THERMAL ENGINEERING-I

ME 502 PC

Unit -I, IC Engines.

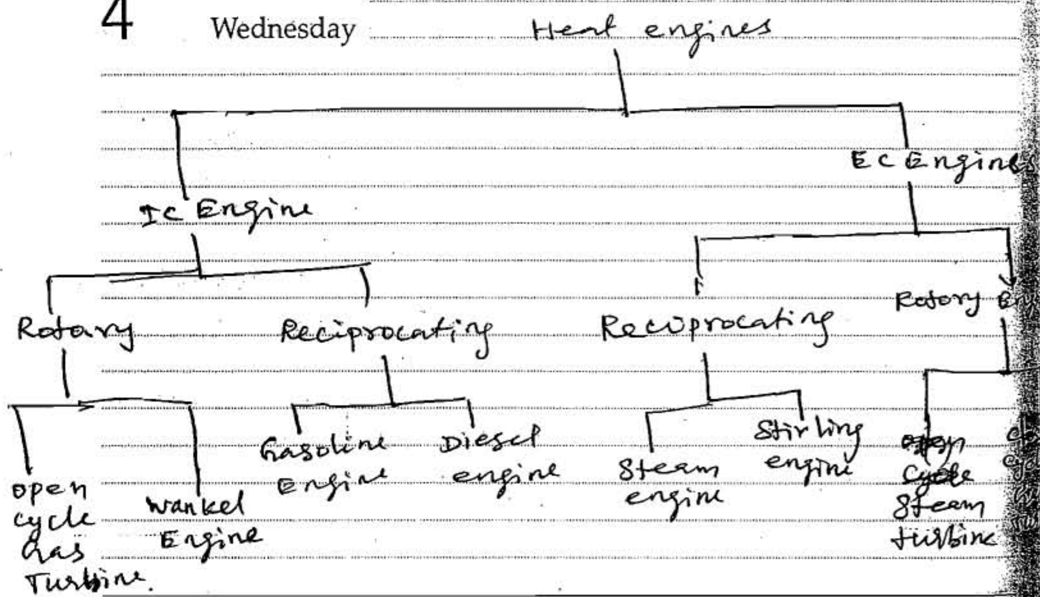
classification of Heat engines.

Based on the place of combustion, it is classified as

(i) Internal Combustion Engines (IC Engines)
(ii) External Combustion Engines (EC Engines)

Based on the type of motion, It can be classified as

1. Rotory engines
2. Reciprocating engines.



IC Engine applications.

IC engine have bee found suitable for use in automobile motor cycles and scooters, power boats, ships, slow speed aircraft, locomotives and power units of relatively small output.

Classification of IC Engines:

Internal combustion engines are usually classified on the basis of the thermodynamic cycle of operation type of fuel used, method of charging the cylinder, type of ignition, type of cooling and the cylinder arrangement etc.

1. cycle of operation.

(i) otto cycle or Constant volume Q_s cycle engine
(ii) Diesel cycle or constant pressure Q_s cycle engine

2. Type of fuel used.

- petrol
- Diesel
- Gasoline
- Kerosene
- alcohol
- benzene etc

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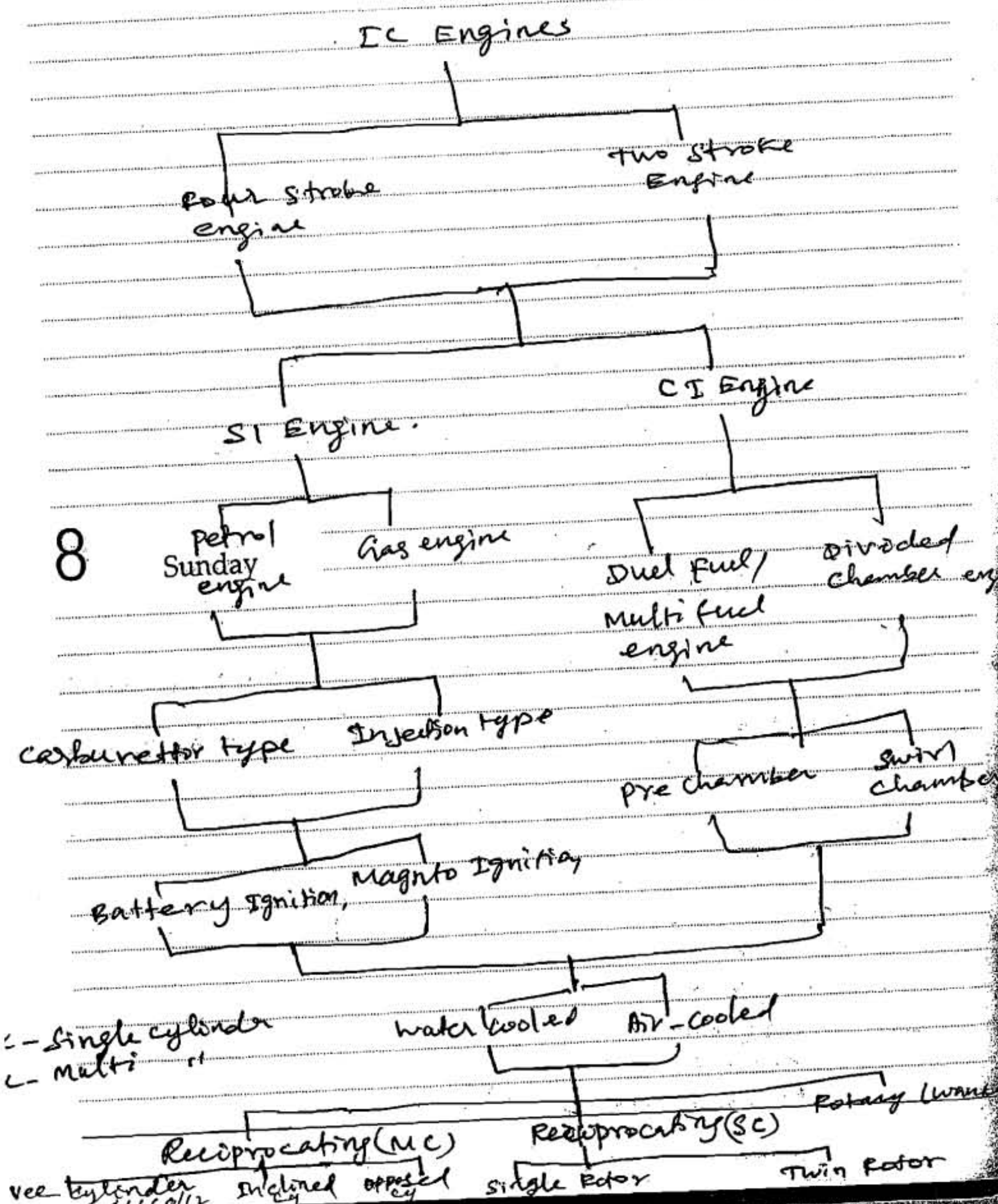

Diesel Engine
Rudolf Diesel (1892)

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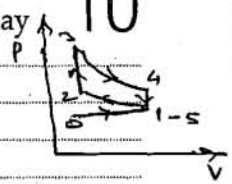
Working principles of IC Engines

Nicolaus A otto (1876)

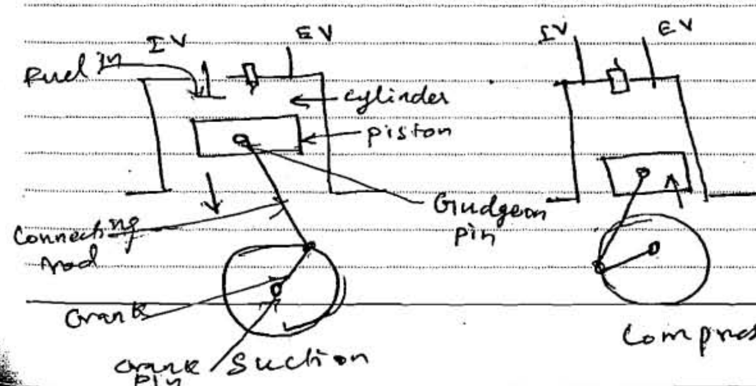
Four Stroke Spark Ignition Engine:-

In a four-stroke engine, the cycle of operation is completed in four strokes of the piston (or) two revulations of the crank shaft. During the the four strokes, there are five events to be completed, viz, suction, compression, combustion, expansion and exhaust. Each stroke consists of 180° of crankshaft rotation and hence a four stroke cycle is completed through 720° of crank rotation.

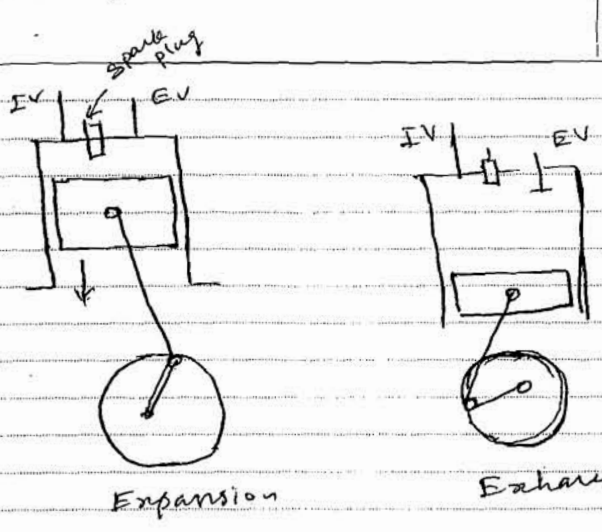


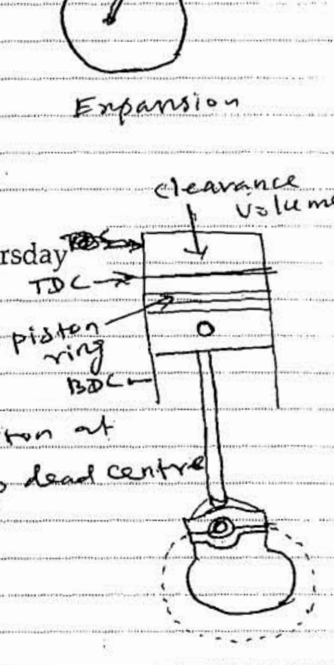
Strokes of operation.

(1) suction or intake stroke
(2) Compression stroke
(3) Expansion or power stroke
(4) exhaust stroke.



d<L → under square engine
d = L → square engine
d > L → over square engine





(i) Suction Stroke (or) Intake stroke $(0-1)$ → process

Suction stroke $0 \to 1$ (pv diagr) starts when the piston is at the top deal centre and about to move downwards. The inlet valve is open at this time and exhaust valve closed.

The charge consisting of Air-fuel mixture entering to the cylinder due to the ~~motions of~~ suction created by the piston moment towards the BDC. At the end of suction stroke the IV is closed.

(ii) Compression stroke $(1-2)$ & $(2-3)$ → process → process

The charge of A/F mixture is compressed ~~the~~ by the return stroke of the piston 1-2. During this stroke both IV and EV's

are in closed position. At the end of the compression stroke the mixture is ignited with the help of a spark plug. In ideal engine it is assumed that burning takes place instantaneously when the piston is at the TDC. During the burning process the chemical energy of the fuel is converted into heat energy producing a temperature rise of about 2000°C and the pressure at the end combustion process increased considerably due to heat release from fuel.

(iii) Expansion or power stroke $(3-4)$ → process

The high pressure of the burnt gases forces the piston towards the BDC, (stroke 3-4). Both the valves are ~~remains closed.~~ Both pressure and Temperature decreases during expansion.

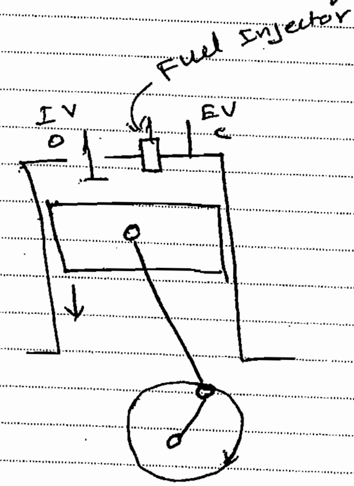
(iv) Exhaust stroke: (4-1).

During this process the exhaust valve opens and the Inlet valve remains closed. The pressure falls to atmospheric level a part of the burnt gases escapes. The piston moves from the BDC to TDC. (Stroke 5→0) and sweeps the burnt gases out from the cylinder almost at atmospheric pressure. The exhaust valve closes when the piston reaches TDC.

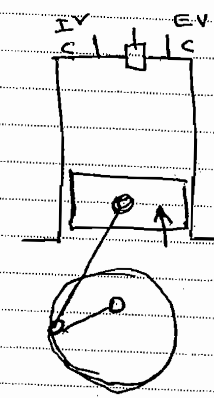
Four-stroke Compression Ignition Engine [CI Eengine]

The four stroke CI engine is similar to 4S-SI engine but it operates at a much higher compression ratio. The compression ratio of an SI engine is between 6 & 10, for CI engine is from 16 to 20. In the CI engine during suction stroke, air, instead of a fuel-air mixture is inducted.

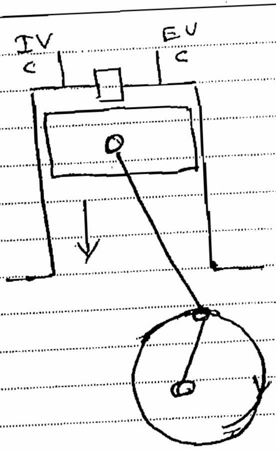
The ideal sequence of operations for the four-stroke CI engine as shown in fig.



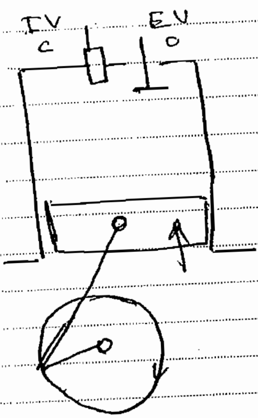
(a) Suction



(b) Compression



(c) Expansion

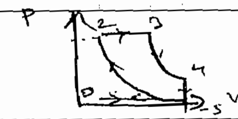


(d) Exhaust

(i) Suction stroke: Air alone is inducted during the suction stroke. During this ~~stroke~~ intake valve is open and exhaust value is closed.

(ii) Compression stroke:
Air inducted during the suction stroke is compressed into the clearance volume. Both valves remain closed during this stroke

(iii) Expansion stroke:-
Fuel injection starts nearly at the end of the compression stroke, the rate of injection is such that combustion maintains the pressure constant in spite of the piston movement on its expansion stroke increasing the volume.



Heat is assumed to have been added at constant pressure. After the injection of fuel is completed (ie after cut-off) the products of combustion expand. Both the valves remain closed during the expansion stroke.

(iv) Exhaust stroke:
The piston travelling from BDC to TDC pushes out the products of combustion. The exhaust valve is open and the intake valve is closed during this stroke.

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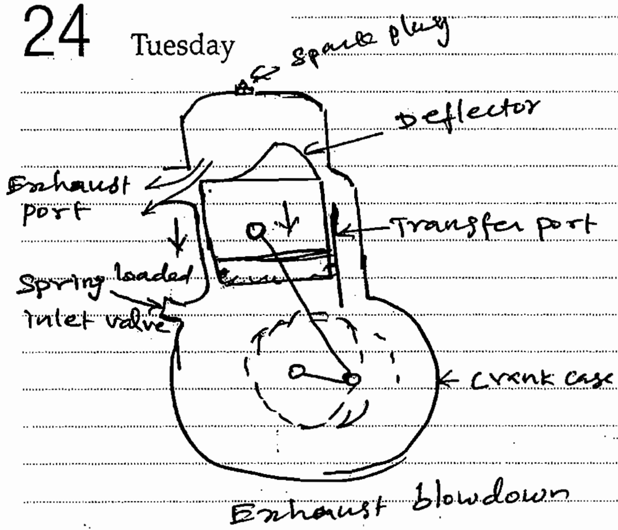
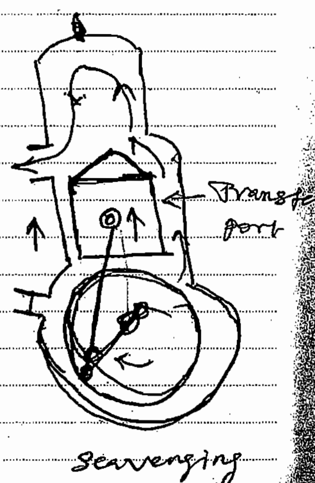
Two-Stroke Engine :-

The two unproductive strokes, viz, the suction and exhaust could be served by an alternative arrangement, especially without the movement of the piston then there will be a power stroke for each revolution of the crankshaft. In such an arrangement, theoretically the power output of the engine can be doubled for the same speed compared to a four-stroke engine. Based on this concept, Dugald Clark (1878) invented the two-stroke engine.





Crank Case Scavenged Two Stroke Engine.

May 2018

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Working principle

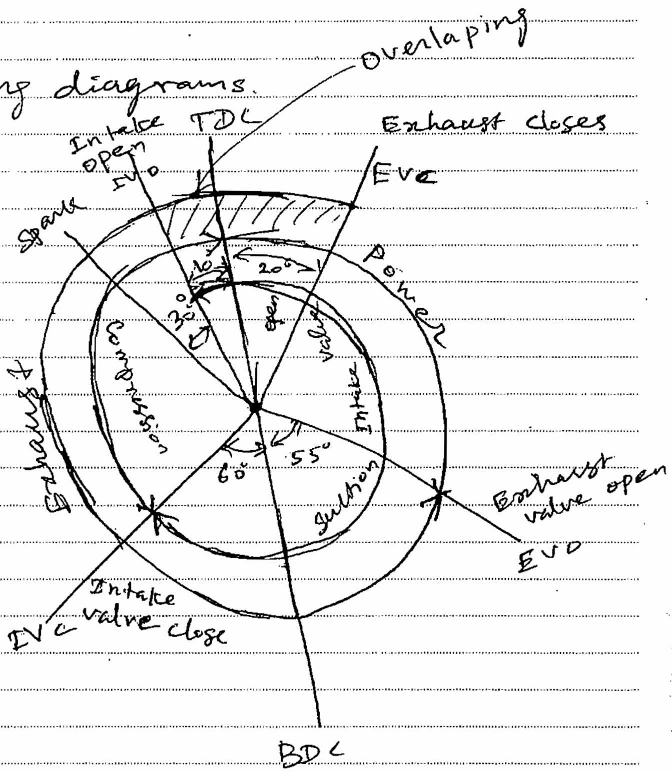
The air or charge is inducted into the crankcase through the spring loaded inlet valve when the pressure in the crankcase is reduced due to upward motion of the piston during compression stroke. After the compression and ignition, expansion takes place in the usual way.

During the expansion stroke the charge in the crankcase is compressed. Near the end of the expansion stroke, the piston uncovers the exhaust ports and the cylinder pressure drops to atmospheric pressure as the combustion products leave the cylinder. Further movement of the piston uncovers the transfer ports, permitting the slightly compressed charge in the crankcase to enter the engine cylinder.

The top of the piston has usually a projection to deflect the fresh charge towards the top of the cylinder before flowing to the exhaust ports. This serves the double purpose of scavenging the upper part of the cylinder of the combustion products and preventing the fresh charge from flowing directly to the exhaust ports.

Valve Timing and port timing diagrams for Four stroke and Two stroke engines.

Valve Timing diagrams.

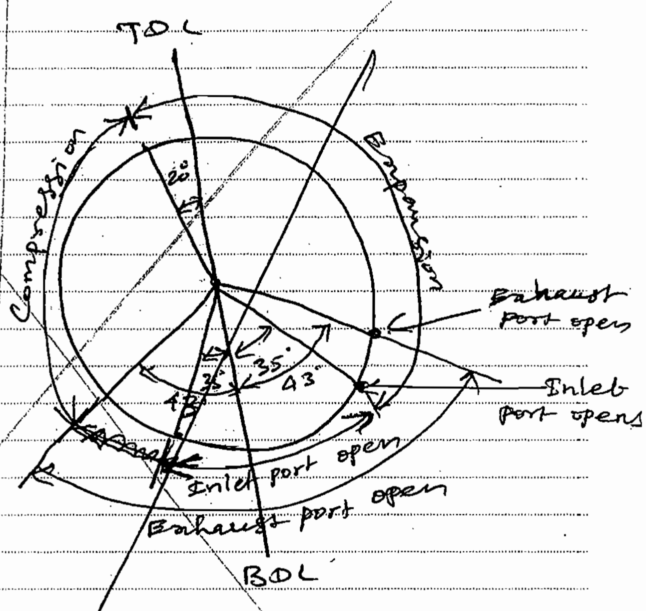


Actual valve Timing diagram for S.I. Engines.

The inlet valve opens 10-30° before TDC. The air-fuel mixture is sucked into the cylinder till the inlet valve closes. The inlet valve closes 30-40° or even 60° after the BDC. The charge is compressed till spark occurs. The spark is produced 20-40° before TDC. It gives sufficient time for the fuel to burn. Both pressure and temperature increase. The burnt gases are expanded till the exhaust valve opens. The exhaust valve opens 30-60° before BDC. The exhaust gases are forced out from the cylinder till the exhaust valve closes. The exhaust valve closes 8-20 after TDC. Before closing, the inlet valve again opens 10-30° before TDC. The period between EVO and EVC is known as valve overlap period. The angle between inlet valve opening and exhaust valve closing and exhaust valve closing is known as angle overlap.

1 2018 May Tuesday

Port Timing Diagram:-

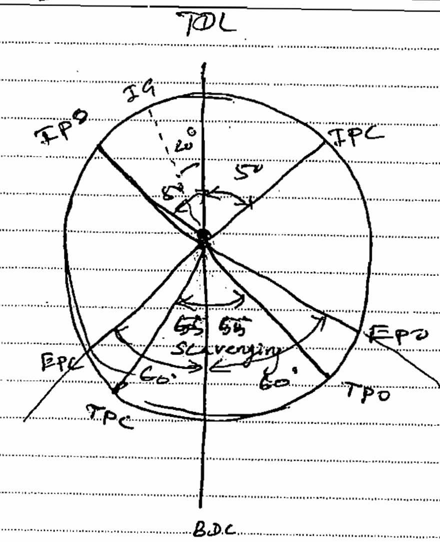


2 Wednesday

~~Diesel Engine~~

Thursday 3

Port Timing Diagram for 2S - SI Engine (petrol)

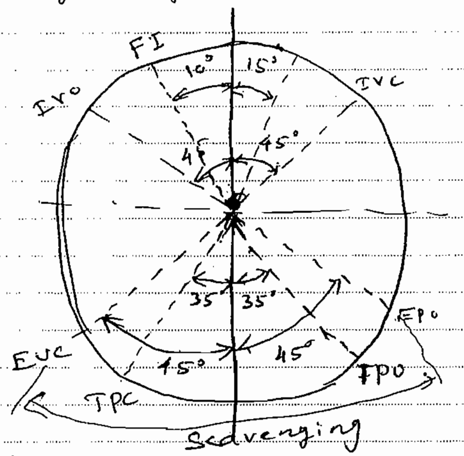


Friday 4

IPO - 45° to 55° BTDC
IPC - 45° to 55° ATDC
TPO - 30° to 40° BBDC
TPC - 30° to 40° ABDC
Ignition - 15° to 20° BTDC
EPO - 40 to 55° BBDC
EPC - 40 to 55° ABDC

- The two stroke cycle engine are provided with ports. The timing sequence of events such as opening and closing of various ports are graphically shown in terms of crank angles from dead centre positions. This diagram is known as port timing diagram.

Port timing diagram for 2 stroke Diesel (CI) Engine



The fuel is injected in the cylinder 10° to 15° BTDC.

The other difference between CI & SI is that the charging and scavenging periods of CI engine (90°) and greater than SI engine (70°).

Scavenging: Removal of unburnt gases from the cylinder.

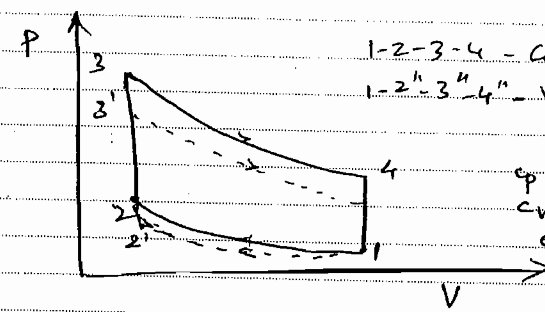
Comparison of A/S cycle, F/A cycle and Actual cycles.

Air standard cycle.

Air standard cycles are ideal cycles based on the following assumptions.

→ the working fluid is air (ideal gas)
→ all the processes are internally reversible
→ the combustion process is replaced by heat input from an external source
→ heat rejection is used to restore fluid to initial state

Effect of specific Heat.



1-2-3-4 - Const-SP. heat
1-2''-3''-4'' - variable SP. heat.

For air
$c_p = 1.005$ kJ/kgK (300K)
$c_v = 0.717$ kJ/kgK (300K)
$c_p = 1.345$ kJ/kgK at 2000K
$c_v = 1.057$ kJ/kgK at 2000K.

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

$c_p = a_1 + k_1 T$ | where a_1, b_1 and k_1 are constant
$c_v = b_1 + k_1 T$ | Now $R = c_p - c_v = a_1 - b_1$

above 1500K the specific heat increases much more rapidly and may be expressed as

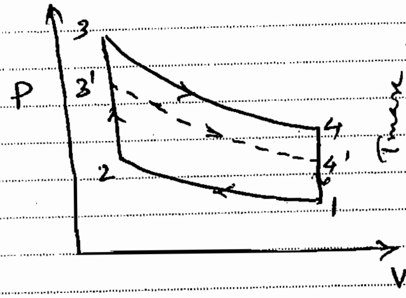
$c_p = a_1 + k_1 T + k_2 T^2$
$c_v = b_1 + k_1 T + k_2 T^2$

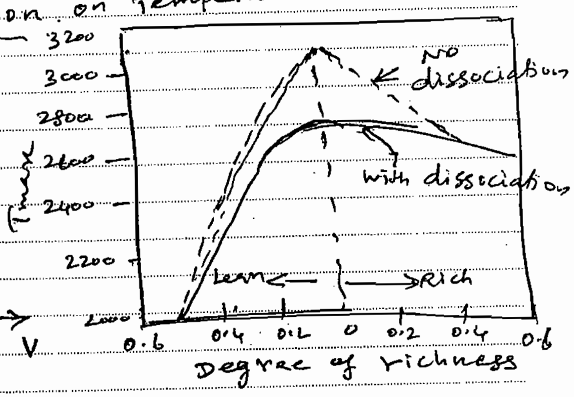
Rich mixture: A mixture having more fuel than that a chemically correct mixture is termed as Rich mixture.

Lean mixture: A mixture having less fuel than that a chemically correct mixture.

A F 1 : 15 ⇒ chemically correct

Effect of Dissociation on Temperature





poor utilization of ~~Power~~ Heat

Heat balance of an engine.

Useful output (Brake power)	34 %
Cooling loss	30 %
Exhaust loss	26 %
Friction, Radiation	10 %
	100 %

Dissociation :-

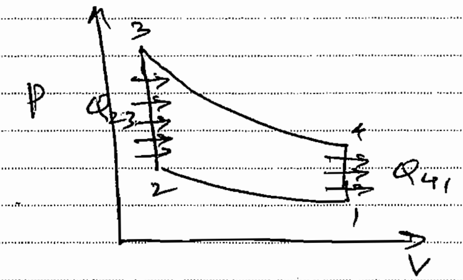
Dissociation process can be considered as the disintegration of combustion products at high temperature.

→ Dissociation can also be looked as the reverse process of combustion.

→ During dissociation the heat is absorbed whereas during combustion the heat is liberated.

Dissociation reduces the maximum temperature by about 300°C even at the chemically correct air-fuel ratio.

Ideal Air standard Otto cycle



In IC Engine, Mainly dissociation of CO_2 into CO and O_2 occurs, whereas there is very little dissociation of H_2O

The dissociation of CO_2 into CO and O_2 starts commencing around 1000°C and the reaction equation can be written as

$$CO_2 \rightleftharpoons 2CO + O_2$$

Similarly, the dissociation of H_2O occurs at temperature above 1300°C and is written as

$$H_2O \rightleftharpoons 2H_2 + O_2$$

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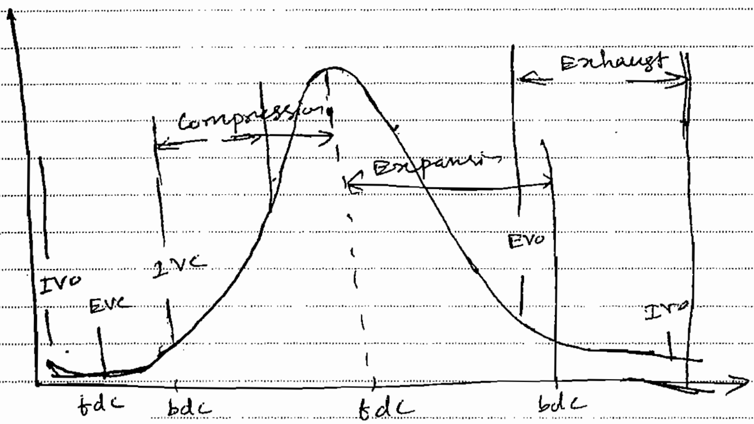
Actual Heat addition.



→ variation of specific heats with temperature

→ Dissociation of the combustion products

→ progressive combustion
→ Incomplete combustion of fuel
→ Heat transfer into the walls of the combustion chamber
→ Blowdown at the end of the exhaust process

→ Gas (mixture) exchange process.

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Losses in Actual cycles.

→ Time loss

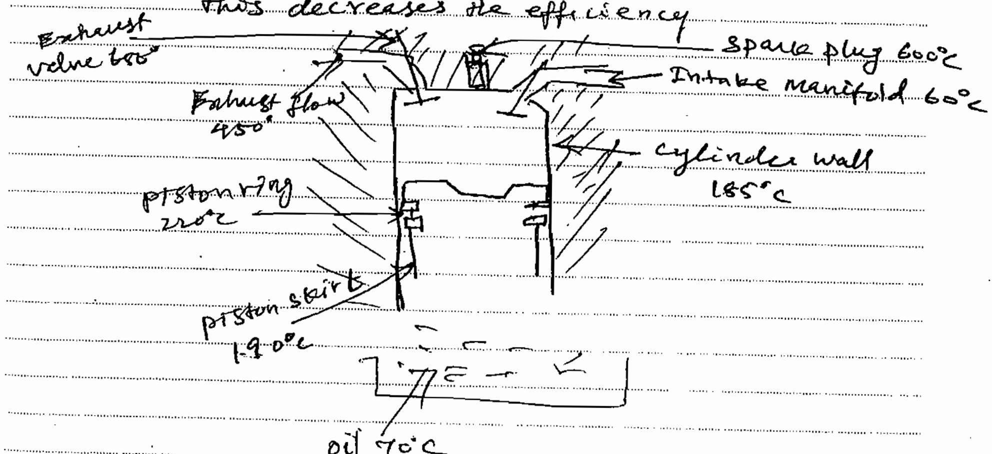
Heat addition is not instantaneous, and spread over a period (30° to 40° of crank shaft revolution). There fore, P_{max} is not at TDC, but just after TDC.

→ Time loss depends upon flame velocity which, in turn, again depends on type of fuel used, A/F ratio and shape of combustion chamber.

→ Heat loss (Heat transfer loss)

This is due to the transfer of heat through water jackets and cooling fins. Also, some heat is being transferred during compression and expansion process.

Due to heat loss, Temperature T_{max} decreases, and specific heat gets reduced. this decreases the efficiency



April 2018

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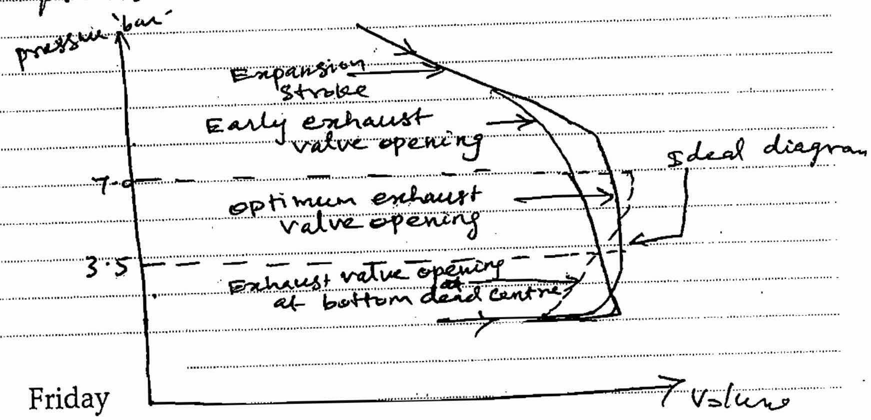
→ Blow Down loss.

Blow down loss is due to the early opening of exhaust valves. This results in drop in pressure, and a loss of work output during expansion stroke. Too early opening results in loss of expansion work. Best compromise is between 40° - 70° BBDC. (3.5 bar pressure



→ Rubbing Friction loss.

Rubbing friction loss is due to friction between the piston and chamber walls, friction in various bearings and also includes the energy spent in operating various auxillary equipments such as cooling fans, water pump etc.

The piston ring friction increases rapidly with engine speed. It also increases to a small extent with increase in mean effective pressure. The bearing friction and the auxillary friction also increases with engine speed.

June 2018

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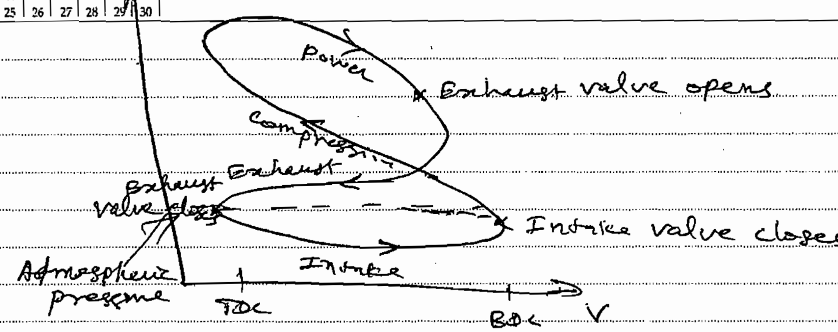
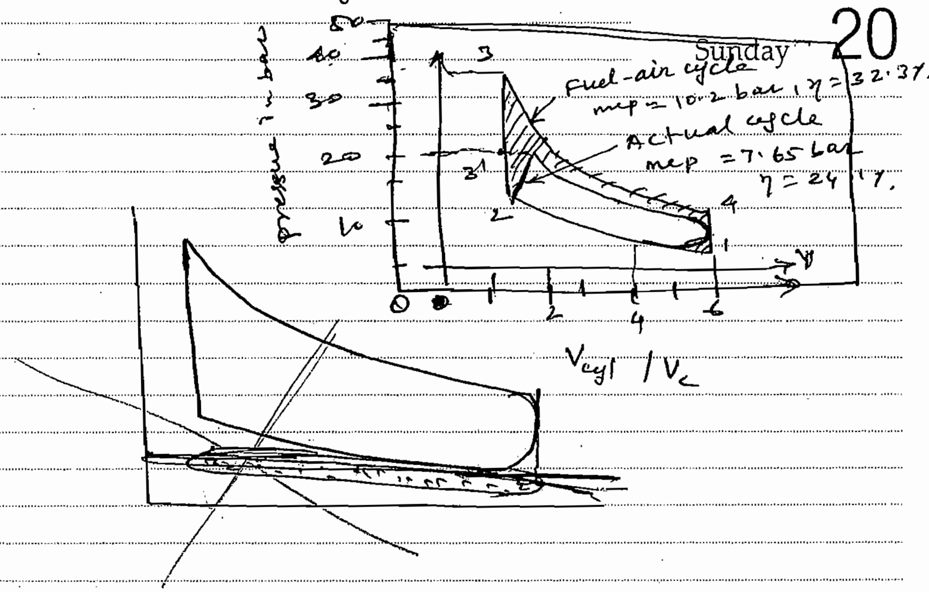


Losses during cycle.



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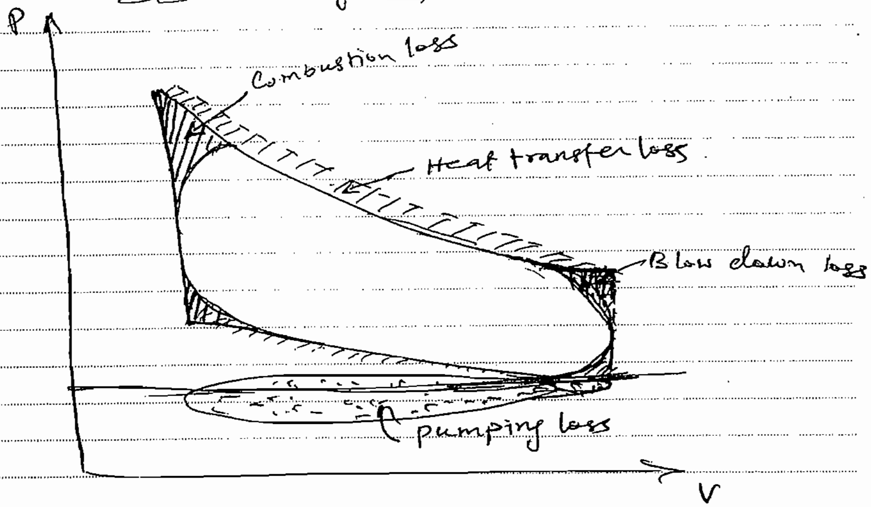
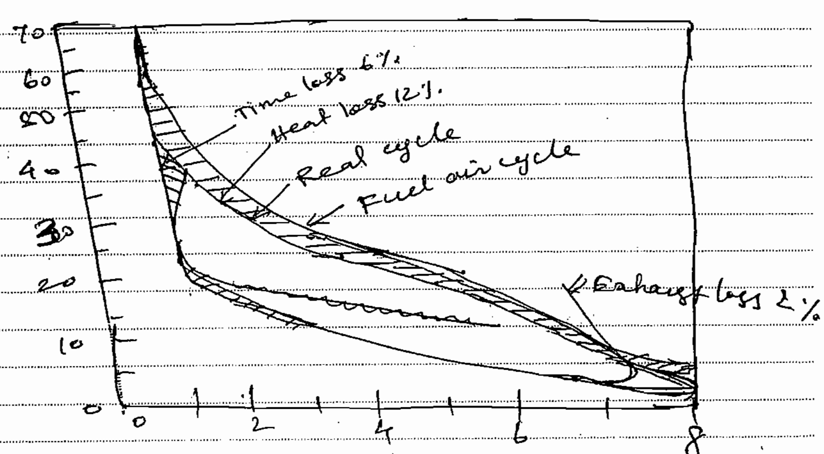
April 2018

* Heat loss during Combustion is to reduce the maximum temperature and there fore the specific heats are lower.

Losses during cycle



Magnitude of losses



Effect of Time losses, Heat loss and exhaust loss

June 2018

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~~Heat transfer~~

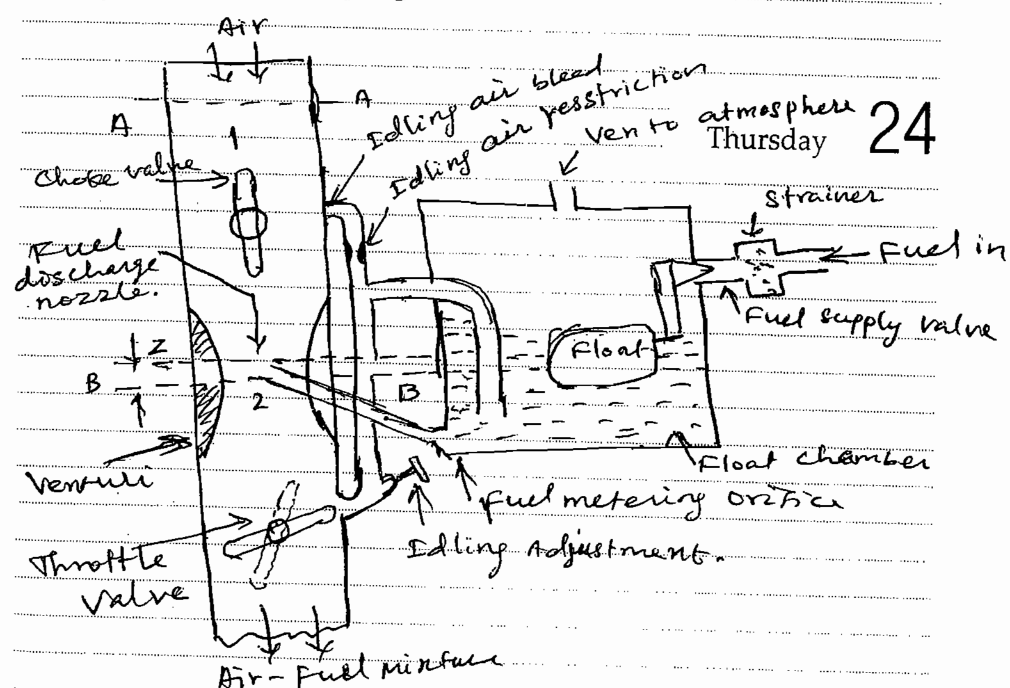
Carburettor :-

principle of carburetion :-

Both air and gasoline are drawn through the carburetor and into the engine cylinder by the suction created by downward movement of the piston.

→ The main function of the carburetor is to atomize ~~and~~ vapourize the fuel and provide the sufficient high fuel vapour-air ratio for efficient starting of the engine.

The Simple carburetor :-



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

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Fuel Injection System:

In a constant pressure cycle (or) diesel engine only air is compressed in the cylinder and then fuel is injected into the cylinder by means of a fuel injection system.

Injection system classified as
(i) Air injection system
(ii) Solid injection system.

(I) Air injection system:-
In this system fuel is forced into the cylinder by means of compressed air. This system is little used nowadays, because it requires a bulky (heavy) multi-stage air compressor. This causes an increase in engine weight and reduces the brake power output.

→ one advantage that is claimed for the air injection system is good mixing of fuel with the air with resultant higher mean effective pressure. These advantage are off-set by the requirement of a multistage compressor there by making the air injection system obsolete.

(ii) Solid Fuel injection system:-
In this system the liquid fuel is injected directly into the combustion chamber without the aid of compressed air. Hence it is also called airless mechanical injection (or) solid injection system.

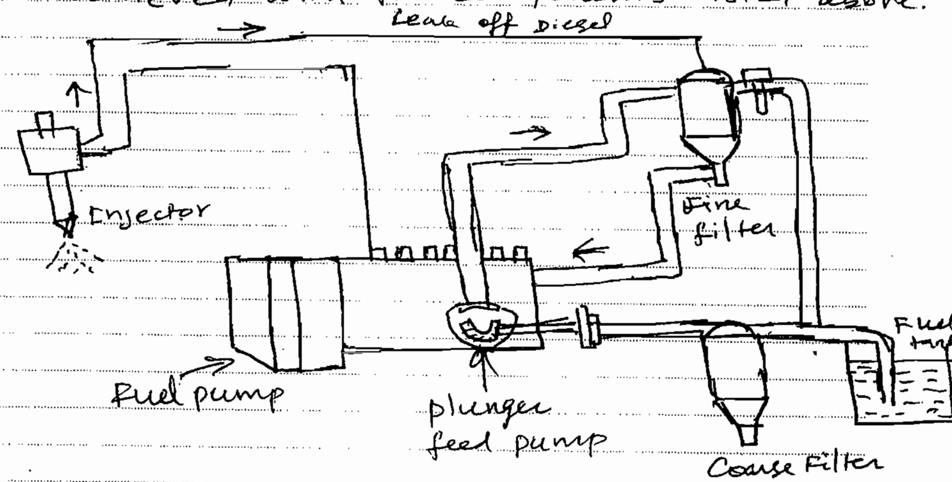
classification of Solid injection System
(i) Individual pump and nozzle system
(ii) Unit injection System
(iii) Common rail System
(iv) Distributor System

All the above systems comprise mainly of the following components.
(i) Fuel tank,
(ii) Fuel feed pump to supply fuel from main fuel tank to the injection System
(iii) Injection pump to meter and pressurize the fuel for injection,
iv) governor to ensure that the amount of fuel injected to is in accordance with variation load.

(v) Injector to take the fuel from the pump and distribute in the combustion chamber by atomizing it into fine droplets
(vi) Fuel filters to prevent dust and abrasive particles from entering the pump and injectors there by minimizing the wear and Tear of the components.

A typical arrangement of various Components for the solid injection System used in CI engine is shown in fig.

Fuel from the fuel tank first enters the Coarse filter from which drawn in to the plunger feed pump where the pressure is raised very slightly. Then the fuel enters the fine filter where all the dust and dirt particles are removed. From the fine filter the fuel enters the fuel pump where it is pressurized to about 200 bar and injected into the engine cylinder by means of injector. Any spill over in the injector is ~~refused to the~~ returned to the fine filter.
A pressure relief valve is also provided for the safety of the system. The above functions are achieved with the components listed above.



Typical Fuel Feed system for a CI Engine

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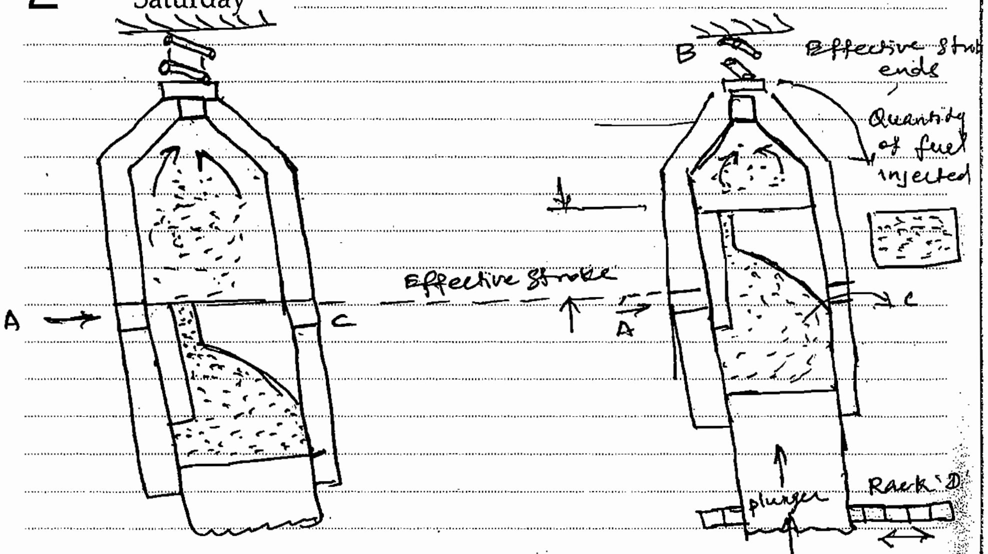
Injection pump:

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



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(a) Above sketch of a typical plunger is shown.

(b) A schematic diagram of the plunger within the barrel is shown. Near the port A, fuel is always available under relatively low pressure. While the axial movement of the plunger is through cam shaft, its rotational movement about its axis by means of rack 'D'. port B is the orifice through which fuel is delivered to the injector. At this stage it is closed by means of a spring loaded check valve.

When the plunger is below port A, the fuel gets filled in the barrel above it. As the plunger rises and closes the port A.

the fuel will flow out through port C. This is because it is has to overcome the spring force of the check valve in order to flow through port B. Hence it takes the easier way out via port C.

(c) At this stage rack rotates the plunger and as a result port C also closes. The only escape rotate for the fuel is past the check valve through orifice B to the injector. This is the begining of injection and also the effective stroke of the plunger.

(d) The injection continues till the helical identation on the plunger uncovers port C. Now the fuel will take the easy way out through C and the check

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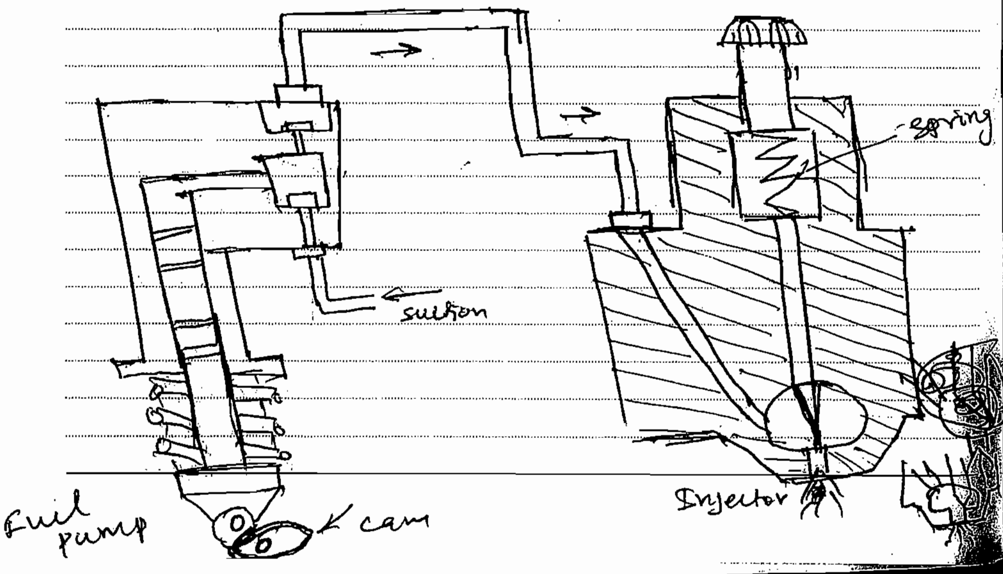
Valve will close the orifice B. The fuel Injection Stops and the effective stroke ends. Hence the effective stroke of the plunger is the axial distance traversed between the time port A is closed and the time port A is uncovered.

(e) & (f) the plunger is rotated to the position Show The same sequence of events occur. But in this case port C is uncovered Sooner. Hence the effective stroke is Shortened.



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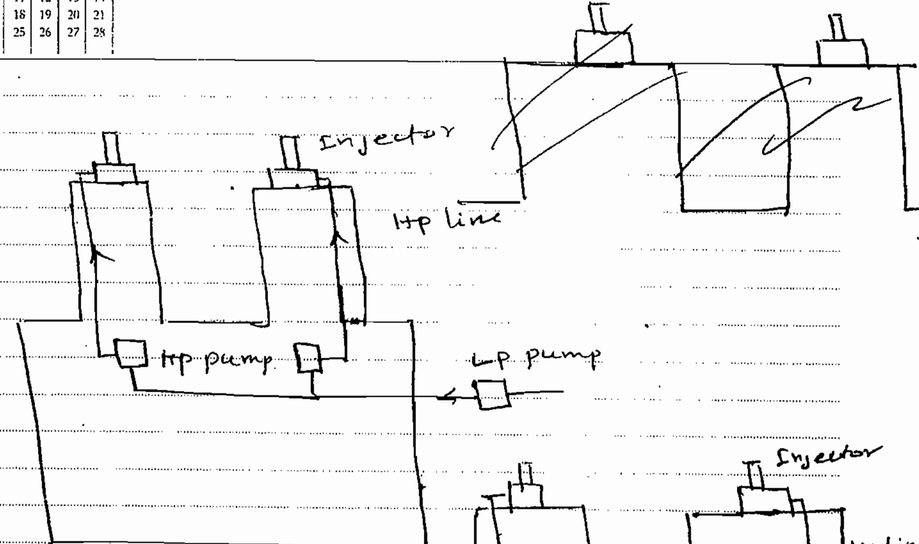
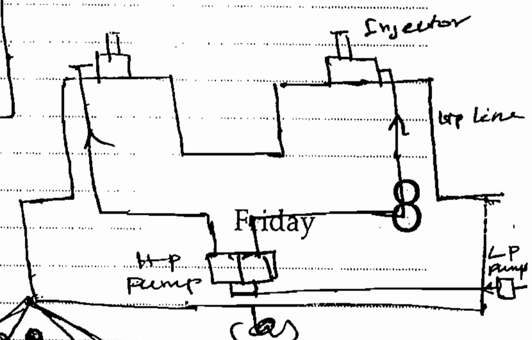
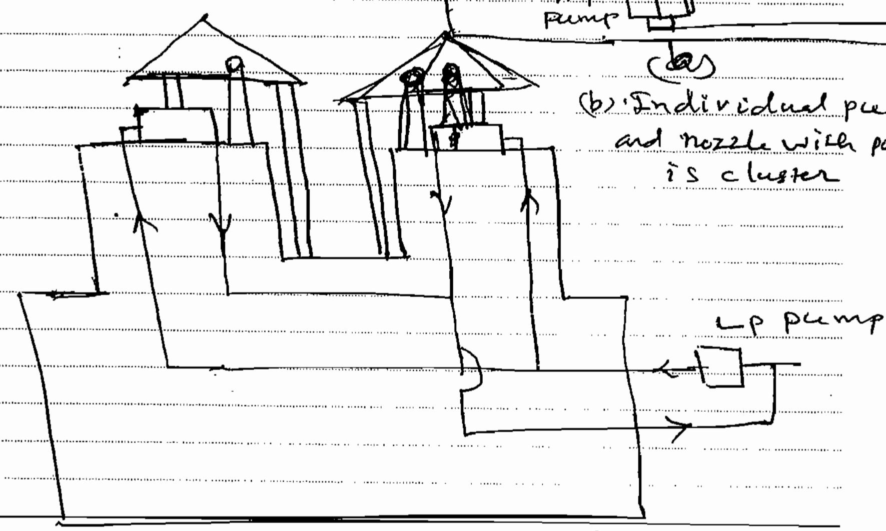


(a) Individual pump and nozzle with Seperated pump



(b) Individual pump and nozzle with pump is cluster



(c) Unit Injector System

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

Unit injector system.

The unit injector system fig 'c' is one in which the pump and the injector nozzle are combined in one housing. Each cylinder is provided with one of these unit injectors. Fuel is brought up to the injector by a low pressure pump, where at the proper time, a rocker arm actuates the plunger and thus injects the fuel into the cylinder. The amount of fuel injected is regulated by the effective stroke of the cylinder plunger. The pump and the injector can be integrated in one unit as shown in fig

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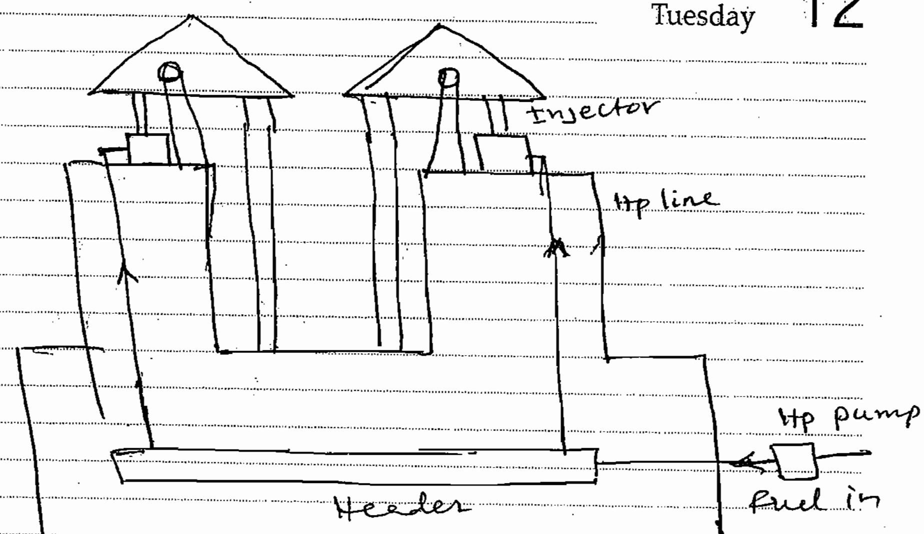
Common rail system.

In the common rail system, fig d, a HP pump supplies fuel, under high pressure, to a fuel header. High pressure in the header forces the fuel to ~~the~~ each of the nozzles located in the cylinders. At the proper time, a mechanically operated (by means of a push rod and rocker arm) valve allows the fuel to enter the proper cylinder through the nozzle. The pressure in the fuel header must be that, for which the injector system was designed, i.e it must enable to penetrate and disperse the fuel in the combustion chamber. The amount of fuel entering the cylinder is regulated by varying length of the push rod stroke.



Common rail system (d)

A high pressure pump is used for supplying to a header (from where the fuel is metered by injectors (assigned one per cylinder)).

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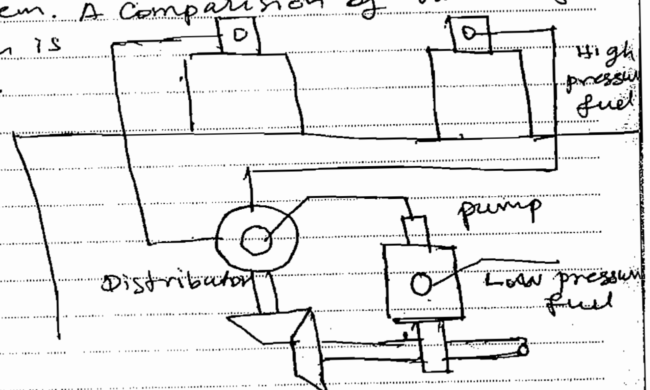
Distributer system.

A schematic diagram of a distributor system. In this system the pump which pressurizes the fuel also meters and times it. The fuel pump after metering the required amount of fuel supplies it to a rotating distributor at the correct time for supply to each cylinder. The number of injection strokes per cycle for the pump is equal to the number of cylinders.

Since there is ~~no~~ one metering element in each pump, a uniform distribution is automatically ensured. Not only that, the cost of the fuel-injection system also reduces to a value less

than two-thirds of that for individual pump system. A Comparision of various fuel injection system is given in table.



Schematic Diagram of Distributor system

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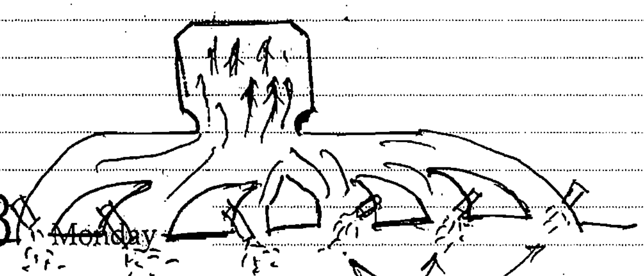
Job	Air injection system	Solid injection system		
		Individual pump	Common rail	Distributor
metering	pump	pump	Injector	pump
Timing	Fuel cam	pump cam	Fuel cam	Fuel cam
injection rate	spray valve	pump cam	spray valve	Fuel cam
Atomization	spray valve	spray tip	spray tip	spray tip
Distribution	spray valve	spray tip	"	"

petrol Fuel injection System.

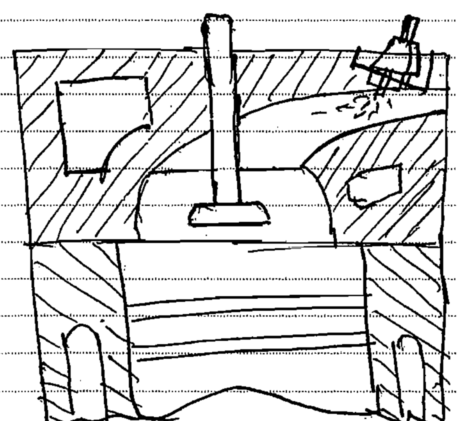
In the petrol injection system, the fuel is injected into the intake manifold through fuel injector valves. There are two types: 1, Multi-point FI, Monopoint FI system.

Multi-point Fuel injection (MPFI) System

It is also as port injection system. In this system, there is an injection valve for each engine cylinder as shown in fig. (a) Each injection valve is placed in the intake port near the intake valve as shown in fig. b. The main advantage of this system is that it allows more time for the mixing of air and petrol.



(A)



Port injection

(b)

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Electronic Fuel injection System for CI Engine.

In conventional diesel injection system, precise controlling of various parameters related to the injection process such as timing, rate of fuel injection, end of injection, quantity of fuel injected etc. is very difficult if the engine is operated at high speed. This ~~many~~ may result in lower efficiency and higher emission levels. Conventional systems only senses a few parameters and meter the fuel quantity or adjust the injection timing, therefore, electronically controlled diesel injection systems have been developed. This system facilitates the precise control of the following parameters.

(a) Quantity of fuel injection
(b) Injection timing
(c) Rate of injection during various stages of injection
(d) Injection pressure
(e) Speed of nozzle opening
(f) pilot injection timing and its quantity

Electronically controlled diesel injection system may use the following parameters which can significantly affect the performance of the engine as input.

(i) Intake air mass flow rate
(ii) Intake air temperature and pressure
(iii) Engine temperature

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(iv) Lubricating oil temperature
(v) Engine speed
(vi) crank shaft position
(vii) Turbo charger boost pressure
(viii) Accelerator pedal position
(ix) Exhaust gas oxigen level.

There are different types of electronically controlled diesel injection systems. They are as follows.
(i) unit injector system
(ii) Rotary distributor system
(iii) Common rail direct injection system

Unit injection system:-

This system is also called

as individual pump injection system. The unit injector system (UIS) combines the injection nozzle and the high pressure pump in a single assembly. One such unit injector is fitted in the head of each engine cylinder as shown in fig. The high pressure is built up by the activation of the pump plunger of the unit injector by the engine camshaft via a tappet or rocker arm.

The sequence operations as follows

(i) Fill phase:- The constant stroke pump element on the way up draws fuel from the supply duct in to the chamber, and as long as eletric solenoid valve remains de-energized fuel line is open.

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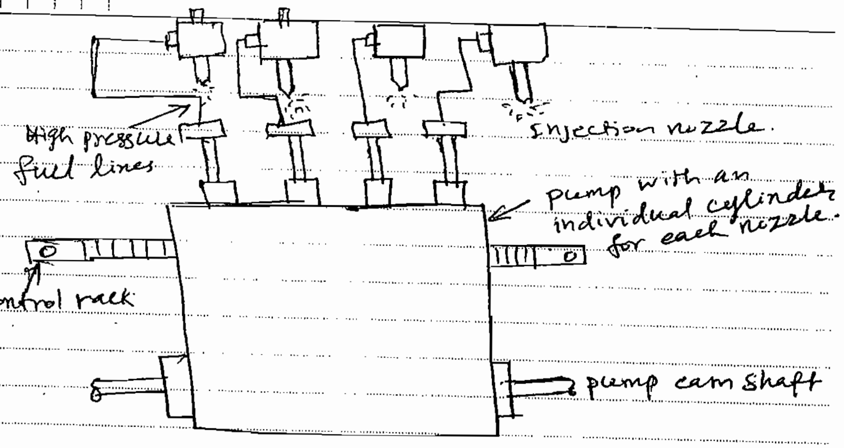

(ii) Spill phase.

The pump element is on the way down, and as long as solenoid valve remains de-energized the fuel line is open and fuel flows in through into the return duct.

(iii) Injection phase:-

The pump element is still on the way down. the solenoid is now energized and fuel line is now closed. The fuel cannot pass back into return duct, and is now compressed by the plunger until pressure exceeds specific "opening" pressure, and the injector nozzle needle lifts, allowing fuel to be injected into the combustion chamber.

(iv) pressure reduction phase:

The plunger is still on its way down, the engine ECU de-energizes the solenoid when required quantity of fuel is delivered, the fuel valve

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opens, fuel can flow back into return duct, causing pressure drop, which in turn causes the Injector nozzle needle to shut, hence no more fuel is injected.

Rotary distributor system

The fig shows a schematic diagram of the rotary distributor system. In distributor systems, the fuel is metered at a central point. A pump which pressurizes the fuel also meters the fuel and times the injection. The fuel pump after metering the required amount of fuel supplies it to a rotating distributor at the correct time for supply to each cylinder. The fuel is distributed to cylinders in correct firing order operated by poppet valves

 which are opend to admit fuel to the nozzles. Distributor pumps use control sleeves for metering the injected quantity. Thus they can be easily be made to work with an electronically controlled solenoid actuator.

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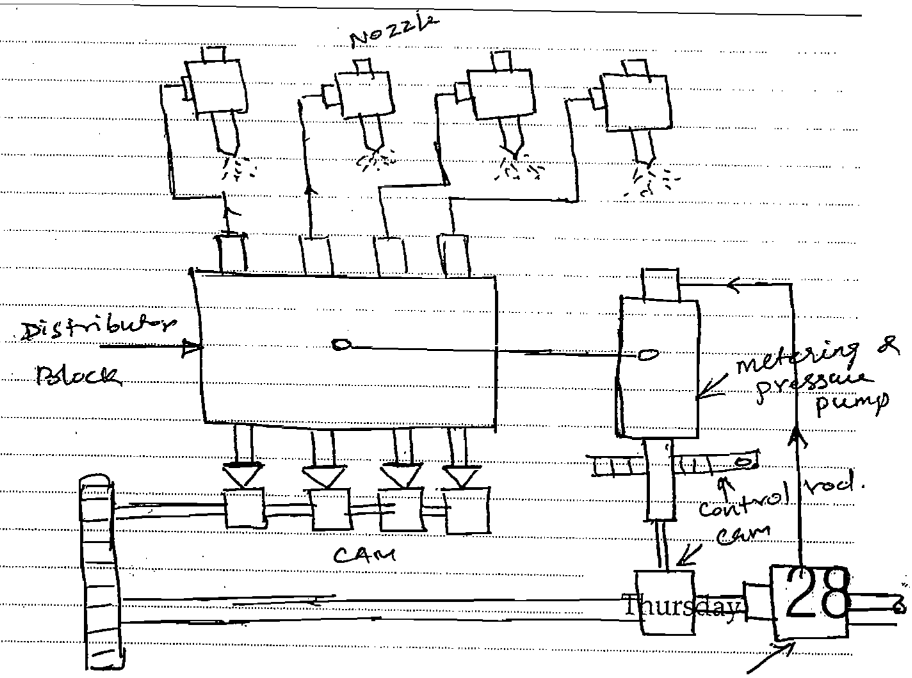

Distributor system

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IGNITION SYSTEM OF IC ENGINES

In IC engines, the ignition of fuel-air mixture should take place at the end of compression stroke. It ensures efficient and smooth running of an engine. There are mainly two different types of ignition used in IC Engines

1. Compression Ignition and
2. Spark ignition

2. Spark ignition:

Spark ignition is mostly used in gas engines, petrol engines and light oil engines working on otto cycle. In this system, the fuel-air mixture is ignited by a high-tension electric spark. Hence, they are known as spark ignition (SI) engines. There are different types of spark ignition system used in otto cycle engines.

(i) Coil Ignition system,
(ii) Magneto ignition system,
(iii) Electronic ignition system, and
(iv) Transistorised ignition system,

(i) Battery Ignition system or Coil ignition system:

It is employed in petrol engines. Fig shows the wiring diagram of a simple coil ignition system of a four cylinder engine. This system is used in automobiles.

Firing order: The sequence in which the firing or power occurs in a multi cylinder engine ~~IS known as firing order.~~

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Firing order: 4C- 13,42
1432
6C: 153624

Construction:-

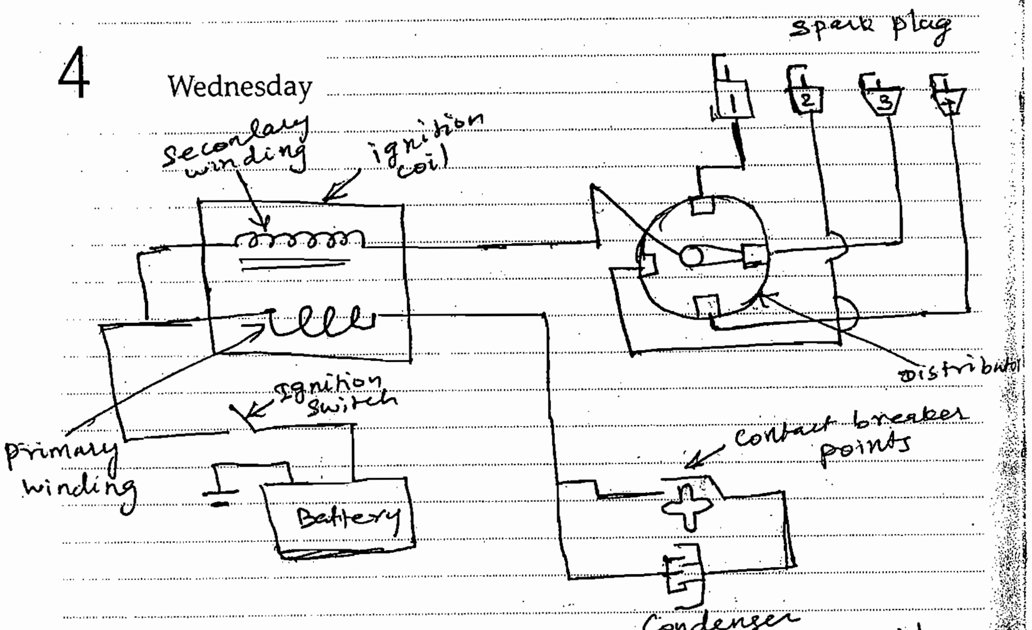
It consists of a battery, ignition coil, condenser, contact breaker, distributor and spark plug. Generally, 6 to 12 volts battery is used. The ignition coil consists of two windings such as primary winding ~~consists of a thick wire with less number of turns~~ and secondary winding. The primary winding consists of a thick wire with less number of turns. The primary winding is formed of 200 - 300 turns of thick wire of # 20-gage to produce a resistance of about 1.5 ohms.

The secondary winding located inside the primary winding consists of about 21,000 turns of thin enameled wire of # 38-40 gage with sufficiently insulated to withstand high voltage. It is wound close to the core with end one end connected to

the secondary terminal and the other end grounded either to the metal case or the primary coil. The condenser is connected across the contact breaker. It prevents the excess arcing and pitting of contact breaker points. The contact breaker is housed in the distributor itself. It makes and breaks the primary ignition circuit. The distributor distributes the high voltage to the respective spark plug having regular intervals in the sequence of firing order of the engine.

The spark plug is fitted in the combustion chamber of the engine. It produces the spark to ignite the fuel-air mixture. The rotor of the distributor and contact breaker cam are driven by the engine. There are two circuits in this system. one is the primary circuit. It consists of battery, primary coil of the ignition coil, condenser and contact breaker. The other circuit is the secondary circuit. It consists of secondary coil, distributor and spark plugs.

Working:-
The ignition switch is switched on and the engine is cranked. The cranking of the engine opens and closes the contact breaker ~~poi~~ point through a cam.



Battery Coil Ignition System or Coil Ignition System

When the contact breaker points are closed

1. The current flows from the battery to the contact breaker points through the switch and primary winding and then it returns to battery through the earth.

2. This current builds up a magnetic field in the primary winding of the ignition coil.

3. When the primary current is at the highest peak, the contact breaker points will be opened by the cam.

When the contact breaker points are opened

1. The magnetic field set up in the primary winding is suddenly collapsed

2. A high voltage (15000 volts) is generated in the secondary winding of the ignition coil.

3. This high voltage is directed to the rotor of the distributor.

4. The rotor directs high voltage to the individual spark plug in the sequence of the firing order of the engine.

5 - The high voltage tries to cross the spark plug gap (0.45 to 0.6mm) and the spark is produced. ~~This spark is produced~~. This spark ignites the fuel-air mixture.

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Advantages

* It provides better sparks at low speeds of the engine during starting and idling due to the availability of maximum current throughout the engine speed range.
* The initial cost is low as compared with magneto Ingnition system.
* The maintenance cost is negligible except battery

Disadvantages:

* Frequent Battery down occurs when the engine is not in use continously. The causes starting trouble
* The weight is greater than magneto ignition System.
* wiring mechanism is more compleacated.

8 Sunday

Magneto ignition system.

In this system, the battery is replaced with a magneto Fig shows the wiring diagram of a magneto Ignition system. It consists of a Switch, magneto, contact breaker, condenser, distributor and spark plugs. This system is used in two wheelers such as motor cycles, scooters etc.

Construction

The magneto ignition system consists of a rotating magnet assembly driven by an engine and a fixed Armature. The armature consists of primary and secondary windings. ~~The primary conecuit~~ consists of a ~~Switch~~ primary winding condenser

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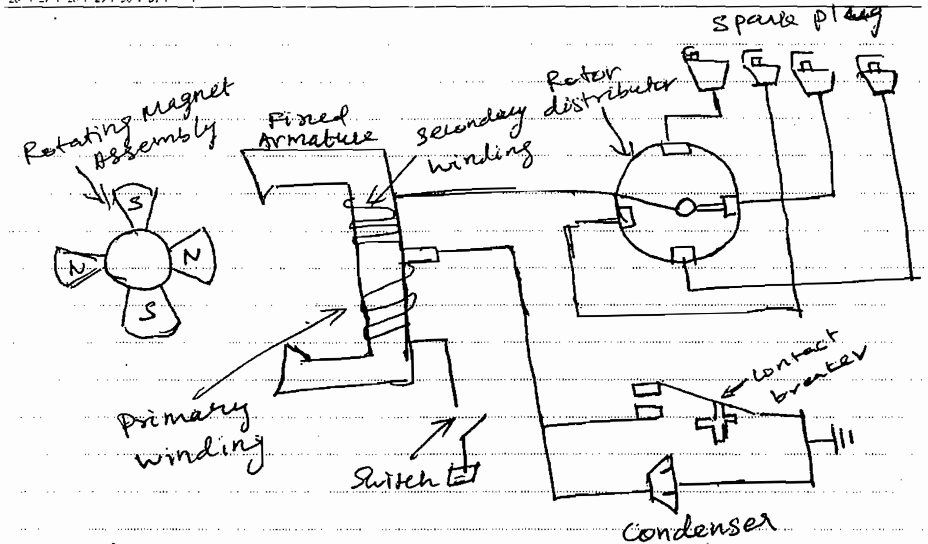

and contact breaker. The secondary circuits consists of a secondary windings distributor and spark plugs.

Tuesday 10

when the contact breaker points are closed.

1. The current flows in the primary circuit.
2. It produces a magnetic field in the primary winding.
3. When the primary current is at the highest peak, the contact breaker points will be opend by the cam

When the contact breaker points are opend.

1. There is a break in the primary circuit.
2. The magnetic field in the primary winding is suddenly collapsed

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3. A high voltage (15000 volts) is generated in the secondary winding

4. The high voltage is distributed to the respective spark plugs through the rotor of the distributor.

The high voltage tries to cross the spark plug gap and a spark is produced in the cap. This spark ignites the fuel-air mixture in the engine cylinder.

Electronic Ignition System:-

There are some drawbacks in above-discussed magneto ignitions system. Firstly, the contact breaker points will wear out or burn when it is operated with heavy current.

Secondly, the contact breaker is only a mechanic device which cannot precisely operate at higher speed due to insufficient dwell period for building up the magnetic field to its full value. The conventional contact breaker can give the satisfactory performance only at about 400 sparks per second which limits the engine speed.

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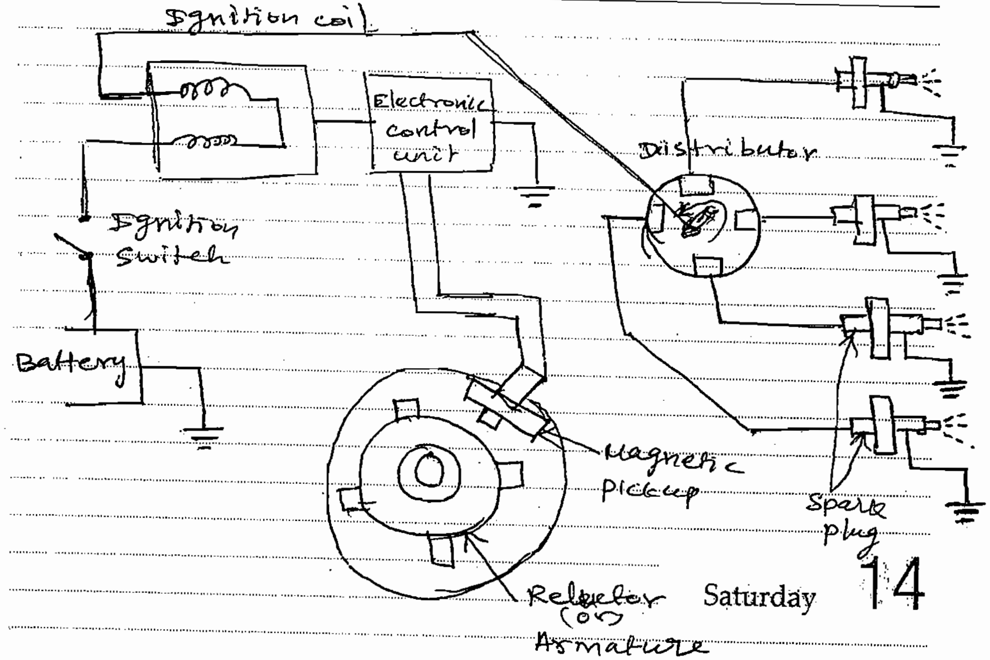

Electronic Ignition System

Construction:-

A schematic diagram of an electronic ignition system is shown in fig. It consists of a battery, ignition switch, electronic control unit, magnetic pick up, reluctor or armature, ignition coil, distributor and spark plugs. The construction of battery, ignition switch, ignition coil, distributor and spark plug is same as previous methods. In this system, a magnetic pick-up is used instead of contact breaker points in the conventional system. cam is also replaced by a reluctor or, armature.

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Working:-

When the ignition switch is closed (ie switch is ON) the reluctor rotates which makes the teeth of the reluctor come closer to the permanent magnet. It reduces the air gap between the reluctor tooth and sensor coil. Thus, the reluctor provides a path for the magnetic lines from the magnet. The magnetic field is passed onto the pick up every time when the reluctor teeth pass the pick up coil in which an electric pulse is generated. This small current then triggers the electronic control unit which stops the flow of battery current to the ignition coil.

The magnetic field in the primary winding collapses and the high voltage is generated in the secondary winding. It leads to the spark in spark plug via distributor. Meanwhile, the reluctor teeth pass past the pickup coil. There fore, the pulse unit is ended. It causes the electronic control unit to close the primary circuit.

Advantages:-

1. The parts such as reluctor, magnetic pick up and electronic control unit are not subjected to wear as incase of mechanical contact breaker.
2. The periodic adjustment of engine timing is not necessary.
3. It gives very accurate control of timing.

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COOLING SYSTEMS:-

Purpose of cooling:- When the air-fuel mixture is ignited and the combustion takes place at about 2500°c for producing power inside an engine, the temperature of the cylinder, cylinder head, piston and valve, continues to raise when the engine runs.

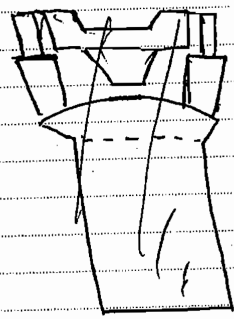
If the parts are not cooled by some means, then they are likely to get damaged and even melted. The piston may cease inside the cylinder. To prevent this, the temperature of the parts around the combustion chamber is maintained at 200°c to 250°c. Too much of cooling will lower the thermal efficiency of the engine. Hence the purpose of cooling is to keep the engine at its most efficient operating temperature at all engine speeds and all driving conditions.

Types of cooling system:-

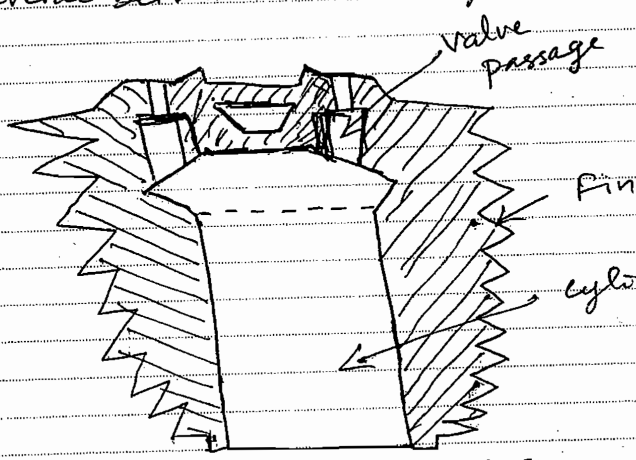
1. Air cooling or direct cooling
2. water cooling or indirect cooling
   (a) Thermosyphon cooling
   (b) pump circulation cooling
3. Liquid cooling
4. pressure sealed cooling
5. Steam cooling or evaporative cooling

Generally for automobile engines, air cooling and water cooling are used.

Air cooling.



The method of cooling an engine by the use of atmospheric air is called air-cooling. Generally, the two stroke engines are air cooled. The heat from inside the cylinder is spread over a large area of the outer surface of cylinder head and cylinder by providing fins as shown in fig. ~~If~~ a ~~foss~~ fan is used to supply a continuous air over the large finned surface, the heat can be quickly removed. More temperature difference between air and cylinder due to



cylinder with fins

good heat conductivity of the metal helps the air-cooling. The use of copper and steel alloys improves heat transfer.

Advantages

1. Light in weight since there are no radiators, cooling water and pipelines.
2. No coolant is used and so no leak and no ~~antifre~~ anti freeze required
3. Warming up is faster
4. Maintenance is easy and hence, it is cheaper.

Disadvantages:

1. It is less efficient since air is poor conductor of heat compared with water.
~~2. Since it is not possible to maintain maintenance is easy and hence~~

2. Since it is not possible to maintain even cooling some time distortion may take place
3. It produces more noise when it is operated
4. It can be used only in small engines.

• water cooling

In water-cooling, water is used for cooling the engine by circulating it through water jackets around each combustion chamber cylinder, cylinder head, valve and valve sheet, By absorbing heat, water will become hot. When it in again passed through radiator, it will be cooled by air blast due to forward motion of the vehicle as well as of this engine to absorb heat.

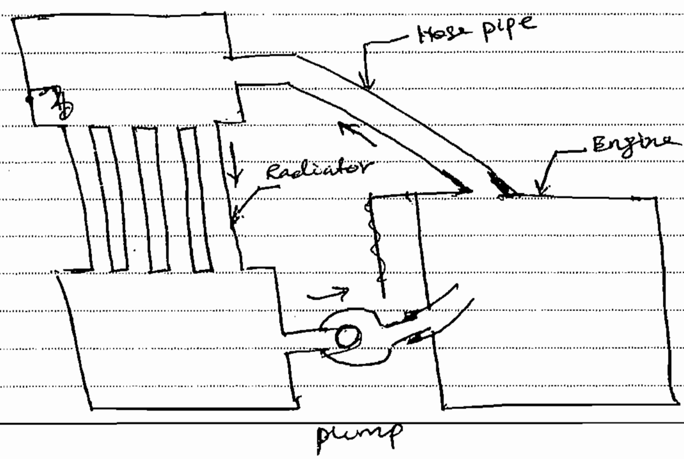
There are two systems of water-cooling

1. Thermosyphon system
2. pump circulation system

1. Thermosyphon system ✓
2. pump circulation system

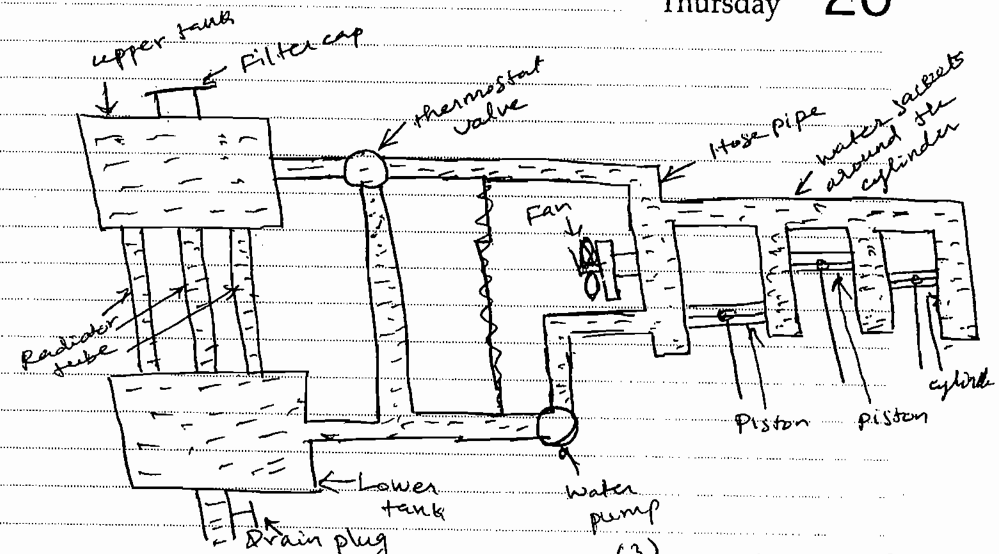
pump circulation system

②



To make the thermosyphon system more effective and improve water circulation, a water pump is introduced as shown in fig. which is driven by a V-belt from a pulley on the engine crank shaft. This is called pump circulation system.

The water-cooling arrangement for a 4 cylinder engine is shown in fig. When the hot water in engine passes through the radiator tubes from upper tank to lower tank, it is exposed to large amount of airflow and it sufficiently gets cooled. Then it is pumped to cylinder jackets by the water pump. The automatic thermostatic valve is used to regulate the circulation of water so that ~~was~~ very cold water will become hot in short time to improve the efficiency of engine.

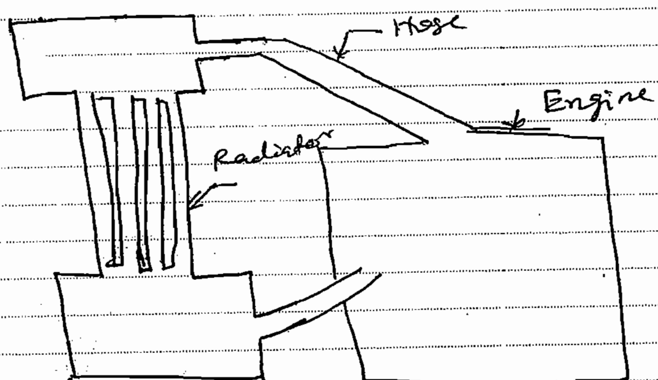


(3)
water-cooling system for 4-cylinder engine

Thermostat in cooling system:-

A Thermostat valve is used in the water cooling system to regulate the circulation of water in system to maintain the normal working temperature of engine parts at various operating conditions. The thermostat valve automatically works in the cooling system.

(1) Thermosyphon system



Thermosyphon cooling system

The principle of hot water going up and cold water coming down due to difference in density is used here. There is no pump to circulate water. The light hot water from the engine goes to the top of the radiator by itself and gets

cooled by the surrounding air and hence, it goes down to bottom of radiator and it again goes to engine cylinder as shown in fig. It is simple and cheap but cooling is low. water should be maintained ~~at~~ to a correct level at all time.

~~Methods of~~ Lubrication System in IC Engines.

In an IC engine, the moving parts rub against each other causing the frictional force. Due to the force, the heat is generated and the engine parts wear easily. The power is also lost due to friction. To reduce the power loss and also wear and tear of the moving parts, a foreign substance called lubricant is introduced in between rubbing surfaces. The lubricant keeps the mating surfaces apart.

Lubricant may be solid (graphite) or semi-solid (grease) or liquid (oil). The liquid lubricant used is generally mineral oil. This is obtained by refining petroleum. Grease is also used to lubricant certain parts parts of the engine.

Properties of Lubricant

The lubricant used in IC engine should have some properties for the successful performance of the engine. The properties required for a good lubricant are listed below.

(1) Viscosity:-

It is defined as the measure of fluid resistance to flow, Viscosity of the lubricant is ~~decrease as the~~ depending upon its temperature. Viscosity of lubricant decreases as the temperature increases and vice versa.

(ii) oilness:-

It is the property of an oil to spread and attach itself firmly to bearing surfaces. In general, high oiliness is required for a better lubrication.

(iii) Flash point.

Flash point of the lubricant is the temperature at which it forms vapours and it produces combustible mixture with air. The high flash point is always desirable because low flash point leads to burning of lubricant. The minimum flash point of lubricating oil used in IC engine varies from 200 to 250°C

(iv) Fire point:

The fire point is the lowest temperature at which the fuel burns continuously. The fire point of oil should be higher than the flash point.

(v) Volatility:-

When the lubricating oil is exposed to a high temperature for long time, it may evaporate. This property is known as volatility. The lubricating oil should have low volatility.

pour point

It is defined as the temperature below which oil will cases to flow in the pipeline under controlled test conditions. Lubricants having lesser pour point are always recommended as its flow will start even when the engine is started in cold weather conditions.

Neutralization → I will have corrosive action on the parts of engine.

Foaming:-

It is the condition in which minute air bubbles are held in the oil. It will reduce the mass flow and also accurate the oxidation process. Therefore, the oil should force from foaming trouble.

Emulsification:

The lubricating oil should not form an emulsion when it is brought in contact with water, good lubricating oil must not emulsify easily.

Methods of lubrication:-

(i) petroil lubrication (or) mist lubrication system
(ii) Wet sump system, and
(iii) Dry sump system.

(i) petroil system

This method is used in light vehicles such as motor cycles and scooters. About 3 to 6% of lubricating oil is mixed with petrol in the fuel

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tank. Here there is no separate sump and pump. The oil mixing with petrol acts as a lubricant.

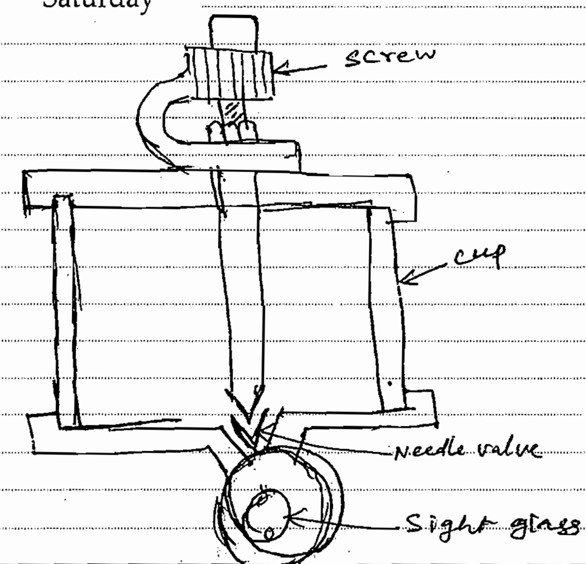
Wet Sump system :-

In this method, the lubrication oil is stored in the oil sump. From the oil sump, the oil is supplied to various parts of the engine.

This system may be further classified as

(a) Gravity lubrication system
(b) splash lubrication system
(c) pressure lubrication system
(d) semi-pressure lubrication system

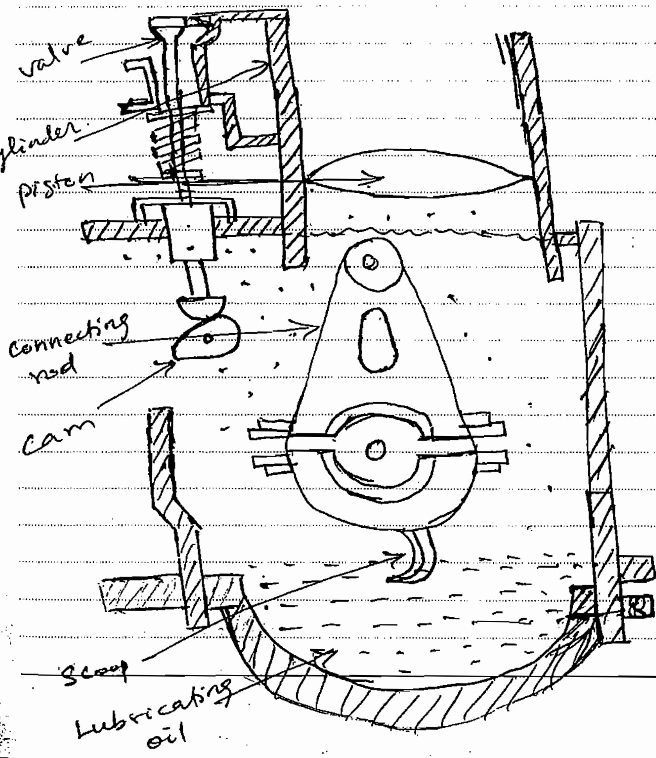
(a) Gravity lubrication system :



Drop feed oiler

In this method, oil is supplied to the parts to be lubricated by means of gravity. This system uses a drop feed oiler shown fig. It consists of a cup and needle valve arrangement. The needle valve is operated by means of a screw. The valve is raised to increase the flow of oil and lowered to decrease the oil flow. This system is used for lubricating the external moving parts such as bearings, cross head, crank pins of simple steam engine.

(b) Splash lubricating system:-



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In this system oil is stored in the crank case. A small scoop is attached with the big end of connecting rod as shown in fig. when the crank is rotated, the scoop dips in the oil and splashes the oil. The oil is splashed on ~~the~~ cylinder wall, connecting rod ends and valve mechanisms. This method is used in some motorcycles and single cylinder stationary engines. Greater care should be taken that the oil in the crank case is filled up to the desired mark. There will be in sufficient lubrication when the oil level is low.

Disadvantage

* It is not efficient if the bearing loads are heavy
* It is very difficult to introduce oil in the minute gaps between the sliding surfaces.

pressure lubricating system:

In this system, the lubricating oil is forced under pressure by a pump at a pressure of 2 to 4 MPa. Fig. shows a line diagram of this system. It consists of oil sump, oil pump, oil gallery, pressure release valve, oil filter, oil pressure gauge and dipstick. The lubricating oil from the sump and filter ~~valve~~ are always immersed in the oil.

From the oil gallery, the oil is distributed under pressure to various parts of the engine to be lubricated by oil tubes. oil from gallery enters the crank pin bearing through a taper hole in the crank shaft. A through hole is provided at the centre of the connecting

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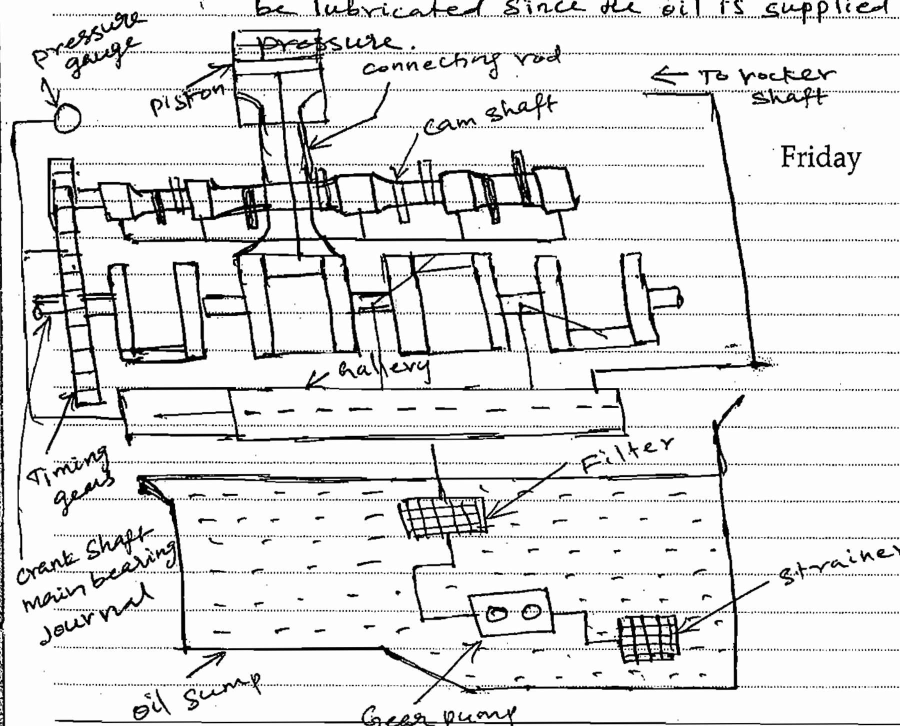
rod. The oil from the big end bearing enters the gudgeon pin bearing (small end bearing) through the hole in the connecting rod. Separate oil tubes carry oil for lubricating timing gears, rocker arm assembly, cam shaft etc. Another oil line is connected to the pressure gauge to show the pressure of the oil.

Advantages:-

* All parts of the engine are efficiently lubricated
* The minute gap between sliding surfaces can be lubricated since the oil is supplied under pressure.



pressure lubricating system.

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Major properties of fuel. (Petrol)

1. volatility
2. knock resistivity
3. sulphur content
4. gum content and
5. Contamination etc.

1. volatility :-

The volatility of a liquid is general is its tendency to evaporate under a given set of Conditions.

The consitituents of gasoline, which is a mixture of many hydrocarbons, boil off at a wide range of temperatures.

2. sulphur content :-

sulphur contents troubles in three way 1. corrosion, 2. odour, 3. poor explosion characteristics of gasoline fuel.

3. Gum content :-

All petroleum motor fuels oxides slowly in presence of air. The oxidation of unsaturated hydrocarbons (also unstable sulphur and nitrogen compounds) results in formation of resinous materials called gum.

oxidized gasoline shows a loss of antiknock quality. High gum content fuels may clog carburettors jets, promote

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sticking of automatic chokes, sticking of the intake valves, piston rings, and promote formation of manifold deposits, reducing volumetric efficiency.

23 2018 August Thursday

24 Friday

August 2018 Saturday 25

Unit - II -

Normal combustion and abnormal combustion is SI engines - Importance of flame speed and effect of engine variables - Abnormal combustion, pre ignition and knocking in SI Engines - Fuel requirements and fuel rating, anti knock additives - Combustion chambers ~~and fuel injection~~ requirements, types of SI Engine.

Four stages of combustion in CI engines - Delay period and its importance - Effect of engine variables - Diesel knock - Need for air movement, suction, compression and combustion induced turbulence in Diesel engine - open and divided combustion chambers and fuel injection - Diesel fuel requirements and fuel rating.

Sunday 26

Normal and abnormal Combustion.

Normal combustion is the process where the fuel is burned layer by layer i.e. in a wave from by the help of spark only. And abnormal combustion is the process where the fuel burns not only by spark but also by self ignition process where a no. of waves of flames are produced from different parts of cylinder.

SI- Engine Normal Combustion

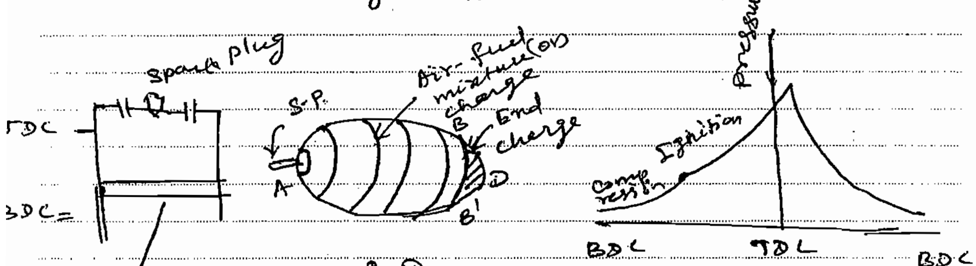

fig ①

Stages of combustion.
1. Ignition Lag
2. propagation of flame
3. After burning

Abnormal Combustion.

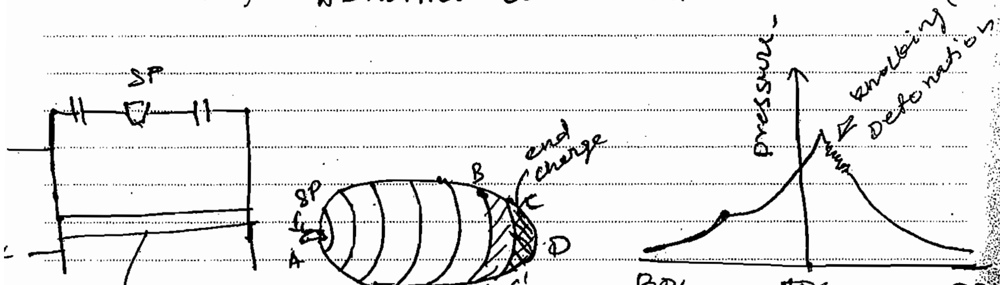

fig ②

Combustion with Detonation.

Knocking

If the temperature of the unburnt mixture exceeds the self-ignition temperature of the fuel and remains at or above this temperature during the period of preflame reactions (Ignition lag), spontaneous ignition or autoignition occurs at various pin-point locations. This phenomenon is called knocking. The process of autoignition leads towards engine knock.

The phenomenon of knock may be explained by reffering the above figures. The fig① shows the cross section of the combustion chamber with flame advancing from the spark plug location A without knock whereas fig ② shows the combustion process with knock. In the normal combustion the flame travels across the combustion chamber from A

towards D, The advancing flame front compresses the end charge BB'D farthest from the spark plug, thus raising its temperature. The temperature is also increased due to heat transfer from the hot advancing flame-front. Also some preflame oxidation may take place in the end charge leading to further increase in temperature. In spite of these factors if the temperature of the end charge had not reached its self-ignition temperature. the charge would not autoignite and the flame will advance further and consume the charge BB'D, this is the normal combustion process. which illustrated in by means of the pressure - time ~~dig~~ diagram.

However if the end charge BB'D reaches its autoignition temperature and remains for some length of time equal to the time of preflame reactions the charge will autoignite, leading to knocking combustion. in fig ② It is assumed that when flame has reached the position BB', the charge ahead of it has reached critical auto ignition temperature. During the preflame reaction period if the flame front could move from BB' to only CC' then the charge ahead of CC' would autoignite.

Because of the autoignition, another flame front starts traveling in the opposite direction to the main flame front. When the two flame fronts collide, a severe pressure pulse is generated. The gas in the chamber is subjected to compression and rarefraction along the pressure pulse until pressure equilibrium is restored. vibration frequency in order of 5000 cps.

Importance of flame speed (or) Factors influencing flame speed ~~and Effect of Engine variables~~.

* The flame is started by the spark at the spark plug terminals. The flame spreads from there to the remotest points of the combustion chamber
* At any instant, the flame has a definite front or boundary (surface area) called flame front. The flame front seperates the burned charge from the unburned charge.
* The speed which the flame front travels affects combustion phenomena development of pressure and production power.
* The time between mixture ignition and complete combustion is roughly two milliseconds.

Factors influencing flame speed.

The following factors affects the speed of the flame front, i.e., flame velocity, within the combustion chamber.

1. Inlet pressure and Temperature
2. Turbulence (prevailing and temperature condition)
3. Engine speed (crank shaft rotational)
4. Residual gas content [products of combustion left in the cylinder at the end of exhaust process]
5. Compression ratio
6. Spark timing
7. Mixture strength (or) Air-Fuel ratio
8. Fuel (physical and chemical) properties.

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1. Inlet pressure and temperature :-
Flame speed increases with an increase in intake pressure and temperature. A higher initial pressure and temperature may help to form a better homogeneous air-vapour mixture which helps in increasing the flame speed.

2. Turbulence :-
The flame speed is quite low in non-Turbulance mixtures and increase with increasing turbulence. A suitable design of the combustion chamber which involves the geometry of cylinder head and piston crown increases the turbulence during the compression stroke.

3. Engine speed.
The flame speed increases almost linearly with engine speed since the increase in engine speed increases the turbulence inside the cylinder. The time required for the flame to traverse the combustion chamber (or) space is reduced considerably (1/2 of the time)

4. Residual gas.

5. Compression ratio :-
A higher compression ratio increases the pressure and temperature of the working

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mixture which reduce the initial propagation phase of combustion and hence less ignition advance is needed. Engines having higher compression ratios have higher flame speed.

6. Air-fuel ratio :- The fuel-air ratio has a very significant influence on the flame speed. The highest flame velocity & minimize the time for complete combustion. Very rich mixtures lead to incomplete combustion which results again in the release of less thermal energy. less thermal energy released in the case of lean mixtures ~~and~~ due to lower flame speed.

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Rating of fuels in SI Engine.

Generally fuels rating is given for their antinock qualities. The rating of fuel is done by defining two parameters called octane number and cetane number for gasoline and diesel oil respectively. The rating of spark-Ignition (SI) engine fuels and rating of compressed ignition (CI) engines fuels discussed

Rating of SI Engine Fuels.

Resistance to knocking is an extremely important characteristic of fuel for spark ignition engines. These fuels differ widely ~~knock~~ in their ~~knock~~ ability to resist knock

depending on their chemical composition. A satisfactory rating method for comparing the antiknock qualities of the various fuels has been established. In addition to the chemical characteristics of hydrocarbons in the fuel, other operating parameters such as engine speed, fuel-air ratio, ignition timing, dilution, shape of the combustion chamber, compression ratio, ambient conditions, etc. affect the tendency to knock in the engine cylinder. Therefore, in order to determine the knock resistance characteristic of the fuels, the SI engine and its operating variables must be fixed at standard values.

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The rating is given an octane number:-

* The fuel under test is compared with a mixture of iso-octane (high rating) and normal heptane (low rating), by volume. The octane number of the fuel is the percentage of octane in the reference mixture which knocks under the same conditions as the fuel. The number obtained depends on the condition of the test and the two main methods in use (the research and motor methods) give different ratings for the same fuel. The motor test is carried out at the higher temperature and gives the lower rating. The difference between the two is taken as the measure of the temperature sensitive of

the fuel.

* High octane fuels (up to 100) can be produced by refining techniques, but it is done more cheaply, and more frequently, by the use of anti-knock additives, such as tetraethyl lead. (An addition of $1.1 cm^3$ of tetraethyl lead to one litre of 80 octane petrol increases the octane number to 90). Fuels have been developed which have a higher anti-knock rating than iso-octane and thus has lead to an extension of the ~~octane~~ ~~of~~ octane scale.

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Advantages of high octane fuel.

1. The engine can be operated at high compression ratio and there fore, with high efficiency without detonation

2. The engine can be supercharged to high output without detonation

3. optimum spark advance may be employed raising both power and efficiency.

~~Cont~~ Effect of Engine Variables on knock.

1. Density factor :- Any factor which reduces the density of the charge tends to reduce knocking parameters which are directly ~~or~~ indirectly connected with temperature, pressure and density.

2. Fuel choise : A low self ignition temperature promotes knock.

3. Engine speed :- Low engine speeds will give low turbulance and low flame velocity (speed)

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knock may occur at low speed.

4. Induction pressure :- Increase of pressure decreases the self ignition temperature and the inducti period knock will tend to occur at full thro

5. Ignition timing :- Advanced ignition timing increases peak pressures and promote knock.

6. Mixture strength :- optimum mixture strength gives high pressure and promote knock

7. Compression ratio :- High compression ratios increase the cylinder pressures and promote knock.

8. Combustion chamber design :- poor design gives long flame paths poor turbulence and insufficient cooling all of which promote knock.

9. cylinder cooling :- poor cooling raises the mixture temperature and promotes knock

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Combustion Chamber Requirements

* The design of the combustion chamber for an SI engine has an important inflence on the engine performance and its knocking tendencies.

* The important requirements of an SI engine combustion chamber are to provide
→ high power output with minimum octane requirement
→ high thermal efficiency and
→ smooth engine operation.

Combustion chamber design:-
The design involves the shape

~~of~~ of the combustion chamber, the location of spark plug and the location of inlet and exhaust valves.

→ Combustion chambers are usually designed with necessary possible attempts made to meet the following objectives.

1. To regulate the rate of pressure rise such as the greatest force is applied to the piston as closely after TDC on the 'power stroke' as possible, with a gradual decreases in the force on the piston during the power stroke. The forces must be applied to the piston smoothly, however thus placing a limit on the rate

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of the pressure rise, as well as the position of the peak pressure with respect to T.D.C.

2. To prevent the possibility of knocking in all times.

To obtain these objectives attempt ~~is~~ made to design the combustion chambers with the following factors in mind.

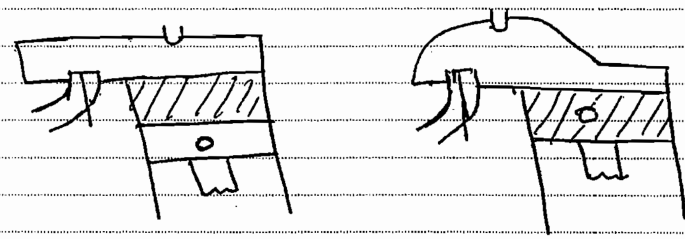
(i) To achieve the highest possible flame front velocity through the creation of high turbulences of the minute 'swirl' type.

(ii) To burn the largest mass of the charge as soon as possible after ignition

(consistance with smooth operation of the engine), with progressive reduction in the mass of the charge burnt toward the end of the combustion.

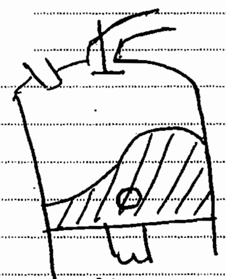
(iii) To reduce the possibility of knock by reducing the temperature of the last portion of the charge to burn, through the application of a high surface to volume ratio in that part of the combustion chamber where this portion burns. Such a ratio increases the heat transfer to the combustion chamber walls and thereby tends to reduce the temperature of the final unburnted charge.

→ Reducing the distance of the flame to travel by centrally locating the spark plug, or in some engines, by using dual spark plugs.



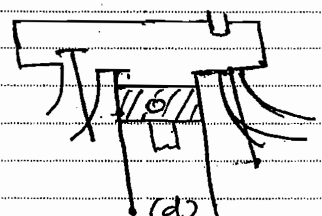
(a) (b)

L-Head Types



(C)

I-Head



(d)

T-Head

The above chambers are designed to obtain the following:

→ A high combustion rate at the start

→ A high surface to volume ratio near the end of burning

→ A rather centrally located spark plug.

Stages of Combustion in CI Engines.

The Combustion in a CI engine is considered in four stages.

1. Ignition delay period
2. The period of rapid combustion
3. The period of controlled combustion
4. The period of after burning

1. Ignition delay period.

The fuel does not ignite immediately upon injection in to the combustion chamber. There is a definite period of inactively between the time when the first droplet ~~bet~~ of fuel hits the hot air

in the combustion chamber and the time it starts through the actual burning phase. This period is known as the ignition delay period. In fig the delay period is shown on pressure crank angle (or time) diagram between points a and b, point a represents the time of injection and point b represents the time at which the pressure curve (caused by combustion) first seperates from the motoring curve. The ignition delay period can be divided into two parts,

(i) physical delay (ii) chemical delay

(i) The physical delay is the time between the beginning of injection and the attainment of chemical reaction conditions. During this period, the fuel is atomized, vaporized, mixed with air and raised to its self ignition temperature

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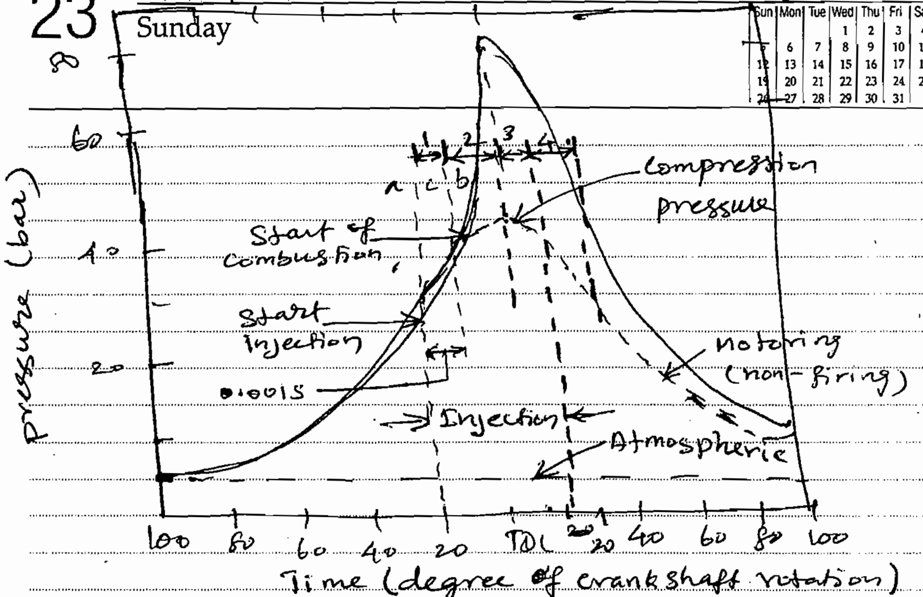

Stages of Combustion in CI Engines

(ii) During the chemical delay, reaction starts Slowly and then accelerate until inflammation or ignition takes place. Generally, the chemical delay is larger than the physical delay. However it depends on the temperature of the Surroundings and at high temperatures the chemical reactions are faster. and the physical delay longer than the chemical delay. for CI engines.

2. Period of Rapid Combustion

The period of rapid combustion also called the uncontrolled combustion, is that phase in which the pressure rise in rapid. During

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the delay period, the droplets have had time to spread over a wide area and fresh air is always available arround the droplets.

* The period of rapid combustion is counted from end of delay period or the begining of the combustion to the point of maximum pressure on the indicator diagram. the rate of heat release is maximum during this perio

3. Period of controlled combustion.

At the end of second stage of combustion the temperature and pressure are so high, that the fuel droplets injected in the third stage burn almost as they enter and any farther pressure rise can be controlled by purely mechanical means

i.e by the injection rate. The period of controlled combustion is assumed to end at maximum cycle temperature.

4. After burning.

The combustion continues even after the fuel injection is over, because of poor distribution of fuels particles. This burning may continue in the expansion stroke upto 70° to 80° of crank travel fro T.D.C. This continued burning is called the after burning, may be considered a a fourth stage of combustion.

Factors affecting the delay period (or) Factors affecting combustion in CI engines:-

I compression ignition combustion the length of the delay period plays a vital role. This period serves useful purpose in that it allows the fuel jet to penetrate well into the combustion space and the complete combustion is takesplace. If delay period is too long the amount of fuel available for simultaneous explosion is too great and the resulting pressure rise is to rapid.

~~Delay is reduced by the following~~

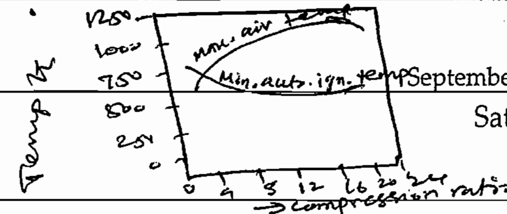
~~→ High charge temperature~~

Factors affecting the delay period.

Many design and operating factors affect the delay period. The imortants are

(1) Compression ratio

The increase in the compression temperature as well as the decrease in the minimum autoignition temperature decreases the delay period. The peak pressure during the combustion process is only marginally affected by the compression ratio (because delay period is shoter with higher compression ratio and hence the pressure rise is lower)



(ii) Engine speed.

Decrease in delay period in terms of milliseconds with increase in engine speed in a varoable speed operation with a given fuel

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

(iii) output.

With an increase in engine output the air-fuel ratio decreases, operating temperature increase and hence delay period decreases.

(iv) Atomization and duration of injection.

Higher fuel-injection pressure increase the degree of atomization. The fineness of atomization reduces ignition delay, due to higher surface volume ratio.

(v) Injection Timing:

As the pressure and temperature at the begining of injection are lower for higher ignition advance the delay period increases with increase in injection advance.

(vi) Quality of fuel.

Self ignition temperature is the most important property of the fuel which affects the delay period

Fuels with higher cetane number give lower delay period and smoother engine operation. Other properties of the fuel which affect the delay period are volatility, latent heat, Viscosity and surface tension.

(VII) Intake pressure :-

Increase in intake pressure or super-charging reduces the autoignition temperature and hence reduce the delay period. The intake pressure increases with the higher compression ration

(viii) Intake temperature.

Increase in intake temperature increases the compressed air temperature which is reduces delay period.

Fuel requirements is S-I Engine

1. It should mix readily with air and afford uniform manifold distribution, ie it should easily vaporise.

2. It must be knock-resistant.
3. It should not pre-ignite easily
4. It should not tend to decrease the volumetric efficiency of engine
5. It should be easy to handle
6. It must be cheap and should be available everywhere

7. It must burn clean and produce no corrosion etc., on engine parts
8. It must have high calorific value
9. It should not form gum and varnish.

Diesel knock

If the delay period is long a large amount of fuel will be injected and accumulate in the chamber. The auto-ignition of this large amount of fuel may cause high rate of pressure rise and high maximum pressure which may cause knocking in diesel engine

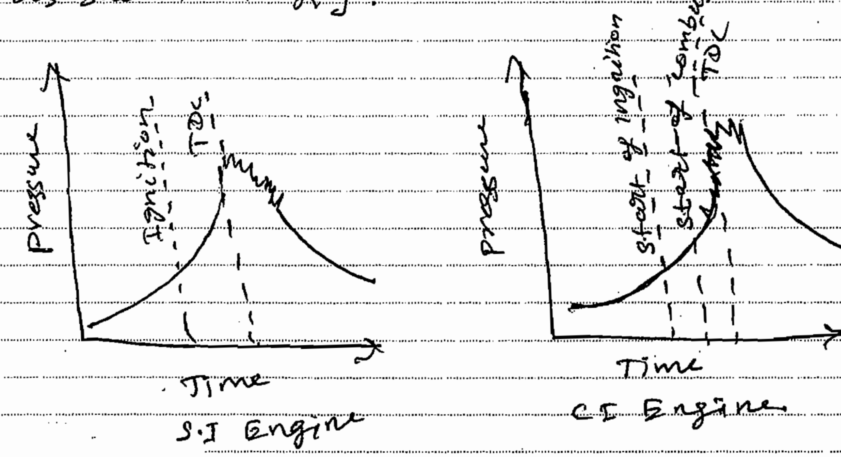
→ A long delay period is also improves the homogeneity of the fuel-air mixture and its chemical preparedness for explosion type self-ignition similar to detonation (or) knock in SI Engine.

Comparision of knocks in CI Engine with SI E

It is very instructive to compare the phenomenon of detonation (knock) in SI engines with that of knocking in CI Engi[ne]. There is no doubt that these two phenomena are fundamentally similar. B[oth] are process of auto-ignition subject to t[he] ignition time-lag charecteristics of the fuel-air mixture.

1. In the SI engine, the detonation (knock) occurs near the end of combustion

where as in CI engine detonation (knock) occurs ~~no~~ near the begining of combustion as shown in fig.



2. The knock in the SI engine is of a homogeneous charge causing very hogh rate of pressure rise and very high maximum pressure. In the CI engine the fuel and air are imperfectly mixed and hence the rate of pressure rise is normally lower than that in the detonation part of the charge in the SI engine.

3. Since in the the CI engine the fuel is injecte into the cylinder only at the end of compression stroke there is no pre-ignition (or) premature ignition, as in the SI engine

4. In the SI engine it is relatively easy to distinguish between knocking and non-knocking operation as the human ear easily finds the distinction. However, in the case of the CI engine the normal ignition is itself by auto-ignition and hence most CI engines have a sufficiently high rate of pressure rise per degree of crank angle to cause audible noise.

Factors tending to reduce knocking in SI and CI engines.

S.No	Factors	SI engine	CI Engine
1	Self-Ignition temperature	High	Low
2	Time lag or delay period	Long	short
3	Compression ratio	Low	High
4	Inlet temperature	low	High
5	Inlet pressure	Low	High
6	Combustion chamber wall temp.	Low	High
7	Speed	High	Low
8	Cylinder size	Small	Large

When the compressibility of air is negligible, then $V_1 = 0$,

therefore, $C_2 = \sqrt{2\left(\frac{P_1 - P_2}{\rho_a}\right)}$

$$m_a = A_2\sqrt{2\rho_a(P_1 - P_2)}$$

$$(m_a)_{actual} = C_{da}\cdot A_2\sqrt{2\rho_a(P_1 - P_2)}$$

$$\text{Air - fuel ratio} = \frac{m_a}{m_f}$$

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

Unit - III

Peformance calculations of IC Engines.

An Internal combustion Engine must be tested after designing and Manufacturing.

The purposes of testing are:-

1. To determine thermal η of the engine at various Load
2. To determine the specific fuel consumption of various Loads and speeds.
3. To determine power developed by the engine
4. To Confirm data used in design.
5. To prepare heat Balance sheet.
6. To satisfy the customer regarding the performance of the engine.

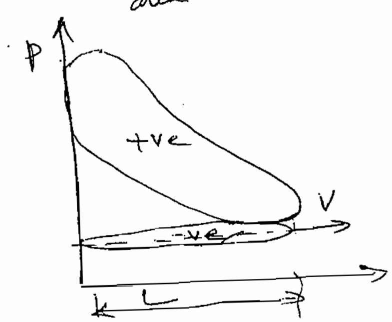
Two Types of Test

1. Commercial Test ← Lubricating oil consumption, Valve and port timing, cooling water consumption, overload carrying capacity
2. Thermodynamic Test.

(i) Indicated Mean effective pressure
(ii) Indicated power.
(iii) Speed of the engine.
(iv) Brake power.
(v) Mechanical Losses.
(vi) Mechanical efficiency
(vii) Fuel consumption.
(viii) Air consumption.
(ix) thermal efficiency
(x) Volumetric efficiency.
(xi) Heat Balance Sheet.

1. Measurement of Indicated power.

area measured by planimeter -



Indicated mean effective pressure (P_m)

It is defined as the algebraic sum of the mean pressure acting on the piston during one complete cycle. It is obtained from the indicator diagram drawn with the help of an engine indicator.

An indicator diagram is the graphical representation of pressure volume variation during working.

The area of the indicator diagram is measured by planimeter.

Let, $A_p \Rightarrow$ Area of positive loop.

$A_n \Rightarrow$ Area of negative loop.

$L \Rightarrow$ Actual length of the diagram

Mean height of the indicator diagram,

$$h = \frac{\text{Area of the indicator diagram}}{\text{Base length of the indicator diagram}}$$

$$= \frac{A_p - A_n}{L}$$

Indicated mean effective pressure,

P_m = mean height × Spring scale (or) Spring index

$= h \times S$.

$$P_m = \frac{A_p - A_n}{L} \times S \text{ bar (or) } kN/m^2$$

$S = S_1 - S_2$

Indicated power (IP.)

Indicated power is the rate of work done by the products of combustion on the piston. It is the power actually developed by the engine cylinder.

$$IP = P_m \cdot l \cdot a \cdot n \cdot k \text{ kW}.$$

$P_m \Rightarrow$ Indicated mean effective pressure in kN/m^2

$l \Rightarrow$ Length of the stroke in m

$a \Rightarrow$ Area of the piston in m^3

$n \rightarrow$ Number of working stroke per cycle

$k \rightarrow$ Number of cylinders.

Efficiencies of I.C Engines: – Mechanical η

(a) It is the ratio of Brake power to the indicated power. Mechanical efficiency.

$$\eta_{mech} = \frac{\text{Brake power}}{\text{Indicated power}}$$

$$= \frac{BP}{IP}.$$

(b.) Indicated thermal efficiency (η_{IT})

It is the ratio of indicated power to the heat supplied to an engine.

$$\eta_{IT} = \frac{\text{Indicated power}}{\text{Heat supplied}}.$$

$$\text{Heat supplied per sec} = \frac{m_f \times CV}{3600} \text{ in kW}$$

Where, $M_f \Rightarrow$ Mass of fuel consumed per hour

$CV \Rightarrow$ Calorific value of fuel kJ/kg

$$= \frac{IP}{\left(\frac{m_f \times CV}{3600}\right)}$$

c. Brake thermal efficiency (η_{BT})

It is the ratio of Brake power to the heat supplied to an engine.

$$\eta_{BT} = \frac{\text{Brake power}}{\text{Heat supplied}}.$$

$$\eta_{BT} = \frac{BP}{\left(\frac{m_f \times CV}{3600}\right)}$$

d. Relative efficiency (η_R)

It is also known as efficiency ratio.

It is the ratio of indicated thermal η to the Air Standard η.

$$\eta_R = \frac{\text{Indicated thermal } \eta}{\text{Air Standard } \eta}$$

e. Volumetric efficiency (η_V)

$$\eta_V = \frac{\text{Volume of charge inducted during suction stroke}}{\text{Swept volume of the piston}}.$$

· Specific fuel consumption (SFC)

It is defined as the amount of fuel consumed per brake power per hour of work, It is called as brake SFC.

$$BSFC = \frac{m_f}{BP} \text{ kg / kW-hr}$$

Brake thermal efficiency

$$\eta_{BT} = \frac{BP}{\left(\frac{m_f \times CV}{3600}\right)} = \frac{3600}{BSFC \times CV}$$

$\frac{BP}{m_f} = \frac{1}{BSFC}$

$$\therefore BSFC = \frac{3600}{\eta_{BT} \times CV}$$

The amount of fuel consumed per indicated power per hour of work is known as indicated SFC.

$$ISFC = \frac{m_f}{IP}.$$

Indicated thermal efficiency,

$$\eta_{IT} = \frac{IP}{\left(\frac{m_f \times CV}{3600}\right)} = \frac{3600}{ISFC \times CV}$$

$$\therefore ISFC = \frac{3600}{\eta_{IT} \times CV}.$$

Brake power →

$$BP = 2\pi NT$$
$$= 2\pi NWR$$

$T = WR$. (kNm)

W – Net load on brake (kN)

R – Effective radius of brake drum (m)

N – speed of the engine rps.

$$BP = 2\pi NT$$
$$= 2\pi NWR.$$

$R = \frac{D_1 + d_1}{2}$

$W = W_1 - W_2$

Weight applied in kN

spring balance reading

Heat Balance Sheet :-

The following values should be calculated for tabulating heat balance sheet.

(i) Heat supplied by the fuel. (Q_s)

$$Q_s = m_f \times CV \ kJ/hr.$$

(ii) Heat absorbed in BP produced (Q_{IP})

Brake power $Q_{BP} = 2\pi NT \ kJ/hr$ [N reval. per hour]

(or)

$$= 2\pi N(W-S) \ kJ/hr.$$

(iii) Heat rejected to the cooling water (Q_W)

$$Q_W = M_W C_W (T_2 - T_1) \ kJ/hr.$$

M_W - mass of cooling water kg/hr

C_W → specific heat capacity of water in kJ/kg k. = 4.19 kJ/kg k

T_1 - inlet temp in k.

T_2 - outlet temp in k.

(iv) Heat carried away by exhaust gas (Q_s)

$$Q_s = m_g C_g (T_g - T_a)$$

where m_g ⇒ Mass of exhaust gases in kg/hr

C_g - sp. heat capacity of exhaust gas = 1.005 kJ/kg k.

(v) Unaccounted losses (Q_{ua})

$$Q_{ua} = Q_s - [Q_{IP} + Q_W + Q_g + \ldots] \ kJ/hr.$$

Find out all the above heat losses into percentage,

$$\% \text{ heat loss} = \frac{\text{Heat loss}}{Q_s} \times 100$$

Heat Balance Sheet

Credit	kJ	%	Debit	kJ	%
1. Heat supplied by the fuel Q_s	--	100	1. Heat equivalent to IP or BP. 2. Heat carried away by cooling water (Q_W) 3. Heat carried away by exhaust gas (Q_g) 4. Unaccounted heat Loss (Q_{ua})		
	-	100	Total	-	100

A Test on a Single cylinder 4 Stroke oil engine having bore of 180mm and stroke of 360mm gave the following results.

Speed = 290 rpm

Brake Torque = 392 N-m

IMEP, P_m = 7.2 bar

oil consumption, m_f = 3.5 kg/hr

coolant flow, m_w = 270 kg/hr

cooling water Temp. rise = 36°C

Air fuel ratio by weight = 25

Exhaust gas Temp. T_g = 415°C

Room Temperature, T_a = 21°C

calorific value, CV = 45200 kJ/kg

C_{pg} (sp. heat of exhaust gas) = 1.005 kJ/kg K

calculate

(i) Indicated thermal η

(ii) Draw up a heat balance sheet in kJ/min basis.

Solution

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

Indicated power, $IP = \frac{P_m L A n}{60}$

$n = N/2$

$= \frac{7.2 \times 10^5 \times 0.36 \times \frac{\pi}{4}(0.18)^2 \times \frac{290}{2}}{60}$

$= 15939.94\ W$

$= 15.94\ kW$

Indicated thermal η

$\eta_{I.d.T} = \frac{IP \times 3600}{m_f \times CV}$

$= \frac{15.94 \times 3600}{3.5 \times 45200}$

$= 0.363$

$= 36.3\%$

Air fuel ratio by weight

$\frac{M_a}{M_f} = 25.$

$M_a = 25 \times m_f \rightarrow \frac{3.5}{60}$ kg/

$= 25 \times 0.0583$

$= 1.458\ kg/min$

(i) Heat supply by the fuel (Q_s)

$$Q_s = m_f \times CV$$
$$= 0.0583 \times 45200$$
$$= \underline{2636.67 \text{ kJ/min}}$$

(ii) Heat absorbed in BP produced (Q_{BP})

$$BP = 2\pi NT$$
$$= 2 \times \pi \times 290 \times 392$$
$$= 714272.5 \text{ J/min}$$
$$= \underline{714.27 \text{ kJ/min}}$$

% heat loss = $\dfrac{\text{Heat loss to BP}}{Q_s}$

$$= \frac{714.27}{2636.67} \times 100$$

$\frac{4.5}{270} = 4.5$

$M_W = \cancel{64}$

$$= \underline{27.09\%}$$

iii) Heat rejected to the cooling water (Q_w)

$$Q_w = M_w C_w (T_{w2} - T_{w1})$$
$$= 4.5 \times 4.187 \times 36$$
$$= 680.4 \text{ kJ/min}$$

% heat loss by cooling water.

$$= \frac{Q_w}{Q_s} = \frac{680.4}{2636.67} \times 100$$
$$= 25.81\%$$

(iv) Heat carried away by exhaust gas (Q_g)

$$Q_g = m_g \times c_g (T_g - T_a) \qquad \boxed{m_g = m_a}$$
$$= m_g\, 1.458 \times 1.005 (415 - 29)$$
$$= 577.32 \text{ kJ/min}$$

% heat loss = $\dfrac{\text{Heat loss } (Q_g)}{Q_s} \times 100$

$$= \frac{577.32}{2636.67} \times 100$$
$$= \underline{21.9\%}$$

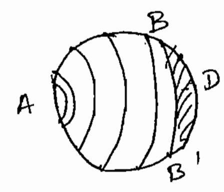
(v) Un accounted loss. (Q_{ua})

$$Q_{ua} = Q_s - [Q_{BP} + Q_w + Q_g + \cdots]$$
$$= 2636.67 - [714.27 + 680.4 + 577.32]$$
$$= \frac{664.68}{2636.67} \times 100$$
$$= \underline{25.2\%}$$

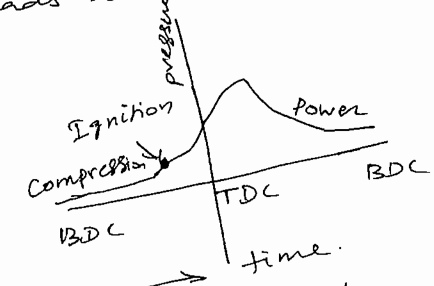
Credits	kJ/min	%	Debits	kJ/min	%
Heat supplied by the fuel Q_s	2636.67	100	1. Heat equivalent to BP	714.37	27.-
			2. Heat carried away by cooling water, (Q_w)	680.4	25.8
			3. Heat carried away by exhaust gas (Q_g)	577.32	21.9
			4. Unaccounted heat loss (Q_{ua})	664.68	25.2
		100	Total.	2636.67	100

Knocking in SI Engines

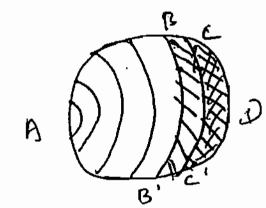
→ Autoignition leads towards engine knocking.



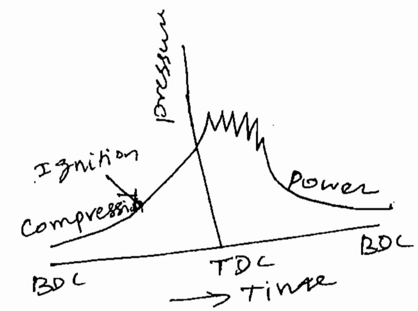
(a)



(b)



(c)



(d)

(vibration frequency - 5000 cps)

(a) The charge would not autoignite and the flame will advance further and consume the charge BB'D This is the normal combustion process. which illustrated fig (a) and (b).

(b) If the end charge BB'D reaches its autoignition temperature and remains for some length of time equal to the time of preflame reactions the charge will autoignite, leading to knocking combustion. During the preflame reaction period if the flame front could move from BB' to only CC' then the charge ahead of CC' would autoignite.

~~Effects~~

Effect of engine variables on knocking.

(1) Compression ratio. → Higher compression ratio increases the pre flame, reactions in the end charge.

→ high compression ratio leads to knock due to high pressure, Temp and density,

→ Low compression ratio will stop the knock.

(2) Mass of Inducted charge:-

A reduction in the mass of the inducted charge into the cylinder of an engine by throttling (or) by reducing the amount of supercharging reduces both density and temp of the charge at the time of Ignition. This decreases the tendency of knocking.

(3) Inlet temp of the mixture:-

Lower inlet temp is always preferable to reduce knocking.

(4) Temp of the combustion chamber walls.

The hot spot in the combustion chamber should be avoided, since the spark plug and exhaust valves are hottest parts in CC.

(5) Retarding the spark timing.

having the spark closer to TDC.

(6) Power output of the Engine:-

A decrease in the output of the engine decreases the temp of the cylinder and the CC walls and also the pressure of the charge.

An eight-cylinder, 4 stroke engine of 0.09 m bore and 0.08 m stroke with a compression ratio of 7 is tested at 4500 rpm on a dynamometer ~~Scale beam reading~~ which has 0.54 m arm. During a 10 min test the dynamometer scale beam reading was 42 kgf and the engine consumed 4.4 kg of gasoline having a calorific value of 44,000 kJ/kg. Air 300K and 1 bar was supplied to the carburettor at the rate of 6 kg/min. Find the brake power delivered, brake mean effective pressure, brake specific fuel consumption, brake specific air consumption, brake thermal efficiency, Volumetric efficiency and the air fuel ratio.

Given data

Number of cylinders, $K = 8$

Four Stroke petrol engine,

Diameter, $D = 0.09\,m$

Length, $l = 0.08\,m$

Compression ratio, $r = 7$

$CV = 44,000\,kJ/kg$

$N = 4500\,rpm = 75\,rps$

Radius, $R = 0.54\,m$

time $t = 10\,min$

Beam force, $W = 42\,kgf$

Mass of fuel, $m_f = 4.4\,kg$

$P_a = 1\,bar$

$T_a = 300K$

Mass of air, $m_a = 6$ kg/min.

Solution:-

Torque, T = Force × radius

$= 42 \times 0.54$

$= 22.68$ Nm

Brake power, $BP = 2\pi NT$

$= 2 \times \pi \times 75 \times 22.68$

$= 10687.698$ W

Brake mean effective pressure.

Brake power, $BP = P_{mb}\, l\, \frac{\pi}{4} D^2 \frac{N}{2} k$

$10687.698 = P_{mb} \times 0.08 \times \frac{\pi}{4} \times 0.09^2 \times \frac{75}{2} \times 8$

$P_{mb} = 70000$ Pa

Brake SFC $= \frac{m_f}{BP}$

$= \frac{4.4}{\frac{10}{60} \times 10687.698}$

$= 0.00247$ kg/W-hr

Brake SAC $= \frac{m_a}{BP}$

$= \frac{6 \times 60}{10687.698}$

$= 0.0337$ kg/W-hr

Brake thermal efficiency η_{BT}

$\eta_{BT} = \frac{BP \times 3600}{m_f \times CV}$

$= \frac{3600}{BSFC \times CV}$

$= \frac{3600}{0.00247 \times 44000000}$

$= 3.31$ %

From ideal gas equation

$P_a V_a = m_a R T_a$

$V_a = \frac{m_a R T_a}{P_a}$

$V_a = \frac{\frac{6}{60} \times 0.287 \times 300}{100}$

$= 0.0861$ m^3/s

Swept volume, $V_s = \frac{\pi}{4} D^2 L n k$

$= \frac{\pi}{4} \times 0.09^2 \times 0.08 \times \frac{75}{2} \times 8$

$= 0.1527$ m^3/s

Air Compressor - Unit - III

Reciprocating Air Compressors.

Introduction:

The process of increasing the pressure of air, gas, or vapour by reducing its volume is called compression and the device used to carry out this process is called a compressor.

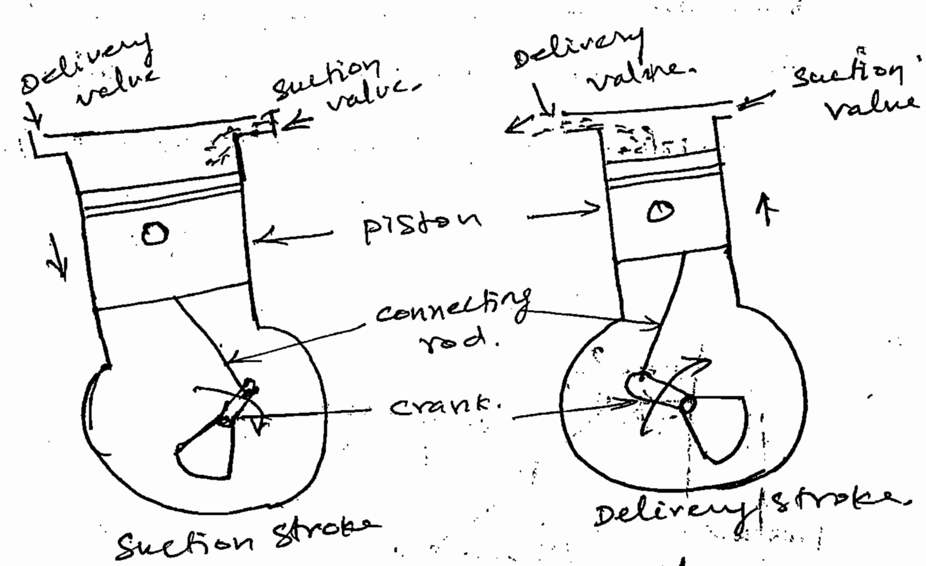
Application

1. Pneumatic Brakes & drills
2. Pneumatic Jacks, pneumatic lifts, spray painting. Shop cleaning, Injecting fuel in diesel engines. Refrigeration and air conditioning system.

Classification of Air compressor :-

1. According to the design and principle of operation
   a. Reciprocating compressor
   b. Rotary compressor
2. According to the action:-
   a. Single acting compressor
   b. Double acting compressor
3. According to number of stages
   a. Single stage compressor
   b. Multi stage compressor
4. According to the pressure limit.
   a. low pressure compressor
   b. Medium pressure compressor
   c. High pressure compressor
5. According to the capacity
   a. Low capacity compressor ($0.15\ m^3/s$)
   b. Medium capacity compressor ($0.15\ m^3/s$ to $5\ m^3/s$)
   c. High capacity compressor (More than $5\ m^3/s$)

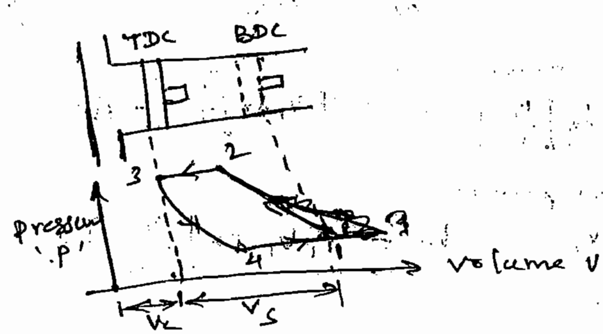
1. Syed Meer abjit
2. ~~Abjit~~ Dinkar
3



Air intake $P_1 V_1 T_1$ → compressor → compressed air. $P_2 V_2 T_2$

clearance space and clearance volume:-

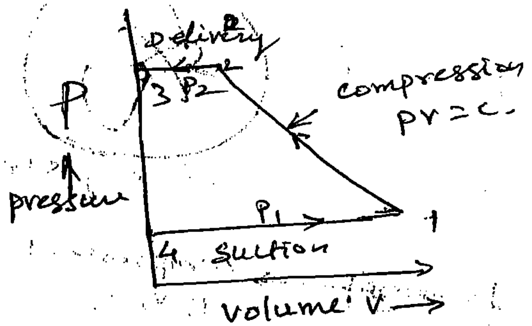
When the piston reaches top dead centre in the cylinder, there is a dead space between piston top and cylinder head. This space is known as clearance space and the volume occupied by this space is known as clearance volume. (V_c)



16, 39, 40, 45, 51

Work done By a Single Stage Reciprocating Air compressor without clearance volume.

Workdone during isothermal compression ($PV=C$)



process 4-1 ⇒ the suction of air at P_1 pressure

process 1-2 → Air is compressed isothermally from pressure P_1 to P_2

process 2-3 ⇒ the discharge of air at pressure P_2,

work done = Area 1-2-3-4-1

$$W = W_{comp} + W_{Delivery} - W_{suction}$$

For isothermal compression work done $= P_1V_1 \ln\left[\frac{V_1}{V_2}\right]$

$$= P_1V_1 \ln\left[\frac{V_1}{V_2}\right] + P_2V_2 - P_1V_1$$

For isothermal process $P_1V_1 = P_2V_2$

$$W = P_1V_1 \ln\left[\frac{V_1}{V_2}\right] \quad \text{—(1)}$$

We know that

$$P_1V_1 = P_2V_2$$

$$\frac{V_1}{V_2} = \frac{P_2}{P_1}$$

Substituting in eqn(1)

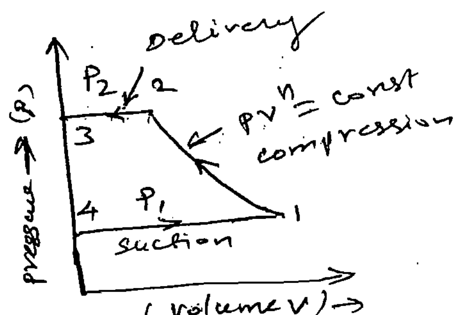
$$W = P_1V_1 \ln\left[\frac{P_2}{P_1}\right]$$

We know $PV = mRT$.

$$\boxed{W = mRT_1 \ln\left[\frac{P_2}{P_1}\right]}$$

Work done polytropic compression ($PV^n = \text{const}$)

The P-V diagram for a single stage single acting reciprocating air compressor is shown in fig.



process 4-1 represents suction of air at pressure P_1

process 1-2 represents compression of air polytropically from pressure 'P_1' to pressure P_2

process 2-3 – represents the discharge or delivery of air at pressure P_2'

Work done = Area 1-2-3-4-1

$W = W_{comp} + W_{Delivery} - W_{suction}$

$\because W_{comp} = \frac{P_2V_2 - P_1V_1}{n-1}$ for polytropic process.

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

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

$= \frac{\cancel{P_2V_2} - \cancel{P_1V_1} + nP_2V_2 - \cancel{P_2V_2} - nP_1V_1 + \cancel{P_1V_1}}{n-1}$

$= \frac{nP_2V_2 - nP_1V_1}{n-1}$

$= \frac{n}{n-1}\left[P_2V_2 - P_1V_1\right]$ $\quad P_1V_1 = mRT_1$

$= \frac{n}{n-1}\left[mRT_2 - mRT_1\right]$

$= \frac{n}{n-1} mR(T_2 - T_1)$ $\quad T_1$ Take outside

$= \frac{n}{n-1} mRT_1\left[\frac{T_2}{T_1} - 1\right]$

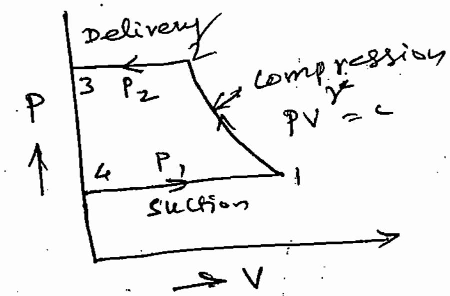
For polytropic process

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

$= \frac{n}{n-1} mRT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$ $\quad [P_1V_1 = mRT_1]$

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

c. Work done during isentropic compression (or) Reversible adiabatic compression $[PV^{\gamma} = c]$



4-1 — Suction

1-2 — compression (Rev. adiabatic process)

2-3 — Discharge

Work done = Area 1-2-3-4-1

$W = W_{comp} + W_{Delivery} - W_{suction}$

$\because W_{comp} = \frac{P_2V_2 - P_1V_1}{\gamma - 1}$ for isentropic process.

$= \frac{P_2V_2 - P_1V}{\gamma - 1} + P_2V_2 - P_1V_1$

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

$$= \frac{\cancel{P_2V_2} - \cancel{P_1V_1} + \gamma P_2V_2 - \cancel{P_2V_2} - \gamma P_1V_1 - \cancel{P_1V_1}}{\gamma - 1}$$

$$= \frac{\gamma P_2V_2 - \gamma P_1V_1}{\gamma - 1}$$

$$= \frac{\gamma}{\gamma-1}\left[P_2V_2 - P_1V_1\right]$$

$pV = mRT$

$$= \frac{\gamma}{\gamma-1}\left(mRT_2 - mRT_1\right)$$

$$= \frac{\gamma}{\gamma-1} mR\left[T_2 - T_1\right]$$

$$= \frac{\gamma}{\gamma-1} mRT_1\left[\frac{T_2}{T_1} - 1\right]$$

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

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

$P_1V_1 = mRT_1$

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

A single stage, single acting air compressor has a cylinder bore of 175 mm and a stroke of 225 mm. The compressor ~~has a~~ sucks air at 100 kN/m² and 27°c and delivers at 8 bar. Find the power required to drive a compressor If its speed is 120 rpm. Also calculate mass of the air compressed per minute and delivery Temperature. Compression follows the law $pV^{1.3} = C$.

Give data:-

Cylinder bore $d = 175$ mm $= 0.175$ m

Storke $l = 225$ mm $= 0.225$ m

Suction pressure $P_1 = 100$ kN/m²

" Temp $T_1 = 27°$c

Delivery pressure $P_2 = 8$ bar $= 800$ kN/m²

Speed $N = 120$ rpm

Required data:-

1. power required to drive the compressure
2. Mass ~~flow rate~~ of the air compressed per minute

Solution:-

Stroke or swept volume.

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

$$= \frac{\pi}{4} \times (0.175)^2 \times 0.225$$

$$= \underline{\underline{5.41 \times 10^{-3}\ m^3}}$$

$V_s = V_1$ [$\because$ clearance volume is neglected,]

$V_1 = 5.41 \times 10^{-3}\ m^3$

Work done during polytropic compression,

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

$\boxed{kN\text{-}m = kJ}$

$$= \frac{1.3}{1.3-1}\ 100 \times 5.41 \times 10^{-3} \left[\left(\frac{800}{100}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$= 1.444\ kJ$

Indicated power of the compressor,

$$P = \frac{W \times N}{60}$$

$$= \frac{1.44 \times 120}{60}$$

$= 2.888\ kW$

We know that

$P_1 V_1 = m R T_1$

$$m = \frac{P_1 V_1}{R T_1}$$

$$= \frac{100 \times 5.41 \times 10^{-3}}{0.287 \times 300}$$

$= 6.283 \times 10^{-3}\ kg$

Mass of air delivered per minute.

$m = 6.283 \times 10^{-3} \times N$. [For each revolution we get one delivery.]

$= 6.283 \times 10^{-3} \times 120$

$= 0.754\ kg/min$

Temp. at the end of compression

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

$$T_2 = \left(\frac{800}{100}\right)^{\frac{1.3-1}{1.3}} \times 300$$

$= 986.76\ K$

Work done by single stage Reciprocating Air Compressor with clearance volume:-

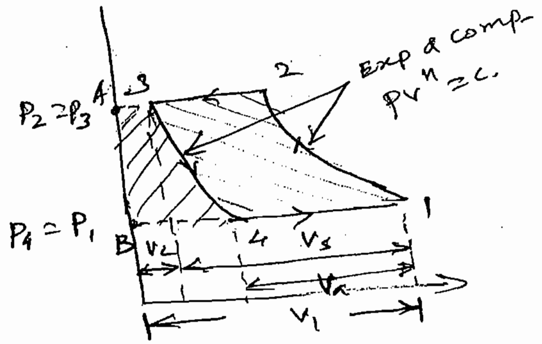
Consider a RAC with clearance volume (V_c)

$P_1 V_1 T_1$ – Initial pressure, Temp, volume of air respectively.

$P_2 V_2 T_2$ – Corresponding values for final cond

V_c – clearance volume.

n – polytropic index for compression and expansion,



Work done by the compressor per cycle

W = Area 1-2-3-4-1 = (Area 1-2-A-B-1) − (Area 3-A-B-4-3)

W = work done during comp. − workdone during Exp.

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

We know that
$P_3 = P_2$
$P_4 = P_1$.

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

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

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

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

where $V_1 - V_4 = V_a \rightarrow$ Actual volume of free air delivered per cycle.

$$W = \frac{n}{n-1} mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad \boxed{P_1 V_a = mRT_1}$$

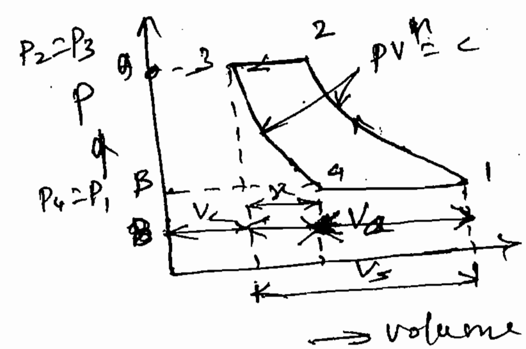
— x —

Volumetric efficiency,

Volumetric efficiency is defined as the ratio of volume of free air sucked in to the compressor per cycle to the Stroke volume of the cylinder.

$$\eta_{vol} = \frac{\text{Volume of free air taken per cycle}}{\text{Stroke volume of the cylinder}}$$

$$\eta_{vol} = \frac{V_a}{V_s}$$



From p-v diagram

$$V_a = V_s - x$$

$$x = V_4 - V_c$$

$$\eta_{vol} = \frac{V_s + x}{V_s} = \frac{V_s - (V_4 - V_c)}{V_s}$$

$$= \frac{V_s - V_c\left[\frac{V_4}{V_c} - 1\right]}{V_s}$$

$$= 1 - \frac{V_c}{V_s}\left[\frac{V_4}{V_c} - 1\right] \quad \text{—— (1)}$$

compression and expansion follows $pv^n = c$.

$$P_3 V_3^n = P_4 V_4^n$$

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

from pv diagram we know that

$$V_3 = V_c, \quad P_4 = P_1, \quad P_3 = P_2$$

$$\frac{V_4}{V_c} = \left(\frac{P_2}{P_1}\right)^{1/n}$$

Apply $\frac{V_4}{V_c}$ value in eqn (1).

$$\eta_{vol} = 1 - \frac{V_c}{V_s}\left[\left(\frac{P_2}{P_1}\right)^{1/n} - 1\right] \quad \text{—— (2)}$$

clearance ratio is defined as the ratio of clearance volume to swept volume.

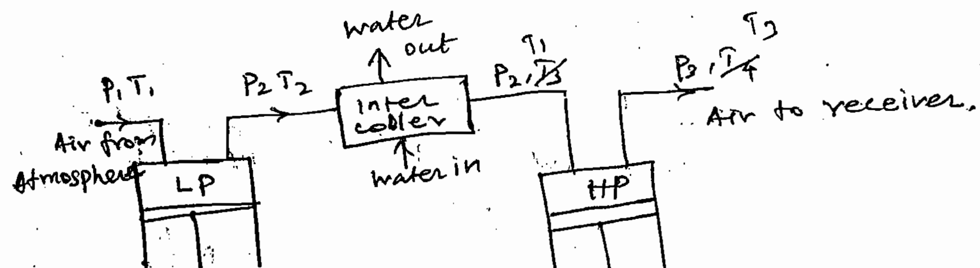
clearance ratio, $c = \frac{V_c}{V_s}$

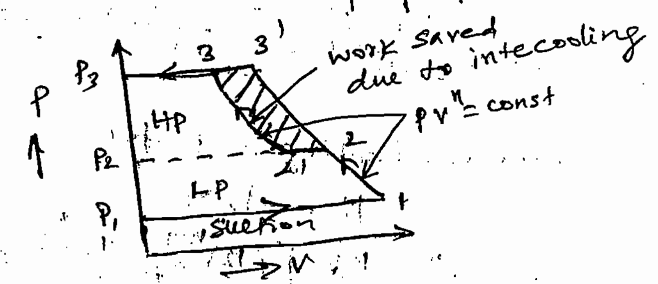
$$\eta_{vol} = 1 - c\left[\left(\frac{P_2}{P_1}\right)^{1/n} - 1\right]$$

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

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

Multi-stage Air compression with intercooling





Assumptions Made in Multistage Compression.

1. Suction and delivery pressures remains constant during each stage.
2. The index of compression is same in each stage
3. The intercooling in each stage is at constant temperature.
4. The mass of air handled by the low pressure and high-pressure cylinders are same.

$P_1\ V_1\ T_1 \rightarrow$ Pressure, Volume and Temperature of air entering the low pressure (LP) cylinder

$P_2\ V_2\ T_3\ T_1 \rightarrow$ pressure volume and Temp of air entering the high pressure cylinder

$P_3 \rightarrow$ Final delivery pressure of air

$n \rightarrow$ polytropic index of both the cylinder.

Total work input,

= work input for LP comp. + work input for HP comp.

$$W = \frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} P_2V_2\left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right] \text{ --- (1)}$$

When the cooling is perfect

$P_1V_1 = P_2V_2$

$$W = \frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 2\right] \text{ --- (2)}$$

Inter-Stage pressure (P_2) or condition fo
Minimum work input for two-stage c

We know

$$W = \frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 2\right]$$

Differentiating with respect to P_2 and equating to zero.

$$\frac{dW}{dP_2} = 0.$$

$$\frac{d}{dP_2}\left[\frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 2\right]\right] = 0.$$

Let $\frac{n}{n-1} = K$ (cons

$$\frac{d}{dP_2}\left[KP_1V_1\left[\left(\frac{P_2}{P_1}\right)^K + \left(\frac{P_3}{P_2}\right)^K - 2\right]\right] = 0.$$

$$KP_1V_1\frac{d}{dP_2}\left[\left(\frac{P_2}{P_1}\right)^k + \left(\frac{P_3}{P_2}\right)^k - 2\right] = 0.$$

$\left(\frac{1}{P_1}\right)^k P_2^k + P_3^k \cdot P_2^{-k}$, $-k-1$

$$KP_1V_1\left[\left(\frac{1}{P_1}\right)^k \cdot k(P_2)^{k-1} + (P_3)^k \cdot (-k)(P_2)^{-k-1}\right] = 0.$$

$$\left[\left(\frac{1}{P_1}\right)^k \cdot k(P_2)^{k-1} + P_3^k \cdot (-k)(P_2)^{-k-1}\right] = 0.$$

$$k(P_1)^{-k}(P_2)^{k-1} - k(P_3)^k(P_2)^{-k-1} = 0.$$

$$k(P_1)^{-k} \cdot (P_2)^{k-1} = k(P_3)^k \cdot (P_2)^{-k-1}.$$

$$\frac{(P_2)^{k-1}}{(P_2)^{-k-1}} = \frac{(P_3)^k}{(P_1)^{-k}}$$

$$(P_2)^{k-1} \times (P_2)^{k+1} = (P_3)^k \cdot (P_1)^k$$

$$(P_2)^{2k} = (P_3 P_1)^k$$

(or)

$$P_2^2 = P_1 P_3$$

$$\boxed{P_2 = \sqrt{P_1 P_3}}$$

Minimum work Required for a Two-Stage Reciprocating Air compressor:

Work done by a two stage Reciprocating Air compressor with intercooling.

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 2 \right] \quad \text{--- (1)}$$

Work input will be minimum, if $P_2 = \sqrt{P_1 P_3}$

$$P_2^2 = P_1 P_3$$

$$P_2 \times P_2 = P_1 P_3$$

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} \quad \text{--- (2)}$$

$$P_2^2 = P_1 P_3$$

Dividing the equation on both sides by P_1^2

$$\frac{P_2^2}{P_1^2} = \frac{\cancel{P_1} P_3}{P_1^{\cancel{2}}}$$

$$\frac{P_2^2}{P_1^2} = \frac{P_3}{P_1}$$

$$\left(\frac{P_2}{P_1}\right)^2 = P_3/P_1$$

$$\frac{P_2}{P_1} = (P_3/P_1)^{1/2}$$

$$= \left(\frac{P_3}{P_*}\right)^{\frac{n-1}{n} * \frac{1}{2}} + \left(\frac{P_3}{P_1}\right)^{\frac{n-1}{n} * \frac{1}{2}}$$

$$= \left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}} + \left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}}$$

We know that

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \left(\frac{P_3}{P_1}\right)^{1/2} \quad \text{--- (3)}$$

Substitute the eqn (3) in eqn (1).

$$W_{min} = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}} + \left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}} - 2 \right]$$

$$W_{min} = \frac{n}{n-1} P_1 V_1 \left[2 \left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}} - 2 \right]$$

$$= \frac{2n}{n-1} P_1 V_1 \left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}} - 1 \right]$$

In general the workdone can be written for compressor having x no of stages

$$W = \frac{xn}{n-1} P_1 V_1 \left[\left(\frac{P_{x+1}}{P_1}\right)^{\frac{n-1}{xn}} - 1 \right]$$

A single acting reciprocating air compressor has a piston diameter of 200mm and a stroke of 300mm and runs at 350 rpm. Air is drawn at 1.1 bar and is delivered at 8 bar. The law of compression is $pv^{1.35}$ = constant, and the clearance volume is 6% of stroke volume. Determine the mean effective pressure and power required to drive the compressor.

Given data

$d = 200mm$

$l = 300 mm$

$N = 350$ rpm

$P_1 = 1.1$ bar

$P_2 = 8$ bar

$pv^{1.35} =$ const, $n = 1.35$

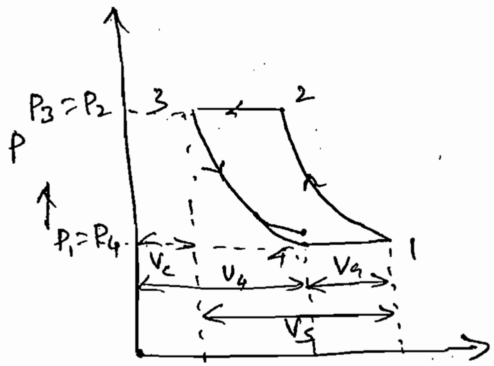
$V_c = 6\%$ of V_s

Required data

Mean effect. pre. $P_m = ?$

power, $P = ?$

Solution



$$\boxed{V_a = V_1 - V_4}$$

$$P_m = \frac{\text{Work done}}{\text{stroke volume}}$$

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

Stroke volume, $V_s = \frac{\pi}{4} d^2 \times l$

$$= \frac{\pi}{4} \times (0.2)^2 \times 0.3$$

$$= 0.00943 \, m^3$$

from P-V diagram,

$$V_1 = V_c + V_s$$

$$= 0.06 \times 0.00943 + 0.00943$$

$$= \underline{0.01 \, m^3}$$

we know that

$$P_3 V_3^n = P_4 V_4^n$$

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

$V_3 = V_c$
$= 0.06 \times 0.00943$

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

$P_3 = P_2$
$P_4 = P_1$

$$= \left(\frac{800}{110} \right)^{1/1.35} \times 0.06 \times 0.00943$$

$$= \underline{0.00246 \, m^3}$$

$$V_a = V_1 - V_4$$

$$= 0.01 - 0.00246$$

$$= \underline{0.00754 \, m^3}$$

$$W = \frac{1.35}{1.35-1} \times 110 \times 0.00754 \times \left[\left(\frac{800}{100} \right)^{\frac{1.35-1}{1.35}} - 1 \right]$$

$$= \cancel{1.937 \, kJ} \quad \underline{2.152 \, kJ}$$

power out put of the compressor

$$P = \frac{W \times N}{60}$$

$$= \frac{2.152 \times 350}{60} = \underline{12.55\ kW}$$

$$= \cancel{9.685\ kW}$$

$$P_m = \frac{WD}{V_s}$$

$$= \frac{2.152}{0.00943}$$

$$= \underline{228.2\ kPa}$$

In a single acting air compressor, the clearance volume is 6% of stroke volume. Air is drawn in a constant pressure of 1 bar at a temperature of 30°c. Compression follows the law $pv^{1.3} = c$ and the reciever pressure is 8 bar. The compressor delivers 10 kg/s of air. Find the volumetric efficiency and power required to drive the compressor.

Given data

$V_c = 6\%$ of V_s

$P_1 = 1$ bar

$T_1 = 30°c$

$pv^{1.3} = $ const.

$P_2 = 8$ bar

$m = 10$ kg/s

Required data:-

1. Volumetric efficiency, $\eta_{vol} = \frac{V_a}{V_s}$ (or) $1 + c - c\left(\frac{P_2}{P_1}\right)$

2. Power required to run the compressor $\Big\}$ $P = \frac{W \times N}{60}$ (or) W in k…

Solution

$$\eta_{vol} = 1 + c - c\left[\frac{P_2}{P_1}\right]^{1/n} \quad \text{whe } c = \frac{V_c}{V_s}$$

$$= 1 + \frac{V_c}{V_s} - \frac{V_c}{V_s}\left[\frac{P_2}{P_1}\right]^{1/n}$$

$$= 1 + \frac{0.06 V_s}{V_s} - \frac{0.06 V_s}{V_s}\left[\frac{800}{100}\right]^{1/1.3}$$

$$= 0.763 = \underline{76.3\%}$$

$$\text{work done} = \frac{n}{n-1} P_1 V_a \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{\cancel{1.3}}{\cancel{1.3-1}}$$

WKT

$$P_1 V_a = m R T_1$$

$$V_a = \frac{10 \times 0.287 \times 303}{100}$$

$$= 8.7 \ m^3/s$$

$$WD = \frac{1.3}{1.3-1} \times 100 \times 8.7 \left[\left(\frac{800}{100}\right)^{\frac{1.3-1}{1.3}} - 1 \right]$$

$$= 2321.82 \ kW$$

A single stage double acting compressor has a free air delivery (FAD) of $14 \ m^3/min$ measured at 1.013 bar and 15°C. The pressure and temperature in the cylinder during induction are 0.95 bar and 32°C respectively. The delivery pressure is 7 bar and index of compression and expansi $n = 1.3$, The clearance volume is 5% of the swept volume. Calculate the indicated power requir and the volumetric efficiency.

FAD - It is disch volume of the compr corresponding to amb conditions.

Given data:-

$P_2 = 7$ bar $= 700$ kPa

$P_1 = 0.95$ bar $= 95$ kPa

$T_1 = 32°C + 273 = 305 \ K$

$V_0 = 14 \ m^3/min = 0.233 \ m^3/s$

$V_c = 5\% \ V_s = 0.05 \ V_s$

$c = \frac{V_c}{V_s} = 0.05$

$P_0 = 1.013$ bar $= 101.3$ kPa

$T_0 = 15°C + 273 = 288 K$

$n = 1.3$

Solution

$$\eta_{vol} = 1 + c - c \left[\left(\frac{P_2}{P_1}\right)^{1/n} \right]$$

$$= 1 + 0.05 - 0.05 \left[\left(\frac{700}{95}\right)^{1/1.3} \right]$$

$$= 0.818$$

$$= 81.8 \%$$

We know that

$$\frac{P_0 V_0}{T_0} = \frac{P_1 V_a}{T_1}$$

$$\frac{101.3 \times 0.233}{288} = \frac{95 \times V_a}{305}$$

$$V_a = 0.263 \, m^3/s$$

work done

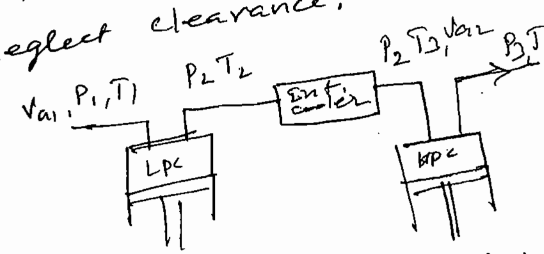
$$P = \frac{n}{n-1} P_1 V_a \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.3}{1.3-1} \, 95 \times 0.263 \left[\left(\frac{700}{95}\right)^{\frac{1.3-1}{1.3}} - 1 \right]$$

$$= 63.39 \, kW$$

Multi stage problem

A single-acting two stage air compressor with 4 m^3/min of air at 1.013 bar and 15°c with a speed of 250 rpm. The delivery pressure is 80 bar. Assuming the complete intercooling, find the minimum power required by the compressor and the bore and stroke of the compressor. Assume a piston speed of 3 m/s, mechanical efficiency of 75% and volumetric efficiency of 80% per stage. Assume the polytropic index of compression in both the stages to be n = 1.25 and neglect clearance.



Solution

Given data

$V_{a_1} = 4 \, m^3/min$

$P_1 = 1.013$ bar

$T_1 = T_3 = 15°c = 273 + 15 = 288K$

$N = 250$ rpm

$P_3 = 80$ bar

piston speed $= 3$ m/s $= \frac{2LN}{60}$

$\eta_{mech} = 75\%$

$\eta_{vol} = 80\%$

$n = 1.25$ both the stages

Required data

① power required or Input

② bore, D = ?

③ stroke, L = ?

Solution

Inter cooler pressure, $P_2 = \sqrt{P_1 P_3}$

$= \sqrt{1.013 \times 80}$

$= 9 \text{ bar}$

From ideal gas equation, $P_1 V_{a_1} = mRT_1$

$m = \dfrac{P_1 V_{a_1}}{RT_1}$

$= \dfrac{1.013 \times 10^2 \times 4/60}{0.287 \times 288}$

$= 0.0817 \text{ kg/s}$

piston speed $= \dfrac{\cancel{2}LN}{60} = \dfrac{2LN}{60}$

$3 = L \times 250/60 \times 2$

$L = 0.72 \text{ m} = 720 \text{ mm}$

But we know that the volumetric η,

$\eta_{vol} = \dfrac{V_{a_1}}{V_{s_1}}$

$0.8 = \dfrac{4/60}{V_{s_1}}$

Swept volume of LP cylinder

$V_{s_1} = 0.0833 \text{ m}^3/\text{s}$

Swept volume $= \dfrac{LAN}{60}$ $\boxed{V_s = \frac{\pi}{4} d^2 \times l. \text{ m}^3}$

$0.0833 = 0.72 \times \frac{\pi}{4} D^2 \times \frac{250}{60}$

$D = 187.7 \text{ mm}$

Similarly for HP cylinder

$V_{a2} = \dfrac{mRT_3}{P_2}$

$= \dfrac{0.0817 \times 0.287 \times 288}{9 \times 10^2}$

$= 0.0075 \text{ m}^3/\text{s}$

But we know that the Volumetric η

$\eta_{vol} = \dfrac{V_{a2}}{V_{s2}}$

$0.8 = \dfrac{0.0075}{V_{s2}}$

$V_{s2} = 0.009375 \text{ m}^3/\text{s}$

Swept volume,

$$V_{s2} = \frac{LAN}{60} \qquad A = \frac{\pi}{4} D^2$$

$$0.009375 = 0.72 \times \frac{\pi}{4} \times D^2 \times \frac{250}{60}$$

$$D = 0.06308 \text{ m}$$
$$= \underline{\underline{63.08 \text{ mm}}}$$

Work done during polytropic compression,

$$W = \frac{xn}{n-1} P_1 V_a \left[\left(\frac{P_{x+1}}{P_1} \right)^{\frac{n-1}{xn}} - 1 \right]$$

$$= \frac{2 \times 1.25}{1.25-1} \times 1.013 \times 10^2 \times \frac{4}{60} \left[\left(\frac{80 \times 10^2}{1.013 \times 10^2} \right)^{\frac{1.25-1}{2 \times 1.25}} - 1 \right]$$

$$= \cancel{37002} \ 37002.75 \text{ W} \Rightarrow \underline{\underline{37.002 \text{ kW}}}$$

Mechanical efficiency,

$$\eta_{mech} = \frac{\text{Indicated power}}{\text{power Input}}$$

$$\text{power input} = \frac{\text{Ind. power}}{\eta_{mech}}$$

$$= \frac{37.00275}{0.75}$$

$$= \underline{49.337 \text{ kW}}$$

A Three stage air compressor delivers 5.2 m^3 of free air per minute. The suction pressure and temperature are 1 bar and 30°C. The pressure and temperature are 1.03 bar and 20°C at the free air condition. The air is cooled at 30°C after each stage of compression. The delivery pressure of the compressor is 150 bar. RPM of the compressor is 300. The clearances of L.P., I.P., and H.P. cylinders are 5% of the respective strokes. The index of compression and re-expansion in all stages is 1.35. Neglecting pressure losses, find the B.P. of the motor required to run the compressor if the mechanical efficiency is 80%.

Given data

$V_0 = V_0 = 5.2 \text{ m}^3/\text{min} \qquad \frac{5.2}{60} =$

$P_1 = 1 \text{ bar}$

$T_1 = 30°C = 30+273 = 303 K \Rightarrow T_1 = T_2 = T_3 = T_4$

$P_0 = 1.03 \text{ bar}$

$T_0 = 20°C = 20+273 = 293 K$

$P_4 = 150 \text{ bar}$

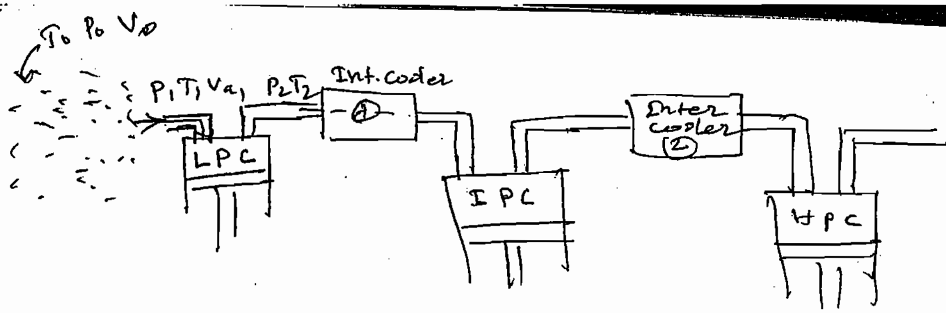
$N = 300 \text{ rpm}$

$C = 5\% = 0.05$

$n = 1.35$

$\eta_{mech} = 80\% = \frac{BP}{IP}$

$\underline{IP = W \text{ in kJ/sec}}$



Solution

Inter cooler pressure,

$$\frac{P_2}{P_1} = \left(\frac{P_4}{P_1}\right)^{1/3}$$

$$= \left(\frac{150}{1}\right)^{1/3}$$

$$\frac{P_2}{P_1} = 5.31$$

$$\therefore \frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} = 5.31.$$

We know that $V_0 = 5.2\ m^3/min$

$$= \frac{5.2}{60} = 0.0867\ m^3/s$$

Then, $\dfrac{P_0 V_0}{T_0} = \dfrac{P_1 V_{a1}}{T_1}$

$$\frac{103 \times 0.0867}{293} = \frac{100 \times V_{a1}}{303}$$

$$V_{a1} = 0.0923\ m^3/s$$

III^ly, $\dfrac{P_0 V_0}{T_0} = \dfrac{P_2 V_{a2}}{T_2}$

$$\frac{103 \times 0.0867}{293} = \frac{531 \times V_{a2}}{303}$$

$$V_{a2} = 0.0174\ m^3/s$$

III^ly, $\dfrac{P_0 V_0}{T_0} = \dfrac{P_3 V_{a3}}{T_3}$

$$\frac{103 \times 0.0867}{293} = \frac{28.19 \times V_{a3}}{303}$$

$$V_{a3} = 0.00328\ m^3/s$$

$$W = \frac{n}{n-1} P_1 V_{a1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} P_2 V_{a2}\left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} P_3 V_{a3}\left[\left(\frac{P_4}{P_3}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.35}{1.35} \times 103 \times 0.0923 \times \left[(5.31)^{\frac{1.35-1}{1.35}} - 1\right] + \cancel{\frac{1.35}{1.35-1} \times 5.31 \times 0}$$

$$+ \frac{1.35}{1.35-1} \times 5.31 \times 0.0174 \left[(5.31)^{\frac{1.35-1}{1.35}} - 1\right]$$

$$+ \frac{1.35}{1.35-1} \times 28.19 \times 0.0174 \left[(5.31)^{\frac{1.35-1}{1.35}} - 1\right]$$

$$= 57.91\ kW$$

$$\frac{P_2}{P_1} = 5.31$$

$$P_2 = 5.31 \times 1$$

$$= 5.31\ kPa$$

$$\therefore \frac{P_3}{P_2} = 5.3$$

$$P_3 = 5.31 \times$$

$$= 5.31 \times 5.$$

$$= 28.19$$

we know that mechanical efficiency,

$$\eta_{mech} = \frac{BP}{IP}$$

$$0.8 = \frac{BP}{57.912}$$

$$BP = \underline{\underline{46.33}}\ kW$$

Introduction:- Rotary compressors are used to sup
Continuous pulsation free compressed air. They
have rotor(s) and casing in place of piston an
cylinder arrangement. They are compact,
well balanced, and high speed compressors.
They have low starting torque thus they are
directly coupled ~~to~~ with prime-mover. The
handle large mass of gas and are suitable
for low and medium pressure ratios.

The special features of rotary compressor

1. Designed to provide pulsation-free air
2. 100% Continuous duty
3. Quiet operation
4. Energy efficient at full load
5. Extended service intervals,
6. Reliable long life
7. Improved air quality
8. Low starting torque.

Fans and blowers are also rotary machines, which are used for supplying air or gas. these machines are differentiated by the method used to move the air, and by the system pressure, at which they are operating the ratio of the discharge pressure due to the suction pressure is used to define the fans, blowers and compressors as shown is table.

Equipment	pressure ratio	pressure rise in mm of water
Fan	up to 1.11	1136
Blower	1.11 to 1.20	1136-2066
Compressor	More then 1.20	More then 2066

Some Compressors are suitable only for low pressure ratio work such as for scavenging and Supercharging of engines and for various applications of exhausting and vaccum pumping.

For a pressure ratio above 9 bar, Vane-type rotary machines can be used to boost the pressure.

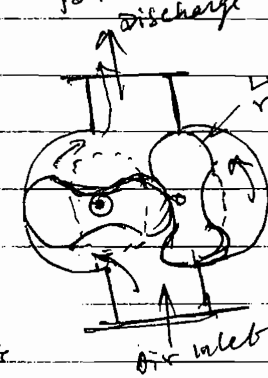
Classification of Rotary Compressors

- Rotary Compressors
  - Positive displacement type
    - Roots Blower
    - Lysholm
    - Screw type
    - Vane type
  - Dynamic action or Steady flow type
    - Centrifugal type
    - Axial flow type

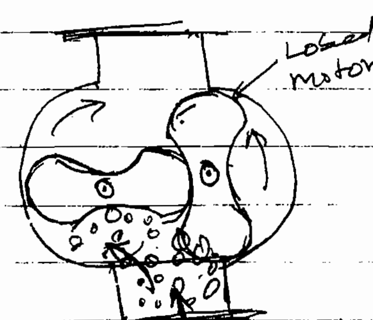
Comparison between Reciprocating and Rotary Compressors

Sno	Aspect	Reciprocating A.C.	Rotary A.C
1.	pressure ratio	high pressure ratio 4 to 7 bar per stage	Low pressure ratio 3 to 5 bar per stage
2.	Handled volume	Low volume limited to 50 m^3/s	Large volume about 500 m^3/s
3.	Speed of Compressor	Low Speed	High Speed Compress
4.	Vibrational problem	Greater Vibrational due ~~to~~ Reciprocating action	Less Vibrational problem
5.	Size	Bulky for given discharge volume	Small for given D.V.
6.	Air supply	Air supply is intermittent	Air supply is steady and continuous
~~7~~	~~purity of compressed~~		
7.	~~Air~~ purity of compressed air	Air delivered from compresser is dirty	Very clean and free from dirt
8.	Compression efficieny	High with pressure ratio more than 2	Higher with Comp. ratio < 2
9.	Maintenance	Higher due to Reciprocating parts	Lower due to balanced rotary parts
10	Mechanical efficieny	Lower due to several sliding parts	Higher due to less sliding parts
11.	Lubrication	complicated Lub. system	Simple Lub. Sys.
12	Initial cost	Higher	Lower
13	Flexibility	Greater flexibility in capacity and pressure range	No flexibility capacity and pressure range
14.	suitability	For medium and high ressure ratio and low and medium gas volume	For low and medi pressure and la Volumes

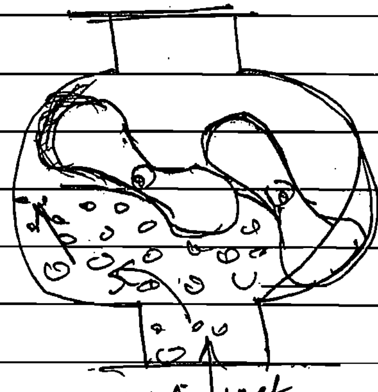
Roots Blower Compressor



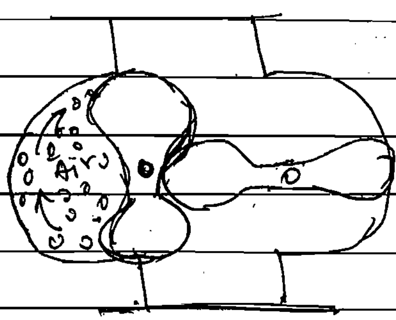
(a) Two lobed roots blower



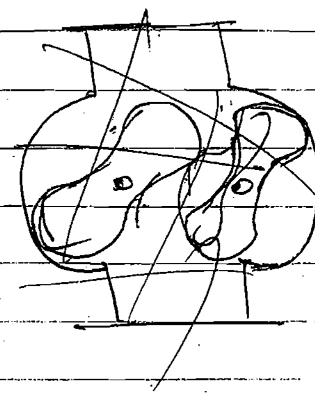
(b) Initial part of suction process

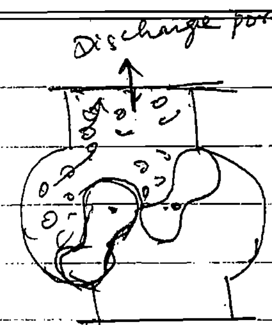


(c) Continuation of suction process

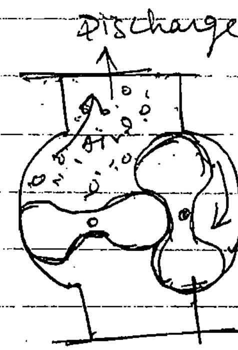


d. compression process





e. start of discharge process



f. Completion of discharge position.

Roots Blower Compressor :-

Roots blower Is a positive displacement Compressor, it is also called Lobe compressor. The roots blower is essentially a low pressure blower and is limited to a discharge pressure of 1bar in single-stage design and up to 2.2 bar in a two-stage design. Its discharge capacity is limited to 1500 m^3/ min and it can run up to 7000 rpm.

Construction :

This type of rotary compressor consists of two or more lobed rotors and casing with inlet and outlet passage of air. The lobed rotors rotate in an air tight casing with the help of gears in external housing. The compressor inlet is open to atmospheric air at one side and it is open to delivery side at the other side. The two lobes of the roots blower is shown in fig (a).

One of the rotor is connected to drive. The second rotor is gear driven from the first. Thus, both rotors rotate with the same speed. The profile of the lobe is made cycloidal or involute in order to seal the inlet side from the delivery side

Working :-

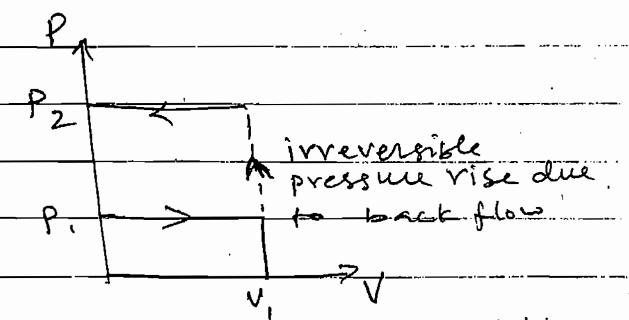
The rotation of rotors creates space in the casing at the entry port as shown in fig (b). The air is drawn in to the casing to fill the space. The flow of gas in the casing space continues till both rotors change their position as shown in fig (c).

With further ~~moment~~ movement of the lobed rotor, the air is trapped between one rotor, when its tip touches the casing as shown in fig (d). This part of the blower is not open to suction port. Du the air flows into the space created by rotation of other rotor. This rotor is also carrying out the same cycle as first rotor after 90°.

The trapped volume of air is not internally compressed, it is only displaced at high speed from suction side to delivery side. Continued rotation of lobes opens the discharge port as shown in fig (e).

Since the compressed air at higher pressure is present at the delivery side, when the rotor lobe uncovers the exit port, some pressurised air enters into the space between the rotors and casing of the compressor. This flow of air is called back flow of air. This back flow continues until the pressure in the blower gets equalised. After back flow, the air is compressed irreversibly at constant volume.

Finally, at high pressure, the air is delivered from the blower to receiver as shown in fig (f). The process of compression can be represented by constant volume line on p-V plane as shown in fig.



P-V digram for roots blower.

Consider a volume of V_1 is trapped between the lobe and casing at atmospheric pressure P_1. The air is compressed to delivery pressure P_2. The actual work done on air,

$$W_{act} = -\int_1^2 V\,dp = V_1(P_2 - P_1) \quad \text{—— (1)}$$

The ideal work input for compression is isentropic work input. The theoretical work input to compress air from atmospheric pressure P_1 to delivery pressure P_2

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

Efficiency

The efficiency of the roots blower can be defined as the ratio between adiabatic work to the actual work input. Mathematically

$$\eta_{roots} = \frac{\text{Adiabatic work input}}{\text{Actual work input}}$$

$$= \frac{\frac{\gamma}{\gamma-1} P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{V_1(P_2-P_1)}$$

$V_1(P_2-P_1)$

take P_1 is outside

$P_1V_1\left[\frac{P_2}{P_1} - 1\right]$ $\quad \frac{P_2}{P_1} = r_p$

Dividing numerator and denominator by P_1V_1, we get

$$= \frac{\frac{\gamma}{\gamma-1} \times \frac{\cancel{P_1V_1}}{\cancel{P_1V_1}} \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\frac{\cancel{P_1V_1}\left[r_p - 1\right]}{\cancel{P_1V_1}}}$$

$$= \frac{\gamma}{\gamma-1} \times \frac{\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\left[r_p - 1 \right]} \qquad \text{— (3)}$$

The efficiency of roots blower decreas[es] with increase in pressure ratio. However, th[e] compressor is suitable to give a pressure ratio between range of 1 to 3.6.

2S - Two stroke
ICE - Internal combustion engines

Applications

1. Scavenging and supercharging of 2S.ICE.
2. To supply low pressure of air in steel furnaces, sewage disposal plants, low pressure gas boosters, and for blower service in general.

problem

A root blower compresses $1 m^3$ of air per second from a pressure of 1.01325 bar to 1.8 bar. Find the power required to run the compressor and its efficiency

Given data

Volume, $V = 1 m^3/s$

pressure, $P_1 = 1.01325$ bar $= 101.325$ kN/m^2 (or) kPa

$P_2 = 1.8$ bar $= 180$ kN/m^2 (or) kPa

$\Rightarrow 1 \text{ bar} = 1 \times 10^5 N/m^2 \Rightarrow 100 kN/m^2$

Required data

1. power required to run the compressor,
2. Efficiency of compressor.

Solution

Assume, $\gamma = 1.4$

1. The actual power required to run the compressor

$$P = V(P_2 - P_1)$$

$$= 1 \times (180 - 101.325)$$

$$m^3/s \times kN/m^2 \Rightarrow kN\text{-}m/s = kJ/s \Rightarrow kW$$

$$= 78.675 \text{ kW}$$

Isentropic (or) Ideal power required to run the compressor.

$$P_{ise} = \frac{\gamma}{\gamma - 1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$= \frac{1.4}{1.4-1} \times 101.325 \times 1 \left[\left(\frac{180}{101.325}\right)^{\frac{1.4-1}{1.4}} - 1 \right]$$

$$= 63.28 \text{ kW}$$

(ii) Compressor efficiency.

$$\eta_{comp} = \frac{\text{Adiabatic work Input}}{\text{Actual work input}}$$

$$= \frac{63.28}{78.675} = 0.8043 \text{ (or) } 80.43\%$$

—— x ——

prob. 2

1kg of air per second is taken into a root blower compressor at 1bar and 27°c. The delivery pressure of air is 1.5 bar. Calculate the motor power required to run the compressor, if mechanical efficiency is 80%.

Given data:-

mass, $m = 1$ kg/s

suction pressure, $P_1 = 1$ bar $= 100$ kN/m² or kPa

(or) Initial Temperature, $T_1 = 27°c \Rightarrow 27+273 = 300$ K

delivery pressure, $P_2 = 1.5$ bar $= 150$ kN/m² or kPa

Required data:-

1. Motor power required to run the compressor i.e. mechanical η is 80%.

Solution

Volume flow rate, $V = \frac{mRT}{P}$ $\quad PV = mRT$

$$= \frac{1 \times 0.287 \times 300}{100}$$

$$= 0.861 \text{ m}^3/\text{s}$$

The power required to run the compressor

$$P = V_1 (P_2 - P_1)$$

$$= 0.861 (150 - 100)$$

$$= 43.05 \text{ kW}$$

WKT, The mechanical efficiency

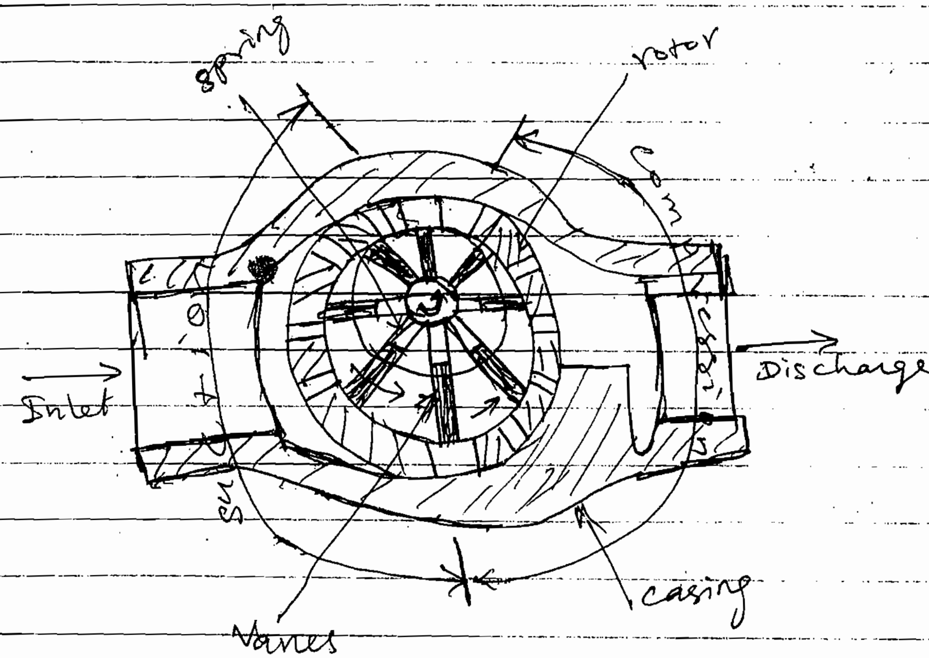
$$\eta_{mech} = \frac{IP}{BP} = \frac{43.05}{BP}$$

$$0.8 = \frac{IP}{BP}$$

$$BP = \frac{43.05}{0.8}$$

$$= \underline{\underline{53.81}}\ kW$$

Vane - Type Compressor :-



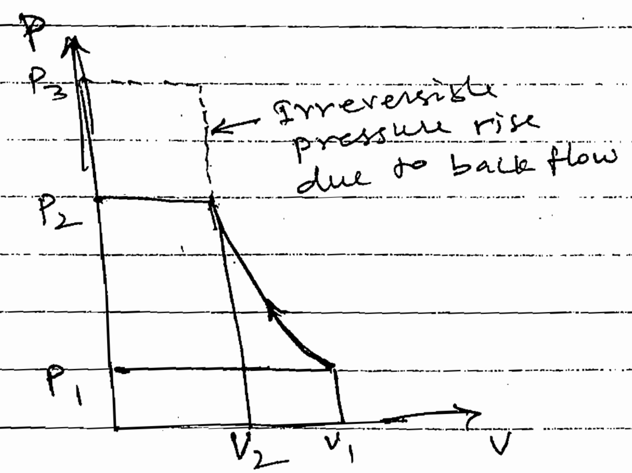
Vane type compressor

Construction :-

An arrangement of a typical Vanes type compressor is shown in fig. It consists of an air tight circular casing, in which a drum rotates about an ecentric centre of casing. The drum consists of a set of Spring-loaded Vanes. The slots are cut in the drum to accomodate the vanes. The drum rotates in anticlockwise direction. During the rotation of the drum, the vane's remain in contact with the casing. Size of inlet pasage is larger than the size of outlet in the compressor.

Working :-

As the drum rotates, the volume of air V_1 at atmospheric pressure P_1 is trapped between the vanes, drum and casing. Air gets compressed due to two operations performed on air. First the compression begins due to decreasing volume between the drum and casing. The volume is reduced to V_2 and pressure increases to P_2. Secondly, the air is compressed due to back flow of compressed air in the receiver. Then air is compressed at constant volume to a pressure P_3. The first part of compression follows adiabatic compression process and the second part follows constant volume process. The process of compression is shown on the p-V diagram in fig.



p-V diagram for Vane type comp

Work input for adiabatic Compression

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

Work Input for Constant Volume Compression,

$$W_{2-3} = V_2 (P_3 - P_2)$$

Total work input for compression within a vane

$$W_{vane} = \frac{\gamma}{\gamma-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + V_2 (P_3 - P_2)$$

If there are N vanes within the drum, then total work input :

$$W_{N\,vane} = N \left[\frac{\gamma}{\gamma-1} P_1 V_1 \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} \right] + V_2 (P_3 - P_2)$$

The efficiency of a vane compressor can be compressed as

$$\eta_{vane,\,comp} = \frac{\text{Work input for constant Volume Compression}}{\text{Total work input for compres}}$$

$$= \frac{V_2 (P_3 - P_2)}{\frac{\gamma}{\gamma-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + V_2 (P_3 - P_2)}$$

* The vane type compressor requires less work input compared to roots blower for same cap and pressure ratio
* It is commanly used to deliver air up to 150 m³ at a pressure ratio up to 8.5
* It can run up to 3000 rpm
* It is used for supercharging of IC engines

and supply of air to cupola (Furnace)

* It's used for construction purpose.

problem

calculat the power required to run the vane compressor and its efficiency, when it handles $6 m^3$ of air per minute from 1 bar to 2.2 bar. The pressure rise due to compression in the compressor is limited to 1.6 bar. Take the mechanical efficiency of compressor as 80%

Given vane compressor type

$V = 6 m^3/min = 0.1 m^3/s$

$P_1 = 1$ bar $= 100$ kPa (or) kN/m^2

$P_2 = 1.6$ bar $= 160$ kPa

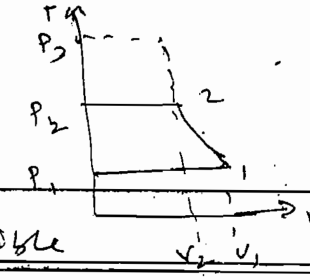
$P_3 = 2.2$ bar $= 220$ kPa

$\eta_{mech} = 0.8$

To find

1. Brake power required to run the Compressor

ii. ~~specific heat ratio for~~ Efficiency of the compressor



Values to be assumed

(i) Compression is reversible

(ii) Specific heat ratio for air as 1.4.

Solution

The isentropic compression power from 1 to 2

$$IP_1 = \frac{\gamma}{\gamma-1} p_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$= \frac{1.4}{1.4-1} \times 100 \times 0.1 \left[\left(\frac{160}{100}\right)^{\frac{1.4-1}{1.4}} - 1 \right]$$

$= 5.03$ kW (or) kJ/s

The volume of air after compression to 1.6 bar

$$V_2 = V_1 \left(\frac{P_1}{P_2}\right)^{1/\gamma}$$

$$= 0.1 \left(\frac{1}{1.6}\right)^{1/1.4}$$

$= 0.07148 \ m^3/s$

Work done due to back flow from 2 to 3

$$IP_2 = V_2 (P_3 - P_2)$$

$$= 0.07148 (220 - 160)$$

$$= 4.29 \text{ kW}.$$

Total power input for compression of air

$$IP = IP_1 + IP_2$$

$$= 5.03 + 4.29 = \underline{\underline{9.32 \text{ kW}}}$$

The mechanical efficiency is given by

$$\eta_{mech} = \frac{IP}{BP}$$

(i) The Brake power required to run the motor or shaft power.

$$BP = \frac{IP}{\eta_{mech}} = \frac{9.32}{0.8} = 11.65 \text{ kW}$$

(ii) The efficiency of Vane Compressor.

$$\eta_{vane\ comp} = \frac{\text{Work Input for Constant Volume Compress'n.}}{\text{Total work Input for Compress'n}}$$

$$= \frac{4.29}{9.32} = 0.46 \text{ or } 46\%.$$

Compare the work required for compression in a root blower and vane blower compressors for the following particulars.

Intake volume, $= 0.05 m^3$ per revolution

Inlet pressure $= 1.013$ bar

Pressure ratio $= 1.5$ for the vane type compr[essor] internal compression takes place through half the pressure range.

Given

Compression with $V_1 = 0.05 m^3/rev$

$P_1 = 1.013 \text{ bar} = 101.3 \text{ kPa}$

pressure ratio $= 1.5$

To find

(1) Work input to root blower and vane-type comp[ressor]

Take, $\gamma = 1.4$, Reversible compression.

(i) For roots blower

pressure ratio, $\frac{P_2}{P_1} = 1.5$

$P_2 = 1.5 P_1$

$= 1.5 \times 1.013$

$= 1.5195$ bar $=$ ~~151.95 kPa~~

Work input per revolution

~~Root~~, $W_{comp} = V_1 (P_2 - P_1)$

$= 0.05\,(151.95 - 101.3)$

$= 2.53$ kJ

(ii) For Vane-type compressor,

pressure ratio, $\frac{P_3}{P_1} = 1.5$

(or) $P_3 = 1.5 P_1$

$= 1.5 \times 1.013 = 1.5195$ bar

$= 151.95$ kPa

Intermediate pressure.

$P_2 = P_1 + 0.5(P_3 - P_1)$

$= 101.3 + 0.5\,(151.95 - 101.3)$

$= 126.625$ kPa.

The volume of air after intermediate compression

$$V_2 = V_1 \left(\frac{P_1}{P_2}\right)^{1/\gamma}$$

$$= 0.05 \times \left(\frac{101.3}{126.625}\right)^{1/1.4}$$

$= 0.0426\ m^3/rev.$

Work input per revolution for Vane-type Compresso

$$W_{vane} = \frac{\gamma}{\gamma - 1} \cdot P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + V_2 (P_2 - P_1) =$$

$$= \frac{1.4}{1.4-1} \times 101.3 \times 0.05 \left[\left(\frac{126.625}{101.3}\right)^{\frac{1.4-1}{1.4}} - 1 \right]$$

$+ 0.0426\,(151.95 - 126.6$

$= 1.167 + 1.079$

$= 2.246$ kJ/rev

Scientist name : Alf Lysholm

LYSHOLM COMPRESSOR - A Screw Compressor.

The Screw Compressors are most commonly used rotary air compressor. They are single stage helical or spiral lobe, oil

There are available in wide ~~ratio~~ range of pressure ratio and capacity. Lubricated types are available in size ranging from 200 to 2000m^3 per hour with discharge pressure up to 10 bar.

The oil free rotary screw air compressor uses specially designed machine. It compresses air without oil in the compression chamber. These compressors are air or water cooled machines. They deliver oil free air up to 30,000m^3/hr and pressure up to 15 bar.

fig Screw compressor

Construction

Screw compressors are equipped with two matting helical grooved rotors housed within a cylindrical casing equipped with inlet and discharge ports as shown in fig.

The main (male) rotor is normally driven by either an electric motor or an engine and transforms about 85-90%. of the energ received at the coupling into pressure and heat energy. The number of lobes (or valleys) on the rotors will vary from one compressor manufactu to another. usually, the male rotor is made with four lobes along the length of the rotor the meshes with similarly formed correspondingl Six helical flute on the auxilliary (female roto

The auxiliary rotors seal the working spa between the suction and pressure side. In th course of rotation, main and auxiliary rotor generate a V-shaped space for the air draw in, which becomes smaller and smaller right pulsation - increase or decrease of pressure, volume

up to the end, between the rotor lobes and cylinder walls.

Because of the number of male lobes, there are four compression cycles per revolution which means that the resulting compressed air has small pulsations compared to reciprocating compressor.

Working:

As rotors rotate, the air is drawn through the inlet port to fill the space between the male lobe and female flute. As rotors continue to rotate, the air is moved past the suction port and sealed in the interlobe space. The trapped air is moved axially and radially and is compressed by direct volume reduction as enmeshing of lobes of compressor progressively reduces the space occupied by the gas with increase in pressure. Simultaneously, with this process, the oil is

injected to the system. The oil seals the interna[l] clearances and it absorbs the heat energy generated during compression. The compression of air continues until the interlobe space communica[tes] with discharge port in the casing. The compress[ed] air leaves the casing through the discharge por[t]

Ruggedness - Roughly irregular surface, strong construction, hard.

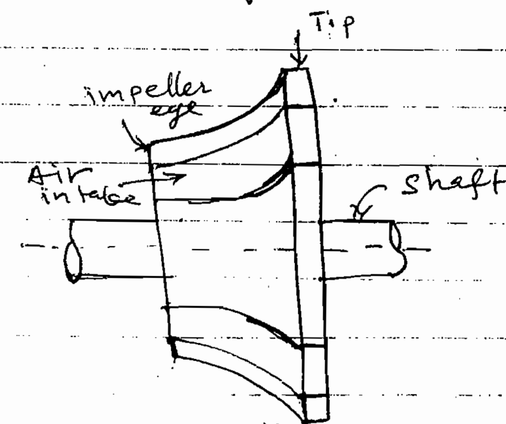
Centrifugal Compressor:-

The centrifugal compressors are dynamic action compressors. These compressors have appreciably different characteristics as compared to reciprocating machines. Centrifugal machines are better suited for very high capacity applications. It's above 3000 m^3/min and a moderate pressure ratio of 4 to 6.

The centrifugal air compressor is an oil-free compressor by design. The oil-lubricated running gear is separated from the air by shaft seals and atmospheric vents. It is a continuous duty compressor, with few moving parts, and is particularly suited for 'high volume specifications', especially where oil-free air is required.

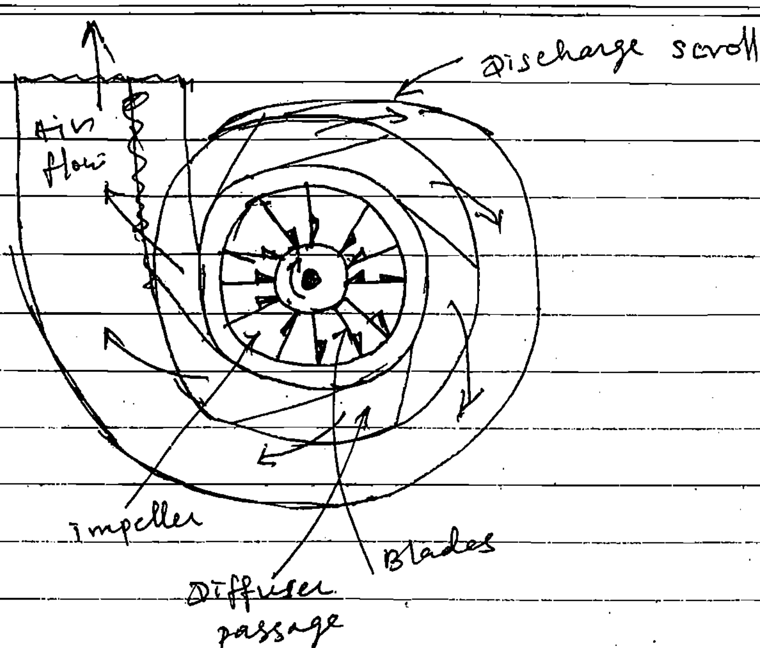
Construction:-

The basic components of a typical centrif Compressor includes an impeller, diffuser and casing as shown in fig. The impeller is a rad disc with a series of radial blades (vanes



Impeller eye.

Static pressure:- on the other hand If you have some fluid with no KE then the pressure exerted by the fluid on its surrounding due its molecular structure of that fluid is known as static pressure. So it is inherent tendency. [Inherent – in-built, inborn]



Sectional view of centrifugal compressor.

Construction:-

The basic components of a typical centrifugal compressor includes an impeller, diffuser and casing as shown in fig. The impeller is a radial disc with a series of radial blades (vanes). The impeller rotates ~~the~~ inside the casing.

* Stagnation properties:- Suppose that a fluid is moving with very high velocity. Due to this velocity, it posses some KE and may be some other types of energy as well. But if through some mechanism this fluid is suddenly brought to ~~rest~~ rest. then according to law of conservation of energy all KE of the fluid converted to pressure energy eff. At ~~this instant all the properties of the corresponding fluid is known~~ this instant all the properties of the corresponding fluid is known as stagnation properties.

The impeller is usually forged or die casting of alluminum alloy.

The centre of the impeller is called the eye. The eye of the impeller is connected with the drive shaft. The casing of the compressor has a volute shape. A diffuser ring is housed in the radial portion of the casing.

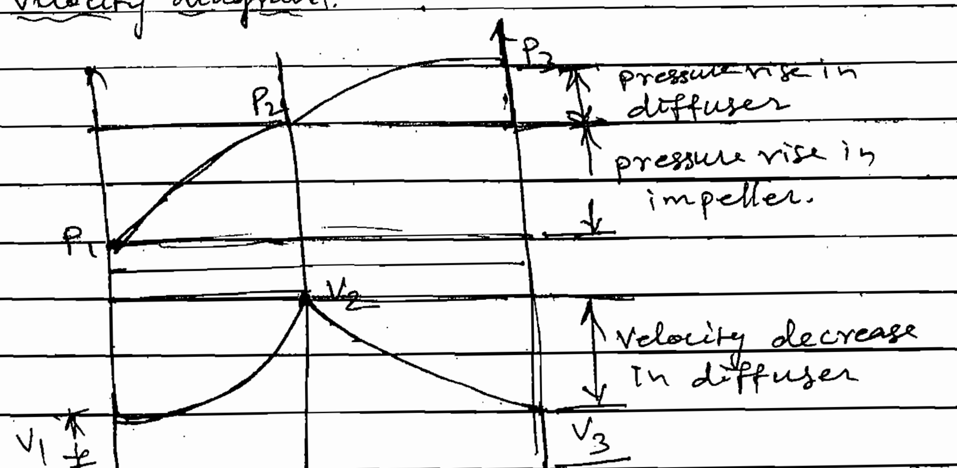
Working:-

As the impeller rotates, the air enters radially into the impeller eye with low velocity V_1 at atmospheric pressure P_1. Due to centrifugal action of the impeller, the air comes radially out and during its movement, it is guided by the blades within the impeller.

The high velocity of the impeller increases the momentum of air, causing rise in static pressure, temperature and kinetic energy of air. The pressure, temperature and velocity of air leaving the impeller are P_2, T_2 and V_2 respectively.

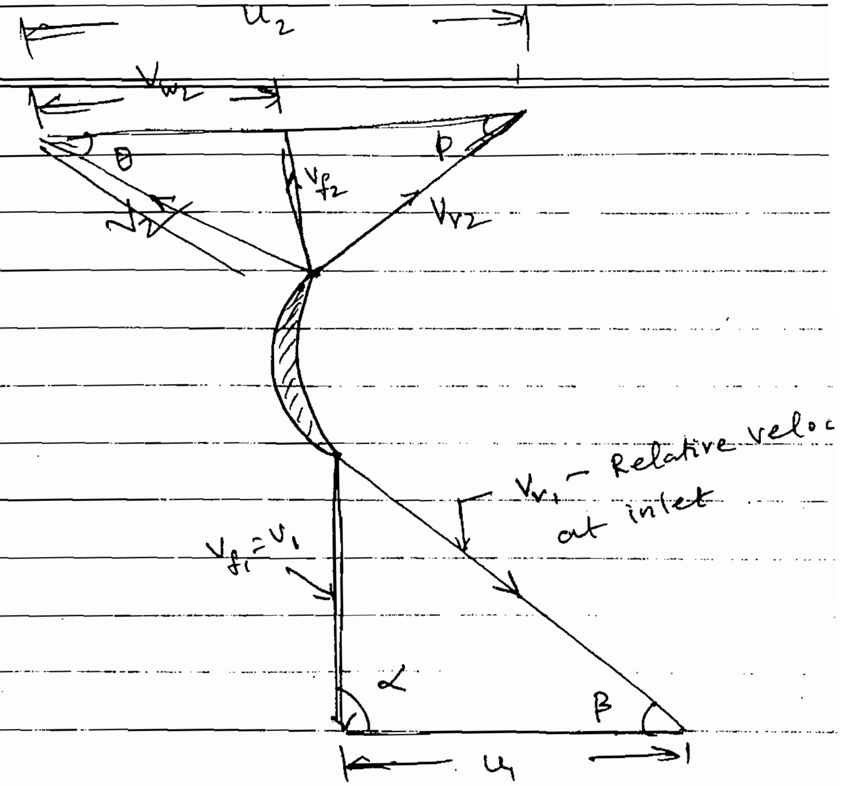
The air leaving the outside edge of the impeller enters into the diffuser ring where its kinetic energy is converted into pressure energy. Thus, the static pressure of air is further increased. The air is then collected in the casing and discharged from the compressor. The change in pressure and velocity of air passing the impeller and diffuser passage are shown in fig. (below)

Velocity diagram:



Velocity and pressure variation of air passing through impeller and diffuser.

$V_{w_1} = 0$, since the working fluid entering to the bla at $\alpha = 90°$.



Velocity diagram of at inlet and outlet of impeller blade.

blades are designed in such a way that the air and leave the blades without shock.

Using usual notation, let

u_1 = Blade velocity at inlet

V_{r_1} = Relative velocity of blades at inlet

V_1 = Absolute velocity of air at inlet
V_{f1} = inlet flow velocity
α = Air inlet angle
β = Blade angle at inlet.
u_2, V_{r2}, V_2, V_{f2}, θ and ϕ are the corresponding values at outlet.
V_{w2} – whirl velocity of air at outlet

Work input per kg of air
$$W = u_1 V_{w1} + u_2 V_{w2}$$
Since the working fluid enters radially, ie, $\alpha = 90°$, thus $V_{w1} = 0$.
Hence work input by the blade per kg of air
$$W = u_2 V_{w2} \ (J/kg)$$
The above equation is known as Euler's equation (or) Euler's work.
If 'm' is the mass flow rate of air in kg/s, then power input to compressor
$$P = m\, u_2 V_{w2} \ \text{Joules}$$

The work transfer per kg of air to centrifugal compressor can also be obtained by using steady flow energy equation.

$$q - w = (h_2 - h_1) + \left(\frac{V_2^2}{2} - \frac{V_1^2}{2}\right) + \Delta Pe$$

For centrifugal air compressor, $q = 0$ change in potential energy, $\Delta Pe = 0$, Omitting the negative sign for work input the compressor work per kg of air,

$$W = \left(h_2 + \frac{V_2^2}{2}\right) - \left(h_1 + \frac{V_1^2}{2}\right)$$

$$W = (h_{02} - h_{01}) = c_p (T_{02} - T_{01})$$

$$= c_p T_{01} \left(\frac{T_{02}}{T_{01}} - 1\right)$$

The static state and stagnation state are shown in fig. The stagnation temperature

T_{01} and T_{02} can be expressed as

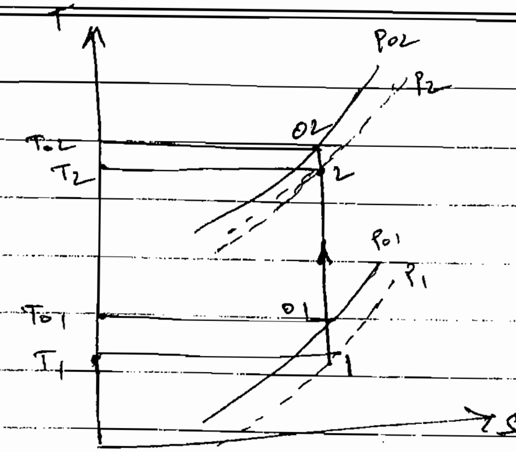
$$T_{01} = T_1 \left(\frac{P_{01}}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_{02} = T_2 \left(\frac{P_{02}}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

and $$\frac{T_{02}}{T_{01}} = \left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}},$$

Hence the work input to the compressor per kg of air can be expressed as

$$W = C_p T_{01} \left\{ \left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}$$



Static and stagnation properties during adiabatic compression.

If $V_1 = V_2$, then stagnation pressure,

$P_{02} = P_1$ (static pressure) and $P_{02} = P$

$T_{01} = T_1$.

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

Width of Blades of impeller and Diffuser.

For constant mass flow rate of air, the width of the blades of impeller can be calculated as follows -

$$m = \frac{AV_1}{v_1} \quad \text{[continuity equation]}$$

where v_1 = specific volume of air at inlet

$A = \pi D_1 B_1$, with D_1, diameter of impeller at inlet and B_1 as width of impeller blades at inlet.

As air trapped radially, $V_1 = V_{f1}$

$$\therefore \dot{m} = \frac{\pi D_1 B_1 V_{f1}}{v_1} \quad \text{—(1)}$$

$$\text{or} \quad B_1 = \frac{\dot{m} v_1}{\pi D_1 V_{f1}} \quad \text{—(2)}$$

Similarly, the width of impeller blades at the outlet can be obtained by using the suffix 2, in eqn. (2)

$$B_2 = \frac{m\, v_2}{\pi D_2 V_{f2}} \quad \text{—(3)}$$

If the number and thickness of blades are considered then the effective area of the blade will be

$$A = \pi D - nt$$

where, n - number of blade, t - thickness of blade

$$\text{Then} \quad B_1 = \frac{\dot{m} v_1}{(\pi D_1 - nt) V_{f1}} \quad \text{—(4)}$$

$$B_2 = \frac{\dot{m} v_2}{(\pi D_2 - nt) V_{f2}} \quad \text{—(5)}$$

Degree of reaction:-

The degree of reaction is defined as the ratio of static pressure rise in the impeller to the total static pressure rise in the compressor. The pressure rise the impeller is equal to change in KE of air in the impeller

ie, $\Delta KE = \frac{V_{r_1}^2 - V_{r_2}^2}{2} + \frac{u_2^2 - u_1^2}{2}$ —(6)

$\frac{V_{r_1}^2 - V_{r_2}^2}{2} \rightarrow$ pressure rise in the impeller due to diffusion action

$\frac{u_2^2 - u_1^2}{2} \rightarrow$ pressure rise in the compressor due to centrifugal action of the impeller.

Total pressure rise in the compressor = work input to the compressor

$\therefore$ degree of reaction

$$R_d = \frac{\text{pressure rise in the impeller}}{\text{pressure rise in the compressor}}$$

$$= \frac{\frac{(V_{r_1}^2 - V_{r_2}^2)}{2} + \frac{u_2^2 - u_1^2}{2}}{u_2 V_{w_2}}$$

$$= \frac{(V_{r_1}^2 - V_{r_2}^2) + (u_2^2 - u_1^2)}{2(u_2 V_{w_2})}$$

$$= \frac{(u_2^2 - u_1^2) + (V_{r_1}^2 - V_{r_2}^2)}{2 u_2 V_{w_2}}$$

(or)

$$= \frac{(u_2^2 - V_{r_2}^2) + (V_{r_1}^2 - u_1^2)}{2 u_2 V_{w_2}}$$

From the inlet velocity triangle

$V_{r_1}^2 - u_1^2 = V_{f_1}^2 = V_{f_2}^2 \quad [\because V_{f_1} = V_{f_2}]$ —

$(a-b)^2 = a^2+b^2-2ab$

From outlet triangle

$V_{r_2} = \sqrt{V_{f_2}^2+(u_2-V_{w_2})^2}$

$V_{f_2}^2 + (u_2 - V_{w_2})^2 = V_{r_2}^2$

(or) $V_{r_2}^2 = V_{f_2}^2 + u_2^2 + V_{w_2}^2 - 2u_2V_{w_2}$

(or) $u_2^2 - V_{r_2}^2 = 2u_2V_{w_2} - V_{f_2}^2 - V_{w_2}^2$ — (9)

Substitute the values of $V_{r_1}^2 - u_1^2$ from eqn (8) and $u_2^2 - V_{r_2}^2$ from eqn (9) in eqn. 7.

$$= \frac{(u_2^2 - V_{r_2}^2) + (V_{r_1}^2 - u_1^2)}{2u_2V_{w_2}}$$

$$= \frac{2u_2V_{w_2} - \cancel{V_{f_2}^2} - V_{w_2}^2 + \cancel{V_{f_2}^2}}{2u_2V_{w_2}}$$

$$= \frac{2u_2V_{w_2} - V_{w_2}^2}{2u_2V_{w_2}}$$

$$= \frac{2u_2V_{w_2}}{2u_2V_{w_2}} - \frac{V_{w_2}^2}{2u_2V_{w_2}}$$

$$\boxed{= 1 - \frac{V_{w_2}}{2u_2}} \quad \text{— (10)}$$

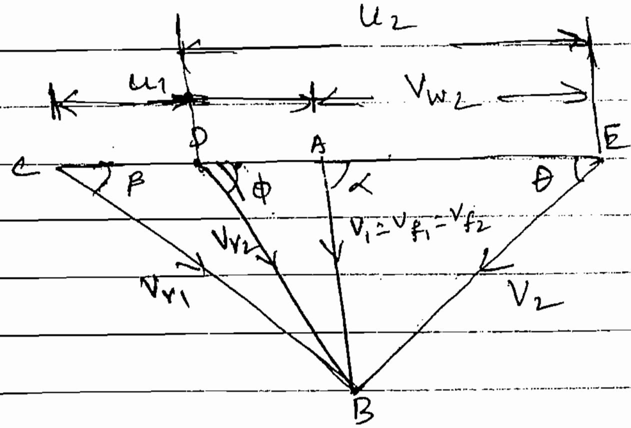
Combined velocity triangle for centrifugal Compressor.

1. Draw a vertical line AB to represent the flo[w] velocity and it remains constant at inlet a[nd] exit.
2. The horizontal line CA represent the bla[de] velocity u_1 at inlet.
3. The line CB inclined at the blade angle β represents the relative velocity V_{r_1} of blad[e] at the inlet.
4. The line DB inclined at blade angle φ

represents the relative velocity V_{r2} of the blade at the outlet.

5. The line DE represents the blade velocity u_2 of impeller at outlet.

6. Join the line EB it represents absolute velocity V_2 of air at the outlet inclined at angle θ with respect to the horizontal



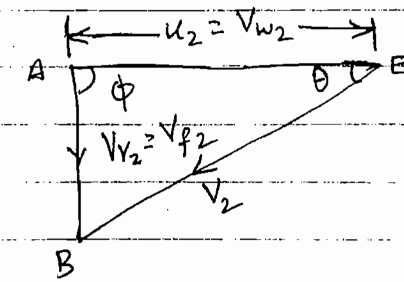
combined velocity triangle for centrifugal compressor.

From the combined velocity triangle, the work inp[ut] to the compressor per kg of air

$$W = u_2 V_{w_2} \cdot (J/kg)$$

power input, $P = \dot{m} W = \dot{m} u_2 V_{w_2}$ watts

If the air flow through the impeller blade radial (Ideal case), the velocity diagram at th[e] outlet takes the shape as shown in fig.



velocity triangle at outlet for ideal air

In this case, the blade velocity at the outle[t] becomes equal to whirl velocity at the outl[et] ie $u_2 = V_{w_2}$, The work input per kg of air

$$W = u_2^2 \ (J/kg) \quad \text{——} \quad (1)$$

The exit whirl velocity V_{w2} of air cannot be geater than the blade tip velocity. Thus, it is the limiting case and it is the maximum work supplied to air per kg.

Losses and efficiency of centrifugal compressor

The following losses occur in a centifugal compressor, when air flows through the impeller.

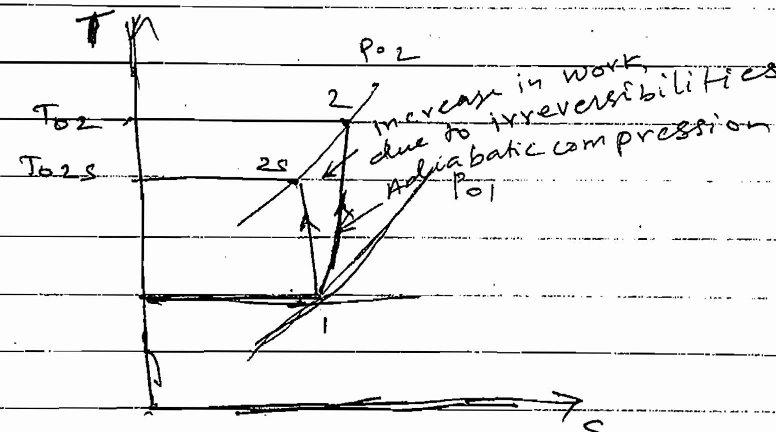
1. Friction between moving air layers and impeller blades and friction between air layers moving with relative velocities

2. Shock at entry, and
3. Turbulence caused in air.

These losses cause in increase in enthalpy of air without increasing the pressure of air. Therefore the actual temperature of air coming out of the compressor is more than the temperature of air

at the inlet. The actual work input for the same pressure ratio is more due to irreversibilities. T actual and isentropic compression for pressure ratio is shown in fig.

Since the cooling arrangement is not provic in dynamic compressors, the ideal compression



Actual and Isentropic compression on a T-s diagram

process in dynamic compressors is isentropic compression. But the actual work input for compr is always more than isentropic work input for compression through same pressure ratio.

$$\eta_{Isent} = \frac{\text{Isentropic work}}{\text{Actual work I/P}} = \frac{h_{02s} - h_{01}}{h_{02} - h_{01}} \quad \text{—} (2)$$

If sp-heat C_p of air remains constant then $\boxed{h = C_p T}$

$$\eta_{Isent} = \frac{T_{02s} - T_0}{T_{02} - T_{01}} \qquad \frac{C_p h_{02s} - C_p h_{01}}{C_p h_{02} - C_p h_{01}} \quad \text{—} (3)$$

If inlet velocity of air ~~remains constant then~~ is equal to air exit velocity,

ie $V_1 = V_2$ then

$$\eta_{Isen} = \frac{T_{2s} - T_1}{T_2 - T_1} \quad \text{—} (4)$$

$$= \frac{\text{Isentropic temperature rise}}{\text{Actual temperature rise.}}$$

T_1 - Inlet state,

T_{2s} - After isentropic compression

T_2 - After actual ~~temperature rise~~ expansion.

T_{01}, T_{02} - Corresponding stagnation properties.

Slip and slip factor.

A centrifugal compressor has maximum work i[...] when $u_2 = V_{w2}$. In actual practice, the whirl veloc[...] V_{w2} is always less than the blade tip velocity u[...]

* slip :- The difference between blade tip veloc[...] u_2 the whirl velocity V_{w2} is known as sl[...]

ie $\text{slip} = u_2 - V_{w2}$ —(5)

Slip factor : It is defined as the ratio of whirl velocity to blade tip velocity. It is denoted b[...] Φ_s.

$$\text{slip factor}, \ \Phi_s = \frac{V_{w2}}{u_2} \quad \text{—} (6)$$

Work Factor and pressure coefficient

The actual work done per kg of air by compre[...] is always greater than the impeller work $u_2 V_{w2}$ de[...]

to fluid friction and windages losses. work factor or power input factor is defined as ratio of actual work input to Euler work input. It is designated as ϕ_w and is given as

$$\phi_w = \frac{\text{Actual work input}}{\text{Euler work input}}$$

$$= \frac{C_p(T_{02} - T_{01})}{u_2 V_{w2}} \quad \text{—— (6)}$$

pressure co efficient is defined as the ratio of isentropic work to Euler work. It is designed as ϕ_p.

$$\phi_p = \frac{\text{Isentropic work input}}{\text{Euler work input}}$$

$$C_p = \frac{C_p(T_{02s} - T_{01})}{u_2 V_{w2}} \quad \text{—— (7)}$$

using eqn (3) and assuming radial vanes of impeller $u_2 = V_{w2}$, then

$$\phi_p = \frac{\eta_{isen} C_p (T_{02} - T_{01})}{u_2^2} \quad \text{—— (8)}$$

From eqn (6) we get

$$C_p(T_{02} - T_{01}) = \phi_w u_2 V_{w2}$$

substituting the V_{w2} as $\phi_s u_2$ from eqn. (5) we g

$$C_p(T_{02} - T_{01}) = \phi_w \phi_s u_2^2$$

from eqn. (8), we get

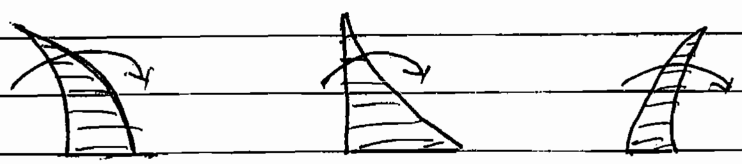
$$\phi_p = \frac{\eta_{isen} \phi_w \phi_s u_2^2}{u_2^2} = \phi_w \phi_s \eta_{isen} \quad \text{—— (9)}$$

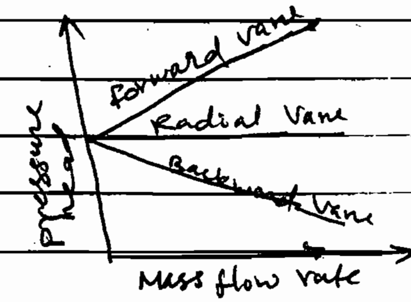
Effect of Impeller Blade Shape on Compressor performance:-

The Three type of impeller blade shapes used in a centrifugal Compressor. These are

1. Backward curved blades ($\theta < 90°$)
2. Radial blades ($\theta = 90°$)
3. Forward curved blades ($\theta > 90°$)

Figure shows the geometry of backward radial and forward curved vanes and performance of these vanes. The centrifugal action on the curved vanes creates bending moment and induces bending stresses.





Charecteristics of backward, radial and forward curved vanes.

The pressure head of delivered air decreas with increase in mass-flow rate for backwar curved vanes, while the pressure head increa for forward curved vanes. But for radial curved vanes, the pressure head of delivered air remains constant with mass-flow rate.
The backward vanes are normally used w $\theta = 20°$ to $25°$, except for delivery of air at high head. The radial vane is the Compromise between backward and forwar

curved vanes. ∴ the radial vane impeller is most commonly used in a centrifugal compressor due to the following reasons.

1. Radial-vane geometry is simple, thus vanes can be manufactures easily

2. Radial vane have lowest bending stress for given diameter and speed as compared to forward and backward curved vanes

3. It have constant pressure head in the impeller as well as in the diffuser

4. It has good efficiency and high pressure head.

Diffuser System:

In a centrifugal compressor, the diffuser converts kinetic energy of air into static pressure head. For a radial vane impeller, the diffuser contributes about one half of the overall static pressure rise.

Unit-V

Refrigeration And Air Conditioning:-

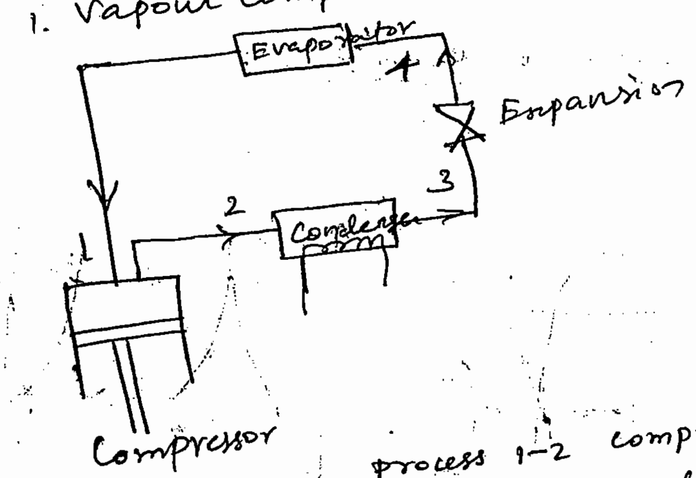
Introduction To Refrigeration:-

Refrigeration aims at keeping space at a temperature lower than that of the surroundings.

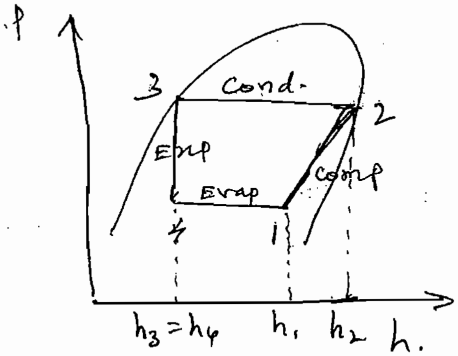
Types of Refrigeration system.

1. Vapour compression Refrigeration systems
2. Vapour Absorption systems.

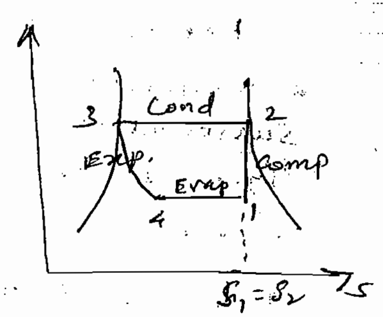
1. Vapour compression Refrigeration system.



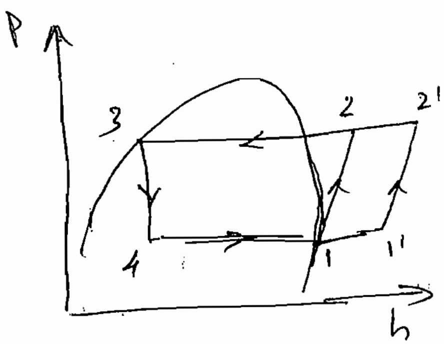
process 1-2 compression
2-3 condensation
3-4 Expansion
4-1 Evaporation.

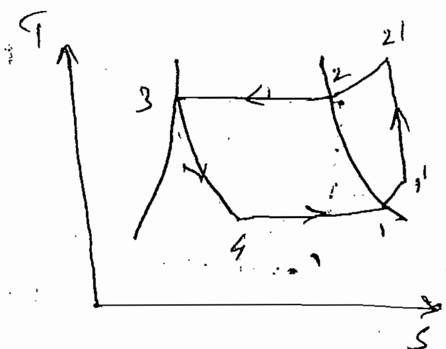


$h_{f3} = h_4.$



Superheating of Refrigerant





i) Specific volume increases from v_1 to v_1'

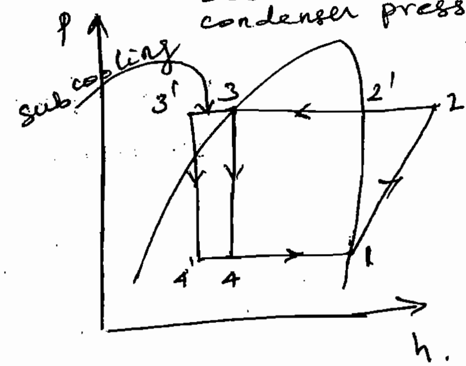
ii) Refrigeration effect increases from $h_1 - h_4$ to $h_1' - h_4$.

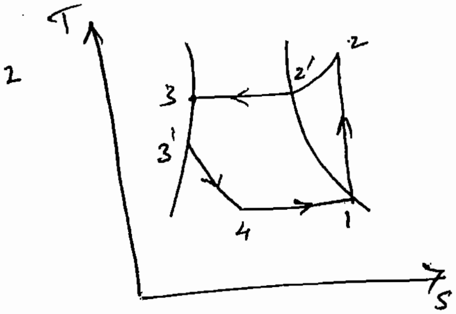
iii) Specific work increases from $h_2 - h_1$ to $h_2' - h_1'$

iv) Coefficient of performance is reduced due to increase in compressor work.

Sub cooling of Refrigerant.

The vapour is cooled at constant pressure below the saturation temperature at the condenser pressure.





1. Refrigeration effect is increased from $h_1 - h_4$ to $h_1 - h_4'$

2. Compression work remains constant.

3. Coefficient of performance is increased.

performance calculations.

The performance of the refrigeration system is generally measured by a factor called coefficient of performance. The refrigeration load on the system is given interms of tonnes of refrigeration.

A tonne of refrigeration is defined as the quantity of heat required to be removed from one tonne of water (1000 kg) at 0°c to convert that into ice at 0°c in 24 hours. in actual practice

$\frac{1000 \times 333.43}{24 \times 60}$

1 tonne of refrigeration = 210 kJ/min = 3.5 kW

Latent heat of water at 0°c } 333.43 kJ/kg

1. COP is defined as the ratio of heat absorbed by the evaporator or refrigeration effect to the compressor work.

$$COP = \frac{\text{Refrigeration effect}}{\text{work done}}$$

$$= \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

2. The quantity of refrigerant required to be circulated in the refrigerator plant is given by

$$m = \frac{3.5T}{h_1 - h_{f3}} \text{ kg/sec}$$

T = Load in tonne of refrigeration

h_1 = Enthalpy at entry of Compressor

h_{f3} = Enthalpy at entry of the evaporator (or) enthalpy at exit of the condenser,

3. Power required to run the compressor.

$$P = m(h_2 - h_1) \text{ in kW.}$$

4. The quantity of cooling air or water required by the condenser per second is

$$m_w C_{pw} \Delta T = m(h_2 - h_3)$$

C_{pw} = specific heat of cooling air (or) water

m_w = mass of cooling air (or) water

ΔT = Rise in temperature of the cooling air (or) water.

5. The dimensions of a single acting compressor are given by.

$$\left(\frac{\pi}{4} D^2 L\right) \eta_v \frac{N}{60} = m V_{s1}$$

D - Diameter of the cylinder

L - stroke length of the cylinder

N - rpm of the compressor.

η_v - Volumetric efficiency of the compressor.

V_{s1} - Specific volume of saturated refrigerant at point 1.

Psychrometry

The atmosphere air generally consists of water vapour. The amount of water vapour plays important role in psychrometry. If it exceeds (or) lowers certain limit, it will creat discomfort to the men. so it is very important to keep the moisture content in the air within the specified limit in case of processing industries and air conditioned buildings.

→ The science which deals with the study of behaviour of moist air (mixture of dry air and water vapour) is known as psychrometry.

Psychrometry properties:-

(1) Dry air:- Air without moisture (or) water vapour. The pure dry air mixture of nitrogen, oxygen, carbon dioxide, hydrogen, etc., 77% of nitrogen and 23% of oxygen others are negligible quantity.

(2) Moist air:- It is a mixture of dry air and water vapour. The amount of water vapour present in the moist air varies with temperature.

(3) Saturation capacity of air:- The quantity of water vapour present in air at particular air temperature is known as saturation capacity of air.

(4) Moisture:- The water vapour present in the air is known as moisture.

(5) Dry bulb temperature:- (DBT) (t_d)
The temperature measured by an ordinary thermometer is known as dry bulb temperature. It is generally denoted by t_d.

(6) Wet bulb temperature (WBT) t_w:
It is the temperature of air measured by a thermometer when its bulb is covered with wet cloth and is exposed to a current rapidly moving air. It is denoted by t_w.

(7) Wet bulb depression (WBD)

$$WBD = DBT - WBT$$

The value of WBT is zero when the air becomes saturated.

(8) Dew point temperature (DPT) t_{dp}.
It is the temperature at which the water vapour present in air begins to condense when the air is cooled.

(9) Dew point depression (DPD)

$$DPD = DBT - DPT$$

(10) Specific humidity (or) Humidity ratio (or) moisture content (ω).
It is defined as the mass of water vapour present in one kg of dry air. It is the ratio of the mass of water vapour to the mass of dry air in a given volume of the moisture.

$$\text{Sp. humidity}, \omega = \frac{\text{Mass of water vapour}}{\text{Mass of dry air}} = \frac{m_v}{m_a}$$

$$\omega = 0.622 \frac{p_v}{p_b - p_v}$$

11. Degree of Saturation (or) percentage saturation (μ) or saturation ratio.

It is defined as the ratio of specific humidity of the moist air to the specific humidity of saturated air at the same temperature.

$$\mu = \frac{\text{specific humidity of moist air}}{\text{specific humidity of saturated air}}$$

$$\mu = \frac{P_v}{P_s}\left[\frac{P_b - P_s}{P_b - P_v}\right]$$

12. Relative humidity : ϕ

It is defined as the actual mass of water vapour in a given volume to the saturated mass of water in same volume and temperatures.

$$\phi = \frac{m_v}{m_s} \quad \text{ie, } \phi = \frac{P_v}{P_s}$$

$$\phi = \frac{\mu}{1-(1-\mu)\frac{P_s}{P_b}} //.$$

13. Total enthalpy (total heat) of moist air (h) :-

Total enthalpy of moist air is the sum of the enthalpy of dry air and the enthalpy of water vapour associated with the dry air.

$$H = C_p t_d + w h_g$$

C_p - specific heat at constant pressure = 1.005 kJ/kg K

t_d - Dry bulb temperature

w - specific humidity

h_g - specific enthalpy of a air corresponding to dry bulb temperature.

Daltons Law of partial pressure :-

ie $P_b = P_a + P_v$.

$$P_v = P_{sw} - \frac{(P_b - P_{sw})(t_d - t_w)}{1527.4 - 1.3 t_w}.$$

P_{sw} - Saturation pressure corresponding to wet bulb temperature from steam table.

t_d - dry bulb temp.

t_w - wet " "

A Simple R-12 plant is to be develop 5 tonnes of refrigeration. The condenser and evaporator temperature are to be 40°c and −10°c respectively. Determine.

(i) The refrigerant flow rate in kg/s

(ii) The volume of flow rate handled by the compressor in m^3/s

(iii) The compressor discharge temperature.

(iv) The heat rejected to the condenser in kW

(v) The COP and

(vi) The power required to drive the compressor

How does this COP compare with that of a carnot refrigerator operating between 40°C and −10°C

Solution

Given data

Capacity of refrigerator = 5 tonnes

Condenser Temp $T_2 = 40°C$

Evaporator Temp $T_1 = -10°C$

Required data.

(1) Mass of refrigerant in kg/s

At $-10°C$

$h_{f_1} = 190.822$ kJ/kg $h_{g_1} = 347.134$ kJ/kg

$V_{f_1} = 0.7 \times 10^{-3}$ m^3/kg

$S_f = 0.96601$ kJ/kgK $S_g = 1.55997$ kJ/kgK, $V_g = 0.0766\ m^3/kg$

At 40°C

$h_{f_2} = 238.535$ kJ/kg, $h_{g_2} = 367.146$ kJ/kg, $V_{f_2} = 0.798 \times 10^{-3}\ m^3/kg$

$S_{f_2} = 1.12984$ kJ/kgK, $S_{g_2} = 1.54051$ kJ/kgK, $V_{g_2} = 0.0187\ m^3/kg$

(i) Mass of refrigerant.

capacity of Ref $= \dot{m}(h_1 - h_{f_3})$

$3.5 \times 5 = m(347.134 - 238.535)$

$= \underline{0.161\ kg/s}$

(ii) Volume flow rate.

$V = m(V_{g2} - V_{f2})$

$= 0.161 \times (\cancel{0.0766}\ 0.01817 - 0.798 \times 10^{-3})$

$= \underline{0.0122\ m^3/s}$

$V = mV_{g}$

Compressor discharge temperature.

$T_2 = 40°C = 313K$

(iv) Heat rejected to the condenser,

$= m(h_2 - h_3)$

$= 0.161(367.146 - 238.535)$

$= \underline{20.7 kW}$

(v) $COP = \dfrac{h_1 - h_{f3}}{h_2 - h_1}$

$= \dfrac{347.134 - 238.535}{367.146 - 347.134}$

$= \underline{5.426}$

(vi) Power, $P = m(h_2 - h_1)$

$= 0.161(367.146 - 347.134)$

$= \underline{3.22 kW}$

(vii) Carnot $COP = \dfrac{T_2}{T_1 - T_2}$

$= \dfrac{263}{313 - 263}$

$= \underline{5.26}$

$s_1 = s_{g,2} \; 1.557$ kJ/kgK

$$s_1 = s_2 = s_{g2} + c_p \ln \frac{T_{sup}}{T_{sat}} \qquad s_{g2} = s_2'$$

$$1.557 = 1.542 + 1.884 \ln\left(\frac{305}{T_{sat}}\right)$$

$$1.884 \ln\left(\frac{305}{T_{sat}}\right) = 0.015$$

$$\ln\left(\frac{305}{T_{sat}}\right) = \frac{0.015}{1.884}$$

$$\frac{305}{T_{sat}} = e^{0.007961783}$$

$$T_{sat} = \frac{305}{1.00799}$$

$$= \underline{\underline{302.58\,K}}$$

$h_2' = h_{g2}$

$$h_2 = h_2' + c_p\left[T_{sup} - T_{sat}\right]$$

$$= 264.5 + 1.884\left[305 - 302.58\right]$$

$$= \underline{\underline{270.152}}$$

$$cop = \frac{h_1 - h_{f2}}{h_2 - h_1}$$

$$= \frac{249.3 - 130.5}{270.1 - 249.3}$$

$$= \underline{\underline{6.}}$$

$h_4 = h_{f2}$

Refrigerant effect $= m(h_1 - h_4)$

$$5 \times 210 = m(249.3 - 130.5)$$

$$m = \underline{\underline{8.84}}\ \text{kg/min}$$

power required to drive the compressor

$$C.O.P. = \frac{\text{Ref Effect}}{\text{work input}}$$

$$\text{work input} = \frac{5 \times 210}{6}$$

$$= 175\ \text{kJ/min}$$

$$= \underline{\underline{2.92\ kW}}$$

An NH_3 refrigerator produces 30 tons of ice from and at 0°c in a day of 24 hours. The temperature range in the compressor is from 25°c to −15°c. The vapour is dry saturated at the end of compression. Assume a COP 60% theoretical. Calculate the power required to drive the compressor. Assume latent heat of ice is 335 kJ/kg. For properties of NH_3 refer table or charts.

Temp	h_f kJ/kg	h_g kJ/kg	S_f kJ/kgK	S_g kJ/kgK
25	298.9	1465.8	1.124	5.039
−15	112.34	1426.5	0.4572	5.549

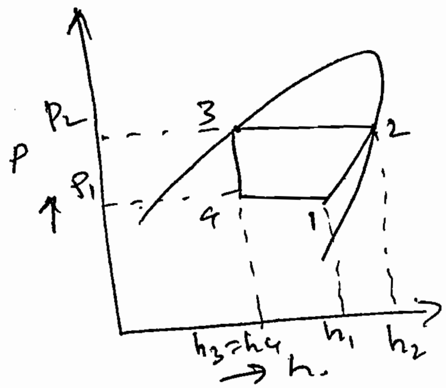
Latent heat of ice = 335 kJ/kg

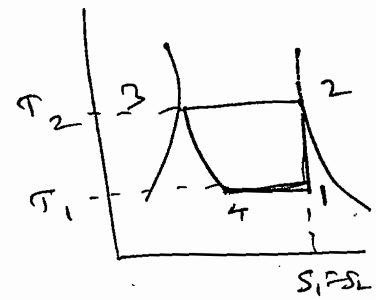
$COP_{theo} = 60\%$.

Reqd. data

(1) power required to drive the compressor

Solution





$S_1 = S_2 = S_{f_1} + x_1 S_{fg_1}$ $\quad [S_{fg_1} = S_{g_1} - S_{f_1}]$

$5.039 = 0.4572 + x_1[5.549 - 0.4572]$

$x_1 = \underline{\underline{0.90}}$

$h_1 = h_{f_1} + x_1 h_{fg_1}$

$= 112.34 + 0.90[1426.5 - 112.34]$

$= 1295.084$ kJ/kg

$h_2 = h_{g_2} = 1465.8$ kJ/kg

$$COP = \frac{h_1 - h_4}{h_2 - h_1} \qquad h_4 = h_{f2}$$

$$= \frac{1295.084 - 298.9}{1465.8 - 1295.084}$$

$$= \frac{996.184}{170.716}$$

$$COP = 5.835$$

Actual COP = 60% of theor

$$= 0.6 \times 5.835$$

$$= \underline{3.501}$$

Power required.

Actual Heat removal rate

$$= \cancel{\frac{30 \times 335}{105}}$$

$$= \frac{30 \times 1000 \times 335}{24 \times 3600}$$

$$= 116.31$$

$$\text{Actual cop} = \frac{\text{Heat removal rate}}{\text{Power required}}$$

$$= \frac{116.31}{3.501}$$

power required = $\underline{33.221}$ kw

A Freon 12 refrigerant producing a cooling effect of 20 kJ/s operates on a simple cycle with pressure limits of 1.509 bar and 9.607 bar. The vapour leaves the evaporator dry saturated and there is no undercooling. Determine the power required by the machine. If the compressor operates at 300 rpm and has a clearance volume of 3% of compressor swept volume, assume that the expansion following the law $pV^{1.3}$ = const.

	T	P bar	v_g m³/kg	h_f kJ/kg	h_g kJ/kg	s_f kJ/kg K	s_g kJ/kg K
2	40	9.607	-	74.53	203.05	0.2716	0.682
1	-20	1.509	0.1088	17.8	178.61	0.073	0.7082

~~Sp~~ heat = 0.747 kJ/kgK