Unit I IC engines

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Classification of IC engines:-

1) Type of cycle
2) Type of fuel
3) Cycle of combination
4) Method of ignition.
5) Number of cylinders
6) Method of cooling
7) Arrangement of cylinder.
8) Speed of engine
9) Method of governing etc...

Classification of engnies according to cylinder arrangements:-

Inline engines:- It has an advantage of incorportaing all the cylinders linearly in order to transmit power to a single crank shaft.

'V' Engines:-
These engines are used in high powered automobiles as it incorporates number of cylinders. It has two-in line cylinders inclined at an angle to a single crank shaft.

Opposed cylinder engine:-
It is inherently a well-balanced engine and has an advantage of a single crank shaft because it comprises of two-in-line arrange-ments at 180° apart.

Opposed piston engine:-
This engine does not require any cylinder head apart from being well balanced. Also, for a given crank and piston speed the relatively piston velocity rate is doubled.

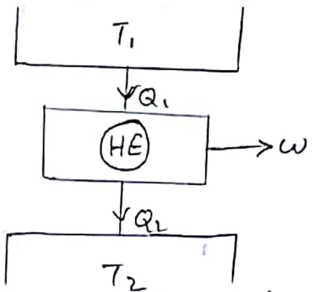
Requirements of an I.C engine:-

The requirements of an IC engine are as follows

1) The engine should develop high HP.
2) The weight per unit power required should be low.
3) It should have high thermal and mechanical efficiency
4) It should be simple and compact in size.
5) Initial cost should be low.

Heat engine:-

A device which gives o/p of work by absorbing some form of energy as input in a thermodynamic cycle, is called as 'heat engine'.



Consider an internal combustion engine in which the working fluid is air-fuel mixture. This, air fuel mixture is compressed and ignited so that heat energy is relased. Then the expansion stroke and lastly exhaust stroke takes place. All these four process continute a thermodynamic cycle in which the engine is absorbing heat energy as input and giving output as shaft power. Therefore an I.C engine qualifies to be heat engine.

Merits of two-stroke engine:-

1) low power to start.
2) Cost is less.
3) Power obtained is more in two-stroke engine, theoretically two times that of four-stroke engines.

4) The torque obtained is uniform and hence lighter fly wheel is required.

5) Simple in construction due to absence of valve, valve mechanism, camshaft, cams etc...

6) weight per kilowatt ratio of two-stroke engine is considerably less.

Demerits of two-stroke engines

1) Volumetric efficiency is less due to lesser time for induction.

2) Thermal efficiency is less.

3) Capacity of cooling system required is more.

4) Because of very high temperatures, more lubricating oil is required

Ignition required in an I.C engine:-

In an internal combustion engine the engine cycle is executed in the following four processes. They are.

1) Suction stroke
2) Compression stroke
3) Expansion stroke.
4) Exhaust stroke.

An IC engine is a device in which chemical energy is converted to heat energy and than into mechanical energy. In order to release the heat energy from chemical energy, the air fuel mixture must be ignited at the end of compression stroke. In a spark ignition the mixture is ignited using a spark.

Cylinder row:-

This is a type of arrangement of cylinders in multi-cylinder engines. In this arrangement, the centre-line of the crank ~~shaft journals is perpendicular to the plane containing the centre~~ lines of the engine cylinders.

In this arrangement, the centre line of the crankshaft journals is parallel to the plane containing the centre lines of the engine cylinders.

Carburetion:

Carbureation is the process of mixing of combustible fuel air -mixture to form a proper ratio of fuel and air before entering into the combustion chamber of engine. The device which performs Carb -uretation is called as "Carburettor".

Carbureation is generally done in S.I Engine so that the required mixture of fuel and air are entred into the cylinder at all operating conditions.

Importance of Cooling in an IC engine:

In IC engines, large amount of heat is relased due to Comb -ustion of fuel in the cylinder. Among this, only 25% of the heat Produced is converted into useful work and the remaining heat dissipate as loosses. The various heat looses are, heat lost to the cylinder walls (30%), loss in exhaust gases (35%) and friction losses (10%). The heat loss to the cylinder wall is considerable and if it is not removed then, it may leads to seizure of the Piston burning of lubricating oil, Preignition of the charge and loss in mech -anical Properties of cylinder material.

Cooling system

In this method, air allowed to flow over the heated surface. The circulating air absorbs certain amounts of the heat from the surface, thereby reducing the temperature. Generally, fins provided over the engine cylinder to increase the conductive and radiative surface for effective cooling. These fins are either cast as an integral part of the cylinder of the surface depends on the surface area in contact with the air, mass flow rate of air, the conductivity of the metal and the temperature difference between the air and heated surface. These systems are generally used in industrial and agricultural engines, small engines and low duty engines.

Lubrication:

Lubrication is defined as the application of lubricant between two surfaces which are in contact and in a relative motion with respect to one another.

Need for lubrication in I.C engines:

1) To reduce friction between the moving parts.
2) To reduce wear of the moving parts.
3) To carry much of the heat generated by friction.
4) To reduce engine noise.
5) To reduce the temperature of the moving parts and thus prevent sizure.

Filtration of oil is necessary in lubrication:-

All the oil is used for lubrication before reaching the engine bearing is allow to pass through an oil filter from the Pump. The bearings are likely to be damaged, if any foreign materials are allowed to enter the lubrication line as because they are machined to a very close tolerance. The function of the filter is to seperate the oil from the abrasive particles ~~if not~~ which are extremely small in nature and these abresive particles if not removed flows freely and cause wear of the working surfaces. Also the function of the filter is to prevent the deposition of sludge on the bearings.

Purpose of engine testing:-

The engine tests determines the measures to be taken to improve the performance according to the manufacture specifications. The basic measurements to evalute the performance are,

1.) Measurement of speed by tachometers
2.) Fuel measurement for fuel consumed.
3.) Measurement for the consumption of air.
4.) Measurement for the brake power and
5.) Exhaust smoke measurement.

Test of I.C engine:-

The various measurements that are to be taken in a test of an I.C engine are as follows

1.) friction power

2) Indicated Power.
3) Brake Power
4) Fuel Consumption
5) Air flow
6) Speed
7) Emissions
8) Noise
9) Combustion phenomenon.
10) Coolant and exhaust temperature.

Mean effective Pressure:-

It is the average pressure acting on a Piston of an Internal Combustion engine.

The mean effective pressure is calculated based on power output & can be calculated from Indicated diagram. In IC engine pressure in the cylinder varies with the position of Piston as hypothetical pressure is acting on the Piston throughout the power stroke. Mean effective pressure is given by

$$P_m = \frac{\text{work done}}{\text{Stroke volume}}$$

They are two types of mean effective pressure. They are

1) Indicated mean effective pressure.
2) Brake mean effective pressure.

friction power:-

It is defined as the diff b/w the indicated power and brake power of an engine. In an engine, the friction loss occurs between the cylinder walls and Piston, Piston rings and cylinder walls and also in between crankshaft and camshaft and their bearings.

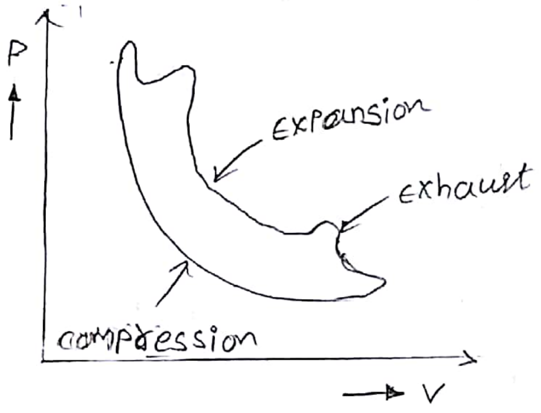
...ds for measuring friction Power:-

The various methods for measuring friction Power to deter-mine the indicated Power and mechanical efficieny are as follows.

1.) Morse test
2.) Motoring test
3.) willans line method.
4.) Diff b/w IP and BP.

P-V diagram of an IC engine:-

The fig shown is a P-V diagram for a two stroke engine where P stands for pressure and V stands for volume of the gasses inside the combustion Chamber. It shows the variations of the Pressure and volume of the gases.



Importance of P-V diagram:-

1.) By this diagram it can be known at what pressure and volume the gases are compressed for ignition.

2.) we can also know the direct stages from which point to a stage of stroke is occuring.

Three Principal factors that influence the engine performance:

The three principal factors that influence Performance are,

1.) Combustion rate and spark timing
2.) Air-fuel ratio
3.) Compression ratio.

1) Combustion Rate and Spark timing:-

For smooth running of the engine, the spark timing and control of the combustion rate should be such that maximum pressure occurs at the beginning of the power stroke.

2) Air-fuel Ratio:-

According to the engine requirements, the air-fuel ratio is maintained.

3) Compression Ratio:-

As compression ratio increases it increases thermal efficiency. But, increase in compression ratio also increases friction of the engine, therefore beyond certain point it is not preferable.

$$'r' = \frac{V_1}{V_2} = \frac{V_s + V_c}{V_c} = \frac{\text{stroke volume} + \text{Clearance volume}}{\text{Clearance volume}}$$

Classification of IC engines:-

Internal combustion engines are classified based upon the following parameters.

(1) Nature of thermodynamic cycle

(a) otto cycle engines.

(b) Diesel cycle engines

(c) Dual combustion cycle engines.

(2) Types of fuel

(a) petrol engine

(b) Diesel engine

(c) Bio-fuel engine

(3) Number of strokes

(a) Two stroke engine

(b) Four stroke engine

(4) Number of cylinders

(a) single cylinder engines

(b) Multi cylinder engines.

5) Methods of Ignition

(a) Spark Ignition (SI) engines
(b) Compression Ignition (CI) engines

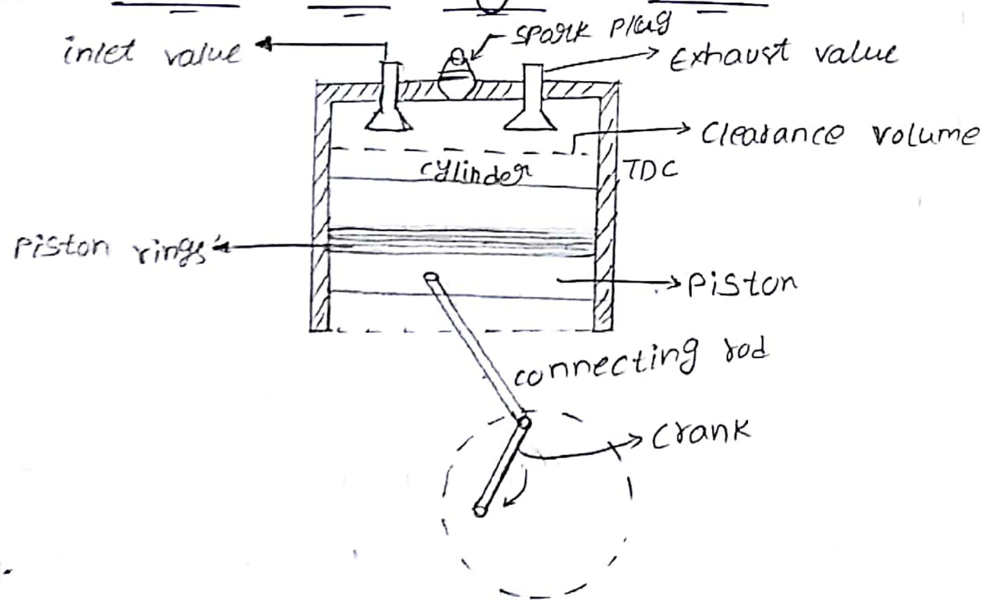
6) Position of cylinders
(a) Horizontal engine.
(b) vertical engine.
(c) V-engines
(d) In-line engines
(e) opposite cylinder engines
(f) Radial engines.

7) Method of Cooling
(a) Air cooled engines
(b) water Cooled engines

8) Speed of engine
(a) Slow speed engines
(b) Medium speed engines
(c) High speed engines

9) Methods of use
(a) Stationary engines
(b) Automobile engines
(c) Portable engines
(d) Aero engines
(e) Marine engines
(f) locomotive engine.

Construction details of IC engine mechanism:-



Cylinder:-

Cylinder is considered as the heart of engine in which the combustion of fuel takes place. The inside diameter of the cylinder is called as bore. In order to protect cylinder from wear liners & sleeves are sometimes inserted into the cylinder.

Piston:

Piston is a close fitted cylindrical plunger which moves to and fro inside an engine cylinder. The main function of piston is to transmit the force exerted due to combustion of fuel to the connecting rod which in turn transmits it to the crankshaft to produce mechanical power.

Piston rings:-

The piston rings are the metallic rings inserted into the circumferential grooves provided at the top end of the piston. These rings maintain a gas tight joint between the piston and the cylinder while piston and the cylinder while the piston is reciprocating in the cylinder.

Connecting rod:-

It is the link that connects the piston and the crankshaft by means of pin joints. Therefore, it should be designed and manufactured carefully.

Crank and crankshaft:-

A crank is a lever that is connected to the end of the connecting rod by a pin joint with its other end and connected rigidly to a shaft called crankshaft.

Valves:-

The valves are the devices which controls the flow of the intake charge and the exhaust gases to release from the engine cylinder respectively.

Fly wheel:-

It is a heavy wheel mounted on the crankshaft of the engine to maintain uniform rotation of the crank shaft.

Crank Case:

It is the lower part of the engine serving as a enclo-sure for the crankshaft and also as a pump for lubricating oil.

TDC (Top Dead Centre):

Top dead centre position is defined as the posi-tion of the piston when it is at its top most popular position & when the volume in the cylinder is the lowest.

BDC (Bottom Dead Centre):

It is the position of the pistion when the volume in the cylinder is maximum & when the piston is at its lower position.

Sweept Volume:

It is the volume swept by the piston when moving from BDC to TDC & vice-versa. It is denoted by V_s.

Clearance Volume:

It is the volume ~~swept~~ remains in the cylinder when the piston is in TDC position. It is denoted by V_c.

$\therefore$ Total volume inside engine cylinder

= Swept volume + Clearance volume

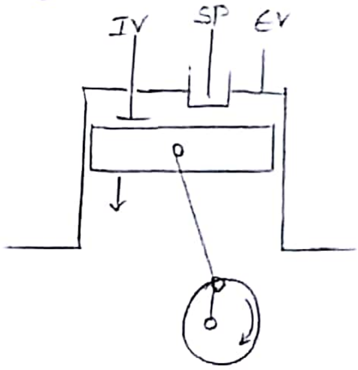
$= V_s + V_c$.

Stroke:

It is the distance travelled by the piston when moving from BDC to TDC position & vice versa.

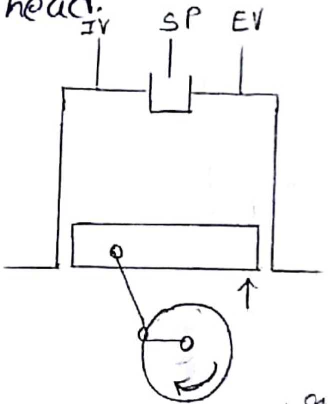
Suction (or) Intake stroke:-

The suction stroke starts when Piston is at TDC and move towards BDC. The inlet valve is opened and exhaust valve is closed. The fuel air-mixture is drawn by the suction created in the cylinder by the downward motion of piston. The suction stroke ends when Piston reaches BDC and inlet valves is closed. In P-V diagram, 0→1 represents the suction stroke.



Compression stroke:-

It starts when the Piston moves upward towards TDC by the return stroke compressing the charge in the cylinder. Both inlet and exhaust valves are closed in this stroke. The entire charge is compressed into the clearance volume by the Piston. At the end of this stroke, the charge is ignited with the help of spark plug placed on the cylinder head.



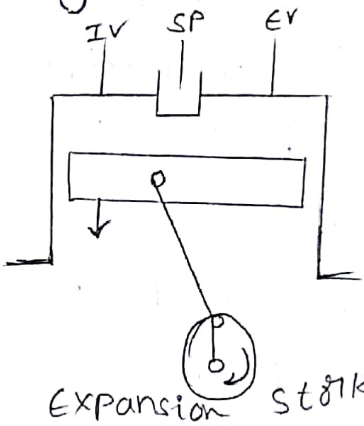
compression stroke

Expression & Power stroke:-

In this stroke, the Piston moves towards the BDC due to ~~high pressure of the burnt gas. Both inlet and exhaust valves are closed~~

Stroke. This is the only stroke in which power is produced

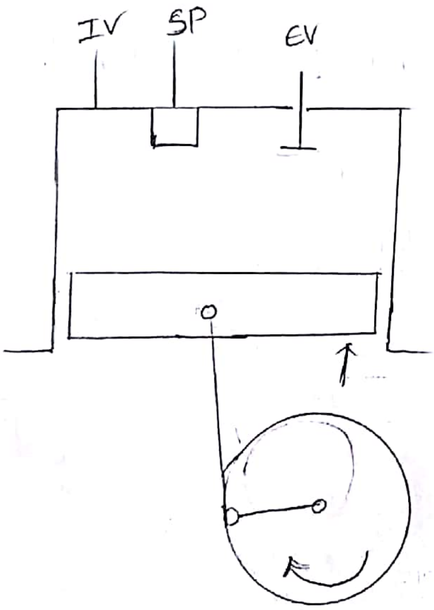
among the four strokes. Both pressure and temperature increase during power stroke. In Ideal P-V diagram, the process 3→4 represents power stroke.



Expansion stroke

Exhaust stroke:

After expansion stroke, the inlet is remained closed and exhaust valve is opened due to which pressure of burnt gases reduces to atmospheric level. As the piston moves towards TDC, it sweeps the remaining burnt gases are forced out of the cylinder. Exhaust valve is closed as the piston reaches TDC. Some residual gases by which may get trapped in the cylinder at the end of this stroke. In Ideal P-V diagram, the process 5→0 represents the exhaust ~~valu~~ stroke.



IV = inlet value

EV = exhaust value

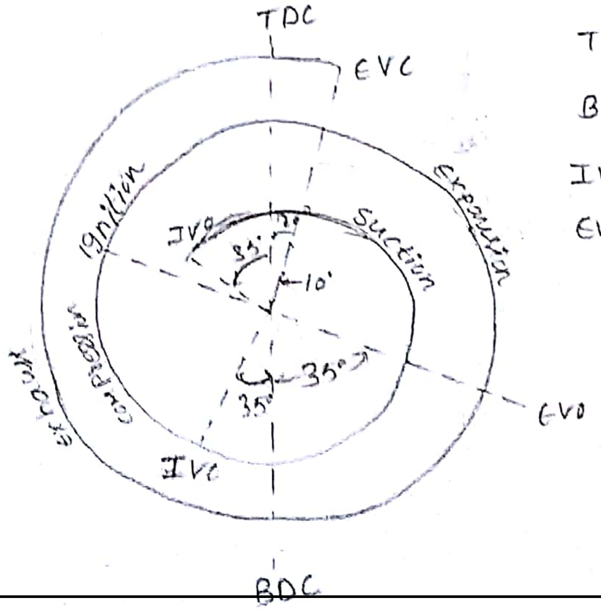
SP = spark plug

Exhaust stroke

Comparison of External and Internal Combustion engines:-

External Combustion	Internal Combustion
1) In external Combustion engine the Combustion of fuel is external.	1) In Internal combustion engines, the Combustion of fuel is internal.
2) working fluid is steam.	2) working of fluid is Products of Combustion of air, fuel mixture.
3) These engines are operated at low temperature (about 600°C)	3) IC engines are operated at high temperatures (about 2300°C)
4) It doesnot require cooling system	4) It require Cooling system.
5) Starting of engine is not easy	5) starting of engine is easy
6) It requires more space	6) It requires less space.
7) Its design is Complex	7) Its design is compact and simple.
8) These are self-starting engines	8) These are not self-stating engines.
9) Efficiency is low.	9) Efficiency is high.
10) Ex:- steam engine, steam turbine	10) Ex:- petrol engine, diesel engine.

Valve timing diagram:-



TDC - Top Dead centre

BDC - Bottom dead centre

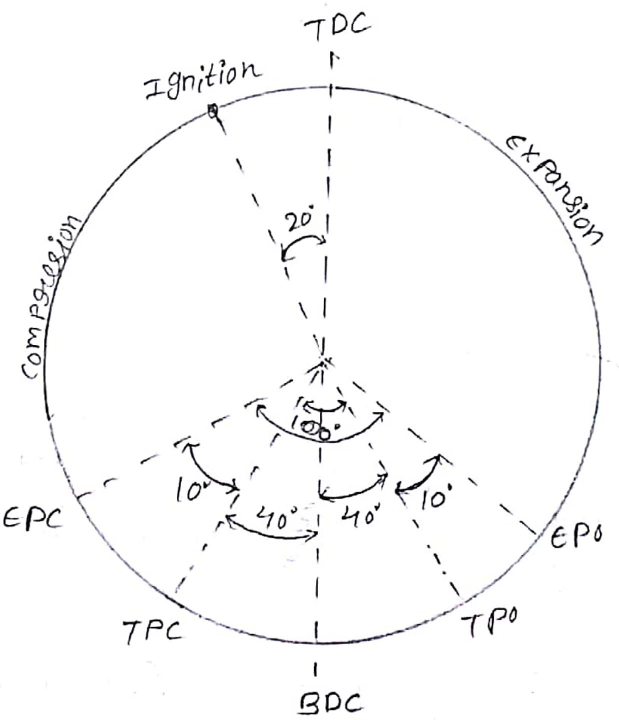
IVO - Inlet valve opens

EVO - exhaust valve opens

EVC - Exhaust valve opens

Practice it is not possible to open & close the valves insta-ntaneously. Therefore in order to improve the performance of an engine, the valve timing diagrams are modified as shown in fig. from fig it is seen that the inlet valve opens 10°-30° before the Piston reaches TDC Position. This enables the fresh charge to enter into the cylinder and helps in escaping the burnt gases to the atmosphere. The intake & suction process continues until the Piston reaches 35° after BDC position. At this point, the inlet valve closes and the compression of the charge proceeds. The ignition of the charge by producing a spark from the spark plug is carried out at 35° before TDC position of the piston. This helps in effective burning of the charge (air-fuel mixture). The exhaust process starts when the piston reaches 35° before BDC position and closes at 10° after TDC position of the piston.

Port timing diagram:

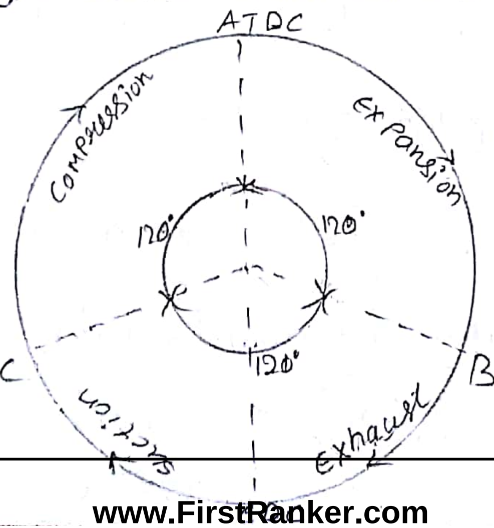


TDC - Top dead center
BDC - bottom dead center
TPO - transfer port opens
TPC - transfer port closes
EPO - Exhaust port opens
EPC - exhaust port closes

The Port timing diagram of a two stroke S.I engine is shown in fig. The exhaust Port opens 50° before the bottom dead centre Position of the Piston. After another 10° of crank revolution, transfer Port opens i.e at 40° before TDC position ~~first average~~ of the Piston. The entering fresh air-fuel mixture helps in removal of burnt gases. Further movement of the Piston first covers the transfer Port at 40° after BDC position of the Piston, then the exhaust Port at 50° after BDC Position of the Piston. The charge is then compres-sed for the upward movement of the Piston. The ignition of charge occurs 20° before TDC Position of the Piston.

Ideal Port timing diagram of a 2-stroke S.I engine:

In Ideal Port timing diagram, the cycle starts from Point A where the combustion of fuel takes Place. During combu-stion process, expansion of glases takes Place and continue till the crank shaft reaches Point B. At approximately 180° crank angle from Point A, the Piston from top dead centre reaches to bottom dead centre. The crank shaft moves from Point B towards Point C, compression of air fuel mixture takes place and continue till ~~at~~ it reaches point A. At Point A, the Piston reaches to top end centre from bottom dead centre and the cycle is repeated



A cycle is defined as a process in which repeated series of operations occurs in an order such that intial and final conditions are same. If the cycle is of imaginary perfect engine, it is called as an actual cycle. Air or mixture of air-fuel is used as a working substance in IC engines. If air alone works as a working fluid in the IC engines, then the cycle of operation is called as "Air standard cycle".

Assumptions:-

1) The working medium used in the engine cylinder is air.
2) Air behaves like an ideal gas and obeys the gas laws and has constant specific heats at all temperatures
3) The walls of cylinder are good insulators of heat.
4) The cycle is considered to be closed and same air is used to repeat the cycle.
5) There is no suction and exhaust strokes in the cycle.

Otto-cycle (or) Constant volume cycle:-

This cycle was introduced by Otto and hence this cycle is called as "Otto cycle". Otto cycle is the theoretical cycle for petrol engine. The original cycle was designed by Alphose Bean, but Otto was the first man to build an engine on this cycle. The cycle is represented on P-V and T-s diagrams as own in fig. It consists of two adiabatic and two constant volume process and all of these processes are assumed to be reversible.

Process 1-2:-

It is a reversible adiabatic compression process during which pressure and temperature increases from P_1, T_1 to $P_2 T_2$ respectively. But, the volume decreases from V_1 to V_2.

Process 2-3:-

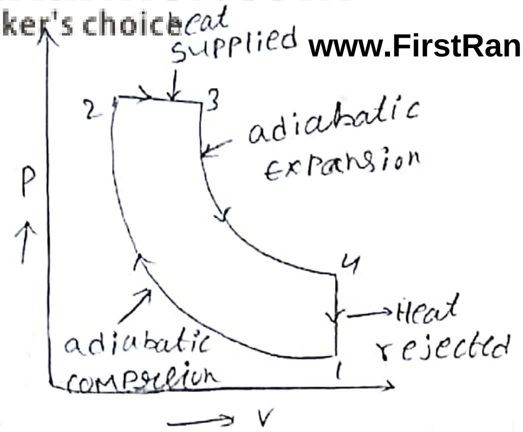
During this process, heat is added at constant volume. Such that the pressure and temperature increases from $P_2 T_2$ to P_3, T_3 respectively.

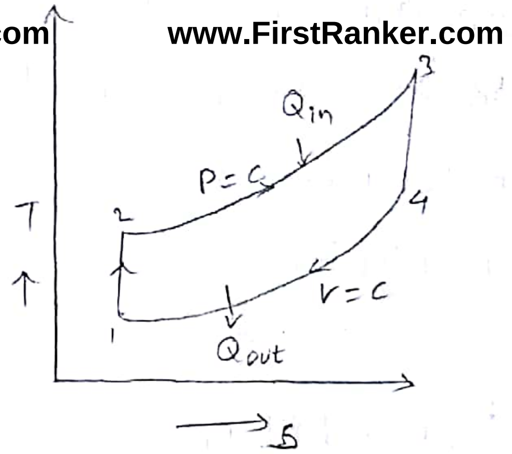
Process 3-4:-

The hot gases after combustion expands adiabatically in the engine cylinder. During this expansion stroke, the gases drives the piston downwards i.e., from TDC to BDC producing work. Due to this both the pressure and temperature decreases from P_3, T_3 to P_4, T_4 respectively and volume increases from V_3 to V_4.

Diesel Cycle & Constant Pressure Cycle:-

This cycle was introduced by Rudolph Diesel in 1897. It is generally suitable for low speed diesel engines. It differs from Otto cycle in the sense that in Otto cycle heat is supplied at constant volume but in Diesel cycle, heat is supplied at constant pressure. The cycle is represented on PV and TS diagrams as shown in figure.





Process 1-2:-

It is a reversible adiabatic compression process during which the pressure and temperature of air increases from P_1T_1 to P_2T_2 respectively.

Process 2-3:-

During this process, heat is added at constant pressure by a suitable heating source that volume and temperature increases from V_2T_2 to V_3T_3 respectively.

Process 3-4:-

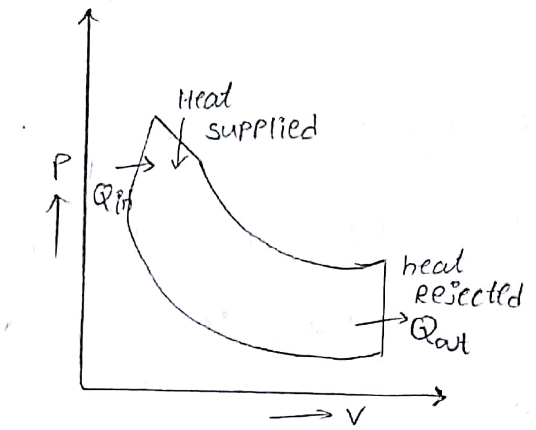
It is a reversible adibatic expansion process during which the pressure and temperature reduces from P_3T_3 to P_4T_4 respectively.

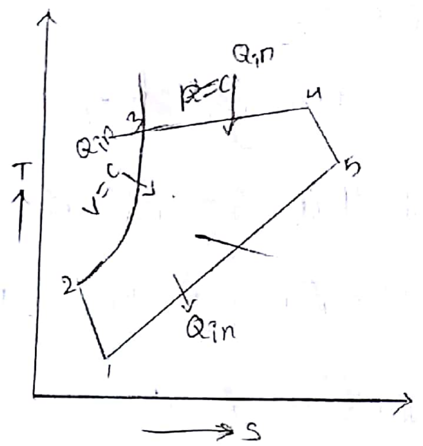
Process 4-1:-

During this process, heat is rejected at constant volume such that pressure and temperature reduces from P_4T_4 to P_1T_1 respectively. This process completes the cycle and returns air to its original state.

Dual Combustion Cycle or limited Pressure Cycle:-

This cycle is a combination of constant volume and constant pressure cycles. Thus, it is called as mixed cycle. Here heat is added partly at constant volume and partly at constant pressure. The main advantage of this cycle is that sufficient amount of time will be available for the combustion of fuel. It is suitable for high speed diesel engi-nes.





Process 1-2:-

It is a reversible adiabatic compression process during which the pressure and temperature of air increases from P_1T_1 to P_2T_2 respectively.

Process 2-3:-

During this process, heat is added to the compressed air at constant volume by a suitable heating source such that the pressure and temperature increases from P_2T_2 to P_3T_3 respectively.

Process 3-4:

During this process, heat is added to the compressed pressure by a suitable heating source such that the pressure and temperature increases from P_2T_2 to P_3T_3 respectively.

Process 4-5:

It is a reversible adibatic expansion process during which the pressure and temperature decreases from P_4T_4 to P_5T_5 respectively.

Process 5-1:

During this process, heat is rejected at constant volume such that the pressure and temperature decreases from P_5T_5 to P_1T_1 respectively. This process complete the cycle and returns the air to its orginal state.

Air-Injection System:

In this method, a compressor is used in which air is compressed to a very high pressure and it is injected into the enginee cylinder through the fuel nozzle. The fuel admiss-ion rate is controlled by varying the pressure of injection air. High pressure air is supplied by storage air-bottles which are charged by air compressor.

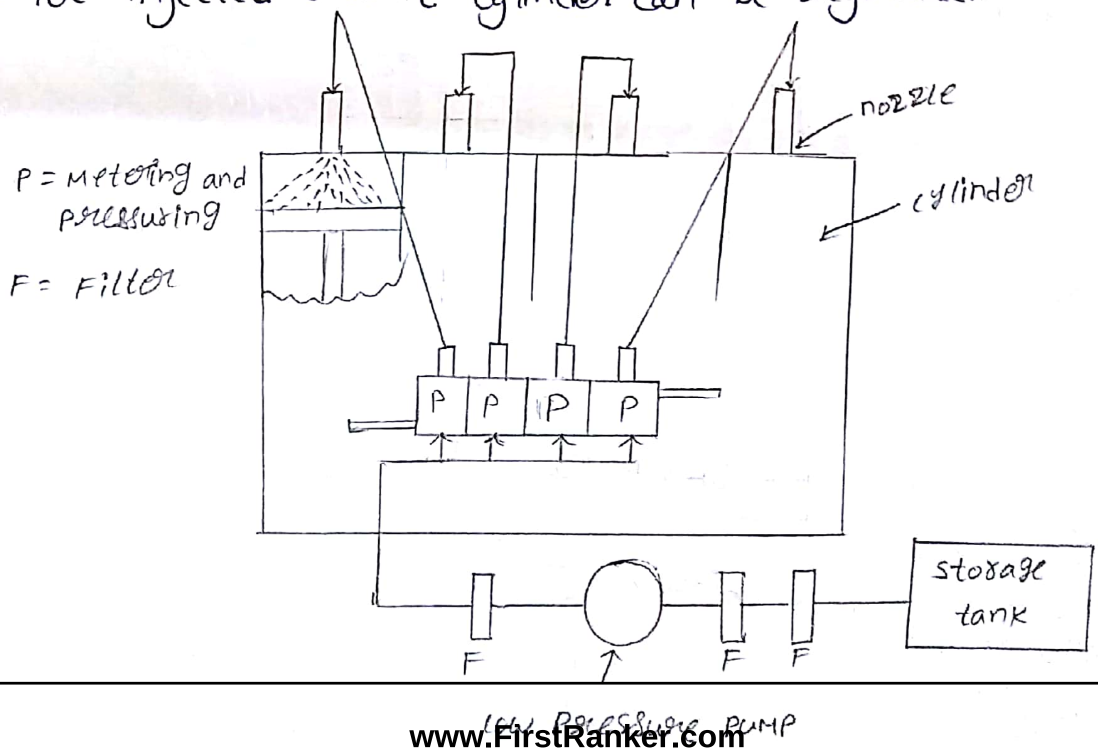
Solid & Airless Injection System:-

Solid injection is also called as mechanical Injection. The fuel is directly injected into the combustion chamber without any primary atomization by a fuel pump at a very high velocity. The injection pressure varies from 1000 to 145 bar. The different types of solid injection system are

1) Individual Pump system.
2) Common Rail system.
3) Distributor system.

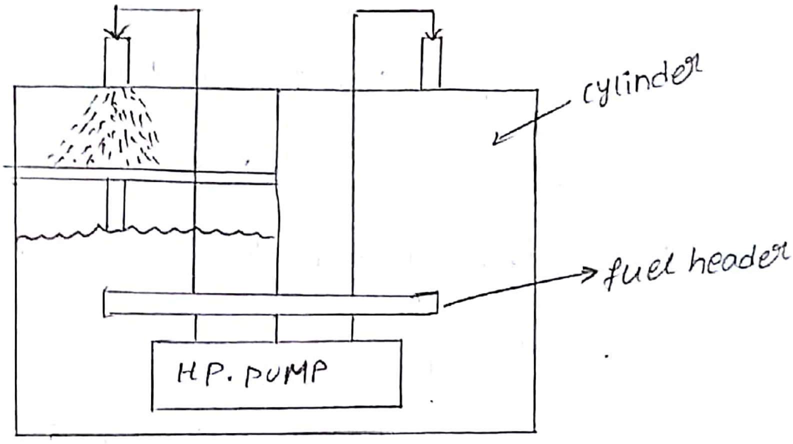
1) Individual Pump System:-

It consists high pressure pumps for each cylinder for metering, controlling the time of injection and to pump the fuel at required pressure to the cylinder. One injector is provided to each of the cylinders to inject the fuel in an atomized form. The injector and the pump may be integrated as one unit. By properly maintaining the stroke of the plunger, the amount of fuel injected into the cylinder can be regulated.



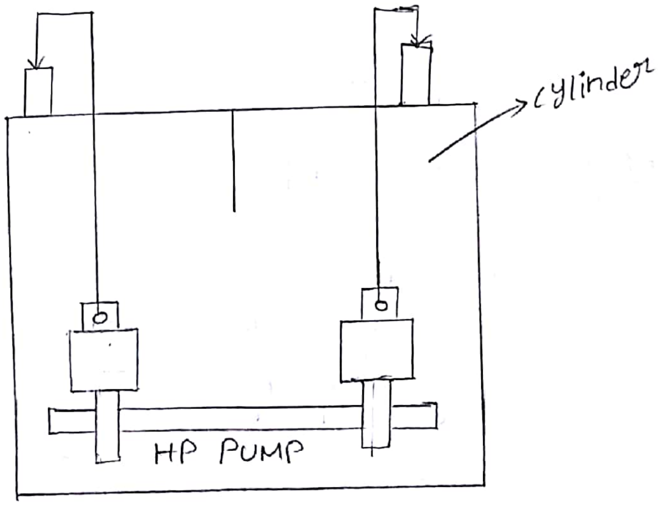
Common Rail System:

It Consists high Pressure Pump which supplies fuel to a Common rail or header at high Pressures. Due to this high pressure, the fuel is forced into the engine Cylinders. The fuel entering into the Cylinder is meterd and is Supplied by a fuel injection value with the help of Cam mechanism.



3) Distributor System:-

In this system, fuel pressure is increased by pump as well as meters and times the injection. The Pump after metering the required amount of fuel, supplies it to the distributor ~~sy~~ at Correct time. The Cost of the system is less nearly 1/3rd of that individual pump system.



Distributor system

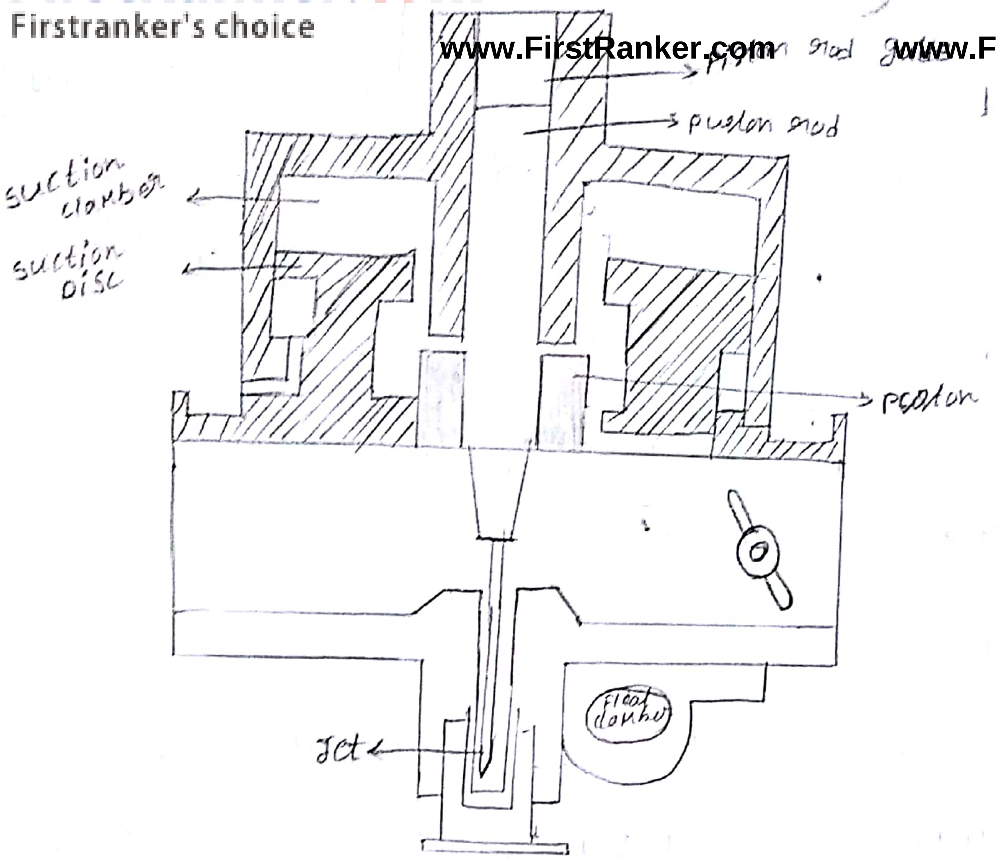
Carburetor:

A Carburettor is a device which is used to mix air and fuel outside the engine cylinder. It is most commonly used in SI enginess. The different types of carburettors are:

1) Carter Carburettor
2) Solex Carburettor
3) Stromberg Carburettor
4) S.U Carburettor
5) Zenith Carburettor.

S.U Carburettor:-

S.U Carburettor is a type of automatic variable choke. It is also called as constant vaccum or depression type of carburettor. It consists of piston, piston rod, tapered needle throttle value, jet etc.. lower end of piston rod is connected to the tapered jet needle which is inserted into the main jet. The piston moves up and down through the piston rod guide. The upper end of the piston is termed as suction disc an, the area above this disc is called as suction chamber. When the piston moves due to the force created by vaccom inside the suction chamber the tapered needle along with the jet moves up. The weight the piston acts vertically downwards, due to gravitational force. These two forces are constant at particular instant. Hence these are balanced forces. During the start of the engine of adequate amount of air-fuel mixture is required for the combustion.



Requirements of Ignition system in I.C engines:-

1.) The Spark Produced at the spark Plug electrode must be sufficiently strong as to start the ignition of charge present in engine cylinder with less delay.

2.) The spark must be regular and synchronously timed all the engine speeds and loads.

3.) The maintanenace of the system should be easy.

4.) The weight of the ignition system should be less.

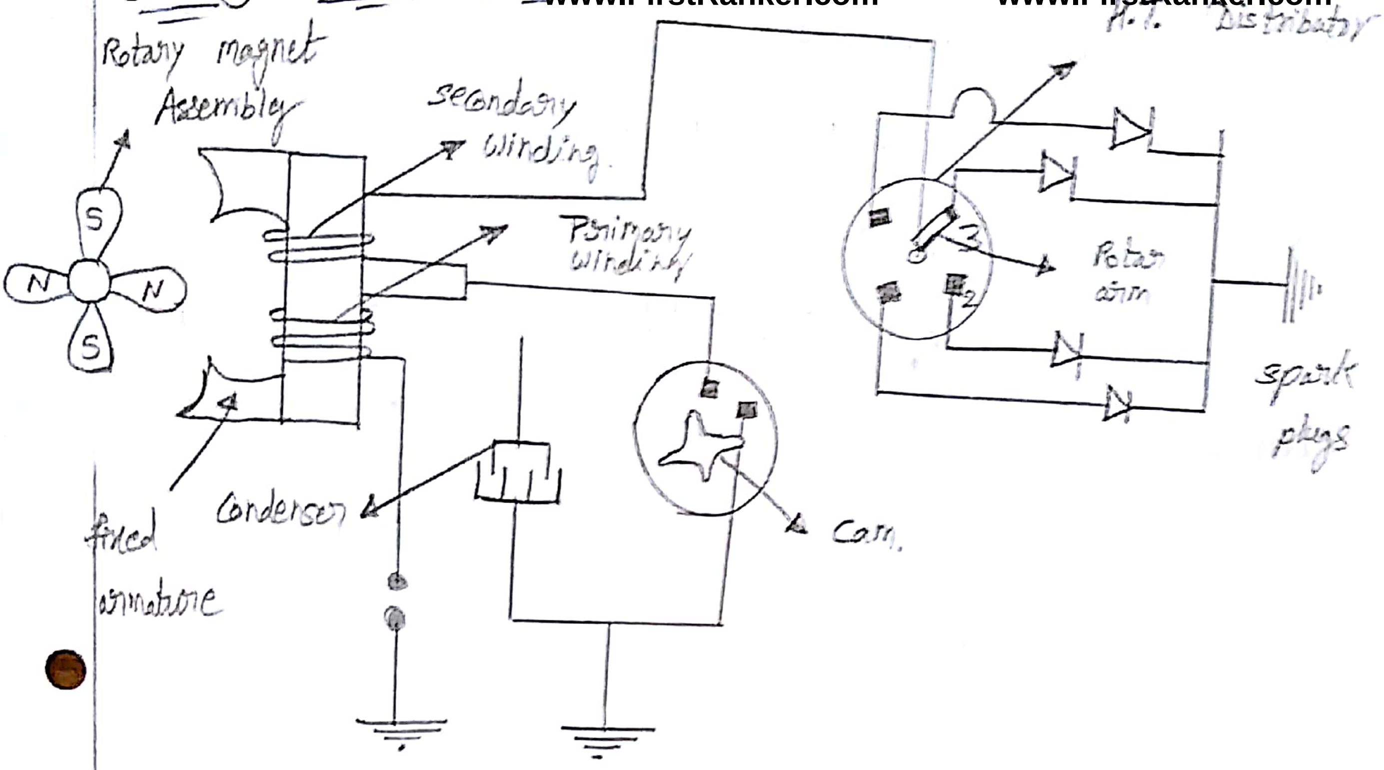
5.) It should occupy less space.

6.) longer life of breaking points and spark along.

7.) The service life of the system should be high.

8.) The performance of the system should be good even ... and high speeds.

Battery Ignition System operation:-



Battery Ignition System is used mostly in spark Ignition engines When the Current flows in a Primary Circuit, the Ignition switch and the breaker Points are in the Closed state. The field thus Produced exp-ands through and around the soft iron Core of the Coil. The induce-d e.m.f opposes the battery Current. This in turn, delays the build-ng Process of the field itself. The expanding magnetic field also Cuts the Secondary winding such, a voltage is induced in the secondary winding.

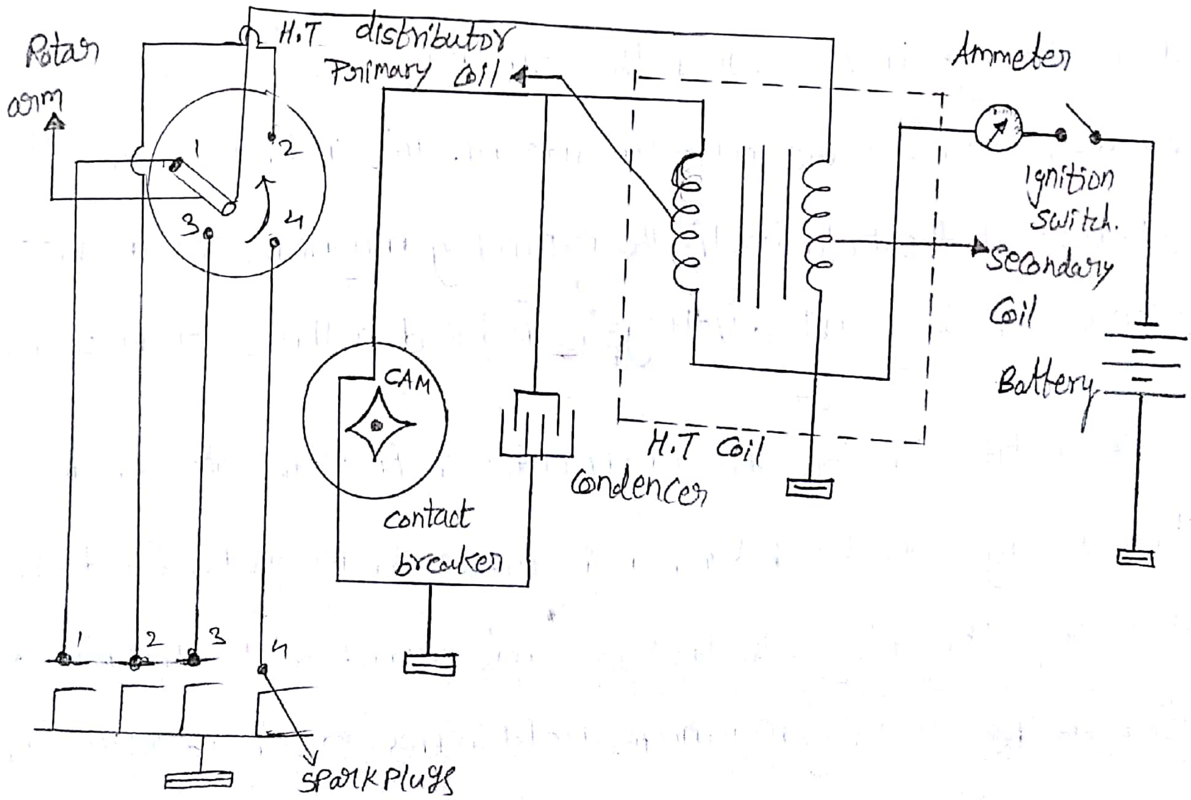
When the Cam on the distributor shaft opens the breaker Points, the Primary Circuit is broken and Causes the magnetic field to start Collapsing. The magnetic lines are now attracted to the iron Core of the Coil. The Collapsing magnetic field induces a Current in the Primary Circuit in the same direction as that of the Primary Current.

The current induced in the primary circuit charges the condenser which is connected across the contact breaker points. The charging of the condenser greatly reduces the occurence of a spark between the breaker points.

The collapse magnetic field induces a voltage in the primary winding. The magnitude of this induced voltage is decided by the number of turns of the primary winding and the rate at which the flux field collapses.

High tension Magnet Ignition System:-

This system is based on the principle of mutual electromagnetic induction. Magneto ignition system for a four-cylinder engine consists of a magneto, switch, ignition coil, condenser, contact breaker, distrubutor and spark plug.



The Primary ignition circuit starts at the magneto and Passes through the switch, Primary winding, Contact breaker Points to the ground. A Condenser is also Connected in Parall-el to the Contact breaker points. One end of the Condenser is grounded and the other end is Connected to Contact breaker.

The magneto Consisting a fixed armature wound on its primary and secondary winding.

When the magnet rotates inside the fixed armature current flows in the Primary winding. When the Primary voltage is interrupted by Contact break Points, high voltage is induced in the secondary windings and from the distribution.

In magneto system, the Permanent magnets Provide the magnetic field whereas in conventional type generator, magnetic field is produced by Passing the current through the field winding which develops the magnetic field.

Advantages of magneto Ignition system over Ignition syste-m:-

1) Magneto Ignition system Consists of an electric generator through which current is produced.

2) less maintainance is required due to battery when compared to coil ignition system.

3) Space Consumed by this type is lesser compared to coil type.

4) writing is simple compared to coil ~~gg~~ type system.

Dis-advantages of Magneto Ignition system over Coil System:-

1) Intensity of spark at low speed is poor when compared to coil ignition system.

2) ~~Intensity~~ This system is more costlier than coil type system.

Liquid-Cooled System:-

In these system water is circulated through the jackets provided around the cylinder head, cylinder valve ports and seats. The heat is transferred by conduction and convection from cylinder walls and other parts to the jackets.

Air-Cooled System:-

In this system, the current of air is flowed over the cylinder barrel and outer surface which carries away the heat by convection. Cooling fins are also provided to increase the rate of cooling.

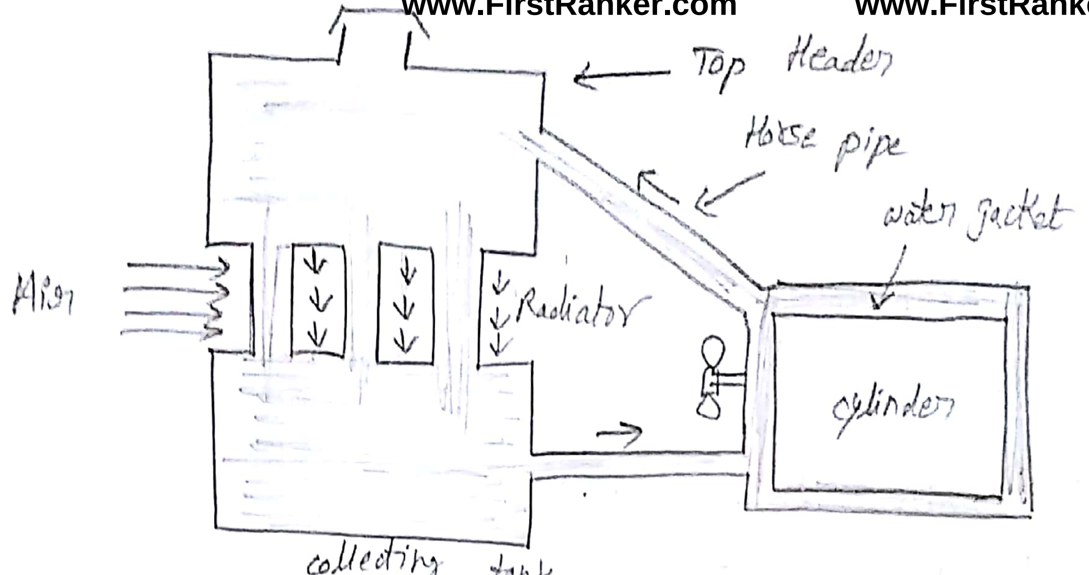
Liquid Cooled System	Air-Cooled System
1) Engine design is compact with less frontal area.	1) Engine design is simple due to absence of water jackets.
2) There is danger of coolant leakage in these systems	2) No danger of coolant leakage occurs in these system.
3) Installation is difficult.	3) Installation is easier.
4) Cost is high	4) Cost is less

Methods used for water Circulation Around Cylinder and Cylinder Head.

1) Thermo Syphon Cooling system.
2) Forced or Pump Cooling System.
3) Cooling with thermostatic regulator.
4) Pressurised water Cooling.
5) Evaporative Cooling.

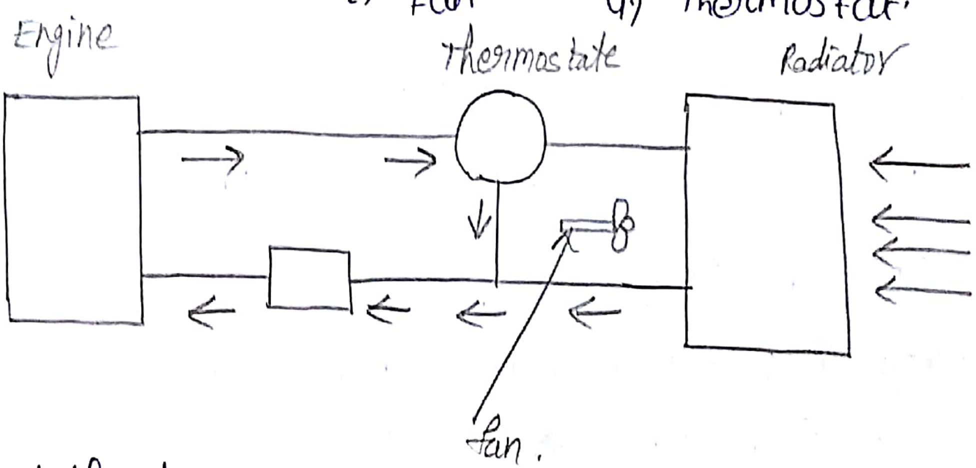
Thermosyphon Cooling system:-

It Consists radiator connected to the engine by means of hose connections. In this system, circulation of water is carried out due to the difference in densities of hot and cold water. During the operation of an engine, the water near the cylinder gets heated by taking the heat of engine, thereby cooling the same. Due to this heat, the density of water decreases and it moves upwards. The cold water from the bottom of the radiator comes into replace it. The hot water then enters into the radiator, where it dissipates the heat to the atmosphere. The water circulation rate is directly proportional to the heat output from the engine and not to the speed. The fan used is driven from the crankshaft by means and pulley drives.



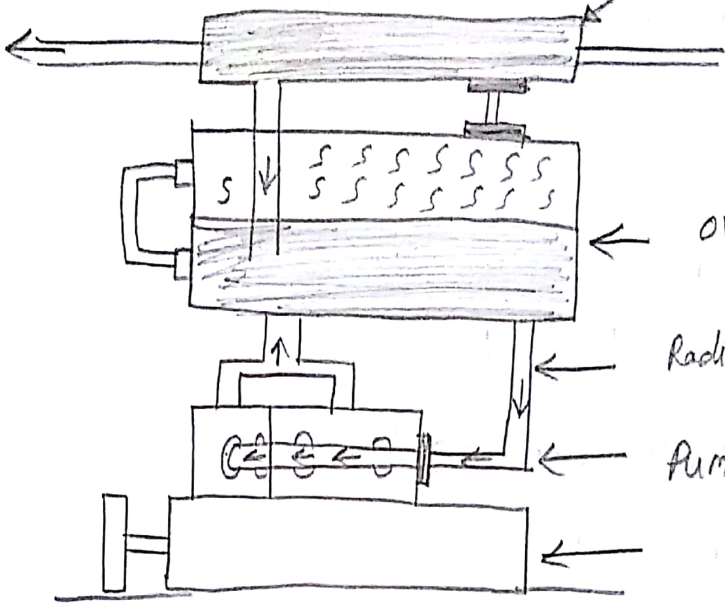
Force Circulation Cooling System:-

Forced circulation cooling system is mainly used for cooling in heavy trucks, buses etc. The main components of this system are 1) Radiator 2) Fan 3) water pump 4) Thermostat.



The centrifugal pump is driven by engine keeps the coolant in motion which is circulated to various parts of engine through the jackets. The radiator cools the coolant passing through it by the air drawn through it by a fan and the air draft motion by the forward motion of vehicle. The temperature required for cooling is maintained by using a thermostat.

Evaporative Cooling System:



The evaporative cooling system mainly used in stationary engines. The engine is cooled by the evaporation of water in the jackets into steam. The high latent heat of vaporizatio-n of water is used as an advantage to evaporate it from cylinder jackets.

Pressure Cooling System:

Pressure cooling system employ moderate pressure upto 2 bar. This system consists of a cap fitted with two safety valves loaded by a compression spring and a vaccum valve. Both the values are shunt when the coolant is cold and when engine warms up, the coolant temperature rises until the preset value is reached corresponding to desired pressure. The safety valve opens when the coolant temperature rises and the safety valve is coled back, when the engine is switched off, vaccum is formed in the cooling system and if pressure

-us, below atmospheric, the vaccum valve opens and the Cooling system attains Atmospheric Pressure.

Various Components to be lubricated in an engine:-

1) Piston and Cylinder walls.
2) Main crank shaft bearings.
3) Cam shaft and camshaft bearings.
4) Big-end bearings.
5) Crank pin and their bearings.
6) small end bearings.
7) Wrist pin and their bearings.
8) valve mechanism.
9) Timing areas.
10) Valve guides and valve tappets.

Mist lubrication System:-

The two stroke engines mainly use this system to lubricat-e all the parts of the engine.

Wet Sump lubrication System:-

This system is further Classified as

(a) splash system

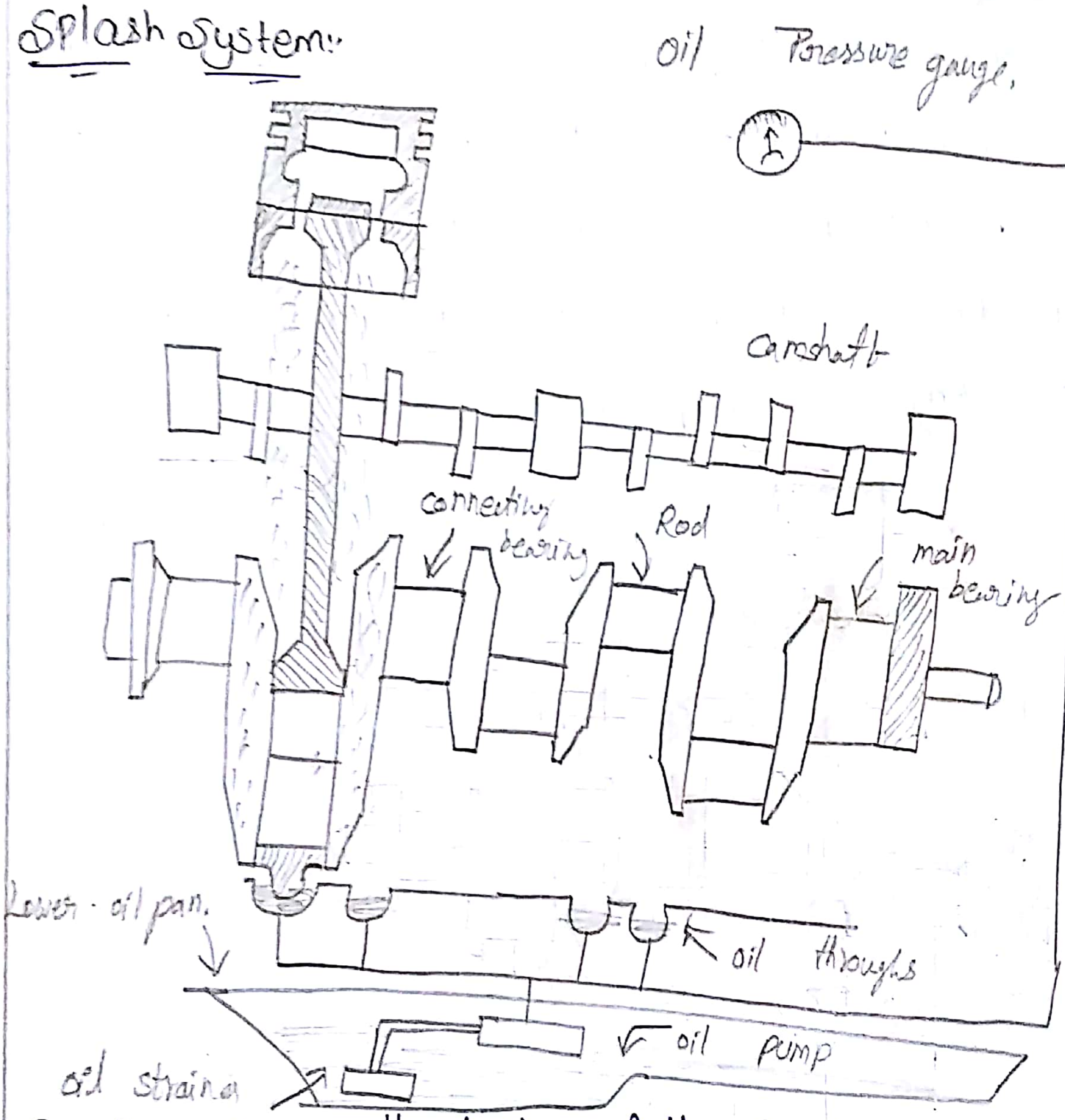
(b) splash and pressure system.

(c) splash and pressure lubrication system.

Dry Sump lubrication System:-

The various bearings of engine and cylinders are lubricated by this system. low and medium speed Engine pistons are lubricated by positive feed mechanical oilers.

Splash System:-

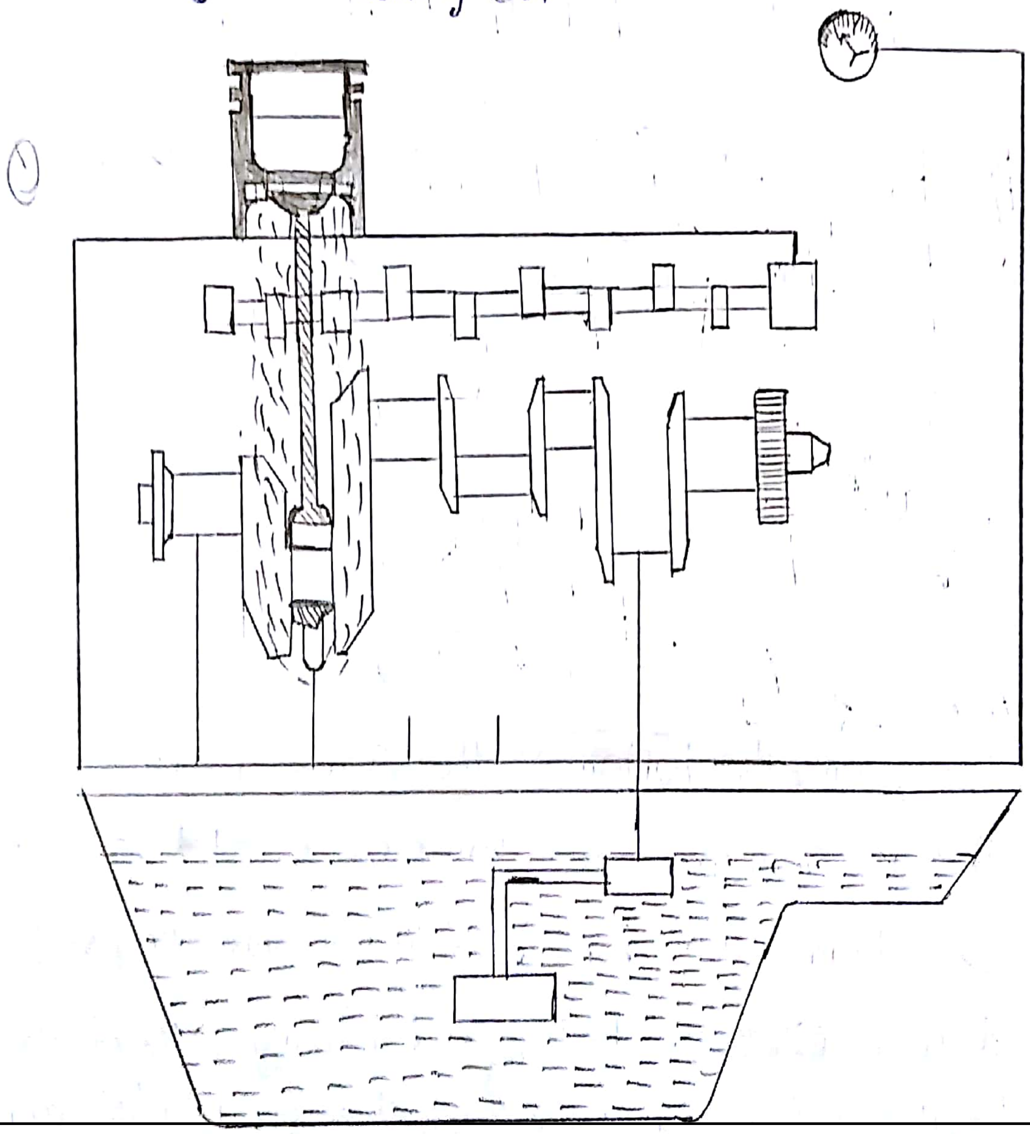


In this system, the bottom of the crank case is provided with the oil from where it is pump to the oil through provi-ded just below the crank shaft. The connecting rod is provided with scoops on the big end bearing caps. At the lowest position of the connecting rod, the scoops dips into the through and the oil splashes on to the lower portion of

the cylinder walls, crankshaft and other parts requiring lubrication.

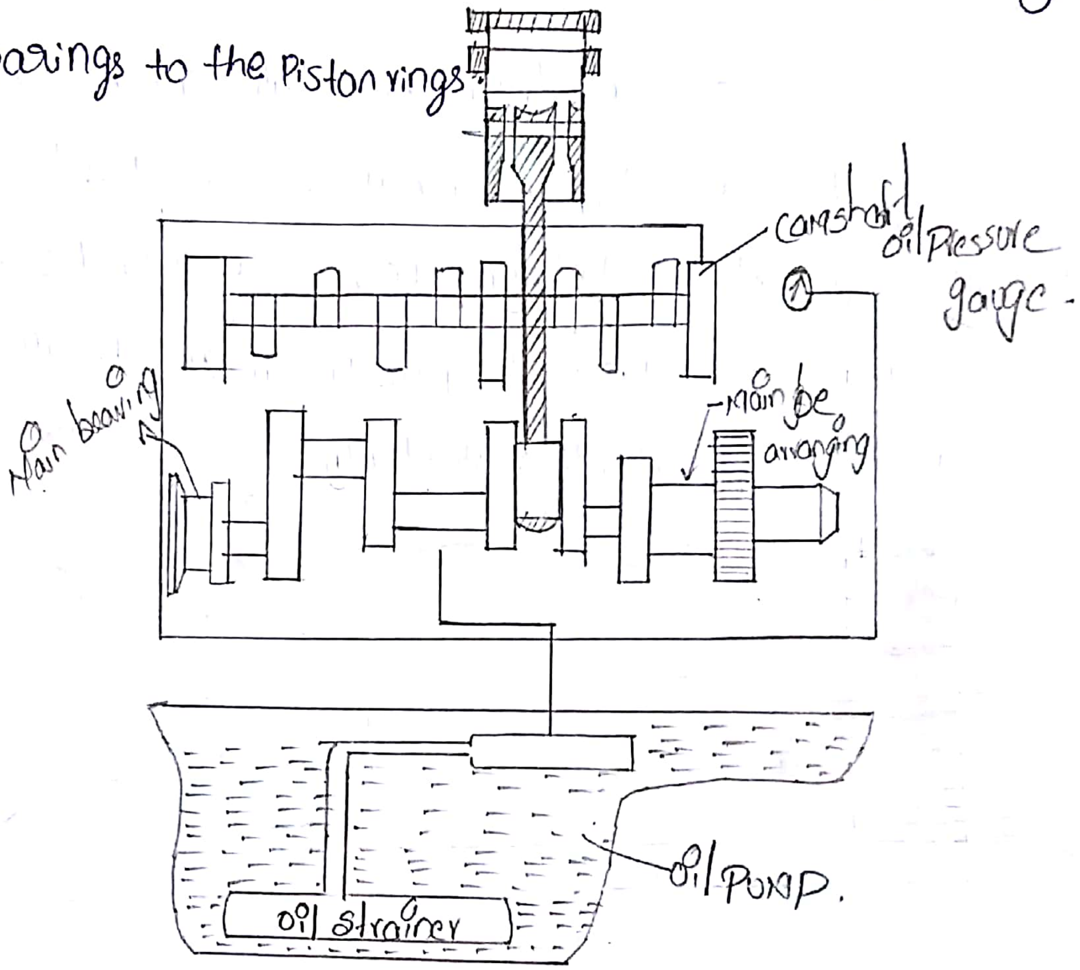
Splash and Pressure system:

The lubrication is carried out this system by splash and pressure feed systems. The oil from the sump is supplied by pump through filter to the main and camshaft bearings. The pump also supply oil under pressure to the pipes which direct a stream of oil against the dippers on the connecting rod bearing cup.



Full Pressure System:-

In this oil is pump under pressure to the crank shaft. The holes drilled the crankshaft works as a passage for the oil to flow to the various bearings, there by lubricating the same. A hole is drilled in the connecting rod which allows the oil to flow from the connecting rod bearings to the piston rings.



Measurement of Indicated Power:-

The engineer has to designe in such a way that the intial capital investement as well as the running cost should be reduced. But the parameters are so large and different in nature and it is impossible to consider during designining engine

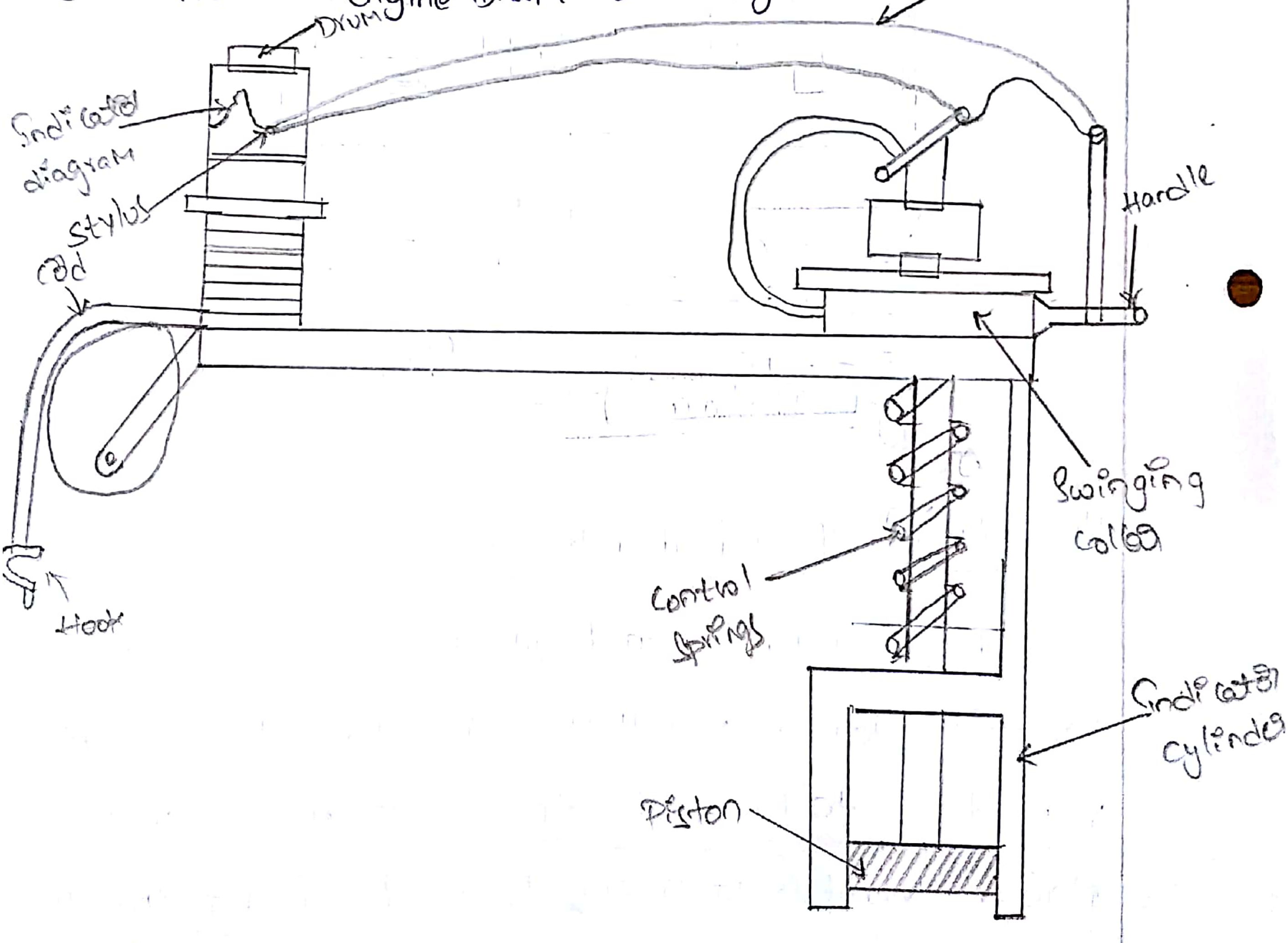
The testing of an engine is necessary to verify the desi performance according to the manufactures specifications.

The indicated power can be determined by using the formula.

$$I.P = \frac{P_m L A n}{60} \quad \text{where, } P_m = \frac{A_p}{L} \cdot S_p - \frac{A_n}{L} \cdot S_n$$

Mechanical type Indicator:

The paper on which the P-V diagram is drawn is wrapped on the drum. The stylus which is used for drawing the indicator (P-V diagram) is connected to one end of the straight line linkage. The vertical and the horizo-ntal movements are operated by the piston and the rotating drum from the engine shaft respectively.



Farnborough balanced Engine Indicator:-

It is generally used for high speed engines where the mechanical indicator is not suitable due to inertia of the parts such as links, piston and springs. It consists of a drum driven by the engine crankshaft with the help of pull-eys mounted on them. It has a cylinder and piston mechanism in which piston operates against a spring. A gas cylinder containi-ng nitrogen or other gas is connected to this cylinder by pipes. For taking an indicator diagram, the drum is rotated and spark is put ON and the engine pressure increases from maximum to maximum to minimum by operating the gas cylinder valve.

Cathode Ray oscillograph engine Indicator:

It is a high speed engine Indicator working on the principle that by varying the electric current, the cathode ray can be deflected. This deflection of the ray is caused by

(a) electromagnet.

(b) electrostatic effect.

Parameters of performance:-

1) Power and mechanical efficiency:

a) Brake power:-

The brake power is the power available by an engine ~~at output shaft and is measured with the help of dynamometers~~

brake power is given by

$$B.P = \frac{2\pi N T}{60,000} \text{ kW}$$

where
N = Speed in r.p.m
T = Torque in N-m.

(b) Mechanical efficiency:-

Mechanical efficiency is defined as the ratio of brake power to the Indicated power.

$$\eta_m = \frac{B.P}{I.P}$$

(c) Frictional power:-

The difference between I.P and B.P is called Frictional power F.P, $F.P = I.P - B.P$

2.) Mean effective Pressure:-

Mean effective pressure is defined as the pressure acting on the piston throughout the expansion stroke.

$$P_m = \frac{\text{Area of Indicator diagram} \times \text{Spring Constant}}{\text{length of Indicator diagram}}$$

$$= \frac{a_i}{l_i} \times C_s$$

3) Specific fuel Consumption:-

It is defined as the mass fuel consumed per kW developed per hour.

4) Thermal efficiency:-

It is defined as the ratio of the power output to the heat input. It is denoted by η_{th}.

$$\eta_{th} = \frac{B.P}{\text{Heat Supplied}}$$

5) Volumetric efficiency:-

It is defined as the ratio of mass of charge indicated into the engine cylinder to the mass of charge corresponding to the swept volume of the engine at intake pressure and temperature.

6) Air-fuel ratio:-

Air-fuel ratio is defined as the ratio of mass of air to the mass fluid supplied to the engine.

7) Exhaust Emission:-

The smoke and other emissions such as NO_x unburned hydrocarbons etc... are harmful to the human life and are to be minimized to control pollution.

8) Specific weight:-

It is defined as the weight of the engine per kW brake power developed. It is the indication of engine bulk and should be kept as minimum as possible in aircraft applications.

Principle for measurement of Brake Power:-

The measurement of brake power involves the determination of the torque and the angular speed of the engine output shaft. A dynamometer is used in the measurement of torque.

Work done per revolution is $(W) = 2\pi RF$

The external moment of torque balances the turning moment $(R \times F)$

$\therefore S \times l = R \times F$.

where S - Scale reading, l - Arm length

Substituting equation (2) in equation (1) we get

$W = 2\pi Sl$

Work done per minute $(W) = 2\pi SL \times N$

Where. N = speed of rotor in r.p.m T = Torque in r.p.m.

$\therefore$ Brake Power $(bp) = 2\pi NT$ watts

Morse test:-

In this test, by adjusting the throttle in SI engine or the pump crack in CI engine is first run at the required speed and the output is measured. The throttle rack is fixed in this position. By short-circuiting of spark plug in S.I engine or by disconnection of injector in C.I engine, one cylinder. By reducing the load, the engine speed is maintained at original value, so that frictional power is same and brake power of

Engine is k-cylinder less.

Consider, there are k-cylinders.

$$ip_1 + ip_2 + ip_3 + ip_4 + \dots + ip_k = \sum_1^k bp_k + fp_k$$

Where

ip - Indicated Power

bp - Brake Power

fp - frictional power.

When the first cylinder is cut off, it will not produce any power but has friction.

$$\therefore ip_2 + ip_3 + ip_4 + \dots ip_k = \sum_2^k bp + fp_k$$

subtracting equations (2) in equation (1) we get

$$ip_1 = \sum_1^k bp_k - \sum_2^k bp_k$$

Similarly the indicated power of remaining cylinder i.e $ip_2, ip_3, ip_4 \dots$ can be determined.

Therefore, total indicated power developed by the engine is $ip = \sum_1^k ip_k$

Therefore, the frictional power of the engine is given by

$$fp = ip_k - bp_k$$

The following data were recorded during a test on a 4-stroke cycle engine.

Area of indicator diagram = $90 cm^2$

length of indicator diagram = 7cm

Spring scale = 0.3 bar/mm

Diameter of piston = 20cm,

length of stroke = 25cm

speed = 300 r.p.m

Determine i) Indicated mean effective pressure

ii) Indicated power.

Given that,

type of cycle gas engine = 4 stroke

Area of indicator diagram, $A_i = 90 cm^2$

$= 9000 mm^2$

length of indicator diagram $l = 7cm = 70mm.$

Spring scale, $S = 0.3$ bar/mm.

Diameter of Piston, $D = 20cm = 0.2m.$

length of stroke, $L = 25cm = 0.25m.$

Speed of engine, $N = 300$ r.p.m

(i) Indicated Mean effective Pressure:-

$$P_m = \frac{A_i \times S}{l} = \frac{9000 \times 0.3}{70}$$

$P_m = 38.571$ bar.

$P_m = 38.571 \times 10^5 \, N/m^2$

ii) Indicated Power:

$$I.P = \frac{P_m \times L \times A \times N}{6000}$$

$$= \frac{(38.571 \times 10^5) \times 0.25 \times \frac{\pi}{4} \times (0.2)^2 \times \left(\frac{300}{2}\right)}{6000}$$

$$I.P = 75.73 \text{ kW}.$$

II Unit Vapour power cycles (24)

Wet steam:-

The steam consisting of water particles during its evaporation is known as wet steam. In wet steam, latent heat is not absorbed and evaporation is incomplete.

Dry steam :-

The steam which does not contain the water particles in it is known as dry steam (or) by dry saturated steam. In dry steam latent heat is absorbed completely and the gas behaves as a perfect gas.

Superheated steam :-

At constant pressure, dry steam is further heated and results in increase of temperature of steam to saturation point. This high temperature steam is known as superheated steam.

Latent heat of vapourization

It is defined as the amount of heat absorbed by a kg of water at its saturation temperature to get converted into a kg of saturated steam for a given pressure. Latent heat of vapourization is denoted by h_{fg} and its value depends on pressure.

Internal latent heat of steam

It is defined as the actual of latent heat that remains after the part of latent heat of vapourisation is utilised for performing external

amount of workdone. It is denoted by h_{fg} and mathematically.

h_{fg} - latent heat of vapourization.

$$\therefore h_{(fg)i} = h_{fg} - PV_g$$

Dryness fraction :-

It is defined as the ratio of mass of dry steam to the total mass of wet steam. It is denoted by 'x' and the mathematically relation is given as,

$$\therefore x = \frac{m_g}{m}$$

where,

m_g = mass of dry steam

m = total mass of wet steam

$m = m_g + m_f$

sensible heat :-

It is defined as the total amount of heat absorbed by a kg of water for converting to steam and raising the temperature from 0°c to the saturation point. It is denoted by 'h_f' and mathematically.

$$\therefore h_f = C_p T_{saturation}$$

h_f : ~~latent heat~~ Sensible heat.

C_p = specific heat

Steam calorimetry :-

The principle of measuring the dryness fraction of steam is termed as steam calorimetry.

Steam calorimetry is used to measure the dryness fraction of steam and it can be explained by any one of the following methods.

i, Throttling calorimeter

ii, Bucket calorimeter

iii, separate - throtting calorimeter

Internal energy of steam :-

It is defined as the actual amount of energy stored in steam above 0°c (or) freezing point of water. internal energy is calculated by subtracting the enthalpy with the external workdone during evaporation, i.e.,

Internal energy per kg of steam is calculated as follows,

Enthalpy (h) = internal Energy (U) + work done

for wet steam,

$$u = h - (100 \times P \times x \times V_g)$$

W.D = P×V

steam → wet
dry → x = 1
super

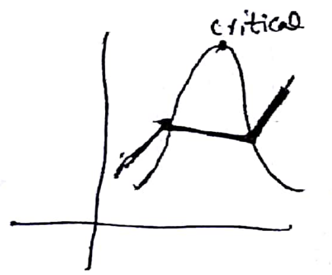
for dry steam,

$$u = h_g - (100 \times P \times V_g)$$

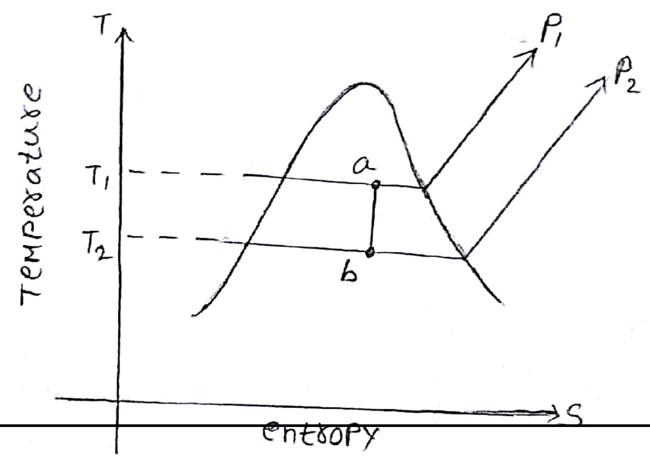
= P×x Vg → wet
P×Vg
= P Vs

For superheated steam,

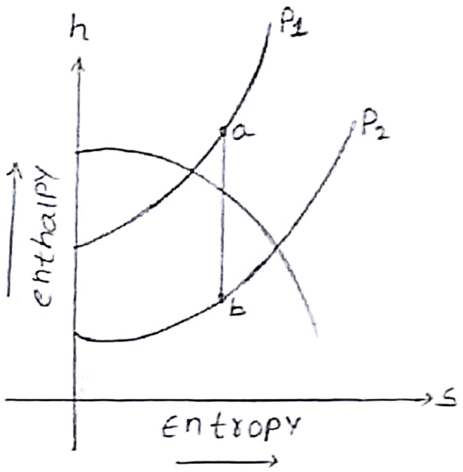
$$u = h_{sup} - (100 \times P \times V_{sup})$$



Adiabatic process for wet steam :-



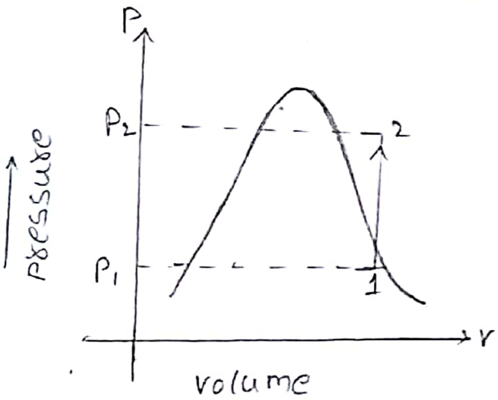
b: temperature decreases and entropy remains constant

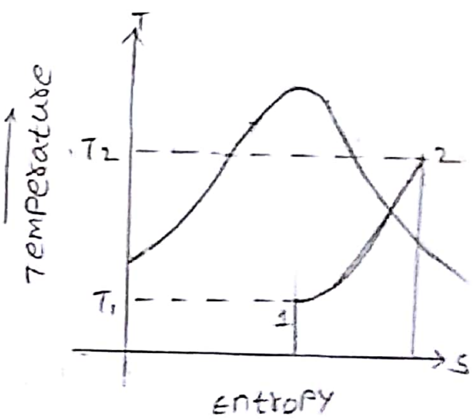


a:b enthalpy decreases and entropy remains constant

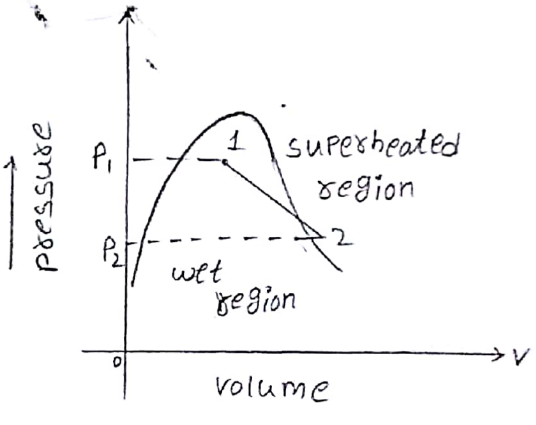
constant volume process :-

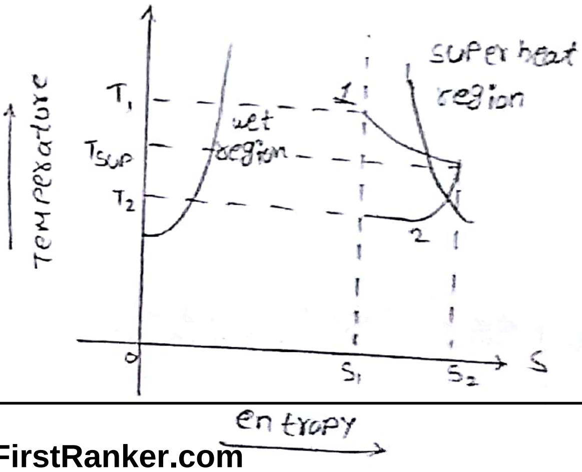
At the heat is supplied to wet steam at constant volume, the pressure and temperature will increase from P_1 to P_2 and T_1 to T_2 respectively as shown in figures 1 & 2.





formation of steam in hyperbolic process :-





Difference between carnot cycle and Rankine cycle :- 26

carnot cycle	Rankine cycle
1. In carnot cycle, the total heat addition occurs at constant temperature.	In rankine cycle, temperature increases during constant pressure heat addition
2. At a point, condensation stops because the insentropic compression brings back the fluid to its original condition	At a point, condensation completes because the pressure raises by the feed pump returns to the original condition
3. The rate of steam consumption is higher for a specific output	The rate of steam consumption is smaller for a specific output.
4. The required plant size is larger for a given work output.	The required plant size is smaller for a given work output.
5. By compressing a wet vapour, the liquid is fed to the boiler	By using a pump, a liquid is fed to the boiler.

carnot cycle is not suitable for steam power plants :-

1. It is difficult to perform isothermal heat transfer from a two phase system into practice.
2. At a specific state the quality of steam is very poor during the isentropic expansion process.

In the isentropic compression, a compressor cannot handle the two phases of the working medium and not easy to control the condensation process.

4. In power plants, the quality of the steam should not be less than 90%. In carnot cycle, steam is available with high moisture content, due to this reason the carnot cycle is not suitable.

5. A major source of wear on the turbine blades occurs liquid drop falls on it.

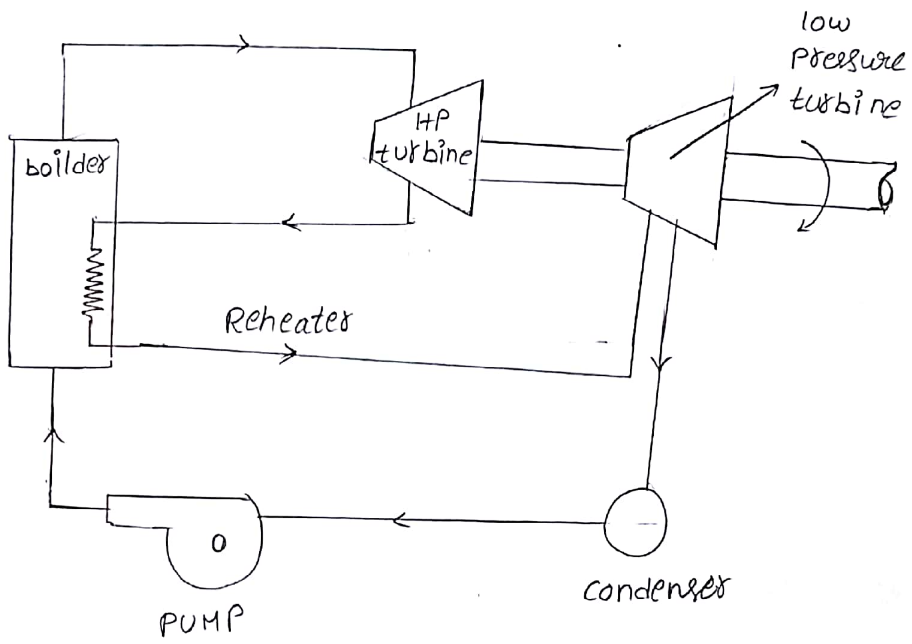
Methods of improving the thermal efficiency of a rankine cycle :-

The following are the different methods for improving the thermal efficienty of the rankine cycle.,

1. By reheating the steam
2. By regenerative feed heating
3. By increasing the maximum pressure of steam
4. By increasing the superheating temperature of steam
5. By reducing the exhaust pressure of steam
6. By water extraction
7. By using binary - vapour etc.

Reheating of steam :-

In the last stage of turbine, steam has higher velocity and it contains large moisture, this liquid particles have lesser velocity than vapour particles. Thus, the wet steam causes erosion of turbine blades and increases internal losses. Hence, it reduces the efficiency of turbine.



The following are the advantages of reheating of steam,

1. It increases the work done through the turbine and hence efficiency increases.
2. It reduces erosion of blades
3. As the specific steam consumption is reduced, the quantity of water required in the condenser is reduced.

) and H-S diagrams of Reheat cycle :-

In a reheat cycle the steam enters the turbine in a superheated state at point 1. The steam then expands isentropically while flowing through the turbine, as shown by the vertical line in figure . After expansion, the steam becomes wet, which is reheated at a constant pressure shown by the point 3, when it is again in superheated state. The steam again expands isentropically while flowing through the next stage of the turbine as shown by the vertical line 3 to 4 in H-S diagram.

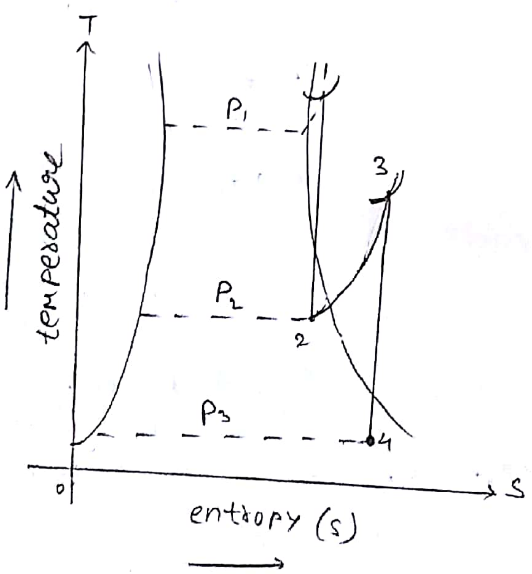
The salient points

1 - steam entering into turbine

2 - steam condition after partical expansion,

3 - steam condition after reheating

4 - steam condition after expansion in turbine.



Classification of steam turbines :-

steam turbines are classified upon the following parameters.

1. action of steam :-

Impulse turbine

Reaction turbine

combination of impulse reaction turbine

2. direction of steam flow :-

axial steam turbine - steam flows axially i.e., parallel to axis of turbine.

Radial steam turbine - steam flows perpendicularly to axis of turbine.

3. number of pressure stages :-

single stage turbine with one or more velocity stages.

Multistage impulse and reaction turbines.

4. Pressure of steam at inlet :-

low pressure turbines (1.2 - 2 atm)

medium pressure turbines (< 40 atm)

high pressure turbines (> 40 atm)

very - high pressure turbines (≥ 170 atm)

super - critical pressure turbines (≥ 225 atm)

5. number of cylinders :-

single cylinder turbines

Double cylinder turbines

Three cylinder turbines

Four cylinder turbines

6. Exhaust condition of steam :-

condensing turbines

non-condensing turbine

Axial discharge :-

If the steam leaves the blade at its exit tip at 90° to the direction of the blade motion, then the turbine is said to have an axial discharge.

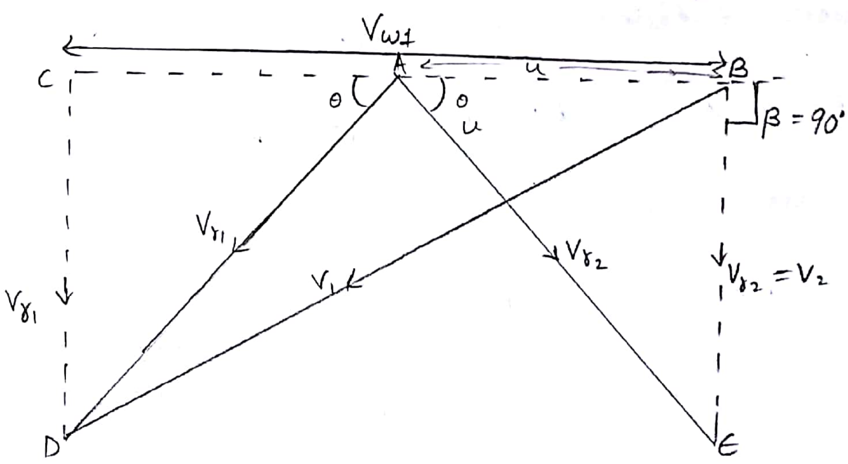
conditions for axial discharge :-

The condition for axial discharge is velocity of whirl at outlet (V_{w_2}) is equal to zero

$V_{w_2} = 0$ here, Power developed by the turbine,

$$P = m \times (V_{w_1} + V_{w_2}) \times u \text{ watts}$$

$$P = m \times V_{w_1} \times u \text{ watts} \quad (\because V_{w_2} = 0)$$



combined velocity diagram for axial discharge.

Expression for stage efficiency in case of a steam turbine :-

stage efficiency

stage efficiency is defined as the ratio of work done on the blades through the stage to the enthalpy drop in that stage.

$$\therefore \eta_{stage} = \frac{\text{work done on the blades}}{\text{enthaply drop}}$$

work done on the blades $= m_s (V_{w_1} + V_{w_2}) u$

where, m_s = mass flow rate of steam

u = blade velocity

V_{w_1}, V_{w_2} = inlet and outlet whirl velocities respectively.

$$\therefore \eta_{stage} = \frac{m_s (V_{w_1} \pm V_{w_2}) u}{m \Delta H}$$

now, multiply and divide by $2V_1^2$, we get,

$$\eta_{stage} = \frac{2(V_{w_1} \pm V_{w_2}) u}{V_1^2} \times \frac{V_1^2}{2\Delta H}$$

= blade efficiency X nozzle efficiency

$$\therefore \eta_{stage} = \eta_b \times \eta_n$$

if losses in nozzle are neglected, then

$$\eta_{stage} = \eta_b$$

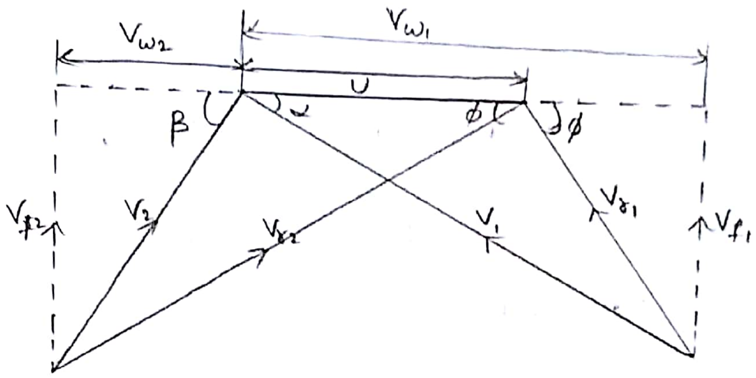
Diagram efficiency in case of a steam turbine

diagram efficiency :-

Diagram efficiency is the ratio of the work done by the blade to the kinetic energy entering the blade. it is also known as blade efficiency.

where, m = mass of steam supplied in kg/s

V = absolute velocity of inlet steam in m/s



energy supplied to the blade per second,

$$KE = \frac{1}{2} m V_1^2 \ J/s$$

The work done on the blades per second $= m(V_{w_1} + V_{w_2})U \ J/s$

$$\text{Diagram efficiency} = \frac{\text{work done on blade}}{\text{energy supplied to blade}}$$

$$= \frac{m(V_{w_1} + V_{w_2})U}{\frac{1}{2} m V_1^2}$$

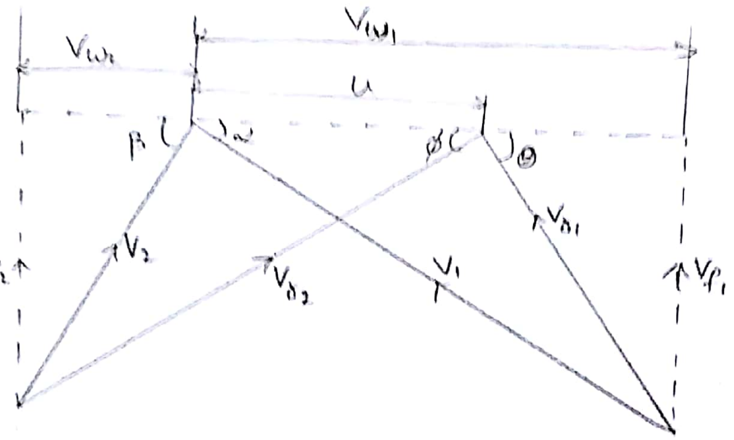
Blade efficiency or diagram efficiency

$$\eta = \frac{2U(V_{w_1} + V_{w_2})}{V_1^2}$$

... diagram with reference to Steam turbines :-

combined velocity diagram :-

The industrial method of constructing velocity diagrams is to combine both the velocity diagrams at inlet as well as at outlet of the blades into a single diagram as shown in figure. In this figure, both the velocity diagrams are drawn on a common base line representing the blade velocity 'u'.



combined velocity triangle for Parson's reaction turbine and explain the salient features :-

where,

V_{w_1} = velocity of whirl at entrance

V_{w_2} = velocity of whirl at exit

V_{f_1} = velocity of flow at inlet

V_{f_2} = velocity of flow at outlet

V_1 = absolute velocity at entrance

V_2 = absolute velocity at exit

u = Blade velocity

V_{r1} = relative velocity w.r.t the blade at inlet

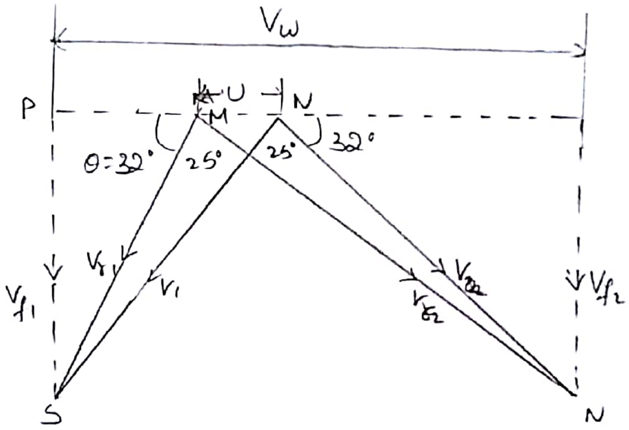
V_{r2} = relative velocity w.r.t the blade at outlet

α = angle made by the entering steam

β = angle made by the steam leaving the blade.

ϕ = outlet angle of blade

θ = inlet angle of blade.



Salient features :-

The combined velocity diagram of Parson's reaction turbine is symmetrical about the central line. Hence, following relations exists in combined velocity diagram.

$V_{f_1} = V_{f_2}$ $\qquad$ $\alpha = \phi$

$V_{r_1} = V_2$ $\qquad$ $\theta = \beta$

$V_{r_2} = V_1$

$FA = BE$

Different losses in reaction turbine :-

The following are the losses in reaction turbine,

1. Blade friction losses
2. wheel friction losses
3. Mechanical bearing friction losses
4. Residual velocity losses
5. Leakage losses
6. Radiation losses
7. Moisture losses
8. Governing losses

Preventive measures :-

1. Select the suitable materials for blades to reduce friction.
2. Select self lubricated bearings
3. Apply proper wheel balancing to reduce axial thrust and vibrating forces on wheel shaft
4. Proper radiation shields reduce radiation losses.
5. Use proper sealing to reduce leakages.
6. Use reheating and regeneration methods to reduce moisture.

dryness fraction 0.9 and pressure of 10 bar.

$P = 10$ bar

$x = 0.9$

steam tables,

$P = 10$ bar and $x = 0.9$, we get,

$h_f = 762.682$ kJ/kg

$h_{fg} = 2777.119$ kJ/kg

Specific enthapy of a wet steam with dryness fraction is given by,

$$h = h_f + x h_{fg}$$

$$h = 762.682 + 2499.407$$

$$= 3262.227 \text{ kJ/kg}$$

$\therefore$ Specific enthaply, $h = 3262.227$ kJ/kg

Specific volume, enthalpy and internal energy of wet steam at 18 bar, dryness fraction 0.85.

Pressure of wet steam, $P = 18$ bar

dryness fraction, $x = 0.85$

$P = 18$ bar and $x = 0.85$, we get,

$$V_g = 0.11 \text{ m}^3/\text{kg}$$

$$h_f = 884.6 \text{ kJ/kg}$$

$$h_{fg} = 1910.3 \text{ kJ/kg}$$

Specific volume :-

$$V = x V_g$$

$$V = 0.85 \times 0.11 = 0.094 \text{ m}^3\text{kg}$$

enthalpy of wet steam :-

$$h = h_f + x h_{fg}$$

$$= 884.6 + 0.85(1910.3)$$

$$= 2508.335 \text{ kJ/kg}$$

External work,

$w = P \times V$

$= 169.2 \, kJ/kg$

Internal energy wet steam :-

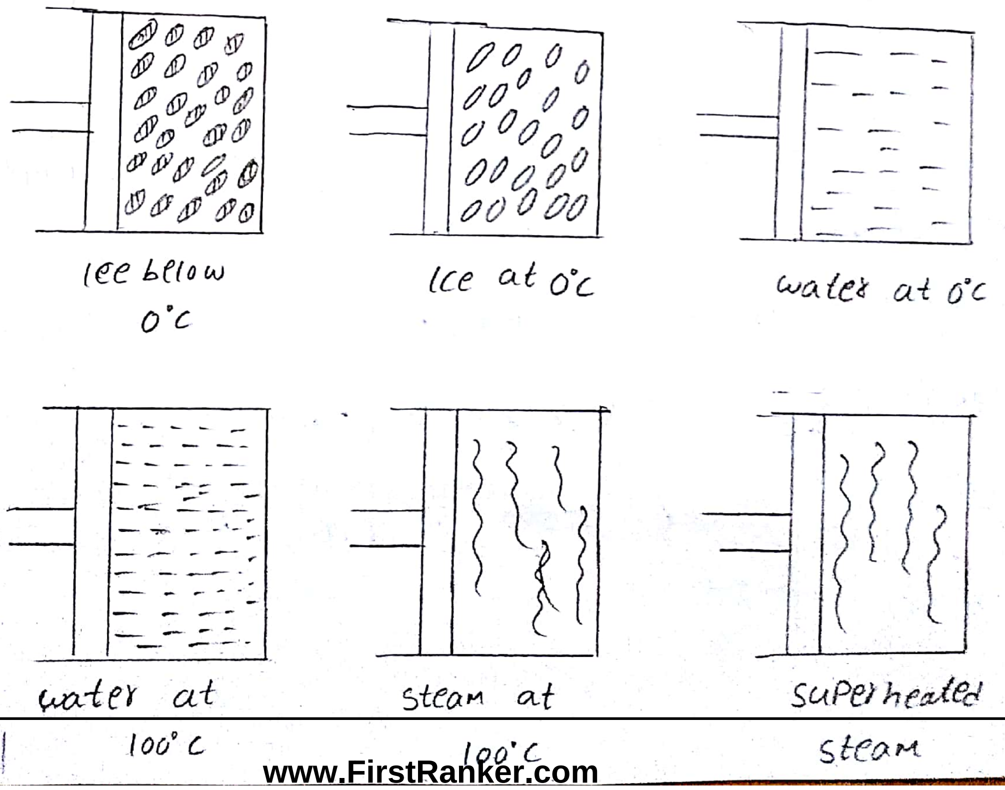
$u = h - PV_1$

$= 2508.3 - 169.2$

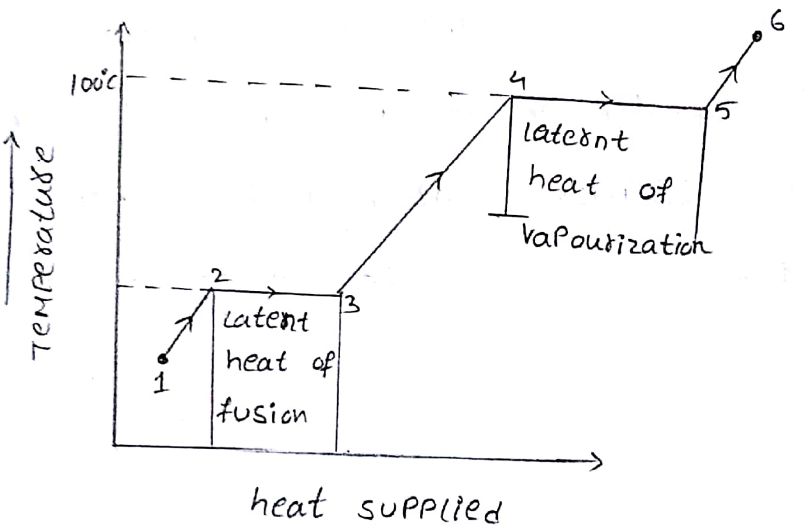
$= 2339.1 \, kJ/kg$

formation of steam and give its graphical representation:

The phase transformation of the water from solid ice to steam, is analysed by considering certain mass of ice which is at temperature less than 0°c. This is kept in a cylinder and pressure is applied on it. now heat is supplied slow to the cylinder containing ice. then, the phase transformation of ice takes place as shown in figure (1).



...ure (2) shows the graph drawn between the temperature and heat supplied. Point-1, in the graph shows the initial temperature of the ice before the heat is supplied. Due to the supply of heat, the temperature of ice increases to 0°c. This state is represented by point-2. The line '1-2' represents the supply of heat. on farther heating of ice, the temperature of ice remains constant, but the ice changes to water at 0°c.



Different terms pertaining to the steam formation :-

1. Dry saturated steam
2. sensible heat of water (h_f)
3. latent heat of vaporization (h_{fg})
4. Enthalpy of wet Steam or total heat
5. Specific volume of steam

1. Dry saturated steam :-

A steam that does not contain particles of liquid suspended in it is called as dry saturated steam. It behaves like a perfect gas.

2. sensible heat of water (h_f) :-

It is defined as the total amount of heat absorbed by a kg of water for converting to steam and raising the temperature from 0°c to the saturation point. it is denoted by 'h_f' and mathematically.

$$\therefore h_f = C_p T_{saturation}$$

3. Latent heat of vaporization (h_{fg}) :-

It is defined as the amount of heat absorbed by kg of water at its saturation temperature to get converted into a kg of saturated steam for a given pressure. Latent heat of vaporization is denoted by h_{fg} and its value depends on pressure.

4. Enthalpy of wet steam or total heat (h) :-

It is defined as sum of enthalpy of water and enthalpy of latent heat,

$$\therefore h = h_f + x h_{fg}$$

5. Specific volume of steam :-

It is defined as the volume occupied by the steam per unit mass, as pressure increases, gradually decreases. The expression for specific volume in case of wet, dry and superheated steam are given as follows,

Superheated steam :-

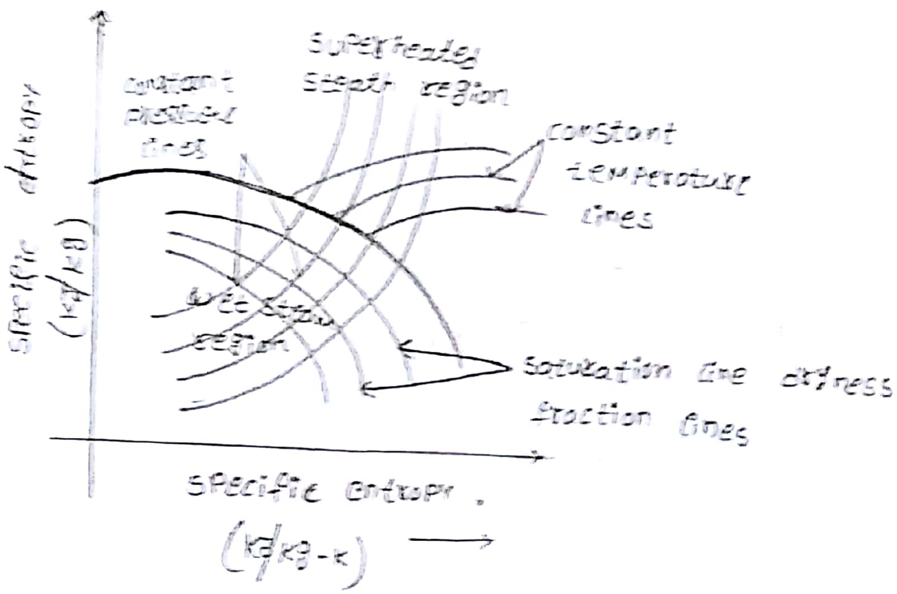
Superheated steam obeys Charles law i.e.,

$$\frac{pv}{T} = \text{constant}$$

$$\frac{V_{sup}}{T_{sup}} = \frac{V_g}{T_s}$$

$$V_{sup} = V_g \frac{T_{sup}}{T_s}$$

Mollier diagram :-



Mollier diagram is very useful in solving the problems on isentropic expansion and isentropic compression of steam. The figure is divided by saturation line into two portions. In the lower region, the temperature of steam remains constant at a given pressure. It is also termed as wet region. In the upper region the temperature of steam increases at the given pressure the region is also called superheated region.

The mollier diagram has the following lines,

1. constant pressure lines
2. constant temperature lines
3. Dryness fraction lines
4. Isothermal lines
5. Isothermal lines
6. Throttling lines

a) Isothermal process :-

It is a constant temperature process workdone during this process is given by,

$$W_{1-2} = Q_{1-2} - du$$

where,

Q_{1-2} - is the heat transferred during process.

'du' - is the change in internal energy.

change in internal energy is given by,

$$du = U_2 - U_1$$

where,

$$U_1 = h_1 - 100 P_1 V_1$$

$$U_2 = h_2 - 100 P_2 V_2$$

heat transferred during the process is given by,

$$Q_{1-2} = T_g \times (S_2 - S_1)$$

where,

T_g = saturation temperature.

Hyperbolic process,

workdone during the process,

$$W_{1-2} = 2.3 \times 100 \times P_1 \times V_1 \log\left(\frac{P_1}{P_2}\right) KJ/kg$$

change in internal energy

$$dU = U_2 - U_1$$
$$= h_2 - h_1$$

heat transferred during the process,

$$Q_{1-2} = dU + W_{1-2}$$

(c) Adiabatic process :-

work done during the process is given by,

$$W_{1-2} = -dU$$
$$= -(U_2 - U_1)$$
$$W_{1-2} = U_1 - U_2$$

change in internal energy

$$dU = U_2 - U_1 \; KJ/kg$$

where,

$$U_1 = h_1 - 100 \times P_1 \times x_1 \times V_{g1} \; KJ/kg$$

$$U_2 = h_2 - 100 \times P_2 \times x_2 \times V_{g2} \; KJ/kg$$

$$U_2 = h_2 - 100 \times P_2 \times V_{g2} \; KJ/kg$$

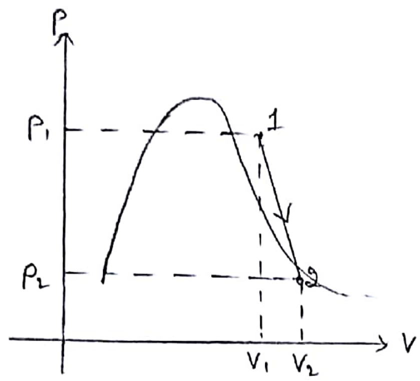
$$U_2 = h_2 - 100 \times P_2 \times V_{sup} \; KJ/kg$$

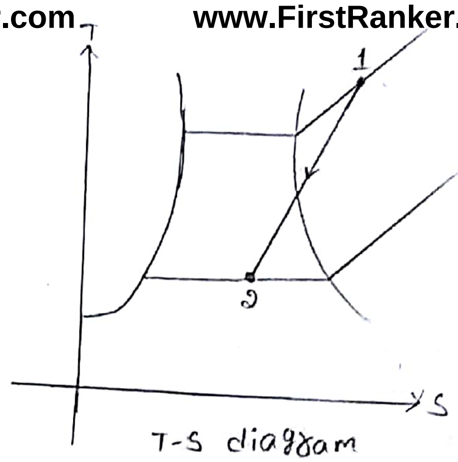
heat transferred during adiabatic process is '0'

where,

$$Q_{1-2} = 0$$

Polytropic process applied to steam :-





T-S diagram

A polytropic process is represented by,

$$PV^n = \text{constant}$$

where, n is the index.

In this process, there is a change in entropy. Hence, it is known as polytropic process.

heat interaction is given by,

$$Q = W + (u_2 - u_1)$$

workdone in a polytropic process is given as,

$$W = \frac{P_1V_1 - P_2V_2}{n-1} \quad \text{and} \quad U = h - PV$$

$$Q = \frac{P_1V_1 - P_2V_2}{n-1} + \{(h_2 - P_2V_2) - (h_1 - P_1V_1)\}$$

constant volume process :-

It is process, the volume of steam remains constant before and after the completion of process. It is also known as isochoric process and there is no workdone during the process.

consider a kg of steam that is heated at constant volume.

Since, the steam is heated at constant volume,

$$\therefore \quad V_1 = V_2$$

$$x V_{g_1} = x_2 V_{g_2}$$

Change in internal energy :-

change in internal energy of steam during isochonic process is given by,

$$du = U_2 - U_1$$

where,

$$U_1 = h_1 - 100 \times P_1 \times x_1 \times V_{g_1}$$

and

$$U_2 = h_2 + 100 \times P_2 \times x_2 \times V_{g_2}$$

$$U_2 = h_2 - 100 \times P_2 \times V_{g_2}$$

$$U_2 = h_2 - 100 \times P_2 \times V_{sup}$$

Isobaric process :-

In this process steam is generated by maintaining pressure constant. The pressure remains same before and after the process.

workdone :-

The workdone during isobaric process is given by,

$$W_{1-2} = 100 \times P \times (V_2 - V_1) \ kJ/kg$$

Change in internal energy :-

The change in internal energy for this process is given by,

$$dU = U_2 - U_1 \ kJ/kg$$

where, $U_1 = h_1 - 100 PV_1 \ kJ/kg$

heat absorbed :-

heat absorbed during the process is given by,

$$Q_{1-2} = (U_2 - U_1) + W_{1-2}$$

$$= (h_2 - 100\,PV_2) - (h_1 - 100\,PV_1) + 100\,(P)(V_2 - V_1)$$

$$= h_2 - 100\,PV_2 - h_1 + 100\,PV_1 + 100\,P\,(V_2 - V_1)$$

$$= (h_2 - h_1) + 100\,P\,(V_2 - V_1) - (100\,P\,(V_2 - V_1))$$

$$Q_{1-2} = h_2 - h_1 \text{ kJ/kg}$$

(a) specific entropy in the cases:-

1. $P = 20$ bar, $q = 0.8$ dry

2. $P = 32.8$ bar, dry and saturated

3. $P = 50$ bar, $T = 290°c$

$P = 20$ bar, $q = 0.8$ dry

At $P = 20$ bar & $q = 0.8$ dry, we get

$$S_f = 2.4473 \text{ kJ/kg-K}$$

$$S_g = 6.3367 \text{ kJ/kg-K}$$

specific entropy for wet steam in given by,

$$S_{wet} = S_f + q\,(S_g - S_f)$$

$$= 2.4473 + 0.8 \times (6.3367 - 2.4473)$$

$$= 2.4473 + 3.112$$

$$S_{wet} = 5.559 \text{ kJ/kg}$$

$P = 32.8$ bar, dry and saturated

from steam tables, at $P = 32.8$ bar for dry and saturated conditions, we get.

$T_{sat} = 238.876^\circ C$

$= 511.876\ K$

$h_{fg} = 1770.14\ kJ/kg$, $S_f = 2.692\ kJ/kg-K$

$S_{fg} = 3.457\ kJ/kg-K$

Specific entropy for dry and saturated condition of steam is given by,

$$S_g = S_f + \frac{h_{fg}}{T_{sat}}$$

$$= 2.692 + \frac{1770.14}{511.876}$$

$$S_g = 6.15\ kJ/kg-K$$

At = 50 bar and $T_{sup} = 290^\circ C$

Given that,

$P = 50\ bar$

$T_{sup} = 290^\circ C = 290 + 273 = 563\ K$

$S_g = 6.143\ kJ/kg-K$

$T_{sat} = 263.95^\circ C$

$T_{sat} = 263.95^\circ + 273 = 536.95\ K$

Specific entropy for super heated steam is given by,

$$S_{sup} = S_g + C_p \log_e \left(\frac{T_{sup}}{T_{sat}}\right)$$

$C_p = 2.0934\ kJ/kg-K$

$$S_{sup} = 6.143 + 2.0934 \times \log_e \left(\frac{563}{536.95}\right)$$

$$\therefore S_{sup} = 6.242\ kJ/kg-K$$

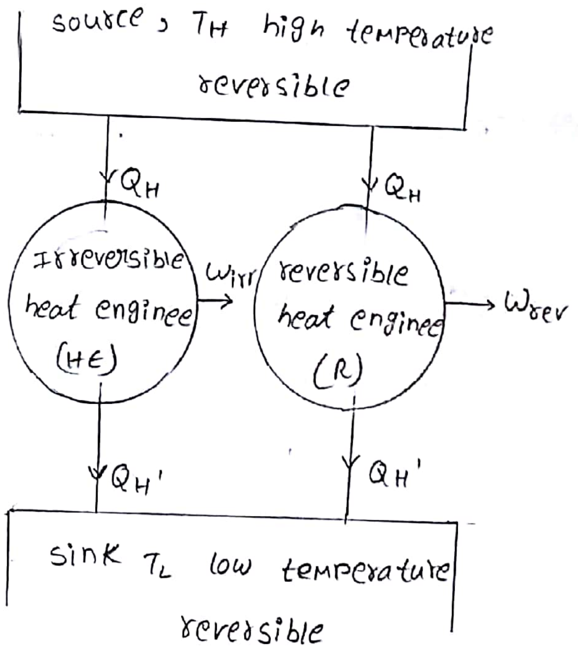
Carnot theorem :-

Statement :-

It states that "heat engines operating between a given constant temperature source and is given constant temperature sink, has a higher efficiency than a reversible engine".

Proof :-

consider two heat engines operating between the same reservoirs. let the source temperature be T_H and sink temperature be T_L. An engine is considered to be reversible and the other is considered to be irreversible.



In order to prove that efficiency of reversible heat engine is greater that of irreversible heat engine, let as assume that,

$$\eta_{irr} > \eta_{rev}$$

$$\frac{W_{irr}}{Q_{irr}} \quad \frac{W_{rev}}{Q_{rev}}$$

$$\frac{W_{irr}}{Q_{Hr}} > \frac{W_{rev}}{Q_{Hr}} \Rightarrow W_{irr} > W_{rev}$$

where $\eta_{irr} > \eta_{rev}$, irreversible heat engine delivers more work than the reversible one. Therefore, for assumption that efficiency of irreversible heat engine is greater than that of a reversible engine is proved wrong.

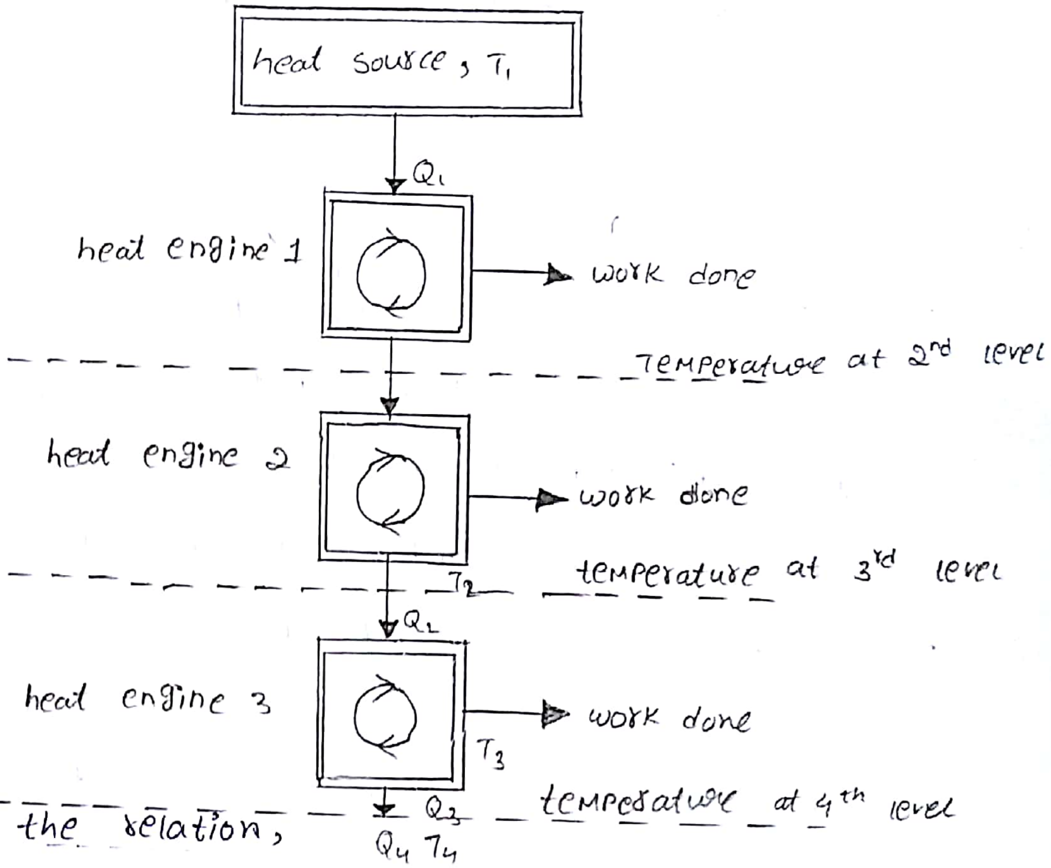
$$\therefore \eta_{rev} > \eta_{irr}$$

absolute temperature scale :-

Absolute temperature scale measures the temperature of the reservoirs of a reversible heat enginee cycle and is employed to determine its cycle efficiency. This scale is independent of thermometric substance and can be defined by an equation.

$$\frac{Q_1}{Q_2} = \phi(t_1, t_2) = \frac{T_1}{T_2}$$

Generally, steam point temperature (T_s) and ice point temperature (T_i) are considered to be fixed reference between the invertervel of 100° C



using the relation,

$$\eta_{th} = \eta_c = 1 - \frac{Q_2}{Q_1} = 0.268$$

or

$$\eta_c = 1 - \frac{T_i}{T_s} = 0.268$$

$$\frac{T_i}{T_s} = 0.732$$

$$T_i = 0.732\ T_s$$

substituting equation (2) in equation (1) we get,

$$T_s - 0.732\ T_s = 100$$

$$T_s = 373.134 \simeq 373.15 \quad \text{and}$$

$$T_i = 0.72\ (373.15)$$

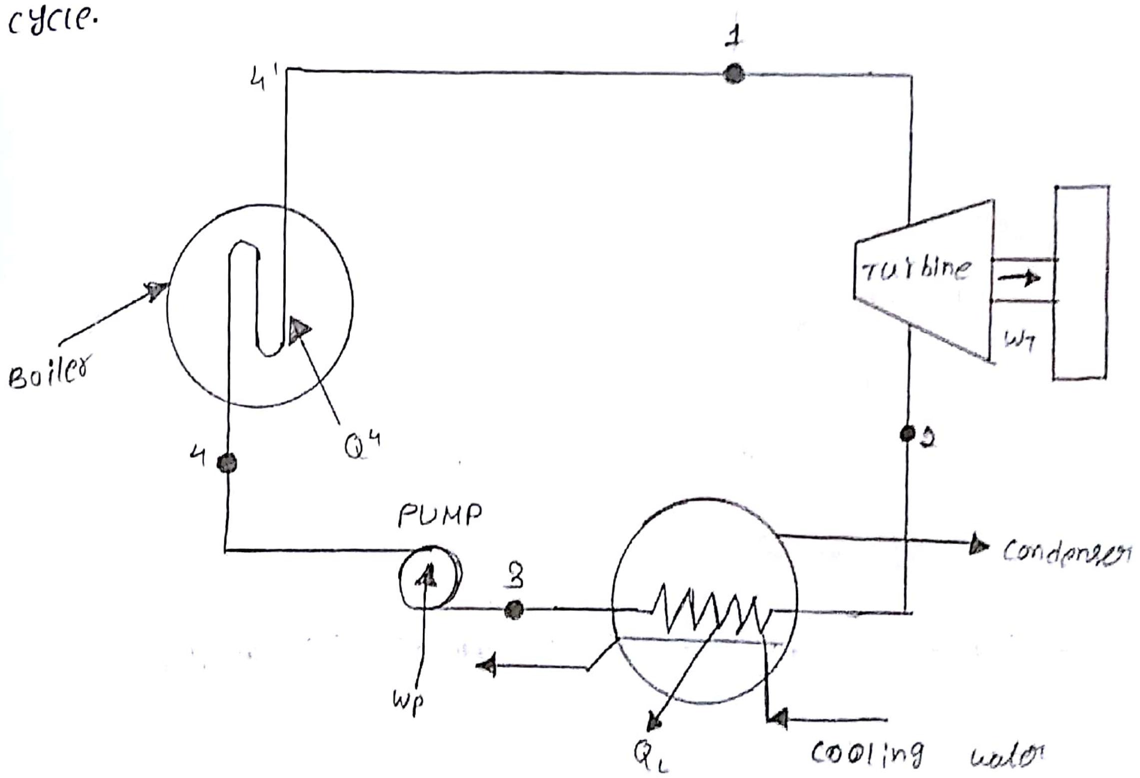
$$T_i = 268.67$$

Thus, the relation for absolute scale & Kelvin scale can be expressed in degree centigrade as,

$$T^\circ K = T^\circ C + 268.67$$

Rankine cycle :-

The thermodynamic cycle for steam power plant is rankine cycle. The various limitations of carnot cycle are eliminated in rankine cycle by superheating the steam in a turbine and condensing it completely in the condenser. each of the four processes constitude the cycle is a theoretical processes that may be nearly achieved in an actual power cycle.



Ideal rankine cycle is shown in figure (1). The four processes involved in the complete cycle are different from each other and each processes required a separate component.

Process 1-2 :- Reversibe adiabtic expansion in the turbine from pressure P_1 to P_2 to produce the work output.

Process 2-3 :- heat transfer from the steam in the condenser, steam is condensed to saturated water at 3.

Process 3-4 :- Reversible adiabatic compression of water in the pump. The pressure of water increases from P_2 to P_1.

Process 4-1 :- Heat transfer to water at constant pressure in the boiler to produce steam.

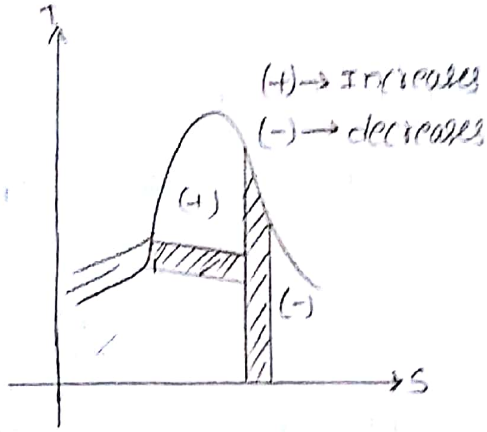
$$\eta_R = \frac{\text{work done}}{\text{heat absorbed}}$$

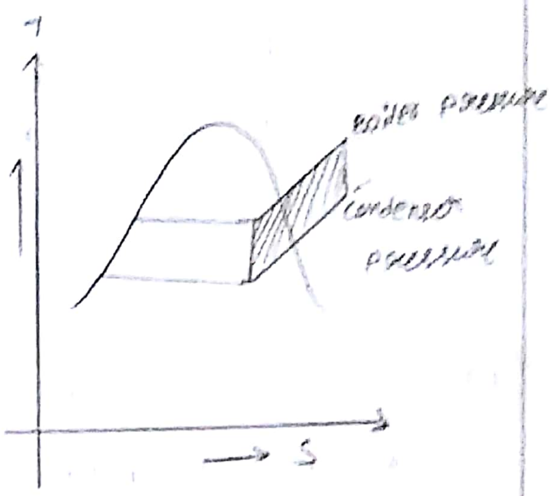
$$= \frac{h_1 - h_2}{h_1 - h_{f3}}$$

Maximum temperature & maximum pressure in the rankine cycle fixed :-

A higher boiler pressure using dry-saturated and a constant condenser pressure gives high work output and higher termal efficiency as shown in figure. But use dry saturated steam at highe pressure gives more moisture in the lower stages of turbine. a maximum temperature is fied from metallurgical considerations.

when the maximum temperature of steam is ed the steam pressure increases at which heat is added in the boiler.





Different b/w impulse & reaction turbine :-

S.No	Impulse turbine	reaction turbine
1.	During flow of steam, pressure drops only in the nozzle	During flow of steam, Pressure drops over fixed nozzle and Moving blades.
2.	The blades of impulse turbine are of Profile types.	The blades of reaction turbine are of aerofoil type.
3.	The blades are symmetrical	The blades are not symmetrical
4.	The number of stages required for a given Power is less.	The number of stages required for a given power is more.
5.	efficiency of impulse turbine is less	efficiency of reaction turbine is more
6.	Steam velocity is more	Steam velocity is less

Steam power plant layout :-

The modern steam power plant mainly divided into circuits. They are,

1. coal and ash circuit
2. Air and gas circuit
3. cooling water circuit.

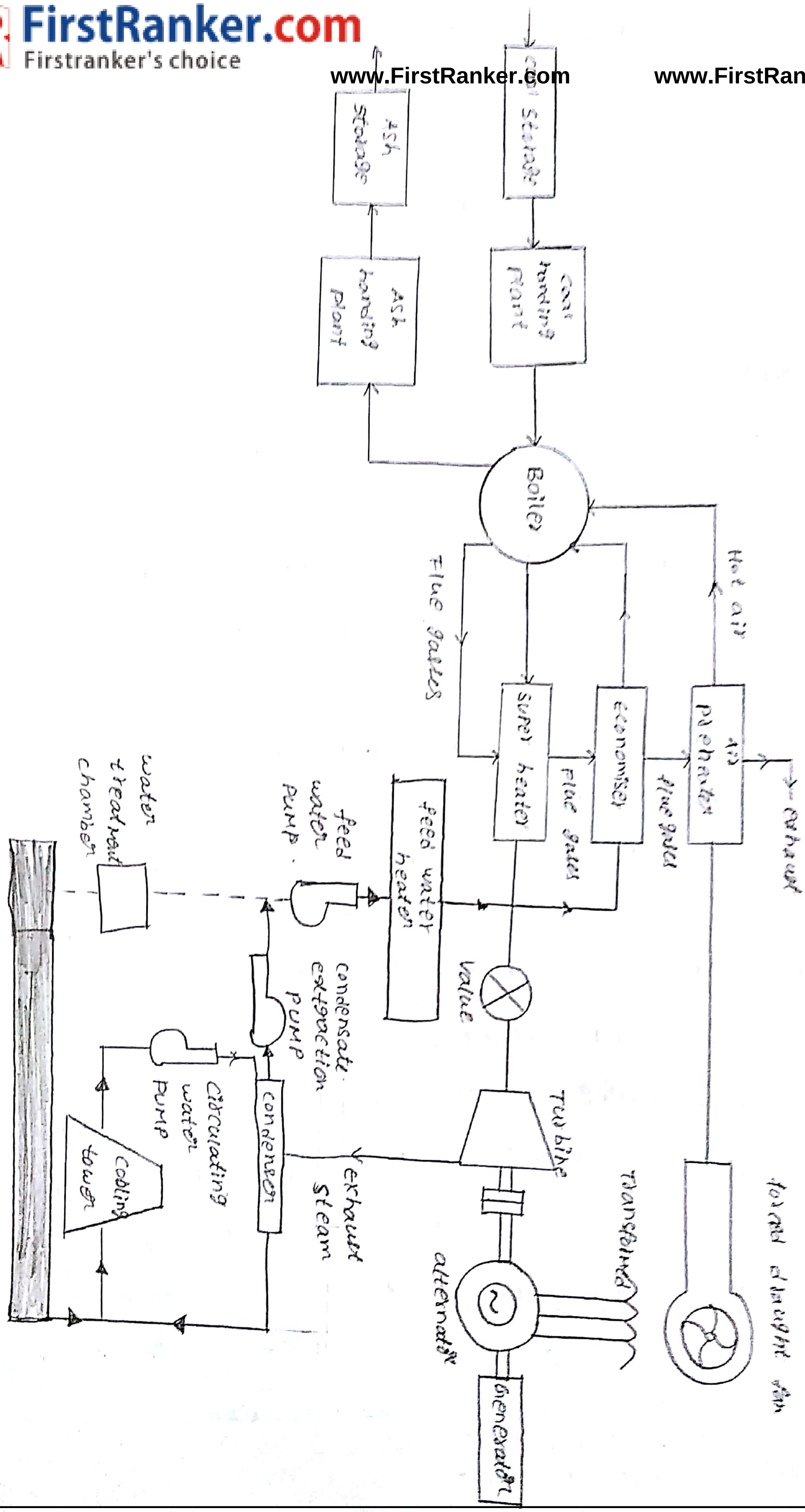
coal and ash circuit :- It comprises of coal hadling equipment and ash handling equipment. coal is supplied to the boiler from the storage point by coal handing equipment and steam is generated in the boiler.

Air and Gas circuit :- Air and is blown to the combustion chamber by induced draught fan or forced draught fan or both. The dust present in the air is removed by dust catching device or precipitator. The heat in the exhaust gases is used to pre-heat the air.

cooling water circuit :-

The cooling water circuit to the condenser helps in maintaining a low pressure of steam in it. the water may be taken from the taken source like river, lake or same water may be cooled and circulated again. Based on these, this circuit is divided into,

1. open systeam and
2. closed systeam


Coal storage
Ash storage
Coal handling Plant
Ash handling Plant
Boiler
Flue gases
Hot air
Super heater
Economiser
Air Preheater
exhaust
Flue gases
flue gases
feed water heater
feed water PUMP
water treatment chamber
condensate extraction PUMP
value
Turbine
exhaust steam
condenser
circulating water PUMP
Cooling tower
Transformer
alternator
Generator
forced draught fan

Single stage impulse turbine :-

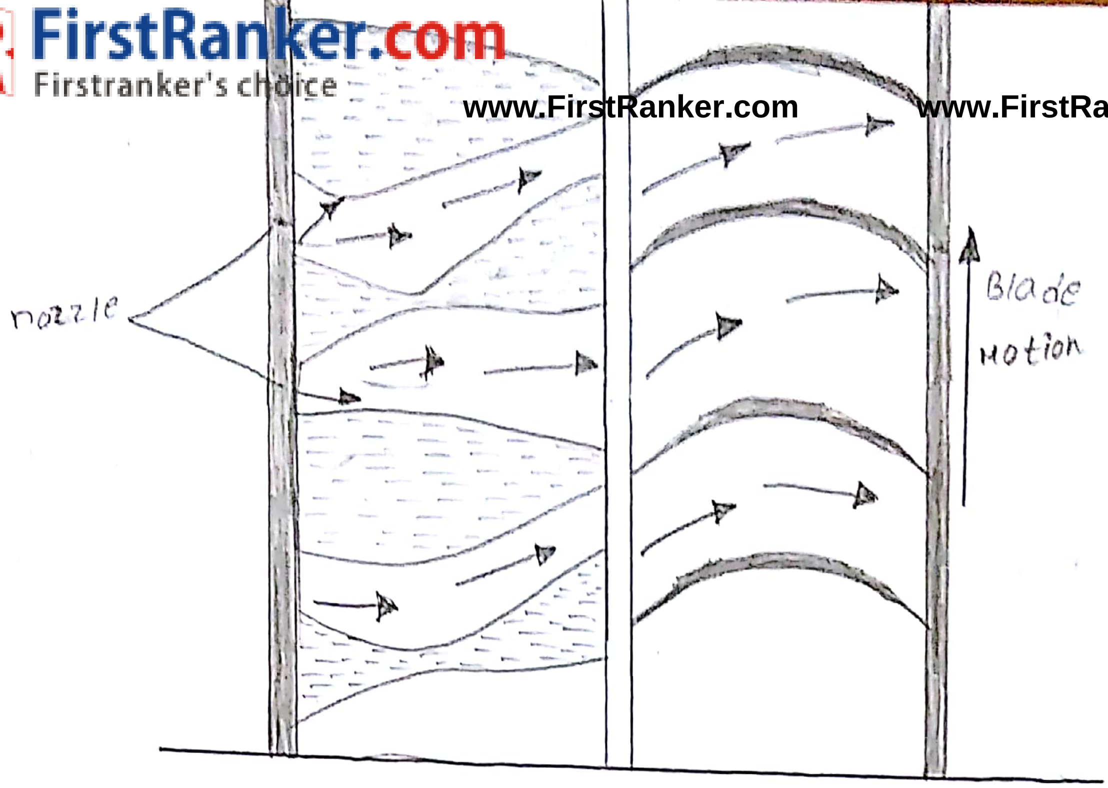
In a single stage impulse turbine, the steam from the nozzle enters into the moving blade and exhaust into the condenser.

The shape of the nozzle and blading during the flow of steam is as shown in figure (1). Here, complete expansion occurs in a stationary nozzle decreasing the enthalpy and increasing the kinetic energy which is then converted into shaft work.

working :-

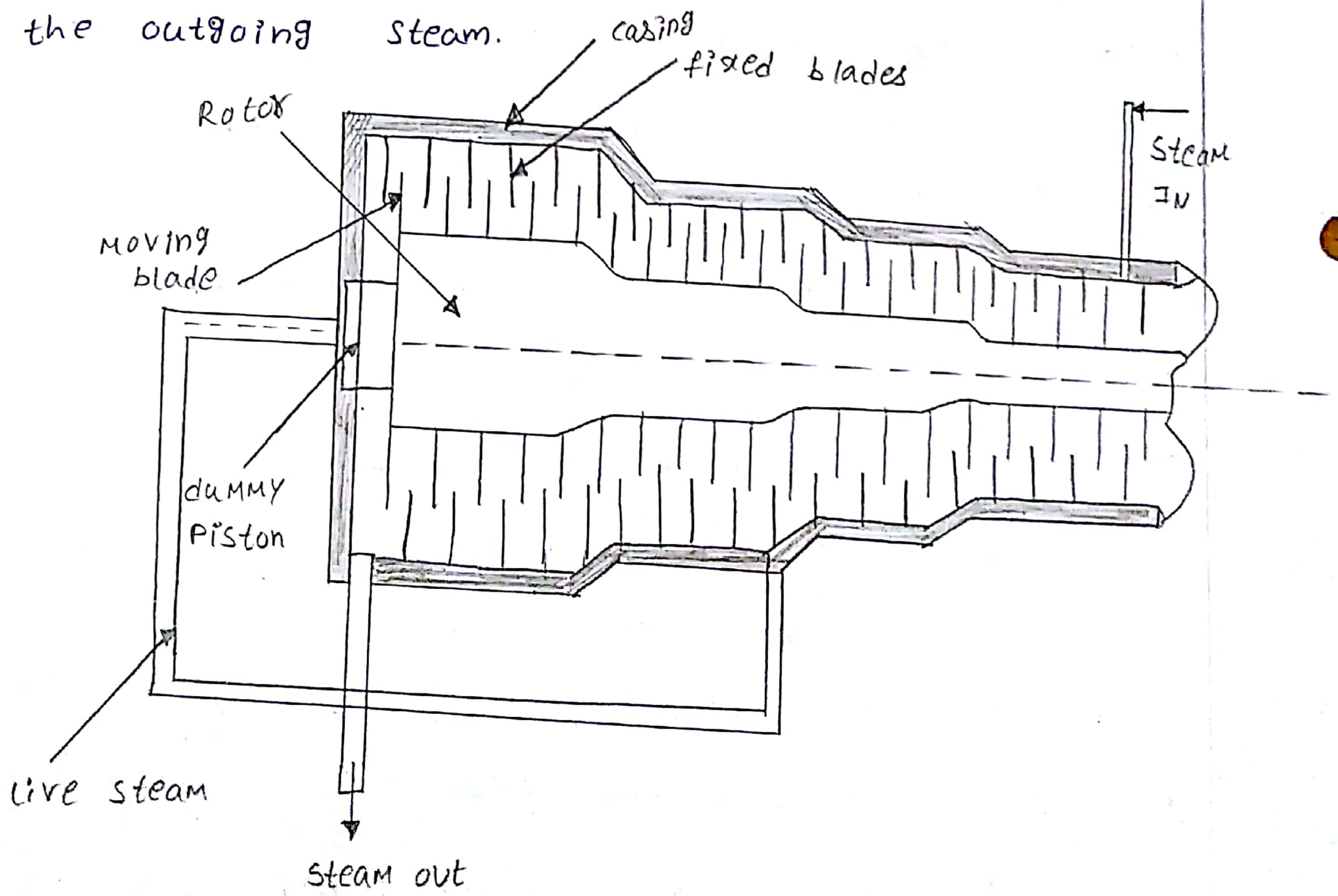
A single stage impulse turbine consists of one set of nozzles and one set of moving blades. The function of nozzle is not only to allows the flow of steam at a specified direction but also to regulate the steam flow. The blades of steam turbines are fixed uniformally on the steam the runner. As the steam impinge on the blades, it starts moving which causes the runner to rotate.

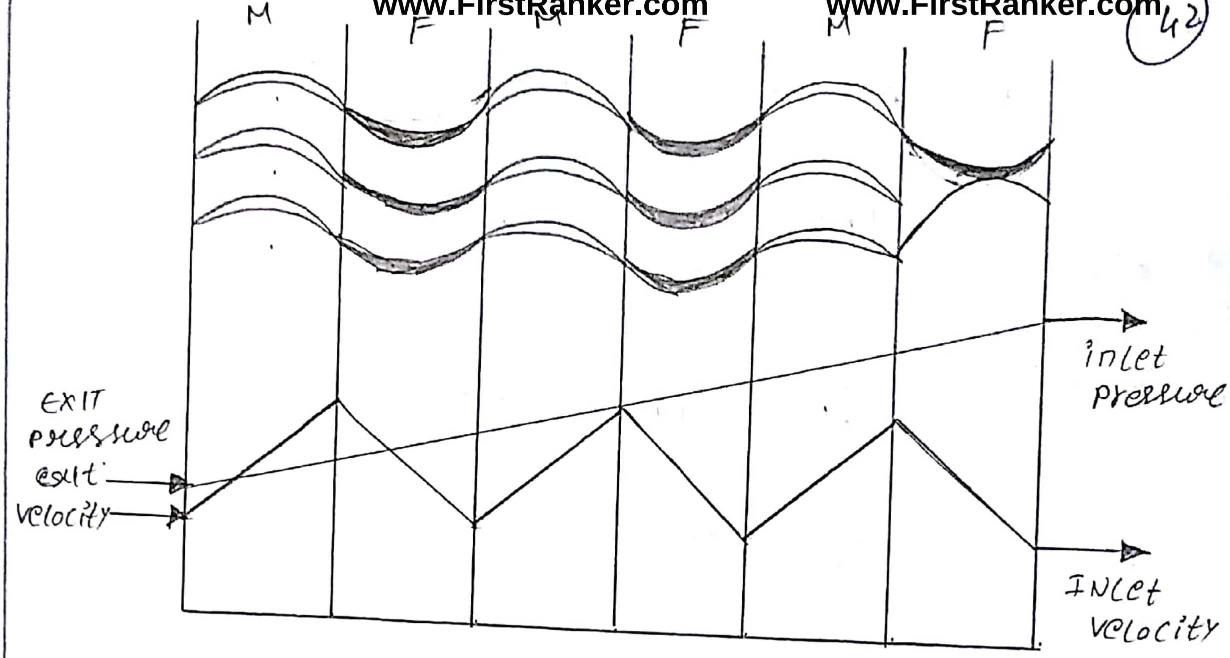
when the steam enters the nozzle, it expands from initial pressure to final pressure and increases the velocity of steam due to higher pressure drop occur in the nozzle as shown in figure.



Parson's reaction turbine :-

In Parson's Reaction turbine power is mainly obtained by an impulsive force of the incorming steam and small reactive force of the outgoing steam.





This turbine consists of a rotor and casing of varying diameter as shown in figure (2). The diameters of the casing and rotor increases gradually to accommodate the increases in volume of steam at reduced pressure.

Degree of Reaction :-

degree of reaction is defind as the ratio of enthalpy drop in moving blades to the total enthalpy drop in the stage. Mathematically

$$\text{degree of reaction} = \frac{\text{enthalpy drop in moving blades}}{\text{enthalpy drop in the stage}}$$

$$= \frac{\Delta h_m}{\Delta h_m + \Delta h_f}$$

$\Delta h_f = \frac{V_1^2 - \ }{2000}$ kJ/kg → (2)

enthalpy drop in moving blades,

$$\Delta h_f = \frac{V_{r_2}^2 - V_{r_1}^2}{2000} \text{ kJ/kg}$$

$$\Delta h = \Delta h_m + \Delta h_f$$

$$\Delta h = \frac{V_{r_2}^2 - V_{r_1}^2}{2000} + \frac{V_1^2 - V_2^2}{2000}$$

from Parson's reaction turbine

$V_1 = V_{r_2}$ and $V_2 = V_{r_1}$

equations becomes,

$$\therefore \Delta h_m + \Delta h_f = \frac{V_{r_2}^2 - V_{r_1}^2}{2000} + \frac{V_{r_2}^2 - V_{r_1}^2}{2000}$$

$$\Delta h_m + \Delta h_f = 2\left(\frac{V_{r_2}^2 - V_{r_1}^2}{2000}\right)$$

$$\Delta h_m + \Delta h_f = 2\,\Delta h_m \text{ kJ/kg}$$

now, substituting equation in equation (1) we get,

$$\text{degree of reaction} = \frac{\Delta h_m}{2\,\Delta h_m} = \frac{1}{2} = 0.5$$

$$= 50\%$$

For a Parson's reaction turbine, the degree of reaction is 50%

Gas Turbines

Gas turbines is a rotary external combustion engine. It uses continuous flow of gases as working medium in order to convert heat energy into mechanical energy.

* classification of gas turbine:

The gas turbines are classified as follows,

1. According to the combustion process,
   (i) Constant pressure type (Joule or Brayton Cycle)
   (ii) Constant Volume type (Atkinson Cycle).
2. According to the direction of flow,
   (i) Axial flow
   (ii) Radial flow.
3. According to action of Expanding gases,
   (i) Impulse turbine
   (ii) Impulse - reaction turbine
4. According to path of working Substance,
   (i) open Cycle gas turbine
   (ii) closed Cycle gas turbine.

* Assumption in ideal gas turbine Cycle Analysis:

The following assumption are made to analyse an ideal gas turbine cycle,

1. The working fluid is a prefect gas with constant specific heat and has same composition through

out the cycle.

2. The mass flow of gases are assumed to be constant throughout the cycle.
3. compression and expansion of working fluid are considred to be isentropic.
4. No pressure losses are considered.
5. Bearing losses, transmission losses and auxiliary equipment losses are neglected.
6. The change in kinetic energy at inlet and outlet of devices is negligible.
7. The efficiencies of compressor, turbine and heat exchanger are assumed to be 100%.

* Advantages of the closed cycle over the open cycle gas turbine:

1. The air remains clean, hence turbine unit can be kept clean for indefinite period.
2. specific volume of air entering the compressor is low. Thus the compressor size is comparatively smaller.
3. Compressor and turbines sizes are smaller for given output.
4. Any type of fuel can be used since combustion is effected externally to the working fluid.
5. Easier to adopt to marine propulsion.
6. Smaller in weight any occupy less space.
7. Thermal efficiency of the plant is higher

Disadvantages of the Closed cycle over the open cycle gas turbine :-

1. Requires very large air heater
2. The complexity of the system increases the coast and engineering problems.
3. Large amount of cooling water is required.

* Assumption made in Actual Brayton Cycle :-

1. Air is used as working substance, behaves like an ideal gas and obeys gas laws.
2. All thermodynamic processes are reversible
3. No pressure losses are considered
4. Specific heat of air remain constant
5. Bearing losses, transmission losses and auxilary equipment power losses are not considered.
6. Compression and expansion of air are considered to be isentropic
7. Change in kinetic energy between the various points in the cycle is negligible.
8. No chemical reactions are considered.

* The influence of isentropic efficiency of compressor and turbine on thermal efficiency of gas turbine unit is shown in figure below.

The thermal efficiency of gas turbine unit is very sensitive to varitions in the isentropic efficiencies of the turbine and compressor. The figure shows that there is a particular pressure ratio for which maximum efficiency occurs. The peak of the thermal efficiency occurs at lower pressure ratio for lower turbine and compressor efficiencies. Overall thermal efficiency of the ~~input~~ unit improves as the efficiency of turbine and compressor increases

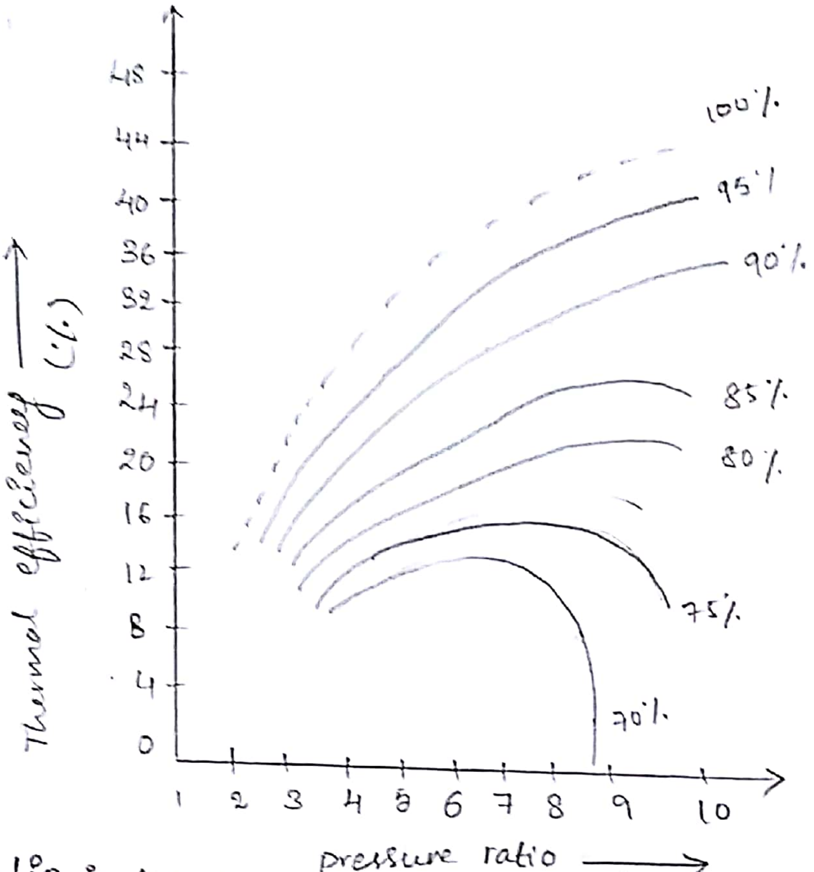


figure: Effect of Turbine and compressor efficiencies

* work Ratio :

It is defined as the ratio of the net work obtained to the turbine work generated. It is given by,

$$W_R = \frac{W_T - W_C}{W_T}$$

Where,

W_C = Work done by compressor

$= C_p (T_2 - T_1)$

W_T = Work done by turbine

$= C_p (T_3 - T_4)$

$$W_R = \frac{C_p\left[(T_3 - T_4) - (T_2 - T_1)\right]}{C_p(T_3 - T_4)}$$

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

$$= \frac{T_4\left[\frac{T_3}{T_4} - 1\right] - T_1\left[\frac{T_2}{T_1} - 1\right]}{T_4\left[\frac{T_3}{T_4} - 1\right]}$$

But, pressure ratio,

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

$$= \frac{P_3}{P_4}$$

For an isentropic process,

$$\frac{T_2}{T_1} = \left[\frac{P_2}{P_1}\right]^{\frac{\gamma-1}{\gamma}} \text{ and } \frac{T_3}{T_4} = \left[\frac{P_3}{P_4}\right]^{\frac{\gamma-1}{\gamma}}$$

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

Substituting the above values in equation (1)

$$\therefore W_R = \frac{T_4\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right] - T_1\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]}{T_4\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]}$$

$$= \frac{T_4 - T_1}{T_4}$$

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

$$\therefore W_R = 1 - \frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\left[\because \frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{(r_p)^{\frac{\gamma-1}{\gamma}}}\right]$$

: Thermal efficiency of gas turbine plant increases with an increase in pressure ratio till it becomes maximum. There after, it gradually decreases. Higher thermal efficiency is obtained by lowering the specific work output. This results in increase in the size of plant which increases the capital cost due to low specific work output. However, optimum pressure ratio is generally desired rather than high pressure ratio.

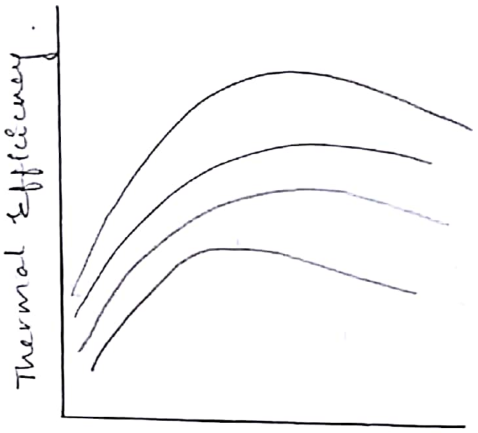


figure:

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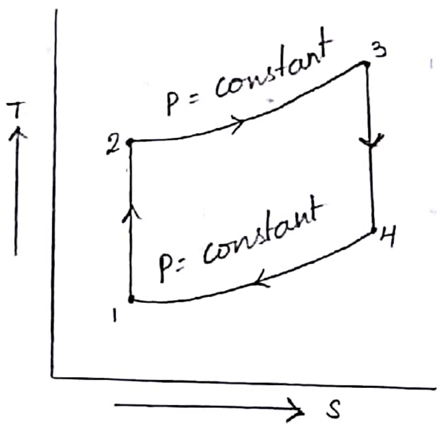


Figure

The net specific work output is given by

$$W_{net} = W_T - W_C = m C_p (T_3 - T_2) - m C_p (T_4 - T_1)$$

$$W_{net} = m C_p (T_3 - T_2) - m C_p \left[T_3 \left(\frac{T_1}{T_2} \right) - T_1 \right] \quad \left[\because \frac{T_3}{T_4} = \frac{T_2}{T_1} \right]$$

The net work done depends on only T_2 as T_3 and T_1 are fixed.

So for maximum specific work output, differentiate the above equation with respect to T_2 and equate is to zero.

$$\frac{dW_{net}}{dT_2} = -mc_p + mc_p \frac{T_3 T_1}{T_2^2} = 0$$

$$mc_p = mc_p \frac{T_3 T_2}{T_2}$$

$$T_2^2 = T_3 T_1$$

$$T_2 = \sqrt{T_3 \times T_1}$$

$$T_2^2 = T_3 \times T_1$$

$$\text{But } T_3 = T_4 \left[\frac{T_2}{T_1}\right]$$

$$\frac{T_2^2}{T_1} = T_4 \left[\frac{T_2}{T_1}\right] \qquad \left[\because T_3 = \frac{T_2^2}{T_1}\right]$$

$$\boxed{\therefore T_2 = T_4}$$

∴ For maximum specific work output, the outer temperture of compressor must be equal to the outlet temperture of turbine.

* In simple gas turbine plant, air is first compressed adiabatically in process (1-2) as shown in the figure below. It then enters the combustion chamber where fuel is injected and burnt essentially at constant pressure in process (2-3) The products of combustion expands in the turbines adiabatically to the atmospheric pressure in process (3-4).

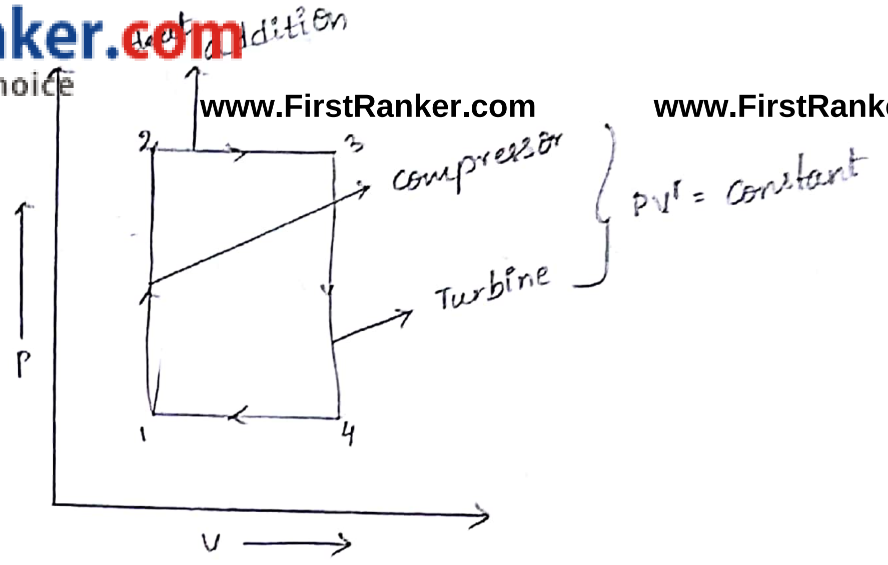


Figure : P-V Diagram

The Brayton or Joule cycle is the ideal thermodynamic cycle for the (closed) gas turbines with constant pressure combustion as shown in figure. Gas is first compressed reversibly and adiabatically in (process 1-2) and heat is added to high pressure gas at constant pressure. Then the gas expands in the turbine reversibly and adiabelity to the initial pressure. The heat of the gas in then rejected reversibly at constant pressure in the cooler to bring the gas to its initial pressure and temperature conditions. Therefore, the brayton or joule cycle consists of two reversible adiabatic and two reversible isobars.

* The different parameters that influence the performance of gas turbine are as follows,

(a) Pressure Ratio :

It is defined as the ratio of higher pressure to the lower pressure of a cycle. It is denoted as "r_p" and it is given as,

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

(b) Work Ratio:

It is defined as the ratio of net output of work to the turbine work. It is denoted by W_R. Mathematically,

$$W_R = \frac{W_{net}}{W_T} = \frac{W_T - W_C}{W_T}$$

where,

W_T = Turbine Work

W_C = Compressor Work

(C) Air Ratio:

It is the amount of air (in kg) entering into the inlet of compressor per unit net output work of a cycle

(D) Compressor Efficiency:

It is defined as the ratio of work required for ideal air compression at a specified range of pressure to the actual work utilized by the Compressor.

(e) Engine Efficiency:

It is defined as the ratio of actual work generated by turbine, expanding gas at a specified range of Pressure to the yielded ideal expansion conditions

(f) Machine Efficiency:

It is defined as the product of an engine efficiency and efficiency of turbine and compressor.

(g) Combustion Efficiency:

It is defined as the ratio of amount of heat released by 1 kg of fuel to the amount of heat released by entire combustion of fuel.

(h) Thermal Efficiency:

It is defined as the ratio of network output to the amount of heat supplied.

Intercooler :

In gas turbine power plants inter-cooler is used only when the pressure ratio is very large and the compression is accomplished. cold water is used for cooling the compressed air. cross-flow type of inter-cooler is mostly used for obtaining higher efficiency, with effective heat transfer rate

(b) Reheat cycle :

The work output and thermal efficiency of a gasturbine can be improved by introducing reheating process during expansion without changing the compressor work (constant pressure) or maximum temperture in a cycle.

Therefore, the constant pressure heating between two turbine expansion is known as reheating and this modified cycle is known as reheat cycle.

(c) Regeneration :

Regeneration is the process of heating the air coming out from the compressor by utilizing the heat of the turbine exhaust gases. As the exhaust gases from the turbine has a large quantity of heat, it canbe utilized effitively to raise the thermal efficiency of the cycle. The high pressure air from the compressor flows through heat exchanger where the heat exchange between the turbine exhaust gases and the air takes place, thereby heating the air as close to the temperture of the exhaust gases. This hot air is then supplied to the combustion chamber where combustion ~~chamber where~~ combustion of fuel takes place and

... air is again heated to the turbine inlet temperture. Regenaration does not have any effect on the compressor work, turbine work and network. It only helps in reduction in heat supplied in the combustion champer, thereby increasing the thermal efficiency of the cycle.

* layout of gas turbine powerplant:

The performance of a gas turbine power plant is effected by the type of layout. large number of sharp bends in the interconnecting ducts can lead to a loss of as much as 20% of power developed. Therefore, proper care has to be taken while designing the layout of air as well as gas circuits.

following figure shows a typical layout of gas turbine plant.

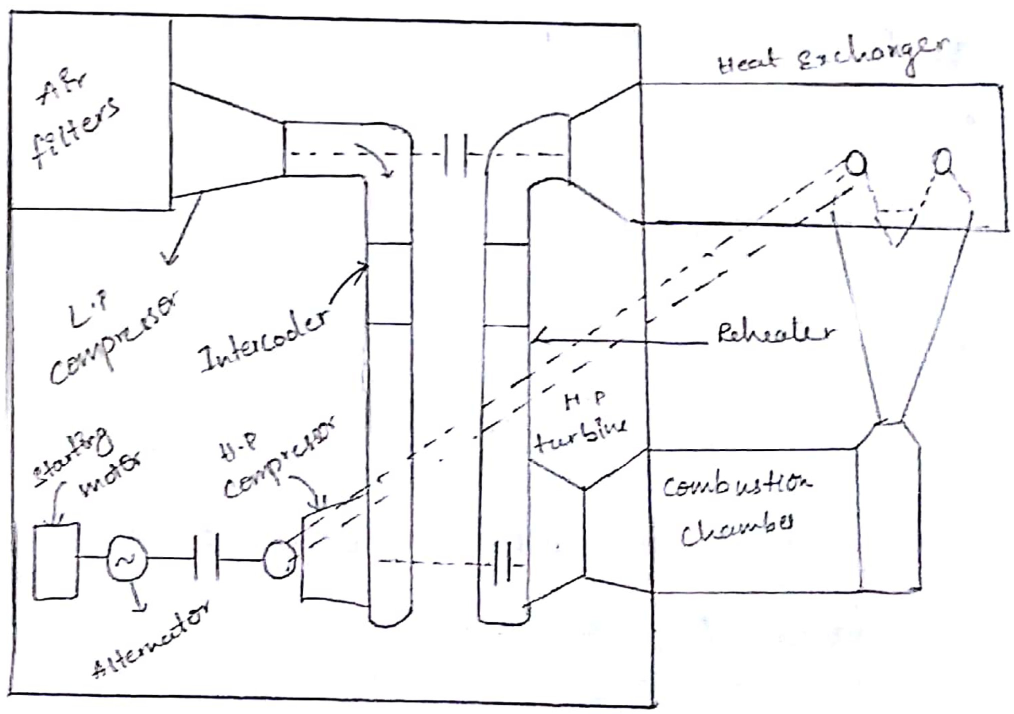


Fig: Layout of gas Turbine Power plant.

The main building in the layout is the turbine house in which major portion of the plant along with auxiliaries are installed. It is simillar to the steam turbine plant house. fuel storage tanks are arranged outside but adjoing the turbine house. In some Cases, even the heat exchangers are installed outdoors.

Intercoolers, combustion chamber, heat chamb exchanger, waste heat boilers and interconnecting ducts occupy a major portion of the space and therefore have to be arranged wisely. Rotating parts of the plant occupy a very small portion of the total volume.

Air after getting filtered by passing through the air filters flow into the low pressure (L.P) Compressor. This compressed air enters the high pressure (H.P) Compressor through intercooler. Air leaving the H.P compressor enters heat exchanger where it gets heated and flows to the combustion chamber. The products of combustion first expand in H.P turbine followed by the L.P. turbine.

* closed Cycle Gas turbine power plant :-

Closed cycle gas turbine plant consists of a Compressor, Condenser, air heater and a gas turbine as shown in the figure (1)

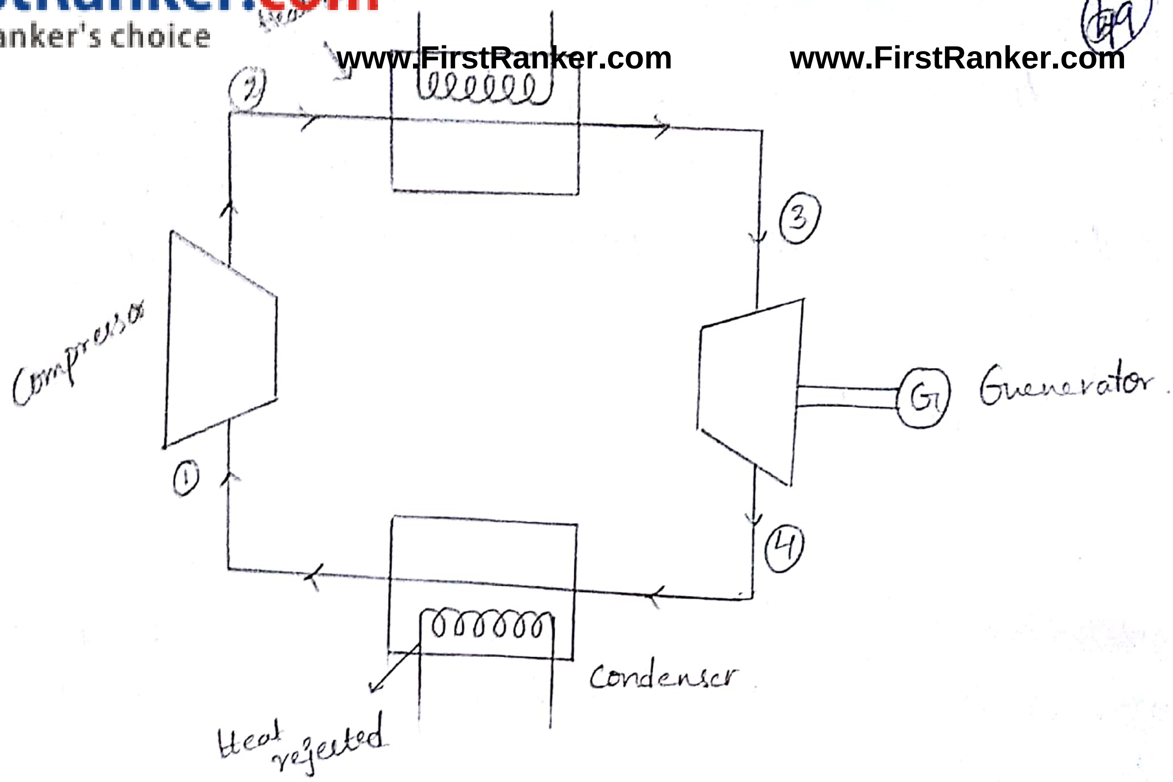


fig: (1) closed cycle gas turbine power plant.

In this System, air or any Suitable gas is used as a working fluid, which circulates Continously within the system. The Compressed working fluid from the compressor is heated in an air heater by an external Source at constant pressure. The working fluid exerts high temperture and pressure in the air heater. This high pressure fluid from the heater, enters into the gas turbine where it will get expaned on the turbine blades producing power. Thus, the working fluid from the turbine is cooled to its initial temperture in the condensor and recirculated to compressor to repeat the cycle.

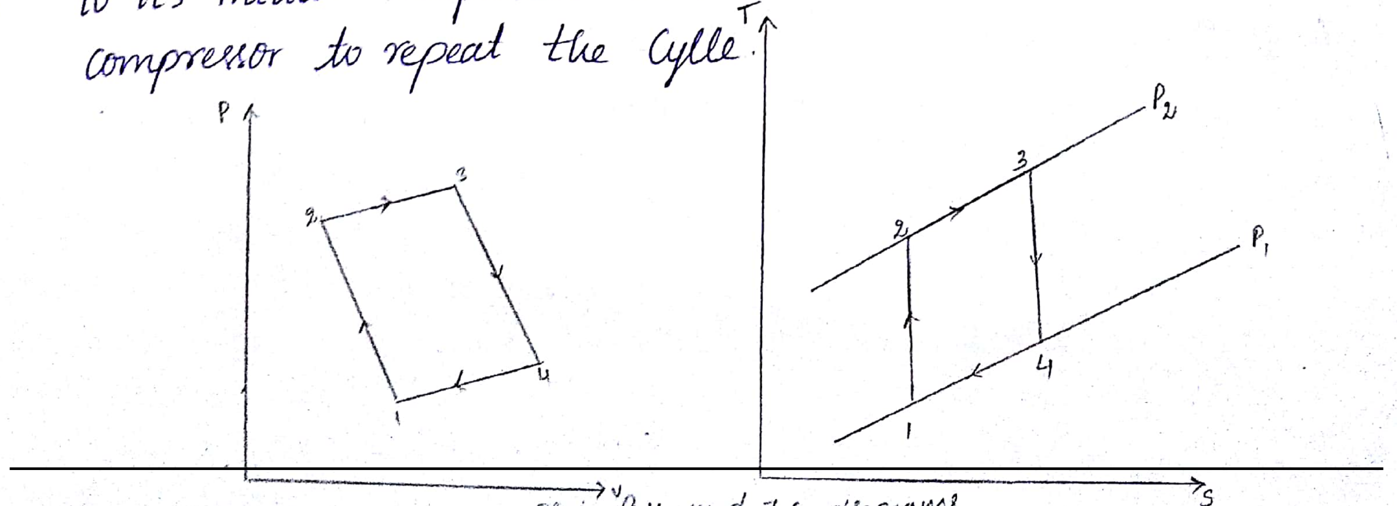


fig: P-V and T-s diagrams

Process 1-2 (Isentropic compression): The pressure of the fluid raises from P_1 to P_2 and temperature from T_1 to T_2

Process 2-3 (Constant pressure process): Under heating, the volume of the fluid increases from V_2 to V_3 and temperture from T_2 to T_3

Heat supplied $= mc_p(T_3 - T_2)$

Process (3-4) (Isentropic Expansion process): The fluid pressure decreases from P_3 to P_4 and temperture from T_3 to T_4.

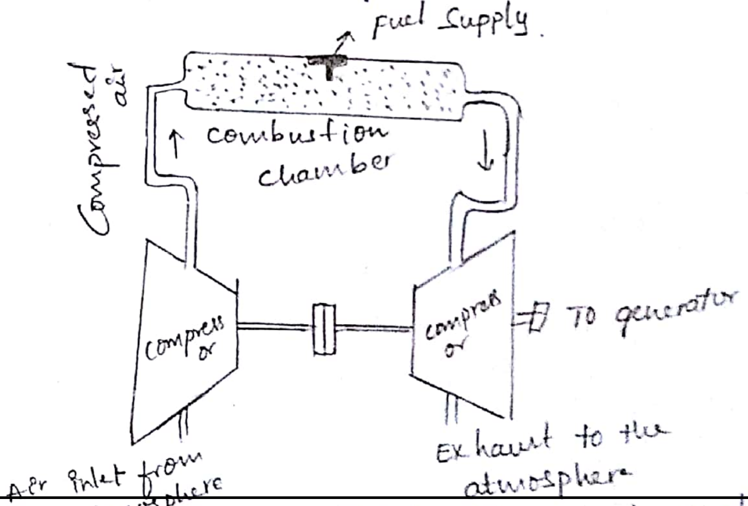
Process 4-1 (Constant pressure process): with the heat rejection, volume decreases from V_4 to V_1 and temperture from T_4 to T_1.

Heat Rejected $= mc_p(T_4 - T_1)$.

* Simple constant pressure open cycle gas turbine:

The fundamental cycle for gas turbine was given by joule, which comprises four processes. viz., Isentropic Compression Isentropic expansion and the remaining two process would take place at constant pressure. Therefore, this cycle is also known as constant pressure cycle or Brayton cycle.

The working of a simple constant pressure gas turbine or open cycle constant pressure gas turbine is shown in the figure below and also represented on PV and T-S diagram.



(fig): open cycle gas turbine or simple constant pressure gas turbine

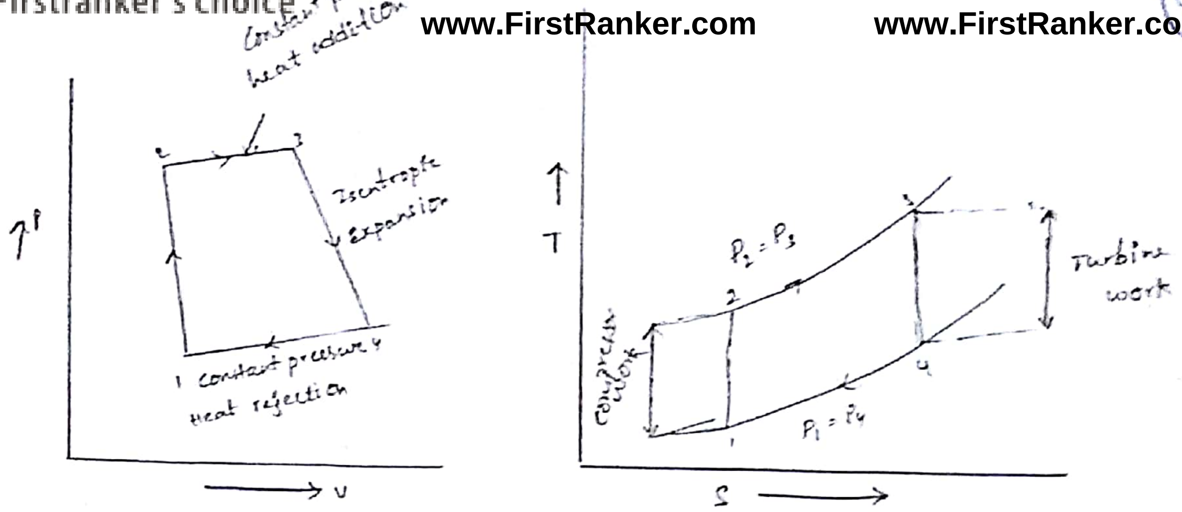


Figure (6) open cycle Gas Turbine

Working:-

Open cycle gas turbine works on the principle of joule (Brayton) cycle. Air enters the compressor at ambient conditions and is compressed to high pressure. This compression follows isentropic (adiabatic) to curve 1-2 on T-S diagram, this high pressure air is sent to combustion chamber where fuel is used to burn at high tempretures. Temperatures reaches the maximum at this point and this high temperture and high pressured gas is expanned in turbine thereby producing mechanical work at the loss of the pressure and temperture. After expansion in turbine blades, the exhaust is let into atmosphere and the working medium must be replaced continuosly hence, the name, open cycle gas turbine.

Power developed by the turbine is partly utilised in driving the compressor and other accessories and the remaing is used in power generation.

Advantages of open cycle Gas Turbine power plant:-

1 Low weight. The weight in kg per kW developed is less

2 open cycle plant occupy less space.

Any hydro carbon fuel can be used in the combustion chamber of gas turbine plant.

4. The plant does not requires cooling water.

5. Warm up time is considerably less.

Disadvantages of open cycle Gas turbine power plant.

1. plant load efficiency is low

2. The system is sensitive to component efficiency.

3. The efficiency depends on ambient conditions (pressure and temperture

4. The open cycle gas turbine plant has high airrate therefore loss of heat to the ~~external~~ exhaust gases is very high and large diameter duct work is necessary.

* The difference between closed cycle and open cycle gas turbine power plants are tabled in the following.

closed cycle gas turbine powerplant	open cycle gas turbine power plant.
1. In this plant, the working fluid is confined within the plant.	1. In of this plant, the working fluid enters from atmosphere and exhausts to the atmosphere.
2. The air is heated in an air heater by burning fuel exter nally.	2. The working substance (air) is first compressed after which its temperture is raised by burning fuel in it.
3. The working air does not come in contact with the products of combustion	3. The working air comes in contact with the product of combustion.
4. The hot air expands in the turbine itself.	4. The products of combination along with excess air are passed through the turbine.

5. In this case, erosion and corrosion of turbine blades are very much reduced.	5. The hot gases cause erosion and corrosion of turbine blades.
6. After developing power from the turbines, the hot air is cooled in a precooler and supplied back to the compressor.	6. After developing the power from turbine, the hot air is exhausted into the atmosphere.
7. For next cycle, same working fluid circulates over and again in the system.	7. For next cycle, fresh air is taken in the compressor
8. compulsorily requires cooling arrangement	8. Does not Require cooling medium except those which have an intercooler.
9. The closed cycle plant does not depend on the changes in the atmosphere	9. The open cycle plant is sensitive to the changes in the atmosphere i.e. air temperture, pressure and humidity.
10. The disadvantage of limitation of unit rating by the atmospheric back pressure is eliminated in this type of plants.	10. The atmospheric back pressure limits the unit rating.
11. Requires more space and the weight in kg per kW developed is also more.	11 Requires less space and the weight in kg per kW developed is also less.

* Expression for efficiency of brayton cycle:-

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Process involved are as follows,
process 1-2 Isentropic compression
process 2-3 Heat addition at constant pressure
Process 3-4 Isentropic expansion
Process 4-1 Heat rejection at constant pressure.

Let Q_1 be the heat supplied. This occurs during process 2-3 $\therefore Q_1 = mc_p(T_3 - T_2)$

And Q_2 be the heat rejected. This occurs during process 4-1

$$\therefore Q_2 = mc_p(T_4 - T_1)$$

Cycle efficiency.

$$\eta = 1 - \frac{Q_2}{Q_1}$$

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

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

Now, $\frac{T_2}{T_1} = \frac{T_3}{T_4} = \left[\frac{P_2}{P_1}\right]^{\frac{\gamma-1}{\gamma}}$

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

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

Now, Subtracting by 1 on both sides in the above equation, we have,

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

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

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

$$\Longrightarrow \frac{T_4 - T_1}{T_3 - T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{r-1}{\gamma}} \quad \left[\because \frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{r-1}{r}}\right]$$

If r_p = pressure ratio $= \frac{P_2}{P_1}$

$$\eta = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{r-1}{r}}$$

$$\therefore \eta = 1 - \frac{1}{(r_p)^{\frac{r-1}{r}}}$$

*

C_{20}

From above figure, we have,

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

Let $\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = x$

$$\therefore \quad \frac{T_2}{T_1} = \frac{T_3}{T_4} = x$$

The actual work absorbed by the compressor,

$$W_c = (h_2 - h_1) = \frac{(h_2 - h_1)}{\eta_c} = \frac{c_p(T_2 - T_1)}{\eta_c} \text{ kJ/kg}$$

The actual work developed by the turbine,

$$W_T = (h_3 - h_4) = (h_3 - h_4)\eta_T = c_p(T_3 - T_4)\eta_T \text{ kJ/kg}$$

Thus, the net work output is

$$W_{net} = W_T - W_c$$

$$= c_p(T_3 - T_4)\eta_T - \frac{c_p(T_2 - T_1)}{\eta_c}$$

$$= c_p \eta_T T_3\left[1 - \frac{T_4}{T_3}\right] - \frac{c_p T_1\left[\frac{T_2}{T_1} - 1\right]}{\eta_c}$$

$$W_{net} = c_p \eta_T T_3\left(1 - \frac{1}{x}\right) - \frac{c_p T_1 (x-1)}{\eta_c}$$

$$\left[\because x = \frac{T_3}{T_4} = \frac{T_2}{T_1}\right]$$

For maximum specific output, differentiating above equation with respect to 'x' and equating to zero

i.e, $\frac{dW_{net}}{dx} = 0$

$$\Rightarrow c_p \eta_T T_3\left(\frac{1}{x^2}\right) - \frac{c_p T_1}{\eta_c} = 0$$

$$C_p \cdot \eta_T T_3 \left(\frac{1}{x^2}\right) = \frac{C_p T_1}{\eta_c}$$

$$\eta_T \eta_c \cdot \frac{T_3}{T_1} = x^2$$

$$x = \sqrt{\eta_T \cdot \eta_c \cdot \frac{T_3}{T_1}}$$

$$(r_p)^{\frac{\gamma-1}{\gamma}} = \left(\eta_T \cdot \eta_c \cdot \frac{T_3}{T_1}\right)^{1/2}$$

$$\therefore r_p = \left[\eta_T \cdot \eta_c \cdot \frac{T_3}{T_1}\right]^{\frac{1}{2(\gamma-1)}}$$

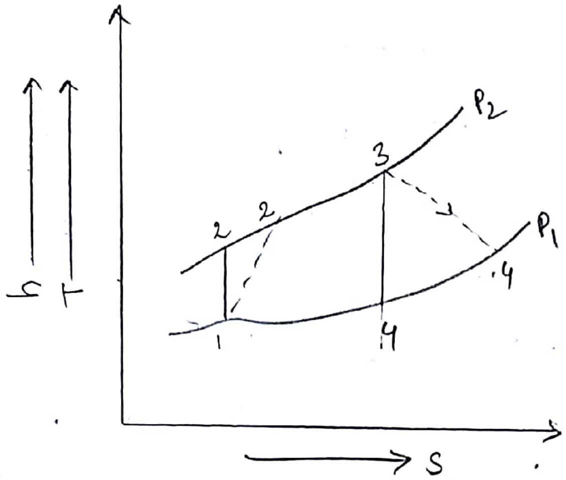


Fig :- Actual Brayton cycle.

The net work output is given as,

$$W_{net} = C_p \eta_T T_3 \left[1 - \frac{1}{x}\right] - \frac{C_p T_1}{\eta_c}(x-1)$$

Heat supplied $= C_p (T_3 - T_{2'}) = C_p (T_3 - T_2)$

$$(\because T_2 = T_{2'}) \quad \ldots (1)$$

Let, $\frac{T_3}{T_4} = \frac{T_2}{T_1} = (r_P)^{\left(\frac{\gamma-1}{\gamma}\right)} = x$ and $\frac{T_3}{T_1} = y \quad \ldots (2)$

Substituting equation (2) in equation (1), we get,

Heat Supplied $= C_p (T_3 - T_1 x) = C_p (T_1 y - T_1 x)$ $\left[\because \frac{T_2}{T_1} = x\right]$

$C_p T_1 (y-x)$

$\therefore$ The actual thermal efficiency of the cycle is

$$\eta_{th} = \frac{\text{Network output}}{\text{Heat Supplied}}$$

$$\eta_{th} = \frac{\eta_T C_p T_3 \left(1 - \frac{1}{x}\right) - \frac{C_p T_1}{\eta_c}(x-1)}{C_p T_1 (y-x)}$$

$$= \frac{C_p T_1 \left[\eta_T \frac{T_3}{T_1}\left(1 - \frac{1}{x}\right) - \frac{1}{\eta_c}(x-1)\right]}{C_p T_1 (y-x)}$$

$$= \frac{\eta_T \frac{T_3}{T_1}\left[1 - \frac{1}{x}\right] - \frac{1}{\eta_c}(x-1)}{(y-x)}$$

$$\eta_{th} = \frac{\eta_T y \left(1 - \frac{1}{x}\right) - (x-1)\frac{1}{\eta_c}}{(y-x)} \qquad \left[\because \frac{T_3}{T_1} = y\right]$$

For maximum thermal efficiency, differentiate the above equation with respect to 'x' and equate to zero.

$$\frac{d\,\eta_{th}}{dx} = 0$$

$$\frac{\left[\eta_T y \left(\frac{1}{x^2}\right) - \frac{1}{\eta_c}(1)\right](y-x) - \left[\eta_T y\left(1 - \frac{1}{x}\right) - (x-1)\frac{1}{\eta_c}\right](-1)}{(y-x)^2} = 0$$

$$\left(\because \frac{d}{dx}\left(\frac{u}{v}\right) = \frac{u'v - v'u}{v^2}\right)$$

$$\Rightarrow \left[\left(\frac{\eta_T (y-x) y}{x^2} - \frac{(y-x)}{\eta_c}\right) + \left(\frac{\eta_T y (x-1)}{x} - \frac{(x-1)}{\eta_c}\right)\right] = 0$$

$$\Rightarrow \left[\frac{n_T n_c y^2 \ldots}{x^2 n_c}\right](y-x) = \left[\frac{n_T n_c\, y - x}{x n_c}\right]$$

$$\Rightarrow \left[\frac{n_c n_T y^2 - n_c n_T xy - x^2 y + x^3}{x^2 n_c}\right] = \left(\frac{-n_c n_T xy + n_c n_T y + x^2 - x}{x n_c}\right)$$

$$\Rightarrow (n_c n_T y^2 - n_c n_T xy - x^2 y + x^3) = (-n_c n_T x^2 y + n_c n_T xy + x^3 - x^2)$$

Multiplying the above equation by $\frac{1}{x^2}$, we have,

$$\left(n_c n_T \frac{y^2}{x^2} - n_c n_T \frac{y}{x} - y + x\right) = \left[-n_c n_T y + n_c n_T \frac{y}{x} + x - 1\right]$$

$$\frac{1}{x^2}(n_c n_T y^2) - 2n_c n_T\left(\frac{y}{x}\right) + [\,y(n_c n_T - 1) + 1] = 0$$

The above equation is in the form of quadratic equation, thus the root will be,

$$\frac{1}{x} = \frac{2n_c n_T y \pm \sqrt{(2n_c n_T y)^2 - 4(n_c n_T y^2)[\,y(n_c n_T - 1) + 1]}}{2n_c n_T y^2}$$

$$x = \frac{2n_c n_T y^2}{2n_c n_T y \pm 2\sqrt{(n_c n_T y)^2 - (n_c n_T y^2)[\,y(n_c n_T - 1) + 1]}}$$

$$x = \frac{n_c n_T y^2}{n_c n_T y \pm \sqrt{(n_c n_T y)^2 - (n_c n_T y)^2 y + n_c n_T y^3 - n_c n_T y^2}}$$

$$x = \frac{n_c n_T y^2}{n_c n_T y\left[1 \pm \sqrt{1 - y + \frac{y}{n_c n_T} - \frac{1}{n_c n_T}}\right]}$$

Considering only positive roots,

$$x = \frac{y}{1 + \sqrt{y\left[\frac{1}{n_c n_T} - 1\right] - \left[\frac{1}{n_c n_T} - 1\right]}}$$

$$x = \frac{y}{1 + \sqrt{(y-1)\left(\frac{1}{n_c n_T} - 1\right)}}$$

Substituting 'x' and 'y' Values in equation (3), we get

$$(r_p)^{\frac{\gamma-1}{\gamma}} = \left[\frac{(T_3/T_1)}{1+\sqrt{\left[\frac{T_3}{T_1}-1\right]\left[\frac{1}{\eta_C\eta_T}-1\right]}}\right]$$

$$(r_p) = \left[\frac{T_3/T_1}{1+\sqrt{(T_3/T_1-1)\left[\frac{1}{\eta_C\cdot\eta_T}-1\right]}}\right]^{\left(\frac{\gamma}{\gamma-1}\right)}$$

* The actual Brayton cycle differs from an ideal Brayton cycle in the following aspects.

Due to frictional effects, a pressure drop occurs in a combustion chamber. However, this pressure drop is significantly less and hence it is neglected for simplicity of analysis.

Also, in ideal cycle, the compressor and turbine efficiencies are assumed to be 100% but in actual cycle the compressor and turbine efficiencies are less than 100%, because of frictional effect within the compressor and turbine.

The specific heat of air is slighty less than the specific heat of combustion gas, However, this increases in specific heat of combustion gas is significantly less and hence it is taken as that of air

As shown in T-s diagram, the process

1-2 is isentropic compression

1-2' is actual compression

2-3 is actual compression

3-4 is Isentropic expansion

The ~~efficiency of compressor, η_c is given as~~

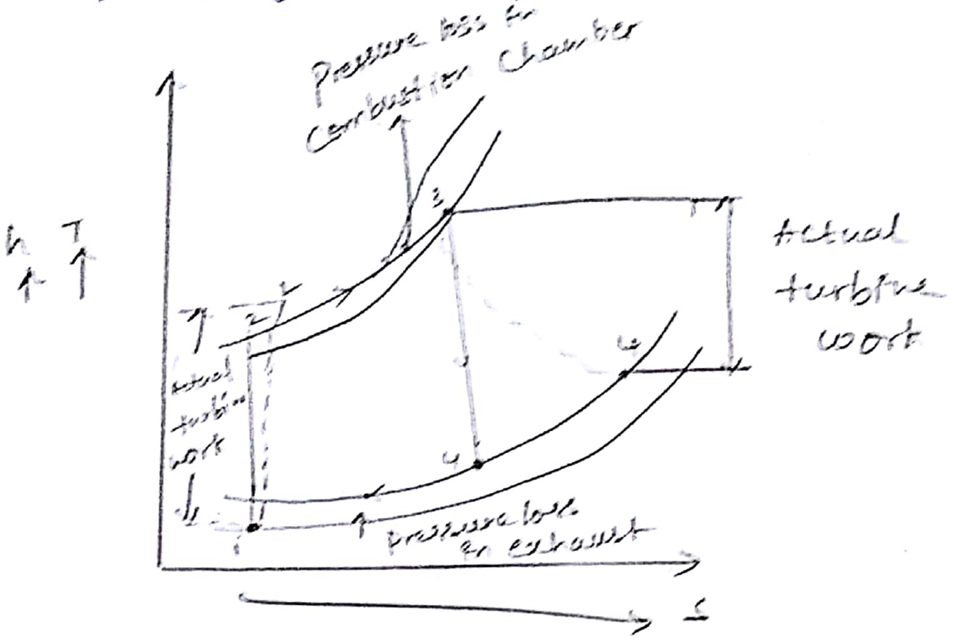


Fig: Actual Brayton cycle

The efficiency of compressor, η_c is given as,

$$\eta_c = \frac{\text{Isentropic Compression Work}}{\text{Actual Compression Work}} = \frac{W_c}{W_{ca}}$$

$$\eta_c = \frac{h_2 - h_1}{h_2 - h_1} = \frac{C_p(T_2 - T_1)}{C_p(T_2 - T_1)}$$

$$\therefore \eta_c = \frac{T_2 - T_1}{T_2 - T_1}$$

Similarly, the efficiency of turbine, η_T is given as

$$\eta_T = \frac{\text{Actual turbine work}}{\text{Isentropic turbine work}} = \frac{W_T}{W_{Ta}}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_4} = \frac{C_p(T_3 - T_4)}{C_p(T_3 - T_4)}$$

$\eta_T = \frac{T_3 - T_4'}{T_3 - T_4}$

$\therefore$ Actual network done = Work done by turbine − Work supplied to compressor.

$$W_{net} = \left[\left(1+\frac{m_f}{m_a}\right)(h_3-h_4)-(h_2-h_1)\right] \text{ kJ/kg of air}$$

where,

$\frac{m_f}{m_a}$ is fuel-air ratio

Also, the actual heat supplied is given as,

$$Q = \left[\left(1+\frac{m_f}{m_a}\right)(h_3-h_2)\right] \text{ kJ/kg of air}$$

$\therefore$ Thermal efficiency is given as,

$$\eta_{th} = \frac{\text{Actual network done}}{\text{Actual heat supplied}}$$

$$= \frac{\left(1+\frac{m_f}{m_a}\right)(h_3-h_4)-(h_2-h_1)}{\left(1+\frac{m_f}{m_a}\right)(c_{pg}T_3 - c_pT_2)}$$

If specific heat of air is assumed to be same as that of combustion gas and also neglecting the mass of fuel supplied in comparison with mass of air, then,

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

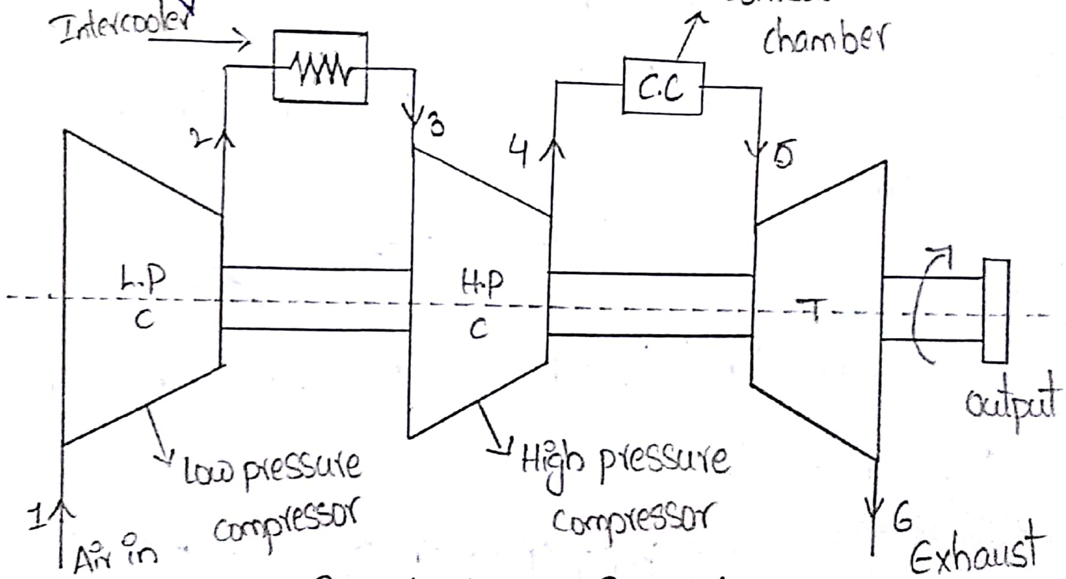
Substituting Equation (1), (2) in Equation (3), we have,

$$\eta_{th} = \frac{\eta_T(T_3-T_4) - (T_2-T_1)/\eta_c}{(T_3-T_2)}$$

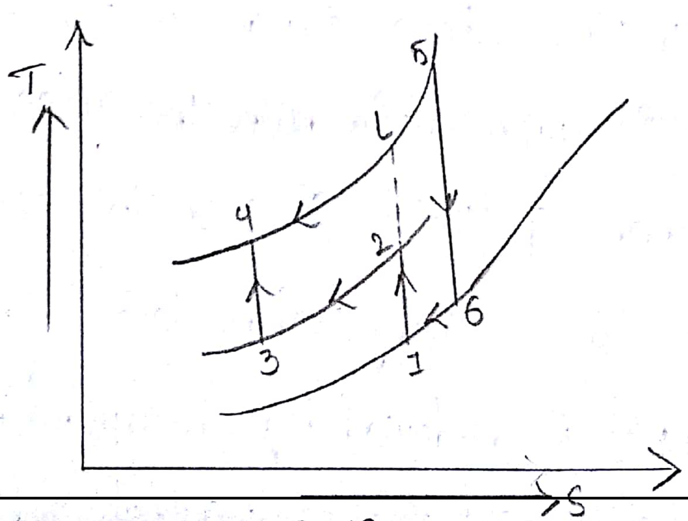
Intercooler has to be placed for efficient specific output and thermal efficiency in a gas turbine:-

The work output and thermal efficiency of a gas turbine can be improved by reducing the compressor work input. It can be done by compressing the air in two stages and incorporating an intercooler between the stages as shown in figure.

Therefore, the process of constant pressure cooling of compressed air between two stages of compressor is known as intercooling.



Gas turbine with intercooler.



T-S diagram

As shown in T-S diagram, the process 1-2 represents an isentropic compression in a low pressure compressor. The Process 2-3 represents cooling of air at constant pressure in the intercooler (heat exchanger) and the process 3-4 represents further isentropic compression in high pressure compressor.

Thus,

Cycle :- 1-2-3-4-5-6 represents gas turbine cycle with intercooler.

Cycle :- 1-L-5-6 represents gas-turbine cycle without intercooler.

Work input to the compressor (with intercooler) per kg of air,

$$= c_p(T_2 - T_1) + c_p(T_4 - T_3)$$

Work input to the compressor (without intercooler) per kg of air.

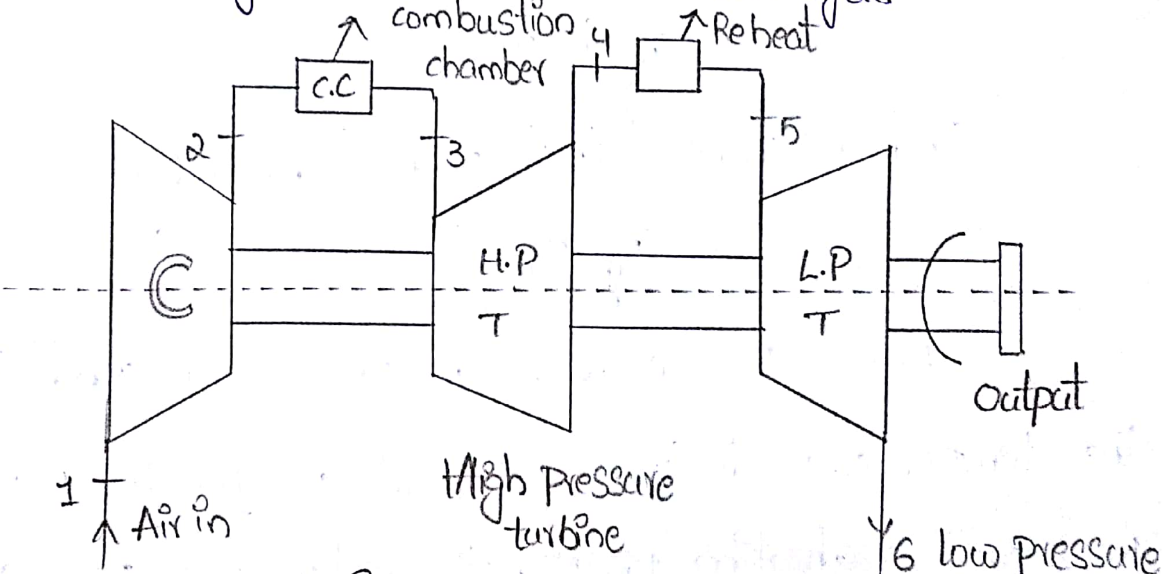
$$= c_p(T_L - T_1) = c_p(T_2 - T_1) + c_p(T_L - T_2)$$

In the above expression for the compressor work $c_p(T_2 - T_1)$ is common for both the cycles. But $c_p(T_L - T_2)$ is greater than $c_p(T_4 - T_3)$, since on the T-S diagram, the pressure lines diverges from left to right. Hence, the work input is reduced by incorporating intercooler which in turn increases the work output and thermal efficiency

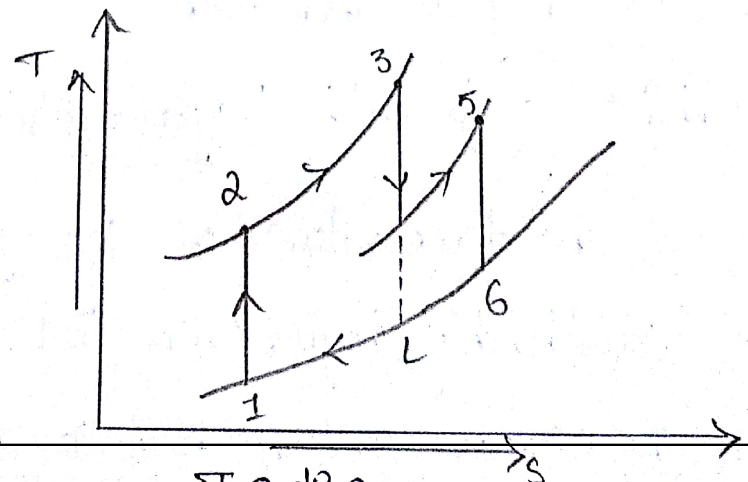
The gas turbine cycle can be modified by Employing "Reheating process":-

The work output and thermal efficiency of a gas turbine can be improved by introducing reheating process during expansion without changing the compressor work (constant pressure) or minimum temperature in a cycle.

Therefore, the constant pressure heating between two turbine expansion is known as reeheating and this modified cycle is known as reheat cycle.



Gas Turbin with Reheat.



T-S diagram

As shown in T-S diagram, the process 1-2-3-4 is similar to simple gas turbine. After Expansion from state '3' to '4' in high pressure turbine, the air is reheated at constant pressure in the reheater i.e., from state 4 to 5. Thus, Expansion take place in low pressure turbine from state '5' to '6'.

cycle : 1-2-3-4-5-6-1 represent the gas turbine cycle with reheater.

cycle : 1-2-3-4-L-1 represents the gas turbine cycle without reheating.

$$= C_{p_g}(T_5 - T_6)$$

And net work output without reheating

$$= C_{p_g}(T_4 - T_L)$$

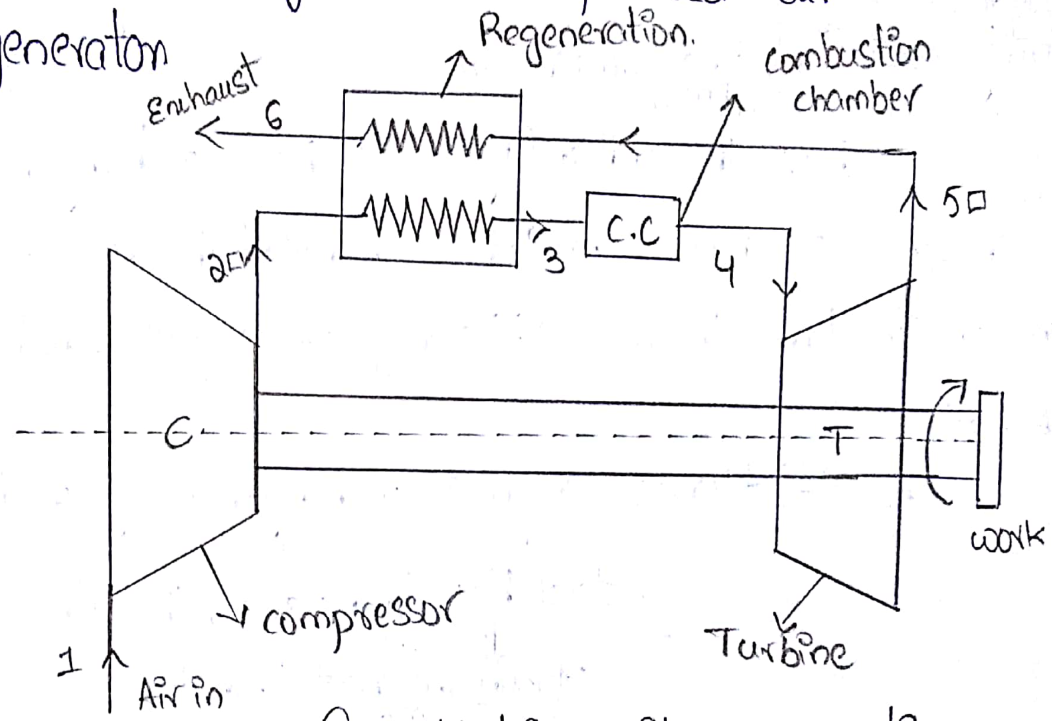
From T-S diagram it can be concluded that the temperature difference $(T_5 - T_6)$ is greater than $(T_4 - T_L)$. Thus, with reaheating process, the net work output and thermal efficiency is increased

<u>Gas Turbine cycle with Regeneration</u>:-

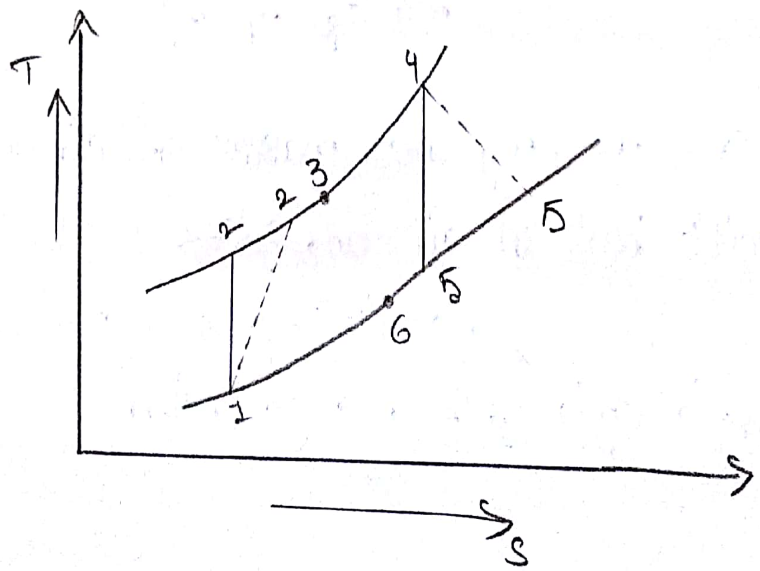
The temperature of the gases leaving the turbine is considerably higher than that of air coming out from the compressor. The heat of Exhaust gases

… to preheat the air coming out from the compressor, thus reducing the mass of fuel supplied in the combustion chamber.

A counter flow heat exchanger which transfer heat from hot gases to compressed air is known as regeneration.



Gas Turbine with Regeneration.



T-S Diagram.

As shown in T-S diagram, the air coming out from the compressor at state-2' enters the regenerator (heat exchanger). In the heat exchanger, the air is heated thus, the temperature is increased to the state-3. The gases leaving the turbine at state "5" are used to pre-heat the air coming out from the compressor.

Thus the advantages of heat exchanger is that the amount of heat supplied is reduced in the combustion chamber, in turn thermal effeciency of a gas turbine cycle increases.

The effectiveness of heat exchanger is given by

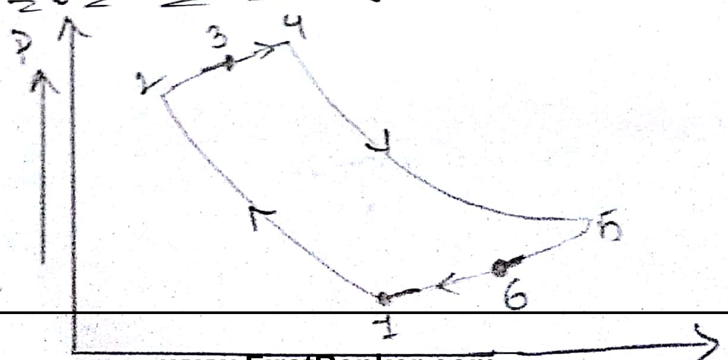
$$\epsilon = \frac{\text{Actual heat transfer to the air}}{\text{Maximum heat transfer.}}$$

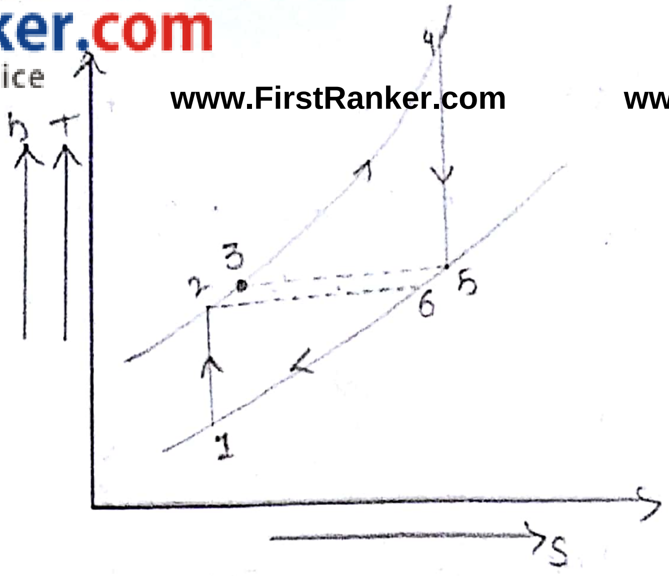
$$\epsilon = \frac{m_a C_{P_a} (T_3 - T_2')}{m_g C_{P_g} (T_5' - T_2')}$$

where, m_a, m_g are masses of air and gases, C_{P_a}, C_{P_g} are specific heat of air and gases.

The expression for specific work output and the efficiency of a simple cycle with a regenerator

P-V diagram





T-S Diagram.

Consider 1 kg of air is flowing through the compressor in a cycle with a regenerator and let the specific constant of air and gases be same.

$\therefore$ From p-v and T-s diagrams,

The work done by compressor, $\omega_c = h_2 - h_1$

$$\omega_{1-2} = C_p (T_2 - T_1) \ \mathrm{kJ/kg}$$

Heat supplied,

$$Q_{34} = (h_4 - h_3) = C_p (T_4 - T_3) = C_p (T_4 - T_5) \quad (\because T_3 = T_5)$$

The work done by turbine,

$$\omega_T = (h_4 - h_5)$$

$$\omega_{4-5} = C_p (T_4 - T_5) \ \mathrm{kJ/kg}$$

Net work done (Specific work output) =

Turbine work − compressor work.

$$\omega_N = C_p (T_4 - T_5) - C_p (T_2 - T_1)$$

∴ The Expression for efficiency, $\eta = \frac{\text{Net workdone}}{\text{Heat supplied}} = \frac{W_N}{Q_1}$

$$\eta = \frac{C_p(T_4-T_5) - C_p(T_2-T_1)}{C_p(T_4-T_5)}$$

$$\eta = 1 - \frac{(T_2-T_1)}{(T_4-T_5)} \longrightarrow (1)$$

But, from isentropic compression and expansion, we have,

$$\left[\frac{T_2}{T_1}\right] = \left[\frac{P_2}{P_1}\right]^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} \text{ and}$$

$$\frac{T_4}{T_5} = \left(\frac{P_4}{P_5}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore \frac{T_4}{T_5} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

So,

$$\therefore \frac{T_2}{T_1} = \frac{T_4}{T_5} = (r_p)^{\frac{\gamma-1}{\gamma}} \text{ (or)}$$

$$\frac{T_2}{T_4} = \frac{T_1}{T_5} = (r_p)^{\frac{\gamma-1}{\gamma}} \longrightarrow (2)$$

we know that,

$$\Rightarrow \frac{T_2-T_1}{T_4-T_5} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$(T_2-T_1) = (r_p)^{\frac{\gamma-1}{\gamma}} \cdot (T_4-T_5) \longrightarrow (3)$$

Substituting equation (3) in equation (1), we get,

$$\eta = 1 - \frac{(r_p)^{\frac{\gamma-1}{\gamma}} \cdot (T_4-T_5)}{(T_4-T_5)}$$

60

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

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

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

$$\therefore \eta = 1 - \frac{T_1}{T_4} \times (r_p)^{\frac{\gamma-1}{\gamma}} \quad \left[\because \frac{T_2}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}\right]$$

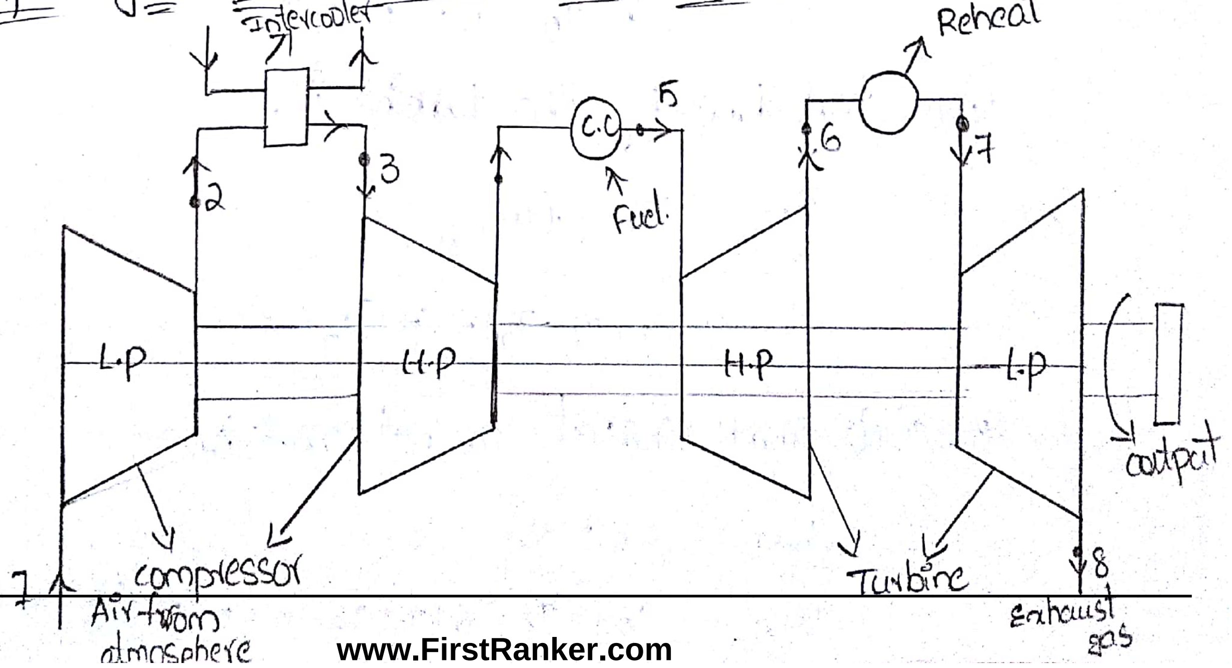
Let, $t = \frac{T_4}{T_1} = \frac{T_{max}}{T_{min}}$ and $c = \frac{T_2}{T_1} = \frac{T_4}{T_5} = (r_p)^{\frac{\gamma-1}{\gamma}}$

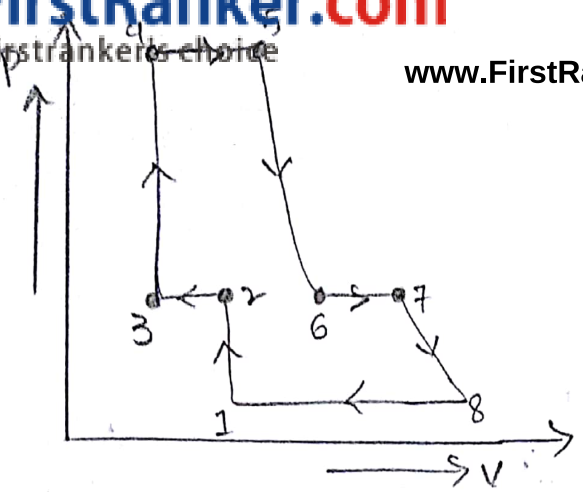
- Thus, the efficiency becomes,

$$\eta = 1 - \frac{c}{t}$$

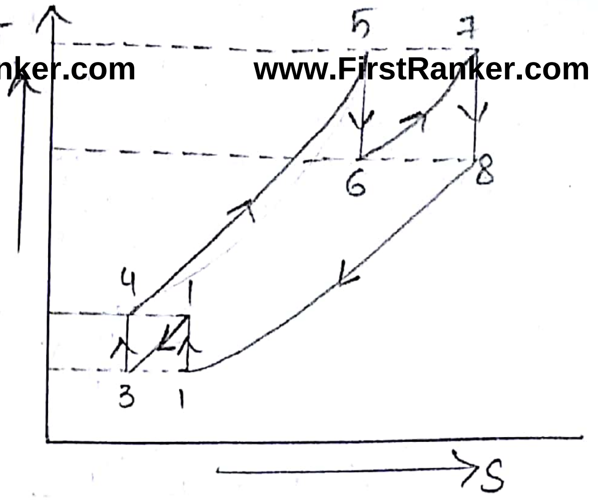
From above equation, it can be concluded that efficiency of a regenerator increases with 't' for a fixed value of 'c'.

- The equation for specific work output and efficiency of simple cycle with intercooled and reheat.





P-V Diagram



T-S Diagram.

Consider a unit quantity of air is flowing through a compressor

From the above figures,

The workdone by the compressor is given as,

$$\omega_C = \omega_{1-2} + \omega_{3-4}$$

$$= C_p(T_2 - T_1) + C_p(T_4 - T_3)$$

Heat supplied,

$$Q = Q_{4-5} + Q_{6-7}$$

$$= C_p(T_5 - T_4) + C_p(T_7 - T_6)$$

The work done by the turbine is,

$$\omega_T = \omega_{5-6} + \omega_{7-8}$$

$$= C_p(T_5 - T_6) + C_p(T_7 - T_8)$$

Specific work output or net workdone,

$$\omega_N = \omega_T - \omega_C$$

$$\omega_N = C_p\left[(T_5 - T_6) + (T_7 - T_8)\right] - C_p\left[(T_2 - T_1) + (T_4 - T_3)\right]$$

Multiply and divide above equation by T_1, we have,

$$w_N = c_p T_1 \left[\frac{T_5}{T_1} - \frac{T_6}{T_1} + \frac{T_7}{T_1} - \frac{T_8}{T_1} - \frac{T_2}{T_1} + 1 - \frac{T_4}{T_1} + \frac{T_3}{T_1} \right]$$

The above equation can also be written as,

$$\frac{W_N}{c_p T_1} = \frac{T_5}{T_1} - \frac{T_6}{T_5} \times \frac{T_5}{T_1} + \frac{T_7}{T_1} - \frac{T_8}{T_5} \times \frac{T_5}{T_1} - \frac{T_2}{T_1} + 1 - \frac{T_4}{T_1} + \frac{T_3}{T_1}$$

As shown in T-S diagram,

$$T_1 = T_3,\ T_2 = T_4,\ T_6 = T_8 \text{ and } T_5 = T_7.$$

$$\therefore \frac{w_N}{c_p T_1} = \frac{T_5}{T_1} - \frac{T_6}{T_5} \times \frac{T_5}{T_1} + \frac{T_5}{T_1} - \frac{T_8}{T_7} \times \frac{T_5}{T_1} - \frac{T_2}{T_1} + 1 - \frac{T_4}{T_3} + \frac{T}{T}$$

let,

$$\frac{T_5}{T_1} = t \text{ and } \frac{T_5}{T_6} = \frac{T_7}{T_8} = \frac{T_2}{T_1} = \frac{T_4}{T_3} = \sqrt{c}$$

$$\frac{T_5}{\quad} \quad \frac{w_N}{c_p T_1} = t - \frac{t}{\sqrt{c}} + t - \frac{t}{\sqrt{c}} - \sqrt{c} + 1 - \sqrt{c} + 1$$

$$\frac{w_N}{c_p T_1} = 2t - \frac{2t}{\sqrt{c}} - 2\sqrt{c} + 2$$

For maximum power output and perfect intercooling, we have

$$\frac{w_{max}}{c_p T_1} = 2\left(t - \frac{t}{\sqrt{c}} - \sqrt{c} + 1\right)$$

$$\boxed{\therefore w_{max} = 2 c_p \cdot T_1 \left[t - \frac{t}{\sqrt{c}} - \sqrt{c} + 1 \right]}$$

$$Q = C_p(T_5 - T_4) + C_p(T_7 - T_6)$$

$$= C_p(T_5 - T_4 + T_7 - T_6)$$

Multiplying and dividing by 'T_1', we get,

$$Q = C_pT_1\left[\frac{T_5}{T_1} - \frac{T_4}{T_1} + \frac{T_7}{T_1} - \frac{T_6}{T_1}\right]$$

$$= C_pT_1\left[\frac{T_5}{T_1} - \frac{T_2}{T_1} + \frac{T_5}{T_1} - \frac{T_6}{T_5} \times \frac{T_5}{T_1}\right]$$

$$= C_pT_1\left[t - \sqrt{c} + t - \frac{t}{\sqrt{c}}\right]$$

$$= C_p \cdot T_1\left[2t - \sqrt{c} - \frac{t}{\sqrt{c}}\right]$$

∴ The maximum efficiency of cycle is given as,

$$\eta_{max} = \frac{W_{max}}{Q}$$

$$= \frac{2C_pT_1\left[t - \frac{1}{\sqrt{c}} - \sqrt{c} + 1\right]}{C_p \cdot T_1\left[2t - \sqrt{c} - \frac{t}{\sqrt{c}}\right]}$$

$$= \frac{2\left[t - \frac{1}{\sqrt{c}} - \sqrt{c} + 1\right]}{\left(2t - \sqrt{c} - \frac{t}{\sqrt{c}}\right)}$$

$$= \frac{2(t - \sqrt{c}) - 2\left(\frac{t}{\sqrt{c}} - 1\right)}{\left(2t - \sqrt{c} - \frac{t}{\sqrt{c}}\right)}$$

$$= \frac{\left(2t - \sqrt{c} - \frac{t}{\sqrt{c}}\right) - \left(\frac{t}{\sqrt{c}} + \sqrt{c} - 2\right)}{\left(2t - \sqrt{c} - \frac{t}{\sqrt{c}}\right)}$$

$$\therefore \eta_{max} = 1 - \frac{\frac{t}{\sqrt{c}} + \sqrt{c} - 2}{2t - \sqrt{c} - \frac{t}{\sqrt{c}}}$$

Com Problems:

Compare the maximum work delivered by an aircraft gas turbine which works with two stage compression with intercooling the compressor pressure ratio is 4 and the temperature limit is 1000k, for the given ambient condition 1 bar and 301k. If the temperature and pressure at 6000m altitude is -25°C and 0.5 bar, find the percentage change in network output, efficiency and exhaust gas temperature if the volume flow rate is 2.5 m^3/s.

Two inlet conditions have been given, therefore we solve the above problem taking one case at a time.

Case 1:

Given that,

Inlet temperature of air, $T_1 = 301k$.

Inlet pressure of air, $P_1 = 1 bar$.

Compression takes place in two stages with intercooling

For a perfect inter cooling process,

$$T_1 = T_3 = 301k$$

$$T_2 = T_4$$

$$T_3 = 1000k.$$

Also, given $\frac{P_2}{P_1} = \frac{P_3}{P_2} = 4$

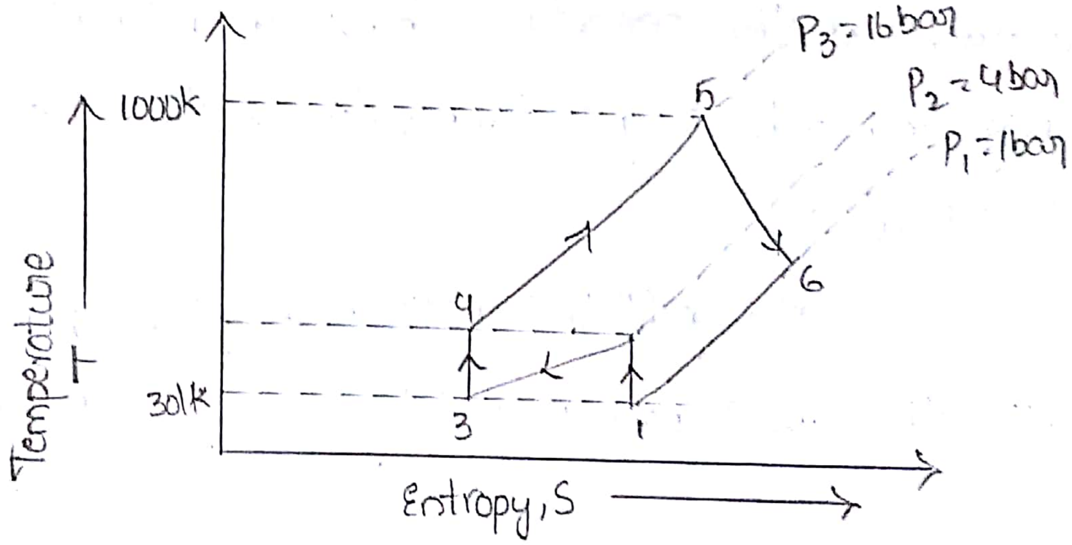
$$\frac{P_2}{1} = 4$$

$$\Rightarrow P_2 = 4 \text{ bar}$$

Also,

$$\frac{P_3}{4} = 4$$

$$\Rightarrow P_3 = 16 \text{ bar}$$



From (1-2) process, we have,

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

$$\Rightarrow T_2 = 301 \times (4)^{\frac{0.4}{1.4}}$$

$= 447.28K$

$$T_2 = T_4 = 447.28k$$

Also from process (5-6), we have,

$$\frac{T_6}{T_5} = \left(\frac{P_6}{P_5}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= \left(\frac{P_1}{P_3}\right)^{\frac{\gamma-1}{\gamma}} \quad [\because P_1 = P_6 \text{ and } P_3 = P_5]$$

$$\Rightarrow \quad T_6 = 1000 \times \left(\frac{1}{16}\right)^{\frac{0.4}{1.4}}$$

$$= 452.86k$$

work done by turbine,

$$w_T = C_p (T_5 - T_6)$$

$$= 1 \times (1000 - 452.86) \quad [\because C_p = 1kJ/kg\text{-}K]$$

$$\Rightarrow \quad w_T = 547.14 \text{ kJ/kg}$$

work absorbed by compressor,

$$w_C = C_p [(T_2 - T_1) + (T_4 - T_3)]$$

$$= [(447.28 - 300) + (447.28 - 300)] \quad [\because T_4 = T_2 \text{ and } T_3 = T_1]$$

$$= 292.56 \text{ kJ/kg}$$

Network,

$$w_1 = w_T - w_C$$

$$= 547.14 - 292.56$$

$$= 254.58 \text{ kJ/kg}$$

Efficiency

$$\eta_1 = \frac{\text{workdone}}{\text{Heat Supplied}}$$

$$= \frac{\omega_1}{c_p(T_5 - T_4)}$$

$$= \frac{254.58}{1 \times (1000 - 447.28)}$$

$$= 0.46 = 46\%$$

Exhaust gas temperature, $T_6 = 452.86\,k.$

Case-II:

$T_1 = -25C = -25 + 273 = 248k = T_3$

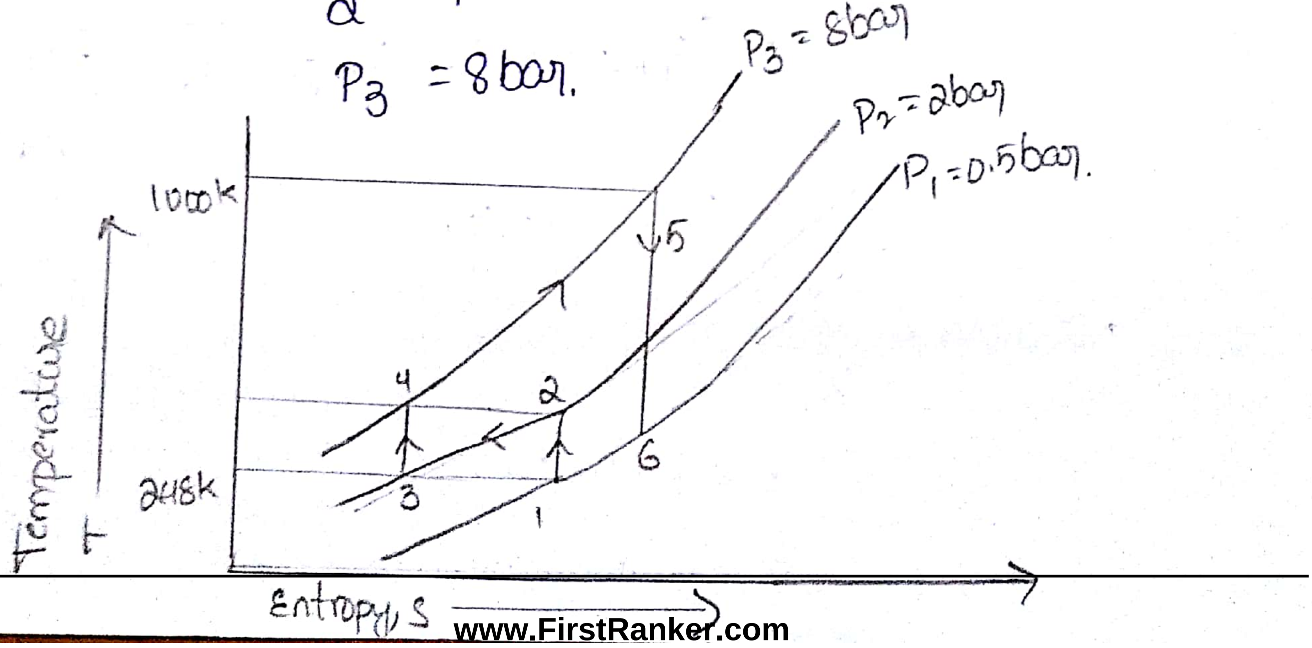
$P_1 = 0.5\,bar$, $\frac{P_2}{P_1} = \frac{P_3}{P_2} = 4$

$$\Rightarrow \frac{P_2}{0.5} = 4$$

$$P_2 = 2\,bar$$

$$\frac{P_3}{2} = 4$$

$$P_3 = 8\,bar.$$



From process (1-2), we have,

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

$$\Rightarrow T_2 = 248 \times (4)^{\frac{0.4}{1.4}}$$

$$= 368.52\text{k}$$

$$T_2 = T_4 = 368.53\text{k}$$

From process (5-6), we get,

$$\frac{T_6}{T_5} = \left(\frac{P_6}{P_5}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_6}{T_5} = \left(\frac{0.5}{8}\right)^{\frac{0.4}{1.4}} \quad [\because P_6 = P_1 \text{ and } P_5 = P_3]$$

$$\Rightarrow T_6 = 1000 \times 0.45 = 450\text{k}$$

work done by turbine,

$$\omega_T = c_p \cdot (T_5 - T_6)$$

$$= 1 \times (1000 - 450) \quad [\because c_p = 1\text{kJ/kgk}]$$

$$= 500 \text{ kJ/kg}$$

work absorbed by the compressor,

$$\omega_c = c_p\left[(T_2 - T_1) + (T_4 - T_3)\right]$$

$$= 1 \times \left[(368.53 - 248) + (368.53 - 248)\right]$$

$$= 241.06 \text{ kJ/kg}$$

Net workdone,

$$w_2 = w_T - w_C$$

$$= 550 - 241.06$$

$$= 308.94 \text{ kJ/kg.}$$

Efficiency, $\eta_2 = \dfrac{\text{work done}}{\text{Heat supplied}}$

$$= \frac{w_2}{C_p(T_5 - T_4)}$$

$$= \frac{308.94}{1 \times (1000 - 368.53)}$$

$$= 0.49 = 49\%$$

Percentage change in Network output.

$$= \frac{w_2 - w_1}{w_1} \times 100$$

$$= \frac{308.94 - 254.58}{254.58} \times 100$$

$$= 21.35\%$$

Network output increases by 21.35%

Percentage change in efficiency

$$= \frac{\eta_2 - \eta_1}{\eta_1} \times 100$$

$$= \frac{49 - 46}{46} \times 100 = 6.52\%$$

Efficiency increases by 6.52 %

Exhaust gas temperature in both the cases is same and its value is 952.86K.

In a gas turbine cycle, the air is compressed in a single stage compressor from 1 bar to 9 bar with initial temperature of 300K. The same air is then heated to a temperature of 800K and then expanded in the turbine. The air is then reheated to a temperature of 800K and then expanded in second turbine. calculate maximum power, if mass of air circulated per second is 2kg. [Take $c_p = 1$ kJ/kg.K]

Given that,

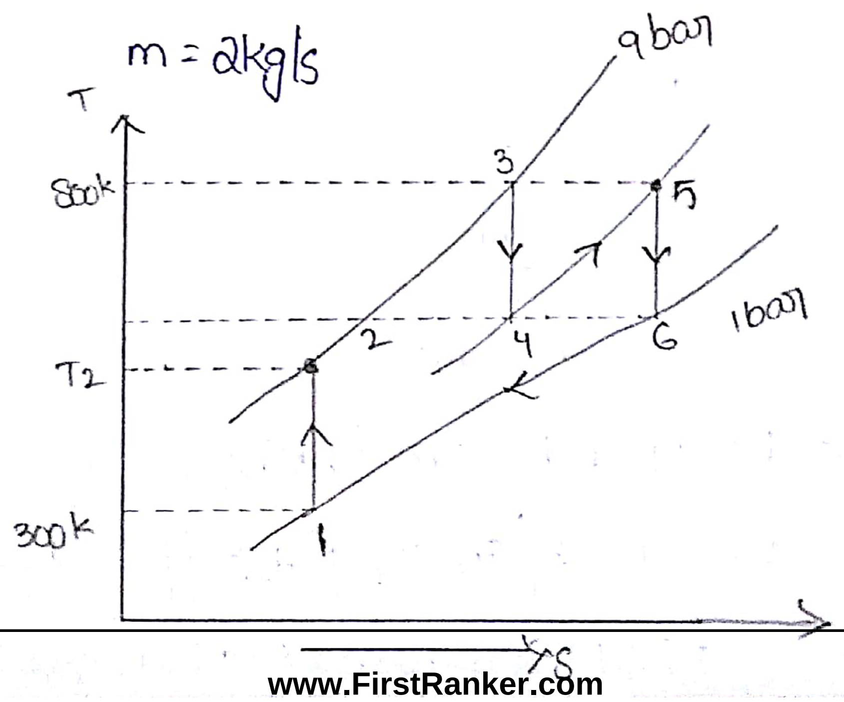
$P_1 = P_6 = 1$ bar

$P_2 = P_3 = 9$ bar

$T_1 = 300K$

$T_3 = T_5 = 800K$

$m = 2kg/s$



For maximum power, the intermediate pressure is given as,

$$P_4 = P_5 = \sqrt{P_1 P_2} = \sqrt{1 \times 9} = 3\ bar$$

Considering isentropic compression process (1-2),

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{9}{1}\right)^{\frac{1.4-1}{1.4}} = (9)^{0.286} = 1.875$$

$$T_2 = 1.875 \times 300 = 562.5 k.$$

Similarly, considering isentropic expansion process (3-4) in first turbine,

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{9}{3}\right)^{\frac{1.4-1}{1.4}} = (3)^{0.286} = 1.37.$$

$$T_4 = \frac{T_3}{1.37} = \frac{800}{1.37} = 583.94 k.$$

For isentropic expansion process (5-6) in second turbine,

$$\frac{T_5}{T_6} = \left(\frac{P_5}{P_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3}{1}\right)^{\frac{1.4-1}{1.4}} = (3)^{0.286} = 1.369$$

$$T_6 = \frac{T_5}{1.37}$$

$$= \frac{800}{1.37}$$

$$= 583.94 k.$$

$\therefore$ work developed by the turbine,

$$W_T = m.C_p[(T_3 - T_4) + (T_5 - T_6)]$$

$$= 2 \times 1[(800 - 583.94) + (800 - 584.368)]$$

$\omega_T = 864.24$ kJ/s

And work required by the compressor,

$$\omega_c = m.c_p (T_2 - T_1)$$

$$= 2 \times 1 (562.5 - 300)$$

$$\omega_c = 525 \text{ kJ/kg}$$

∴ Net work done by the turbine,

$$\omega = \omega_T - \omega_c = 864.24 - 525 = 339.24 \text{ kJ/kg}$$

Therefore, the power developed by the turbine,

$$P = 339.24 \text{ kW}.$$

In a regenerator gas turbine cycles, air enters the compressor at a temperature of 30°C and pressure of 1.5 bar and discharges at 220°C and 5.2 bar. After passing through the regenerators, the air temperature is 395°C. The temperature of air entering and leaving gas turbine are 900°C and 510°C. Assuming no pressure drop through the regenerator. Determine,

(a) The output per kg of air

(b) efficiency of cycle and

(c) work required to drive the compressor

Given that;

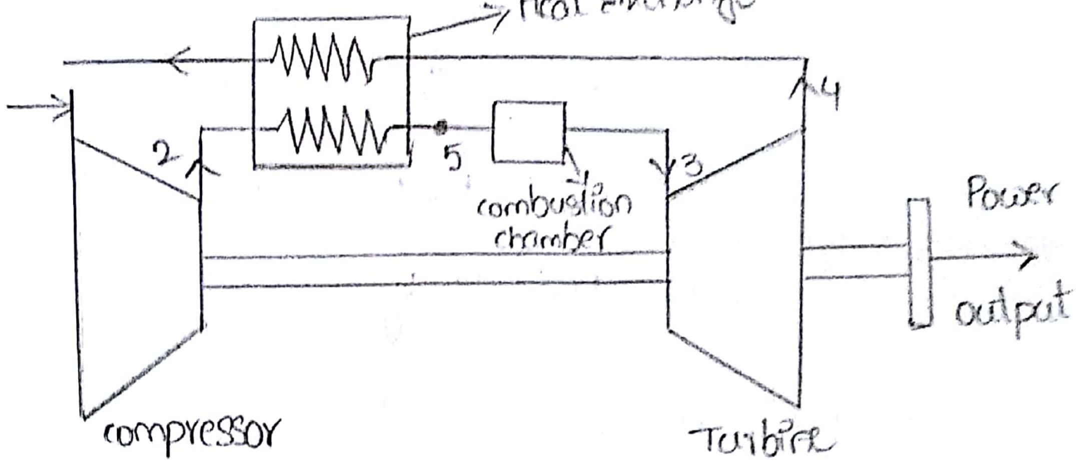
$T_1 = 30°C = 303K$, $P_1 = 1.5$ bar.

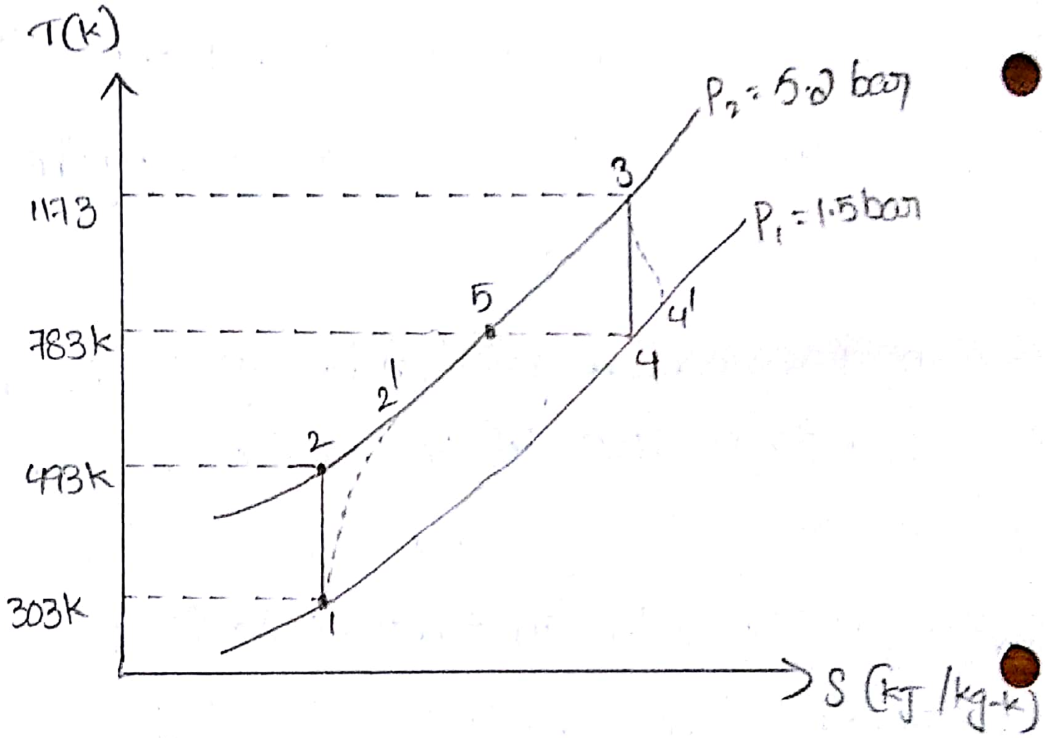
$T_2 = 220°C = 493K$, $P_2 = 5.2$ bar.

$T_3 = 900°C = 1173K$.

$T_4 = 510°C = 783k$

$T_5 = 395°C = 668k.$





work done by compressor,

$w_c = c_p (T_2 - T_1)$

$= 1.005 \times (493 - 303)$

$= 190.95 \ kJ/kg$

workdone by turbine,

$w_T = c_p (T_3 - T_4)$

$= 1.005 \times (1173 - 783)$

$= 391.95 \ kJ/kg$

work done by turbine,

$$W_T = C_p(T_3 - T_4)$$
$$= 1.005 \times (1173 - 783)$$
$$= 391.95 \text{ kJ/kg.}$$

Heat Added 'Q'

$$= C_p(T_3 - T_5)$$
$$= 1.005 \times (1173 - 668)$$
$$= 507.525 \text{ kJ/kg.}$$

(a) output per kg of Air:-

Network done = Turbine work − compressor work

$$W_N = W_T - W_C$$
$$= 391.95 - 190.95$$
$$= 201 \text{ kJ/kg.}$$

(b) Efficiency of cycle

Efficiency of cycle,

$$\eta = \frac{\text{work output}}{\text{Heat Supplied}}$$

$$\eta = \frac{201}{507.525} = 0.396$$

$$\eta = 39.6\ \%$$

(c) work required to drive the compressor,

$$= 190.95 \text{ kJ/kg.}$$

A closed gas turbin with reheat operates with helium as working medium. The pressure ratio is 10. The maximum permitted temperature is 1000k. Assuming the work output to be maximum. calculate the efficiency. If air is used instead of the helium, calculate the efficiency and difference in heat added Assuming ideal Braylon cycle, consider the temperature at the inlet of compressor as 27°C. cp of the helium = 5.204 KJ/kg-k and γ for helium is 1.617.

Given that,

Pressure ratio, $r_p = 10$

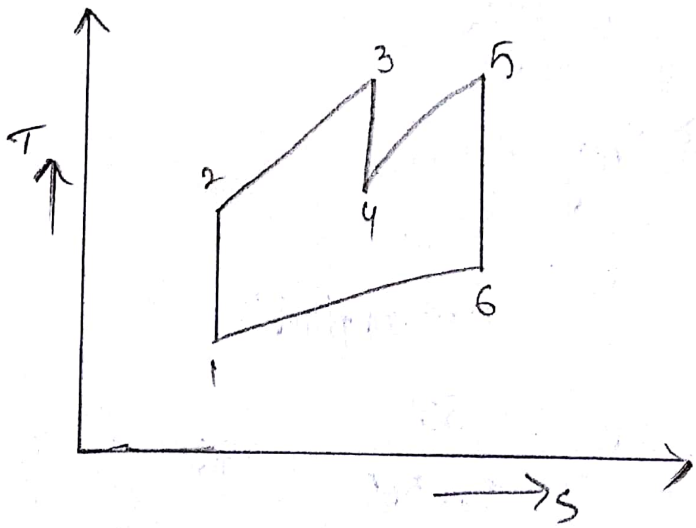
maximum permitted temperature, $T_3 = 1000k$

Inlet temperature of compressor,

$T_1 = 27°C = 27+273 = 300k$

Cp of helium = 5.204 kJ/kg-k

γ for helium = 1.617



we know that the efficiency of reheat cycle for maximum work output is given by

$$\eta_{max} = \frac{2t\left(1-\frac{1}{\sqrt{c}}\right) - (c-1)}{2t - c - \frac{t}{\sqrt{c}}}$$

Here, $t = \frac{T_3}{T_1} = \frac{1000}{300} = 3.333$

and $c = (r_p)^{\frac{\gamma-1}{\gamma}} = (10)^{\frac{1.617-1}{1.617}} = 2.408$

$$\Rightarrow \eta_{max} = \frac{2 \times 3.333 \times \left(1 - \frac{1}{\sqrt{2.408}}\right) - (2.408-1)}{2 \times 3.333 - 2.408 - \frac{3.333}{\sqrt{2.408}}}$$

$$= \frac{0.962}{2.11}$$

$$= 0.456$$

$\therefore$ Efficiency using helium, $\eta_{max} = 45.6\%$

we also know that heat added during the reheat cycle is given by,

$$Q_1 = C_p T_1 \left[2t - c - \frac{t}{\sqrt{c}}\right]$$

$$\Rightarrow Q_1 = 5.204 \times 300 \left[2 \times 3.333 - 2.408 - \frac{3.333}{\sqrt{2.408}}\right]$$

$$= 329.342 \text{ kJ/kg}$$

Now let us consider the case when air is used instead of helium,

For air we know that, $\gamma = 1.4$

$C_p = 1.005$ kJ/kg-K

we know that

$$\text{Efficiency}, \eta_{max} = \frac{2t\left(1-\frac{1}{\sqrt{c}}\right)-(c-1)}{2t-c-\frac{t}{\sqrt{c}}}$$

Here, $t = \frac{T_3}{T_1} = \frac{1000}{300} = 3.333$

$$c = (r_p)^{\frac{\gamma-1}{\gamma}} = (10)^{\frac{1.4-1}{1.4}} = (10)^{0.286} = 1.932$$

$$\eta_{max} = \frac{2\times3.333\left(1-\frac{1}{\sqrt{1.932}}\right)-(1.932-1)}{2\times3.333-1.932-\frac{3.333}{\sqrt{1.932}}}$$

$$= \frac{0.938}{2.336}$$

$$= 0.402.$$

$\therefore$ Efficiency using air, $\eta_{max} = 40.2\%$

And also heat added, $Q_2 = c_p T_1\left[2t-c-\frac{t}{\sqrt{c}}\right]$

$$\Rightarrow Q_2 = 1.005\times300\left(2\times3.333-1.932-\frac{3.333}{\sqrt{1.932}}\right)$$

$$= 704.333 \text{ kJ/kg}$$

Difference in heat added, $\Delta Q = Q_1 - Q_2$

$$\Delta Q = 3294.342 - 704.333$$

$$\Delta Q = 2590.009 \text{ kJ/kg}.$$

...cycle gas turbine there is two stage compressor

and a two stage turbine. All components are mounted on the same shaft. The pressure at the inlet of the first stage compressor are 1.5 bar and 20°c. The maximum cycle temperature and pressure are limited to 750°c and 6 bar. A perfect intercooler is used between the two stage compressor and heater is used between the two turbines. Gases are heated in a heater to 750°c before entering into the LP turbine. The compressor and turbine efficiencies are 0.82 calculate,

(i) The Efficiency of the cycle without regenerator

(ii) The Efficiency of the cycle with a regenerator whose effectiveness is 0.70 Take $\gamma = 1.4$ and $c_p = 1.005$ kJ/kg-k for working fluid.

Given that,

Temperature in inlet,

$T_1 = 20°c = 20 + 273$

$T_1 = 293k.$

maximum cycle temperature,

$T_5 = T_7 = 750°c = 750 + 273$

$T_5 = T_7 = 1023k$

Pressure at inlet of first stage compressor,

$P_1 = 1.5$ bar

maximum cycle pressure $P_2 = 6$ bar

Efficiencies of compressor and turbine

$\eta_C = \eta_T = 0.82$

The given effectiveness of regenerator,

$\epsilon = 0.70$

The working fluid of the compressor is air,

$\gamma = 1.4$ and $C_p = 1005$ kJ/kgK

where, $\frac{T_2}{T_1} = \left(\frac{P_x}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$ and $P_x = \sqrt{P_1 P_2}$

$= \sqrt{1.5 \times 6}$

$P_x = 3 bar$

Now, $T_2 = T_1 \times \left(\frac{P_x}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$

$T_2 = 293 \times \left(\frac{3}{1.5}\right)^{\frac{1.4-1}{1.4}}$

$= 293 \times 1.219$

$T_2 \cong 357 k$

At low pressure, efficiency of compressor.

$\eta_{c(L.P)} = \frac{T_2 - T_1}{T_2' - T_1}$

$$0.82 = \frac{357-293}{T_2'-293}$$

$$T_2' = \frac{357-293}{0.82} + 293$$

$$\therefore T_2' \cong 371k$$

$$T_2' = T_4' = 371k$$

NOW,

$$\frac{T_5}{T_6} = \left[\frac{P_5}{P_6}\right]^{\frac{\gamma-1}{\gamma}} = \left[\frac{P_2}{P_x}\right]^{\frac{\gamma-1}{\gamma}} \quad [\because P_5 = P_2, P_6 = P_x].$$

$$\frac{1023}{T_6} = \left[\frac{6}{3}\right]^{0.286} = 1.219 \quad [\because T_5 = 1023k]$$

$$\frac{1023}{T_6} = 1.219$$

$$\therefore T_6 = \frac{1023}{1.219}$$

$$\cong 839k.$$

At high pressure, efficiency of turbine,

$$\eta_{T(H.P)} = \frac{T_5 - T_6'}{T_5 - T_6}$$

$$0.82 = \frac{1023 - T_6'}{1023 - 839}$$

$$= \frac{1023 - T_6'}{184}$$

$$T_6' = 1023 - 0.82\,(184)$$

$T_6' = 872k$

efficiency of turbine at high pressure and low pressure is same, because of the same temperature at that point.

$$\therefore T_6' = T_8' = 872k \qquad [\because \eta_{T(H.P)} = \eta_{T(L.P)}]$$

where temperature, $T_7 = T_5 = 1023k$

the effectiveness of regenerator, $\epsilon = \dfrac{T_1' - T_4'}{T_8' - T_4}$

where

T_1' - Temperature of air coming out of regenerator.

$$\epsilon = \frac{T_1' - T_4'}{T_8' - T_4'}$$

$$0.70 = \frac{T_1' - 371}{872 - 371}$$

$$T' = 0.70(872 - 371) + 371$$

$$T' = 0.70(501) + 371 = 350.7 + 371$$

$$T' = 721k$$

Net work available,

$$W_{net} = [W_{T(H.P)} + W_{T(L.P)}] - [W_{C(H.P)} + W_{C(L.P)}]$$

$$W_{net} = 2[W_{T(L.P)} + W_{C(L.P)}]$$

As the work developed by each turbine is sam and work absorbed by each compressor is same,

$$W_{net} = 2Cp[(T_5 - T_6) - (T_2' - T_1)]$$

$$= 2 \times 1.005[(1023 - 872) - (371 - 293)]$$

$= 2 \times 1 \times 73$

$\therefore w_{net} = 146.73$ kJ/kg

$\because$ Heat supplied per kg of air without regenerator,

$$Q_{wo} = c_p[(T_5 - T_4') + c_p[T_7 - T_6']]$$

$$= 1.005[(1023 - 371) + (1023 - 872)]$$

$$= 1.005(803)$$

$$Q_{wo} = 807 \text{ kJ/kg}$$

Heat supplied per kg of air with regenerator,

$$Q_w = c_p(T_5 - T') + c_p(T_7 - T_6')$$

$$= 1.005[(1023 - 721) + (1023 - 872)]$$

$$= 1.005(453)$$

$$Q_w = 455.265 \text{ kJ/kg}$$

Therefore,

(i) η_{th}(without regenerator) $= \dfrac{w_{net}}{Q_{wo}} = \dfrac{146.73}{807}$

$= 0.182$ or 18.2%

(ii) η_{th}(with generator) $= \dfrac{w_{net}}{Q_w} = \dfrac{146.73}{455.265}$

$= 0.322$ or 32.2%

Impact of Jets

A fluid jet is a stream of fluid issuing from a nozzle with a high velocity and hence a high kinetic energy

→ When a jet impinges on a plate or vane, it exerts a force on it (due to change in momentum). This force can be evaluated by Impulse momentum principle

Impulse-Momentum Equation

- The momentum equation is based on the law of conversation of momentum or momentum principle

- It states that

"The net force acting on a mass of fluid is equal to change in momentum of flow per unit time in that direction".

As per Newton's second law of motion,

$$F = ma$$

where

m = mass of fluid

F = force acting on the fluid

a = acceleration

But acceleration

$$a = \frac{dV}{dt}$$

$$F = m \cdot \frac{dV}{dt}$$

$$\boxed{\therefore \quad F = \frac{d(mv)}{dt}}$$

'm' - is taken inside the differential being constant.

This equation is known as momentum principle. It can also be written as

$$\boxed{\therefore F\,dt = d(mv)}$$

This equation is known as Impulse momentum Equation.

Cases on Impact of jet

A. Force exerted by the jet on the stationary plate

(i) When the plate is held normal to the jet

(ii) when the flat plate is held inclined to the jet

(iii) when plate is curved.

B. Force Exerted by the jet on the moving plate

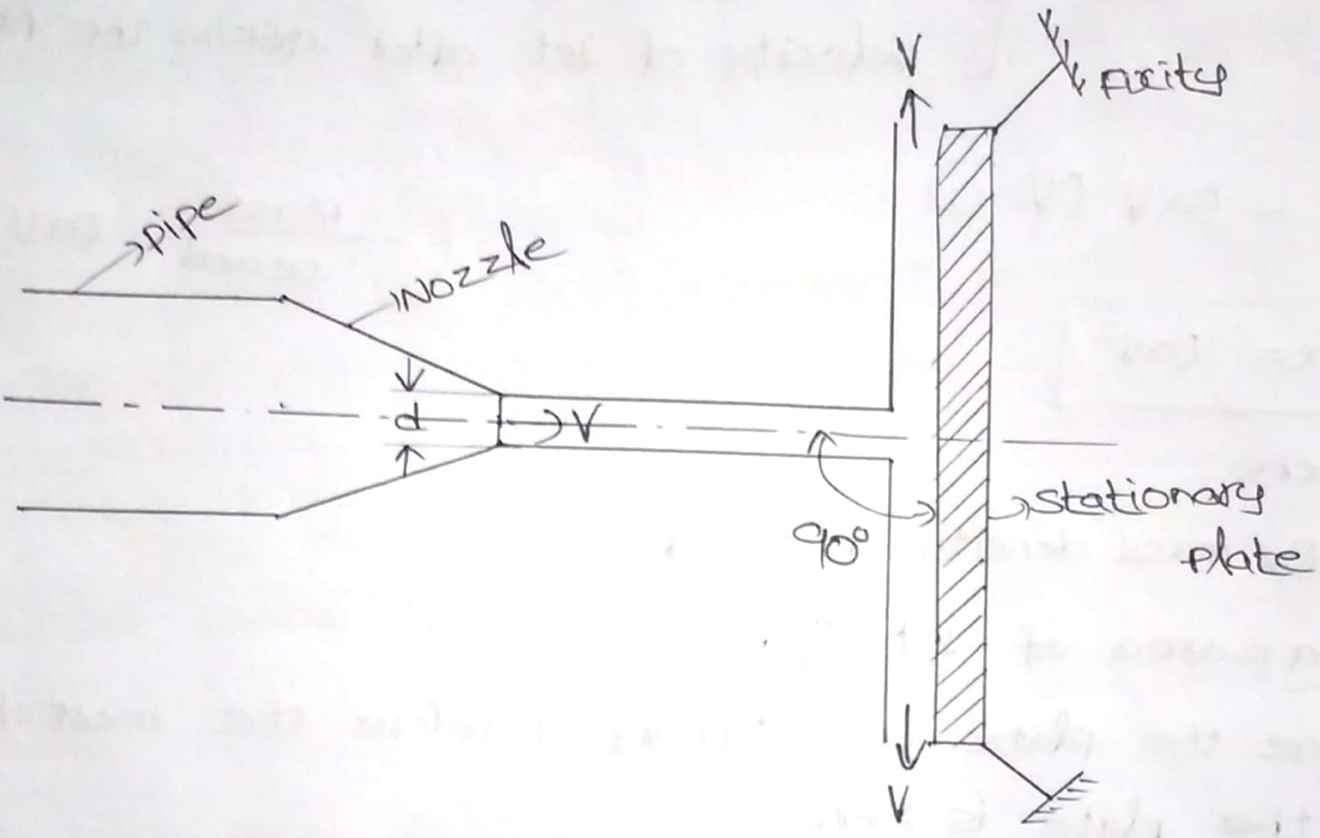
(i) when plate is held normal to the jet

(ii) when plate is held inclined to the jet

(iii) when plate is curved.

... exerted by the Jet on the stationary plate

(i) Force exerted on a stationary flat plate held normal to the Jet



Fluid jet striking a stationary plate

Above figure shows the fluid jet striking a stationary flat plate held perpendicular to the flow direction.

- Let a & V be cross sectional area and velocity of the jet respectively.
- The jet, after striking this plate, will get its direction changed through 90°. but it will move on and off the plate with velocity 'V'.

The force exerted by the jet on the plate in the direction of jet.

F_x = Rate of change of momentum

= (Initial momentum - Final momentum)

$= \frac{\text{mass}}{\text{sec}} \times$ [Velocity of jet before striking the plate - Velocity of jet after striking the plate]

$= \rho a V (V - 0)$ $\qquad \left(\because \frac{\text{Mass}}{\text{second}} = \rho a V \right)$

$$\boxed{F_x = \rho a V^2}$$

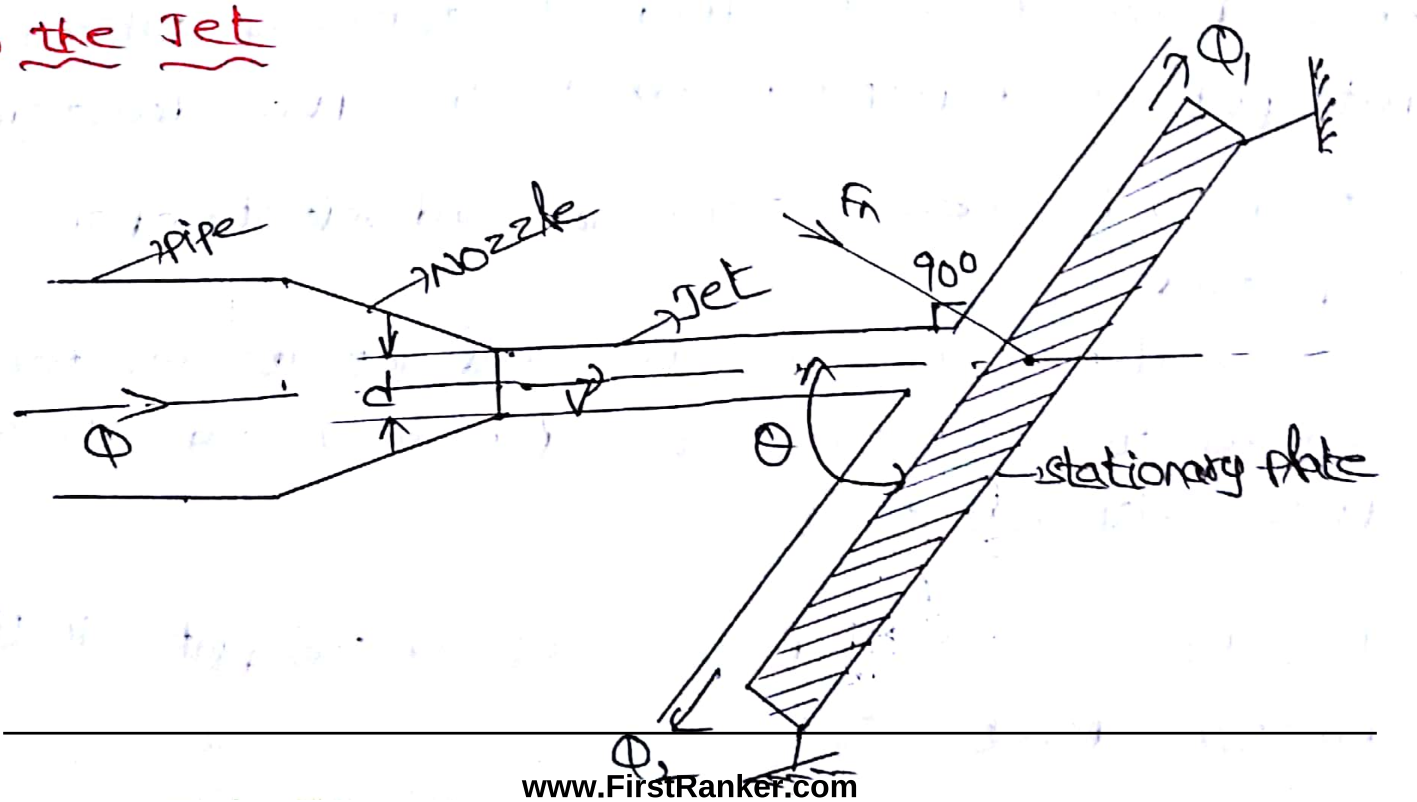
where

ρ = mass density of liquid

a = area of jet $= \frac{\pi}{4} d^2$

since the plate is stationary, therefore the work done on the plate is zero.

Force Exerted on a stationary flat plate held inclined to the Jet



shows the stationary flat plate inclined at $\theta°$ to the direction of the horizontal jet.

- If a and V are the cross-sectional area and velocity of jet respectively, then the mass of liquid per second striking the plate

$$= \rho \times aV$$

$$m = \rho aV$$

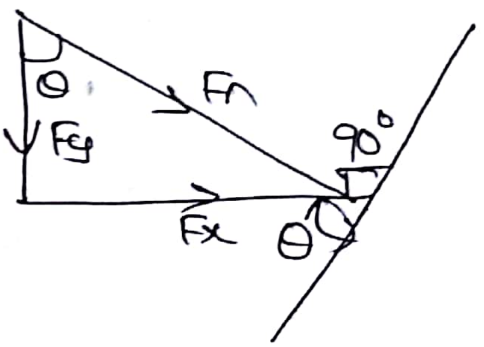
→ After striking the plate, (assuming it smooth), the jet leaves the plate with a velocity equal to initial velocity (V).

Let us apply the impulse-momentum equation in the direction normal to the plate

Force in normal direction

$$F_n = \rho aV(V\sin\theta - 0)$$

$$\boxed{\therefore F_n = \rho aV^2\sin\theta}$$



This normal force can be resolved into two components:

- component 'Fx' parallel to the direction of jet and component 'Fy' ~~normal~~ perpendicular to the direction of jet

$$F_x = F_n\sin\theta = \rho aV^2\sin\theta \times \sin\theta$$

$$\boxed{\therefore F_x = \rho aV^2\sin^2\theta} \ N$$

$F_y = F_n \cos\theta$

$F_y = \rho a V^2 \sin\theta \times \cos\theta$

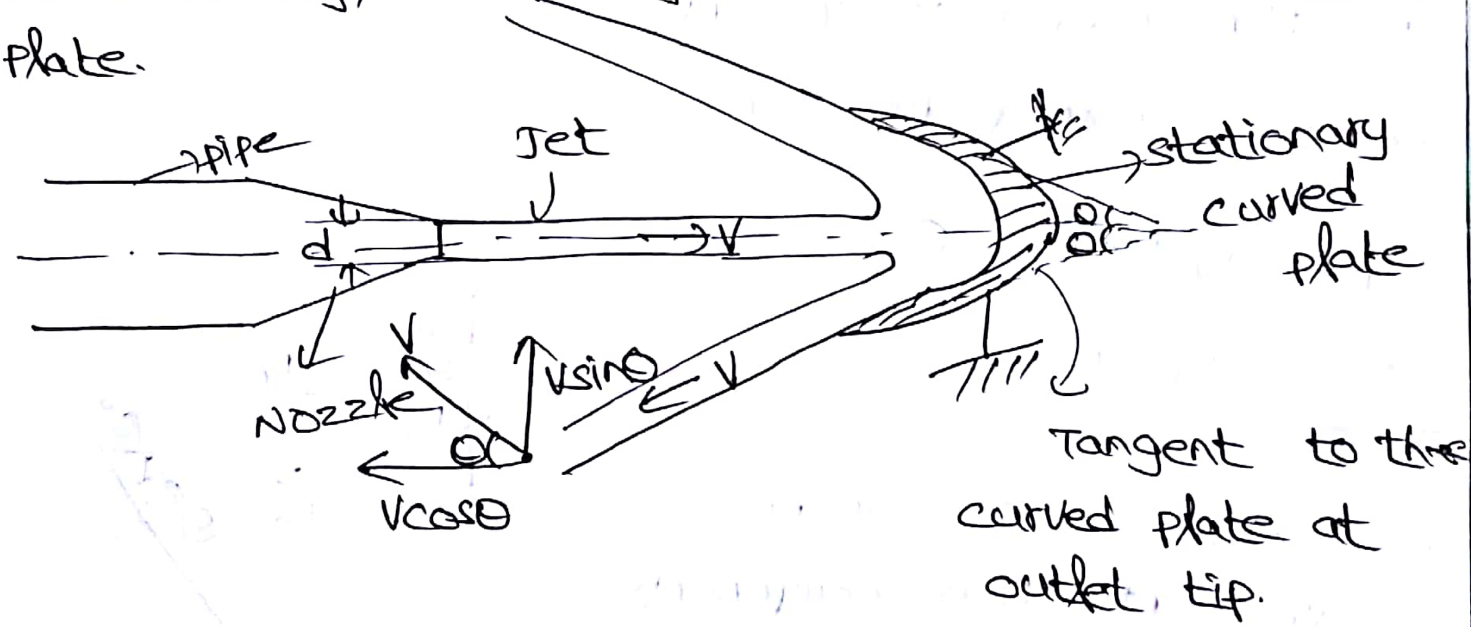
$$\boxed{\therefore F_y = \rho a V^2 \sin\theta \cdot \cos\theta}$$

Force Exerted on stationary curved plate

case-I
Jet strikes the curved plate at the centre

consider a fluid jet striking a stationary curved plate at the centre as shown in fig.

→ The jet after striking the plate comes out with the same velocity, in the tangential direction of the curved plate.



Fluid jet striking a stationary curved plate.

The velocity at the outlet of the plate can be resolved into the following components

(i) Component of velocity in the direction of jet

$= -V\cos\theta$

indicates that the velocity at the outlet in a

direction opposite to that of the fluid jet.

(ii) component of velocity perpendicular to the jet $= V\sin\theta$

Applying impulse-momentum equation, we have

Force exerted by the jet (in the direction of jet)

$$\boxed{\therefore F_x = \rho a V (V_{1x} - V_{2x})}$$

where

ρ = mass density of the fluid

a = cross-sectional area of the jet $= \frac{\pi}{4} d^2$

V = velocity of the jet

V_{1x} = Initial velocity in the direction of jet $= V$

V_{2x} = Final velocity in the direction of jet $= -V\cos\theta$

$$F_x = \rho a V [V - (-V\cos\theta)]$$

$$= \rho a V [V + V\cos\theta]$$

$$= \rho a V^2 [1 + \cos\theta]$$

similarly

$$F_x = \rho a V (V_{1y} - V_{2y})$$

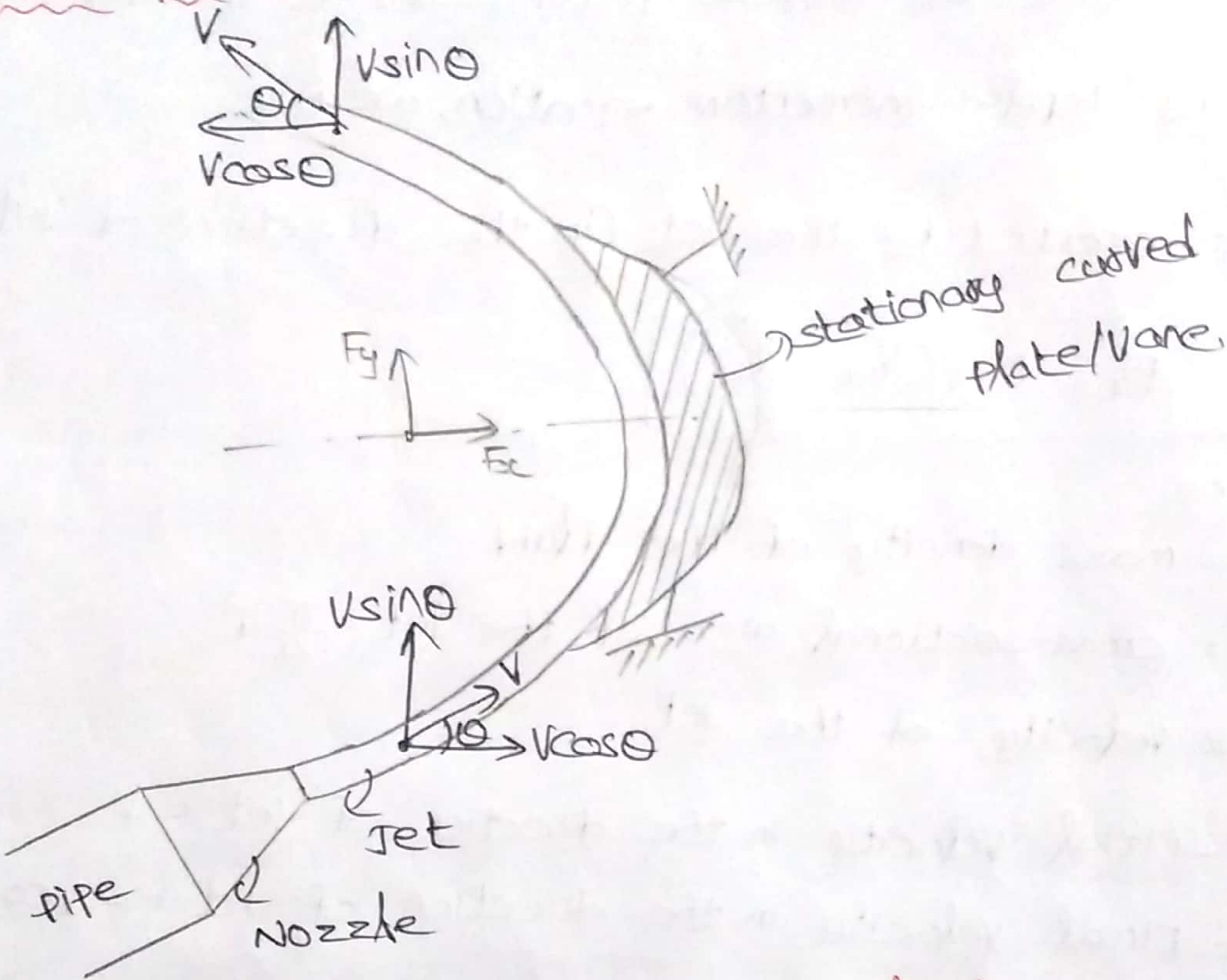
V_{1y} = Initial velocity in the direction of ~~jet~~ $y = 0$

V_{2y} = Final velocity in the direction of $y = V\sin\theta$

$$F_y = \rho a V (0 - V\sin\theta) = -\rho a V^2 \sin\theta$$

~~-ve sign indicates that force is acting in the downward~~ direction.

Jet strikes the curved plate at one end tangentially when the plate is symmetrical



Fluid jet striking stationary curved plate/vane

The above figure shows a fluid jet striking a stationary symmetrical curved plate at one end tangentially the plate is symmetrical about x-axis.

- Let V be the velocity of jet and 'θ' be the angle made with the jet with x-axis at inlet tip of the curved plate.
- The velocity of fluid at outlet tip of the curved plate will be equal to 'V'.

forces exerted by the fluid jet in x and Y directions are

$$F_x = \rho a V (V_{1x} - V_{2x})$$

$$= \rho a V [V\cos\theta - (-V\cos\theta)]$$

$$= \rho a V [V\cos\theta + V\cos\theta]$$

$$= \rho a V \, 2V\cos\theta$$

$$\boxed{F_x = 2\rho a V^2 \cos\theta}$$

$$F_y = \rho a V (V_{1y} - V_{2y})$$

$$= \rho a V [V\sin\theta - V\sin\theta]$$

$$\boxed{\therefore F_y = 0}$$

case-III

Jet strikes the curved plate or vane at one end tangentially when plate is unsymmetrical

In this case, as the plate is unsymmetrical about x-axis, therefore, the angles made by the tangents drawn at inlet and outlet tips of the plate with x-axis will be different.

- Let θ & ϕ be the angles made by the tangets at inlet tip and outlet tip respectively with x-axis.

components of velocity at inlet $V_{1x} = V\cos\theta$; $V_{1y} = V\sin\theta$

components of velocity at outlet $V_{2x} = -V\cos\phi$

$V_{2y} = V\sin\phi$

The force exerted by the fluid jet in x and y direction are

$$F_x = \rho aV (V_{1x} - V_{2x})$$

$$F_x = \rho aV [V\cos\theta - (-V\cos\phi)]$$

$$= \rho aV [V\cos\theta + V\cos\phi]$$

$$\boxed{F_x = \rho aV^2 (\cos\theta + \cos\phi)}$$

and

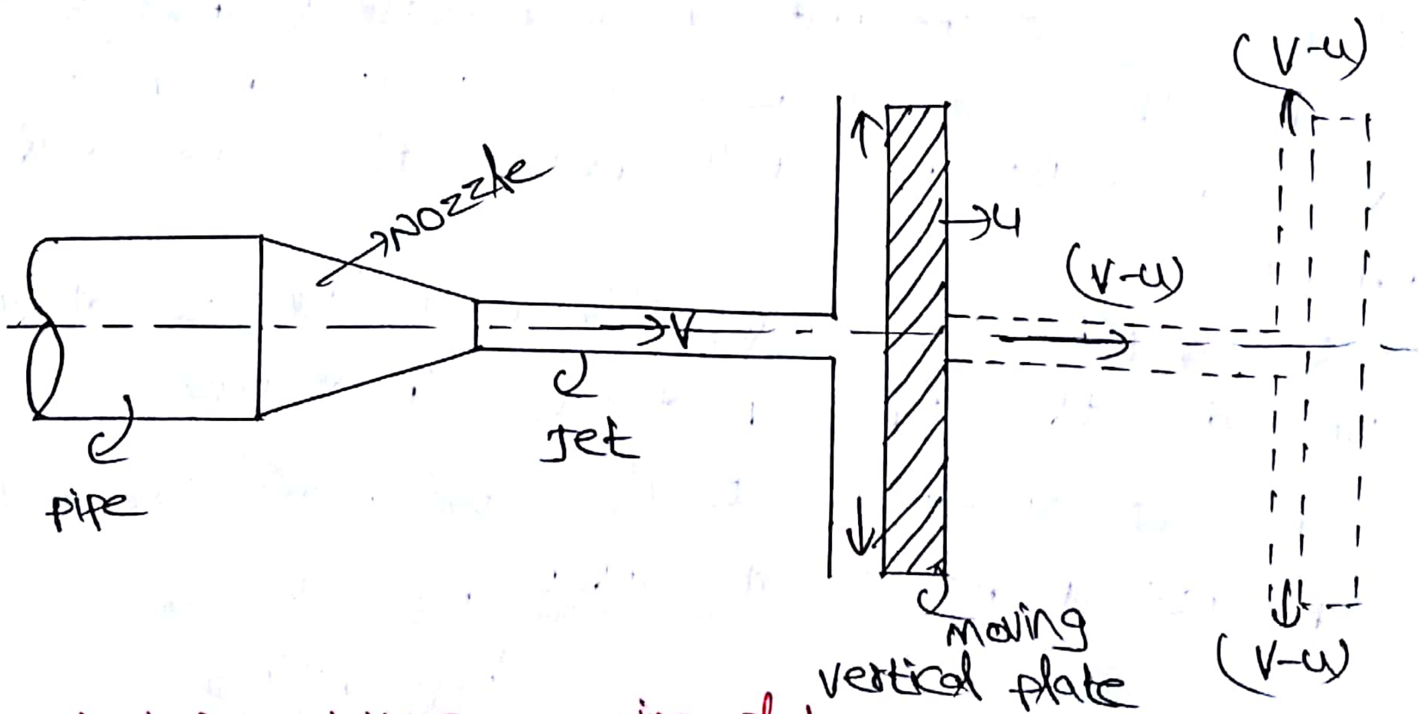
$$F_y = \rho aV (V_{1y} - V_{2y})$$

$$= \rho aV (V\sin\theta - V\sin\phi)$$

$$\boxed{F_y = \rho aV^2 (\sin\theta - \sin\phi)}$$

Force Exerted by the Jet on the Moving Plate

1) Force Exerted on moving flat plate held normal to jet



Fluid jet striking a moving plate

Fig shows a fluid jet striking a flat vertical plate moving with a uniform velocity away from the jet.

Let

V = Absolute velocity of the jet

a = cross-sectional area of the jet

u = velocity of the flat plate held normal to the jet.

→ The relative velocity with which the jet strikes the plate is $(V-u)$

Mass of water striking the plate per second $= \rho a(V-u)$

∴ Force exerted by the jet on the plate in the direction of jet,

F_x = Mass of water striking the plate/sec × (Initial velocity with which water strikes − final velocity)

$$= \rho a (V-u)[(V-u)-0]$$

$$\boxed{\therefore F_x = \rho a (V-u)^2}$$

Workdone = Force × the distance through which the body moves in the direction of force

$$\boxed{\therefore \text{Workdone} = \rho a (V-u)^2 \times u}$$

2) Force Exerted on a moving plate held inclined to the direction of jet

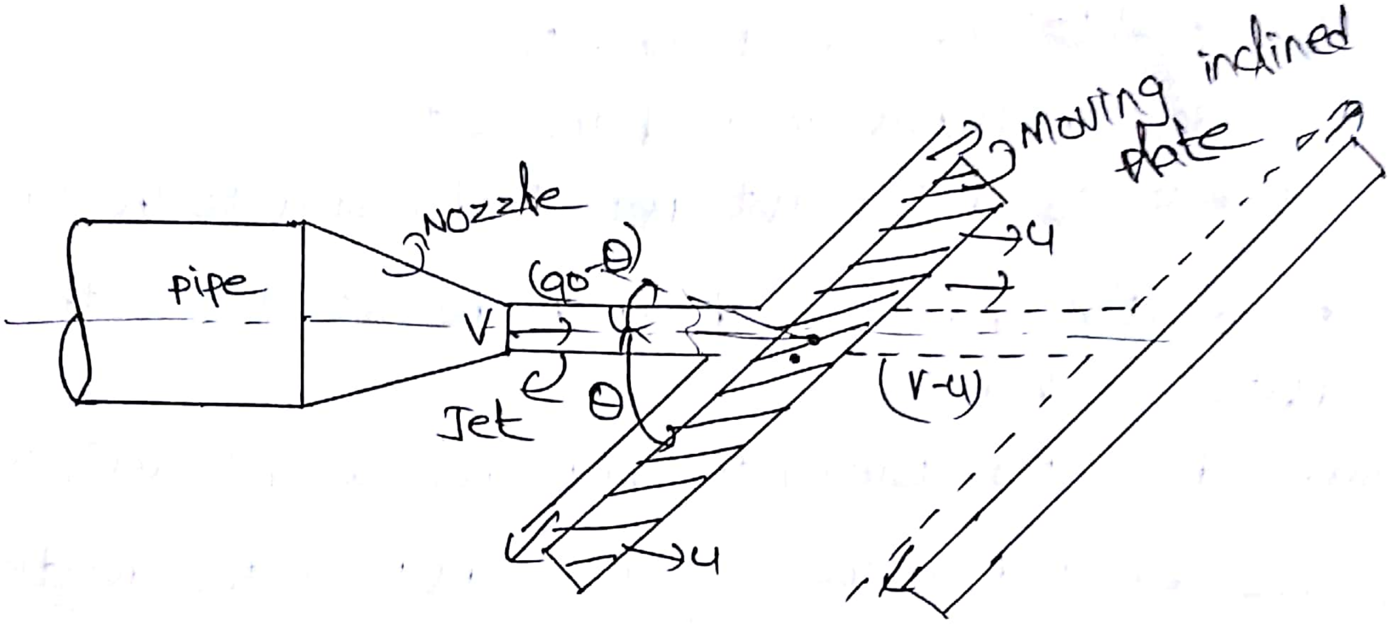
→ A fluid jet striking an inclined plate, which is moving with uniform velocity in the direction of jet

Let V = absolute velocity of the jet

u = velocity of plate in the direction of jet

a = cross sectional area of jet

θ = Angle between jet and the plate



Fluid jet striking a moving plate held inclined to the direction of jet

Relative velocity with which the jet strikes the plate = $V-u$

The mass of fluid striking the plate/second $= \rho a(V-u)$.

→ The force exerted by the jet on the plate in the direction normal to the plate is given as

$$F_n = \rho a(V-u)\left[(V-u)\sin\theta - 0\right]$$

$$\boxed{\therefore F_n = \rho a(V-u)^2 \sin\theta}$$

component of this force in the direction of jet

$$F_x = F_n \sin\theta = \rho a(V-u)^2 \sin\theta \times \sin\theta$$

$$\boxed{F_x = \rho a(V-u)^2 \sin^2\theta}$$

$$\boxed{\therefore \text{Workdone} = F_x \times u = \rho a(V-u)^2 \sin^2\theta \times u}$$

...e Exerted on a Curved plate when the plate is moving in the direction of Jet

A fluid Jet striking at the centre of curved vane moving with a uniform velocity in the direction of jet

Let

V = absolute velocity of jet

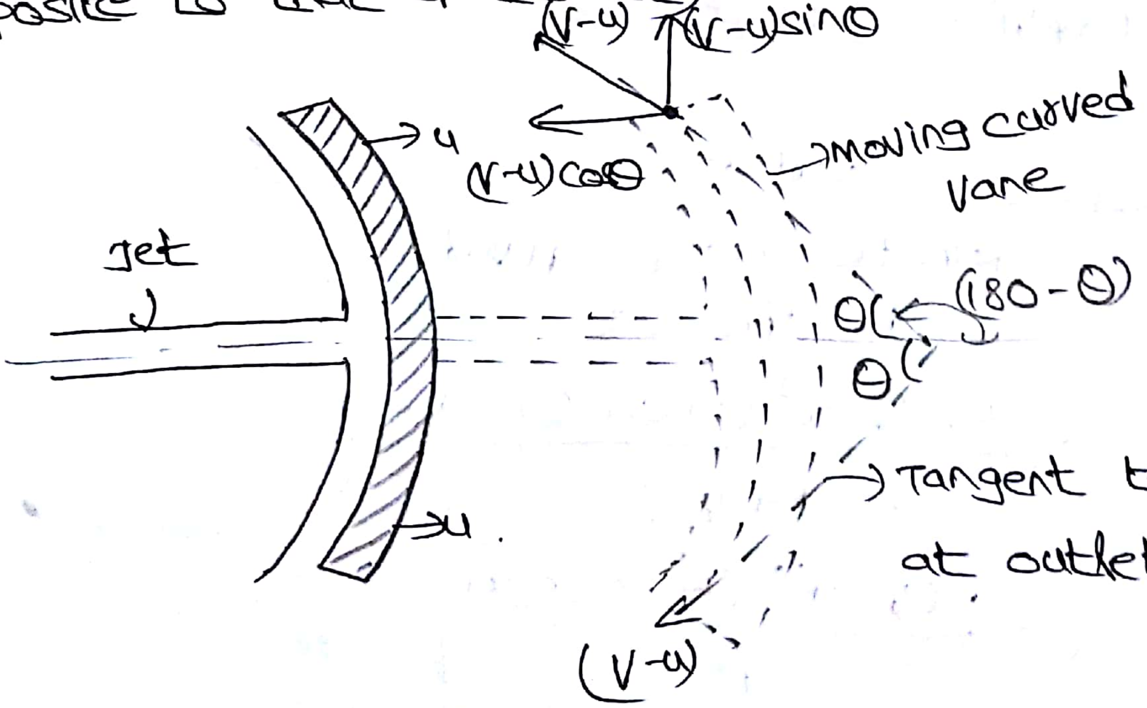
a = Area of jet, and

u = velocity of the plate in the direction of the jet.

- When the jet strikes the moving vane, the effective velocity is the relative velocity $(V-u)$.
- The component of this velocity $[(V-u)]$ in the direction of jet

$$= -(V-u)\cos\theta$$

('−' sign indicates that the component is in the direction opposite to that of the jet.)



Jet striking a curved moving Vane

∴ Force exerted by the jet on the vane

F_x = Mass/sec × (Initial velocity with which the jet strikes the vane in the direction of jet − final velocity)

$$= \rho a (V-u)\left[(V-u) - (-(V-u)\cos\theta)\right]$$

$$= \rho a (V-u)\left[(V-u) + (V-u)\cos\theta\right]$$

$$\boxed{\therefore F_x = \rho a (V-u)^2 (1+\cos\theta)}$$

Workdone on the vane per second

$$= F_x \times u$$

$$= \rho a (V-u)^2 (1+\cos\theta) \times u$$

kinetic energy of issuing jet $= \frac{1}{2} m V^2$

$$= \frac{1}{2} \rho a V (V)^2 = \frac{1}{2} \rho a V^3$$

$$\text{Efficiency} = \frac{\text{Workdone}}{\text{kinetic energy supplied by the jet}}$$

$$= \frac{\rho a (V-u)^2 (1+\cos\theta) \times u}{\frac{1}{2} \rho a V^3}$$

$$\boxed{\therefore \eta = \frac{2(V-u)^2 (1+\cos\theta) u}{V^3}}$$

A pump is a contrivance which provides energy to a fluid in a fluid system.

- It assists to increase the pressure energy or kinetic energy or both of the fluid by converting the mechanical energy

- The basic difference between a turbine and pump, from hydrodynamic point of view, is that in the former flow takes place from the high pressure side to the low pressure side

- whereas, in pump flow takes from the low pressure towards the higher pressure.

- Thus in a turbine there is accelerated flow, while in a pump, the flow is decelerated.

Classification of pumps

1. Rotodynamic pumps
   (a) Radial flow pumps — (centrifugal pumps)
   (b) Axial flow pumps
   (c) mixed flow pumps

Centrifugal pump

1. Type of casing
   (a) volute pumps
   (b) Turbine pump

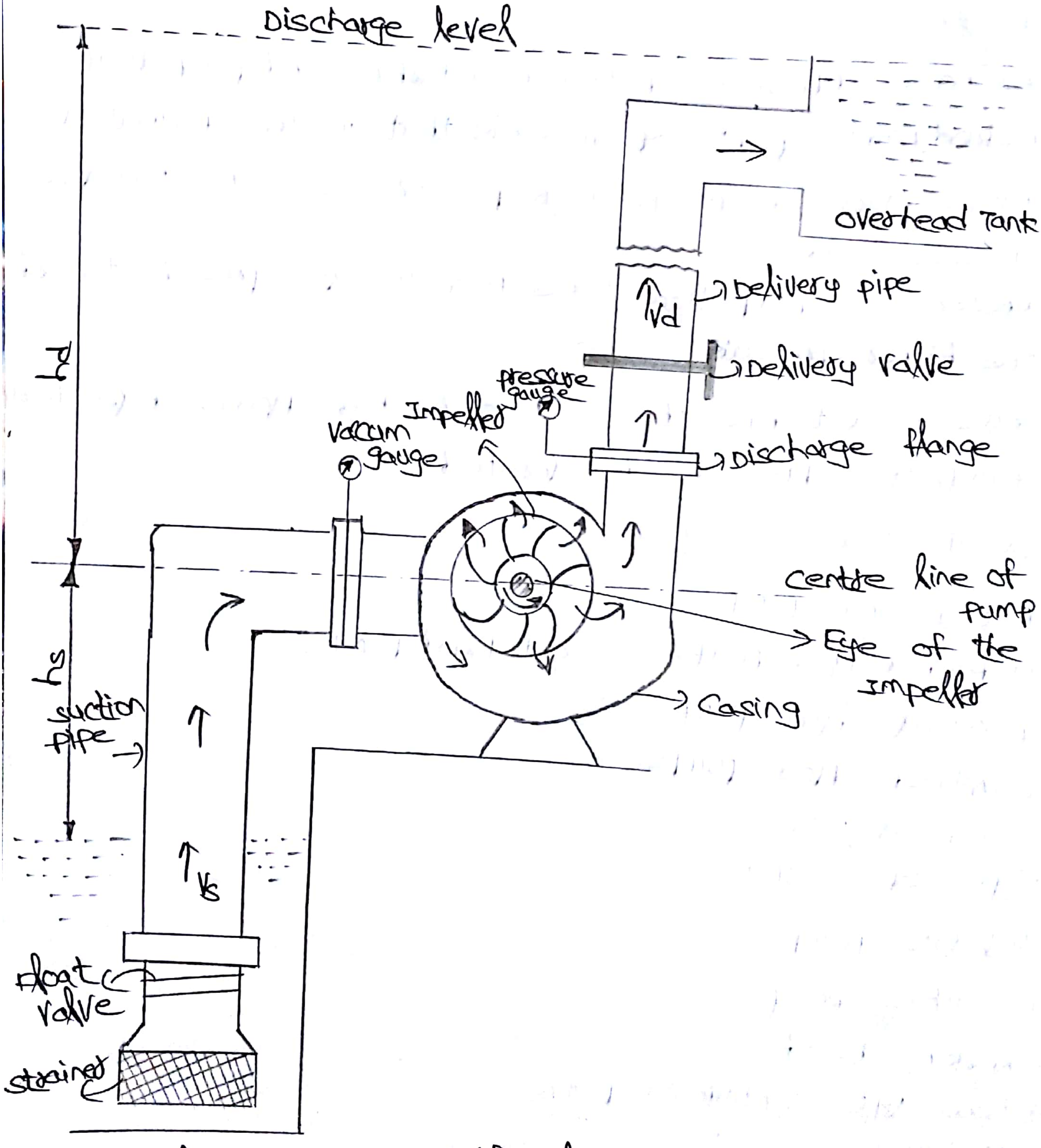
2. Working head
   (a) Low lift centrifugal pumps
   (b) medium " " "
   (c) High " " "

No. of impellers per shaft

(a) single stage centrifugal pump

(b) multi " " "

Centrifugal pump



Volute type centrifugal pump

Centrifugal pump consists of the following main components

1) Impeller 2) Casing

3) suction pipe

4) Delivery pipe

Impeller

An impeller is a wheel or rotor with a series of backward curved vanes.

- It is mounted on a shaft which is usually coupled to an electric motor.

The impellers are following three types.

(i) shrouded or closed impeller

(ii) semi open type impeller

(iii) open type impeller.

casing

- The casing is an air tight chamber surrounding the pump impeller.
- It consists of suction and discharge arrangements, supporting for bearings, and facilitates to house the rotor assembly
- It has provision to fix stuffing box packing materials which prevent external leakage.

suction pipe

- The pipe which connects the centre/eye of the impeller to sump from which liquid is to be lifted is known as suction pipe.
- In order to check the formation of air pockets the pipe is laid air tight
- To prevent the entry of solid particles debris etc.

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Delivery pipe

- The pipe which is connected at its lower end to the outlet of the pump and it delivers the liquid to the required height is known as delivery pipe.
- A regulating valve is provided on the delivery pipe to regulate the supply of water.

UNIT-V

Centrifugal Pumps:

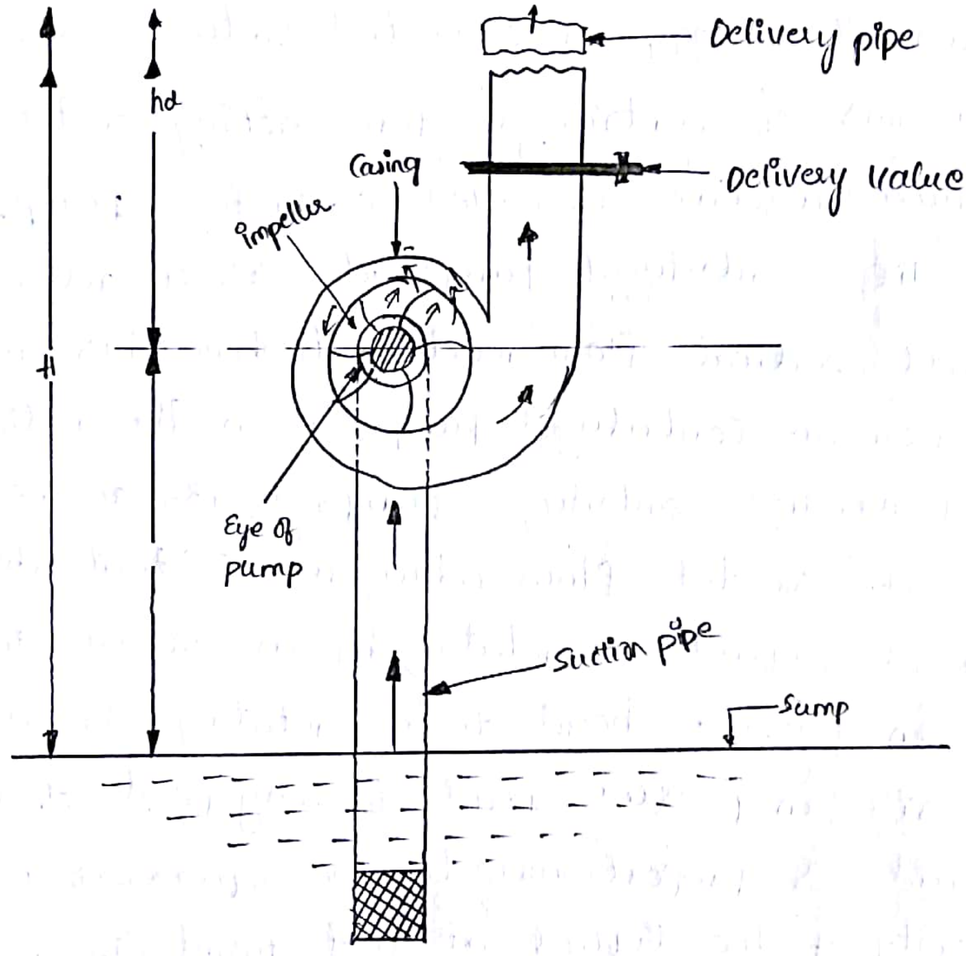
The hydraulic machines which convert the mechanical energy into hydraulic energy are called pumps. The hydraulic energy is in the form of pressure energy. If the mechanical energy is converted into pressure energy by means of centrifugal force acting on the fluid, the hydraulic machine is called centrifugal pump.

The centrifugal pump acts as a reversed of an inward radical flow reaction turbine. This means that the flow in centrifugal pumps is in the radical outward directions. The centrifugal pumps works on the principle of forced vortex flow which means that when a certain mass of liquid is rotated by an external torque, the rise in pressure head of the rotating liquid takes place. The rise in pressure head of any point of the rotating liquid is proportional to the square of tangential velocity of the liquid at that point [i.e, rise in pressure head $= \frac{V^2}{2g}$ or $\frac{\omega^2 r^2}{2g}$]. Thus at the outlet of the impeller where radius is more, the rise in pressure head will be more and the liquid will be discharged at the outlet with a high pressure head. Due to this high pressure head, the liquid can be lifted to a high level.

Main Parts of a Centrifugal Pump.

The following are the main Parts of a centrifugal Pump:

1. Impeller
2. Casing
3. Suction pipe with a foot value and a strainer.
4. Delivery Pipe.



* 1. Imprellor: The rotating Part of a centrifugal Pump is called Imprellor. It consists of a series of backward curved vanes. The impeller is mounted as a shoft which is connected to the shoft of an electric motor.
* 2. Casing: It is an air-tight Passage Surrounding the impeller and is designed in such away that the

kinetic energy of the water discharged at the outlet of the impeller is converted into Pressure energy before the water leaves the casing and enters the delivery pipe, The following three types of the casing are commonly adopted.

a. volute casing
b. vortex casing
c. casing with guide blades.

a. volute casing: The above figure shows the volute casing, which surrounds the impeller. It is of spiral type in which area of flow increases gradually, The increase in area of flow decreases the velocity of flow. The decrease in velocity increases the Pressure of the water flowing through the casing. It has been observed that in case of volute casing, the efficiency of the pump increases slightly as a large amount of energy is lost due to the formation of eddies in this type of casing.

b. Vortex Casing:

If a circular chamber is introduced between the casing and the impeller as shown in fig. the casing is known as vortex casing. By introducing the circular chamber, the loss of energy due to the formation of eddies is reduced to a considerable extent. Thus the

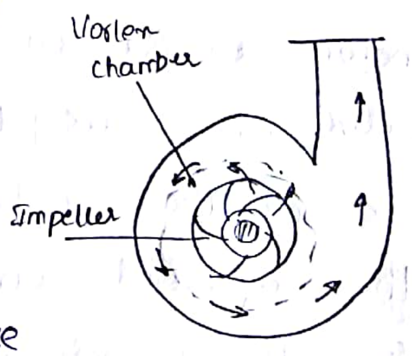
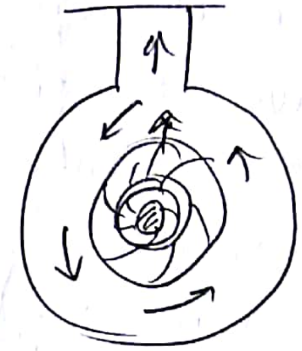


Fig: Vortex casing.

efficiency of the pump is more than the efficiency when only volute casing is provided.

c. Casing with guide blades:



This casing is shown in figure, in which the impeller is surrounded by a series of guide blades mounted on a ring which is known as diffuser. The guide vanes are designed in which a way that the water from the impeller enters the guide vanes without stock. Also the area of the guide blades vanes increases, thus red reducing the velocity of flow through guide vanes and consequently increasing the pressure of water. The water from the guide vanes then passes through the surrounding casing which is in most of the cases concentric with the impreller.

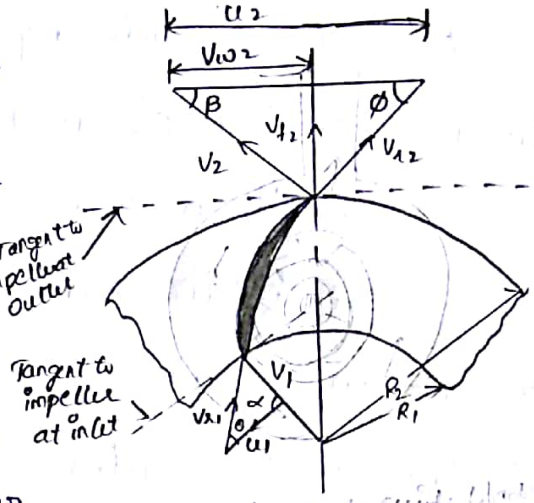
3. Suction pipe with a foot-value and a stainer:

A pipe whose one end is connected to the inlet of the pump and other end dips into water in a sump is known as suction pipe. A foot valve which is a non-return value or one-way type of valve is fitted at the lower end of the suction pipe. The foot valve opens only in the upward direction. A strainer is also fitted at the lower end of the suction pipe.

4. Delivery Pipe: A pipe whose one end is connected to the outlet of the pump and other end delivers

the water at a required height is known as delivery pipe.

Workdone by the centrifugal pump (or by impeller) on water:



In case of the centrifugal pump, work is done by the impeller on the water. The expression for the workdone by the impeller on the water is obtained by drawing velocity triangles at inlet and outlet of the impeller in the same way as for a turbine.

The water enters the impeller radially at inlet for best effecency of the pump, which means the absolute velocity of water enters the imakes an angle of 90° with the direction of motion of the impeller at inlet. Hence angle $\alpha = 90^\circ$ and $V_{w_1} = 0$, for drawing the velocity triangles, the same notations are used as that for turbines. Figure shows the velocity triangles at the inlet and outlet tips of the vane fixed to an impeller.

let N = Speed of the impeller in rpm.

D_1 = Diameter of impeller at inlet

u_1 = Tangential velocity of impeller at inlet,

$$u_1 = \frac{\pi D_1 N}{60}$$

D_2 = Diameter of impeller at outlet.

u_2 = Tangential velocity of impeller at outlet.

$$= \frac{\pi D_2 N}{60}$$

V_1 = Absolute velocity of water at inlet.

V_{r_1} = Relative velocity of water at outlet,

α = angle made by absolute velocity (V_1) at inlet with the direction of motion of vane.

θ = angle made by relative velocity (V_{r_1}) at inlet with the direction of motion of vane, and

V_2, V_{r_2}, β and ϕ are the corresponding values at outlet.

As the water enters the impeller radially which means the absolute velocity of water at inlet is in the radial direction and hence angle $\alpha = 90°$ & $V_{w_1} = 0$

A centrifugal pump is the reverse of a radially inward flow reaction turbine, the workdone by the water on the runner per second per unit weight of the water striking per second is given by.

$$= \frac{1}{g}\left[V_{w_1}u_1 - V_{w_2}u_2\right]$$

∴ Workdone by the impeller on the water per second per unit weight of water striking per second

$$= -\left[\text{workdone in case of turbine}\right]$$

$$= -\left[\frac{1}{g}\left(V_{w_1}u_1 - V_{w_2}u_2\right)\right]$$

$$= \frac{1}{g}\left[V_{w_2}u_2 - V_{w_1}u_1\right]$$

$$= \frac{1}{g}V_{w_2}u_2 \qquad \left[\because V_{w_1} = 0 \text{ hw}\right]$$

workdone by impeller on water per second $= \frac{W}{g}V_{w_2}u_2$

where W = weight of water $= \rho \times g \times Q$

Q = volume of water

$Q =$ Area $\times$ velocity of flow

$$= \pi D_1 B_1 \times V_{f_1}$$

$$= \pi D_2 B_2 \times V_{f_2}$$

where B_1 and B_2 are width of impeller at inlet and outlet and V_{f_1} and V_{f_2} are velocities of flow at inlet and outlet.

__Definitions of heads and efficiencies of a centrifugal pump:__

1) Suction head (h_s): It is the vertical height of centre line of centrifugal pump above the water surface in the tank or pump from which water is to be lifted. This height is also called suction lift and is denoted by 'h_s'.

2) Delivery head (h_d): The vertical distance between the centre line of the pump and the water surface in tank to which water is delivered is known as delivery head is known as delivery head. This is denoted by 'h_d'.

3) Static head (H_s): The sum of suction head and delivery head is known as static head. This is represented by H_s is written as

$$\boxed{H_s = h_s + h_d}$$

4) Manometric head (H_m): This is defined as head against which a centrifugal pump has to work. It is denoted by 'H_m'. It is given by following expressions.

(a) $H_m =$ Head imparted by impeller to the water − loss of head in pump

$$= \frac{V_{w_2} u_2}{g} - \text{loss of head in impeller and casing.}$$

$$= \frac{V_{w_2} u_2}{g} \text{ ----- if loss of pump is zero.}$$

(b) $H_m =$ Total head at outlet of pump − Total head at inlet of pump.

$$= \left[\frac{P_o}{\rho g} + \frac{V_o^2}{2g} + z_o\right] - \left[\frac{P_i}{\rho g} + \frac{V_i^2}{2g} + z_i\right]$$

where $\frac{P_o}{\rho g}$ = pressure head at outlet of pump = h_d

$\frac{V_o^2}{2g}$ = Velocity head at outlet of pump

= Velocity head in delivery pump = $\frac{V_d^2}{2g}$

z_o = vertical height of outlet of pump from datum line and

$\frac{P_i}{\rho g}, \frac{V_i^2}{2g}, z_i$ = corresponding values of pressure head, velocity head and datum head at inlet of pump.

i.e., h_s, $\frac{V_s^2}{2g}$ and z_s respectively.

$$\boxed{H_m = h_s + h_d + h_{f_s} + h_{f_d} + \frac{V_d^2}{2g}}$$

where h_s = suction head.

h_d = Delivery head.

h_{f_s} = Frictional head loss in suction pipe,

h_{f_d} = Frictional head loss in delivery pipe

V_d = velocity of water in delivery pipe

5) Efficiencies of a centrifugal pump: In case of a centrifugal pump, the power is transmitted from the shaft of electric motor to shaft of the pump and then to impeller. From impeller, the power is given to the water. Thus power is decreasing from the shaft of the pump to the impeller and then to water. The following are important efficiencies of centrifugal pump

(a) Monometric efficiency, η_{man}

(b) Mechanical efficiency, η_m and

(c) Overall efficiency, η_o

(a) Monochrometic Efficiency (η_{man}): The ratio of the manometric head to the head imparted by impeller to the water is known as manometric efficiency. Mathematically, it is written as

$$\eta_{man} = \frac{\text{Manometric head}}{\text{Head imparted by impeller to water}}$$

$$\eta_{man} = \frac{H_m}{\frac{V_{w_2}u_2}{g}} = \frac{gH_m}{V_{w_2}u_2}$$

The power at the impeller of pump is more than power given to the water at outlet of pump. The ratio of power given to water at outlet of pump to power available at impeller, is known as manometric efficiency.

The power given to water at outlet of pump $= \frac{WH_m}{1000}$ kW.

The power at the impeller $= \frac{\text{Workdone by impeller per second}}{1000}$ kW

$$= \frac{W}{g} \times \frac{V_{w_2}u_2}{1000} \text{ kW}$$

$$\eta_{man} = \frac{\frac{W \times H_m}{1000}}{\frac{W}{g} \times \frac{V_{w_2} \times u_2}{1000}}$$

$$\boxed{\eta_{man} = \frac{g \times H_m}{V_{w_2}u_2}}$$

(b) Mechanical Efficiency (η_m) : The power at shaft of centrifugal pump is more than the power available at impeller of pump. Ratio of power available at impeller to power at shaft of centrifugal pump.

$$\eta_m = \frac{\text{power at impeller}}{\text{power at shaft}}$$

The power at impeller in kW $= \frac{\text{Workdone by impeller per second}}{1000}$

$$= \frac{W}{g} \times \frac{V_{w_2}u_2}{1000}$$

$$\boxed{\eta_m = \frac{\frac{W}{g}\left(\frac{V_{w_2}u_2}{1000}\right)}{S.P}}$$

S.P = Shaft Power

(c) Overall Efficiency (η_o) : It is defined as ratio of power output of pump to power input to pump. The power output of pump in kW.

$$= \frac{\text{Weight of water lifted} \times H_m}{1000}$$

$\frac{WHm}{1000}$

Power input to pump = Power supplied by electric motor
= Shaft power of pump

$$\eta_o = \frac{\left(\frac{WHm}{1000}\right)}{S.P}$$

$$\eta_o = \eta_{man} \times \eta_m.$$

i) The internal and external diameters of impeller of centrifugal pump are 200 mm and 400mm respectively. Pump is running at 1200 rpm. The vane angles of impeller at inlet and outlet are 20° and 30° respectively. The water enters impeller radially and velocity of flow is constant. Determine the workdone by impeller per unit weight of water.

Sol: Internal diameter of impeller, $D_1 = 200mm = 0.20m$
External diameter of impeller, $D_2 = 400mm = 0.40m$
Speed, $N = 1200$ rpm

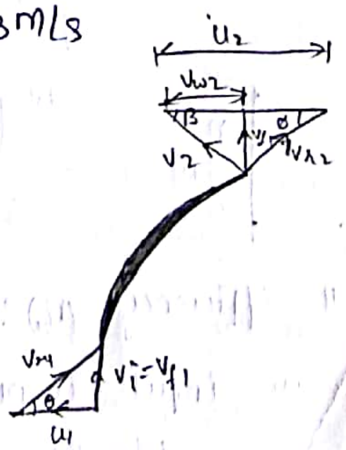
Vane angle at inlet, $\theta = 30°$
Vane angle at outlet, $\phi = 30°$.

Water enters radially means, $\alpha = 90°$, $V_{w_1} = 0$.
Velocity of flow, $V_{f_1} = V_{f_2}$

Tangential velocity of impeller and inlet and outlet are,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.2 \times 1200}{60} = 12.56 \text{ m/s}.$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.40 \times 1200}{60} = 25.13 \text{ m/s}$$



From inlet velocity triangle,

$$\tan\theta = \frac{V_{f_1}}{u_1} = \frac{V_{f_1}}{12.56}$$

$$\tan 20 = \frac{V_{f_1}}{12.56}$$

$$V_{f_1} = (12.56) \times \tan 20 = 4.57 \text{ m/s}$$

$$V_{f_2} = V_{f_1} = 4.57 \text{ m/s}$$

From outlet triangle, $\tan\phi = \frac{V_{f_2}}{u_2 - V_{w_2}}$

$$\tan 30 = \frac{4.57}{25.13 - V_{w_2}}$$

$$25.13 - V_{w_2} = \frac{4.57}{\tan 30}$$

$$V_{w_2} = 25.13 - \frac{4.57}{\tan 30}$$

$$V_{w_2} = 25.13 - 7.915$$

$$= 17.215 \text{ m/s}$$

The workdon by impeller per kg of water per second is given by $= \frac{1}{g} V_{w_2} u_2$

$$= \frac{17.215 \times 25.13}{9.81}$$

$$= \underline{\underline{44.1}} \text{ N-m/N}$$

2. A centrifugal pump is to discharge $0.118 \text{ m}^3/\text{s}$ at a speed of 1450 rpm against a head of 25 m. The impeller diameter is 250 mm its width at outlet is 50 mm and manometric efficiency is 75%. determine the vane angle at the outer periphery of the impeller

Sol : discharge, $Q = 0.118 m^3/s$

speed, $N = 1450$ rpm

Head, $H_M = 25$ m

diameter at outlet, $D_2 = 250$ mm $= 0.25$ m

width at outlet, $B_2 = 50$ mm $= 0.50$ m

Manometric efficiency, $\eta_{man} = 75\% = 0.75$

let vane angle at outlet $= \phi$

Tangential velocity of impeller at outlet

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.25 \times 1450}{60} = 18.98 \text{ m/s}$$

Discharge is given by $Q = \pi D_2 B_2 \times V_{f_2}$

$$V_{f_2} = \frac{Q}{\pi D_2 B_2} = \frac{0.118}{\pi \times 0.25 \times 0.5}$$

$$= 3.0 \text{ m/s}$$

manometric efficiency is given by $\eta_{man} = \dfrac{g H_M}{V_{w_2} u_2}$

$$0.75 = \frac{9.81 \times 25}{V_{w_2} \times 18.98}$$

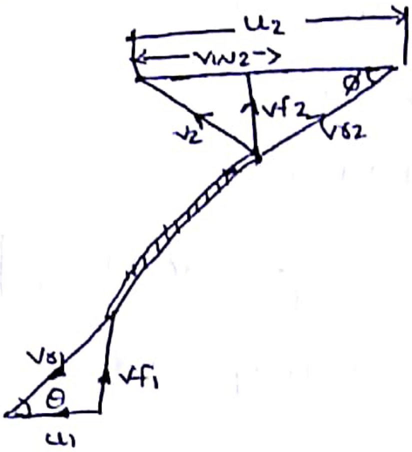
$$V_{w_2} = \frac{9.81 \times 25}{0.75 \times 18.98}$$

$$= 17.23 \text{ m/s}$$

From outlet velocity triangle

$$\tan \phi = \frac{V_{f_2}}{u_2 - V_{w_2}}$$

$$\tan\phi = \frac{30}{18.98-17.23}$$



$$\tan\phi = 1.7143$$

$$\phi = \tan^{-1}(1.7143)$$

$$= 59.74^\circ.$$

<u>specific speed of a centrifugal pump</u>

The specific speed of a centrifugal pump is define as the speed of a geomatrically similar pump which would diliver and lubic metre of lequid per second against a head of one metre It is denoted by Ns.

<u>Expression for specific speed of pump</u> : The discharge Q, for a centrifugal pump is given by

$$Q = \text{Area} \times \text{velosity of flow}$$

$$= \pi D \times B \times V_{f_1}$$

(or) $\boxed{Q \propto D \times B \times V_{f_1}}$ — (i)

Where D = diameter of the impeller of the pump

B = width of the impeller

We know that $B \propto D$

From equation (i), we have $\boxed{Q \propto D^2 \times V_{f_1}}$ —(ii)

We also know that tangential velocity is given by

$\boxed{u = \frac{\pi D N}{60} \propto DN}$ - (iii)

Now the tangential velocity (u) and velocity of flow of (Vf) are related to the manometric head (Hm) as

$$\boxed{u \propto Vf\sqrt{H_M}} \quad \text{--- (iv)}$$

Substituting the value of u in equation (iii)

$$\sqrt{H_M} \propto DN$$

(or) $$\boxed{D \propto \frac{\sqrt{Hm}}{N}} \quad \text{--- (v)}$$

Substituting the value of D in equation (ii)

$$Q \propto \frac{H_M}{N^2} \times Vf$$

$$\propto \frac{Hm}{N^2} \times \sqrt{Hm}$$

$$\propto \frac{H^3m^{/2}}{N^2}$$

$$\boxed{Q = k\frac{Hm^{3/2}}{N2}} \quad \text{--- (vi)}$$

Where k = constant of proportionality

If Hm = 1m and Q = $1m^3/s$ N becomes = Ns

Substituting these values in equation (vi)

We get

$$1 = \frac{k\,(1)^{3/2}}{Ns^2}$$

$$k = Ns^2$$

Substituting the value of k in equation (ii)

$$Q = Ns^2 \cdot \frac{H_m^{3/2}}{N^2}$$

$$\sqrt{Q} = Ns \cdot \frac{H_m^{3/4}}{N}$$

$$\boxed{Ns = \frac{N\sqrt{Q}}{H_m^{3/4}}}$$

Model testing of centrifugal pumps :

Before manufacturing the large size pumps, their models which are complete similarity with the actual pumps also (called prototypes) are made tests are conducted on the models and performence of the prototypes are predicted. The complex similarity between the model and actual pump (prototype) will exist if the following conditions are satisfied.

1. specific speed of model = specific speed of prototype

$$(Ns)_m = (Ns)_p$$

$$\boxed{\left(\frac{N\sqrt{Q}}{H^{3/4}}\right)_m = \left(\frac{N\sqrt{Q}}{H^{3/4}}\right)_p}$$

2. Tangential velocity (u) is given by, $u = \frac{\pi DN}{60}$

also $u \propto \sqrt{H_M}$

3. $Q \propto D^2 V f_1$

$V f_1 \propto u \propto DN \quad Q\, D^2 \times DN$

$Q \propto D^3 N$

$$\frac{Q}{D^3 N} = \text{constant}$$

$$\boxed{\left(\frac{Q}{D^3 N}\right)_m = \left(\frac{Q}{D^3 N}\right)_p}$$

4. Power of the pump

$P \propto Q \times H_m$

$\propto D^3 N \times H_m \qquad \because Q \propto D^3 N$

$\propto D^3 N \times D^2 N^2 \qquad \sqrt{H_m} \propto DN$

$\propto D^5 N^3 \qquad \frac{P}{D^5 N^3} = \text{constant}$

$$\boxed{\left(\frac{P}{D^5 N^3}\right) = \left(\frac{P}{D^5 N^3}\right)_p}$$

A single stage centrifugal pump with impeller diameter of 30 cm rotates at 200 rpm and lifts 3 m³ of water per second to a height of 30m with an efficiency of 75%. Find the number of stages and diameter of each impeller of a similar multi stage pump to lift 5 m³ of water per second to a height of 200 metres when rotating at 1500 rpm.

Sol :

Single - stage pump

Diameter of impeller, $D_1 = 30\ cm = 0.30m$

speed , $N_1 = 2000\ rpm$

discharge , $Q_1 = 3m^3/s$

height $Hm_1 = 30m$

Efficiency ., $\eta_{man} = 75\% = 0.75$

multystage similar pump :

Dischange , $Q_2 = 5m^3/s$

Total height $= 200m$

Let the height per stage $= H_{M_2}$

speed $N_2 = 1500$

Diameter of each impeller $= D_2$

specific speed should be same

$$\left(\frac{N\sqrt{Q}}{Hm^{3/4}}\right)_1 = \left(\frac{N\sqrt{Q}}{Hm^{3/4}}\right)_2$$

$$\frac{N_1\sqrt{Q_1}}{Hm_1^{3/4}} = \frac{N_2\sqrt{Q_2}}{Hm_2^{3/4}}$$

$$\frac{2000\sqrt{3}}{(30)^{3/4}} = \frac{1500\sqrt{5}}{Hm_2^{3/4}}$$

$$Hm_2^{3/4} = \frac{1500\sqrt{5} \times (30)^{3/4}}{2000\sqrt{3}}$$

$$H_{m_2}^{3/4} = \frac{1500}{2000}\sqrt{\frac{5}{3}} \times (30)^{3/4}$$

$$= 12.818(30)^{3/4}$$

$$= 12.411$$

$$\doteq 154$$

$$H_{m_2} = (12.411)^{4/3} = 28.71 \text{ m}$$

$\therefore$ Number of stages $= \dfrac{\text{total head}}{\text{head per stage}}$

$$= \frac{200}{28.71} = 6.76 \simeq 7$$

We know the equation

$$\frac{\sqrt{H_{m_1}}}{D_1 N_1} = \frac{\sqrt{H_{m_2}}}{D_2 N_2}$$

$$\frac{\sqrt{30}}{0.30 \times 2000} \times \frac{\sqrt{28.71}}{D_2 \times 1500} \quad D_2 = \frac{28.71 \times 0.30 \times 2000}{1500}$$

$$D_2 = 0.3913 \text{ m}$$

$$D_2 = 391.3 \text{ mm}$$

Two geometrically similar pumps are running at the same speed of 1000 rpm. on pump has can impeller diameter of 0.30 metre and lifts water at the rate of 20 litres per second against a head of 15 mitres. Determine the head and impeller diameter of the other pump the deliver half the discharge

Sol: For pump No. 1:

Speed, $N_1 = 1000$ rpm

diameter $D_1 = 0.30$ m

discharge, $Q_1 = 20$ litres/s $= 0.020\ m^3/s$.

Head, $H_{m_1} = 15$ m.

For pump No. 2,

Speed, $N_2 = 1000$ rpm

discharge, $Q_2 = \frac{Q_1}{2} = \frac{20}{2} = 10$ litres/s.

$= 0.01\ m^3/s$.

Let D_2 = Diameter of impeller

H_{m_2} = Head developed.

$$\frac{N_1\sqrt{Q_1}}{H_{m_1}^{3/4}} = \frac{N_2\sqrt{Q_2}}{H_{m_2}^{3/4}}$$

$$\frac{1000\sqrt{0.02}}{(15)^{3/4}} = \frac{1000\sqrt{0.01}}{H_{m_2}^{3/4}}$$

$$H_{m_2}^{3/4} = \frac{1000\sqrt{0.01}}{1000\sqrt{0.02}} \times (15)^{3/4}$$

$$H_{m_2}^{3/4} = 5.389 \Rightarrow H_{m_2} = (5.389)^{4/3} = \underline{9.44\,m}$$

We know the equation,

$$\left(\frac{\sqrt{H_m}}{DN}\right)_1 = \left(\frac{\sqrt{H_m}}{DN}\right)_2$$

$$\frac{\sqrt{H_{m_1}}}{D_1 N_1} = \frac{\sqrt{H_{m_2}}}{D_2 N_2}$$

$$\frac{\sqrt{15}}{0.3 \times 1000} = \frac{\sqrt{9.44}}{D_2 \times 1000}$$

$$D_2 = \frac{\sqrt{9.44} \times 0.3}{\sqrt{15}} = 0.238\,m = \underline{238.0\,mm}$$

Priming of a centrifugal Pump:

Priming of a centrifugal pump is defined as operation in which the suction pipe, casing of pump and a portion of delivery pipe upto delivery valve is completely filled up from outside source with the liquid to be raised by pump before starting pump. Thus air from these parts of the liquid to be pumped.

workdone by the impeller per unit weight of liquid per sec is known as the head generated by the pump as $= \frac{1}{g} V_{w_2} u_2$ m.

This equation is independent of the density of the liquid. This means that when pump is running in air, the head generated is in terms of metre of air. If the pump is primed with water, the head generated is same metre of water. But as the density of air is very low, the generated head of air in terms of equivalent metre of water head is negligible and hence the water may not be sucked from the pump. To avoid this difficulty, priming necessary.

Characteristic curves of centrifugal Pumps:

These are defined which are plotted from the results of a number of tests on the centrifugal pump. These curves are necessary to predict the behaviour and performance of the pump, when the pump is working under different flow rate, head and speed. The followings are the important characteristics curves for pumps:

1. Main characteristic curves.
2. Operating characteristic curves.
3. Constant efficiency (or) Muschel curves.

* 1. Main characteristic curves:

The main characteristic curve of a centrifugal pump consists of variation of head (monometric head, H_m) power and discharge with respect to speed. For plotting curves of monometric head versus speed, discharge is kept constant. For plotting curves of power versus speed, the manometric head and discharge are kept constant.

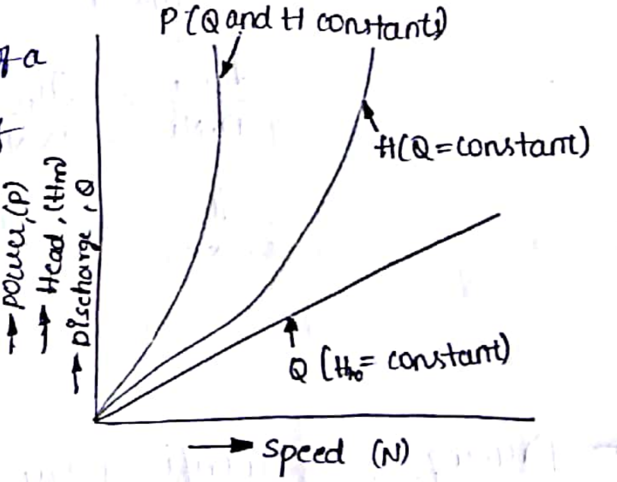


Fig: Main characteristic curves of a pump.

* 2. Operating characteristic curves:

If the speed is kept constant, the variation of manometric head, power and efficiency with respect to discharge gives operating characteristics of pump.

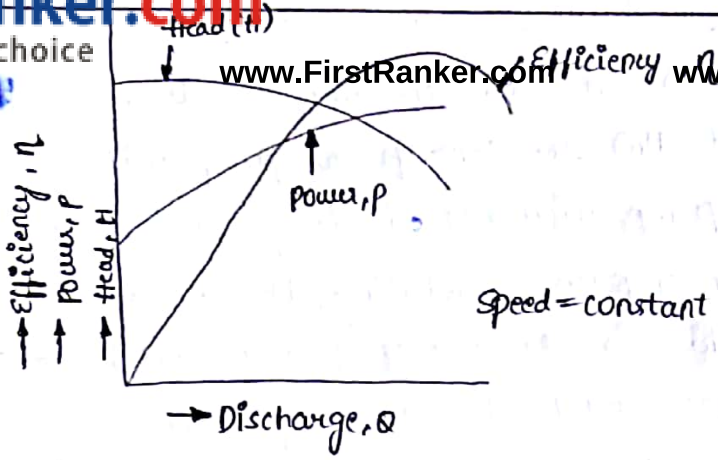


Fig: Operating characteristics of a pump.

* 3. Constant Efficiency Curves:

For obtaining constant efficiency curves for a pump the head versus discharge curves for a pump, and efficiency versus discharge curves for different speeds are used. Fig(a) shows the head versus discharge curves for different speeds. The efficiency versus discharge curves for the different speeds are shown in fig(b). By combining these curves ($H \sim Q$ curves and $\eta \sim Q$ curves), constant efficiency curves are obtained as shown in fig(a).

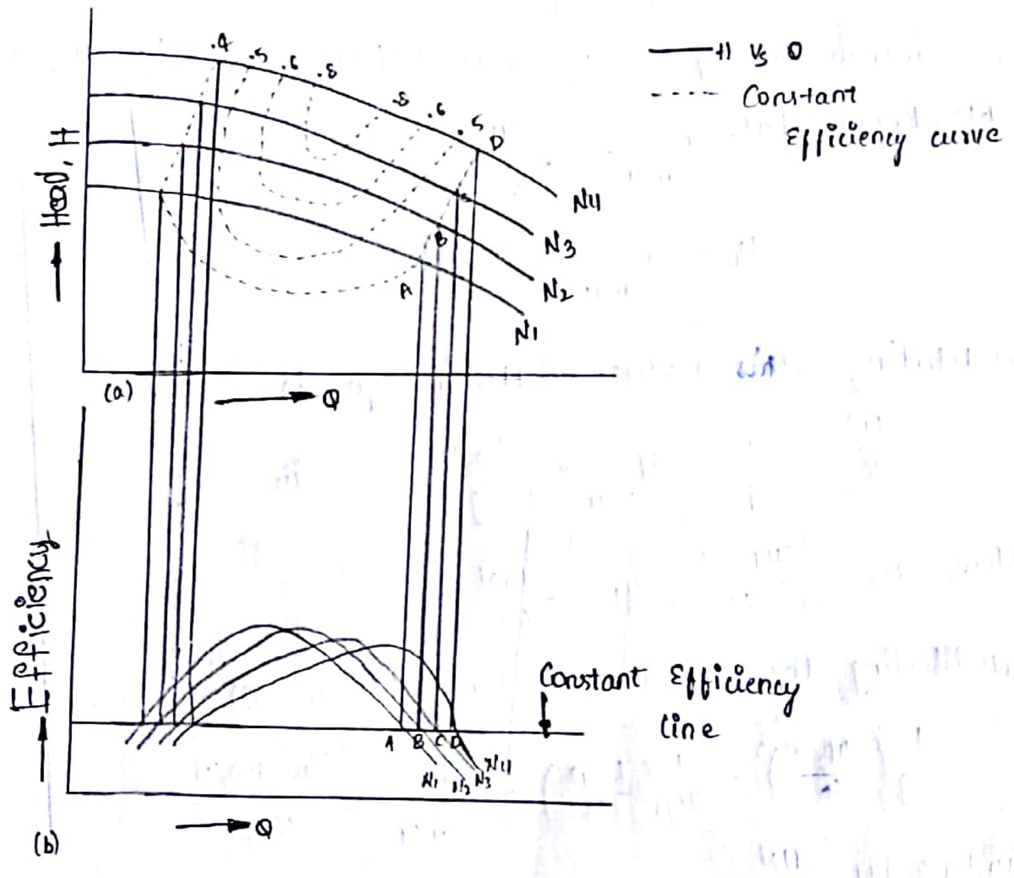


Fig: Constant Efficiency curves of a pump

Minimum Speed for starting a centrifugal pump:

If the pressure rise in the impeller is more than or equal to manometric head (H_m) the centrifugal pump will start delivering water. Otherwise, the pump will not discharge any water, though the impeller is rotating. When impeller is rotating, the water in contact with the impeller is also rotating. This is the case of forced vortex. In case of forced vortex, the centrifugal head or head due to pressure rise in the impeller

$$= \frac{\omega^2 r_2^2}{2g} - \frac{\omega^2 r_1^2}{2g}$$

where, ωr_2 = Tangential velocity of impeller at outlet $= u_2$.

ωr_1 = Tangential velocity of impeller at inlet $= u_1$.

$\therefore$ Head due to pressure rise in impeller $= \frac{u_2^2}{2g} - \frac{u_1^2}{2g}$

The flow of water will commence only if head due to pressure rise in impeller $= \geq H_m$

$$\text{(or)} \quad \frac{u_2^2}{2g} - \frac{u_1^2}{2g} \geq H_m$$

For minimum speed, we must have, $\frac{u_2^2}{2g} - \frac{u_1^2}{2g} = H_m$ ——(i)

We know that, $\eta_{man} = \frac{gH_m}{V_{w_2} u_2}$

$$H_m = \eta_{man} \times \frac{V_{w_2} u_2}{g}$$

substituting this value of H_m in eqn. (i)

$$\frac{u_2^2}{2g} - \frac{u_1^2}{2g} = \eta_{man} \times \frac{V_{w_2} u_2}{g} \quad \text{——(ii)}$$

Now, $u_2 = \frac{\pi D_2 N}{60}$, $u_1 = \frac{\pi D_1 N}{60}$

substituting these values of u_1 and u_2 in (ii)

$$\frac{1}{2g}\left(\frac{\pi D_2 N}{60}\right)^2 - \frac{1}{2g}\left(\frac{\pi D_1 N}{60}\right)^2 = \eta_{man} \times \frac{V_{w_2} \pi D_2 N}{60 g}$$

Dividing by $\frac{\pi N}{g \times 60}$,

$$\frac{\pi N D_2^2}{120} - \frac{\pi N D_1^2}{120} = \eta_{man} \times V_{w_2} \times D_2$$

$$\frac{\pi N}{120}(D_2^2 - D_1^2) = \eta_{man} V_{w_2} D_2$$

$$\boxed{N = \frac{120 \times \eta_{man} \times V_{w_2} D_2}{(D_2^2 - D_1^2)}}$$

The above equation gives the minimum speed of the centrifugal pump.

→ The diameter of an impeller of a centrifugal pump at inlet and outlet are 30cm and 60cm respectively. Determine the mini starting speed of the pump if it works against a head of 30m.

Sol: Diameter of Impeller at inlet, $D_1 = 30cm = 0.30m$

Diameter of Impeller at outlet, $D_2 = 60cm = 0.60m$

Head, $H_m = 30$ cm

Let the minimum starting speed $= N$

We know the eqn,

$$\frac{u_2^2}{2g} - \frac{u_1^2}{2g} = H_m$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.6 N}{60} = 0.03141 N$$

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.3 \times N}{60} = 0.0157 N$$

$$\frac{N^2}{2g}\left[(0.03141)^2 - (0.0157)^2\right] = 30$$

$$N^2 = \frac{30 \times 2 \times 9.81}{0.03141^2 - 0.0157^2} = \frac{588.6}{0.0009866 - 0.0002465}$$

$$N^2 = 795297.9$$

$$N = \sqrt{795297.9}$$

$$N = \underline{891.8 \text{ rpm}}$$

→ **Reciprocating pumps:**

The pumps are the hydraulic machines which convert the mechanical energy is converted into hydraulic energy which is mainly in the form of pressure energy. If the mechanical energy is converted into hydraulic energy (or pressure energy) by sucking the liquid into a cylinder in which a piston is reciprocating (moving backwards and forwards), which exerts the thrust on liquid and creates in hydraulic energy (pressure energy), the pump is known as reciprocating pump.

Main parts of a Reciprocating pump:

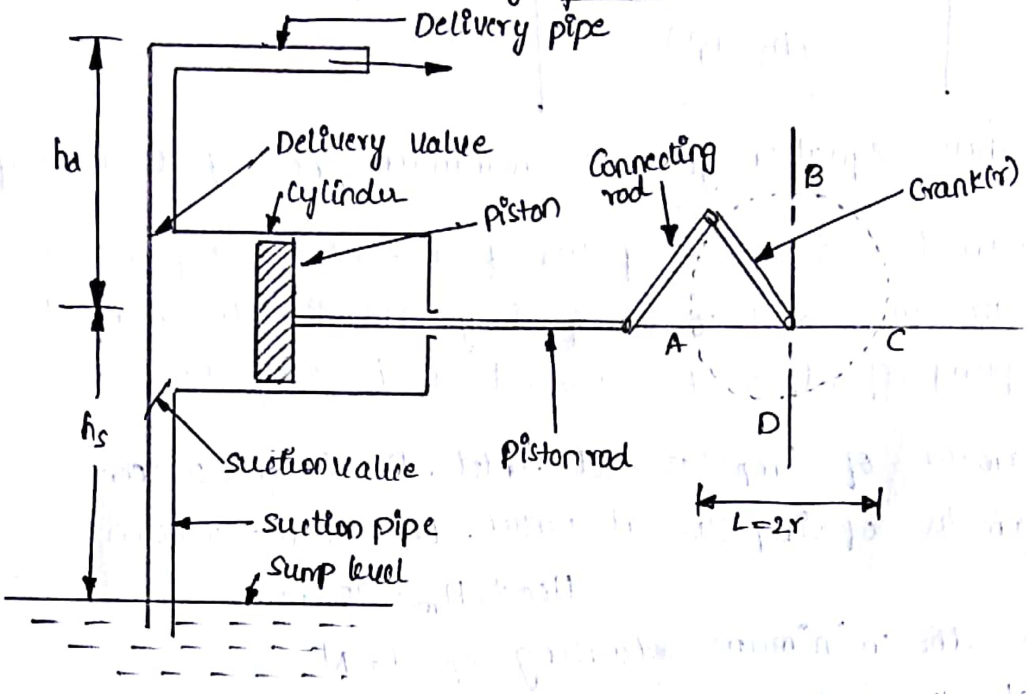


Fig: Main parts of a reciprocating pump.

1. A cylinder with a piston, piston rod, connecting rod and a crank
2. Suction pipe
3. Delivery pipe
4. Suction valve and
5. Delivery valve

Working of a reciprocating pump:

The above fig. shows a single acting reciprocating pump, which consists of a piston which moves forwards and backwards in a close fitting cylinder. The movement of piston is obtained by connecting the piston rod and crank by means of a connecting rod. The crank is rotated by means of an electric motor. Suction & delivery pipes with suction valve and delivery valve are connected to the cylinder. The suction & delivery valves are one way valves or non-return valves, which allow the water to flow in one direction only. Suction valve allows water from suction pipe to cylinder which delivery valve allows water from cylinder to delivery pipe only.

When crank starts rotating, the piston moves to and fro to cylinder

When crank is at A, the piston is at the extreme left position in the cylinder. As crank is rotating from A to C (i.e., from $\theta = 0°$ to $180°$), the piston is moving towards right in the cylinder. The movement of the piston towards right creates a partial vacuum in the cylinder. But on the surface of liquid in sump atm pressure is acting, which is more than the pressure inside the cylinder. Thus the liquid is forced in the suction pipe from sump. This liquid open the suction valve and enters the cylinder.

When crank is rotating from C to A (i.e., from $\theta = 180°$ to $360°$), the piston from its extreme right position starts moving towards left in the cylinder. The movement of the piston towards left increases the pressure of liquid inside the cylinder more than the atm pressure. Hence suction valve closes and delivery valve opens. The liquid is forced into the delivery pipe and is raised to required height.

Discharge through a Reciprocating pump:

Consider a single acting reciprocating pump.

Let D = diameter of cylinder.

A = Cross-sectional area of piston or cylinder,

$= \frac{\pi}{4} D^2$

r = Radius of crank

N = rpm of the crank

L = length of the stroke $= 2 \times r$

h_s = Height of axis of cylinder from water surface in sump.

h_d = Height of delivery outlet above the cylinder axis (also called delivery head)

Volume of water delivered in one revolution or discharge of water in one revolution.

$= $ Area $\times$ length of stroke

$= A \times L$

Number of revolution per second $= \frac{N}{60}$

$\therefore$ Discharge of pump per second,

Q = Discharge in one revolution $\times$ No. of revolution per second

$$\boxed{Q = A \times L \times \frac{N}{60} = \frac{ALN}{60}}$$

Weight of water delivered per second

$$W = \rho \times g \times Q = \frac{\rho g ALN}{60}$$

Workdone by Reciprocating pump:

Workdone by reciprocating pump per second is given by the relation as

Workdone per second = Wt. of water lifted per second x Total height through which water is lifted.

$$= W(h_s + h_d) \longrightarrow (i)$$

where, $h_s + h_d$ = Total height through which water is lifted.

Weight W is given by,

$$W = \frac{\rho g \times ALN}{60}$$

Substituting the value of W in (i)

$$\text{Workdone per second} = \frac{\rho g \times ALN}{60}(h_s + h_d)$$

$$\boxed{W.D/s = \frac{\rho g ALN}{60}(h_s + h_d)}$$

∴ Power required to drive the pump in kW

$$P = \frac{\text{Workdone per second}}{1000}$$

$$= \frac{\rho g ALN (h_s + h_d)}{1000 \times 60}$$

$$\boxed{P = \frac{\rho g ALN (h_s + h_d)}{60,000}} \text{ kW}$$

Slip of Reciprocating pump:

Slip of a pump is defined as the difference between the theotrical discharge and actual discharge of pump. The discharge of a single acting pump and a double acting pump are theoritical discharge. The actual discharge of a pump is less than the thearitical discharge due to leakage. The difference of theoritical discharge and

actual discharge is known as slip of pump.
Hence, mathematically

$$\boxed{Slip = Q_{the} - Q_{act}}$$

But slip is mostly expressed as percentage slip which is given by percentage slip $= \frac{Q_{the} - Q_{act}}{Q_{the}} \times 100$

$$\boxed{\% \; Slip = \left(1 - \frac{Q_{act}}{Q_{the}}\right) \times 100}$$

$\because C_d = \frac{Q_{act}}{Q_{the}}$

$= (1 - C_d) \times 100$

where C_d = co-efficient of discharge

* Negitive slip of reciprocating pump: slip is equal to the diff of theoritical discharge and actual discharge. If actual disharge is more than the theoritical discharge, the slip of pump will become –ve. In that case, the slip of pump is known as –ve slip.

–ve slip occurs when delivery pipe is short, suction pipe is long and pump is running at high speed.

→ Classification of Reciprocating pumps:

The reciprocating pumps may be clasified as

1. according to water being in contact with one side or both sides of the piston, and
2. according to no. of cylinders provided.

If water is in contact with one side of piston, the pump is known as single-acting. On other hand, if water is in contact with both side of piston, the pump called as double-acting. Hence classification according to contact of water:

(i) single-acting pump
(ii) Double-acting pump.

according to no. of cylinders provided.

(i) single cylinder pump.
(ii) Double cylinder pump.
(iii) Triple cylinder pump.

A single acting reciprocating pump, running at 50 rpm, delivers $0.01\ m^3/s$ of water. The diameter of piston is 200 mm and stroke length 400 mm. Determine :

(i) The theoritical discharge of pump
(ii) co-efficient of discharge and
(iii) slip and percentage slip of pump.

Sol: Speed of pump, $N = 50$ rpm

Actual discharge, $Q_{act} = 0.01\ m^3/s$

Diameter of piston, $D = 200mm = 0.20m$

$\therefore$ Area, $A = \pi/4\ (0.20)^2 = 0.031416\ m^2$

Stroke, $L = 400\ mm = 0.40\ m$

(i) Theoritical discharge for single-acting reciprocating pump is given by,

$$Q_{th} = \frac{A \times L \times N}{60} = \frac{0.031416 \times 0.40 \times 50}{60} = 0.01047\ m^3/sec$$

(ii) Co-efficient of discharge is given by,

$$C_d = \frac{Q_{act}}{Q_{th}} = \frac{0.01}{0.01047} = 0.955$$

(iii) We know the eqn.

$$\text{Slip} = Q_{th} - Q_{act} = 0.00047\ m^3/s$$

$$\text{percentage slip} = \left(\frac{Q_{th} - Q_{act}}{Q_{th}}\right) \times 100$$

$$= \left(\frac{0.01047 - 0.01}{0.01047}\right) \times 100$$

$$= 4.489\ \%$$

→ A double-acting reciprocating pump, running at 40 rpm, is discharging $1.0\ m^3$ of water per minute. The pump has a stroke of 400mm. The diameter of piston is 200mm. The delivery and suction heads are 20m and 5m respectively. Find the slip of pump and power required to drive the pump.

Sol: Speed of pump, $N = 40$ rpm

Actual discharge, $Q_{act} = 1.0\ m^3/min = \frac{1.0}{60}\ m^3/s = 0.01666\ m^3/s$

Stroke, $L = 400mm = 0.40m$

diameter of piston, $D = 200mm = 0.20m$

$\therefore$ Area, $A = \frac{\pi}{4} D^2 = 0.031416 m^2$

Suction head, $h_s = 5m$

Delivery head, $h_d = 20m$

Theoritical discharge for double acting pump is given by

$$Q_{th} = \frac{2ALN}{60} = \frac{2 \times 0.031416 \times 0.4 \times 40}{60} = 0.01675 \, m^3/s$$

We know that

$$slip = Q_{th} - Q_{act}$$
$$= 0.01675 - 0.01666$$
$$= 0.00009 \, m^3/s$$

Power required to drive the double acting pump is given by

$$P = \frac{2 \times \rho g \times ALN \times (h_s + h_d)}{60{,}000}$$

$$= \frac{2 \times 1000 \times 9.81 \times 0.031416 \times 0.4 \times 40 \times (50+20)}{60{,}000}$$

$$= \underline{4.109 \, kW}$$

Indicator Diagram:

The indictor diagram for a reciprocating pump is defined as the graph between the pressure head in the cylinder and the distance travelled by piston from inner dead centre for the complete revolution of the crank. As the maximum distance travelled by the piston is equal to the stroke length of the piston for the complete revolution. The pressure head is taken as ordinate and stroke length as abscissa.

2. Ideal indicator diagram :

The graph between pressure for a reciprocating pump is defined head in the cylinder and stroke length of the piston for one complete revolution of the crank. under conditions is known as ideal indicator diagram

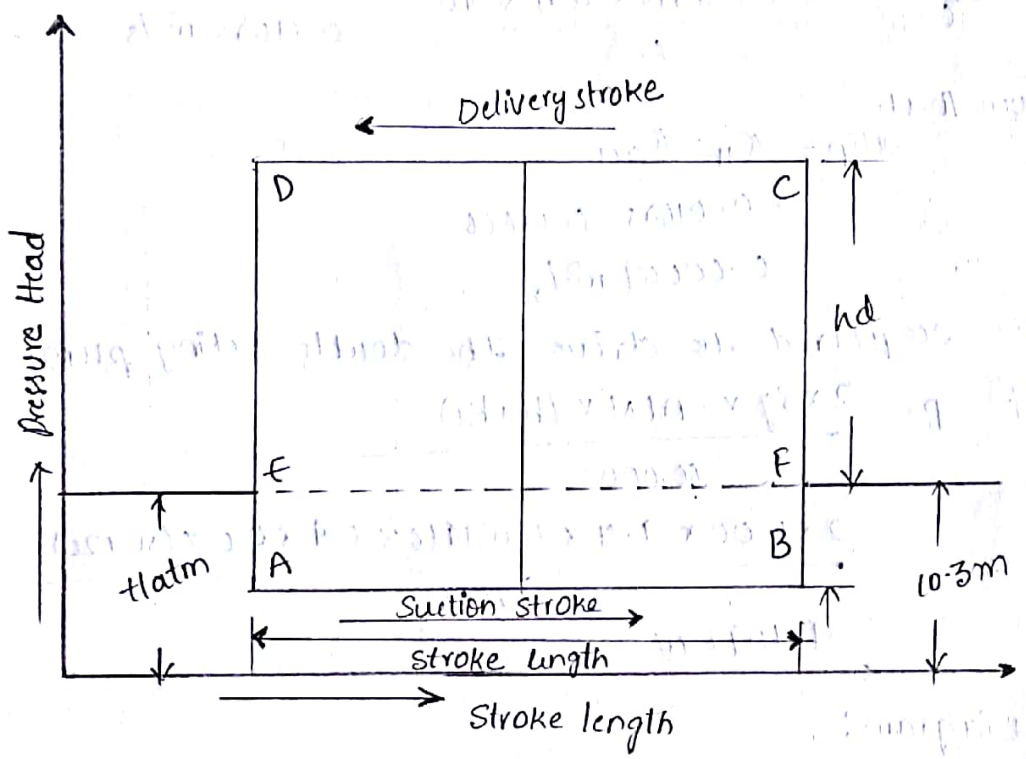


Figure shows the ideal indicator diagram in which line EF represents at atmospheric pressure head equal to 10·3m of water

Let H_{atm} = Atmospheric pressure head.
= 10·3m of water

L = length of the stroke

h_s = Suction head, and

h_d = Delivery head

During suction stroke, the pressure head in the cylinder is constant and equal the suction head (h_s). which is below.

the atmospheric pressure head (H atm) by a height of hs. The pressure head during suction stroke is represented by a horizontal line AB which is below the line BF by a height of hs

(3) During dilvery stroke, the pressure head in the cylinder is constant and equal to delivery head (hd) which is above the atmospheric head by a height of (hd) Thus, the pressure head during delivery stroke is represented by horizontal line CD which is above the line EF by a height of hd. Thus. for one complete revolution of the crank. the pressure head in the cylinder is represented by the diagram A-B-C-D-A. This diagram is known as ideal indicator diagram.

We know that the work done by the pump per second

$$= \frac{S \times g \times ALN}{60} \times (h_s + h_d)$$

$$= KL(h_s + h_d) \qquad \left[\text{Where } K = \frac{SgAN}{60} = \text{constant}\right]$$

$$\propto L \times (h_s + h_d) \qquad \text{———} \; (i)$$

By from fig area of indicator diagram

$$= AB \times BC = AB \times (BF + FC)$$

$$= L \times (h_s + h_d)$$

Substituting this value in equation (i)

Workdone by pump $\propto$ Area of indicator diagram.

Effect of Acceleration in Suction and Delivery pipes on Indicator Diagram

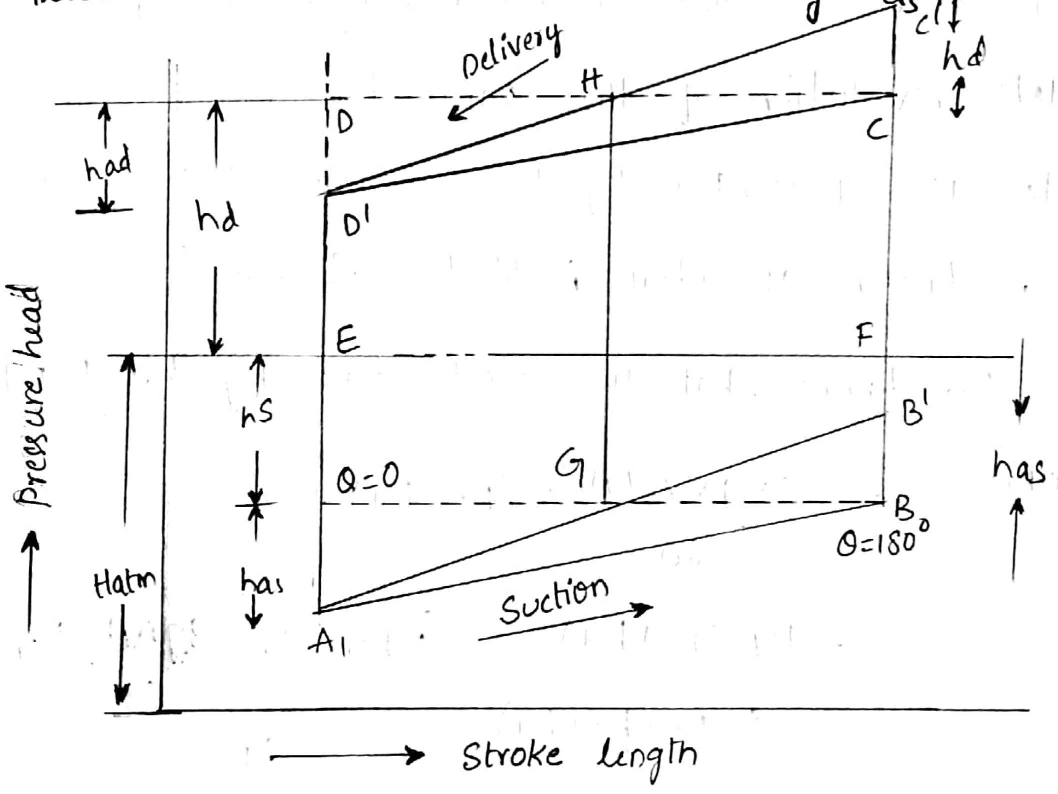
The pressure head due to acceleration in the suction pipe is given by.

$$h_{as} = \frac{l_s}{g} \times \frac{A}{a_s} \omega^2 r \cos\theta$$

When $\theta = 0°$, $\cos\theta = 1$ and $h_{as} = \frac{l_s}{g} \times \frac{A}{a_s} \omega^2 r$

When $\theta = 90°$; $\cos\theta = 1$ and $h_{as} = 0$

When $\theta = 180°$; $\cos\theta = -1$ and $h_{as} = -\frac{l_s}{g} \frac{A}{a_s} \omega^2 r$



Thus, the pressure head inside the cylinder during suction stroke will not be equal to h_s, as was the case for ideal indicator diagram. but it will be equal to the sum of h_s and h_{as}.

Let H_{atm} = Atmospheric pressure head

= 10.3 m of water.

(4) L = length of the stroke.

h_s = Suction head, and

h_d = Delivery head.

During suction stroke, the pressure head in the cylinder is constant and equal to suction head (h_s), which is below the atmospheric pressure head (H_{atm}) by a height of h_s. The pressure head during suction stroke is represented by a horizontal line AB which is below the line BF by a height of h_s.

During delivery stroke, the pressure head in the cylinder is constant and equal to delivery head (h_d) which is above the atmospheric head by a height of a (h_d) Thus, the pressure head during delivery stroke is represented by a horizontal line CD which is above the line BF by a height of h_d. Thus for one complete revolution of the crank, the pressure head in cylinder is represented by the diagram A-B-C-D-A. This diagram is known as ideal indicator diagram.

We know that the work done by the pump per

Second $= \frac{S \times g \times ALN}{60} \times (h_s + h_d)$

$= KL\,(h_s + h_d)$ [where $K = \frac{\omega g A N}{60}$ = constant]

$\propto L \times (h_s + h_d)$ —— (2)

But from fig area of indicator diagram.

$= AB \times BC = AB \times (BF + FC)$

$= L \times (h_s + h_d)$

Substituting this value in equation (i)

Workdone by pump $\propto$ Area of indicator diagram.

3. Expansion chamber Surge tank :

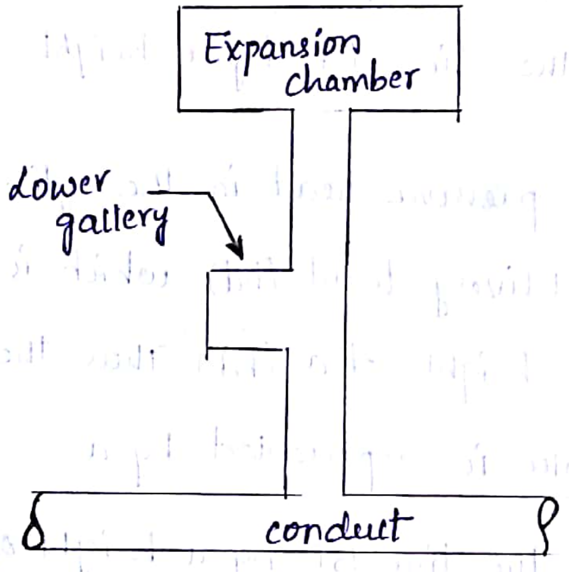


Fig. Expansion chamber Surge tank.

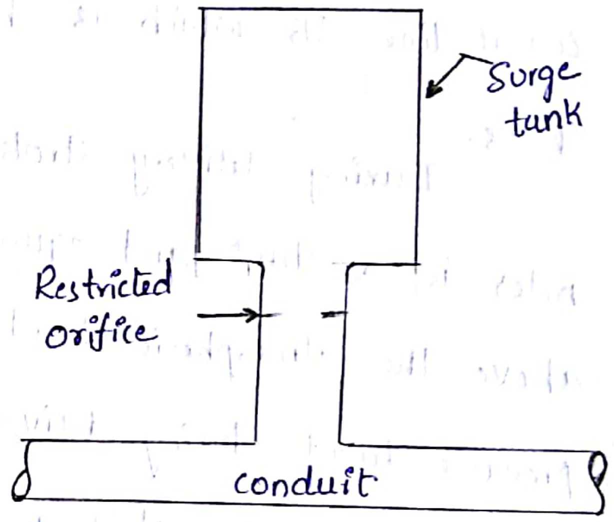


Fig :- Restricted orifice Surge tank

This type of surge tank has an expansion tank at top and expansion gallery at the bottom; these expansions limit the extreme surges. The upper expansion chamber must be above the maximum reservoir level and bottom gallery must be below the lowest steady running level in the surge tank. Besides this the intermediate shaft should have stable minimum diameter.

* 1. Restricted orifice Surge tank: It is also called throttled surge tank. The main object of providing a throttle or restricted orifice is a create an appreciable friction loss when the water is flowing to or from the tank. When the water is flowing to or from the tank. When the load on the turbine is reduced the surplus water passes through the throttle and retarding head equal to the loss due to throttle is built up in the conduit. The size of the throttle can be designed for any desired retarding head. The size of the throttle adopted is usually such as the intial retarding head is equal to the full load is rejected by the turbine (a case when there is closure of the gate valve).

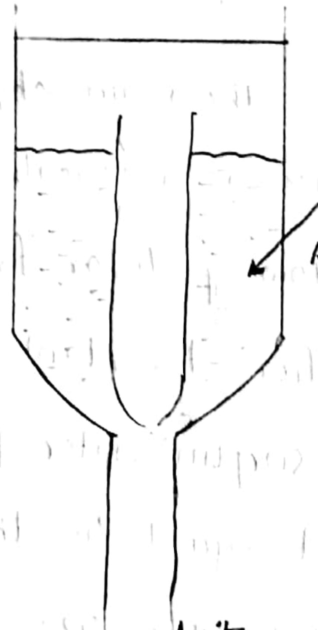
Advantage:- storage function of the tank can be separated from acceleration and retarding functions.

Disadvantages:- considered portion of water hammer pressure is transmitted directly into the low pressure conduit.

In comparisam to other types of surge tanks there less popular.

Differential Surge tank:

A differential surge tank has a riser with a small hole at its lower and through which water enters in it. The function of the surge tank depends upon the area of hole.



Hydro Electric Power Plant:

In hydro-Electric plants, energy of water is utilised to move the turbines which in turn run the electric generators. The energy of water utilised for power generation may be kinetic or potential. The kinetic energy of water is its energy in motion and is a function of mass and velocity, while the P.E is a function of the difference in level/head of water between two points. In either case continuous availability of water is a basic necessity; to ensure this, water collected in natural lakes and reservoirs of high altitudes may be utilized or water may be artificially stored by constructing dams across flowing stream. The ideal site is one in which a good system of natural lakes with substantial catchment area, exists at a high altitude. Rainfall is the primary source of water and depends upon such factors as temperature, humidity, cloudiness, wind etc. The usefulness of rainfall for power purposes further depends upon several complex factors which include its intensity, time distribution, topography of land etc, However its has been observed that only a small part of the rainfall can actually be utilized for power generation. A significant part is accounted for by direct evaporation, while another similar quantity seeps in to the soil and forms the underground storage, some water is also absorbed by vegetation. Thus only a part of water falling as rain actually flows over the ground surface as direct runoff and forms the streams which can be utilized for hydroschemes.

First hydroelectric station was probably started in America in 1882 and thereafter development took place very rapidly. In India, the first major hydro-electric development of 4.5MW capacity named as Sivasamudram Scheme in Mysore was commissioned in 1902. In 1914, a hydropower, plant named khopoli project of 50 MW capacity was commissioned in Maharastra. The hydropower capacity, upto 1947, was nearly 500 MW.

Hydropower is a conventional renewable source of energy which is clean, free from pollution and generally has a good environmental effect. However the following facts are major obstacles in the utilisation of hydropower resources:

(i) Large investments
(ii) Long gestation period
(iii) Increased cost of power transmission.

Next to thermal power, hydropower is important in regard to power generation. The hydroelectric power plants provide 30% of the total power of world. The total hydropotential of the world is about 5000 GW. In some countries (like Norway) almost total power generation is hydrobased

Application of Hydroelectric Power Plants:

Earlier hydro-electric plants have been used as exclusive source of power, but the trend is towards use of hydro-power in an inter-connected system with thermal stations. As a self-contained and independent power source a hydroplant is most effective with adequate storage capacity otherwise the maximum load capacity of the station has to be based on the year. This increases the per unit cost of installation. By interconnecting hydropower with steam, a great deal of saving in cost can be effected due to:

(i) reduction in necessary reserve capacity
(ii) diversity in construction programmes
(iii) higher utilisation factors on hydroplants, and
(iv) higher capacity factors on efficient steam plants.

In an inter-connected system the base load is supplied by hydropower when the maximum flow demand is less than the stream flow while steam supplies the peak. When stream flow is lower than the maximum demand the hydroplant supplies the peak load and steam plant the base load.

Applications of hydroelectric power plants:

→ Advantages and Disadvantage of Hydro-electric Power Plants:

* Advantages:

1. No fuel charges.
2. A hydroelectric plant is highly reliable
3. Maintenance and Operation charges are very low.
4. Running cost of the plant is low.
5. The plant has no standby losses.
6. The plant efficiency doesnot change with age.
7. It takes a few minutes to run and synchronise the plant.
8. less supervising staff is required.
9. No fuel transportation problem.
10. No ash problem and atmosphere is not polluted since no smoke is produced in the plant.
11. In addition to power generation; these plants are also used for flood control and irrigation purposes.
12. such a plant has comparitively a long life (100 to 125 years as against 20-45 years of a thermal plant).
13. The number of operations required is considerably small compared with thermal power plants.
14. The machines used in hydro-electric plants are more robust and generally run at low speeds at 300 to 400 rpm. where as the machines used in thermal plants run at a speed 3000 to 4000 rpm.
15. Therefore, there are no specialised mechanical problems or special alloys required for construction.

The cost of land is not a major problem since the hydroelectric stations are situated away from the developed areas.

• Disadvantages:

1. The initial cost of the plant is very high.
2. It takes considerably long time for the erection of such plants.

such plants are usually located in hilly areas far away from the load centre and as such they require long transmission lines and losses in them will be more.

4. Power generation by the hydro-electric plant is only dependent on the quantity of water available which in turn depends on the natural phenomenon of rain. So, if the rainfall is in time and proper and the required amount of it can be collected, the plants will function satisfactorily otherwise not.

Surge Tanks:

A surge tank is a small reservoir or tank in which the water level raises or falls to reduce the pressure swings so that they are not transmitted in full to a closed. In general a surge tank is employed to serve the following purposes.

1. To reduce the distance between the free water surface and turbine thereby reducing the water hammer effect (the water hammer is defined as the change in pressure rapidly above or below normal pressure caused by sudden changes in rate of flow through the pipe according to the demand of the prime mover) on penstock and also protect upstream tunnel from high pressure rises.
2. To serve as supply tank to the turbine when the water in the pipe is accelerating during increased load conditions and storage tank when the water is decelerating during reduced load continious.

* Types of Surge Tanks:

The different types of surge tanks in use are:

1. Simple surge tank.
2. Inclined surge tank.
3. The expansion chamber and gallery type surge tank.
4. Restricted Orifice surge tank
5. Differential surge tank.

1. Simple Surge Tank: A simple surge tank is a vertical stand pipe connected to the penstock. In the surge tank if the overflow is allowed, the rise in pressure can be eliminated but overflow surge tank is seldom satisfactory and usually uneconomical.

UNIT QUANTITIES:

In order to predict the behaviour of a turbine working under varying conditions of head, speed, output and gate opening, the results are expressed in terms of quantities which may be obtained when the head on the turbine is reduced to unity. The conditions of the turbine under unit head are such that the efficiency of the turbine remains unaffected. The following are the three important quantities which must be studied under unit head:

1. Unit Speed
2. Unit Power
3. Unit discharge

1. Unit Speed: It is defined as the speed of a turbine working under a unit head (i.e., under a head of 1m) It is denoted by N_u. The expression for unit speed (N_u) is obtained as:

Let N = speed of a turbine under a head H.
H = Head under which a turbine is working.
u = Tangential velocity.

The tangential velocity, absolute velocity of water and head on the turbine are related as

$$u \propto V$$

$$\boxed{u \propto \sqrt{H}} \text{ ——— (1)} \qquad \text{where } V \propto \sqrt{H}$$

Also tangential velocity (u) is given by

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

where D = Diameter of the turbine.

For a given turbine, the diameter (D) is constant.

$$u \propto N$$

(or) $$N \propto u$$

$$\propto \sqrt{H} \qquad \because u = \sqrt{H}$$

where k_1 = constant of proportionality

If head on the turbine becomes unity, the speed becomes unit speed (or) when $H = 1$, $N = N_u$

Substituting these values in equation (ii)

$$N_u = k_1\sqrt{1} = k_1$$

Substituting the value of k_1 in eqn. (ii)

$$N = N_u\sqrt{H}$$

$$\boxed{N_u = \frac{N}{\sqrt{H}}}$$

2. Unit Discharge: It is defined as the discharge passing through a turbine, which is working under a unit head (i.e., 1m). It is denoted by the symbol Q_u. The expression for unit discharge is given as;

Let H = Head of water on the turbine.

Q = Discharge passing through turbine when head is H on the turbine.

a = Area of flow of water.

The discharge passing through a given turbine under a head 'H' is given by,

$$Q = \text{Area of flow} \times \text{velocity}$$

But for a turbine, area of flow is constant and velocity is proportional to $\sqrt{H}$

$$Q \propto \text{Velocity} \propto \sqrt{H}$$

$$Q = k_2\sqrt{H} \quad \text{—— (iii)}$$

where k_2 = constant of proportionality

If $H = 1$, $Q = Q_u$

Substituting these values in eqn (iii)

$$Q_u = k_2\sqrt{1.0} = k_2$$

Substituting the value of k_2 in eqn (iii)

$$Q = Q_u\sqrt{H} \qquad \boxed{Q_u = \frac{Q}{\sqrt{H}}}$$

Unit Power: It is defined as the power developed by a turbine working under a unit head (i.e., H = 1m). It is denoted by the symbol 'P_u'. The expression for unit power is obtained as;

Let H = Head of water on turbine.

P = Power developed by turbine under a head of H.

Q = discharge through turbine under a head H.

The overall efficiency (η_o) is given as

$$\eta_o = \frac{\text{Power developed}}{\text{Water power}}$$

$$= \frac{P}{\frac{\rho \times g \times Q \times H}{1000}}$$

$$P = \eta_o \times \frac{\rho \times g \times Q \times H}{1000}$$

$$\propto Q \times H$$

$$\propto \sqrt{H \times H^2} \qquad \because Q \propto \sqrt{H}$$

$$\propto \sqrt{H^3}$$

$$\propto H^{3/2}$$

$$= K_3 H^{3/2} \quad \text{——— (iv)}$$

where K_3 = constant of proportionality.

when H = 1m, P = P_u

$$P = K_3 (1)^{3/2} = K_3$$

Substituting the value of K_3 in eq (iv)

$$P = P_u H^{3/2}$$

$$\boxed{P_u = \frac{P}{H^{3/2}}}$$

→ **Use of Unit Quantities: (N_u, Q_u, P_u)**

If a turbine is working under different heads the behaviour of the turbine can be easily known from the values of unit quantities, i.e., from the values of unit speed, unit discharge and unit power.

Let H_1, H_2, ---- are the heads under which a turbine works,

N_1, N_2, --- = the corresponding speeds,

$Q_1, Q_2, ---$ = the discharge

$P_1, P_2, ----$ = are the power developed by the turbine.

We know that $N_u = \frac{N}{\sqrt{H}}$; $Q_u = \frac{Q}{\sqrt{H}}$; $P_u = \frac{P}{H^{3/2}}$

using these relations.

$$N_u = \frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}}$$

$$Q_u = \frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}}$$

$$P_u = \frac{P_1}{\sqrt{H_1}} = \frac{P_2}{\sqrt{H_2}}$$

Hence, if the speed, discharge and power developed by a turbine under a head are known, then by using above relations, the speed, the discharge and power developed by the same turbine under a different head can be obtained easily.

1) A turbine develops 9000 kW when running at 100 rpm. The head on the turbine is 30m. If the head on the turbine is reduced to 18m, determine the speed and power developed by the turbine.

Sol: Power developed $P_1 = 9000$ kW.

speed $N_1 = 100$ rpm.

Head $H_1 = 30$m

let for a head $H_2 = 18$m

Speed $= N_2$

Power $= P_2$

We know that $\frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}}$

$$N_2 = N_1 \times \frac{\sqrt{H_2}}{\sqrt{H_1}} = 100 \times \frac{\sqrt{18}}{\sqrt{30}} = \frac{100 \times 4.2426}{5.4772} = 77.46 \text{ rpm.}$$

Also we know that $\frac{P_1}{H_1^{3/2}} = \frac{P_2}{H_2^{3/2}}$

$$P_2 = P_1 \times \left(\frac{H_2}{H_1}\right)^{3/2} = 9000 \times \left(\frac{18}{30}\right)^{3/2} = \frac{687307.78}{164.316}$$

$$= 4182.84 \text{ kW}$$

A turbine develops 500 kW power under a head of 100 metres at 200 rpm. What would be its normal speed and output under a head of 81 metres?

Sol: power, $P_1 = 500$ kW

speed, $N_1 = 200$ m.

Head, $H_1 = 100$ m.

For a head, $H_2 = 81$ m

N_2 = speed

P_2 = power

We know that $\frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}}$

$$N_2 = N_1 \times \frac{\sqrt{H_2}}{\sqrt{H_1}} = 200 \frac{\sqrt{81}}{\sqrt{100}} = \frac{9}{10} \times 200 = 180 \text{ rpm}$$

also we know that $\frac{P_1}{H_1^{3/2}} = \frac{P_2}{H^{3/2}}$

$$P_2 = P_1 \times \frac{H_2^{3/2}}{H_1^{3/2}} = 500 \times \frac{(81)^{3/2}}{(100)^{3/2}} = 500 \times \frac{729}{1000} = 364.5 \text{ kW}.$$

3) A turbine is to operate under a head of 25m at 200 rpm. The discharge is 9 cumec. If the efficiency is 90%. determine the performance of the turbine under a head of 20m.

Sol: Head on turbine, $H_1 = 25$ m.

speed, $N_1 = 200$ rpm.

Discharge, $Q_1 = 9$ m³/s.

Overall efficiency, $\eta_o = 90\% = 0.90$

Performance of the turbine under a head, $H_2 = 20$m. means to find the speed, discharge and power developed by the turbine when working under the head of 20m.

Let for the head, $H_2 = 20$m,

N_2 = speed

P_2 = power

Q_2 = discharge

We know the relation, $\eta_o = \frac{P}{W.P} = \frac{P_1}{\frac{\rho \times g \times Q_1 \times H_1}{1000}}$

$$P_1 = \frac{\eta_o \times \rho \times g \times Q_1 \times H_1}{1000}$$

$$= \frac{0.90 \times 1000 \times 9.81 \times 9 \times 25}{1000}$$

$$P_1 = 1986.5 \text{ kW}$$

using the relation, $\frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}}$

$$N_2 = N_1 \times \frac{\sqrt{H_2}}{\sqrt{H_1}} = 200 \times \frac{\sqrt{20}}{\sqrt{25}} = \underline{178.88 \text{ rpm}}$$

and $\frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}}$

$$Q_2 = Q_1 \times \frac{\sqrt{H_2}}{\sqrt{H_1}} = 9 \times \frac{\sqrt{20}}{\sqrt{25}} = \underline{8.05 \text{ m}^3/\text{s}}$$

and $\frac{P_1}{H_1^{3/2}} = \frac{P_2}{H_2^{3/2}}$

$$P_2 = P_1 \times \left(\frac{H_2}{H_1}\right)^{3/2} = 1986.5 \times \left(\frac{20}{25}\right)^{3/2} = \underline{1421.42} \text{ kW}.$$

4) A pelton wheel is revolving at a speed of 190rpm and develops 5150.25 kW when working under a head of 220m with an overall efficiency of 80%. Determine unit speed, unit discharge and unit power. The speed ratio for the turbine is given as 0.47. Find the speed, discharge and power when this turbine is working under a head of 140m.

Sol: Speed, $N_1 = 190$ rpm

Power, $P_1 = 5150.25$ kW

Head, $H_1 = 220$ m

Overall efficiency, $\eta_o = 80\% = 0.80$

Speed ratio $= 0.47$

New head of water, $H_2 = 140$ m

Overall efficiency is given as;

$$\eta_o = \frac{P_1}{\frac{\rho \times g \times Q_1 \times H_1}{1000}} = \frac{1000 P_1}{\rho \times g \times Q_1 \times H_1}$$

$$Q_1 = \frac{5150.25}{0.80 \times 1000 \times 9.81 \times 220}$$

$$Q_1 = 2.983 \text{ m}^3/s$$

We know that the unit speed is given by,

$$N_u = \frac{N_1}{\sqrt{H_1}} = \frac{190}{\sqrt{220}} = \underline{12.81 \text{ rpm}}$$

Unit discharge is given by,

$$Q_u = \frac{Q_1}{\sqrt{H_1}} = \frac{2.983}{\sqrt{220}} = \underline{0.201} \text{ m}^3/\text{sec}$$

unit power is given by,

$$P_u = \frac{P_1}{(H_1)^{3/2}} = \frac{5150.25}{(220)^{3/2}}$$

$$= \underline{1.578 \text{ KW}}$$

When the turbine is working under a new head of 140 m, the speed, discharge and power are given by,

For speed, $\frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}}$

$$N_2 = N_1 \times \frac{\sqrt{H_2}}{\sqrt{H_1}} = 190\sqrt{\frac{140}{220}} = \underline{151.56} \text{ rpm}.$$

For discharge, $\frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}}$

$$Q_2 = Q_1\sqrt{\frac{H_2}{H_1}} = 2.983\sqrt{\frac{140}{220}} = 2.3\underline{79 \text{ m}^3}/\text{sec}$$

For power, $\frac{P_1}{H_1^{3/2}} = \frac{P_2}{H_2^{3/2}}$

$$P_2 = P_1 \times \left(\frac{H_2}{H_1}\right)^{3/2} = 5150.25\left(\frac{140}{220}\right)^{3/2} = \underline{2614.48} \text{ KW}.$$

5) A pelton wheel is supplied with water under a head of 35m at the rate of 40.5 kilo litre/min. The bucket deflect the jet through an angle of 160° and the mean bucket speed is 13 m/s. Calculate the power and hydraulic efficiency of the turbine.

Sol: Net head, $H = 35$ m

Discharge, $Q = 40.5$ kilo litre/min

$= 40.5 \times 1000$ lit/min.

$$= \frac{40.5 \times 1000}{1000} \, m^3/min$$

$$= 40.5 \, m^3/min$$

$$= \frac{40.5}{60} \, m^3/sec$$

$$= 0.675 \, m^3/sec$$

Angle of deflection $= 160^\circ$.

$\therefore$ Angle $\phi = 180^\circ - 160^\circ = 20^\circ$

Mean bucket speed, $u = u_1 = u_2 = 13 \, m/s$.

Calculate (i) power at runner

(ii) hydraulic efficiency

Taking the value of $C_v = 1.0$

The velocity of jet, $V_1 = C_v\sqrt{2gH} = 1 \times \sqrt{2 \times 9.81 \times 35}$

$$= 26.2 \, m/s$$

$V_{r1} = V_1 - u_1 = 26.2 - 13 = 13.2 \, m/s$

$V_{w1} = V_1 = 26.2 \, m/s$

$V_{r2} = V_{r1} = 13.2 \, m/s$

$V_{w2} = V_{r2} \cos\phi - u_2$

$= 13.2 \cos 20 - 13$

$= 12.554 - 13 = -0.446 \, m/s$

(i) Power at runner:

We get the workdone by the jet on the runner per second.

Workdone /s $= \rho \times a \times V_1 [V_{w1} + V_{w2}] \times u$

$= \rho Q [V_{w1} + V_{w2}] \times u$

$= 1000 \times 0.675 [26.2 + (-0.446)] \times 13$

$= 225991 \, N\text{-}m/s$

$= 225991 \, W$

$= 225.991 \, kW$

$\therefore$ Power at runner $= 225.991 \, kW$.

(ii) Hydraulic efficiency:

Input power $= \frac{\rho \times g \times Q \times H}{1000} = \frac{1000 \times 9.81 \times 0.675 \times 35}{1000}$

$= 231.761 \, kW$

Power at runner = 225.991 kW

∴ Hydraulic efficiency = $\frac{\text{Power at runner}}{\text{Input power}}$

$$= \frac{225.991}{231.761}$$

$$= 0.975$$

$$= 97.5\%$$

Characteristic curves of Hydraulic Turbines:

Characteristic curves of a hydraulic turbine are the curves, with the help of which the exact behaviour and performance of the turbine under different working conditions, can be known. These curves are plotted from the results of the tests performed on the turbine under different working conditions.

The important parameters which are varied during a test on a turbine are;

1. Speed (N)
2. Head (H)
3. Discharge (Q)
4. power (P)
5. Overall efficiency (η_o)
6. Gate opening.

Out of the above six parameters, three parameters namely speed (N), Head (H) and discharge (Q) are independent parameters.

Out of the three independent parameters, N, H, Q are one of the parameter is kept constant say H and the variation of the other four parameters with respect to any one of the remaining two independent variables say N and Q are plotted and various curves are obtained. The curves are called characteristic curves.

1. Main characteristic curves (or) constant Head curve.
2. Operating characteristic curves (or) Constant speed curve.
3. Muschel curves (or) constant Efficiency curve.

*1. Main characteristic curves (or) constant Head curves:

Main characteristic curves are obtained by maintaining a constant head and a constant gate opening (G.O) on the turbine. The speed of the turbine is varied by changing load on the turbine. For

..., the corresponding values of the power (P) and discharge (Q) are obtained. Then the overall efficiency (η_o) for each value of the speed is calculated. From these readings the value of unit speed (N_u) unit power (P_u) and unit discharge (Q_u) are determined. Taking N_u as abscissa, the values of Q_u, P_u, P and η_o are plotted as shown fig. By changing the gate opening the values of Q_u, P_u and η_o are plotted.

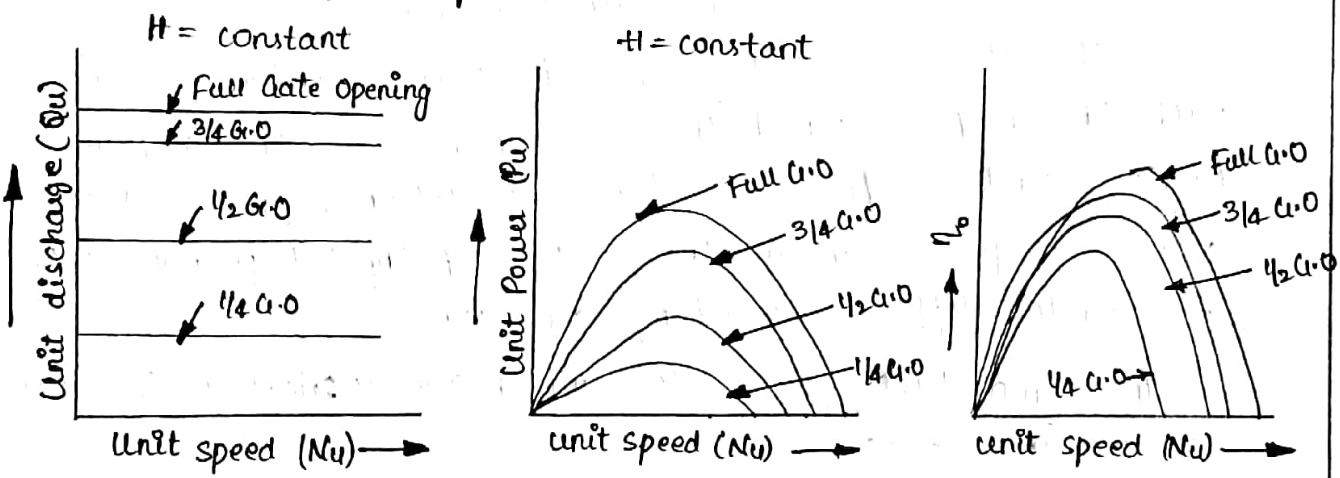
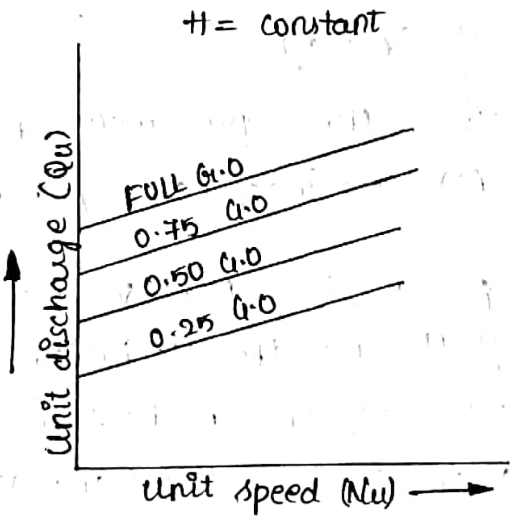
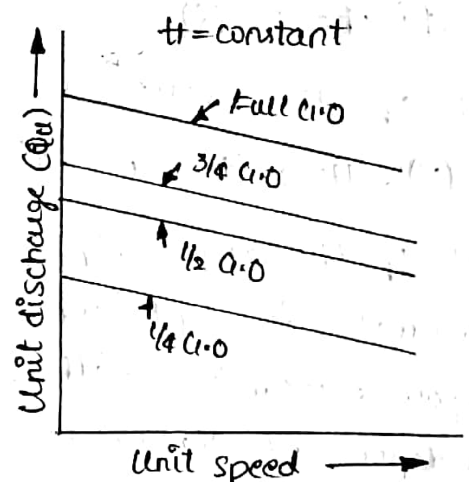


Fig: Main characteristic curves for a pelton wheel.



(a) For Kaplan Turbine



(b) For Franscis Turbine

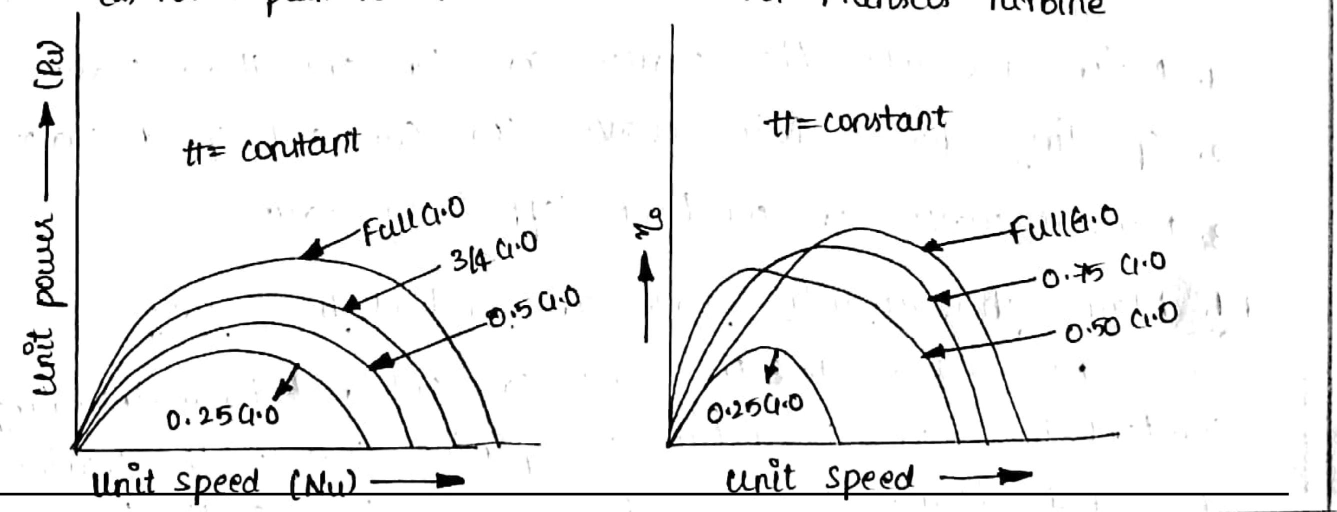


Fig: Main characteristic curves for reaction turbine

2. Operating Characteristic Curves or Constant speed curves:

Operating characteristic curves are plotted when the speed on the turbine is constant. In case of turbines, the head is generally constant. These are three independent parameters namely, N, H and Q. For operating characteristics N and H are constant and hence the variation of power and efficiency with respect to discharge are plotted. The power and efficiency curves will be slightly away from the origin on the x-axis, as to become initial friction certain amount of discharge will be required.

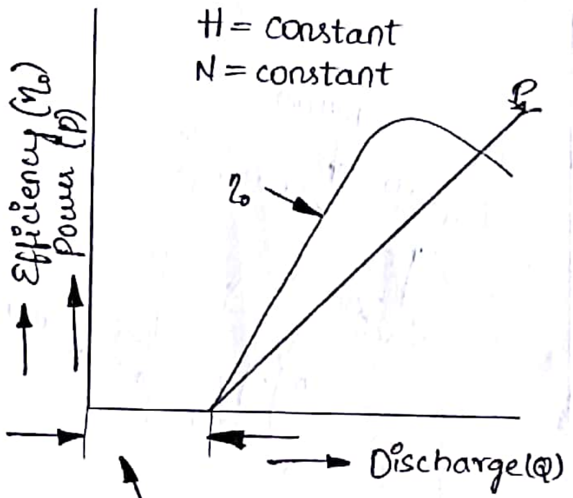


Fig: Operating characteristic curves.

3. Constant Efficiency curves or Muschel curves or Iso-efficiency curves:

These curves are obtained from the speed vs efficiency and speed vs. discharge curves for different gate openings. For a given efficiency from the Nu vs η_o curves, there are two speeds. From the Nu Vs Qu curves, corresponding to two values of speeds there are two values of discharge. Hence for a given efficiency there are two values of discharge for a particular gate opening. This means for a given efficiency there are two values of speeds, a two values of discharge for a given gate opening. If the efficiency is maximum there is only one value. These two values of speed and two values of discharge corresponding to a particular G.O are plotted as shown in fig. The procedure is repeated for diff. gate openings and the curves Q vs. N are plotted. The points having the same efficiencies are joined. The curves having same efficiency are called iso-efficiency curves. These curves are helpful for determining the zone of constant efficiency and for predicting the performance of the turbine at various efficiencies.

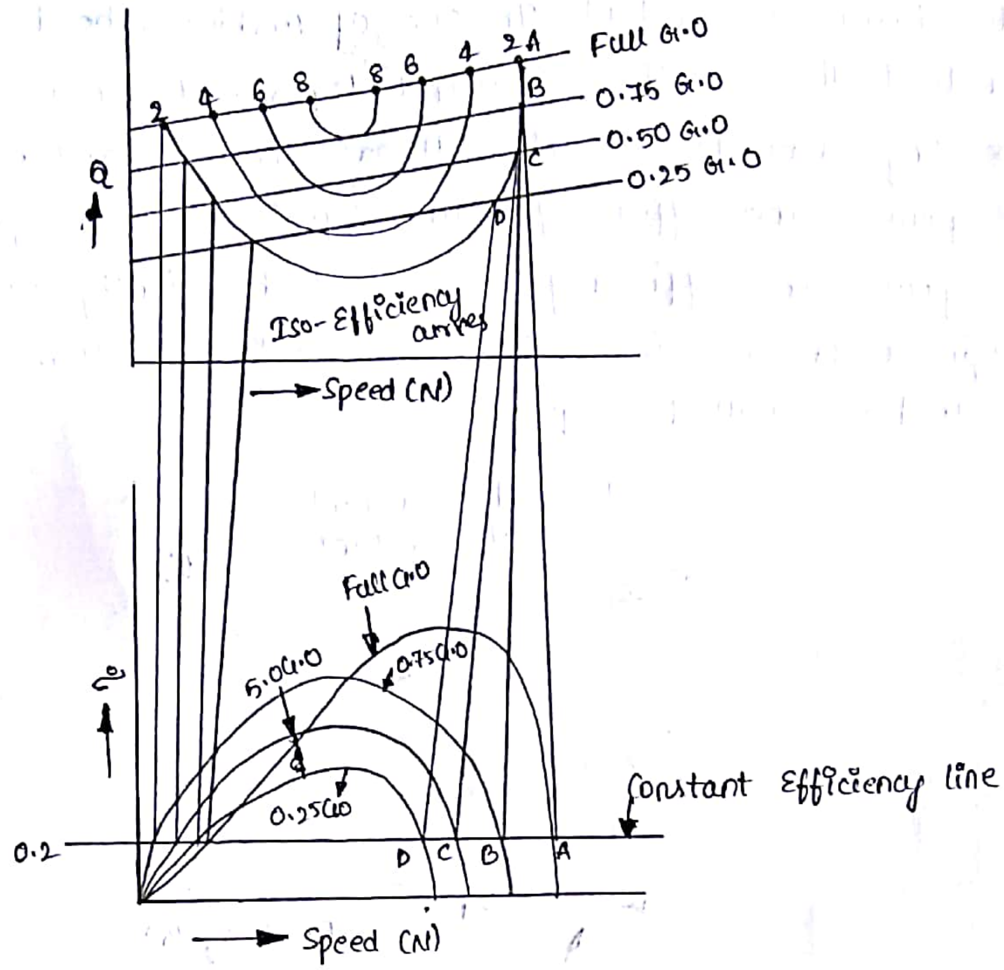


Fig: Constant Efficiency curve.

Governing of Turbines:

The governing of a turbine is defined as the operation by which the speed of the turbine is kept constant under all conditions of working. It is done automatically by means of a governor, which regulates the rate of flow through the turbines according to the changing load conditions on the turbine.

Governing of a turbine is necessary as a turbine is directly coupled to an electric generator which is required to run at constant speed under all fluctuating load conditions. The frequency of power generation by a generator of constant number of pair of poles under all varying conditions should be constant. This is only possible when the speed of the generator, under all changing load condition is constant. The speed of the generator will be constant, when the speed of the turbine (which is coupled to the generator) is constant.

When the load on the generator decreases the speed of the generator increases beyond the normal speed (constant speed). Then the speed of the turbine also increases beyond the normal speed (constant speed). When the speed of the turbine also increases beyond the normal speed. If the turbine or the generator is to run at constant (normal) speed, the rate of flow of water to the turbine should be decreased it to the speed becomes normal. This process by which the speed of the turbine (and hence of generator) is kept constant under varying condition of load is called governing.

Governing of pelton Turbine (Impulse Turbine):

Governing of pelton turbine is done by means of oil pressure governor, which consists of the following as:

1. Oil sump.
2. Gear pump also called oil pump, which is driven by the power obtained from turbine shaft.
3. The servomotor also called the relay cylinder.
4. The control valve or the distribution valve or relay valve.
5. The centrifugal governor or pendulum which is driven by belt or gear from the turbine shaft.
6. Pipes connecting the oil sump with the control valve and control valve with servomotor and
7. The spear rod or needle.

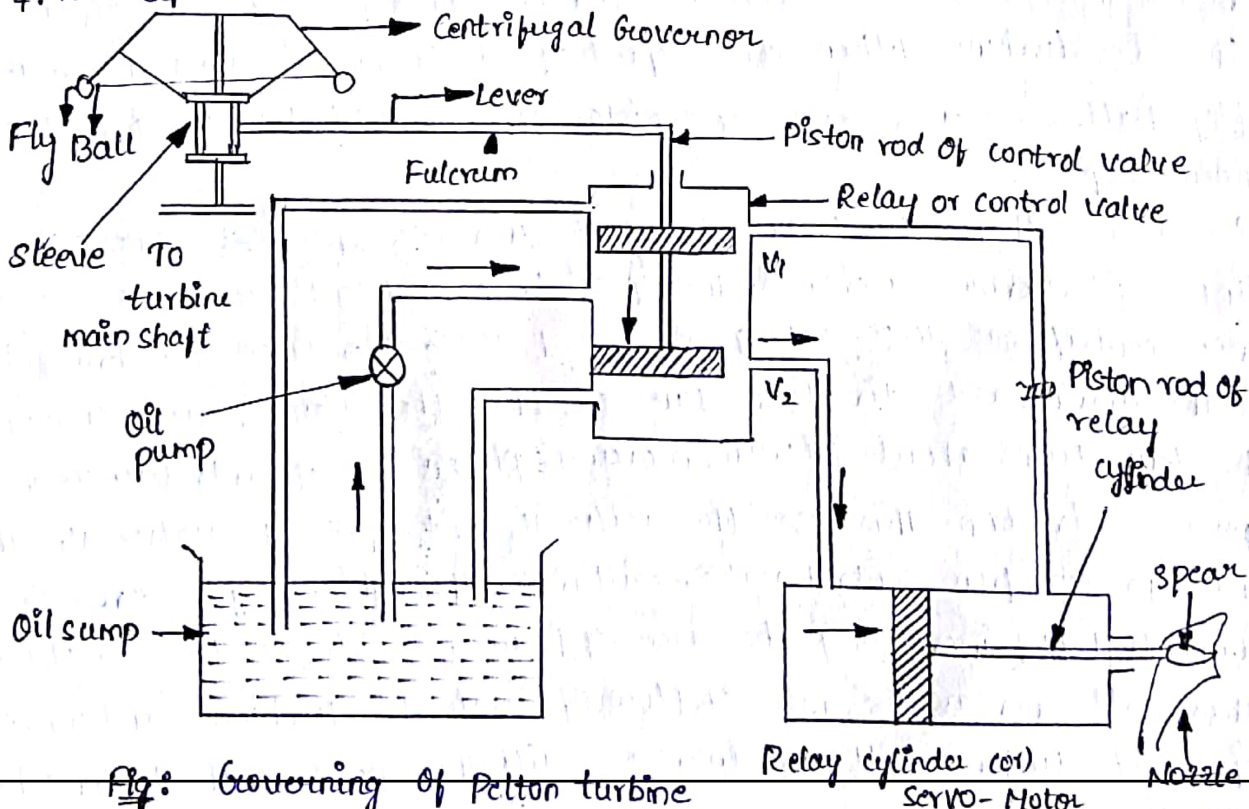


Fig: Governing of Pelton turbine

we shows the position of the piston in the relay cylinder, position of control or relay valve and fly-balls of the centrifugal governor, when the turbine is running at the normal speed.

When the load on the generator decreases, the speed of generator increases. This increases the speed of the turbine beyond the normal speed. The centrifugal governor, which is connected to the turbine main shaft, will be rotating at an increased speed. Due to increase in the speed of centrifugal governor, the fly-balls move upwards due to increased centrifugal force on them. Due to the upward movement of fly-balls, the sleeve will also move upward. A horizontal lever, supported over a fulcrum, connects the sleeve and the piston rod of control valve. As the sleeve moves up, the lever turns about the fulcrum and the piston rod of the control valve moves downward. This close the valve V_1 and opens the valve V_2.

The oil, pumped from the oil pump to the control valve or relay valve, under pressure will flow through the valve V_2 to the servometer (or relay cylinder) and will exert force on the face A of the piston rod and ~~spear will move~~ of relay cylinder. The piston along with piston rod and spear will move towards right. This will decrease the area of flow of water at the outlet of the nozzle. This decrease of area of flow will reduce the rate of flow of water the turbine which consequently reduces the speed of the turbine. When the speed of the turbine becomes normal, the fly-balls, sleeve, lever and piston rod of control valve come to its normal piston.

When load on generator increases, speed of generator and hence of turbine decreases. The speed of centrifugal governor also decreses and hence centrifugal force acting on fly-balls also reduces. This brings the fly-balls in the downward direction. Due to this, the sleeve moves downward and the lever turns about fulcrum, moving piston rod of control valve in the upward direction. This closes the valve V_2 and opens the valve V_1. The oil under pressure from control valve, will move through with piston rod and spear towards left, increasing the area of flow of water at the outlet of nozzle. This will increase the rate of flow of water to turbine and consequently, the speed of turbine will also increases till the speed of the turbine becomes normal.

Cavitation:

The formation, growth and collapse of vapour filled cavities or bubbles in a flowing liquid due to local fall in fluid pressure is called cavitation. When the pressure at any point in a flow field equals the vapour pressure of the liquid at that temperature vapour cavities (bubbles of vapour) begin to appear. It is presumed that a vapour cavity is formed around a dust nuclei which is in the liquid (The vapour pressure values of water at 15°c and 20°c are 1·74m and 2·38m of water column absolute). The cavities thus formed, due to motion of liquid, are carried to high pressure regions where the vapour condenses and they suddenly collapse. The adjoining liquid rushes with a very great velocity and hence with very great force to occupy the empty spaces thus created, causes series of violet, irregular, spherical shock waves. When these irregular implosions occur on the metallic surface, they produce noise and vibration.

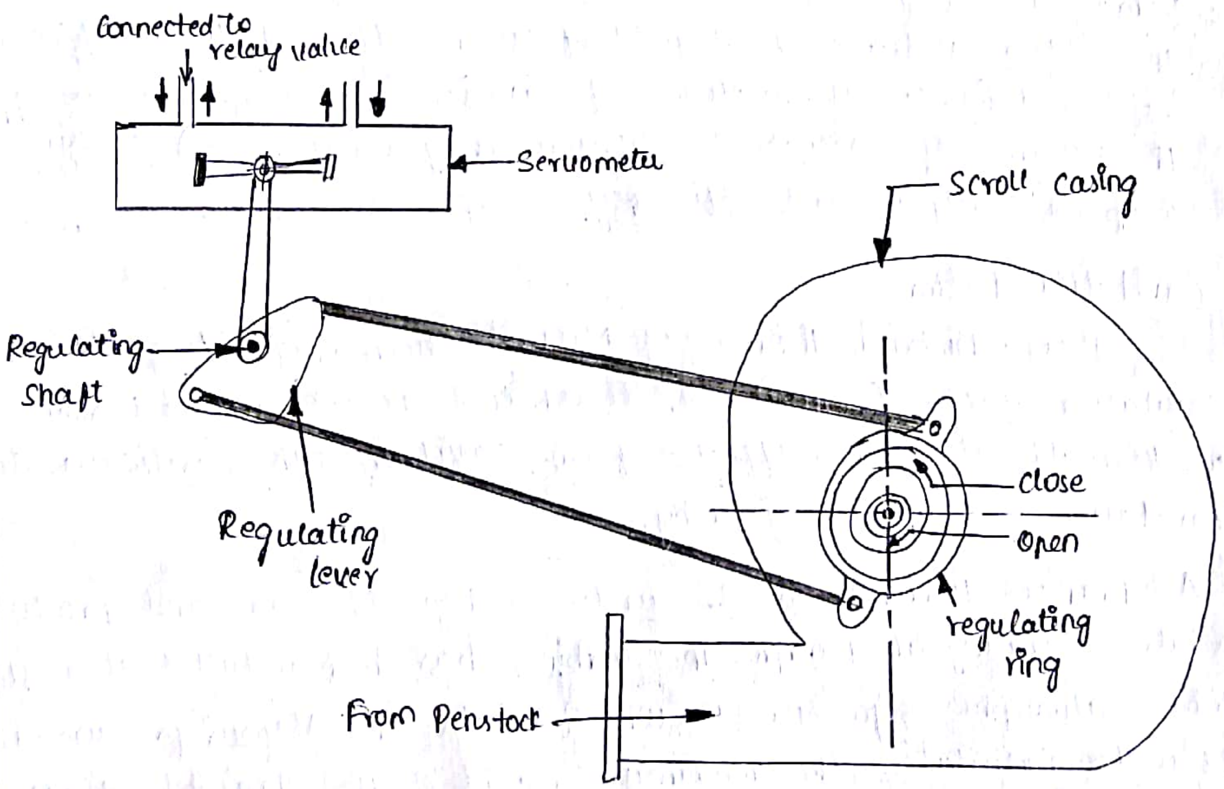


Fig: Governing Mechanism for reaction turbines.

When the cavities collapse the collapsing pressure is of the order of 100 times the atmospheric pressure on the surface of a body, due to repeated hammering action, the metal particles gives way ultimately ~~due to fatigue and indentations are formed; this erosion of~~

material is called pitting.

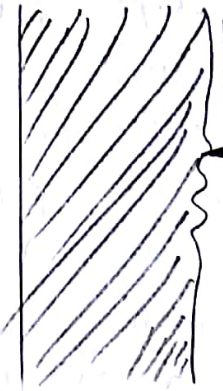


Fig: Pitting action on the inside surface (shown in large scale)

In reaction turbines the cavitation may occur at the runner exit or the draft tube inlet where the pressure is negitive. The hydraulic machinery is affected by the cavitation in the following three ways:

1. Roughening of all the surfaces takes place due to loss of material caused by pitting.
2. Vibration of parts is caused due to irregular collapse of cavities.
3. The actual volume of liquid flowing through the machine is reduced (since the volume of cavities is many times more than the volume of water from which they are formed) causing sudden drop in output and efficiency.

* Cavitation Factor:

Prof. Dietrich Thoma of Munich (Germany) suggested a cavitation factor (sigma) to determine the zone where turbine can work without being affected from cavitation. The critical value of cavitation factor(σ_c) is given by,

→ A Francis Turbine works under a head of 25m and produces 1180 kW while running at 120 rpm. The turbine has been installed at a station where atmospheric pressure is 10m of water and vapour pressure is 0.2m of water. Calculate the maximum height of the straight draft tube for the turbine.

Sol: Head under which the turbine works, $H = 25m$

power output $P = 11800KW$

speed of the turbine, $N = 120 rpm$

atmospheric pressure, $P_a = 10m$ of water.

Maximum height of the draft tube: (H_s)

$$\sigma_c = 0.625 \left(\frac{N_s}{380.78}\right)^2$$

$$= 0.625 \left(\frac{233.2}{380.78}\right)^2$$

$$= 0.2344$$

Also, $\sigma_c = \dfrac{H_a - H_v - H_s}{H}$

$$0.234 = \frac{10 - 0.2 - H_s}{25}$$

$$0.234 \times 25 = 10 - 0.2 - H_s$$

$$H_s = 10 - 0.2 - (0.234 \times 25)$$

$$= \underline{3.94\,m}$$

Hence, max. permissible height of the draft tube, $H_s = \underline{3.94\,m}$

Selection of Hydraulic Turbines:

The following points should be considered while selecting right type of hydraulic turbine for hydro-electric power plant:

(•) 1. **Specific speed:**

High specific speed is essential where head is low and output is large, because otherwise the rotational speed will be low which means cost of turbo-generator and power-house will be high. On the other hand, there is practically no need of choosing a high value of specific speed for high installations, because even with low specific speed high rotational speed can be attained with medium capacity plants.

(•) 2. **Rotational Speed:**

It depends on specific speed. Also rotational speed of an electric generator with which the turbine is to be directly coupled, depends on the frequency and no. of pair of poles. The value of specific speed adopted should be such that it will give the synchronous speed of the generator.

(•) 3. **Efficiency:**

The turbine selected should be such that it gives the highest overall efficiency for various operating conditions.

(c) 4. Partload Operation:

In general the efficiency at partloads and overloads is less than normal. For the sake of economy the turbine should always run with maximum possible efficiency to get more revenue.

When the turbine has to run at part or overload conditions Deriaz turbine is employed. similarly, for low heads, kaplan turbine will be useful for such purposes in place of propeller-turbine.

$$\boxed{\sigma_c = \frac{(H_a - H_v) - H_s}{H}}$$

Where H_a = Atmospheric pressure head in metres of water.

H_v = Vapour pressure head in metres of water corresponding to the water temperature.

H = Working head of turbine (difference between head race and tail level in metres), and

H_s = suction pressure head (or height of turbine outlet above tail race level in metres)

The values of critical factor depends upon the specific speed of the turbine.

The value for σ_c for different materials may be determined with the help of following empirical relations:

- For Francis turbine : $\sigma_c = 0.625\left[\frac{N_s}{380.78}\right]^2$
- For propeller turbine : $\sigma_c = 0.28 + \left[\frac{1}{7.5}\left(\frac{N_s}{380.78}\right)^3\right]$
- For kaplan turbine : Values of σ_c obtained by above equation (propeller turbine) should be increased by 10%.

Where N_s = Specific speed in rpm

* Suction specific speed $(N_s)_{suc}$: In addition to thomas criterion the consideration of suction specific speed provides very useful criterion for establishing similarity in respect of cavitation in the turbines. The suction speed may be defined as speed of a geomatrically similar turbine such that when it is developing a power equal to 1kW the total suction head there is equal to 1m (absolute units). It can be proved that specific speed is given by,

$$N_s = \frac{NSP}{(\sigma H)^{4/5}}$$

$$\sigma = \left[\frac{N_s}{(N_s)_{suc}}\right]^{4/5}$$

The above equations give the relation between the two parameters σ and $(N_s)_{suc}$, both of which are useful for establishing a similarity in respect of cavitation in the model and prototype turbines. The concept of suction speed, however, is more commonly used in pumps.

* Methods to avoid cavitation:

The following methods may be used to avoid cavitation:

1. Runner / turbine may be kept under water. But it is not advisable as the inspection and repair of the turbine is difficult. The other method to avoid cavitation zone without keeping the runner under water is to use the runner of low specific speed.

2. The cavitation free runner may be designed to fulfil the given conditions with extensive research.

3. It is possible to reduce the cavitation effect by selecting materials which resist better the cavitation effect. The cast steel is better than cast iron and stainless steel or alloy steel is still better than cast steel.

4. The cavitation effect canbe reduced by polishing the surface, That is why the cast steel runner and blades are coated with stainless steel.

5. The cavitation may be avoided by selecting a runner of proper specific speed for given head.

(•) 5. Cavitation:

The installation of water turbines of reaction type over the tail race is affected by cavitation. The critical value of cavitation factor must be obtained to see that the turbine works in safe zone. Such a value of cavitation factor also affects the design of turbine, especially of kaplan, propeller and bulb types.

(•) 6. Disposition of turbine shaft:

Experience has shown that the vertical shaft arrangement is better of large sized reaction turbines, therefore, it is almost universally adopted. In case of large size impulse turbines, horizontal shaft arrangement is mostly employed.

(•) 7. Head:

(i) Very high Heads: (350m and above)

For heads greater than 350m, pelton turbine is generally employed and there is practically no choice except in very special cases.

(ii) High Heads: (150m and 350m)

In this range either pelton (or Francis turbine may be employed. For high specific speeds Francis turbine is more compact and economical than the pelton turbine which for same working conditions would have to be much bigger and rather cumbersome

(iii) Medium Heads: (60m and 150m)

A Francis turbine is usually employed in this range. Whether a high or low specific speed unit would be used depends on the selection of the speed.

(iv) Low Heads: (below 60m)

Between 30 and 60m heads both Francis and Kaplan turbines may be used. The latter is more expensive but yield a higher efficiency at partloads and overloads. It is therefore preferable for variable loads. Kaplan turbine is generally employed for heads under 30m.

Propeller turbines are however, commonly used for heads upto 15m. They are adopted only when there is practically no load variations.

(v) Very low heads:

For very low heads from bulb turbines are employed these days. although Kaplan turbines can also be used for heads from 2m to 15m. but they are not economical.

Table: Criteria for Selection of Turbines

S.No	Type of Turbine	Head H (m)	Specific Speed (Ns)	Speed ratio (Ku)	Maximum Hydraulic efficiency(%)	Remarks
1.	Pelton: 1 Jet	upto 2000	12 to 30	0.43 to 0.48	89	Employed for very high head
	2 Jets	upto 1500	17 to 50			
	4 Jets	upto 500	24 to 70			
2.	Francis: High head	upto 300	80 to 50	0.6 to 0.9	93	Full load efficiency high partload efficiency lower than pelton wheel
	Medium head	50 to 150	150 to 250			
	Low head	30 to 60	250 to 400			
3.	Propeller and kaplan	4 to 60	300 to 1000	1.4 to 2	93	High part load efficiency; high discharge with low head.
4.	Bulb or tubular turbines	3 to 10	1000 to 1200	6 to 8	91	Employed for very low head-tidal power plants

Overall efficiency (η_o) of all turbines = 85 percent