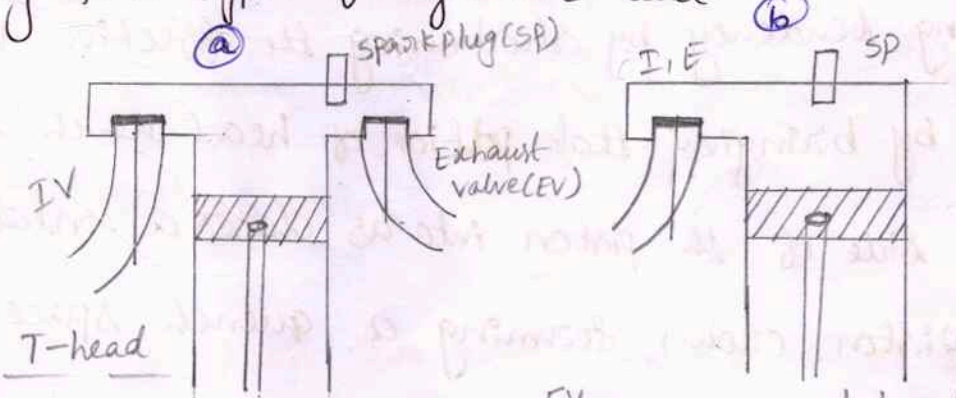
iii) Any design of the combustion chamber that improves its anti-knock characteristics permits the use of a higher compression ratio resulting in increase output and efficiency.

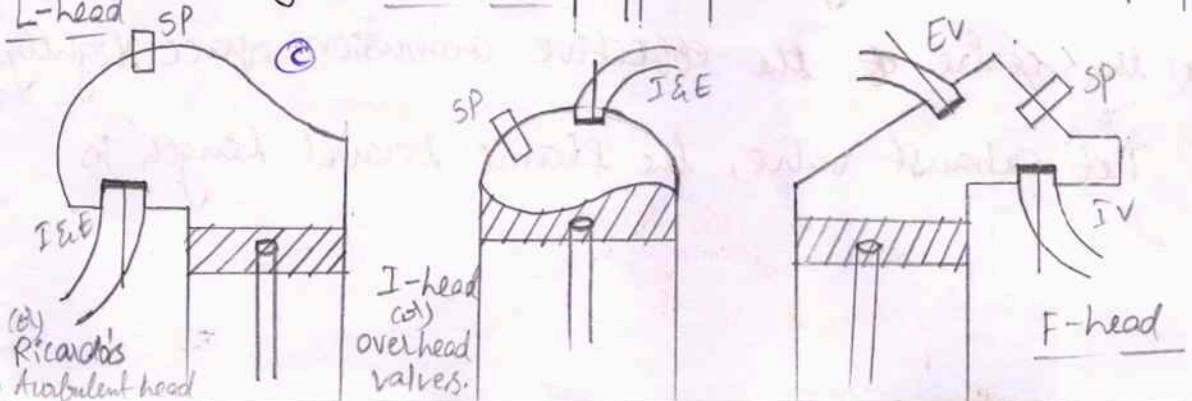
iv) A compact combustion chamber reduces the heat loss during combustion and increases thermal efficiency.

Different Types of heads with Cylinders :- (Different types of combustion chamber)

There are majorly four types of cylinder head.

1. T-head type.
2. L-head type.
3. I-head type.
4. F-head type.





T-head type:- These were used in the early stage of engine development. Since the distance across the combustion chamber is very long, knocking tendency is high in this type of engines. this configaration provides two valves on either side of the cylinder, requiring two cam shafts.

L-head type:- It provides the two valves on the same side of the cylinder and valves are operated by single cam shaft. with the ditachable head it may be noted that cylinder head can be removed without disturbing valve mechanism. fig(b) & fig(c) are L-head type. In fig(b) the air flow has to take two right angles to enter into the cylinder. This causes a loss of velocity head and loss in turbulance level resulting in a slow combustion process.

The main objective of Recardo's turbulant head design is to obtain, fast flame speed and reduce knock. This design reduces knocking tendency by shortening the effective flame travel length by bringing that potion of head which lay over the ~~fother~~ farther side of the piston into as closes a contact as possible with piston crown, forming a quench space. By placing the spark plug in the centre of the effective combustion space slightly towards the hot exhaust valve, the flame travel length is reduced.

Fuels:- There are three types of fuels.

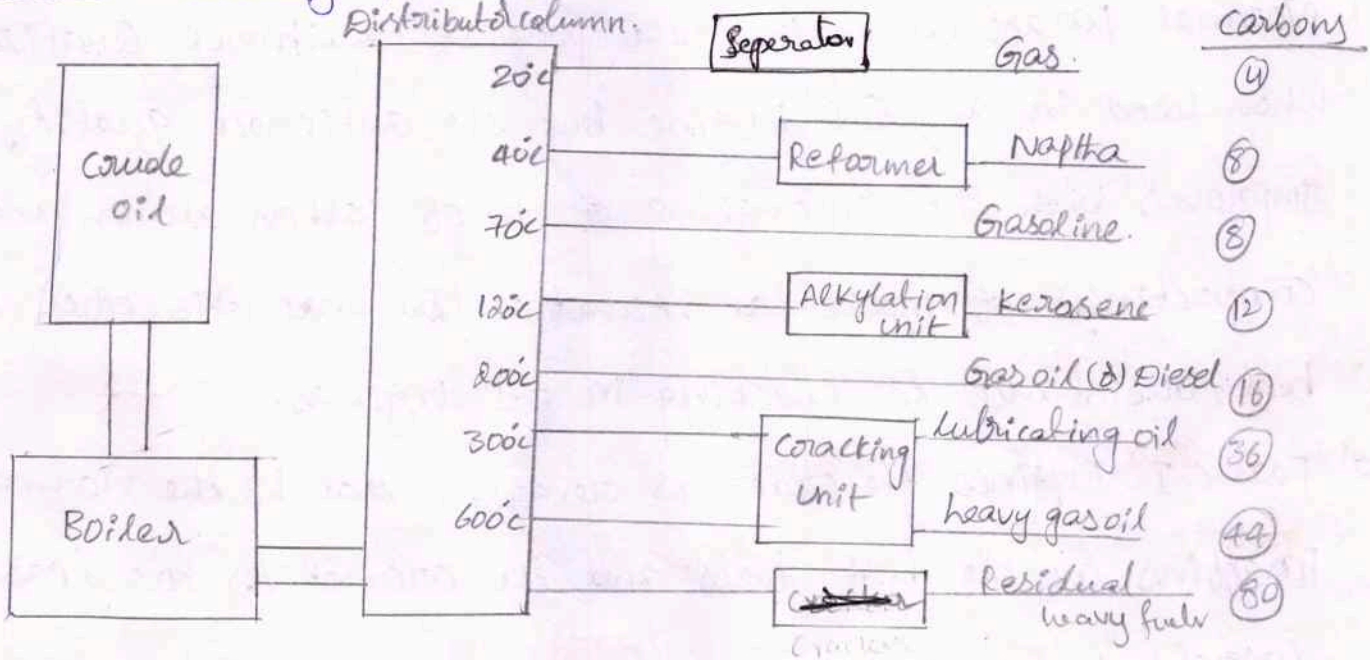
1. Solid fuel (powdered coal)
2. Gaseous fuel (LPG, H_2, CNG, Biogas etc)
3. liquid fuel (Benzyle, ~~Alkahol~~ alcohol and petrolium products).

Alkylation → produce a branched chain iso paraffin

cracking :- breakdown hydrocarbon molecules.

Reformer → convert to low quality stocks into gasoline high octane rating

Petrolium Refining process :-



The chemical structure of the fuels is mainly depends upon different atoms such as carbon, hydrogen, sulphur, oxygen etc. It mainly depends on the following.

1. Molecular structure
2. concentration of different atoms. (O_2, N_2, C, H_2 and water and sand)
3. Linking of carbon and hydrogen atoms influence the chemical and physical properties.
4. Nature of ~~burning~~ bonding determine energy characteristics of fuel

(*) Depending upon no. of carbons and hydrogen atoms, petrolium products are classified as,

hydrocarbons

1) paraffin series. - ~~CH~~ C_nH_{2n+2} (straight chain structure) → (methane, ethane, propane, butane & isobutane)
2) Olefin - C_nH_{2n} (Chain) [~~Hexene~~, Butadiene] (.. double bonds b/w Carbon atoms)

Napthene (lower) – C_nH_{2n} (ring structure) → (cyclopentane) [: called as cyclo paraffins.

Aromatic – C_nH_{2n-6} (ring) (but more stable) → (benzene and tolmene)

↳ chemical formula

2) Olefin :–

hexane, butadiene.

General characteristics due to this molecular structures :–

1. Normal paraffins exhibit the poorest antiknock quality when used in an S.I Engine. but the antiknock quality improves with the increasing of no. of carbon atoms and the compactiness of molecular structure. The aromatic offer the best resistance to knocking in S.I Engines.
2. For C.I Engines the order is reversed that is the normal paraffins are the best fuels and the aromatics are least desirable.
3. As the no. of atoms in the molecular structure increases the boiling temperature is also increases. Thus fuel with fewer atoms in the molecules tends to be more volatile
4. The heating value generally increases as the proportion of hydrogen atoms to carbon atoms in the molecules increases due to higher value of hydrogen than carbon. Thus paraffins have highest heating values and aromatics have least value.

Requirements of Engine fuels (S.I fuels) :–

Important properties of Gasoline are,

i) Volatility :–

1. It determines the stability for use in S.I Engines.
2. Since gasoline is a mixture of different hydrocarbons,

Volatility depends on fractional composition of fuel.

ii) Starting and warming up:-

* certain part of

1) Gasoline should ~~be~~ vapourized at the room temperature for easy starting of the engine.

2) Hence the portion of distillation curve b/w about 0 & 10% boiled off have relatively low boiling temperatures.

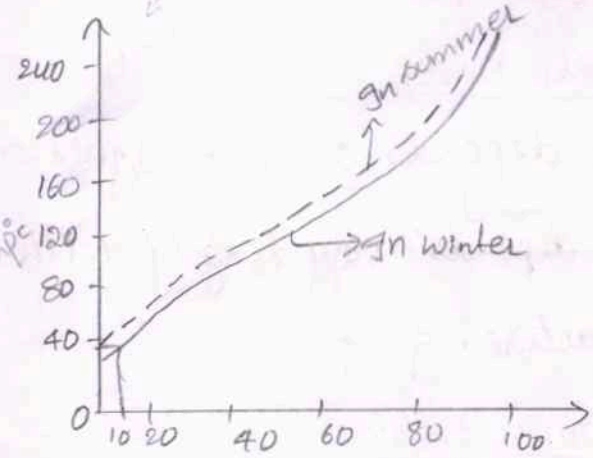


fig:- Distillation process

3) As the engine warms up the temperature will gradually increases to the operating temperature. Low distillation temp's are desirable for best warm up.

iii) Operating Range performance:-

Better vapourization tends to produce uniform distribution of fuel and produce better acceleration characteristics by reducing the quantity of liquid droplets in the intake manifold.

iv) Crankcase Dilution:-

① Liquid fuel causes loss of lubricating oil (By washing away oil from cylinder walls) which deteriorates or the quality of lubricant oil.

② Liquid oil may also dilute lubricating oil and weakens the oil film b/w rubbing surfaces. to prevent (best) vapourization should be proper with minimum distillation temperature.

v) Vapour lock characteristics:-

1. High rate of vapourization of gasoline can upset the carburettor metering or even stop the fuel flow to the engine by setting up a vapour lock in the passages.

2. This demands the presence of high boiling temperature hydrocarbons throughout distillation range

Therefore optimum rate of vaporization should be adopted.

vi) Anti Knock quality :-

1. It depends on self ignition characteristics of the mixture.
2. ~~It depends~~ And vary largely with chemical composition and molecular structure of fuel.

vii) Gum Deposits :-

1. Reactive hydrocarbons and impurities in the fuel have a tendency to oxidize upon storage and form liquid and solid gummy substances.
2. A gasoline with high gum deposites will cause operating difficulties. Such as sticking valves, carbon deposits on piston rings, gum deposits in manifold, clogging of carburetor, enlarging valve stems etc.

viii) Sulphur Content :-

1. Hydrocarbon fuels may contain free sulphur, hydrogen sulphide which are objectionable for several ~~reasons~~ reasons.
2. The sulphur is a corrosive element of the fuel.
3. Sulphur is having low ~~efficient~~ ignition temperature which promote knocking

I head engine :- (overhead valve cc)

It is superior to T-head & L-head engines at higher compression ratios.

Imp. characteristics → less surface to vol ratio & therefore less heat loss

→ less flame travel length & hence greater freedom from knock.

→ higher vol. η from larger valves. → confinement of thermal failures to cylinder head by keeping the hot exhaust valve in the head instead of the cylinder block.

F-head : Compromise b/w L & I head types. In this type one valve is in cylinder head & other in cylinder block. In Modern type : EV in head & IV in cylinder block. The main disadv is IV & EV are separately accessed by two cam shafts.

Combustion In C.I Engines.

Combustion in C.I Engines:- 1. Compression ratio in S.I Engine is from 6:1 to 10:1

2. Compression ratio in C.I Engine is from 16:1 to 20:1

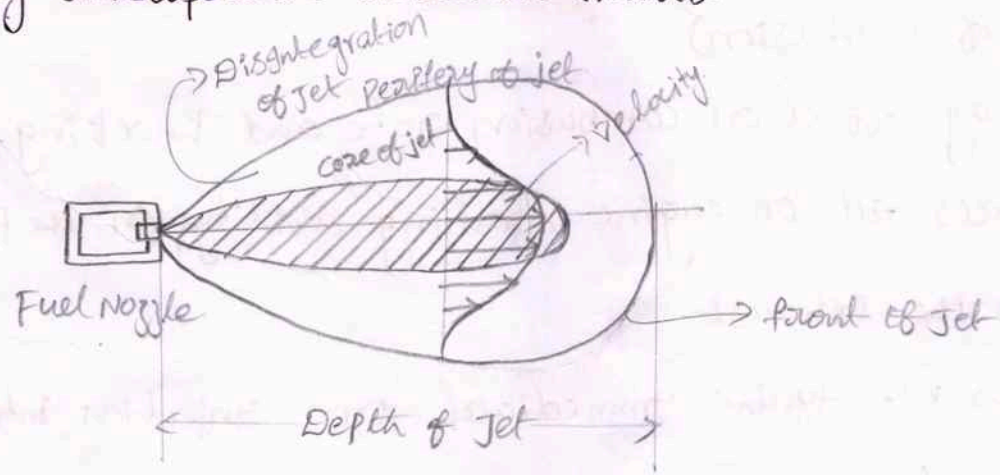
3. In S.I Engines the mixture is ignited at one place and then a single flame front progresses through air-fuel mixture after ignition.

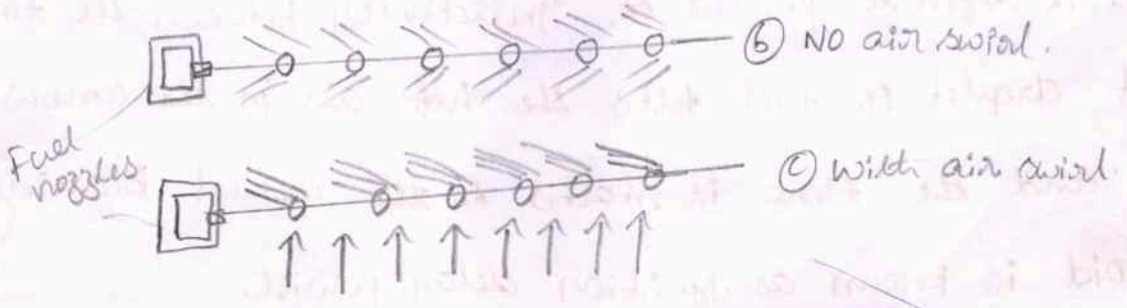
4. In C.I Engines the fuel jet disintegrates into a core of fuel surrounded by a spray envelope of air and fuel particles.

5. This spray envelop is created both by automization and evapourization of fuel.

6. The turbulence of air in combustion chamber parsing across the jet tears the fuel particles from the core.

7. A mixture of air and fuel forms at some location in the spray envelope and oxidation starts.





1. The liquid fuel droplet evaporates by absorbing the latent heat of vapourization from the surroundings air.

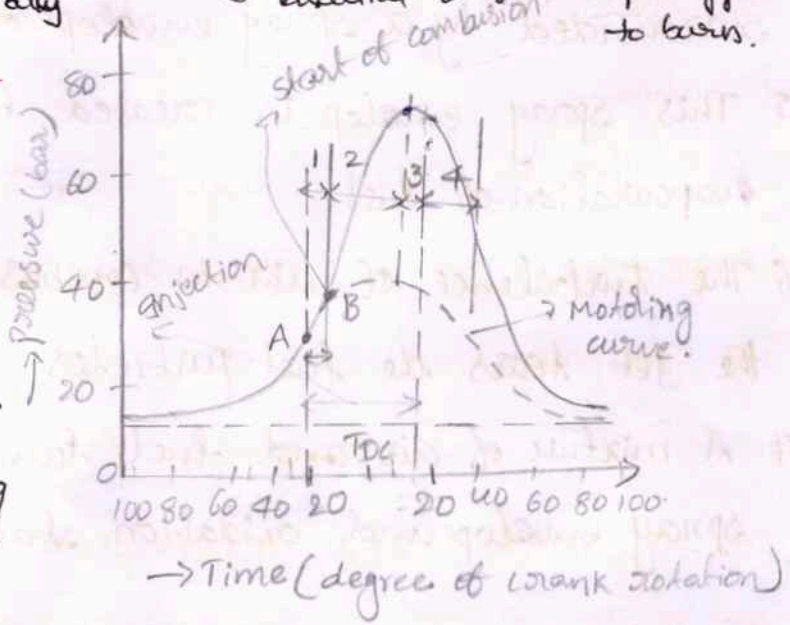
2. As soon as this vapour & air reaches the auto ignition temp and if the local air fuel ratio is with in the combustible range, ignition takes place.

3. Since the fuel ~~deposite~~ droplets can not be injected and distributed uniformly throughout the combustion space, the air fuel mixture is ~~extremely~~ essentially heterogeneous. Air motion is also essential to get enough oxygen to burns.

*Stages of Combustion in C.I Engines:-

1. Ignition Delay period:-

i) It is the period from starting of injection (point A). to the separation of motoring curve. (start of combusion)



2. Ignition lag effect on combusion rate and knocking.

3. It influences ~~all~~ on engine starting ability on the presence of smoke in the exhaust.

4. The fuel does not ignite immediately upon injection into the combusion chamber.

5. There is a definite period of inactivity between the time when the first droplet of fuel hits the hot air in the combusion chamber and the time it starts to the actual burning phase. this period is known as ignition delay period.

It is divided into two parts.

1. Physical delay
2. Chemical delay

i) Physical delay:-

1. It the time b/w the begining of injection and they attainement of chemical reaction conditions.
2. during this peroid fuel is automized, vapourized, mixed with air and raised to self ignition temperature.
3. Fd light fuels physical delay is small where as for heavy fuels it is higher.
4. This physical delay is reduced by high injection pressures and temperature in combusion chamber and turbulence fd improving evapouration.

ii) Chemical delay:- In this peroid reactions starts slowly and then accelerate untill inflamation & ignition takes place.

2. Generally chemical delay is larger than physical delay.
3. It is totally depend on temperatures in the combusion chamber.

2) peroid of rapid combustion:- (or) uncontrolled combusion:-

1. It is also called as uncontrolled combusion. In which pressure rise is rapid.
2. Most of the fuel admitted would have evapourated and form a combustible mixture with air. By this time preflame reactions are also be completed.
3. peroid of rapid combustion is counted from start of combusion to peak pressures on indicated diagram.

4. heat release is maximum during this period and pressure rises depends on delay period.

(3) period of controlled combustion:-

1. Hence ~~When~~ the fuel droplets injected during the second stage burn faster with reduced ignition delay as soon as they find the necessary oxygen and further pressure rise is controlled by rate of injection.
2. This period ends at maximum cycle temperature.

(4) period of after burning:-

1. The unburned and partially burned fuel particles left in the combustion chamber starts burning as soon as they come in contact with oxygen.
2. This process continues for a certain duration called after burning period.
3. It starts from maximum temperature of the cycle and ends over a part of expansion stroke.

Factors affecting the Delay period:-

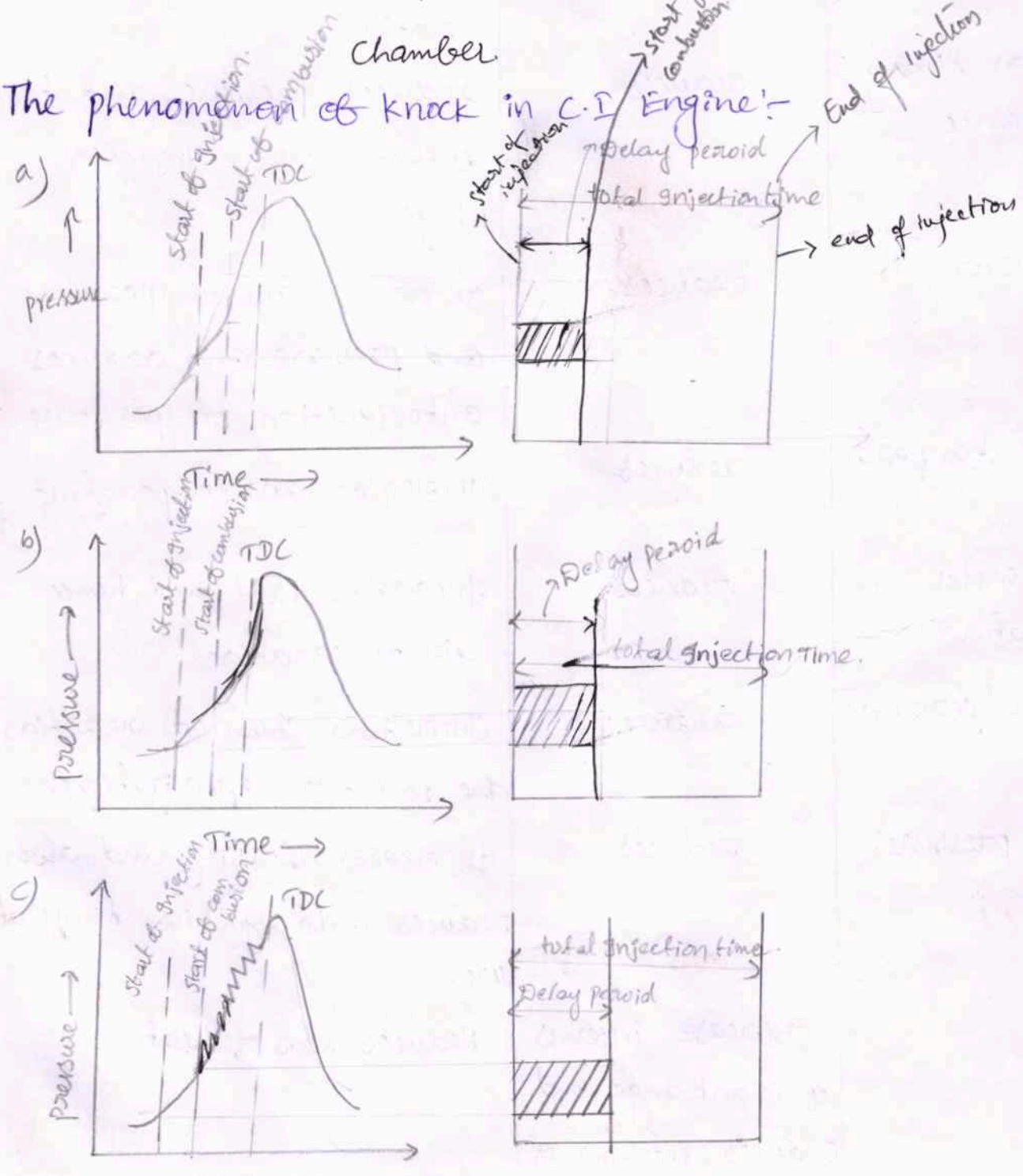
1. compression Ratio (Delay period is shorter due to increase in CR. ↑ Delay period decrease then pressure rise is low.)
2. Engine speed. ↑ ~~Reducing~~ of DP ↓
3. Engine output ↑ Delay period ↓
4. Automization and duration of injection. ↑ DP ↓
5. Injection timing. 9°, 18°, 27° BTDC
6. Intake temperature. ↑ DP ↓
7. Quality of fuel CN ↑ gives lower DP (SIT)

8. Intake pressure. ↑ DP ↓

Increase in variable	Effect on delay period	Reason.
1. Cetane number of fuel.	reduces	reduces the self ignition temperature.
2. Injection pressure.	reduces	reduces physical delay due to greater surface volume ratio.
3. Injection timing in advance	reduces	reduces pressure and temperature when injection begins.
4. Compression ratio.	reduces.	Increases air temperature and pressure and reduces auto ignition temperature.
5. Intake temperature.	reduces.	Increases air temperature.
6. Jacket water temperature.	reduces.	Increases wall and hence air temperature.
7. Fuel temperature	reduces	Increases chemical reaction due to better vapourization.
8. Intake pressure.	reduces	Increases density and also reduces auto ignition temperature.
9. Speed.	Increase interms of crank angle and reduces interms of milliseconds.	Reduces Loss of heat

10. load (air fuel ratio)	decreases.	Increasing the operating temp.
11. Engine size.	decrease interms of crank angle little effect interms of milli seconds.	Larger engines operates normally at low speed.
12. Type of combusion chamber	Lower for engines in Free combusion chamber	Due to compactiness of the chamber.

The phenomenon of knock in C.I Engine:-



1. If the ignition delay is longer the actual burning of first fuel droplets is delayed and a greater Quantity of fuel gets accumulated in the combustion chamber.

2. When the actual burning comences the additional fuel can cause too rapid rate of pressure rise as shown in fig(b) Resulting in a Jarring of forces against the piston and rough engine operation.

3. If the ignition delay is quite long, so much fuel can accumulate, that the rate of pressure rise is almost instantaneous as shown in fig(c). Such situation produces the extreme pressure differentials and violent gas vibrations known as knocking and is evidenced by audible knock.

4. In S.I Engines knocking occurs at the end of combustion.

5. In C.I Engine knocking occurs at the begining of combustion.

6. In order to decrease knocking tendency, it is necessary to start the actual burning as early as possible as the injection begins.

7. It is necessary to decrease the ignition delay and thus decrease the amount of fuel present when the actual burning of first few droplets start..

Combustion chamber for C.I Engine:-

There are two methods of inj C.I Engine combustion chambers are classified into two categeries.

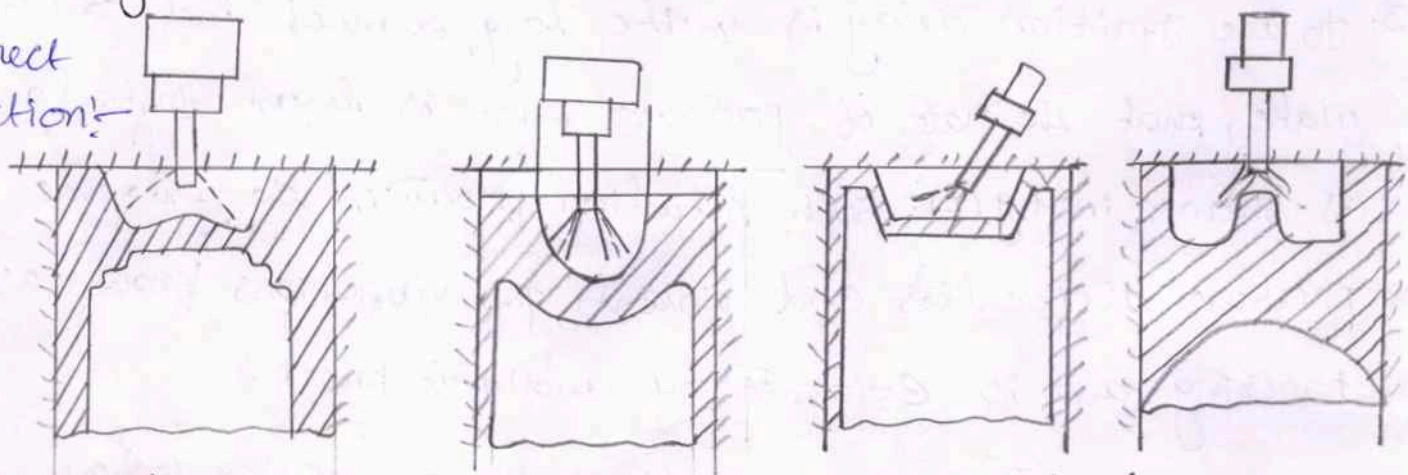
i) Direct Injection Type (DI Type).

ii) Indirect Injection Type

i) Direct Injection Type:- This type of combustion chambers are also called as open combustion chambers. In this type of combustion chambers entire volume is located in the main cylinder and the fuel is injected into this volume.

ii) Indirect Injection Type:- In this, combustion space is devided into two parts one is in the main cylinder & other is in the other part in the cylinder head.

i) Direct Injection:-



a) Shallow depth chamber. b) Hemispherical Chamber c) Cylindrical Chamber d) Toroidal chamber.

Advantages:-

1. Minimum heat loss during combustion because of lower surface area to volume ratio. and hence better efficiency.
2. No cold starting problem.
3. Fine automization because of multihole nozzle.

Disadvantages:-

1. High fuel injection pressure is required.
2. Necessity of accurate metering of fuel by the injection system particularly for small engines.

a) Shallow Depth Chamber:-

1. Depth of the cavity provided in the piston is quite small.

2. This type is applied for large engines running at low speeds.
3. Since the cavity diameter is large the squish is negligable.

b) Hemispherical chamber:-

1. This chamber also gives small squish.
2. However the depth to diameter ratio for cylindrical chamber can be varied to give any desired squish for better performance.

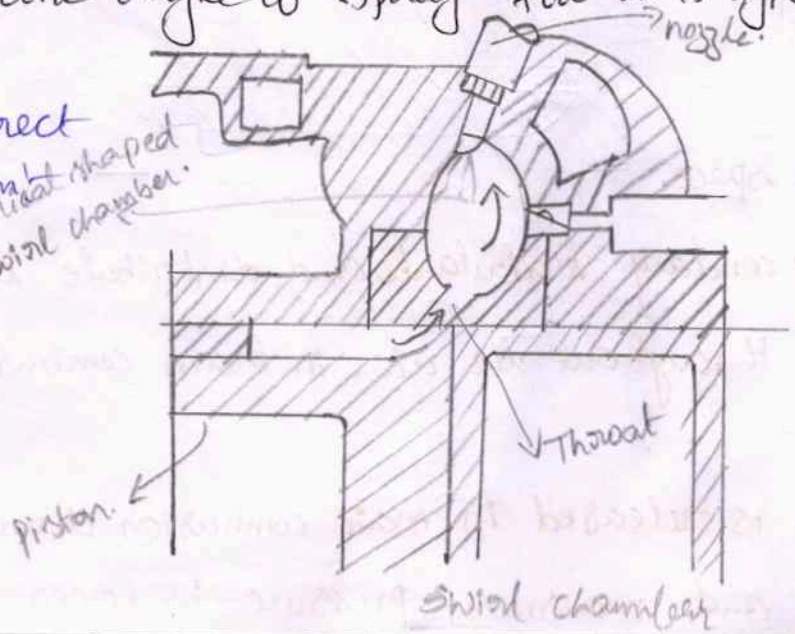
c) Cylindrical chamber:-

1. It is used in recent diesel Engines.
2. Swirl was produced by masking the valve for nearly 180° of circumberence.
3. Squish can also be varied by varying the depth.

d) Toroidal chamber:-

1. This type is to provide a powerful squish along with the air movement similar to that of ~~heavid~~ familid smoke ring with in the Toroidal chamber. Due to this mars needed at ~~inlet~~ and Inlet valve is small and there is better utilization of oxygen.
2. The cone angle of spray for this type of chamber is 150° to 160°.

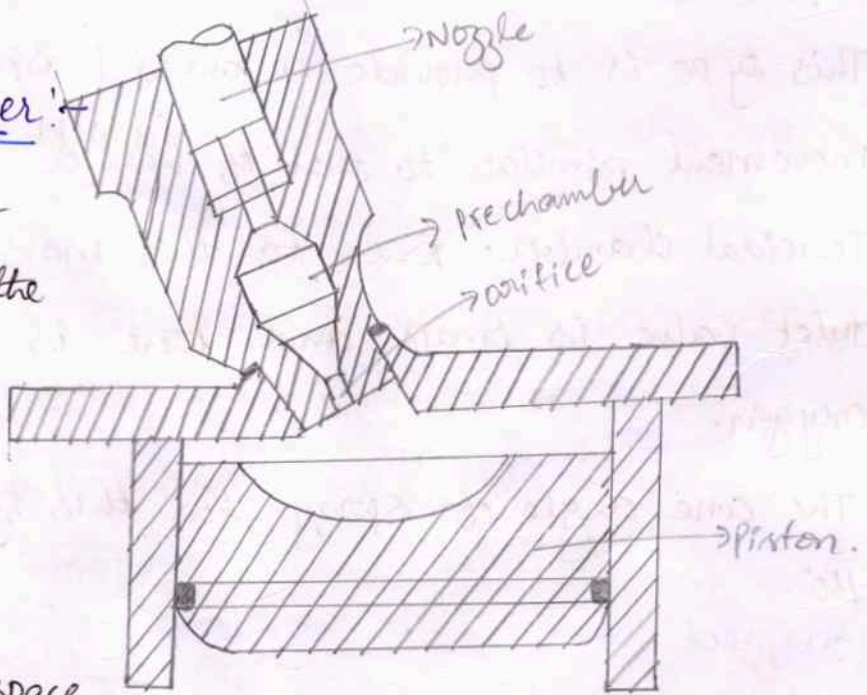
ii) Indirect Injection:- spherical shaped swirl chamber.



Swirl Chamber:-

1. Swirl chamber is located in the cylinder head.
2. About 50% of air is transferred to the swirl chamber.
3. Air enters tangentially through the throat with a strong rotary movement.
4. After combustion in the swirl chamber the products rush out into the cylinder with higher velocities.
5. This causes heat losses to the walls of passage and it can be reduced by employing a heat insulated chamber.
6. This type is employed where fuel quality is difficult and reliability is more important than fuel economy.
7. Single hole of large diameter for nozzle is used in swirl combustion chamber.

Pre combustion Chamber:-



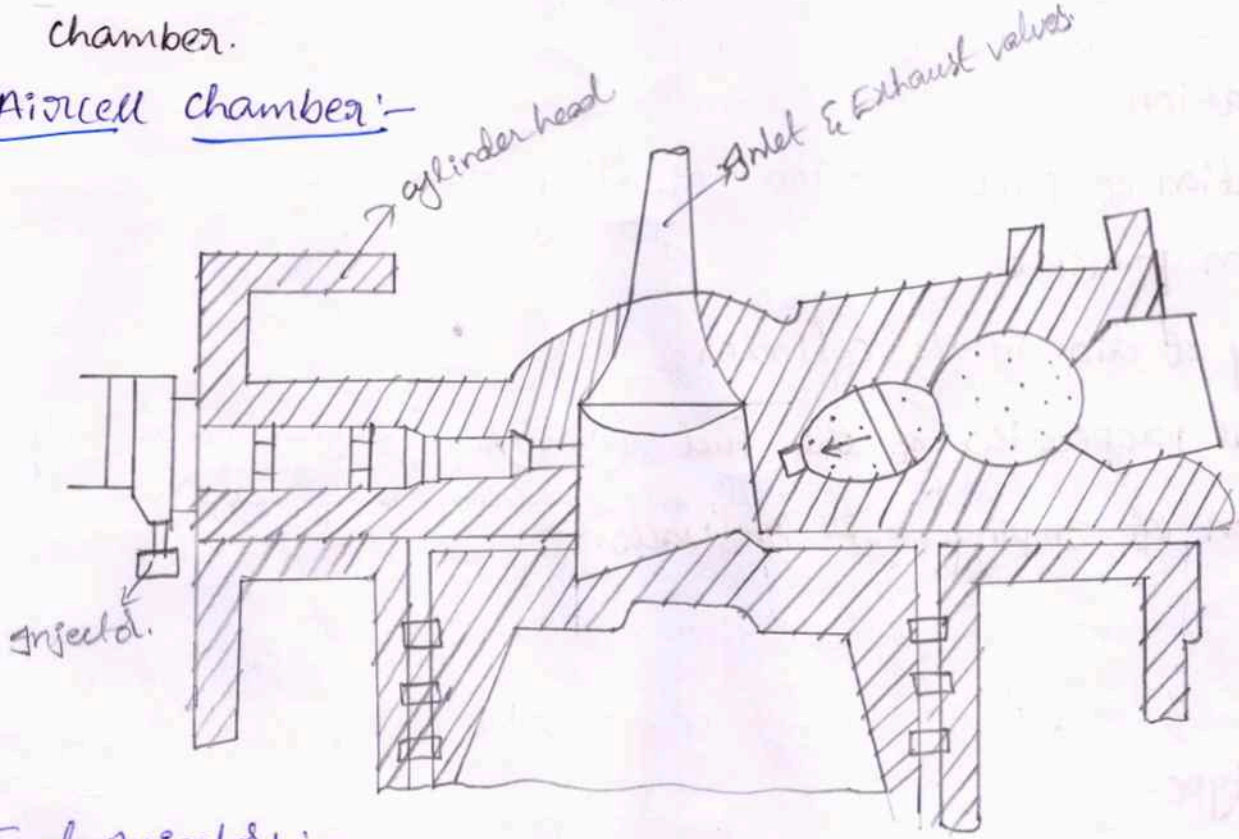
1. It consist of an anti-chamber connected to the main chamber through a number of small holes (Large passage than swirl chamber)
2. It has 40% of combustion space.
3. It creates strong secondary turbulance and distribute the flaming fuel droplets throughout the air in main combustion chamber
4. About 80% of energy is released in main combustion chamber.
5. Rate of pressure rise and maximum pressure is lower than

open type combustion chambers.

6. Initial shock is limited to pre combustion chamber only.

7. It has multi fuel capability because of temperature of pre-chamber.

Aircell chamber:-

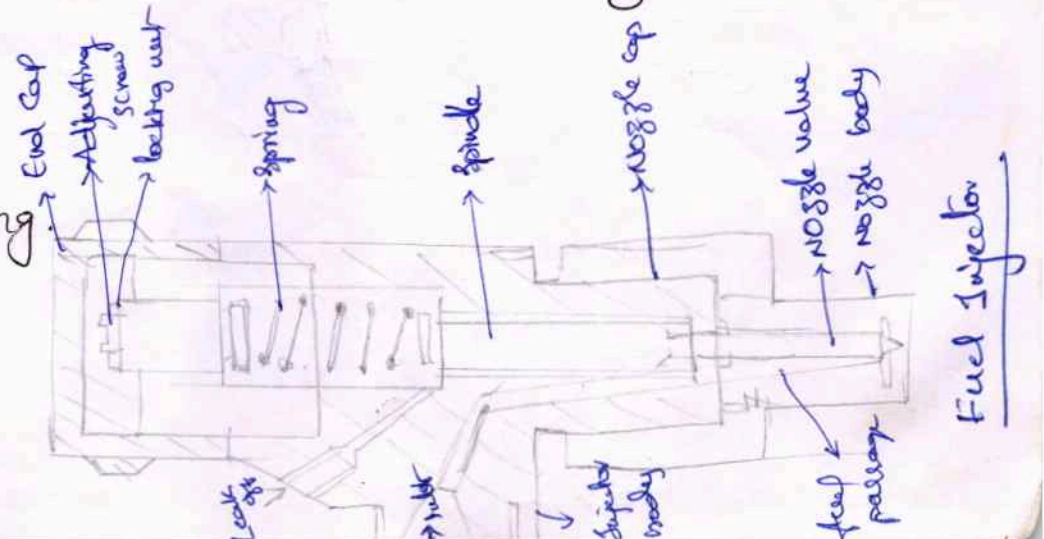


Fuel Injectors:-

Quick and complete combustion is ensured by a well designed fuel injector. By automizing the fuel into very fine droplets, it increases the surface area of the fuel droplets resulting in better mixing and subsequent combustion.

Automization is done by forcing the fuel ~~in~~ through a small orifice under high pressure. The injector assembly consist of mainly four parts.

1. needle valve.
2. compression spring.
3. Nozzle.
4. An injector body.



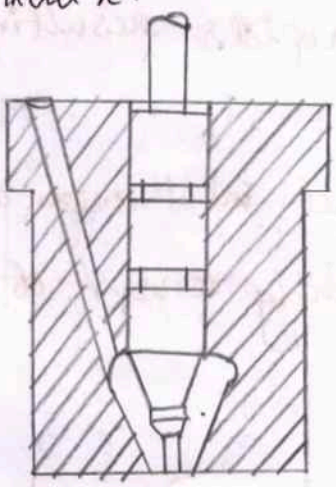
Nozzle:- Nozzle is the part of an injector through which the liquid fuel is sprayed into the combustion chamber.

Functions of nozzle:-

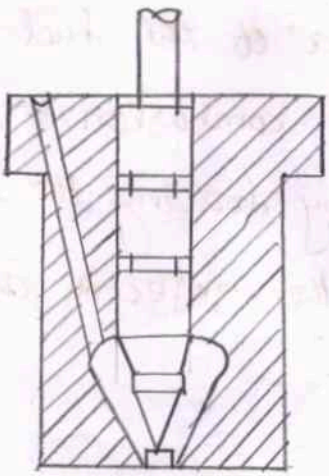
1. Automization
2. Distribution of fuel. Factors affecting this are
   i) Injection pressure.
   ii) density of air in the cylinder.
   iii) Physical properties of the fuel. (viscosity, vapour press, self ignition temp etc)
3. prevention of impingement on walls.
4. Mixing

Types of Nozzles:-

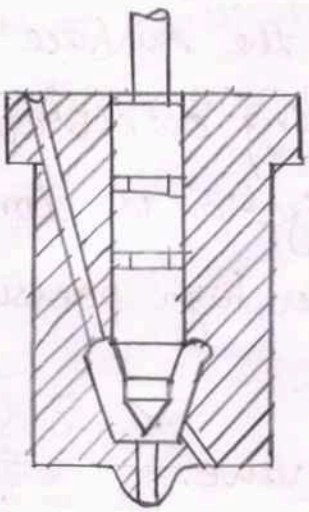
1. pintle type.
2. Single hole
3. Multi hole
4. Pintaux.



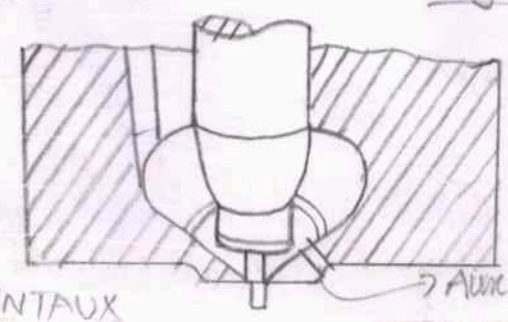
Pintle type



Single hole



Multi hole



PINTAUX

Pintle type Nozzle:- The stem of the Nozzle valve is extended to form a pin (or) pintle which protrudes through the mouth of nozzle. The size and shape of the pintle can be varied according to the requirement. It provides a spray operating at low injection pressures of 8 to 10 Mega pascal. The spray cone angle is generally 60°.

Advantage of this nozzle is that it avoids weak injection and dribbling.

It prevents carbon deposits on the nozzle hole.

Single hole Nozzle:- At the centre of the nozzle body there is a single hole which is closed by the Nozzle valve. The size of the hole is usually 0.2 mm. Injection pressure is around 8 to 10 mpa. Spray cone angle is about 15°.

Disadvantage:-

1. It tends to dripple.
2. Besides, the spray angle is too narrow to facilitate good mixing unless higher velocities are used.

Multi hole Nozzle:- It consists of no. of holes from 4 to 18 and size from 35 to 200 µm. The cone angle may be 20° upwords. Injection pressure is of 18 mpa.

Advantage of this type is the ability to distribute the fuel properly even with lower air motion.

Pintaux Nozzle:- It is similar to pintle type Nozzle which has an auxilary hole in the Nozzle body. It injects a small amount of fuel to the auxilary hole slightly before the main injection.

The Needle valve does not lift fully at low speeds and most of the fuel is injected through this auxiliary hole.

Advantage is better cold starting performance. (20-25°C lower than Multihole design)

Draw back is its injection characteristics are poorer than multi hole nozzle.

Important Qualities of fuel for C.I Engine:-

1. knocking characteristics. [It depends on ignition lag period. minimum ignition lag tends to minimum knocking]
2. Volatility [It should be high]
3. Starting characteristics. [It depends on volatility]
4. Smoking and odour [It depends on volatility]
5. Viscosity [It should be less under the lowest operating temp.]
6. corrosion and wear [It depends on sulphur, ash and residue in the fuel]
7. handling ease [The fuel should be liquid under all conditions and it should have high flash point and fire point]

Rating of C.I Engine fuels:-

Cetane Number is defined as percentage by volume of Normal Cetane in a mixture of Normal cetane and α-methyl Naphthalene which has the same ignition characteristics as the test fuel. When combustion is carried out in a standard Engine under specified Operating conditions.

The Reference fuels,

Normal cetane ($C_{16}H_{34}$) $\rightarrow$ 100.

α-methyl Napthalene ($C_{11}H_{10}$) → 0.

Standard operating characteristics are.

Engine speed 900 rpm.

Jocket water temp. is around 100°c.

Inlet air temp 65.5°c

Injection advance constant at 13° before TDC.

Ignition Delay 13°.

The cetane Number of unknown fuel can be estimated by noting the compression ratio at the above conditions.

The compression ratio is varied for each reference different blend to reach the standard ignition delay. and by interpolation of the compression ratios the cetane rating of unknown fuels is determined.

Rating of SI engine fuels:-

Octane number of fuel is defined as the percentage, by volume, of iso octane in a mixture of iso octane & normal heptane, which exactly matches the knocking intensity of fuel in the std. engine under a set of std. operating conditions.

→ Reference fuels:- Iso octane – 100 (C_8H_{18}) ; normal heptane – 0 (C_7H_{16})

→ Addition of certain compounds (tetra ethyl lead) to iso octane produce fuel of greater antiknock quality (above 100 ON)

→ Becoz of non linear variation, a new scale were derived which expresses the approximate relative engine performance number (PN). ON above 100 can be obtained by $ON\ (>100) = 100 + \frac{28.28A}{1 + 0.736A + \sqrt{1 + 0.736A - 0.035216A^2}}$

where A is TEL in ml/gal of fuel.

(or) from performance number (PN),

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

TESTING AND PERFORMENCE

Engine Performance Parameters:

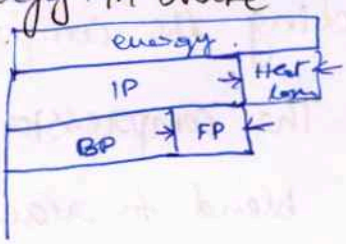
Indicated Thermal efficiency:- It is the ratio of ~~energy~~ the indicated power to the fuel energy consumed.

$$\eta_{ith} = \frac{ip}{\text{fuel consumed}}$$

$$= \frac{ip}{\text{mass of fuel/sec} \times \text{calorific value}}$$

FP = IP - BP

Brake thermal efficiency:- It is the ratio of ~~energy~~ brake power to fuel energy consumed.



$$\eta_{bth} = \frac{bp}{\text{fuel consumed}}$$

$$= \frac{bp}{\text{mass of fuel/sec} \times \text{calorific value}}$$

Mechanical efficiency:- It is defined as ratio of Brake power to indicated power.

$$\eta_m = \frac{bp}{ip} \quad (\text{or}) \quad \eta_m = \frac{\eta_{bth}}{\eta_{ith}}$$

[fp = frictional power]

$$ip = bp + fp$$

$$fp = ip - bp.$$

Volumetric efficiency:- (Breathing Capability of engine.) Volumetric efficiency is defined as the ratio of actual volume flow rate of air into the intake system to the rate at which the volume is displaced by the system.

$$\eta_v = \frac{ma}{\rho_a V_d}$$

V_d = displacement vol

ρ_a = density of air

$$\eta_v = \frac{m_{act}}{\rho_a V_d n} = \frac{m_{act}}{m_{th}}$$

Where N - speed of engine in rpm.

$n = N/2 \rightarrow$ 4-stroke

$N \rightarrow$ 2-stroke

η_{vol} = 70 - 80%.

Supercharged $\rightarrow \eta_v$ = 100%.

m_{act} - Measured quantity of air.

m_{th} - mass of air computed from geometry of the cylinder, No of cylinders, speed of engine and density of surrounding air.

Relative efficiency (or) Efficiency Ratio:- It is defined as the ratio of brake thermal efficiency of an actual cycle to that of an ideal cycle. The efficiency ratio is very much useful citeria which indicates the degree of development of engine.

$$\therefore \eta_{rel} = \frac{\text{Actual brake thermal efficiency } (\eta_{th})}{\text{Air standard } \eta_{th}}$$

Mean effective pressure:-

It is the average pressure inside the cylinder of an I.C Engine based on measured power output. For any particular engine operated at given speed and power o/p, there will be a specific indicated mean effective pressure (IMEP) and a corresponding Brake mean effective pressure (BMEP).

[M.e.p => hypothetical press. which is thought to be acting on the piston throughout the power stroke.]

$$i_p = \frac{P_{im} LANK}{60 \times 1000}$$

where, P_{im} - Indicated mean effective pressure in N/m^2

L - Length of stroke in m.

A - Area of piston in m^2

N - Speed in rpm [N = n for 2-stroke. N = n/2 for 4-stroke]

K - No. of cylinders. working stroke.

n → No. of cylinders.

K = 1 (2 stroke)

K = 1/2 (4 stroke)

$$P_{im} = imep = \frac{i_p \times 60{,}000}{LANK\,\eta}$$

$$P_{bm} = bmep = \frac{i_p \times 60{,}000}{LANK\,\eta}$$

K = no. of cylinder.

N = N = 2 stroke

N = N/2 = 4 stroke.

$imep = bmep + fmep$

$$\frac{bmep}{imep} = \frac{Bp}{ip} = \eta_{mech}$$

Brake mean effective pressure is very much useful in comparing the engines (or) in establishing engine operating limits.

Mean speed of piston:-

$$Sp = 2LN.$$

Specific fuel consumption:- It is defined as the ratio of fuel consumption per unit time to the unit power. (develop)

$$SFC = \frac{\text{fuel consumed/sec}}{\text{power}} = \frac{m_f}{P}$$

It is an important parameter that reflects the performance of the engine. It is inversely proportional to thermal efficiency of engine.

Air fuel ratio (or) fuel air ratio:-

The ratio of actual fuel air ratio by stiochiometric fuel ratio. It is known as equalence air fuel ratio (15:1)

Equivalence AF ratio = $\frac{\text{Actual fuel air ratio}}{\text{Stoichiometric fuel ratio}}$

Caldific value of fuel:- It is the thermal energy released per unit quantity of fuel.

Indicated power:- Energy produced (or) energy developed in chamber.

$$ip = \frac{Pm\ LANK}{60{,}000}\ kW.$$

Brake power:- The power developed by engine at o/p of shaft

$$Bp = \frac{2\pi NT}{60{,}000}\ kW.$$

$T = FXR.$

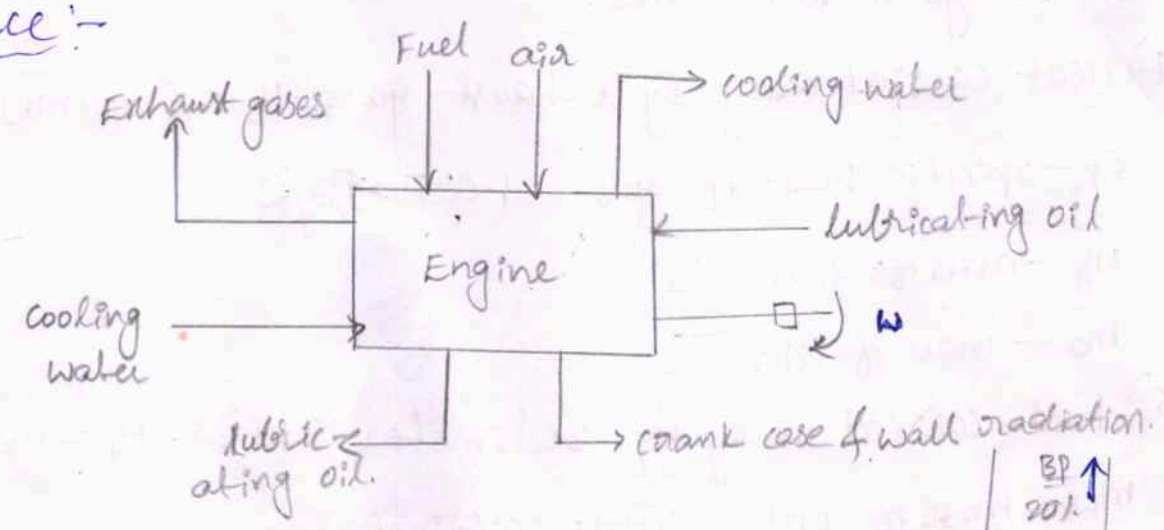
where R - length of moment arm in mtr.

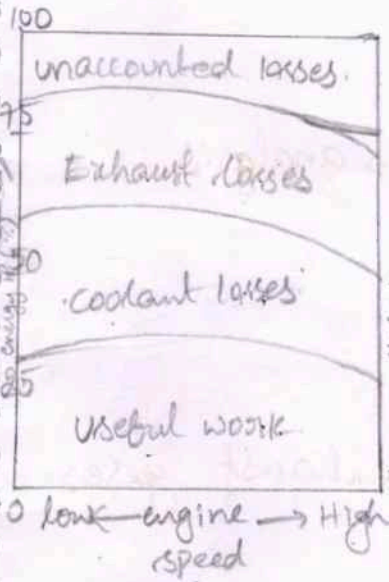
$$B_p = \frac{\pi (W-S)(D+d)N}{60{,}000}$$

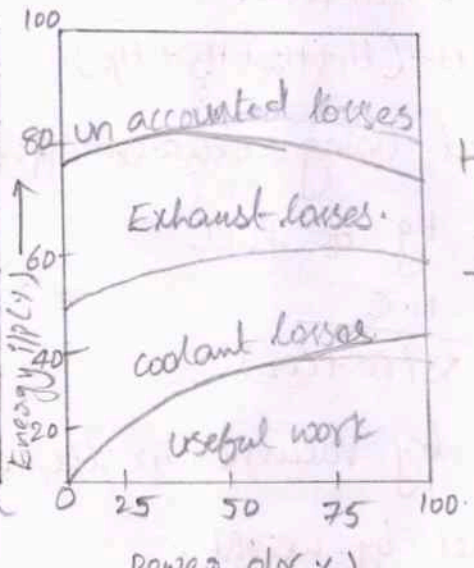
D = drum dia, d = rope dia.
W = load applied.
S - spring balance reading.

Where $T = (W-S)\left(\frac{D+d}{2}\right)$

Heat Balance :-

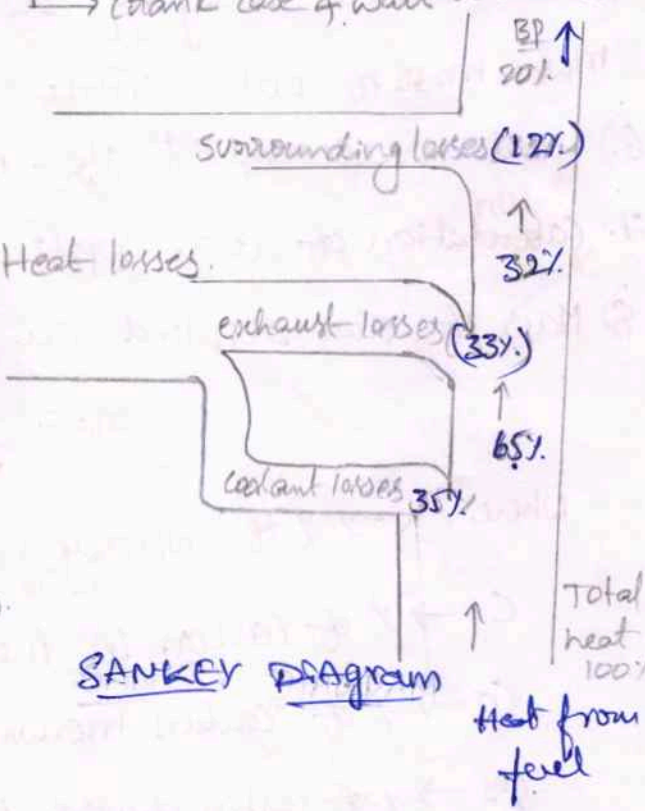






Heat Balance sheet fd S.I (HBS)

HB fd CI



SANKEY Diagram

Heat Balance sheets calculations :-

1. Total heat input to the engine $H = m_f \times CV$

   where m_f = mass of fuel supplied.

   CV = calorific value of the fuel.

2. Heat equivalent to Brake power $H_1 = \frac{2\pi NT}{60} = \frac{\pi (D+d)(W-S)N}{60} = \frac{V \times I}{\eta_g \times 100}$

   V - voltage, T - Torque.

   I - current

3. Heat carried away by the engine Jacket cooling water $H_2 = m_w C_{pw} \Delta T$

m_w – mass of water

C_{pw} – specific heat of water.

ΔT – change in temperature.

4.) Heat carried away by exhaust gases. $H_3 = (m_f + m_a) C_{pg} \cdot \Delta T$

C_{pg} – specific heat of gas – 1.005 kJ/kgK.

m_f – mass of fuel kg

m_a – mass of air kg.

5.) heat carried away by calorimeter water $H_4 = m_{w_1} C_{pw} \Delta T$

m_{w_1} – mass of water in the calorimeter.

6.) Heat unaccounted $H_5 = H - (H_1 + H_2 + H_3 + H_4)$

7.) Calculation of air supplied using exhaust gas analysis.

8) Mass of air supplied per kg of fuel

$$m_a = \frac{N \cdot C}{33(CO + CO_2)}$$

Where, N → % of Nitrogen by volume in the exhaust gases

C → % of carbon in fuel by weight.

CO → % of carbon monoxide by volume in the exhaust gases.

CO_2 → % of carbon dioxide by volume in the exhaust gases.

Measurement of cylinder pressure:-

Determining compression pressures involves using the compression gauge. Remove the spark plug in S.I Engine (or) fuel injector in C.I Engine and hold the gauge tightly in the hole. Crank the Engine in normal manner and observe the gauge reading. The normal pressure reading varies with the type and size of the Engine.

If the pressure value is below that of manufactures value, it indicates that the wear of the cylinder walls and rings or is greater or defects in the gasket and seals & inlet & exhaust valves.

The cylinder pressure during a cycle is usually measured with a piezo electric pressure transduser. The transducer is screwed into the cylinder head (where possible ~~causing~~) until the diaphragm is flush with the walls of combustion chamber. In case of S.I Engines, a spark plug adopted can be fitted with a miniature transduser incaporated with in it.

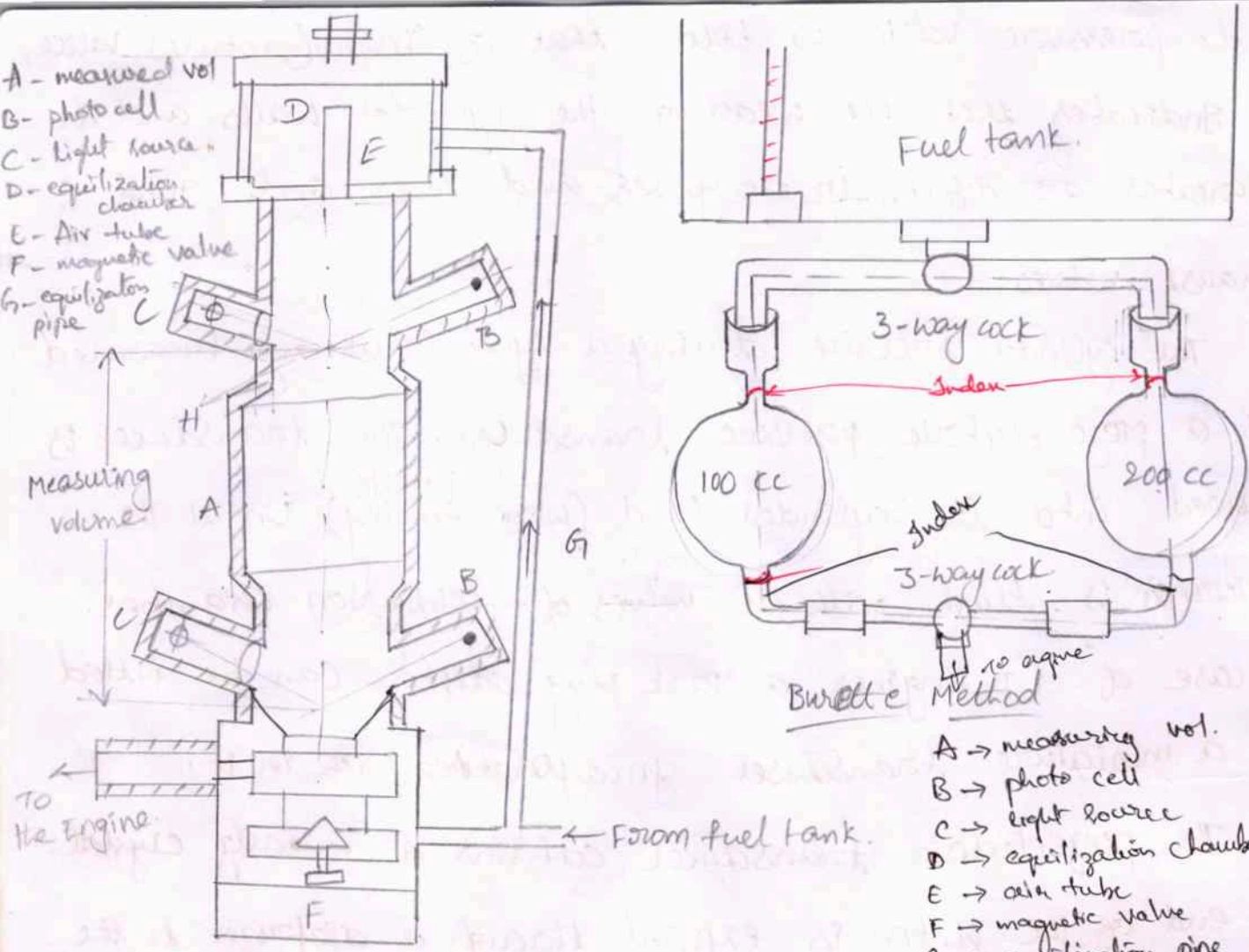
The piezoelectric transducer contains a quearty crystal. one end of the piston is exposed through a diaphragm to the cylinder pressure. As the cylinder pressure increases, the diaphragm is compressed. this generates an electric charge which is proportional to the pressure acting. This unique property is called as piezo electric effect. A charge Amplifier is used to produce an output voltage proportional to the electric charge.

Measurement of fuel Consumption:-

There are two types of fuel measurement devices.

1. Volumetric type.
2. gravimetric type.

→ Due to heat vapour bubbles are formed & back flow takes place
→ Any swirl in flow
→ Temp (-10c to 70c) changes
→ Light beam affected by colour of fuel
→ water hammer effect by float

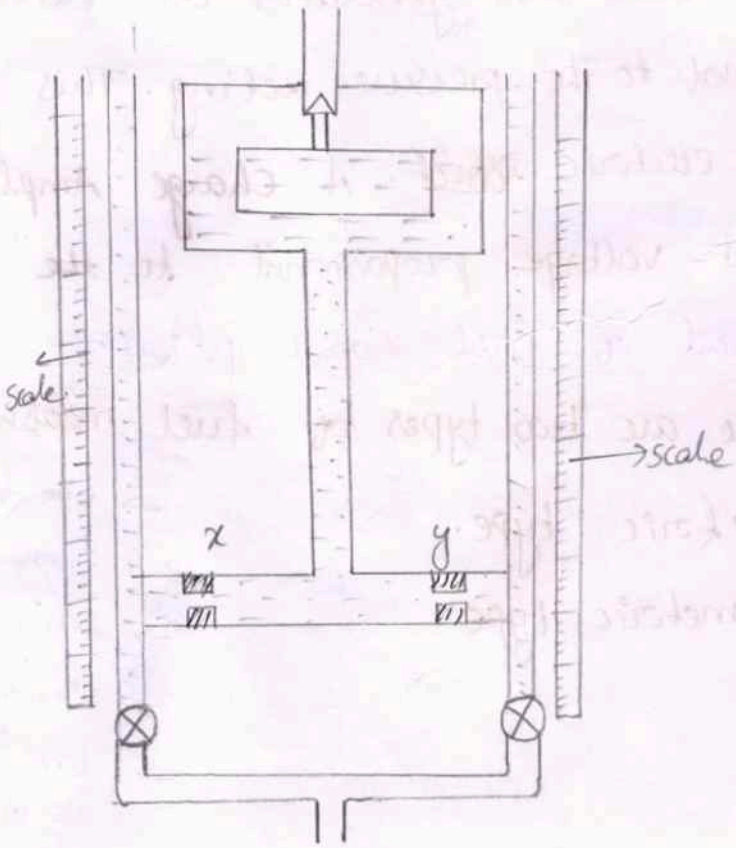


Burette Method

A → measuring vol.
B → photo cell
C → light source
D → equilization chamber
E → air tube
F → magnetic valve
G → equilization pipe

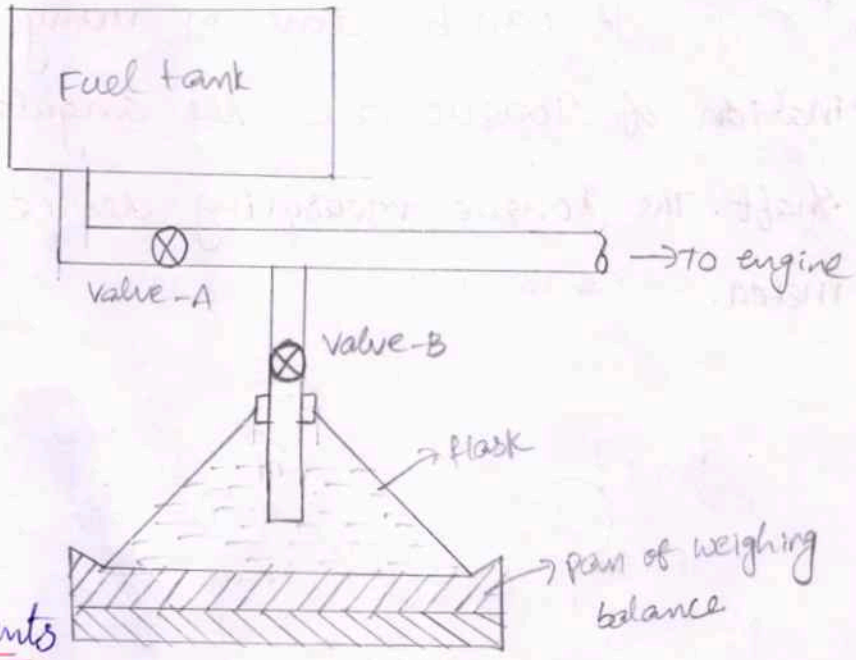
Automatic Burette type Fuel Measurement Device

Orifice Flow Meter :-

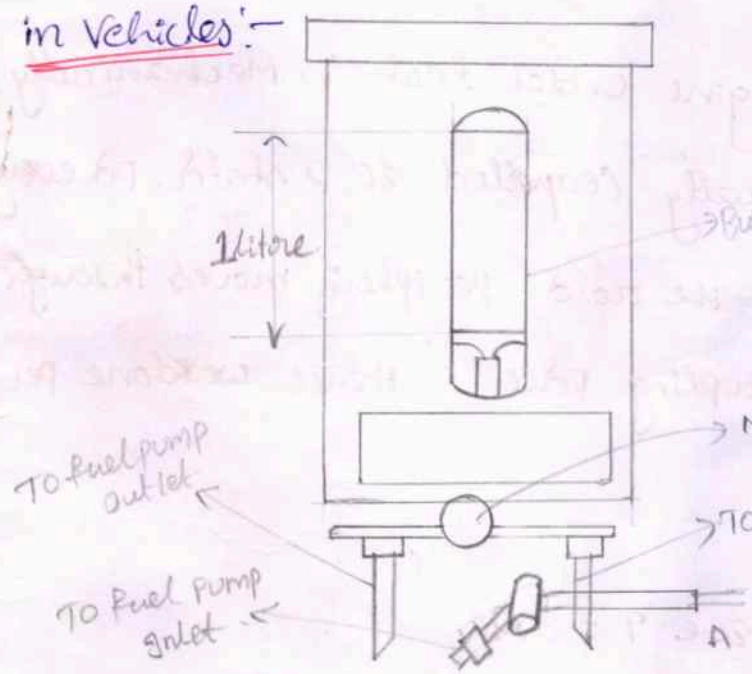


Gravimetric type:-

Gravimetric measurement of fuel flow.



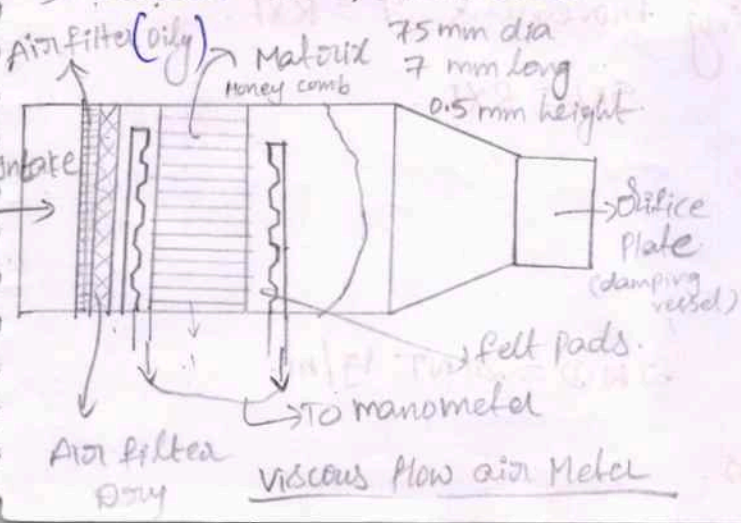
Fuel consumption measurements in Vehicles:-



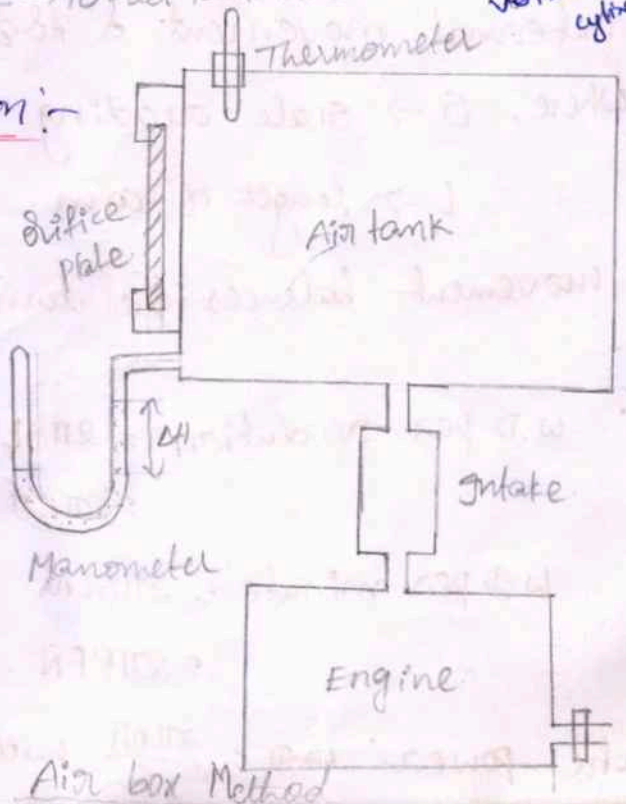
Measurement of air consumption:-

i) Air box method.

ii) Viscous flow Air meter.



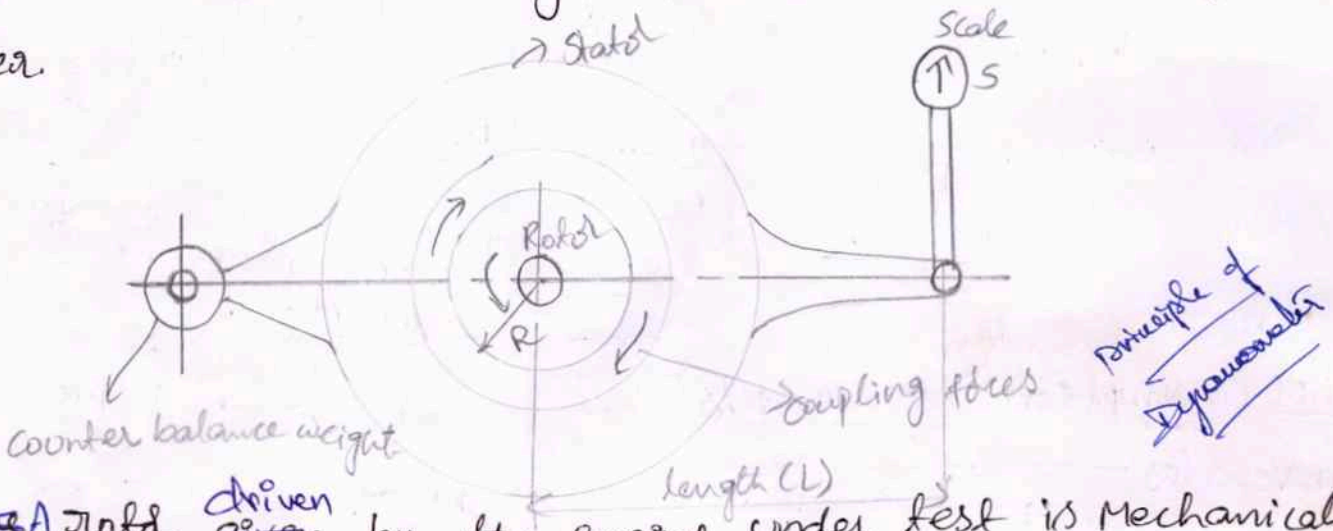
Viscous flow air Meter



Air box Method

Measurement of Brake power:-

It can be done in many ways. It involves determination of Torque and the angular speed of the output shaft. The torque measuring device is called as a dynamometer.



A rotor driven given by the engine under test is mechanically, hydraulically or electromagnetically coupled to a stator. For every revolution of the crank shaft, the rotor periphery moves through a distance $2\pi r$ against the coupling force F. Hence workdone per revolution is $2\pi RXF$

$\therefore W = 2\pi RXF$

The external movement of torque $T = SXL$.

where, S → scale reading

L → length of arm.

This movement balances the turning movement $T = RXF$.

$SXL = RXF$.

$\therefore$ W.D per revolution $= 2\pi SL$

$= 2\pi RF$

W.D per minute $= 2\pi SLN$

$= 2\pi RFN \quad \Rightarrow W.D = 2\pi NT$ J/m.

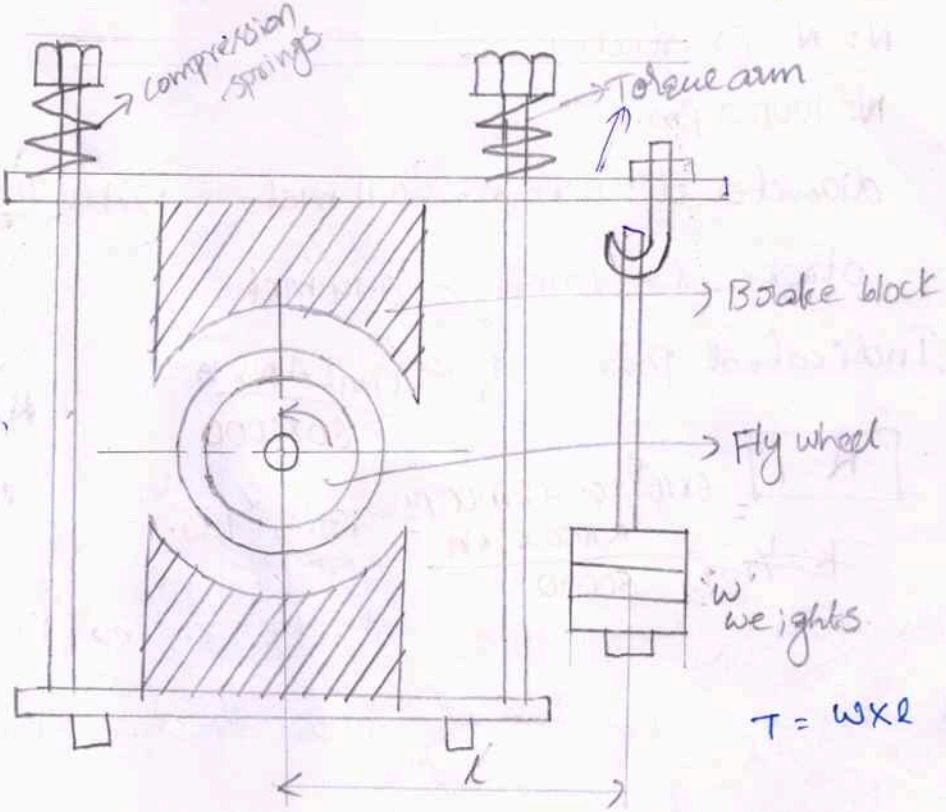
Brake power: ~~W.D~~ $= \frac{2\pi NT}{60}$ watts.

Types of measurement of Brake power:-

→ Absorption type
→ Transmission type (torquemeters)

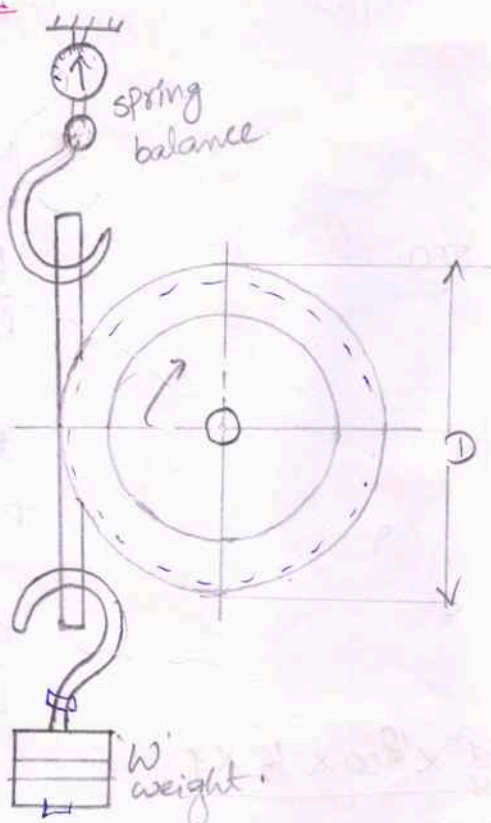
1) Prony Brake:-

(2) Rope Brake
(3) Hydraulic Dynamometers
(4) Eddy current "
(5) Swinging field DC "
(6) Fan dynamometers
(7) Transmission "
(8) Chassis "



$T = W \times l$

ii) Rope Brake:-



$$B.P = \frac{\pi N (W-S) D \pi d}{60}$$

(1) A Two-stroke cycle I.C Engine has a mean effective pressure of 6 bar. The speed of the Engine is 1000 r.p.m. If the diameter of the piston and stroke are 110 mm and 140 mm respectively find the indicated power developed?

1 bar = 10^5 N/m² = Pa
= 10^2 KPa

A) $P = 6 \text{ bar} = 6 \times 10^5 \text{ N/m}^2$

$N = 1000$ rpm

$N = N$ (2 stroke).

$N = 1000$ r.p.m.

diameter $d = 110$ mm. $= 0.11$ met $\Rightarrow A = \frac{\pi (0.11)^2}{4} = 0.0094\ m^2$

stroke, $l = 140$ mm. $= 0.14$ met

Indicated power $i_p = \frac{P_m L A N K}{60 \times 1000}$

$N = N$ (2 stroke)

$N = N/2$ (4 stroke)

$K = 1$

$= \frac{6 \times 10^5 \times 0.14 \times 0.0094 \times 1000 \times 1 \times 1}{60000} = 13.3$ kW.

2) The four cylinder four stroke petrol Engine developes 14.4 kw at 1000 r.p.m the mean effective press. is 5.5 bar. calculate the bore and stroke of the engine if the length of stroke is 1.5 times the bore.

A) $K = 4$

$N = 1000$ r.p.m $\Rightarrow N = N/2 = 500$.

$l = 1.5d$

$P_m = 5.5$ bar. $= 5.5 \times 10^5\ N/m^2$

$i_p = 14.4$ kw.

$i_p = \frac{P_m L A N K}{60,000}$

$14.4 = \frac{5.5 \times 1.5d \times \frac{\pi d^2}{4} \times 1000 \times \frac{1}{2} \times 4}{60,000}$.

$14.4 = \frac{6476.25 d^3}{60,000}$

$0.1079\, d^3 = 14.4$

$d^3 = 133.41$

$d = 87.9$ mm

$\Rightarrow l = 1.5d = 131.85$

$K = 4$

$N = N/2 = 500$ rpm

$L = 1.5d$

$P_m = 5.5$ bar $= 5.5 \times 10^5\ N/m^2$

$P = 14.4$ KW

$P = \frac{P_m L A N K}{60000}$

$d = 87.9$ mm

$L = 131.$ mm.

$d^3 = 400.316$

$d = 15.87$ mm.

$l = 1.5d$

$l = 23.805$ mm

3) A Turbocharged 6-cylinder diesel engine has the following performance details.

i) workdone during compression and Expansion. = 820 kw.

ii) Workdone during Intake & Exhaust = 50 kw

iii) Rubbing Foriction in the Engine = 150 kw.

iv) Net workdone by the turbine = 40 kw.

If the brake mean effective pressure is 0.6 Mpa determine bore and stroke, of the engine taking the ratio of bore to stroke as one and engine speed as 1000 r.p.m.

A) Net work available

Brake power = 820 − (150 + 50 + 40)

= 820 − 240 = 580. kW

K = 6

$P_{bm} = 0.6 \times 10^6$

$P_m = 0.6 \times 10^6 \ N/m^2$

K = 6

N = 1000, $N = N/2$

N = 500.

$\frac{d}{l} = 1 \Rightarrow d = l$

$l = d$

$d = 290.8$ mm.

$$\text{Brake power} = \frac{P_{bm} LANK}{60{,}000}$$

$$580 = \frac{0.6 \times 10^6 \times d \times \frac{\pi d^2}{4} \times 0.5 \times 6 \times 1000}{60{,}000}$$

$$580 = 11775\, d^3$$

$$d^3 = 0.04925$$

$$d = 360 \text{ mm.}$$

290.8 mm.

4) A large 4-stroke cycle diesel engine runs at 2000 r.p.m. The engine has a displacement of 25 lit and Brake mep is of 0.6 Mega N/m^2. It consumes 0.018 kg/sec of fuel. (Calorific value of the fuel is 42000 kJ/kg) Determine Brake power and Brake thermal efficiency ?

A) $A \times l = 25$ lit

$= 25/1000$ m³

$V = A \times l = 0.025$ m³.

$K = 1/2$, $N = 2000 =$ ~~N~~

$n = 1$ (assume).

$P_{bm} = 0.6 \times 10^6$ N/m²

Calolific value (CV) = 42000 kJ/kg.

$N = 2000$ rpm

$Vd = 25$ ltr $= 0.025$ m³

$P_m = 0.6 \times 10^6$ N/m²

$m_f = 0.018$ kg/s.

$CV = 42000$ kJ/kg.

$K = 1$

$$\text{Brake power} = \frac{(0.6 \times 10^6) \times 0.025 \times 2000 \times 0.5 \times 1}{60{,}000}$$

B.P = ~~0~~250 kW

B: $\frac{P_{bm} \times A \times L \times n/2 \times K}{60}$

$m_f = 0.018$ kg/s.

$$\eta_{bth} = \frac{B.P}{m_f \times CV}$$

$$= \frac{0250}{0.018 \times 42000}$$

$\eta_{bth} =$ ~~0.165~~ 0.3306

$=$ ~~16.53~~ 33.06 %.

5) Following observations were recorded during a test on a single cylinder Oil Engine. Bore = 300 mm, Stroke = 450 mm, speed = 300 rpm. Indicated Mean effective press is 6 bar. Net brake load (W-S) 1.5 ~~kilowatts~~ Newtons. Brake drum dia = 1.6 met., Brake rope dia = 2 cm. calculate

i) Indicated power ii) Brake power iii) Mechanical efficiency.

A) Indicated power $= \frac{P_{mf} ANkn}{60{,}000}$, $K = 1/2$ (assume Four stroke)

$l = 450 = 0.45$ met $= \cancel{6 \times 450 \times (6 \times 10^3)}$ $= (6 \times 10^3) \times (0.45) \times \frac{\pi (0.3)^2}{4} \times 300 \times 1 \times 1/2$

$d = 300$ mm

$= 0.3$ met

$i_p = 0.477 = 47.7$ kw $\quad 60{,}000$

Brake power $= \frac{\pi (W-S)(D+d) N}{60000}$

$= \frac{3.14 (1.5) \times (1.8 + 0.02) \times 300}{60000}$

$= 42.861$ kW

Mechanical efficiency $= \frac{B.P}{I.P} = \frac{42.861}{0.477}$

$\eta_m = 89.85\%$

6) The power output of an engine is measured by a rope brake dynamometer. The dia of Brake pulley is 700mm and rope dia is 25 mm. The load on the tight side of the rope is 50 kg (W) mass and spring balance reads 50 N (S). The Engine running at 900 r.p.m. consumes fuel of calorific value of 44,000 kJ/kg (cv). at the rate 4 kg/hour (m_f). assume $g = 9.81$ m/s². Calculate i) Brake specific fuel consumption ii) Brake thermal efficiency.

A) Brake power $= \frac{\pi (W-S)(D+d) N}{60}$, $D = 700$mm, $d = 25$ mm

$= 0.7$ $d = 0.025$ m

$W = 50$ kgs

$= 50 \times 9.81 = 490.5$ N

$B.P = \frac{3.14 (490.5 - 50)(0.7 + 0.025) \times 900}{60000}$

$= 15049.60$ W $= 15.04$ kW.

i) Brake specific fuel consumption $= \frac{m_f}{B.P} = \frac{4}{15.05} = 0.266$ kg/kWhr

$m_f = 4$ kg/hr

$= \frac{4}{3600}$ kg/s. $= 1.11 \times 10^{-3}$ kg/sec

$= \frac{1.11 \times 10^{-3}}{15.04}$

$= 7.38 \times 10^{-5}$

ii) $\eta_{bth} = \frac{B.P}{m_f \times C_v} = \frac{15.04}{(1.11 \times 10^{-3}) \times (44,000)}$

$= 0.3079 = 30.79$

7) A six-cylinder 4-stroke petrol engine having a bore 90 mm and stroke of 100 mm has a compression ratio of 7. The relative efficiency with reference to the indicated thermal efficiency is 55%, when the indicated specific fuel consumption is 0.3 kg/kwh. Estimate the calorific value of fuel and fuel consumption in kg/hr given that the indicated mean effective pressure is 8.5 bar and speed is 2500 R.P.M.

A) Given, $n=6$, $k=\frac{1}{2}$

$D = 90\times10^{-3}$ m.

$L = 0.1$ m.

Compression ratio $r=7$

$\eta_{rela} = 55\% = 0.55$

Specific fuel consumption $= 0.3$ kg/kwh.

$P_{im} = 8.5\times10^{5}$ N/m²

$N = 2500$ r.p.m.

$C_v = ?$, $m_f = ?$

$K = 6$
$N = \frac{N}{2}$ (4-stroke)
$d = 90$ mm
$L = 100$ mm
$r = 7$
$\eta_{rel} = 0.55$
$isfc = 0.3$ kg/kwhr
$P_{im} = 8.5\times10^{5}$ N/m²
$N = \frac{2500}{2} = 1250$ rpm.

$$\eta_{ith} = \frac{ip}{m_f \times CV}$$

$$ip = \frac{nP_mLANK}{60000} = \frac{6\times(8.5\times10^{5})\times(0.1)\times\frac{\pi(90\times10^{-3})^2}{4}\times2500\times0.5}{60000}$$

$= 67.55$ kw.

$\eta_{ith} = \eta_{rel} \times \eta_{air\ std.}$

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

$$= 1-\frac{1}{(7)^{1.4-1}} = 0.5408$$

$\eta_{ith} = 0.55\times0.5408$

$\eta_{ith} = 0.2974$

$$\eta_{ith} = \frac{ip}{m_f \times CV}$$

$$0.297 = \frac{67.55}{\frac{0.3}{3600} \times CV}$$

$$I_p = \frac{m_f}{Sfc} \qquad ISFC = \frac{m_f}{IP} \; ; \; BSFC = \frac{m_f}{BP}$$

$$m_f = I.P \times Sfc$$

$$= 0.2974 \; 67.5 \times 0.3$$

$$= 20.27 \text{ kg/hr.}$$

$$0.29 = \frac{67.5}{\frac{20.27}{3600} \times CV}$$

$$C_V = 40338.48$$

8) A six-cylinder, four-stroke S.I Engine having a piston displacement of 700 cm^3 per cylinder develop 78 kW. at 3200 r.p.m. and consume 27 kg of petrol per hour. the caldific value of petrol is 44 Mega J/kg estimate i) volumetric efficiency of the engine. If air fuel ratio is 12. and intake air is that 0.9 bar and 32°C. ii) Brake thermal efficiency iii) Brake Torque Fd air

$R = 0.287$ kJ/kgk.

$k = 6$

$N = \frac{N}{2} = 1600$

$A \times L = 700 \text{ cm}^3$

$P = 78$ kW

A) Given that,

$n = 6$, $k = \frac{1}{2}$ (4-stroke)

Displacement $= 700 \text{ cm}^3 = 700 \times 10^{-6} \text{ m}^3$.

B.P = 78 kW (or) kJ/sec.

N = 3200 r.p.m, $m_f = 27$ kg/hr $= \frac{27}{3600}$ kg/sec.

CV = 44 Mega J/kg

= 44,000 kJ/kg.

A/F = 12. , $P_1 = 0.9$ bar $= 0.9 \times 10^2 = 90 \text{ kN/m}^2$

T = 32°C = 32 + 273 = 305 K , R = 0.287 kJ/kgk.

i) volumetric efficiency $\eta_v = \frac{\text{vol. of intake air}}{\text{swept volume.}}$

ii) $\eta_{bth} = \frac{BP}{m_f \times C_V}$.

iii) $BP = \frac{2\pi NT}{60}$

vol. of intake air,

$$P_a V_a = m_a R_a T_a$$

$$A/F = \frac{m_a}{m_f}$$

$$12 = \frac{m_a}{27}$$

$$m_a = 324 \text{ kg/hr}.$$

$$P_a V_a = m_a R_a T_a$$

$$90 \times V_a = 324 \times 0.287 \times 305$$

$$V_a = \frac{28361.34}{90}$$

$$V_a = 315.126 \text{ m}^3/\text{hr}$$

[four stroke – N/2]

Swept volume = Piston displacement/Cylinder $\times N/2 \times n$

$$= 700 \times 10^{-6} \times \frac{3200}{2} \times 6$$

$$= 6.72 \text{ m}^3/\text{min.} \times 60$$

$$= 403.2 \text{ m}^3/\text{hr}.$$

i) $\eta_v = \frac{315.126}{403.2} = 0.781 = 78.15\%$ ✓

ii) $\eta_{bth} = \frac{78}{\frac{27}{3600} \times 44000} \times 100$

$\eta_{bth} = \frac{BP}{m_f \times C_V}$

$\eta_{bth} = 23.63\%$ ✓

iii) $B.P = \frac{2\pi NT}{60} \Rightarrow \frac{2 \times 3.14 \times 3200 \times T}{60} = 78$

$$T = \frac{78 \times 60}{2 \times 3.14 \times 3200}$$

$$T = 0.2328 \text{ KNm}$$

9) A Six-cylinder 4-stroke gas Engine with a stroke volume 1.75 lit. develops 26.3 kw. at 504 r.p.m the mean effective pressure is 6 bar. Find the average no. of times each cylinder misfires in 1 minute.

A) $n = 6$, $K = \frac{1}{2}$

$V_s = A \times L = 1.75 \text{ lit} = 1.75 \times 10^{-3} \text{ m}^3$.

$ip = 26.3$ kw, $N = 504$ r.p.m, ~~mep = 6 bar~~

$P_{im} = 6 \text{ bar} = 6 \times 10^5 \text{ N/m}^2$

$$ip = \frac{n P_{im} L A N K}{60000}$$

$$26.3 = \frac{6 \times (6 \times 10^5) \times (1.75 \times 10^{-3}) \times N \times \frac{1}{2}}{60,000}$$

$$N = \frac{60,000 \times 26.3}{6 \times (6 \times 10^5) \times (1.75 \times 10^{-3}) \times 0.5}$$

$$N = 500.95 \text{ r.p.m}$$

$$IP = \frac{P_m L A N K}{60000}$$

$$26.3 = \frac{6 \times 10^5 \times 1.75 \times 10^{-3} \times N \times 6}{60000 \times 2}$$

$N = 500.95$ rpm.

Expected No. of fires in one minute $= \frac{504}{2} \times 6 = 1512$

Actual No. of fires in one minute $= \frac{500}{2} \times 6 = 1500$.

No. of misfires/min $= 1512 - 1500$

$= 12$

No. of misfires/min in one cylinder $= \frac{12}{6} = 2$

10) A Six-cylinder 4-stroke C.I Engine is tested against a water brake dynamometer for which BP $= \frac{WN}{7} \times 10^{-3}$ kw. where 'W' is the brake load in Newtons and N is the speed of the Engine in r.p.m. The air consumption was measured by means of a sharp edged

dilfice. During the test the following observations were taken. Bore = 10 cm, Stroke = 14 cm, N = 2500 r.p.m. Brake load = 480 N. Barometer reading = 76 cm of Hg, dilfice diameter = 3.3 cm. coeffic-ient of discharge of dilfice = 0.62, pressure drop across the dilfice = 14 cm of Hg. Room temperature = 25°C. Fuel consumption = 0.32 kg/min. Calculate the following.

i) volumetric efficiency ii) Brake Mean effective pressure iii) The engine torque and iv) Brake specific fuel consumption.

A) given that,

$n = 6$, $k = 1/2$ (4-stroke)

$K = 6$
$N = N/2$ (4 stroke)

$BP = \frac{WN}{17} \times 10^{-3}$ kW, $d = 10\text{cm} = 0.1$ met.

$l = 14\text{cm} = 0.14$ met.

$N = 2500$ r.p.m., Brake load $= 480$ N.

Barometer reading = 76 cm of Hg. = 760mm of Hg = 101.34 kN/m²

dilfice diameter = 3.3 cm = 0.033 met.

i) $\eta_{vol} = \frac{Va}{Vs}$,

$Va = C_d \cdot \frac{\pi}{4} d_o^2 \sqrt{2gha}$

$P = \rho_a g h_a$

$[P_a = \rho_a R_a T]$

$P = 14$ cm of Hg

$\rho = \frac{P_a}{R_a T} = \frac{101.34}{0.287 \times 298} = 1.184$ kg/m³

$P = \rho_a g h_a$.

press. across dilfice $\Rightarrow \frac{101.34}{1.184 \times 9.81} = h_a$

$h_a = 8.7248$

$P = 14$ cm of Hg

$= \frac{14}{100}$ m of Hg

$= \frac{14}{100} \times 13.6 \times 9.81 \times 10^3$

$= 18.678$ kN/m^2 $= 18678.24$ N/m^2

18665 N/m^2

1 mm of Hg = 133.322 N/m^2

1 cm of Hg = 1333.22 N/m^2

$P = 1.184 \times 9.81 \times ha$ $\quad [\Delta P = \rho g h a]$

$ha = \frac{18678}{1.184 \times 9.81}$

$ha = 1608.08$ m.

$Va = Cd \times \frac{\pi}{4} d_0^2 \times \sqrt{2gha}$

$Va = 0.62 \times \frac{\pi}{4} (0.033)^2 \times \sqrt{2 \times 9.81 \times 1608.08}$

$Va = 0.0941$ m^3/s

$Vs = \frac{\pi}{4} d^2 \times L \times \frac{N}{2} \times \frac{K}{60} = \frac{\pi}{4} (0.1)^2 \times 0.14 \times \frac{2500}{2 \times 60} \times 6$

$Vs = 0.137$ m^3/s.

$\eta_{vol} = \frac{Va}{Vs} = \frac{0.0941}{0.137} = 68.6\%$

(ii) The brake mean effective pressure P_{bmep}

$BP = \frac{\pi N}{17} \times 10^{-3}$ KW $= \frac{480 \times 2500}{17} \times 10^{-3} = 70.588$ KW

$BP = \frac{P_{bmep} L A N K}{60} \Rightarrow P_{bmep} = \frac{60 \times 2 \times 70.588}{0.14 \times \frac{\pi}{4}(0.1)^2 \times 2500 \times 6} = 513.57$ kN/m^2

(iii) Engine Torque T: $BP = \frac{2\pi NT}{60} \Rightarrow T = 269.63$ N-m

(iv) Brake Sp. FC

$bsfc = \frac{m_f}{BP} = \frac{0.272 \times 60}{70.588}$ kg/kwhr.

Problems on Heat balance

i) The following observations were recorded in a test of 1 hr. duration on a single cylinder oil Engine working on Four stroke cycle. Bore = 300mm. Stroke = 450mm. Fuel used = 8.8 kg/hr. Calolific value of fuel = 41,800 KJ/kg. average speed = 200 r.p.m. Mean effective pressure = 5.8 bar. Brake friction load = 1860 N. Quantity of cooking water = 650 kg. Temperature rise = 22°C, dia of brake wheel = 1.22 met. calculate i) Mechanical efficiency ii) Brake thermal efficiency and draw the heat balance sheet.

A) Brake power $= \frac{\pi (W-S)(D+d)N}{60}$, $N=1$, $K = \frac{1}{2}(4-\text{Stroke})$

$d = 300\text{mm} = 0.3$ met , ~~$W = 8.8$ kg~~

$l = 450\text{mm} = 0.45$ met. ~~$= 8.8 \times 9.81 = 86.328$~~ $m_f = 8.8$ kg/hr.

$C_V = 41,800$ KJ/kg

$P_{im} = 5.8$ bar $= 5.8 \times 10^5$ N/m²

$N = 200$ r.p.m.

Brake friction load = 1860 N = (W-S)

dia of Brake wheel = 1.22 met

$$BP = \frac{3.14(1860)(1.22)\times 200}{60} = 23750.96 \text{ W}$$

$= 23.76$ KW

B.P = ~~29591.36~~ = ~~29.59 KW~~ 23.750 KW

$$ip = \frac{nP_{im}LANK}{60,000}$$

$$= \frac{1 \times (5.8\times10^5) \times 0.45 \times \frac{\pi}{4}(0.3)^2 \times 200 \times \frac{1}{2}}{60,000}$$

$ip = 30.73275$ KW

$$\eta_m = \frac{B.P}{ip} = \frac{2991.36/1000}{30.7327} \quad \frac{23.750}{30.732} = 77.28\%.$$

$$\eta_m = 96.28\%.$$

$$\eta_{bth} = \frac{BP}{m_f \times C_v} = \frac{23.750 \times 4000}{\frac{8.8 \times 41800}{3600}} = 6.45\% \quad 0.232 = 23.2\%.$$

$m_f = 8.8$ kg/hr.

Total heat supplied by fuel $= m_f \times C_v$

$= 8.8 \times 41800$

$= 367840$ kJ/hr.

Heat equivalent to ip $= 30.74$ kW $=$ kJ/s

$= 30.74 \times 3600 = 110664$ kJ/hr.

Heat carried away by cooling water $= m_w C_{pw} (t_i - t_o)$

$= 650 \times 4.18 (22)$

$= 59774$ kJ/hr

Heat balance sheet :-

ITEM	kJ	percent (%)
1) Total amount of heat supplied	367840.	100%.
2) Heat equivalent to i.p	110664	$\frac{110664}{367840} = 30\%$.
3) Heat carried away by cooling water	59774	$\frac{59774}{367840} = 16.25\%$.
4) Heat carried away by exhaust gases, radiation etc (by difference)	$(110664 + 59774) - 367840 = 197402$	$\frac{197402}{367840} = 53.66\%$.

2) From the data given below calculate indicated power, Brake power and draw a heat balance sheet for a two storoke diesel engine own for 20 minits at full load. speed = 350 r.p.m

Mean effective pressure = 3.1 bar · Net Brake load = 640N. fuel consumption = 1.52 kg. cooling water = 162 kg. water inlet temperature 30°C. water outlet temperature = 55°C. Air used per kg of fuel = 32 kg. room temperature = 25°C. Exhaust temperature = 305°C. cylinder bore = 200mm. cylinder stroke = 280 mm. Brake drum diameter 1 met. Calorific value of fuel 43,900 kJ/kg. Steam formed per kg of fuel in the exhaust = 1.4 kg. Specific heat of dry exhaust gases = 1 kJ/kgK. Sp. heat of steam in exhaust = 2.09 kJ/kgK

A) given that, K=1, n=1, N = 350 r.p.m.

$P_{im} = 3.1 \text{ bar} = 3.1 \times 10^5 \text{ N/m}^2$.

$(W-S) = 640N$, $m_f = 1.52$ kg., $m_w = 162$ kg.

$t_i = 30°C$, $t_o = 55°C$, $T = 25°C$.

$d = 200mm = 0.2$ met.

$L = 280 mm = 0.28$ met

$D = 1$ met.

$C_v = 43,900$ kJ/kg.

$$ip = \frac{nP_{im}LANK}{60,000}$$

$$= \frac{1 \times (3.1 \times 10^5) \times 0.28 \times \frac{\pi}{4}(0.2)^2 \times 350 \times 1}{60,000} \Rightarrow$$

$= 15.89$ kW

$$B.P = \frac{\pi(W-S)(D+d)N}{60000} = \frac{3.14(640)(1)(350)}{60 \times 1000}$$

$= 11.73$ kW.

IP $\Rightarrow$ IP × 60 × 20 = 19080 KJ

CW $\Rightarrow m_w \times C_{pw} \times (\Delta T)$
= 162 × 4.18 × (55 − 30)
= 16929 kJ.

Total mass gas = 32 × 1.52
= 48.64 kg

mass of exhaust gases
= $m_f + m_a$ = 1.52 + 48.64
= 50.16 kg.

Mass of steam formed = 1.4 × 1.52
= 2.13 kg.

∴ Mass of dry exhaust gases = 50.16 − 2.13
= 48.03 kg.

$t_{sup} = 305$

Total heat supplied 66,728

Heat equalent ip KB 19080

Heat carried cooling 16929

" Exhaust 13648 /

Steam Exhaust 6613 = 2.13

Heat unaccounted 10658

steam: $2.13[h_f + h_{fg} + C_{ps}(t_{sup} - t_s)]$

At Atm 1.013 bar

$h_f = 417.5$

$h_{fg} = 2257.9$

$C_{ps} = 2.09$

$t_{sup} = 305$

$t_s = 99.6$ = 6613 required

1) A Gasoline Engine working on 4-stroke develops a brake Power of 20.9 kw. a morse test was carried out on this engine and brake power obtained when each cylinder was made inoperative by shut circiting the spark plugs are 14.9, 14.3 14.8, 14.5 respectively. The test was conducted at constant speed and find. the indicated power, mechanical efficiency and brake mean effective pressure when all the cylinders are firing The bore of the Engine is 75mm and stroke is 90mm the Engine is running at 3000 r.p.m.

A) given that, $BP_{1234} = 20.9$ kw.

$BP_{234} = 14.9$ kw.

$BP_{134} = 14.3$ kw.

$BP_{124} = 14.8$ kw.

$BP_{123} = 14.5$ kw.

$iP_1 = BP_{1234} - BP_{234} = 6$ kw

$IP_2 = BP_{1234} - BP_{134} = 6.6$ kw

$IP_3 = BP_{1234} - BP_{124} = 6.1$ kw

$IP_4 = BP_{1234} - BP_{123} = 6.4$ kw.

$IP = IP_1 + IP_2 + IP_3 + IP_4$

$IP = 25.1$ kw.

$\eta_{mec} = \frac{BP}{IP} = \frac{20.9}{25.1} = 83.26\%$.

bmep = ?

$BP = \frac{nP_{bm}LANK}{60000}$

$P_{bm} = \frac{60,000 \times BP}{nLANK}$

$$P_{bm} = \frac{60,000 \times 20.9}{4 \times (0.09) \times \frac{3000}{2} \times \frac{\pi (0.075)^2}{4} \times 0.5} = 5,25643.6 \ N/m^2$$

$P_{bm} = 52591.01 N/m^2 = 5.25 bar$

$P_{bm} = 5.25 bar.$

V. Ganesh

25) A Mose test on a 12-cylinder 2-stroke C.I Engine of ~~both~~ bore 40cm and stroke 50cm running at 200 r.p.m gave the following Readings. (39)

Condition.		Brake load (N)	BP = $\frac{WN}{180}$
All firing	—	2040	$2050 \times 200/180 = 2277.8$ kw.
1st cylinder	—	1830	$1830 \times 200/180 = 2033.3$
2nd "	—	1850	$1850 \times 200/180 = 2055.55$
3nd "	—	1850	$1850 \times 200/180 = 2055.55$
4 "	—	1830	$1830 \times 200/180 = 2033.3$
5 "	—	1840	$1840 \times 200/180 = 2044.44$
6 "	—	1855	$1855 \times 200/180 = 2061.11$
7 "	—	1835	$1835 \times 200/180 = 2038.88$
8 "	—	1860	$1860 \times 200/180 = 2066.66$
9 "	—	1820	$1820 \times 200/180 = 2022.22$
10 "	—	1840	$1840 \times 200/180 = 2044.44$
11 "	—	1850	$1850 \times 200/180 = 2055.55$
12 "	—	1830	$1830 \times 200/180 = 2033.33$
All firing	—	2060.	$2050 \times 200/180 = 2277.8$ kw

the output is found from the dynamometer using the relation $BP = \frac{WN}{180}$ kw where W is the brake load in N. and N is the speed in r.p.m. Calculate indicated power, Mechanical efficiency brake mean effective pressure.

A) Given that,

$n=12$, $K=1$, $d=40cm=0.4m$, $L=50cm=0.5m$, $N=200$ r.p.m

$W=\frac{2040+2060}{2}=2050\ N \times \frac{200}{180}=2277.8$ KW.

$IP_1 = BP - BP_1$

$= 2277.8 - 2033.3 = 244.5$ KW

$IP_2 = 2277.8 - 2055.5 = 222.3$

$IP_3 = 2277.8 - 2055.5 = 222.3$

$IP_4 = 2277.8 - 2033.3 = 244.5$

$IP_5 = 2277.8 - 2044.4 = 233.4$

$IP_6 = 2277.8 - 2061.11 = 216.69$

$IP_7 = 2277.8 - 2038.8 = 239$

$IP_8 = 2277.8 - 2066.6 = 211.2$

$IP_9 = 2277.8 - 2022.2 = 255.6$

$IP_{10} = 2277.8 - 2044.4 = 233.4$

$IP_{11} = 2277.8 - 2055.5 = 222.3$

$IP_{12} = 2277.8 - 2033.3 = 244.5$

$IP = 244.5 + 222.3 + 2223 + 244.5 + 233.4 + 216.69 + 239 + 211.2 + 255.6 +$
$233.4 + 222.3 + 244.5$

$IP = 2789.69$ KW

$\eta = \frac{2277.8}{2789.6} = 81\%$.

$bmep = \frac{BP \times 60000}{LANK} = \frac{2277.8 \times 60000}{0.5 \times \frac{\pi}{4}(0.4)^2 \times 200 \times 12}$

$= 9.06 \times 10^5$ Pa

$= 9.06$ bar

3) A 4-stroke gas Engine has cylinder dia of 25 cm and stroke 45 cm the effective Brake drum dia is 1.6 m. The observations made in a test of engine where as follows.

Duration of the test = 40 minutes

Total no. of revolutions = 8080

Total no. of Explosions = 3230.

Net load on the brake = 90 kg.

Mean effective pressure = 5.8 bar

Volume of gas used = 7.5 m^3.

pressure of gas indicated in the meter = 136 mm of water. of gauge.

Atmospheric temperature 17°c.

Calorific value of gas = 19 MegaJ/m^3. (= 19000 KJ/m^3.) At normal temp. and press.

Rise in temperature of Jacket cooling water = 45°c.

Cooling water supplied = 180 kg.

Draw the heat Balance sheet and estimate indicated thermal efficiency and Brake thermal efficiency. Assume atmospheric pressure as 760 mm of Hg.

A) $P_{im} = 5.8 \times 10^5$.

$n_K = \frac{3230}{40} = 80.75$ rev/min. , $i_p = ?$

$\omega = 90$ kg $= 90 \times 9.81$

$BP = \frac{2\pi NT}{60{,}000}$

$T = \omega \times R$

$= (90 \times 9.81) \times \frac{1.6}{2} = 706.32$

$i_p = \frac{n P_{im} LANK}{60{,}000}$

$$i_p = \frac{(5.8\times10^5)\times 80.75 \times 0.45 \times \pi \times (0.25)^2 \times}{4\times 60{,}000}$$

$$N = \frac{\pi D n}{60}$$

$$= \frac{3.14 \times (0.25) \times 8080}{60} =$$

$$i_p = \frac{1 \times (5.8\times10^5) \times 0.45 \times \frac{\pi}{4}(0.25)^2 \times \frac{3230}{40} \times 60}{60{,}000.}$$

$i_p = 17.25$ kw.

$$BP = \frac{2\pi NT}{60} = \frac{2\times\pi \times \frac{8080}{40} \times 706.32}{60}$$

$= 14.94$ kw.

$P_{guage} = P_{atm} + $ Barometric pressure.

pressure of gas supplied $= 760 + 136$ mm. of water.

$= 760 + \frac{136}{13.6}$ mm of Hg.

$= 770$ mm of Hg.

Volume of gas at (NTP) $= 7.5 \times \frac{273}{290} \times \frac{770}{760}$

$= 7.5 \times 0.94 \times 1.013$

$= 7.15\ m^3$.

Total Heat supplied $= m_f \times CV$

$=$ Vol. $\times$ CV

$= 7.15 \times 19000$

$= 135850$ kJ/40 min

$= \frac{135850}{40} = 3396.25$ kJ/min.

$$\eta_{IHP} = \frac{i_p \times 60}{m_f \times CV} = \frac{17.25\times60}{3396.25} = 30.47\%$$

$$\eta_{BHP} = \frac{BP \times 60}{\text{Heat supplied}} = \frac{14.94\times60}{3396.25} = 26.39\%$$

Total Heat supplied = 3396.25 KJ/min.

Heat equivalent to BP = 14.94×60 = 896.46 KJ/m.

Heat carried away by cooling water,

$= m_w C_{pw} (\Delta T)$

$= \frac{180 \times 4.2 (45)}{40} = 850.5$ KJ/m.

$= \frac{180 \times 4.187 \times 45}{40} = 846.5$

Heat unaccounted = 3396.25 − 896.46 − 850.5

= 1649.29 KJ/min. ≃ 1653.35

Heat balance sheet:-

Item	KJ/min	percentage (%)
1) Total amount of heat supplied.	3396.25	100%.
2) Heat equivalent to BP	896.46	26.39 %.
3) Heat carried away by cooling water	846.5	24.92 %.
4) Heat unaccounted	1653.35	48.68 %.

Exhaust gas Compositions:-

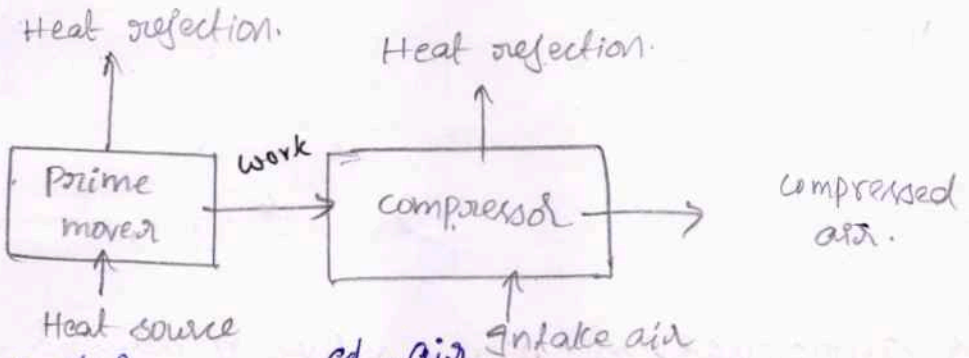
1) Carbodioxide } harmless
2) water vapour }
3) Nitrogen oxide
4) Unburnt hydrocarbons
5) Carbon monoxide
6) Aldehydes
7) Smoke } visible
8) Particulate }

→ Smoke meters are used.

Friction Power:- (including pumping losses also)

1) William's line method
2) Morse test
3) Motoring test
4) From measurement of IP & BP
5) Retardation test

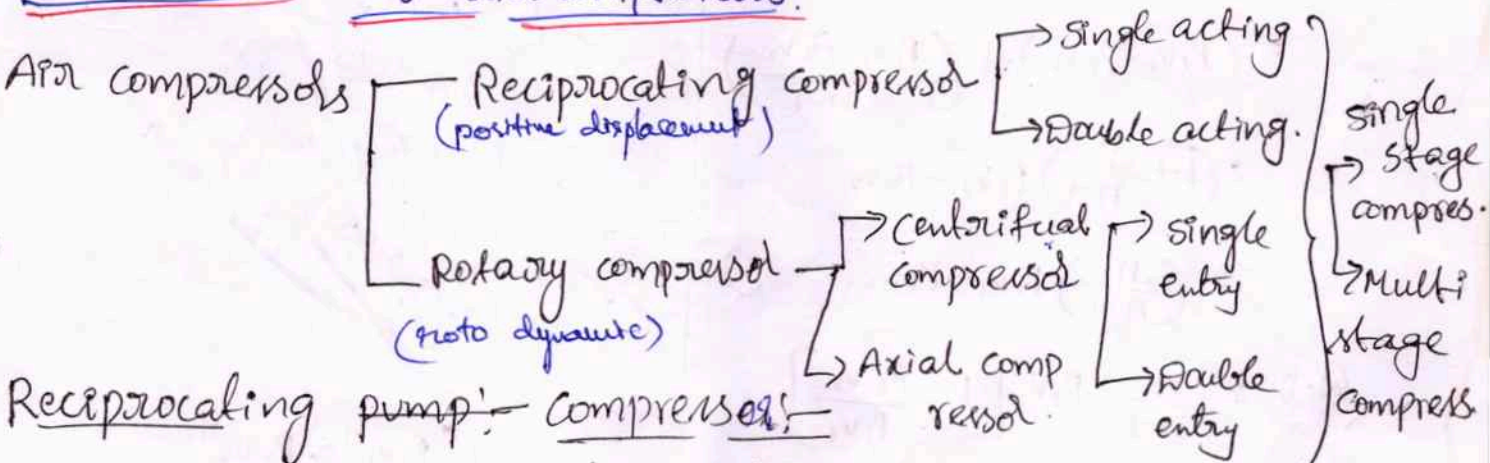
Air Compressors.



Applications of compressed air:-

1. To operate the tools in factories
2. To operate Drills and hammers. in roads & buildings.
3. To Excavate
4. Tunneling and mining.
5. For operating brakes of buses, trains
6. Large Quantity of air at moderate pressure is used in melting of iron, in blowing converters and cupola work.
7. Used in air conditioning, drying and refrigeration.

Classification of air compressors:-



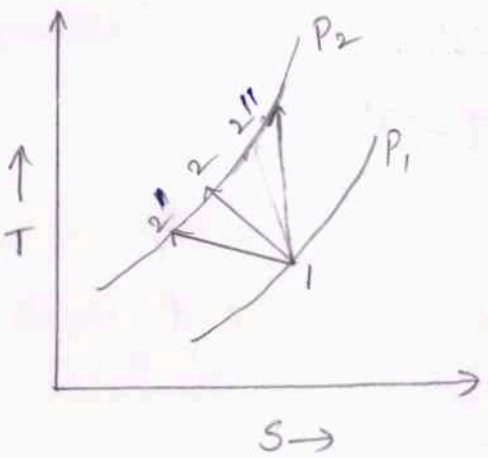
Reciprocating pump:- compressor:-

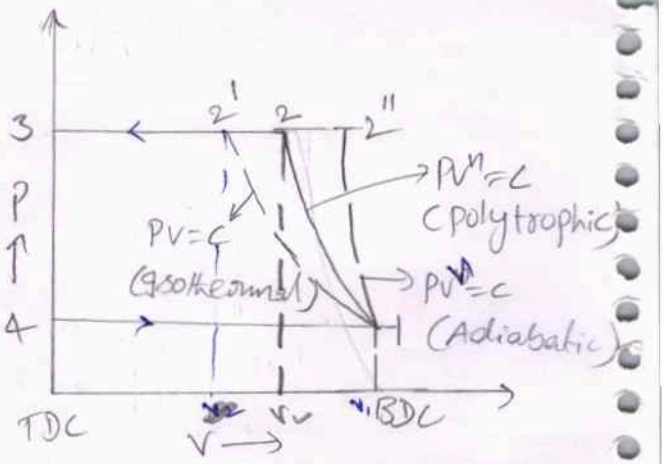
Equation of work (neglecting clearance volume):- (single stage compressor)

(4-1) process: Vol. of air aspirated at P_1 & T_1

(1-2) process: Air compressed according to polytrophic process.

$(PV^n = C)$





(2-3) process: Compressed Air of vol V_2 is delivered from the compressor.

For a reciprocating compressor a comparision b/w the actual workdone during the compression and the ideal Isothermal workdone is made by means of Isothermal efficiency.

$\therefore$ Isothermal efficiency $(\eta_{iso}) = \frac{\text{Isothermal workdone}}{\text{Actual workdone}}$

Total shaft workdone per cycle (W) = Area of 4123 4.

$\therefore W$ = Area under (4-1) − Area under (1-2) − Area under (2-3)

$W.D/cycle = P_1V_1 = \frac{P_2V_2 - P_1V_1}{n-1} - P_2V_2$

$= (P_1V_1 - P_2V_2) - \left(\frac{P_2V_2 - P_1V_1}{n-1}\right)$

$= (P_1V_1 - P_2V_2) + \left(\frac{P_1V_1 - P_2V_2}{n-1}\right)$

$= \left(1 + \frac{1}{n-1}\right) P_1V_1 - P_2V_2$

$= \left(\frac{n}{n-1}\right) P_1V_1 - P_2V_2 \longrightarrow (1)$

$W.D = \frac{n}{n-1} P_1V_1 \left[1 - \frac{P_2V_2}{P_1V_1}\right]$

$= \frac{n}{n-1} P_1V_1 \left[1 - \frac{P_2}{P_1}\left(\frac{P_1}{P_2}\right)^{1/n}\right]$

$= \frac{n}{n-1} P_1V_1 \left[1 - \frac{P_2}{P_1}\left(\frac{P_2}{P_1}\right)^{-1/n}\right]$

$W.D$ = Area (3-2) + Area (1-2) − Area (4-1)

$= P_2V_2 + \frac{P_2V_2 - P_1V_1}{n-1} - P_1V_1$

$= (P_2V_2 - P_1V_1) + \frac{P_2V_2 - P_1V_1}{n-1}$

$= \left(1 + \frac{1}{n-1}\right)(P_2V_2 - P_1V_1)$ (or)

$= \frac{n}{n-1} P_1V_1 \left(\frac{P_2V_2}{P_1V_1} - 1\right)$ ✓

$= \frac{n}{n-1} P_1V_1 \left(\frac{P_2}{P_1}\left(\frac{P_1}{P_2}\right)^{1/n} - 1\right)$

$= \frac{n}{n-1} P_1V_1 \left[\frac{P_2}{P_1}\left(\frac{P_2}{P_1}\right)^{-\frac{1}{n}} - 1\right]$

$= \frac{n}{n-1} P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$

$\left[P_1V_1^n = P_2V_2^n \quad \frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{1/n} \right]$

$$= \frac{n}{n-1} P_1 V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right] \longrightarrow (2)$$

The solution to this equation will always come out negative showing that work must be done on the compressd.

Since only the magnitude of workdone is required from the expression then it is often written as,

$$W.D/cycle = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \longrightarrow (3)$$

$$= \frac{n}{n-1} mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \longrightarrow (4)$$

TO obtain the discharge pressure (or) temperature $\left[\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right]$

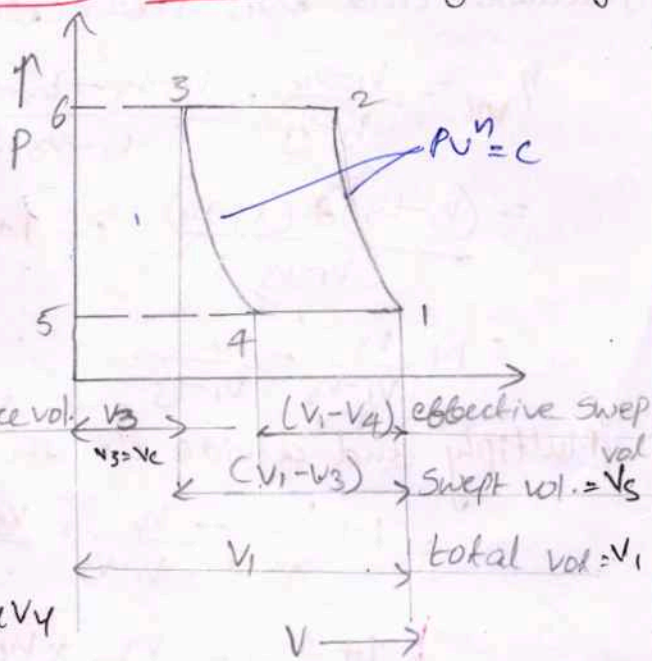
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}.$$

Equation of workdone with clearance volume:- (~~single stage~~)

$(V_1 - V_4)$ → effective swept volume

W.D/cycle W = Area under 41234.

= Area under 51265 − Area under 54365



Assuming polytropic index to be same for both compression and Expansion when workdone

$$W.D = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] - \frac{n}{n-1} P_4 V_4 \left[\left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} - 1 \right]$$

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

$$= \frac{n}{n-1} P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \longrightarrow (5)$$

Volumetric Efficiency:- It is the ratio of free air delivered to the displacement of the compressed. It is also the ratio of effective swept volume to the swept volume.

$$\text{Volumetric efficiency} = \frac{\text{Effect of swept vol.}}{\text{Swept vol.}} = \frac{V_1 - V_4}{V_1 - V_3} \rightarrow (1)$$

Because of presence of clearence volume volumetric efficiency is always less than unity. As a percentage it is usually varies from 60% to 85%. Generally,

$$\text{clearence ratio} = \frac{\text{clearence vol.}}{\text{Swept vol.}} = \frac{V_3}{V_1 - V_3} = k.$$

This clearence ratio varies from 4 to 10 percent.

$V_1 - V_4$ effective swept volume reduces as the pressure ratio increases and thus volumetric efficiency reduces.

$$\eta_{vol} = \frac{V_1 - V_4}{V_1 - V_3} = \frac{V_1 - V_3 + V_3 - V_4}{V_1 - V_3}$$

$$= \frac{(V_1 - V_3) + (V_3 - V_4)}{(V_1 - V_3)} = 1 + \frac{(V_3 - V_4)}{(V_1 - V_3)}$$

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3}.$$

Multiply and divide the above equation with V_3.

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3} \times \frac{V_3}{V_3}$$

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_3}{V_1 - V_3} \times \frac{V_4}{V_3} \qquad \left[\because \frac{V_3}{V_1 - V_3} = k\right]$$

$$\eta_{vol} = 1 + k - k \times \frac{V_4}{V_3}$$

$$\left[P_3 V_3^n = P_4 V_4^n \quad \frac{V_4}{V_3} = \left(\frac{P_3}{P_4}\right)^{1/n}\right]$$

$$\eta_{vol} = 1 + k - k \times \left(\frac{P_3}{P_4}\right)^{1/n_e} \rightarrow (2)$$

From diagram,

$$P_3 = P_2$$

$$P_4 = P_1$$

$$\eta_{vol} = 1 + k - k \left(\frac{P_2}{P_1}\right)^{1/n_c} \rightarrow (3)$$

$$\eta_{vol} = 1 + k - k \left(\frac{V_1}{V_2}\right) \rightarrow (4)$$