# Learning Material

Unit –I

**Objectives**

- To study the working of various heat engines and analyze the performance.

**Syllabus**

**Heat Engines**: IC engine components, classification, SI and CI engines, Four stroke an Two stroke engines, Valve and port time diagrams, comparison of 2-stroke and 4-stroke, SI and CI engines.

**Testing and Performance of I.C. Engines:**

Measurement of fuel consumption, air consumption, break power, frictional power and indicated power, performance tests, heat balance sheet.

At the end of the chapter student is able to

- Identify and explain the function of various parts of an engine
- Understand the working of an internal combustion engine
- Compare S.I engine and C.I Engine, 2-stroke and 4-stroke engines.
- Draw the actual valve and port timing diagrams.
- Determine the efficiency and other performance parameters of the engine.

## 1. INTRODUCTION TO HEAT ENGINE

It is a device, which delivers the mechanical energy as output, by the expansion of the high temperature and high pressure gases obtained by the combustion of a substance called fuel, in the presence of oxidizer (like oxygen).

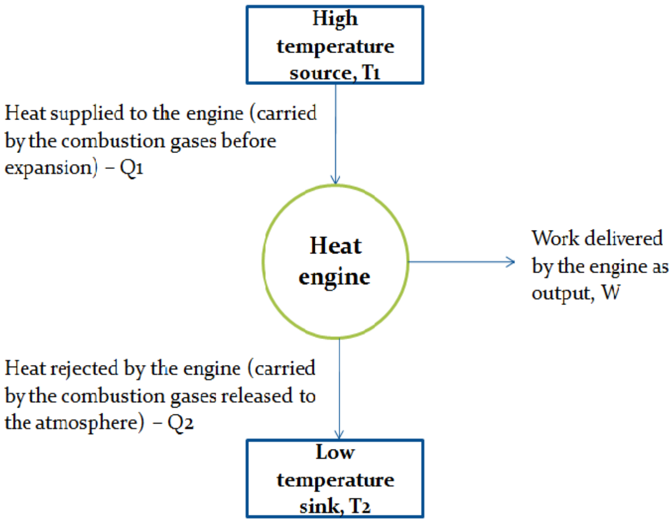



*Fig. 1.1 Heat Engine*

The fuel like gas, petrol or diesel is burned in the engine resulting in the formation of high temperature and high pressure gas, called as combustion gas or flue gas (possessing heat energy – Q1). These gases are expanded in the engine to get the mechanical work as output (W). After the expansion, still the gas possesses some heat in it (Q2), which is rejected to the surroundings.

Applying energy balance to the heat engine,

*Energy input to the Engine = Energy output*

$$Q_1 = W + Q_2$$

The conversion of heat energy in to the required work is assessed by a parameter called thermal efficiency. Thermal efficiency represents the % of input heat energy converted in to the required work as output. Mathematically

$$Thermal \quad efficiency, \eta = \frac{output \quad energy}{input \quad energy}$$

For the engine, $Thermal \quad efficiency, \eta = \frac{work \quad output}{heat \quad input} = \frac{W}{Q_1}$

*If the thermal efficiency is more, then it indicates that the engine is more efficient in converting heat in to required work.*

## 2. CLASSIFICATION OF I.C ENGINES

The heat engines are classified as given in this flow diagram.

If the combustion of the fuel takes place inside the engine, then it is called an internal combustion engine whereas if the combustion takes place outside the engine, then it is called external combustion engine.

Further the engines are sub-classified as reciprocating and rotating engines. In the reciprocating engines, the piston will have a to and fro motion inside the cylinder of an engine between the top and bottom dead centres (if it is a vertical engine) and between inner and outer dead centres (if it is a horizontal engines). Examples are the engines used in automobiles etc.,

*In this subject, we will discuss only reciprocating engines in detail.*

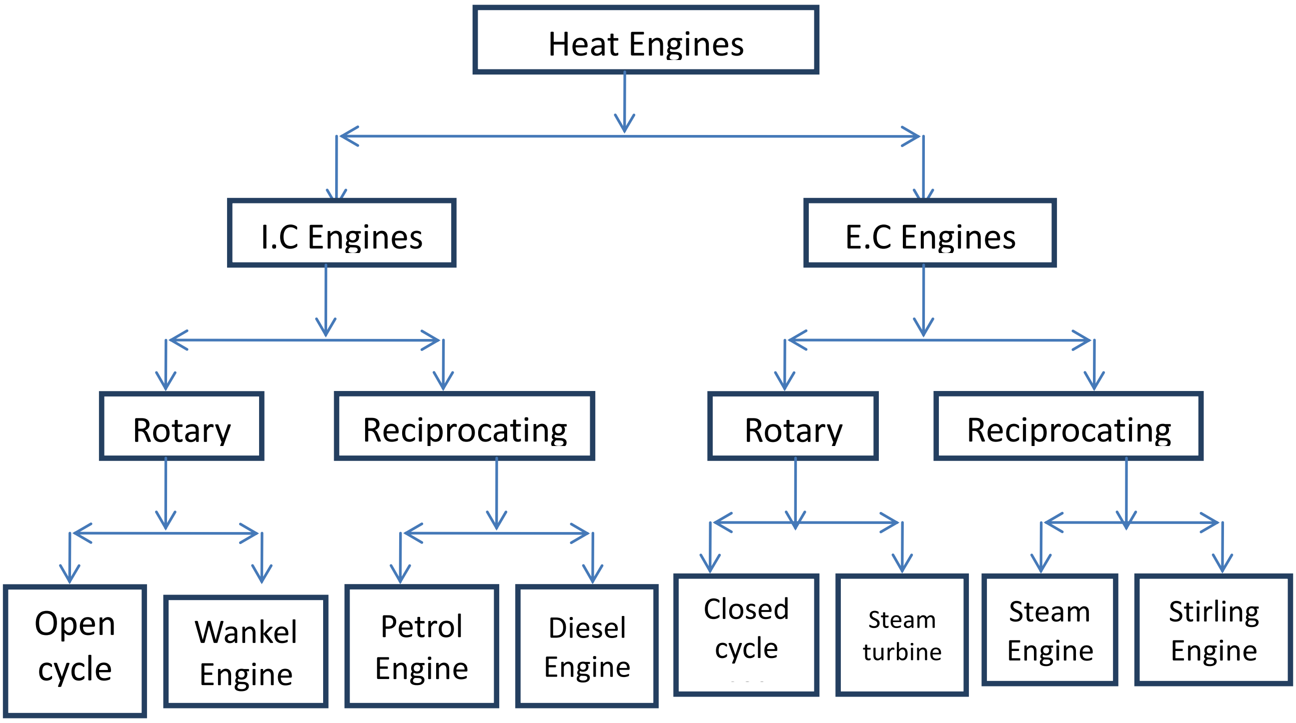



*Fig 1.2 Classifications of Heat Engines*

## 3. INTRODUCTION TO INTERNAL COMBUSTION ENGINES

The internal combustion engine is an engine in which the combustion of a fuel (generally, fossil fuel) occurs with an oxidizer (usually air) in a combustion chamber. In an internal combustion engine the expansion of the high temperature and high pressure gases, which are produced by the combustion, directly applies force to components of the engine, such as the pistons or turbine blades or a nozzle, and by moving it over a distance, generates useful mechanical energy.

*Examples are engines used in bikes, cars, buses, cranes etc.*

In an engine, the following processes are observed.

1. **Suction process**: Air is taken from the atmosphere after filtering it in an air filter. In the engine, pressure is maintained less than the atmosphere, which creates the necessary pressure difference for the flow of charge into the cylinder.

   **Charge –** In petrol engine – Air + Petrol mixture; In diesel engine - Air
2. **Compression process:** In this process, the pressure of the charge is increased with reduction in the volume. This also increases the temperature, resulting in increasing of internal energy and enthalpy. This process requires some energy as input in the form of work. So work done is negative in this process.
3. **Combustion:** The fuel (petrol or diesel or coal or gas) is burned in the closed chamber with the help of oxidant resulting in the formation of high pressure and high temperature gas. In this process, the chemical energy is converted in to the heat energy. The temperature of the substance increases along with increase in pressure.
4. **Expansion process:** In this process, the pressure of the gas is decreased with increase in the volume. During the expansion, work is performed by the hot gas thereby delivering the mechanical work as output with the expense of enthalpy. So the work done is positive in this process.
5. **Heat rejection:** The high temperature gas after expansion is released to the atmosphere resulting in the decrease of temperature.

## 4. Nomenclature of the Engine

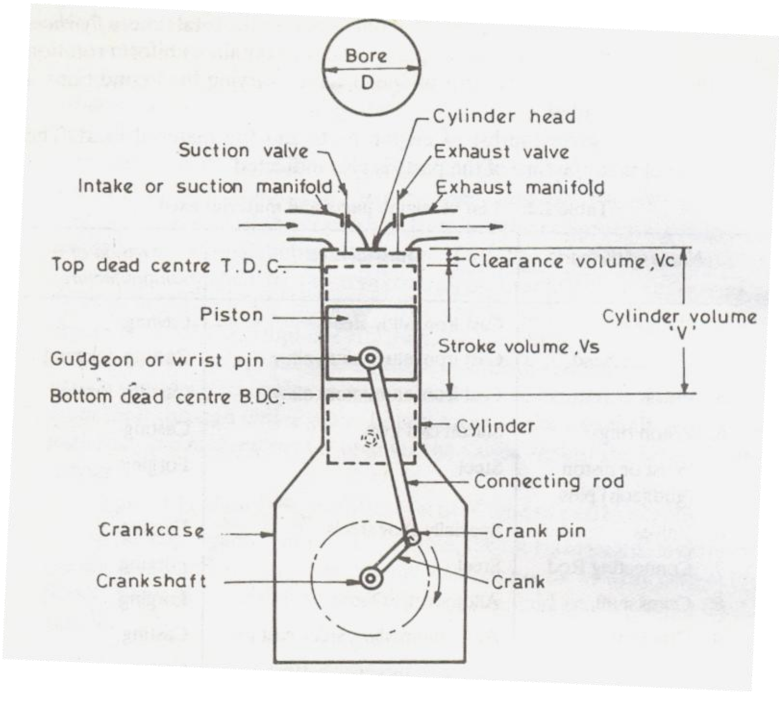



*Fig. 1.3 Components of an Engine*

Engine consists of many components, as depicted in the fig. 1.3.

***Block:*** Body of the engine containing cylinders, made of cast iron or aluminium.

***Cylinder***: The circular cylinders in the engine block in which the pistons reciprocate back and forth.

***Head:*** The piece which closes the end of the cylinders, usually containing part of the clearance volume of the combustion chamber.

***Combustion chamber:*** The end of the cylinder between the head and the piston face where combustion occurs. The size of combustion chamber continuously changes from minimum volume when the piston is at TDC to a maximum volume when the piston at BDC.

***Crankshaft:*** Rotating shaft through which engine work output is supplied to external systems. The crankshaft is connected to the engine block with the main bearings. It is rotated by the reciprocating pistons through the connecting rods connected to the crankshaft, offset from the axis of rotation. This offset is sometimes called crank throw or crank radius.

***Connecting rod:*** Rod connecting the piston with the rotating crankshaft, usually made of steel or alloy forging in most engines but may be aluminum in some small engines.

***Camshaft:*** Rotating shaft used to push open valves at the proper time in the engine cycle, either directly or through mechanical or hydraulic linkage (push rods, rocker arms, tappets).

***Crankcase:*** Part of the engine block surrounding the crankshaft. In many engines the oil pan makes up part of the crankcase housing.

***Intake manifold:*** Piping system which delivers incoming air to the cylinders usually made of cast metal, plastic, or composite material. In most SI engines, fuel is added to the air in the intake manifold system either by fuel injectors or with a carburetor.

The individual pipe to a single cylinder is called runner.

***Carburetor:*** A device which meters the proper amount of fuel into the air flow by means of pressure differential. For many decades it was the basic fuel metering system on all automobile (and other) engines.

***Fuel injector:*** A pressurized nozzle that sprays fuel into the incoming air (SI engines ) or into the cylinder (CI engines).

***Spark plug:*** Electrical device used to initiate combustion in an SI engine by creating high voltage discharge across an electrode gap.

***Exhaust manifold:*** Piping system which carries exhaust gases away from the engine cylinders, usually made of cast iron.

***Exhaust System:*** Flow system for removing exhaust gases from the cylinders, treating them, and exhausting them to the surroundings.

It consists of an exhaust manifold which carries the exhaust gases away from the engine, a thermal or catalytic converter to reduce emissions, a muffler to reduce engine noise, and a tailpipe to carry the exhaust gases away from the passenger compartment.

***Flywheel:*** Rotating mass with a large moment of inertia connected to the crank shaft of the engine. The purpose of the flywheel is to store energy and furnish large angular momentum that keeps the engine rotating between power strokes and smooths out engine operation.

***Top Dead Center (TDC):*** Position of the piston when it stops at the farthest point away from the crankshaft. When the piston is at TDC, the volume in the cylinder is a minimum called the clearance volume.

***Bottom Dead Center (BDC):*** Position of the piston when it stops at the point closest to the crankshaft.

***Stroke:*** Distance traveled by the piston from one extreme position to the other : TDC to BDC or BDC to TDC.

***Bore:*** It is defined as cylinder diameter or piston face diameter; piston face diameter is same as cylinder diameter (minus small clearance).

***Swept volume/Displacement volume:*** It is the volume displaced by the piston as it travels through one stroke. Swept volume is defined as stroke times bore.

***Clearance volume:*** It is the minimum volume of the cylinder available for the charge (air or air fuel mixture) when the piston reaches at its outermost point (top dead center or outer dead center) during compression stroke of the cycle. It is also the minimum volume of combustion chamber with piston at TDC.

## 5. Classification of I.C. Engines

An internal combustion can be classified in to different types based on different parameters mentioned below.

1. Types of ignition

*(i) Spark Ignition (SI)*

An SI engine starts the combustion process in each cycle by use of a spark plug. At the end of the compression process, the spark plug gives a high voltage electrical discharge between two electrodes, which ignites the air fuel mixture in the combustion chamber.

*(ii)* *Compression Ignition (CI)*

At the end of the compression process, the diesel fuel is sprayed in to the combustion chamber which contains compressed air at high pressure and high temperature, in the form of fine droplets. This results in the complete burning of the diesel fuel in the presence of air completely.

2. Engine cycle

(i) *Four-stroke cycle*: All the processes mentioned in section-2 i.e. suction, compression, combustion, expansion and exhaust will be completed in four piston movements over two revolutions of the crank shaft for each cycle.

(ii) *Two-stroke cycle*: All the processes mentioned in section-2 i.e. suction, compression, combustion, expansion and exhaust will be completed in two piston movements over one revolution of the crank shaft for each cycle.

3. Basic Design

(i) *Reciprocating*: Engine has one or more cylinders in which pistons reciprocate back and forth. The combustion chamber is located in the closed end of each cylinder. Power is delivered to a rotating output crankshaft by mechanical linkage with the pistons.

(ii) *Rotary:* Engine is made of a block (stator) built around a large non-concentric rotor and crankshaft. The combustion chambers are built into the non-rotating block.

4. Position and number of cylinders of reciprocating engines

(i) *Single Cylinder:* Engine has one cylinder and piston connected to the crankshaft via connecting rod.

(ii) *In-Line:* Cylinders are positioned in a straight line, one behind the other along the length of the crankshaft. They can consist of 2 to 11 cylinders or possibly more. In-line four-cylinder *engines are very common for automobile and other applications*. In-line six and eight cylinders are historically common automobile engines. In-line engines are sometimes called Straight (e.g., straight six or straight eight).

*(iii)* *V Engine:* Two banks of cylinders at an angle with each other along a single crankshaft, allowing for a shorter engine block. The angle between the banks of cylinders can be anywhere from 15° to 120° with 60°-90°. V engines usually have even numbers of cylinders from 2 to 20 or more. V6s and V8s are common

automobile engines, with V12s and V16s (historic) found in some luxury and high performance vehicles. *Large ship and stationery engines have anywhere from 8 to 20 cylinders.*

(iv) *Opposed Cylinder Engine:* Two banks of cylinders opposite to each other on a single crankshaft (a V engine with 180 deg V). *These are common on small aircraft and some automobiles* with an even number of cylinders from two to eight or more. These engines are often called flat engines (e.g., flat four).

*(v)* *W engine:* Engines of two different cylinder arrangements have been classified as W engines in the technical literature. One type is the same as a V engine except with three banks of cylinders on the same crankshaft. *They are not common, but some race cars of 1930 s and some luxury cars of the 1990s had such engines either with 12 cylinders or 18 cylinders. Another type of W engine is the modern 16 cylinder engine made for the Bugatti automobile (W16). This engine is essentially two V8 engines connected together on a single crankshaft.*

(vi) *Opposed piston engine:* Two pistons in each cylinder with the combustion chamber in the center between the pistons. A single combustion process causes two power strokes at the same time, with each piston being pushed away from the center and delivering power to a separate crankshaft at each end of the cylinder. Engine output is either on two rotating crankshafts or on one crankshaft incorporating a complex mechanical linkage. *These engines are generally of large displacement, used for power plants, ships, or submarines.*

(vii) *Radial engine:* Engines with pistons positioned in a circular plane around a circular crankshaft. The connecting rods of the piston are connected to a master rod, which in turn, is connected to the crankshaft. A bank of cylinders on a radial engine almost always has an odd number of cylinders ranging from 3 to 13 or more. Operating on a four-stroke cycle every other cylinder fires and has a power stroke as the crankshaft rotates, giving a smooth operation. Many medium and large size propeller driven aircraft use radial engines.

5. Air Intake Process

(i) *Naturally Aspirated*: No intake air pressure boosts system. Air is inhaled in to the engine cylinder at atmospheric condition.

(ii) *Super charged*: Before inhaling air into the engine cylinder, the air is compressed to pressure higher than atmospheric pressure, with the help of a compressor driven by the same engine.

(iii) *Turbo charged:* Before inhaling air into the engine cylinder, the air is compressed to pressure higher than atmospheric pressure, with the help of a compressor. The power input to the compressor is obtained by coupling the compressor shaft with the shaft of the small turbine, in which the exhaust gases coming from the engine are expanded.

(iv) *Crankcase compressed*: Two-stroke cycle engine which uses the crankcase as the intake air compressor. Limited development work has also been done on design and construction of four-stroke cycle engines with crank case compression.

6. Method of fuel input for spark ignition engines

(i) *Carbureted*: A device for mixing air and fuel to facilitate the combustion process.

(ii) *Multipoint port fuel injection*: One or more injectors are provided for the injection of fuel in to the cylinder at each cylinder intake.

(iii) *Throttle body fuel injection*: Injectors upstream in intake manifold.

(iv) Gasoline direct injection: Injectors mounted in combustion chambers with injection directly into cylinders.

7. Method of fuel input for compression ignition engines

(i) Direct injection: Fuel injected into main combustion chamber.

(ii) Indirect injection: Fuel injected into secondary combustion chamber.

(iii) Homogeneous charge compression ignition: Some fuel added during intake stroke.

8. Based on the type of fuel used

(i) Gasoline (petrol)

(ii) Diesel oil or Fuel oil

(iii) Gas, Natural gas, Methane

(iv) Alcohol-Ethyl, Methyl

(v) Dual fuel: There are a number of engines that use a combination of two or more fuels. Some, Usually large, CI engines use a combination of natural gas and diesel fuel. These are attractive in developing third world countries because of the high cost of the diesel fuel. Combined gasoline alcohol fuels are becoming more common as an alternative to straight gasoline automobile engine fuel.

(vi) Gasohol: Common fuel consisting of 90% gasoline and 10% alcohol.

9. Application

Automobile, Locomotive, Stationery, Marine, Aircraft, Small, Portable, chain saw, model airplane.

10. Type of cooling

*(i)* *Air cooled* – The outer surface of the engine cylinder is provided with plate type projections called fins. These are provided on low capacity engines; in general o single cylinder engines.

(ii) *Liquid cooled, Water-cooled* – All the multi-cylinder engines are cooled by the circulation of water around the cylinders, because of the large amount of heat generated during the combustion process. The hot water coming out of the engine is cooled in the radiator fitted to the engine, by the circulation of air flowing over it with the help of a cooling fan.

6. **4-STROKE SINGLE CYLINDER SPARK IGNITION ENGINE – WORKING (Otto Cycle)**

If Suction, compression, expansion and exhaust completed in 4-strokes, then it is called 4-stroke engine and if these 4 processes completes in 2 strokes it is called 2-stroke engine.

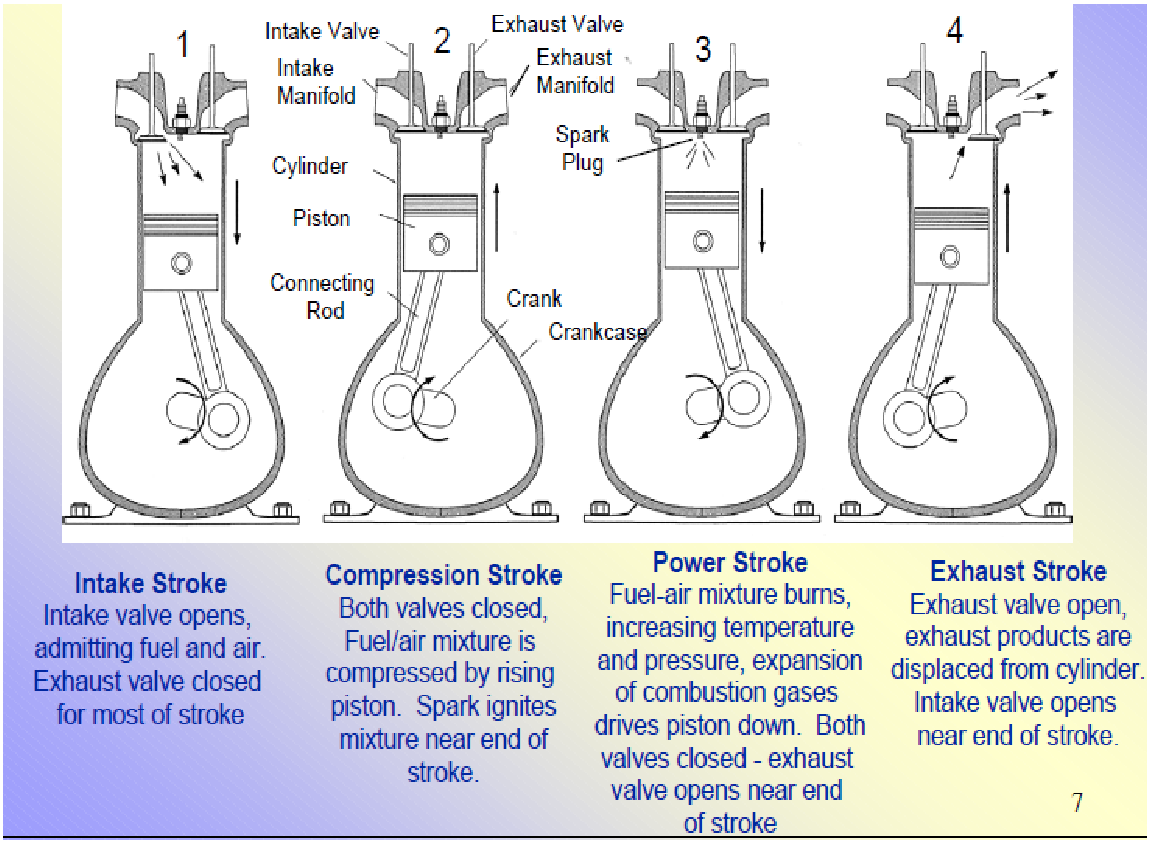



*Fig. 1.4 working of a 4-stroke petrol engine*

- **suction stroke or intake stroke** : During this stroke, Inlet valve opens Air +fule mixture enters into the cylinder due to the downward movement of the piston. The exhaust valve remains closed during this process. Piston moves from TDC to BDC.
- **compression stroke(1-2) (Reversible Adiabatic compression)**

As the crank shaft further rotates, the piston starts moving from BDC to TDC, both the inlet and exhaust valve closes and results in the **compression** of the air-fuel mixture.. During the compression, the pressure and temperature of the mixture rises to the higher value. Once the piston reaches TDC position, the spark is ignited by the spark plug, which is connected to a power source (battery). Since the air fuel mixture is homogeneously mixed, combustion starts instantaneously as soon as we supply spark. Here rate of pressure rise **(process 2-3 in P-v diagram)** is very rapid such that there is no movement of the piston further from TDC. Hence here the combustion is taking place at **Constant Volume**.

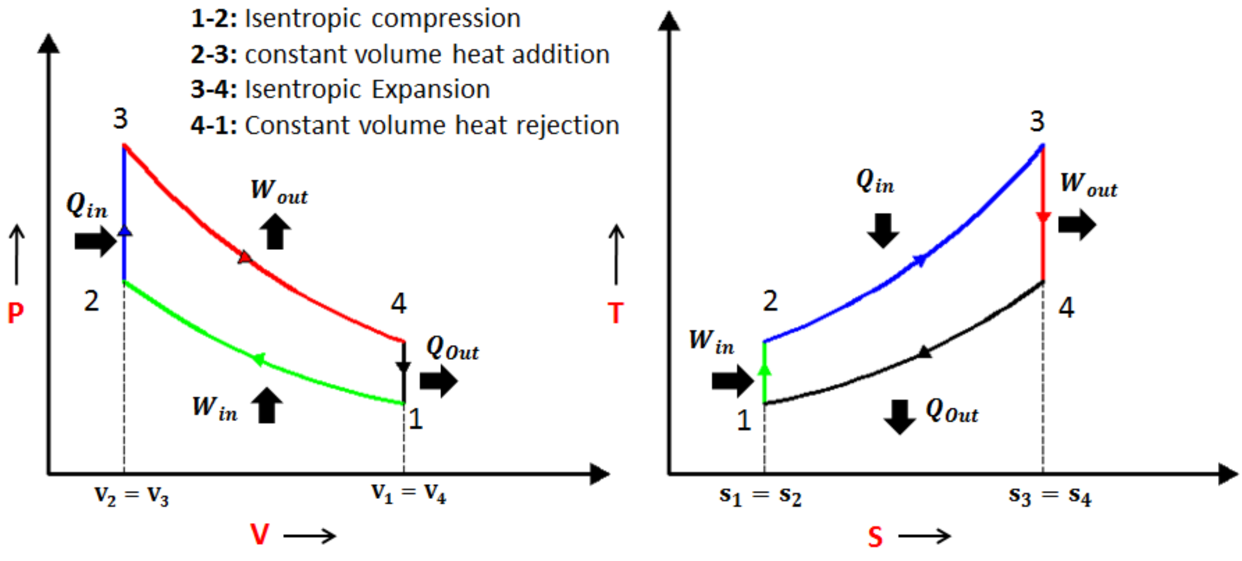
- **power stroke or expansion stroke( Reversible adiabatic Expansion 3-4)**

These high pressure gases exert forces on the top surface of the piston, pushing it in the downward direction. Now the piston moves from TDC to BDC and the forces are transmitted to the crank shaft via connecting rod, resulting in the turning of the crank shaft. This process is called **power stroke or expansion stroke.**

When the piston is at **BDC** Exhaust valve opens which releases high pressure suddenly to atmosphere, Which is represented by process 4-1 in the p-v diagram. This is accompanied with exhaust stroke further.

- **Exhaust stroke**. As the crank shaft rotates further, the piston starts moving from BDC to TDC. Now the exhaust valve is opened, because of the existing pressure difference, the exhaust gases are released to the atmosphere, until the piston reaches TDC. This process is called **exhaust stroke.**

*Applications: used in the 2-wheelers and 4 wheelers etc.,*




P-V and T-S Diagram of Otto Cycle

7. **4-Stroke Single Cylinder Compression Ignition Engine – Working (Diesel Cycle)**

<u>*Compression ignition engine is also called as diesel engine. In this engine, the fuel used is diesel.* The working of the engine is presented in the following figure and explained in detail below:</u>

**Suction Stroke(1-2)**: Only air is sucked into the cylinder during suction stroke when the piston is moving from TDC to BDC. Inlet valve opened and exhaust valve closes.

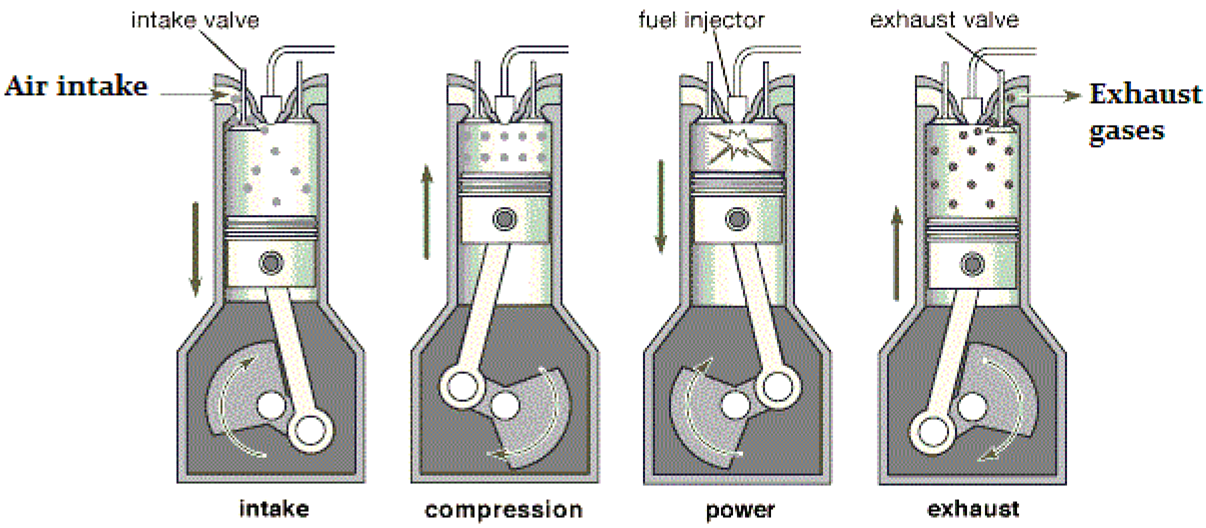
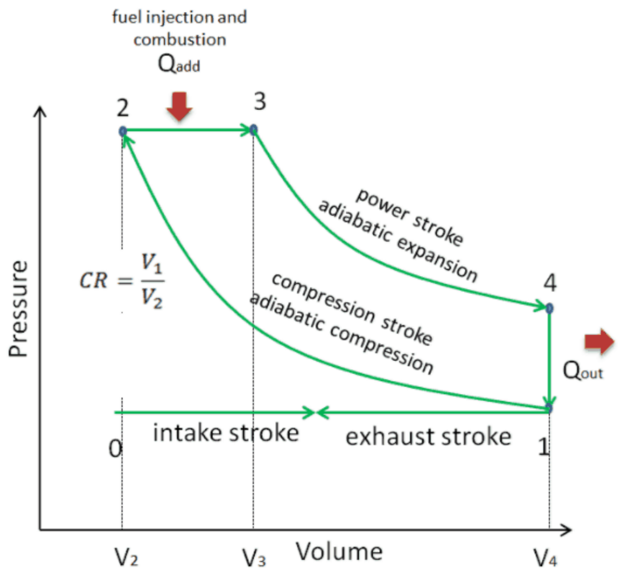
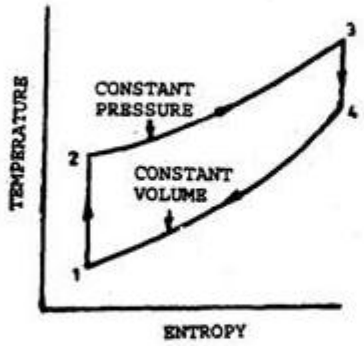



*Fig. 1.5 Working of a 4-stroke diesel engine*

**Compression stroke(1-2)**: As the crank shaft further rotates, the piston starts moving from BDC to TDC, both the inlet and exhaust valve closes and results in the **compression** of the air. During the compression, the pressure and temperature of the air rises to the higher value, the heat being present in it is sufficient to burn the diesel completely. Hence, there is no need of other heat source in the diesel engine.

- **Heat addition Process**: **(Constant pressure process 2-3):** Once the piston reaches TDC position, the fuel Jet( diesel oil) is injected in to the engine cylinder at high pressure in the form of an atomized spray. The turbulence of air in the combustion chamber passing across the jet tears the fuel particles from the core. A mixture of air and fuel forms at some locations and oxidation starts. The fuel droplets evaporate by absorbing the latent heat of vaporization from the surrounding air which reduces the temperature of thin layer of air surrounding the droplet and some time elapses before this temperature can be raised again by absorbing heat from the bulk of air. As soon as this vapor and air reach the level of auto ignition temperature combustion starts. Thus there is some delay period before ignition takes place. So here the combustion process is not Instantaneous like in Otto Cycle. Hence in this cycle there is a movement of the piston from TDC towards BDC during the combustion Process also. The rise in pressure is compensated by lowering of pressure due to piston movement and hence the combustion is at **Constant pressure**.
- **power stroke or expansion** (3-4) : These high pressure gases exert forces on the top surface of the piston, pushing it in the downward direction. Now the piston moves from TDC to BDC and the forces are transmitted to the crank shaft via connecting rod, resulting in the turning of the crank shaft. This process is called **power stroke or expansion.** During expansion, the pressure and temperature of the gases decreases, the pressure being greater than the atmospheric pressure.
- **Exhaust Stroke**:(4-1): The exhaust valve opens during this process. The pressure releases suddenly into the atmosphere (process 4-1). As the crank shaft rotates further, the piston starts moving from BDC to TDC. the exhaust gases are released to the atmosphere, until the piston reaches TDC. This process is called **exhaust stroke.**




*Applications: used in the heavy motor vehicles like buses, lorries, cars, etc.,*

*If all the above mentioned 4-strokes are completed in 2 complete revolutions of the crank shaft, then it is called a 4-stroke single cylinder diesel engine. Because of higher pressures, there generates higher vibrations in the engine. The size of the engine is bulky resulting in more weight.*

## 8. Differences between Diesel and petrol engines

| S. No | Description | S. I Engine | C. I. Engine |
|---|---|---|---|
| 1 | Basic cycle | Based on Otto cycle | Based on Diesel cycle |
| 2 | Fuel | Petrol or gasoline. It has high self ignition temperature. | Diesel oil. It has low self ignition temperature |
| 3 | Introduction Of fuel | Fuel is inducted in to the engine after mixing with the air in the device called carburetor. | Fuel is sprayed in to the cylinder after the compression of air. |
| 4 | Ignition | After the compression of air-petrol mixture, spark is generated for the burning of the fuel. Hence, it is spark ignited engine and requires an ignition system with spark plug in the combustion chamber. | At the end of the compression stroke, diesel oil is sprayed in to the cylinder containing hot air. This heat is sufficiently enough for the complete burning of the fuel. *Ignition system with spark plug in the combustion chamber is eliminated.* |
| 5 | Compression ratio | It varies from 6 to 10.5. | It varies from 14 to 20. |
| 6 | Speed | It has higher maximum revolution per minute due to lighter weight | It has lower maximum revolution per minute due to more weight. |
| 7 | Efficiency | Maximum efficiency is lower due to low compression ratio. | Maximum efficiency is higher due to high compression ratio. |
| 8 | Weight | It is light in weight due to lower | It is heavier due to higher |

| | | | |
|---|---|---|---|
| | | operating pressures. | operating pressures. |

## 8. Working of a 2-stroke petrol engine

*In the petrol engine, the charge intake is a mixture of air and petrol. At the end of the compression, the spark is released in to the hot air-petrol mixture. In the 2-stroke engine, there are no valves and valve mechanisms. Only ports are provided. In the 2-stroke engine, 3 ports namely intake port, transfer port and exhaust port are provided. The opening and closing of ports are controlled by the position of the moving piston in the engine.*

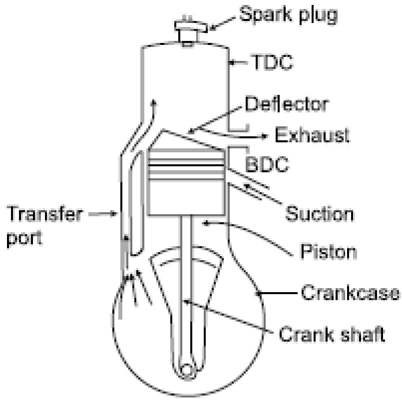



*Fig. 1.6 2-stroke engine*

Initially, air is taken from the atmosphere and after mixing with the petrol in the carburetor, it is supplied to the engine through the inlet port. When the piston moves from the BDC to the TDC, the surface of the piston uncovers the inlet port. This results in the flow of air-petrol mixture into the crank case, called **suction.** As the piston moves further in upward direction, it covers the transfer port as well as exhaust port. This results in the stopping of the transfer of compressed air-petrol mixture to the top of the piston from the crank case through the transfer port and loss of the mixture through the exhaust port. As the piston moves to TDC, the piston completely uncovers the inlet port and covers the transfer and exhaust ports. The air mixture present above the piston surface (intake of the earlier cycle) gets **compressed** and the spark is released. Now this mixture burns completely **(heat addition or combustion)** and the combustion gases at high pressure and temperature exerts force on the piston making it to move in the downward direction. This is **power stroke**.

When the piston moves in downward direction, it uncovers the exhaust port initially, followed by the closing of the inlet port simultaneously. Because of this, some of the exhaust gases leave the engine via exhaust port without expanding completely in the engine; resulting in the loss of power. By the closing of the inlet port, the intake completely stops and as the piston further moves downwards towards BDC; the air-petrol mixture in the crank case gets compressed. At the same time, the transfer port also opens; leaving the path for the compressed charge to enter the top of the piston. When the piston reaches BDC, piston completely covers the inlet port and uncovers the transfer and exhaust ports. At this condition, the expansion of the combustion gases completes and also leaves to the atmosphere from the engine. This process continues.

*But because of simultaneous opening of transfer and exhaust ports, there occurs a loss of charge; resulting in less amount of air-petrol mixture participating in the combustion. Even before the complete expansion of the gases, due to the exhaust port opening; the expansion process will be incomplete resulting in the further reduction in the work output*

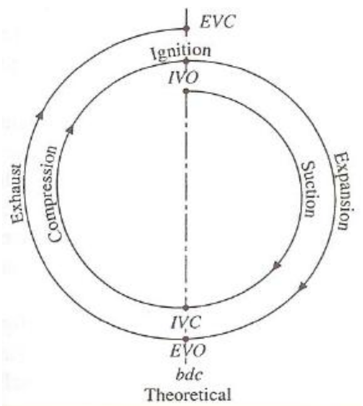
## 9. Differences between 2-stroke and 4-stroke engines

| S. No | 2-stroke engines | 4-stroke engines |
|---|---|---|
| 1 | All the 4-strokes of the engine (suction, compression, expansion and exhaust) are completed in 1 complete revolution of the crank shaft. | All the 4-strokes of the engine (suction, compression, expansion and exhaust) are completed in 2 complete revolution of the crank shaft. |
| 2 | Valve and valve operating mechanisms (cam mechanism) are completely absent. Only ports are provided in the engine. | Valve and valve operating mechanisms (cam mechanism) are present in the engine. |
| 3 | Because of less no. of parts, the weight of the engine is less. | Because of more no. of parts, the weight of the engine is more. |
| 4 | When compared with the 4-stroke engine (for 2 complete crank shaft revolutions), more amount of heat will be generated during combustion process. Hence more amount of coolant is required. | When compared with the 2-stroke engine (for 2 complete crank shaft revolutions), less amount of heat will be generated during combustion process. Hence less amount of coolant is required. |
| 5 | For the two revolutions of the crank shaft, theoretically, more amount of power will be generated (two times the power generated in the 4-stroke engine) | When compared with the 2-stroke engine, for two revolutions of the crank shaft, less amount of power will be generated. |
| 6 | Relatively more amount of lubricant is required. | Relatively less amount of lubricant is required. |
| 7 | Torque generated will be uniform. Hence, a smaller flywheel is required. | Torque generated is not uniform in single cylinder engines. Hence, a larger flywheel is required. |
| 8 | For the same capacity, the size of the engine will be small. | For the same capacity, the engine size is large. |

## 10. Valve timing diagram

These are drawn for the engine provided with the valves and more specifically, for the 4-stroke engines. This diagram is prepared to study the opening of the valves based on the rotation of the crank shaft. The following figure fig. 1.7 (a) represents the theoretical valve timing diagram drawn for the 4-stroke engine. Theoretically, when the piston reaches the TDC position, both the valves should be in closed condition. When the piston just starts moving from TDC to BDC, the inlet valve should open resulting in the intake of the charge. As the piston reaches the BDC, the suction process completes and the inlet valve closes completely. Further as the crank shaft rotates, the piston moves from BDC to TDC, resulting in the compressing of the charge. When the piston reaches the TDC, the charge will be at high pressure and temperature; ***the spark is released for the burning of the charge (air+petrol mixture), if it is a petrol engine; diesel is sprayed in to the charge (hot air), if it is a diesel engine***.

Combustion completes and the exhaust gases starts expanding in the engine, resulting of the moving of the piston in the downward direction from TDC to BDC. As the piston moves to BDC, the expansion completes. Now the piston moves from BDC to TDC with the further rotation of the crank shaft. In this case, when the piston moves just from BDC, the exhaust valve just opens resulting in the exhaust Process. As the piston reaches TDC, the exhaust process completes and the exhaust valve closes. All these processes, completes in the two complete revolutions of the crank shaft. Theoretically, the friction in the valve opening and closing, effect of the pressures on the valve opening and closing, delay in the opening and closing of valves because of the cam mechanisms is neglected. Then the valve timing diagram will be as presented in fig. 1.7 (a). But in practical, there are two factors, one mechanical and other dynamic, for the actual valve timing to be different from the theoretical valve timing.



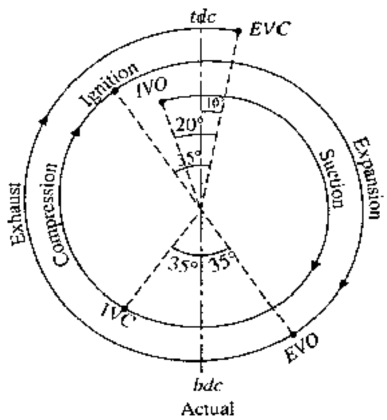




*Fig. 1.7 (a) Theoretical valve timing diagram Fig. 1.7 (b) Actual valve timing diagram*

*(a)* <u>*Mechanical Factor*</u>: The poppet valves of the reciprocating engines are opened and closed by the cam mechanisms. The clearance between cam, tappet and valve must be slowly taken up and valve slowly lifted, if the wear and tear is to be avoided. Similarly, the valve should be closed slowly; otherwise the valve will bounce from the valve seat. The valve opening and closing periods are spread over a

considerable number of crank shaft degrees. As a result, the opening of the valve should commence ahead of the time at which it is fully opened (i.e. before dead centers). Similarly, the valves must be closed after the dead centers.

*(b)* <u>*Dynamic factor*</u>: The dynamic effects (forces) of the gas flowing through the cylinder are also to be taken into consideration, while the valve timing is set up.

| **Valve position** | **Theoretical position of Piston** | **Diesel Engine (Actual angle to be turned by the crank shaft)** | **Diesel Engine (Actual angle to be turned by the crank shaft)** |
|---|---|---|---|
| Inlet valve opening | TDC | $30^0$ before the piston reaches TDC | $10^0$ before the piston reaches TDC |
| Inlet valve closing | BDC | $50^0$ after the piston reaches BDC | $45^0$ after the piston reaches BDC |
| Exhaust valve opening | BDC | $45^0$ before the piston reaches BDC | $45^0$ before the piston reaches BDC |
| Exhaust valve closing | TDC | $30^0$ after the piston reaches TDC | $10^0$ after the piston reaches TDC |
| Valve overlap | At TDC | 45-$60^0$ for the inlet valve | 15-$30^0$ for the inlet valve |
| Fuel injection or Spark generation | TDC | $15^0$ before the piston reaches TDC | $15^0$ before the piston reaches TDC |

***Note: Diagram will be similar to 4-stroke diesel engine (replace spark plug with fuel injector)***

Here, before the complete closing of the exhaust valve, the inlet valve opens for the suction process. The crank angle corresponding to this case is called <u>*valve overlap. At this stage, there occurs a loss of charge.*</u>

**Power and Mechanical Efficiency**

The main purpose of running an engine is to obtain mechanical power.

- Power is defined as the rate of doing work and is equal to the product of force and linear velocity or the product of torque and angular velocity.
- Thus, the measurement of power involves the measurement of force (or torque) as well as speed. The force or torque is measured with the help of a dynamometer and the speed by a tachometer.

a. ***Brake power*** (*bp*)**:**

The power developed by an engine and measured at the output shaft is called the brake power (*bp*) and is given by,

$$bp = \frac{2\pi\,NT}{60}$$

Where, $T$ is torque in N-m and $N$ is the rotational speed in revolutions per minute.

b. ***Indicated power*** (*ip*): It is the total power developed by the combustion of fuel in the combustion chamber.

It forms the basis of evaluation of combustion efficiency or the heat release in the cylinder.

c. ***Friction power*** *(fp):* Energy lost in overcoming the friction during the relative motion between piston and cylinder, crank shaft and bearings, pumping losses etc. can be termed as friction power.

The difference between *ip* and *bp* is called friction power (*fp*).

$$fp = ip - bp$$

*Indicated power and Brake power are expressed in kilo-Watts or horse power.*

**Mean Effective Pressure and Torque:**

Mean effective pressure is defined as a hypothetical/average pressure which is assumed to be acting on the piston throughout the power stroke.

Power developed, $p = \frac{(mep)LAK}{60}\left(\frac{N}{n}\right)$

Where, mep = Mean effective pressure, N/m$^2$,

$p$ = power developed in Watts, $L$ = Length of the stroke, m,

$A$ = Area of the piston, m$^2$, K – number of cylinders

$N$ = Rotational speed of the engine, rpm

n = number of revolutions to be completed for 1 power stroke

*In a 4-stroke engine, 1 power stroke will be completed in 2 revolutions of crank shaft i.e. n=2.*

*In a 2-stroke engine, 1 power stroke will be completed in 1 revolutions of crank shaft i.e. n=1.*

***It is also calculated from Indicator diagram. i.e***

$$\frac{\text{Area of the Idicator diagram}}{\text{length of the Indicator diagram}} = \frac{Net\ work\ of\ the\ Cycle}{Swept\ volume}$$

a. If the mean effective pressure is based on *bp* it is called the brake mean effective pressure (*bmep*).

$$bmep = \left(\frac{n}{N}\right)\frac{60(bp)}{LAK}$$

b. If the mean effective pressure is based on *ip* it is called indicated mean effective pressure (*imep*).

$$imep = \left(\frac{n}{N}\right)\frac{60(ip)}{LAK}$$

c. Similarly, the friction mean effective pressure (*fmep*) can be defined as,

$$fmep = imep - bmep$$

**Specific Output**

Specific output of an engine is defined as the brake power (output) per unit of piston displacement and is given by,

$$Specific\ output = \frac{bp}{swept\ volume} = \frac{(bmep)LAK}{60}\left(\frac{N}{n}\right)\left(\frac{1}{AL}\right) = \frac{(bmep)K}{60}\left(\frac{N}{n}\right)$$

$$\Rightarrow Specific\ output = \left(\frac{K}{60\times n}\right)\times(bmep)\times N = cons\tan t\times(bmep)\times N$$

**Volumetric Efficiency**

- ✓ Volumetric efficiency of an engine is an indication of the measure of the degree to which the engine fills its swept volume.
- ✓ It is defined as the ratio of the **mass** of air inducted into the engine cylinder during the suction stroke to the mass of the air corresponding to the swept volume of the engine at atmospheric pressure and temperature.
- ✓ It can also be defined as the ratio of the actual **volume** inhaled during suction stroke measured at intake conditions to the swept volume of the piston.

Volumetric efficiency, $\eta_v$

$$= \frac{\text{Mass of charge actually sucked in}}{\text{Mass of charge corresponding to the cylinder intake } P \text{ and } T \text{ conditions}}$$

The amount of air taken inside the cylinder is dependent on the volumetric efficiency of an engine and hence puts a limit on the amount of fuel which can be efficiently burned and the power output. For supercharged engine the volumetric efficiency has no meaning as it comes out to be more than unity.

**Fuel-Air Ratio (*F*/*A*)**

Fuel-air ratio (*F*/*A*) is the ratio of the mass of fuel to the mass of air in the fuel-air mixture. Air-fuel ratio (*A*/*F*) is reciprocal of fuel-air ratio. Fuel-air ratio has a significant effect on

- ✓ the flame propagation velocity,
- ✓ the heat release in the combustion chamber,
- ✓ the maximum temperature and
- ✓ the completeness of combustion.

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

$$\text{Relative fuel-air ratio, } F_R = \frac{\text{Actual fuel} - \text{Air ratio}}{\text{Stoichiometric fuel} - \text{Air ratio}}$$

**Fuel consumption:**

In engine testing the fuel consumption is measured in terms of the fuel mass flow rate. It is the quantity of fuel supplied to the engine in unit time.

***Units***: kg/min (or) kg/hr (or) cc/sec etc.,

a. **Specific Fuel Consumption:**

   The specific fuel consumption (sfc), is a measure of how efficiently the fuel supplied to the engine is used to produce power.

   ***Unit***: kg/kW-hr

b. **Brake specific fuel consumption (bsfc):**

   Specific fuel consumption is defined as the amount of fuel consumed for each unit of brake power developed per hour. It is a clear indication of the efficiency with which the engine develops power from fuel. This parameter is widely used to compare the performance of different engines.

$$bsfc = \frac{m_f}{bp}$$

c. **Indicated specific fuel consumption (isfc):**

   It is the ratio of mass of fuel supplied to the engine to deliver unit indicated power per hour.

$$isfc = \frac{m_f}{ip}$$

**Thermal Efficiency:**

Thermal efficiency of an engine is defined as the ratio of the output to that of the chemical energy input in the form of fuel supply. It may be based on brake or indicated output. It is the true indication of the efficiency with which the chemical energy of fuel (input) is converted into mechanical work. Thermal efficiency also accounts for combustion efficiency, i.e., for the fact that whole of the chemical energy of the fuel is not converted into heat energy during combustion.

$$\eta_{th} = \frac{power\ output}{heat\ \text{supplied}}$$

Heat supplied, Q = mass of the fuel x calorific value

- **Brake thermal efficiency**: It is defined as the ratio of brake power developed by the engine to the heat generated in combustion.

$$\text{Brake thermal efficiency} = \frac{bp}{m_f \times C_v}$$

Where, bp – brake power in kW, $C_v$ = Calorific value of fuel, kJ/kg, and $m_f$ = Mass of fuel supplied, kg/sec.

- **Indicated thermal efficiency:** It is defined as the ratio of indicated power developed by the engine to the heat generated in combustion.

$$\text{Indicated thermal } \eta = \frac{I.P.}{\text{Heat supplied}}$$

$$= \frac{\text{I.P}}{mf \ \text{x} \ Cv}$$

- **Mechanical Efficiency:**

It is defined as the ratio of brake power to indicated power and can be expressed in many forms.

$$\eta_{mech} = \frac{bp}{ip} = \frac{bp}{bp + fp} = \frac{bmep}{imep} = \frac{\eta_{bth}}{\eta_{ith}}$$

**3. Measurement of brake power:**

*Principle of working:*

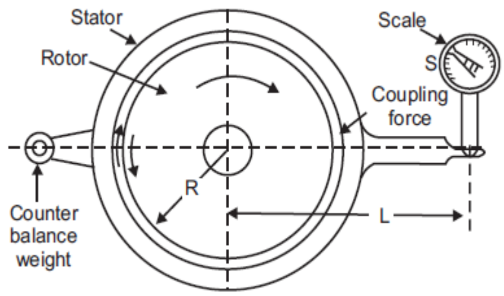


Fig: 1 Principle of a dynamometer

The brake power measurement involves the determination of the torque and the angular speed of the engine output shaft. The torque measuring device is called a dynamometer. Figure shows the basic principle of a dynamometer. A rotor driven by the engine under test is electrically, hydraulically or magnetically coupled to a stator. For every revolution of the shaft, the rotor periphery moves through a distance 2Πr against the coupling force F. The external moment or torque is equal to $S \times L$ where, $S$ is the scale reading and $L$ is the arm. This moment balances the turning moment $R \times F$, i.e. $S \times L = R \times F$

Torque exerted by the engine, $T = SL$

Power delivered by the engine turning at speed N and absorbed by the dynamometer is

$W = \omega.T = 2\pi NSL$

Hence, Brake power, $bp = 2\pi NT$

Dynamometers can be broadly classified into two main types, power absorption dynamometers and transmission dynamometer.

- **Absorption Dynamometers**

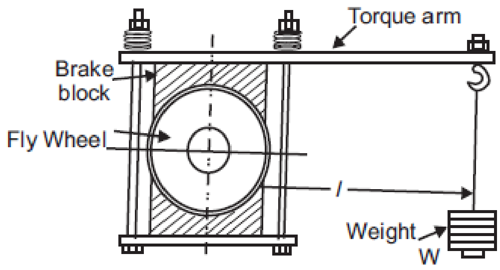
These dynamometers measure and absorb the power output of the engine to which they are coupled. The power absorbed is usually dissipated as heat by some means.

*Example of such dynamometers is prony brake, rope brake, hydraulic dynamometer, etc.*

- **Transmission Dynamometers**

  In transmission dynamometers, the power is transmitted to the load coupled to the engine after it is indicated on some type of scale. These are also called torque-meters.

  a. **Prony Brake**


  One of the simplest methods of measuring brake power (output) is to attempt to stop the engine by means of a brake on the flywheel and measure the weight which an arm attached to the brake will support, as it tries to rotate with the flywheel. The prony brake shown in Fig. 2 works on the principle of converting power into heat by dry friction. It consists of wooden block mounted on a flexible rope or band the wooden block when pressed into contact with the rotating drum takes the engine torque and the power is dissipated in frictional resistance. Spring-loaded bolts Fig: 2 prony brake dynamometer

  are provided to tighten the wooden block and hence increase the friction. The whole of the power absorbed is converted into heat and hence this type of dynamometer must be cooled. The brake horsepower is given by *BP* = 2Π *NT*

  Where, *T* = *W* × *l*, *W* being the weight applied at a radius *l*.

  b. **Rope Brake**

  The rope brake as shown in Fig. 3 is another simple device for measuring *bp* of an engine. It consists of a number of turns of rope wound around the rotating drum attached to the output shaft. One side of the rope is connected to a spring balance and the other to a loading device.

  Initially the engine is started under no load condition.

  *Let W* is the applied weight in Newton,

  *S* is the spring scale reading in Newton.

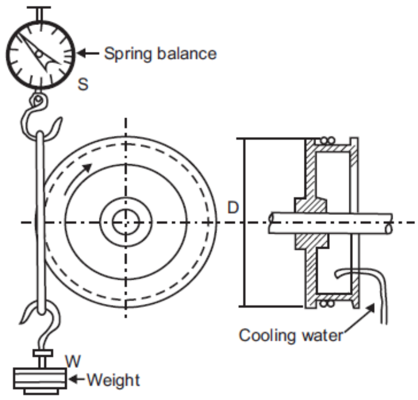
  


  Fig: 3 rope brake dynamometer

  Net force acting on the drum, $F = W - S$

  *With the application of load, the tension in the rope increases resulting in the arresting of the free rotation of the drum. The power is absorbed in friction between the rope and the drum. The drum therefore requires cooling.*

  Torque developed, $T = F \times R = (W - S) \times (R_b + r)$

  The brake power is given by

$$bp = T \times \omega = \frac{2\pi N(W-S)(R_b + r)}{60} = \frac{\pi N(W-S)(D_b + d)}{60}$$

  Where, $D_b$ is the brake drum diameter ($R_b$ – radius of the brake drum),

  *d* is the rope diameter (r – radius of the rope),

  N is the speed in rpm,

*Rope brake dynamometer is cheap and easily constructed but not a very accurate method because of changes in the friction coefficient of the rope with temperature.*

**c. Hydraulic Dynamometer**

Hydraulic dynamometer (shown in Fig. 4) works on the principle of dissipating the power in fluid friction rather than in dry friction.

- In principle its construction is similar to that of a fluid flywheel.
- It consists of an inner rotating member or impeller coupled to the output shaft of the engine.
- This impeller rotates in a casing filled with fluid.
- This outer casing, due to the centrifugal force developed, tends to revolve with the impeller, but is resisted by a torque arm supporting the balance weight.
- The frictional forces between the impeller and the fluid are measured by the spring-balance fitted on the casing.

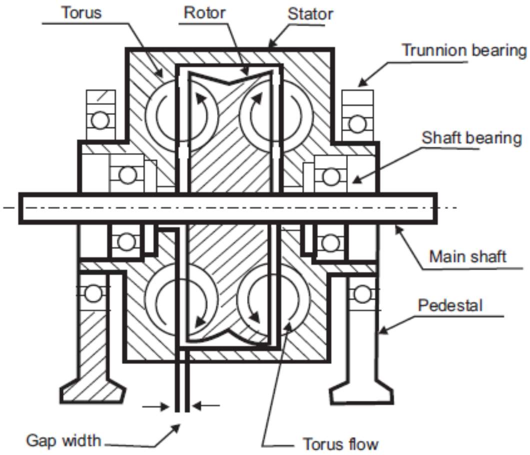



Fig: 4 Hydraulic dynamometer

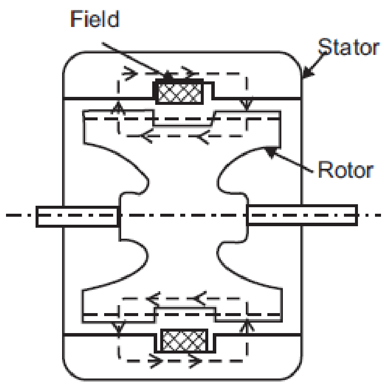
- The heat developed due to dissipation of power is carried away by a continuous supply of the working fluid, usually water.
- The output can be controlled by regulating the sluice gates which can be moved in and out to partially or wholly obstruct the flow of water between impeller, and the casing.

**d. Eddy Current Dynamometer**

The working principle of eddy current dynamometer is shown in Fig. 5. It consists of a stator on which are fitted a number of electromagnets and a rotor disc made of copper or steel and coupled to the output shaft of the engine. When the rotor rotates eddy currents are produced in the stator due to magnetic flux set up by the passage of field current in the electromagnets. These eddy currents are dissipated in producing heat so that this type of dynamometer also requires some cooling arrangement. The torque is measured exactly as in other types of Absorption dynamometers, i.e. with the help of a moment arm. The load is controlled by regulating the current in the electromagnets.

The following are the main advantages of eddy current dynamometers:

- ✓ High brake power per unit weight of dynamometer.
- ✓ They offer the highest ratio of constant power speed range (up to 5: 1).



- ✓ Level of field excitation is below 1% of total power being handled by dynamometer, thus, easy to control and programme.
- ✓ Development of eddy current is smooth Fig: 5 Eddy current dynamometer hence the torque is also smooth and continuous under all conditions.
- ✓ Relatively higher torque under low speed conditions.
- ✓ It has no intricate rotating parts except shaft bearing.
- ✓ No natural limit to size-either small or large.

**e. Transmission Dynamometers:**

Transmission dynamometers, also called torque meters, mostly consist of a set of strain-gauges fixed on the rotating shaft and the torque is measured by the angular deformation of the shaft which is indicated as strain of the strain gauge. Usually, a four arm bridge is used to reduce the effect of temperature to minimum and the gauges are arranged in pairs such that the effect of axial or transverse load on the strain gauges is avoided.

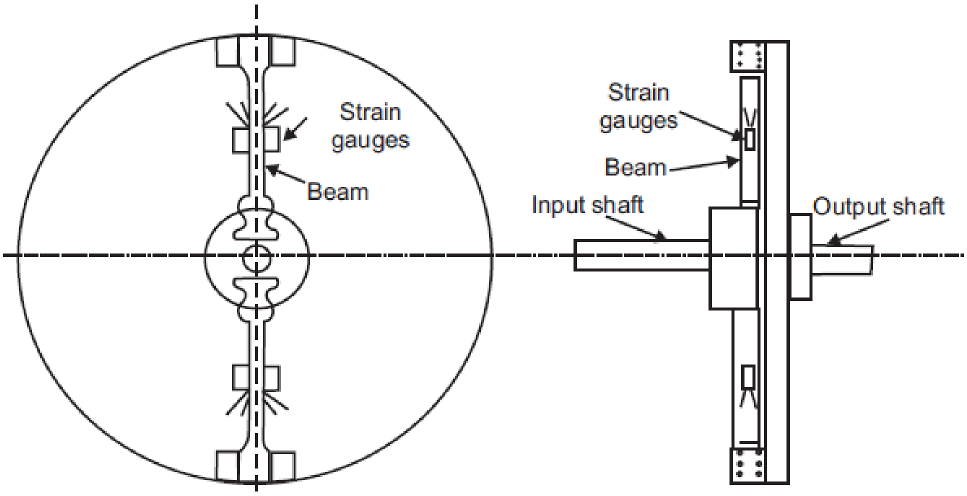



Fig: 6 Transmission dynamometer

Fig. 6 shows a transmission dynamometer which employs beams and strain-gauges for a sensing torque. Transmission dynamometers are very accurate and are used where continuous transmission of load is necessary. These are used mainly in automatic units.

**MEASUREMENT OF FRICTION POWER:**

The difference between indicated power and the brake power output of an engine is the friction power.

- Almost invariably, the difference between a good engine and a bad engine is due to difference between their frictional losses.
- The frictional losses are ultimately dissipated to the cooling system (and exhaust) as they appear in the form of frictional heat and this influences the cooling capacity required. Moreover, lower friction means availability of more brake power; hence brake specific fuel consumption is lower.
- The *bsfc* rises with an increase in speed and at some speed it renders the sue of engine prohibitive. Thus, the level of friction decides the maximum output of the engine which can be obtained economically.

In the design and testing of an engine; measurement of friction power is important for getting an insight into the methods by which the output of an engine can be increased. In the evaluation of *ip* and mechanical efficiency measured friction power is also used. The friction force power of an engine is determined by the following methods:

a. Willan’s line method.
b. Morse test.
c. Motoring test.
d. Retardation test
e. Difference between *ip* and *bp*.

**a. Willan's Line Method or Fuel Rate Extrapolation**

In this method, gross fuel consumption vs. *bp* at a constant speed is plotted and the graph (curve) is extrapolated back to zero fuel consumption as illustrated in Fig. 7. The point where this curve cuts the *bp* axis, is an indication of the friction power of the engine at that speed. This negative work represents the combined loss due to mechanical friction, pumping and blow by. The test is applicable only to compression ignition engines.

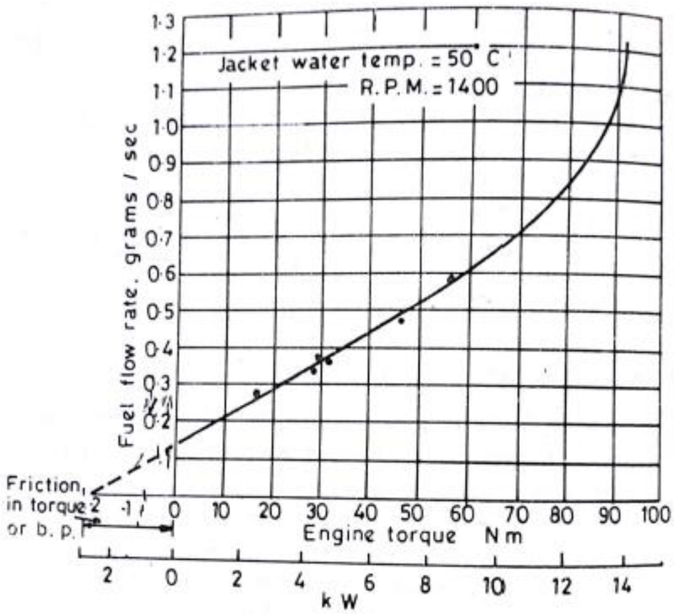



Fig: 7 Willan’s Line Method

- The main drawback of this method is the long distance to be extrapolated from data measured between 5 and 40% load towards the zero line of fuel input.
- The directional margin of error is rather wide because of the graph which may not be a straight line many times.
- The changing slope along the curve indicates part efficiencies of increments of fuel. The pronounced change in the slope of this line near full load reflects the limiting influence of the air-fuel ratio and of the quality of combustion.
- Similarly, there is a slight curvature at light loads. This is perhaps due to difficulty in injecting accurately and consistently very small quantities of fuel per cycle.
- Therefore, it is essential that great care should be taken at light loads to establish the true nature of the curve.

- The Willam's line for a swirl-chamber CI engine is straighter than that for a direct injection type engine.
- The accuracy obtained in this method is good and compares favorably with other methods if extrapolation is carefully done.

**b. Morse Test**

The Morse test is applicable only to multi cylinder engines.

- In this test, the engine is first run at the required speed and the brake power is measured ($bp_n$).
- Then, one cylinder is cut out by short circuiting the spark plug if it is a petrol engine or by disconnecting the injector if it is a diesel engine.
- Under this condition, all other cylinders 'motor' this cut-out cylinder. The brake power is measured by keeping the speed constant at its original value ($bp_{n-1}$).
- The difference in the outputs is a measure of the indicated horse power of the cut-out cylinder.
  The $ip$ of the $n^{th}$ cylinder is given by $(ip)\ n^{th} = bp_n - bp_{n-1}$
- Thus, for each cylinder the $ip$ is obtained and is added together to find the total $ip$ of the engine.
  And the total $ip$ of the engine is, $ip = \sum_{n=1}^{n} ip_n$
- The indicated power of '$n$' cylinders is given by $ip_n = bp_n + fp$
  Friction power is obtained as $fp = ip_n - bp_n$
- Since, the engine is running at the same speed it is quite reasonable to assume that $fp$ remains constant.
  *This method though gives reasonably accurate results and is liable to errors due to changes in mixture distribution and other conditions by cutting-out one cylinder. In gasoline engines, where there is a common manifold for two or more cylinders the mixture distribution as well as the volumetric efficiency both change. Again, almost all engines have a common exhaust manifold for all cylinders and cutting out of one cylinder may greatly affect the pulsations in exhaust system which may significantly change the engine performance by imposing different back pressures.*

**c. Motoring Test**

- In the motoring test, the engine is first run up to the desired speed by its own power and allowed to remain at the given speed and load conditions for some time so that oil, water, and engine component temperatures reach stable conditions.
- The power of the engine during this period is absorbed by a swinging field type electric dynamometer, which is most suitable for this test.
- The fuel supply is then cut-off and by suitable electric-switching devices the dynamometer is converted to run as a motor to drive for 'motor' the engine at the same speed at which it was previously running.
- The power supply to the motor is measured which is a measure of the $fp$ of the engine. During the motoring test the water supply is also cut-off so that the actual operating temperatures are maintained.

- This method, though determines the *fp* at temperature conditions very near to the actual operating temperatures at the test speed and load, does, not give the true losses occurring under firing conditions due to the following reasons.
  - ✓ The temperatures in the motored engine are different from those in a firing engine because even if water circulation is stopped the incoming air cools the cylinder. This reduces the lubricating oil temperature and increases friction due to increase of oil viscosity. This problem is much more severe in air-cooled engines.
  - ✓ The pressure on the bearings and piston rings is lower than the firing pressure. Load on main and connecting road bearings are lower.
  - ✓ The clearance between piston and cylinder wall is more (due to cooling). This reduces the piston friction.
  - ✓ The air is drawn at a temperature less than when the engine is firing because it does not get heat from the cylinder (rather loses heat to the cylinder). This makes the expansion line to be lower than the compression line on the p-v diagram. This loss is however counted in the indicator diagram.
  - ✓ During exhaust the back pressure is more because under motoring conditions sufficient pressure difference is not available to impart gases. The kinetic energy is necessary to expel them from exhaust.

  Motoring method, however, gives reasonably good results and is very suitable for finding the losses due to various engine components. This insight into the losses caused by various components and other parameters is obtained by progressive stripping-off of the under progressive dismantling conditions keeping water and oil circulation intact. Then the cylinder head can be removed to evaluate, by difference, the compression loss. In this manner piston ring, piston etc. can be removed and evaluated for their effect on overall friction.

**d. Retardation test:**

Retardation test is conducted to find out the power loss due to friction in an IC engine.

- Initially the engine is made to run at the rated speed.
- Then the engine is brought to rest condition by cutting the fuel supply or spark to the engine. The time to retard the engine from an initial to final speed is noted.
- This procedure is done under no load condition. Correspondingly torque at no load is obtained as ‘T’ and the time to retard the engine is ‘$t_2$’.

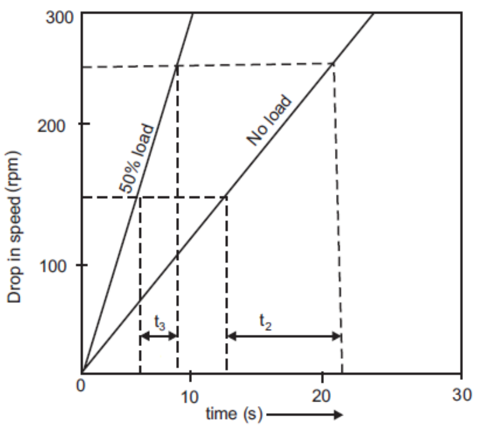


Fig: 8 Speed Vs Time

- This procedure is repeated by applying the load. For example, half of the load corresponding to the rated power is applied on the engine. Under this condition, corresponding torque at applied load is obtained as ‘$T_{1/2}$’ and the time to retard the engine is ‘$t_3$’.
- The torque on engine at no load will be solely due to friction which can be calculated and the power loss due to friction is calculated.

$$P = \frac{2\pi NT}{60000}$$

$$\text{Full load torque, } T = \frac{P \times 60000}{2\pi N}$$

$$T_f = \frac{t_3}{t_2 - t_3} \times \text{ Torque at 50\% load } = \frac{t_3}{t_2 - t_3} \times T_{1/2}$$

$$\text{Friction power } = \frac{2\pi N T_f}{60000}$$

***e.*** **Difference between *ip* and *bp***

(a) The method of finding the *fp* by computing the difference between *indicated power*, as obtained from an indicator diagram, and *bp*, as obtained by a dynamometer, is the ideal method.

(b) In obtaining accurate indicator diagrams, especially at high engine speeds, this method is usually only used in research laboratories. Its use at commercial level is very limited.

**Fuel Consumption Measurement**

Fuel consumption is measured in two ways:

- **Volumetric type**
  - ✓ The fuel consumption of an engine is measured by determining the volume flow in a given time interval and multiplying it by the specific gravity of the fuel which should be measured occasionally to get an accurate value.
  - ✓ Volumetric type flow meter includes Burette method, Automatic Burette flow meter and Turbine flow meter.
- **Gravimetric Fuel Flow Measurement**
  - ✓ By this method, one can measure the time required for consumption of a given mass of fuel.
  - ✓ There are three types of gravimetric type systems which are commercially available include Actual weighing of fuel consumed, Four Orifice Flow meter, etc.

  *The efficiency of an engine is related to the kilograms of fuel which are consumed and not the number of litres. The method of measuring volume flow and then correcting it for specific gravity variations is quite inconvenient and inherently limited in accuracy. Instead if the weight of the fuel consumed is directly measured a great improvement in accuracy and cost can be obtained.*

**Measurement of Air Consumption**

- ✓ In IC engines, the satisfactory measurement of air consumption is quite difficult because the flow is pulsating, due to the cyclic nature of the engine and because the air is a compressible fluid. Therefore, the simple method of using an orifice in the induction pipe is not satisfactory since the reading will be pulsating and unreliable.

- ✓ All kinetic flow-inferring systems such as nozzles, orifices and venturies have a square law relationship between flow rate and differential pressure which gives rise to severe errors on unsteady flow. Pulsation produced errors are roughly inversely proportional to the pressure across the orifice for a given set of flow conditions. The various methods and meters used for air flow measurement include

  (a) Air box method, and

  (b) Viscous-flow air meter.

  - ✓ **Air box method:** The usual method for measurement of air consumption is to ensure that all the air supplied to the engine is derived exclusively from an air box or tank (fig.) which is connected to the induction system of the engine by an air tight pipe of a diameter well in excess of that required theoretically for the predicted air flow. A box itself must be air tight. A sharp edged orifice is fitted to the pipe and the pressure difference across it is measured by means of a water manometer as shown in fig. As it is usually desirable to keep the Calculations simple it is necessary to keep the water manometer reading down to about 15 cm of water pressure difference, in which case the variation in the density of the air across the orifice is negligible. The air box or tank should have internal baffle so as to avoid any air pulsations, and its volume should be large enough in relation to the total capacity of the engine to be tested (say 200 to 600 times the total capacity), to prevent undue pressure pulsations.

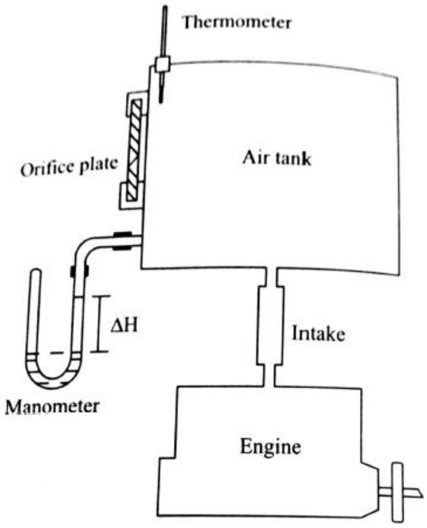



Fig: 9 Air box for air flow measurement

From the manometer the pressure head in mm of water is obtained i.e. ΔH.

But the total pressure remains constant i.e.

$$P = W(\Delta H) = \text{constant}$$

$$\Rightarrow \mathrm{W}_{air}(\Delta H)_{air} = \mathrm{W}_{water}(\Delta H)_{water}$$

$$\Rightarrow (\Delta H)_{air} = \left(\frac{\mathrm{W}_{water}}{\mathrm{W}_{air}}\right)(\Delta H)_{water}$$

Velocity of the air is obtained as $V = \sqrt{2g(\Delta H)_{air}}$

Theoretical Discharge or volume flow rate of air, $Q = A_{orifice} \times V = \frac{\pi}{4} d^2 \sqrt{2g(\Delta H)_{air}}$

By considering the coefficient of discharge of the orifice ($C_d$), we can determine the actual quantity of air entering the cylinder.

$$C_d = \frac{\text{actual discharge}}{\text{theoretical discharge}} = \frac{Q_{act}}{Q}$$

$$\Rightarrow Q_{act} = C_d \times Q = C_d \times \frac{\pi}{4} d^2 \sqrt{2g(\Delta H)_{air}}$$

Mass flow rate of air, $m_{air} = density \times actual discharge = \rho \times Q_{act}$

**Heat Balance sheet**

- To an engine, air is supplied to support the combustion of fuel and water is supplied to cool the various parts of an engine. Lubricant is supplied to reduce the friction in the engine wherever there is relative motion as well as to cool the parts where cooling water cannot be circulated.
- Heat generated in the combustion process is not completely converted into the useful work due to the losses in many forms like, friction, heat lost to cooling media, heat taken away by exhaust gases etc., Hence to understand the performance of the engine, it is needed to quantify this heat distribution by performing heat balance test.

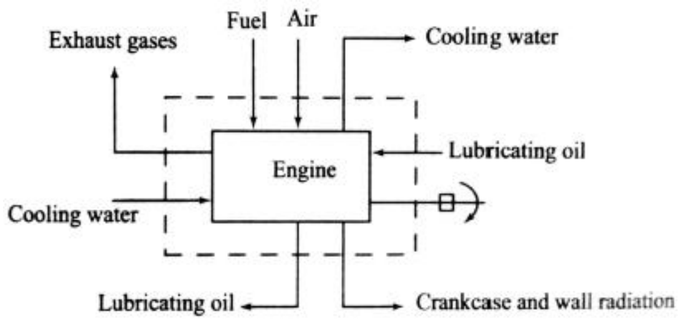


Fig. 10 Various inputs and outputs to an engine

- To draw a heat balance sheet for an I.C engine, it is run at constant load. During the test, the following parameters are to be measured:
  *Speed, load, fuel consumption, air consumption, exhaust temperature, rate of flow of cooling water and its rise in temperature while flowing through the water jackets.*
- Quantity of heat generated in combustion, $Q_s = m_f \times C.V$ kJ/min
  where $m_f$ – mass of fuel supplied in kg/min
  C.V – calorific value of fuel – kJ/kg
- Quantity of heat gained by engine cooling water, $Q_c = m_c C_p \left(T_2 - T_1\right)$ kJ/min
  where $m_c$ – mass of cooling water circulated in kg/min
  $C_p$ – specific heat of water in kJ/kg-K
  $T_1$, $T_2$ – initial and final temperatures of cooling water respectively.
- Quantity of heat taken away by exhaust gases, $Q_g = m_g C_{pg} \left(T_{g2} - T_{g1}\right)$ kJ/min
  where mass of exhaust gas in kg/min, $m_g = m_a + m_f$
  $m_a$ – mass of air supplied in kg/min
  $C_{pg}$ – specific heat of exhaust gas in kJ/kg-K
  $T_{g1}$, $T_{g2}$ – initial and final temperatures of exhaust gas respectively.
- Indicated power and brake power are also calculated.
- Finally unaccountable losses are calculated by subtracting the sum of heat equivalent of brake power, heat lost to engine cooling water, heat carried away by exhaust gases from the heat supplied.
  $Q_{acc} = Q_s - \left(Q_{bp} + Q_c + Q_g\right)$
- Finally the results are tabulated as shown below:

| Item | kJ/min | % of heat supplied |
|---|---|---|
| Heat supplied by the fuel | $Q_s$ | |
| Heat absorbed in B.P | $Q_{bp}$=bp x 60 | $Q_{bp}/Q_s$= |
| Heat taken away by engine cooling water | $Q_c$ | $Q_c/Q_s$= |
| Heat carried away by exhaust gases | $Q_g$ | $Q_g/Q_s$= |

| | | |
|---|---|---|
| Unaccountable losses | $Q_{acc}$ | $Q_{acc}/Q_s$= |
| Total | | |

- Heat distribution in a typical SI and CI engines are represented graphically as shown below.

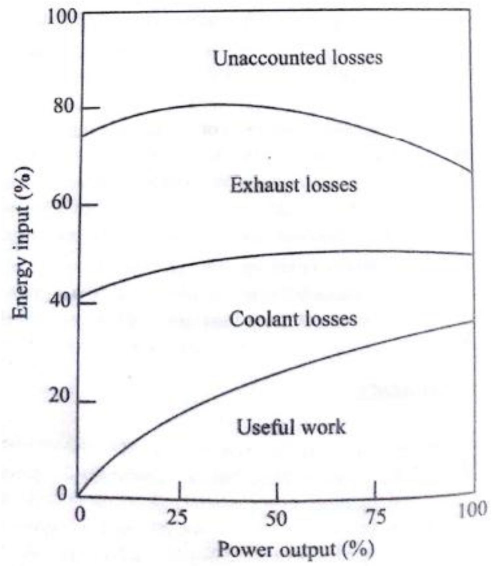



Fig. 11 Heat distribution in CI Engine

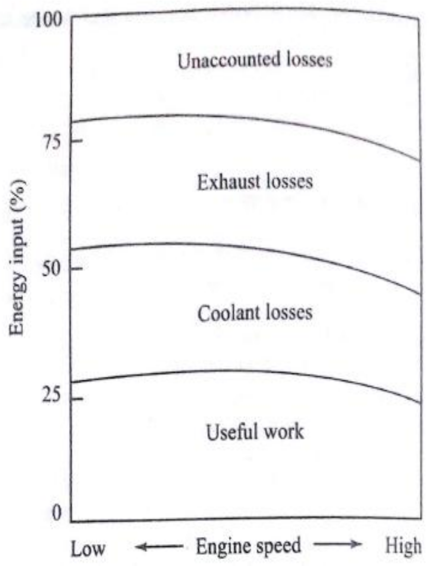



Fig. 12 Heat distribution in SI Engine

- ***Sankey Diagram***
  - ✓ The diagram starts at the bottom with a stream width representing the heat input from the fuel which is 100% of heat input.
  - ✓ While moving up, first the stream representing the heat taken away by the exhaust gases is let off to the left. The width of the stream represents the %loss to the coolant.
  - ✓ Still higher the stream representing the heat taken away exhaust gases is let off to left. The width of the stream represents the % of heat taken away by the exhaust gases.
  - ✓ Still higher the stream representing the heat lost to surroundings is let off to the left. The width of the stream represents the % of heat lost to the surroundings.
  - ✓ All these three streams meet a single loss stream as shown in the diagram.
  - ✓ A single vertical stream is left at the top of the diagram representing the brake power delivered by the engine.

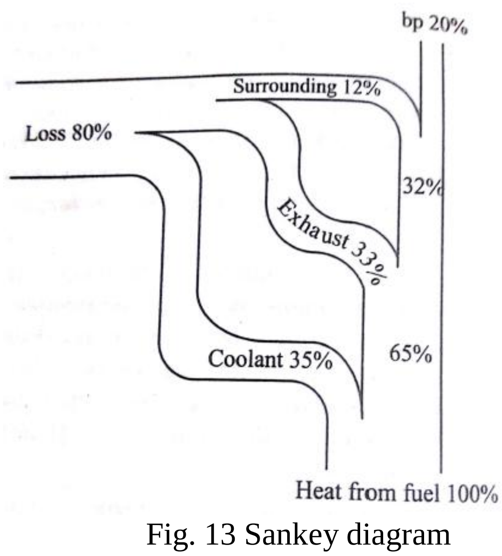



Fig. 13 Sankey diagram

# Unit - II

# Reciprocating Compressors

## Learning Objectives:

- To study the methods of evaluating performance of an internal combustion engine and the effects of various engine emissions as well as controlling measures.

## Syllabus:

**Reciprocating Compressors**: Principle of operation, Single stage of Compression – Work required, Isothermal efficiency, volumetric efficiency and effect of clearance, Free Air Delivered, displacement

**Multi stage compression** - under cooling, saving of work, minimum work condition for Multistage stage compression.

**Learning Outcomes:** At the end of the unit, the student will be able to

- Evaluate the performance of a reciprocating compressor
- Explain the benefits of multi-stage compression

**Reciprocating Compressors**

## Introduction:

Compressors are power consuming thermodynamic devices which convert mechanical energy into pressure head or pressure energy.

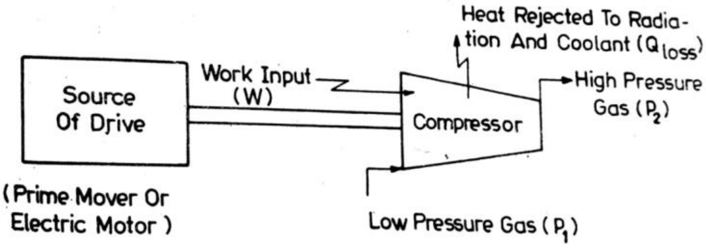



Fig. 1 Compressor

***Function***: The function of a compressor is to compress the gases and vapors from low pressure to high pressure. According to second law of thermodynamics, this is only possible when the work is done on the gas by an external agency such as prime movers, electric motors etc., using direct and indirect transmission.

Let $h_1$, $h_2$ – enthalpy of the working fluid before and after compression respectively.
$\delta Q$ – heat transfer with surroundings
$\delta W$ – work transfer
$Z_1$, $Z_2$ – elevation at the inlet and outlet of compressor respectively
$C_1$, $C_2$ – velocity of working fluid at the inlet and outlet of compressor respectively

Assuming it as a steady flow device, applying the steady flow energy equation to the process in the compressor.

$$h_1 + \frac{C_1^2}{2} + Z_1 g + \frac{\delta Q}{\delta m} = h_1 + \frac{C_1^2}{2} + Z_1 g + \frac{\delta W}{\delta m}$$

Neglecting the changes in kinetic and potential energy, and assuming that the compression is adiabatic, then the above equation modifies to

$$\frac{\delta W}{\delta m} = h_2 - h_1 \Rightarrow \delta W = \delta m\left(h_2 - h_1\right)$$

- ✓ *This equation indicates that the energy supplied in compression is tending to increase the enthalpy of the working fluid.*

**Classification of compressors**:

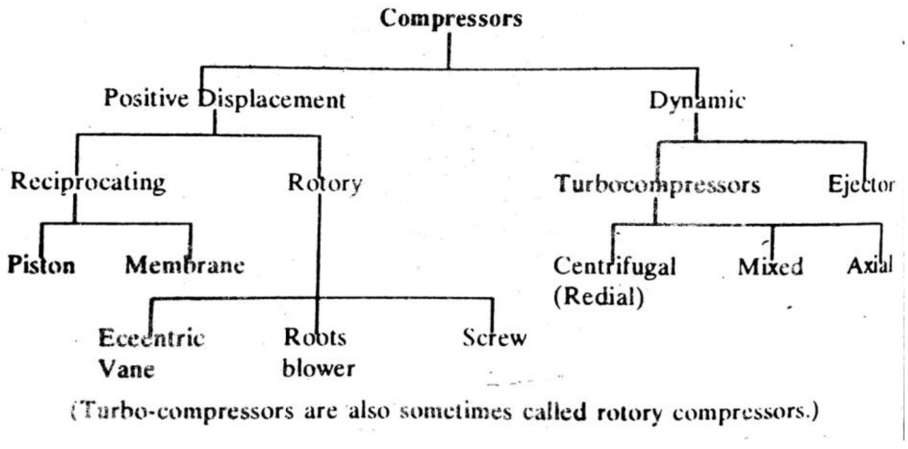



Fig. 2 Classification of Compressors

1. **According to design and Principle of operation**:
   i. *Positive displacement compressor*: The rise in pressure is achieved by positive displacement of gas. i.e. the pressure of gas is increased by decreasing its volume. *Examples: Reciprocating compressor, Rotary screw compressor, vane compressor.*
   ii. *Dynamic Compressors*: In these types, a rotating component imparts its kinetic energy to the air which is eventually converted into pressure energy. These use centrifugal force generated by a spinning impeller to accelerate and then decelerate captured air, which pressurizes it.
   *Examples: Axial compressors and centrifugal compressors.*
2. **According to Pressure:** On the basis of final pressure, the compressors are classified as:
   a. Low pressure compressors whose final pressure does not exceed 10bar,
   b. Medium pressure compressors with a range of 10 to 80 bar.
   c. High pressure compressors with a range of 81 to 1000bar.
   d. Multi stage reciprocating compressors for delivery pressure exceeding 1000 bar are termed as hyper compressors. They have as many as 7 stages.
3. **Depending upon the pressure rise limit**, the multi blade systems may be classified as (according to ASME code)
   a. Fan, in which pressure ratio < 1.1
   b. lower, in which pressure ratio >1.1 < 2.3
   c. Compressor in which the pressure ratio >2.3
4. **Based on Capacity:**
   A classification of compressors according to capacity, based on the quantity of free air delivered is as under:
   a. Small compressors handling upto $9m^3$/min.
   b. Medium compressors handling upto $9m^3$/min to $3000m^3$/min.
   c. Large compressors handling more than 3000 $m^3$/min.
5. **According to Number of stages**: The compressors may be single or multi stage.
6. **According to Medium to be compressed**: Compressors are also classified on the basis of medium to be compressed. It may be air, oxygen, hydrogen, nitrogen, acetylene, lighting gas and vapors and gases used in refrigeration cycle such as methyl chloride, ammonia, Fr-12, Fr-22, etc. Their physical and chemical properties exert a considerable influence upon the design of compressors.
7. **According to Method of Cooling**: They may be air cooled or water cooled.
8. **According to Arrangement of cylinders**: They may be horizontal, vertical, V type, W type, radial, horizontal balanced opposed, etc.
9. **Type of Drives**: They may be Motor driven, I.C. Engine driven, turbine driven, etc.

## Reciprocating Compressors

In this compressor, the gas volume decreases and pressure increases due to the action of one or more reciprocating pistons moving axially in one or more cylinders. It may be

- ✓ Single acting or double acting
- ✓ Single cylinder or multi cylinder
- ✓ Single stage or multi stage.

The compressor is used to meet the need for low, medium, and high compression ratio and low and medium gas volumes. For efficient operation, compressor’s maximum working limit is 300 $m^3$/min.

The reciprocating compressors have a very wide use in comparison to the other types of compressors. It is widely used in refrigeration system such as freeze, air conditioner and cold storage, mining works, chemical factories, fertilizer factories, garages, etc.

**Working of a Reciprocating Compressor:**

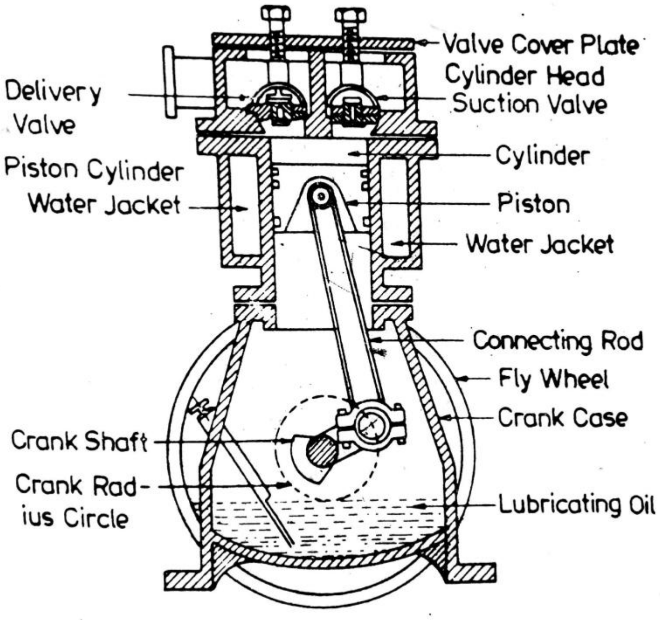



Fig. 3 Parts of a Reciprocating Compressor

Fig. 3 shows a single stage, single acting and single cylinder reciprocating compressor in simplified form.

Due to the rotation of the crank shaft, the piston moves in downward direction during which the pressure in the cylinder falls and the atmospheric air enters through the suction valve (delivery valve closed) till the piston reaches bottom dead centre position. When the piston goes up, the suction and delivery valves are both closed and the air is compressed till the delivery valve opens due to the difference in pressure in cylinder and delivery manifold. As the piston descends on the next downward stroke the air trapped in a clearance volume expands and the pressure falls to the suction pressure; the inlet valve then opens and the cycle is repeated.

**Single stage reciprocating compressor without clearance volume:**

Consider a single stage, single acting, ideal reciprocating compressor having no flow resistance at suction or delivery valve, no friction loses and no clearance. The working fluid is a perfect gas (say air).

Let, $p_1$, $V_1$ and $p_2$, $V_2$ be the pressure and volume before and after the compression respectively. Now consider the piston at the inner dead centre position at point 4, at this instant, the suction and delivery valves are closed.

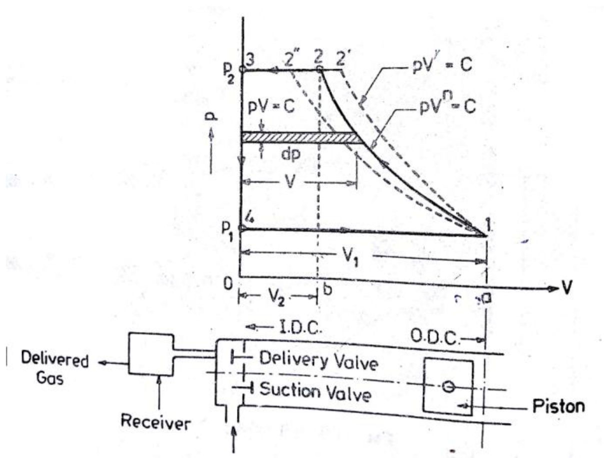



Fig. 4 p-V diagram – Compression process

When the piston moves outward a vacuum is created. Because of the pressure difference across the suction valve, the suction valve opens and the fluid is drawn into the cylinder at constant pressure $p_1$ through the suction manifold. The suction process is represented by 4-1. When the piston reaches the outer dead position, the suction valve closes at point 1, and the piston then starts moving towards inner dead centre position. Since both valves are closed, the induced fluid enclosed in the cylinder volume diminishes, causing a pressure rise. The compression process is shown by 1-2.

The compression process will stop only when the pressure rise in the cylinder exceeds the pressure in the delivery manifold by an amount equal to the resistance offered to the passage of the fluid through the delivery valve. At point 2 the delivery valve opens and the compressed fluid is displaced by the piston at an approximately constant pressure $p_2$ in the delivery manifold. The displacement of the fluid is represented by the process 2-3. It ends when the piston reaches the inner dead centre position where the delivery valve closes. As the piston starts to move on a new suction stroke, the pressure of gas $p_2$ from point 3 falls instantaneously to suction pressure $p_1$ at 4 and the suction valve opens and the entire cycle is repeated.

Work transfer in 1-2 process is given by $-W = \int_{p_1}^{p_2} v\,dp$ per cycle.

Assume $pv^n = c$ for compression process.

$$pv^n = p_1 v_1^n = p_2 v_2^n$$

$$-W = \frac{n}{n-1}\left[p_2 v_2 - p_1 v_1\right] = \frac{n}{n-1} p_1 v_1 \left( \left[\frac{p_2}{p_1}\right]^{\frac{n-1}{n}} - 1 \right)$$

We know that $p_1 v_1 = mRT_1$

$$-W = \frac{n}{n-1} mRT_1 \left( \left[\frac{p_2}{p_1}\right]^{\frac{n-1}{n}} - 1 \right)$$ kW where m is in kg/s.

**Comparison of Work in different Processes:**

1. If the compression is adiabatic, then
$$-W = \frac{\gamma}{\gamma-1} mRT_1 \left( \left[\frac{p_2}{p_1}\right]^{\frac{\gamma-1}{\gamma}} - 1 \right)$$
2. If the compression is isothermal ( $pv = c$ ), i.e all the heat generated during compression is withdrawn, so that temperature of the fluid remains constant, less work will be required to be done on per kg of air to raise its pressure from $p_1$ to $p_2$.
   Work done on the gas per cycle in isothermal compression is

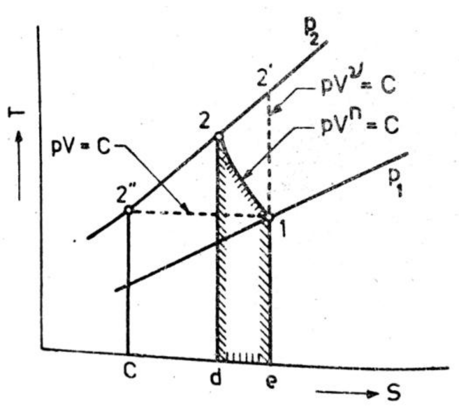


Fig. 5 Comparison of work in different process

$$-W=\int_{v_1}^{v_2} pdv + p_2v_2 - p_1v_1 = \int_{p_2}^{p_1} vdp = mRT_1 \ln\left[\frac{p_2}{p_1}\right]$$

**Single stage compression with clearance**

In actual compressor there is a clearance between the cylinder head and piston when the piston is at the TDC position. It is provided to prevent the piston striking the cylinder head.

The volume swept by the piston in one stroke i.e. when moving from TDC to BDC or BDC to TDC is called swept volume ($V_s$).

The ratio of clearance volume to swept volume is called as clearance ratio (C).

$$C=\frac{clearance\ volume}{swept\ volume\ (or)\ stroke\ volume}=\frac{V_c}{V_s}$$

***Work done with clearance***

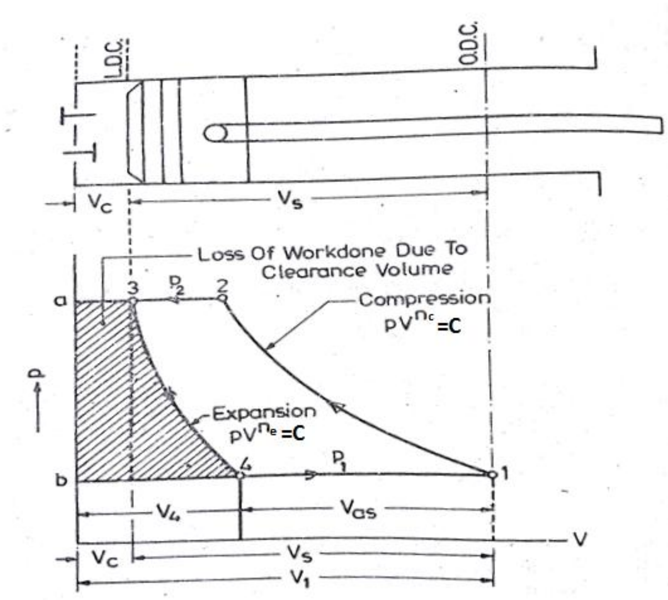



Fig. Effect of Clearance

Because of clearance, the actual volume of the gas sucked per working cycle decreases.

- *Process 1-2*: In the p-V diagram, it represents the compression process. During this process both the suction and delivery valves closes. The pressure and temperature increases from $p_1$ to $p_2$ and $T_1$ to $T_2$ respectively. After compression, the delivery valve opens resulting in the discharging of gas at constant pressure. It follows the law, $pV^{n_c}=c$.
- *Process 2-3*: It represents the discharge of gas at constant pressure i.e. $p_2 = p_3$. By the time delivery valve closes, still some gas remains in the clearance volume. *The volume at point 3 represents the clearance volume when the piston is at TDC ($V_c$=$V_3$).*
- *Process 3-4*: After discharge the piston starts moving from TDC resulting in expansion of gas during which pressure in the cylinder decreases from $p_3$ to $p_4$. At the point '4' suction valve opens and the actual suction process initiates. *This process is called wire drawing or re-expansion process.* It follows the law, $pV^{n_e}=c$.
- *Process 4-1*: From point '4' to '1', the actual suction of gas happens during which the pressure remains constant i.e. $p_4 = p_1$. At the point '2' the piston reaches BDC, the suction valve closes and the compression process initiates.
- *When there is no clearance, the suction process should initiate at point '3' but in practice, due to the clearance the stroke volume decreased from $V_1$-$V_3$ to $V_1$-$V_4$.*
- Net Work done = Work needed for compression without clearance – work done by the gas in expanding the gas from 3- 4

$$-W=\frac{n_c}{n_c-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{n_c-1/n_c}-1\right]-\frac{n_e}{n_e-1}p_4V_4\left[\left(\frac{p_3}{p_4}\right)^{n_e-1/n_e}-1\right]$$

But $p_2 = p_3$ and $p_4 = p_1$. Also assuming that $n_c = n_e$ i.e. index of compression is equal to index of expansion, then the above equation transforms to

$$-W = \frac{n_c}{n_c - 1} p_1 V_1 \left[ \left( \frac{p_2}{p_1} \right)^{n_c - 1 / n_c} - 1 \right] - \frac{n_c}{n_c - 1} p_1 V_4 \left[ \left( \frac{p_2}{p_1} \right)^{n_c - 1 / n_c} - 1 \right]$$

$$-W = \frac{n_c}{n_c - 1} p_1 \left( V_1 - V_4 \right) \left[ \left( \frac{p_2}{p_1} \right)^{n_c - 1 / n_c} - 1 \right]$$

**Volumetric efficiency**

- It is defined as the ratio of actual volume of gas sucked to the swept volume of the piston.

$$\eta_{vol} = \frac{actual\ volume}{swept\ volume} = \frac{V_a}{V_s}$$

Swept volume, $V_s = V_1 - V_3$

Clearance volume, $V_c = V_3$

Clearance ratio, $C = \frac{V_c}{V_s}$

Again process 3-4 is following the law, $pV^n = constant$

$$\Rightarrow p_3 V_3^n = p_4 V_4^n \Rightarrow V_4^n = \frac{p_3 V_3^n}{p_4}$$

$$\Rightarrow V_4 = \left( \frac{p_2}{p_1} \right)^{1/n} V_3 \Rightarrow V_4 = \left( \frac{p_2}{p_1} \right)^{1/n} V_c$$

Actual volume, $V_a = V_1 - V_4 = \left( V_1 - V_3 \right) + V_3 - V_4 = V_s + V_c - V_4$

Finally Volumetric efficiency,

$$\eta_{vol} = \frac{V_a}{V_s} = \frac{V_s + V_c - V_4}{V_s} = \frac{V_s}{V_s} + \frac{V_c}{V_s} - \frac{V_4}{V_s} = 1 + C - \frac{V_c}{V_s} \left( \frac{p_2}{p_1} \right)^{1/n} = 1 + C - C \left( \frac{p_2}{p_1} \right)^{1/n}$$

***Noteworthy points:***

1. If $p_2 = p_1$, i.e. $\eta_{vol} = 100\%$. This indicates that no compression takes place.
2. If $p_2$ increases, then the ratio $(p_2/p_1)$ increases (with other terms remaining constant), and ultimately results in decrease of volumetric efficiency.
3. When the clearance ratio, C and pressure ratio $(p_2/p_1)$ are fixed, then the volumetric efficiency increases with re-expansion coefficient, n.
4. From the above equation, it can be concluded that re-expansion or throttling or wire-drawing reduces the volumetric efficiency and also causes an increase in compressor work.

**Effect of higher compression ratio – Single stage compression**

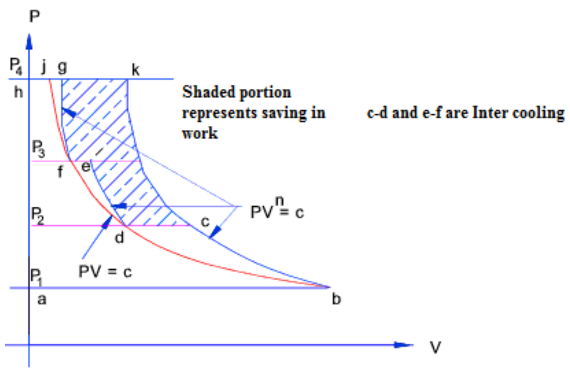
In a single stage compressor, if the compression ratio increases to higher values, the following effects are observed:

- Final temperature increases affecting the operation of delivery valves, diminishes lubricating properties of the oil and increase of ignition in piping and receiver.
- Robust cylinder construction is needed.

To avoid these problems, the higher compression ratios are attained by compressing the fluid in number of stages.

**Principle of multi-stage compression**

- Number of cylinders are arranged such that working fluid is first compressed in the cylinder-1 from pressure $p_1$ to $p_2$, then cooled in a device called intercooler at constant pressure ($p_2$ = $p_3$) and again compressed in the cylinder-2 from pressure $p_3 – p_4$.



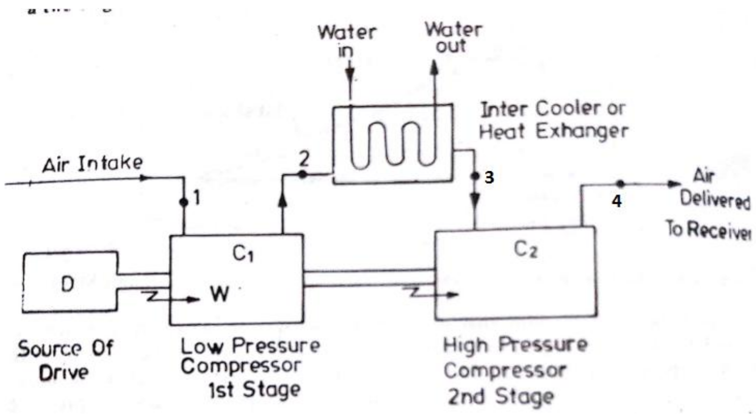

- Again the air at '4' is cooled at pressure $p_4$ and then compressed in the cylinder-3
- Because of cooling the fluid before compression in another cylinder, the index of compression decreases thereby reducing the work input to compress the fluid.

**Advantages of multi-stage compression:**

- Less power is needed to run a compressor
- Better mechanical balance
- Increase in volumetric efficiency
- Improved lubrication
- Less leakage loss
- Lighter cylinder

**Working of a 2-stage compressor**

- Air enters at pressure $p_1$ and temperature $T_1$ in the cylinder-1 or low pressure compressor and is compressed to some intermediate pressure $p_2$. During this process, the temperature also increases from $T_1$ to $T_2$.




- At this pressure and temperature $p_2$, $T_2$ respectively, the air enters the intercooler where it is cooled to the temperature at the Fig. Schematic of 2-stage compressor inlet of low pressure compressor ($T_3$ = $T_1$). During this process the pressure remains constant i.e. $p_2 = p_3$.
- After cooling the air is admitted to cylinder-2 or high pressure cylinder where its pressure is rised to the required value $p_4$. The corresponding temperature is $T_4$.
- Work supplied for compression in L.P cylinder,

$$W_{1-2} = \frac{n}{n-1}\left[p_2V_2 - p_1V_1\right] = \frac{n}{n-1}\left(p_1V_1\right)\left[\frac{p_2V_2}{p_1V_1} - 1\right] = \frac{n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_2}{p_1}\right)^{n-1/n} - 1\right]$$

- Work supplied for compression in H.P cylinder,

$$W_{3-4}=\frac{n}{n-1}\left[p_4V_4-p_3V_3\right]=\frac{n}{n-1}\left(p_3V_3\right)\left[\frac{p_4V_4}{p_3V_3}-1\right]=\frac{n}{n-1}\left(p_3V_3\right)\left[\left(\frac{p_4}{p_3}\right)^{n-1/n}-1\right]$$

- Also $p_2 = p_3$. Assuming the perfect intercooling, i.e. $T_3 = T_1$, we can write $p_1V_1 = p_3V_3$

  Now the work in H.P compressor can be rewritten as $W_{3-4}=\frac{n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_4}{p_2}\right)^{n-1/n}-1\right]$

  .

- Network supplied to the 2-stage compressor,

  $$W=W_{1-2}+W_{3-4}=\frac{n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_2}{p_1}\right)^{n-1/n}-1\right]+\frac{n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_4}{p_2}\right)^{n-1/n}-1\right]$$

  On simplification, we get $W=\frac{n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_2}{p_1}\right)^{n-1/n}+\left(\frac{p_4}{p_2}\right)^{n-1/n}-2\right]$

**Minimum work needed in 2-stage compression**

- Work required in a stage compressor is given as

  $$W=\frac{n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_2}{p_1}\right)^{n-1/n}+\left(\frac{p_4}{p_2}\right)^{n-1/n}-2\right]$$

- If the operating pressures $p_1$ and $p_4$ are fixed, the work is a function of intermediate pressure, $p_2$. Hence for minimum work input, the following condition is to be satisfied.

  $$\frac{dW}{dp_2}=0.$$

  $$\frac{n-1}{n}\left(\left(\frac{p_2}{p_1}\right)^{\left(n-1/n\right)-1}\left(\frac{1}{p_1}\right)+\left(\frac{p_4}{p_2}\right)^{\left(n-1/n\right)-1}\left(\frac{-p_4}{p_2^2}\right)\right)=0$$

  $$\Rightarrow\left(\frac{p_2}{p_1}\right)^{\frac{n-1-n}{n}}\left(\frac{1}{p_1}\right)=\left(\frac{p_4}{p_2}\right)^{\frac{n-1-n}{n}}\left(\frac{p_4}{p_2^2}\right)\Rightarrow\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\left(\frac{p_2}{p_1}\right)=\left(\frac{p_4}{p_2}\right)^{\frac{1}{n}}\left(\frac{p_4}{p_2}\right)$$

  $$\Rightarrow\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}+1}=\left(\frac{p_4}{p_2}\right)^{\frac{1}{n}+1}\Rightarrow\left(\frac{p_2}{p_1}\right)=\left(\frac{p_4}{p_2}\right)$$

  $$\Rightarrow p_2^2=p_1p_4\Rightarrow p_2=\sqrt{p_1p_4}$$

  Above equation gives the optimum intermediate pressure which is to be maintained for minimum work to be supplied in 2-stage compression.

- Minimum work to be supplied is obtained by substituting for $p_2$ in the equation for work

  $$W=\frac{n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_2}{p_1}\right)^{n-1/n}+\left(\frac{p_4}{p_2}\right)^{n-1/n}-2\right]$$

$$\Rightarrow W_{\min} = \frac{n}{n-1}\left(p_1V_1\right)\left[\left(\frac{\sqrt{p_1p_4}}{p_1}\right)^{n-1/n} + \left(\frac{p_4}{\sqrt{p_1p_4}}\right)^{n-1/n} - 2\right]$$

$$\Rightarrow W_{\min} = \frac{n}{n-1}\left(p_1V_1\right)\left[\left(\sqrt{\frac{p_4}{p_1}}\right)^{n-1/n} + \left(\sqrt{\frac{p_4}{p_1}}\right)^{n-1/n} - 2\right]$$

$$\Rightarrow W_{\min} = \frac{2n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_4}{p_1}\right)^{n-1/2n} - 1\right]$$

**Note:**

Similarly for a 3-stage compressor, $W_{\min} = \frac{3n}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_4}{p_1}\right)^{n-1/3n} - 1\right]$

Similarly for an N-stage compressor, $W_{\min} = \frac{Nn}{n-1}\left(p_1V_1\right)\left[\left(\frac{p_4}{p_1}\right)^{n-1/Nn} - 1\right]$

**Indicator Diagram**

It is a diagram representing the variation of pressure with volume during the processes in a compressor.

**Mean effective pressure**

Mean effective pressure is obtained by dividing the area of the indicator pressure with the length of the diagram.

Indicated mean effective pressure,

($imep$) = $p_m$ = mean diagram height x spring constant

$$p_m = \frac{area\ of\ \ indicator\ \ card\left(cm^2\right)\times spring\ \ cons\tan t\left(bar/cm\right)}{length\ of\ diagram\left(cm\right)}$$

Let A – piston area, $m^2$

L – length of the stroke, m

N – crank shaft speed, rpm

K – number of cylinders

n = 1 for single acting compressor and n = 2 for double acting compressor

Work done per cycle, $W = 100\times p_m\times A\times L\times n\times K$

Work done per minute (or) Indicated power,

$$W = \frac{100\times p_m\times A\times L\times n\times K}{60}$$

For single acting and single cylinder,

$$IP = \frac{100\times p_m\times A\times L}{60}$$

**Compressor power efficiencies**

i. Mechanical efficiency: It is defined as the ratio of the indicated power to the brake power.

$$\eta_m = \frac{ip\ of\ \ compressor}{bp\ of\ \ compressor}$$

ii. Isothermal efficiency: It is the ratio of the isothermal work input to the actual work input to the compressor.

$$\eta_{iso} = \frac{isothermal\ \ work\ \ input}{actual\ \ work\ \ input} = \frac{p_1 v_1 \ln\left( {p_2}/{p_1} \right)}{\left(\frac{n}{n-1}\right) p_1 v_1 \left[ \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right]} = \frac{\ln\left( {p_2}/{p_1} \right)}{\left(\frac{n}{n-1}\right) \left[ \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right]}$$

Work done in Isothermal process consumes less amount of work. But in practice temperature increases during the compression, resulting in increase of index of compression. This increases the amount of work supplied in actual practice. Hence isothermal efficiency is always less than 100%.

iii. **Adiabatic efficiency**: It is the ratio of actcual work input to the adiabatic work input.

$$\eta_{adiab} = \frac{Actual\ \ work\ \ input}{Adiabatic\ \ work\ \ input} = \frac{\left(\frac{n}{n-1}\right) p_1 v_1 \left[ \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right]}{\left(\frac{\gamma}{\gamma-1}\right) p_1 v_1 \left[ \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]} = \frac{\left(\frac{n}{n-1}\right) \left[ \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right]}{\left(\frac{\gamma}{\gamma-1}\right) \left[ \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}$$

# Unit - III

# Steam Power cycles

# Learning Material

**Objectives**

- To study and analyze the steam power cycles.

**Syllabus**

**Steam Power Cycles:** Rankine cycle - schematic layout, Thermodynamic analysis, concept of mean temperature of heat addition, Methods to improve cycle, performance - Regeneration & Reheating cycles.
**Boilers:** Classification, working of water tube and fire boilers, Mountings and Accessories.

At the end of the chapter student is able to

- Understand the working of steam power plant.
- Analyze the steam power plants working on Rankine cycle.
- identify and analyze the various methods of increasing the efficiency of Rankine cycle
- Analyze the reheat and regenerative vapor power cycles.
- Understand various Boiler mountings and accessories.

### 3.1 Vapor power cycles (Steam power plant)

A vapor power cycle continuously converts heat (energy released by the burning of fuel) into work (shaft work), in which working fluid repeatedly undergoes change of phases. In the vapor power cycle, the working fluid, which is water, undergoes a phase change.

Figure 1.1 gives schematic of a simple steam power plant working on the vapor power cycle.

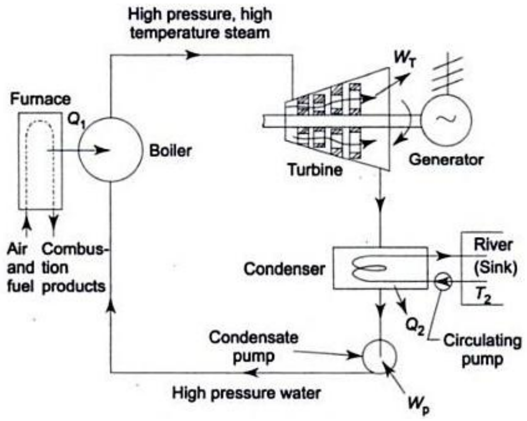



Fig.3.1 Simple steam power plant

**Boiler**

It is a closed vessel inside which combustion of fuel takes place. Tubes are arranged in the walls of the boiler through which water passes. The water by absorbing heat of combustion turns into steam.

**Turbine**

Steam from boiler passes through nozzles and enters into turbine. The high-pressure steam now expands over the blades of the turbine rotor (shaft upon which the circumferential blades are mounted). The pressure of steam drops down along with its enthalpy (total heat content). This drop in heat energy (enthalpy) is converted into mechanical energy. As a result, the shaft of the turbine rotates.

**Condenser**

This is located after the turbine so that steam after expansion in the turbine exhausts into the condenser. The exhaust steam from the turbine enters the condenser and major portion of it gets condensed. The condensed steam is called condensate and is recirculated as feed water to the boiler. With the addition of a condenser, we can extract more work from the turbine.

**Feed Pump**

It is a pump which takes up condensate and then forces into the boiler with pressure. As boiler works at higher pressure, feed pumps are necessary to raise the pressure of water for its entry to boiler.

Since the fluid is undergoing a cyclic process, there will be no net change in its internal energy over the cycle, and consequently the net energy transferred to the unit mass of the fluid as heat during the cycle must equal the net energy transfer as work from the fluid.

By the first law,

$$\sum_{\substack{cycle\\1}}^{2} Q_{net} = \sum_{\substack{cycle\\4}}^{3} W_{net}$$

or$Q_1 - Q_2 = W_T - W_P$

Where $Q_1$ = heat transferred to the working fluid (kJ/kg)

$Q_2$ = heat rejected from the working fluid (kj/kg)

$W_T$= Work transferred from the working fluid (kJ/kg)

$W_P$= Work transferred into the working fluid (kJ/kg)

The efficiency of the vapour power cycle would be given by

$$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{Q_1 - Q_2}{Q_1}$$

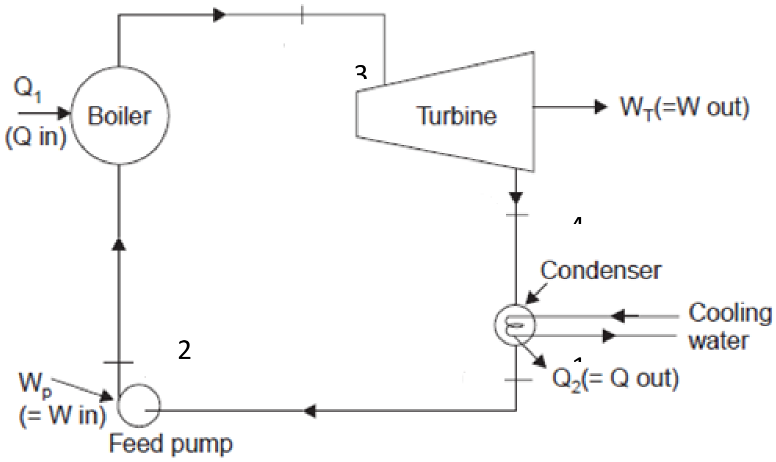
$$= 1 - \frac{Q_2}{Q_1}$$

**3.2 Rankine Cycle**

It is a theoretical cycle upon which steam power plant works. It is a modified form of Carnot cycle and an ideal cycle for comparing the performance of steam power plants. In which heat addition and rejection takes place at constant pressure process.But the thermal efficiency of a Rankine cycle is lower than that of a Carnot cycle operating between the same temperature levels.

**Analysis of Rankine Cycle**

For purposes of analysis the Rankine cycle is assumed to be carried out in a steady flow operation. Applying the steady flow energy equation (S.F.E.E) to each of the processes on the basis of unit mass of fluid, and neglecting changes in kinetic and potential energy, the work and heat quantities can be evaluated in terms of the properties of the fluid.




A simple steam power plant

**Assumptions**

The following assumptions are made in the working of Rankine cycle:

1. The same working fluid is repeatedly circulated in a closed circuit.
2. Heat is added in boiler only and rejected in condenser only. Except boiler and condenser, there is no heat transfer between working fluid and surroundings.
3. There is no pressure drop in the piping system. Expansion in the prime mover occurs without friction or heat transfer i.e., expansion is isentropic in which case entropy of working fluid entering and leaving the prime mover is same.
4. The working fluid is not under cooled in the condenser i.e., the temperature of water leaving the condenser is same as saturation temperature corresponding to the exhaust pressure.

**Based on assumptions following are the various process**

1-2 Isentropic compression in a pump

2-3 Reversible constant pressure heat addition in a boiler

3-4 Isentropic expansion in a turbine

4-1 Reversible constant pressure heat rejection in a condenser

The cycle has been plotted on the p-v, T-s, and h-s planes as shown in Fig.

The numbers on the plots correspond to the numbers on the flow diagram.

- For any given pressure, the steam approaching the turbine may be dry saturated (state 3) wet (state 3'), or superheated (state 3"), but the fluid approaching the pump is, in each case, saturated liquid (state 1).
- Steam expands reversibly and adiabatically in the turbine from state 3 to state 4 ( or 3' to 4', or 3" to 4").
- The steam leaving the turbine condenses to water in the condenser reversibly at constant pressure from state 4 ( or 4', or 4") to state 1.
- The water at state 1 is then pumped to the boiler at state 2 reversibly and adiabatically.
- The water is heated in the boiler to form steam reversibly at constant pressure from state 2 to state 3 (or 3' or 3").

For 1 kg of fluid .

The S.F.E.E for the Turbine (control volume) gives

$$h_3 = W_T + h_4$$

$$\therefore \; W_T = h_3 - h_4$$

Similarly The S.F.E.E for the condenser (control volume) gives

$$h_4 = Q_2 + h_1$$

$$\therefore \quad Q_2 = h_4 - h_1$$

and the S.F.E.E for the pump (control volume) gives

$$h_1 + W_p \ = h_2$$

$$\therefore \quad W_P = h_2 - h_1$$

The S.F.E.E for the boiler (control volume) gives

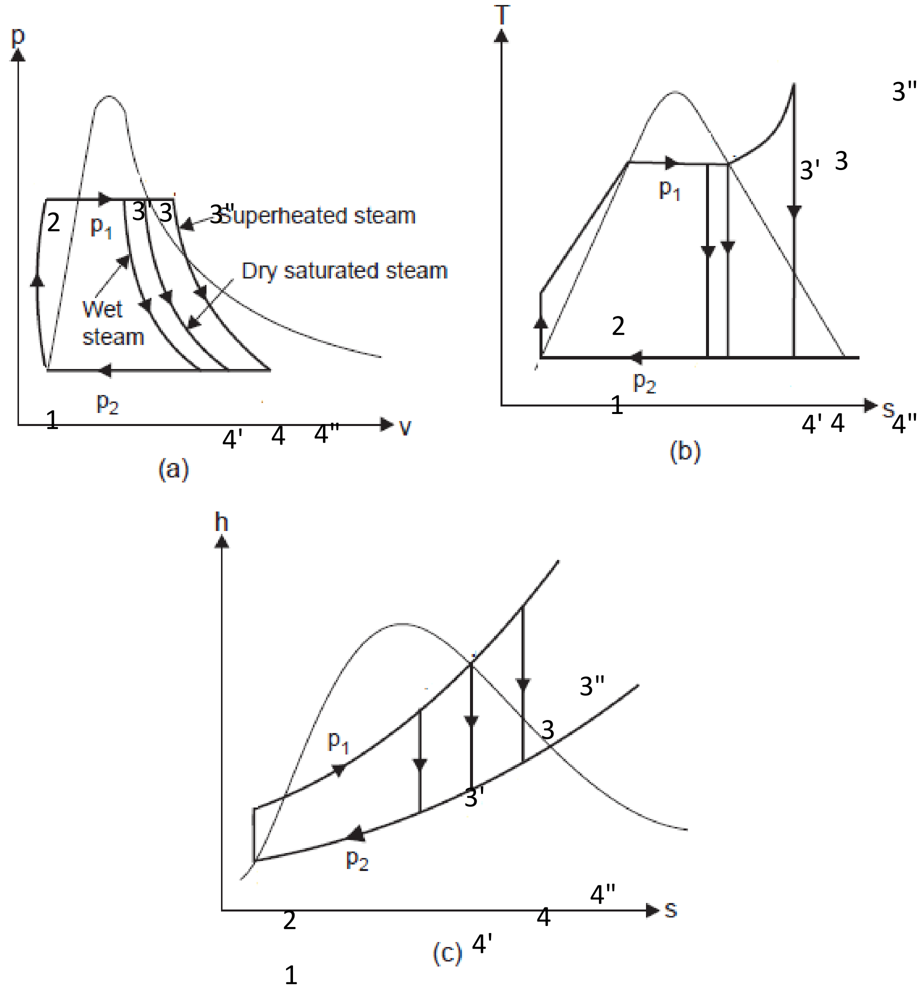
$$h_2 + Q_1 \ = h_3$$

$$\therefore \quad Q_1 = h_3 - h_2$$

The efficiency of the Rankine cycle is then given by

$$\eta_{cycle} \ = \frac{W_{net}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)}$$

The pump handles liquid water which is incompressible, i.e., its density or specific volume undergoes little change with increase in pressure. For reversible adiabatic compression, by the use of the general property relation.




Rankine cycle on p-v, T-s and h-s planes

$$Tds = dh - vdp$$

$$ds = 0$$

Since the change in speifc volume is negligable

$$dh = vdp$$

$$\Delta h = v\ \Delta p$$

$$h_2 - h_1 = v_1\,(p_1 - p_2)$$

$v$ is in $m^3$/ kg and p is in bar

$$h_2 - h_1 = v_1\,(p_1 - p_2) \times 10^5 \text{J/kg}$$

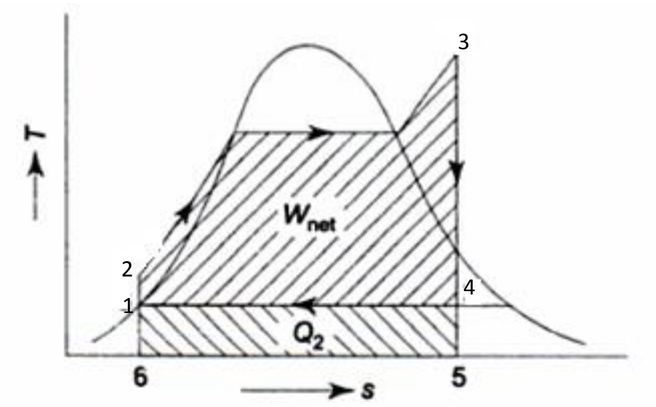
The work ratio is defined as theratio of net workoutput to positive work output

$$\therefore\ \ work\ ratio = \frac{W_{net}}{W_T} = \frac{W_T\ - W_P}{W_T}$$

Usually , the pump work is quite small compared to the turbine work and is somtimes neglected .The $h_4 = h_3$ , and the chycle efficency approximately becomes

$$\eta \equiv \frac{h_3 - h_4}{h_3 - h_2}$$

The efficency of the Rankine Cycle is presented graphically in the T-s plot in Fig.below. Thus $Q_1$ is proportional to area 3562, $Q_2$ is proportional to area 4561, and $W_{net}$ (= $Q_1$- $Q_2$) is proportinal to area 1234 enclosed by the cycle.




$Q_1$, $W_{net}$ and $Q_2$ are proportional areas

The capacity of a steam plant is often expressed in terms of steam rate, which is defined as the rate of steam flow (kg/h) required to produce unit shaft output(1 kW). Therefore

$$\text{Steam Rate}\ \ = \frac{1}{\mathrm{W_T - W_P}}\frac{\text{kg}}{\text{kJ}}\cdot\frac{1\ \text{kJ/s}}{1\ \text{kW}}$$

$$= \frac{1}{\mathrm{W_T - W_P}}\frac{\text{kg}}{\text{kWs}} = \frac{3600}{\mathrm{W_T - W_P}}\frac{\text{kg}}{\text{kWh}}$$

The cycle efficiency is sometimes expressed alternatively as heat rate which is the rate input ($Q_1$) required to produce unit work output ( 1kW)

$$\text{Heat Rate} = \frac{3600\, Q_1}{W_T - W_P} = \frac{3600}{\eta_{cycle}} \frac{\text{kJ}}{\text{kWh}}$$

From the equation $W_{rev} = -\int v\, dp$, it is obvious that the reversible steady flow work is closely associated with the specific volume of fluid flowing through the device.

*The larger the specific volume, the larger the reversible work produced or consumed by the steady-flow device. Therefore, every effort should be made to keep the specific volume of a fluid as small as possible during a compression process to minimize the work input and as large as possible, during an expansion process to maximize the work output.*

***In steam power plants, the pump handles liquid, which has a very small specific volume, and the turbine handles vapor, whose specific volume is many times larger. Therefore, the work output of the turbine is much larger than the work input to the pump. This is one of the reasons for the overwhelming popularity of steam power plants in electric power generation***.

If we were to compress the steam exiting the turbine back to the turbine inlet pressure before cooling it first in the condenser in order to "save" the heat rejected, we need to supply the work which is many times higher than that is produced by turbine . Hence heat rejection in the condenser is necessary.

### 3.3 Actual Vapour Cycle Processes

The processes of an actual cycle differ from those of the ideal cycle. In the actual cycle conditions indicated in Fig 1.6, showing the various losses. The thermal efficiency of the cycle is

$$\eta_{th} = \frac{W_{net}}{Q_1}$$

Where the work and heat quantities are the measured values for the actual cycle, which are different from the corresponding quantities of the ideal cycle.

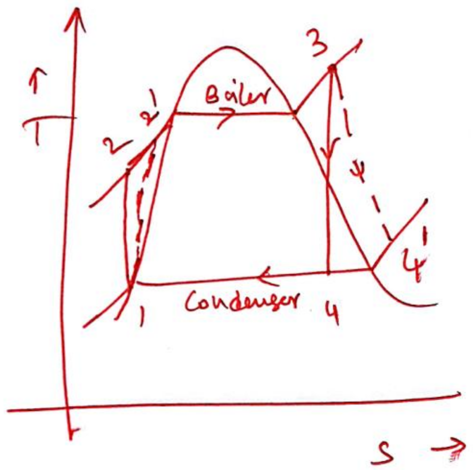
**Deviation of Actual Cycle from Ideal Cycle:**

In an ideal Rankine cycle the pump and turbine would be isentropic, i.e., the pump and turbine would generate no entropy and hence maximize the net work output. Processes 1-2 and 3-4 would be represented by vertical lines on the T- s diagram

The actual vapor power cycle differs from the ideal Rankine cycle because of irreversibilities in the inherent components caused by fluid friction and heat loss to the surroundings;

- fluid friction causes pressure drops in the boiler, the condenser, and the piping between the components, and as a result the steam leaves the boiler at a lower pressure;

- heat loss reduces the net work output, thus more heat addition to the steam in the boiler is required to maintain the same level of net work output, as a result efficiency decreases.




Deviation of actual vapor power cycle from the ideal Rankine cycle (*b*) The effect of pump and turbine irreversibilities on the ideal Rankine cycle

**1) Turbine Efficiency:**
During the expansion of steam in the turbine there will be heat transfer to the surroundings and the expansion instead of being isentropic will be polytropic as shown in the figure.

$3-4 \rightarrow$ Isentropic expansion

$3-4' \rightarrow$ Acutal expansion

Turbine Efficiency $=\eta_T = \frac{h_3 - h_4^1}{h_3 - h_4}$

**2) Pump Efficiency:**
There are losses in the pump due to irreversibility and the process of compression is polytropic instead of isentropic as shown above.

Pump Efficiency $=\eta_p = \frac{h_2 - h_1}{h_2^1 - h_1}$

**3.4 Mean temperature of Heat Addition**

In the Rankine cycle, heat is added reversibly at a constant pressure, but at infinite temperatures. If $T_{m1}$ is the mean temperature of heat addition, as shown in Fig, so that the area under 2s and 3 is equal to the area under 5 and 6, then heat added

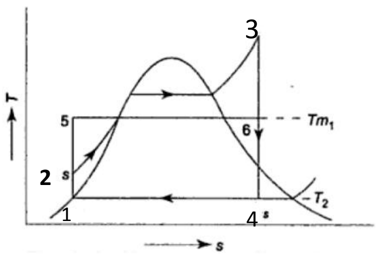



Fig. Mean temperature of heat addition

$$Q_1 = h_3 - h_{2s} = T_{m1}(S_3 - s_{2s})$$

$T_{m1}$ = Mean temperature of heat addition

$$= \frac{h_3 - h_{2s}}{s_3 - s_{2s}}$$

$Q_2$ = heat rejected = $h_{4s} - h_1$

$$= T_2(s_{4s} - s_1) = T_2(s_3 - s_{2s})$$

$$\eta_{Rankine} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2\,(s_3 - s_{2s})}{T_{m1}(s_3 - s_{2s})}$$

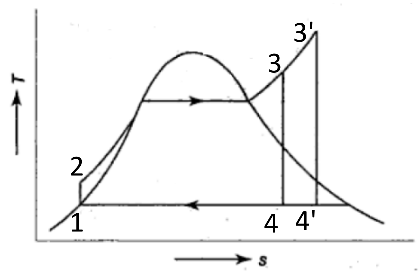
$$\eta_{Rankine} = 1 - \frac{T_2}{T_{m1}}$$

Where $T_2$ is the temperature of heat rejection. The lower is the $T_2$ for a given $T_{m1}$, the higher will be the efficiency of the Rankine cycle. But the lowest practicable temperature of heat rejection is the temperature of the surroundings ($T_0$).This being fixed,

$$\eta_{Rankine} = f(T_{m1})\ only$$

The higher the mean temperature of heat addition, the higher will be the cycle efficiency.

The effect of increasing the initial temperature at constant pressure on cycle efficiency is shown in Fig. When the initial state changes from 3 to 3', $T_{m1}$ between 2 and 3' is higher than $T_{m1}$between 2 and 3. So an increase in the superheat at constant pressure increases the mean temperature of heat addition and hence the cycle efficiency.




Effect of superheat on mean temperature of heat addition

**HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE**?

Steam power plants are responsible for the production of most electric power in the world, and even small increases in the thermal efficiency can mean large savings from the fuel requirements. Therefore, every effort is made to improve the efficiency of the cycle on which steam power plants operate.

The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same:

***Increase the mean temperature at which heat is transferred to the working fluid in the boiler, or decrease the mean temperature at which heat is rejected from the working fluid in the condenser.***

That is, the mean fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection.

Three ways of accomplishing this for the simple ideal Rankine cycle.

1. Lowering the Condenser Pressure (*Lowers* $T_{\text{low, mean}}$)
2. Superheating the Steam to High Temperatures (*Increase* $T_{\text{high, mean}}$)
3. Increasing the Boiler Pressure (*Increase*$T_{\text{high, mean}}$)

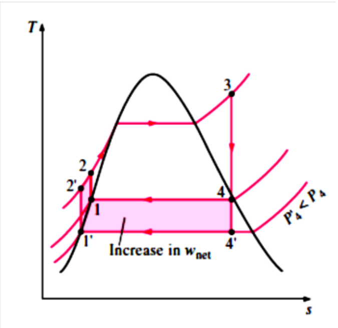
Lowering the Condenser Pressure (*Lowers* $T_{\text{low, mean}}$)

Lowering the operating pressure of the condenser automatically lowers the temperature of the steam, and thus the temperature at which the heat is rejected.

The effect of lowering the condenser pressure on the Rankine cycle efficiency is illustrated on a T-S diagram in Fig .

- For comparison purposes, the turbine inlet state is maintained the same. The colored area on this diagram represents the increase in net work output as a result of the lowering the condenser pressure from $P_4$ to $P'_4$ .

- The heat input requirements also increased (represented by the area under curve 2'-2), but this increase is very small. Thus the overall effect of lowering the condenser pressure is an increase in the thermal efficiency of the cycle.




Effect of lowering condenser pressure

To take advantage of the increased efficiencies at low pressures, the condensers of steam power plants usually operate well below the atmospheric pressure.

However, there is a lower limit on the condenser pressure that can be used. It cannot be lower than the saturation pressure corresponding to the temperature of the cooling medium.

Consider, for example, a condenser that is to be cooled by a nearby river at $15^0$C. Allowing a temperature difference of $10^0$C for effective heat transfer, the steam temperature in condenser must be above 25°C; thus the condenser pressure must be above 3.2 kPa, which is the saturation pressure at 25°C.

**Disadvantages Lowering the condenser pressure**

- It creates the possibility of air leakage into the condenser.

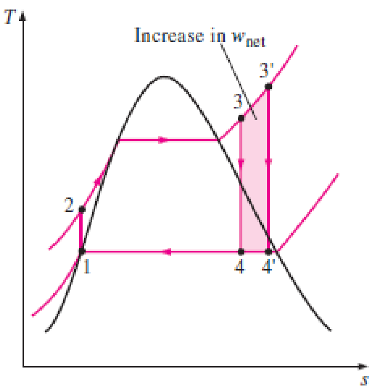
- It increases the moisture content of the steam at the final stage of the turbine, as can be seen from Fig.1.9.

The presence of large quantities of moisture is highly undesirable in turbines because it decreases the turbine efficiency and erodes the turbine blades.

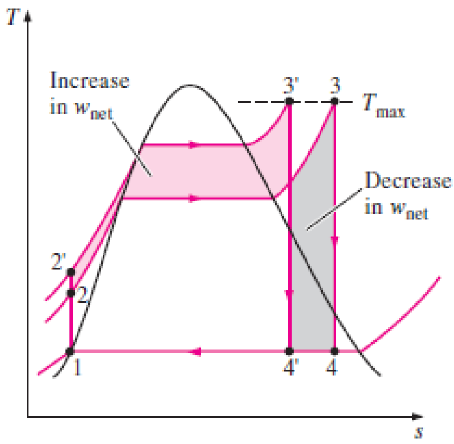
Superheating the Steam to High Temperatures (*Increase* $T_{high, mean}$)

The mean temperature at which heat is added to the steam can be increased without increasing the boiler pressure by superheating the steam to high temperature.

The effect of superheating on the performance of vapor power cycle is illustrated on the *T-s* diagram as shown in Fig. The total area under the process curve 3-3' represents the increase in the heat input. Thus both the net work and heat input increase as a result of super heating the steam to a higher temperature. The overall effect is an increase in thermal efficiency, however, since the average temperature at which heat is added increases.




Effect of superheating




Effect of Boiler pressure

It decrease the moisture content of the steam at the turbine exit, as can be seen from the *T-s* diagram (the quality at state 4' is higher than that at state 4).

However, The temperature to which steam can be superheated is limited, by metallurgical considerations. Presently the highest steam temperature allowed at the turbine inlet is about 620$^{o}$C (1150$^{o}$F). Any increase in this value depends on improving the present materials or finding new ones that can withstand higher temperatures.

Increasing the Boiler Pressure (*Increase*$T_{high, mean}$)

Another way of increasing the mean temperature during the heat-addition process is to increase the operating pressure of the boiler, which automatically raise the temperature at which boiling takes place. This, in turn, raises the average temperature at which heat is added to the steam and thus raises the thermal efficiency of the cycle.

The effect of increasing the boiler pressure on the performance of vapor power cycles is illustrated on *T-s* diagram in Fig.

Notice that for a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This undesirable side effect.

## 3.5 THE IDEAL REHEAT RANKINE CYCLE

We noted in the last section that increasing the boiler pressure increases the thermal efficiency of the Rankine cycle, but it also increases the moisture content of the steam to un acceptable levels. Then it is natural to ask the following question:

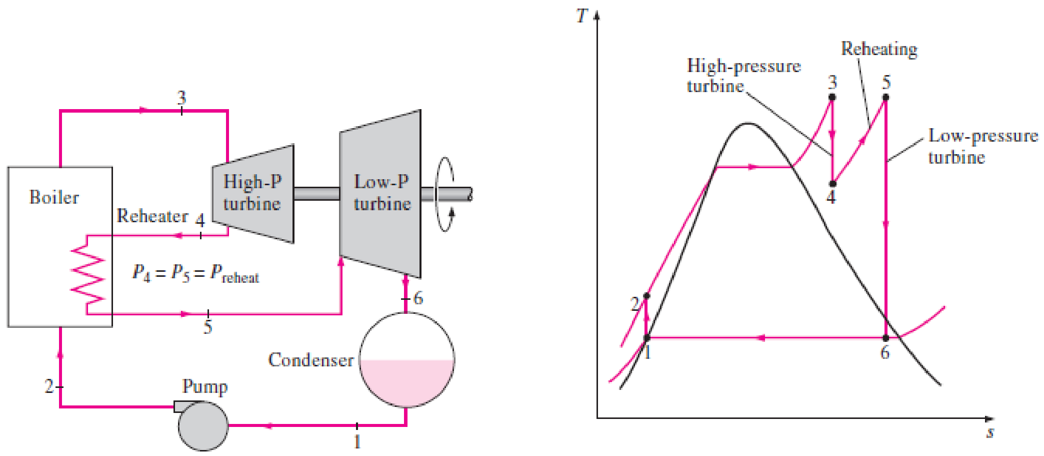
*How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine ?*

Two possibilities:

1.Superheat the steam to very high temperatures before it enters the turbine. This would be the desirable solution since the average temperature at which heat is added would also increase ,thus increasing the cycle efficiency. This is not a viable solution, however, since it will require raising the steam temperature to metallurgical unsafe levels.

2.Expand the steam in the turbine in two stages, and reheat it in between. In other words, modify the simple ideal Rankine cycle with a reheat process. Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern steam power plants.

The T-s diagram of the ideal reheat Rankine cycle and schematic of the power plant operating on this cycle are shown in Fig.




Ideal reheat Rankine cycle

The ideal reheat Rankine cycle differs from the simple ideal rankine cycle in that the expansion process takes place in two stages. In the first stage (the high pressure turbine), steam is expanded isoentropically to an intermediate pressure and sent back to the boiler where it is reheated at constant pressure, usually to the inlet temperature of the first turbine stage. Steam expands isentropically in the second stage (low pressure turbine) to the condenser pressure. Thus the total heat input and the total turbine work output for a reheat cycle become

$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4)$$

and

$$w_{turb.out} = w_{turb,\ I} + w_{turb,\ II} = (h_3 - h_4) + (h_5 - h_6)$$

The incorporation of the single reheat in a modern power plant improves the cycle efficiency by 4 to 5 percent by increasing the average temperature at which heat is added to the steam.

The average temperature during the reheat process can be increased by increasing the number of expansion and reheat stages. As the number of stages is increased, the expansion and reheat processes approach an isothermal process at the maximum temperature.

**Advantages of Reheating Steam:**

Reheating of steam in a turbine has the following advantages:

1. It increases output of the turbine.
2. Erosion and corrosion problems are avoided.
3. The thermal efficiency increases.
4. Nozzle and blade efficiencies increase.

**Disadvantages:**

1. Maintenance is more.
2. Relative to cost of reheating, increase in thermal efficiency is not appreciable.

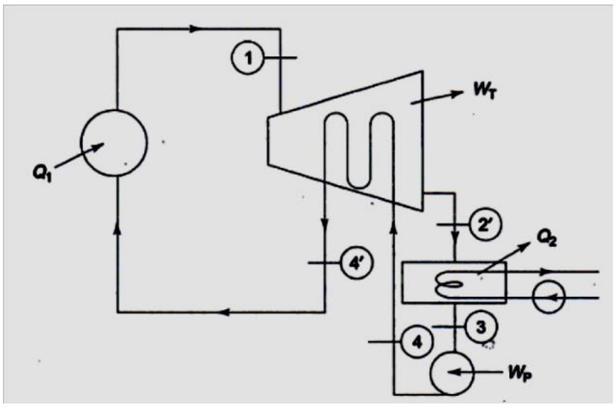
### 3.6 Ideal Regenerative Rankine cycle

In order to increase mean temperature of heat addition ($T_{m1}$), attention was so far confined to increasing the amount of heat supplied at high temperature, such as increasing super heat, using higher pressure and temperature of steam, and using reheat.

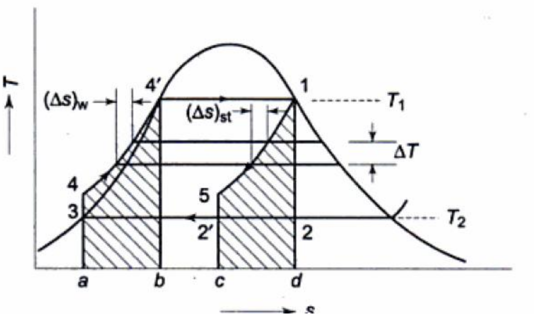
The mean temperature of heat addition can also be increased by decreasing the amount of heat added at low temperature.

In a saturated steam Rankine cycle, a considerable part of the total heat supplied in the liquid phase when heating up water from 2 to 2', at a temperature lower than $T_3$, the maximum temperature of the cycle. For maximum efficiency, all heat should be supplied at $T_3$, and feed water should enter the boiler at state 2'.

This may be accomplished in what is known as ideal regenerative cycle, the flow diagram of which s shown in Fig. and the corresponding T-s diagram also.




Ideal Regenerative cycle -Basic Scheme




ideal regenerative cycle

The unique feature of the ideal regenerative cycle is that condensate, after leaving the pump circulates around the turbine casing, counter flow direction of vapor flow in the turbine. Thus, it is possible to transfer heat from the vapor as it flows through the turbine to the liquid flowing around the turbine. Let us assume that this is a reversible heat transfer, i.e., at each point the temperature of the vapor is only infinitesimal higher than the temperature of the liquid. The process 1-2' thus represents reversible expansion of steam in turbine with reversible heat rejection.

The net work output of the ideal regenerative cycle is thus less, and hence its steam rate will be more, although it is more efficient, when compared with the Rankine cycle. However, the cycle is not practicable for the following reasons

(a) Reversible heat transfer cannot be obtained in finite time.

(b) Heat exchanger in the turbine is mechanically impracticable.

(c) the moisture content of the steam in the turbine will be high.

**The Regenerative Rankine Cycle (Bleeding cycle)**

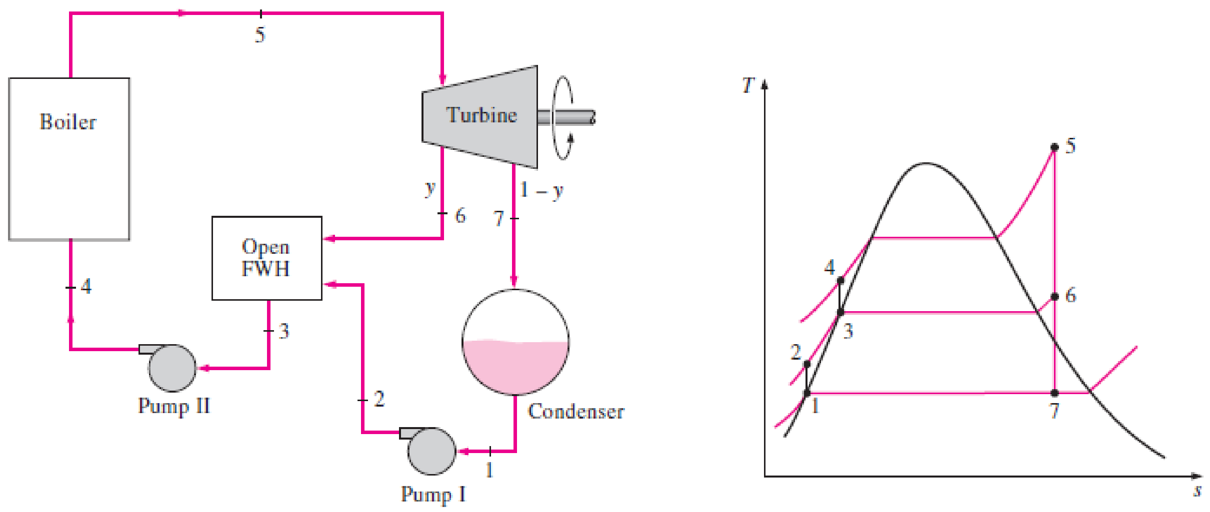
A practical regeneration process in steam power plants is accomplished by extracting, or “bleeding,” steam from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine, is used to heat the feed water instead. The device where the feed water is heated by regeneration is called a **regenerator,** or a **feedwater heater (FWH).**

Regeneration not only improves cycle efficiency, but also provides a convenient means of deaerating the feed water (removing the air that leaks in at the condenser) to prevent corrosion in the boiler. It also helps control the large volume flow rate of the steam at the final stages of the turbine (due to the large specific volumes at low pressures). Therefore, regeneration has been used in all modern steam power plants since its introduction in the early 1920s.

A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feed water either by mixing the two fluid streams (open feedwater heaters) or without mixing them (closed feedwater heaters).

**Open Feedwater Heaters**

An **open** (or **direct-contact**) **feedwater heater** is basically a *mixing chamber,* where the steam extracted from the turbine mixes with the feedwaterexiting the pump. Ideally, the mixture leaves the heater as a saturated liquidat the heater pressure. The schematic of a steam power plant with one openfeedwater heater (also called *single-stage regenerative cycle*) and the *T-s* diagram of the cycle are shown in Fig.




Regenerative Rankine Cycle

- In an ideal regenerative Rankine cycle, steam enters the turbine at the boiler pressure (state 5) and expands isentropically to an intermediate pressure (state 6).
- Some steam is extracted at this state and routed to the feedwater heater, while the remaining steam continues to expand isentropically to the condenser pressure (state 7).
- This steam leaves the condenser as a saturated liquid at the condenser pressure (state 1). The condensed water, which is also called the *feedwater*, then enters an isentropic pump, where it is compressed to the feedwater heater pressure (state 2) and is routed to the feedwater heater, where it mixes with the steam extracted from the turbine. The fraction of the steam extracted is such that the mixture leaves the heater as a saturated liquid at the heater pressure (state 3).
- A second pump raises the pressure of the water to the boiler pressure (state 4). The cycle is completed by heating the water in the boiler to the turbine inlet state (state 5).

In the analysis of steam power plants, it is more convenient to work with quantities expressed per unit mass of the steam flowing through the boiler.

For each 1 kg of steam leaving the boiler, *y* kg expands partially in the turbine and is extracted at state 6. The remaining (1 -*y*) kg expands completely to the condenser pressure.

Therefore, the mass flow rates are different in different components. If the mass flow rate through the boiler is *m* . , for example, it is (1 -*y*)*m* . through the condenser.

The heat and work interactions of a regenerative Rankine cycle with one feedwater heater can be expressed per unit mass of steam flowing through the boiler as follows:

$$q_{in} = h_5 - h_4$$

$$q_{out} = (1 - y)(h_7 - h_1)$$

$$w_{turb.out} = (h_5 - h_6) + (1 - y)(h_6 - h_7)$$

$$w_{pump,\ in} = (1 - y)w_{pump\ I,in} + w_{pump\ II,\ in}$$

where$y = \frac{\dot{m}_6}{\dot{m}_5}$ fraction of steam extracted

$$w_{pump\ I,in} = v_1(P_2 - P_1)$$

$$w_{pump\ II,in} = v_3(P_4 - P_3)$$

- The thermal efficiency of the Rankine cycle increases as a result of regeneration. This is because regeneration raises the average temperature at which heat is transferred to the steam in the boiler by raising the temperature of the water before it enters the boiler.
- The cycle efficiency increases further as the number of feedwater heaters is increased. Many large plants in operation today use as many as eight feedwater heaters. The optimum number of feedwater heaters is determined from economical considerations.
- The use of an additional feedwater heater cannot be justified unless it saves more from the fuel costs than its own cost.

The main advantages of bleeding are :

1. It increases efficiency as heat of bled steam is not lost in the condenser but utilized in feedwater heating which increases the average temperature at which heat is added.
2. Due to bleeding, volume flow rate is reduced and due to this, dimensions of turbine blades canbe reduced. Also, the size of condenser can be reduced.
3. Due to higher temperature of feed water, thermal stresses in the boiler are minimized.

Disadvantages:

1. For given output higher capacity boiler is required.
2. With more heaters, maintenance is more and cost is also more.

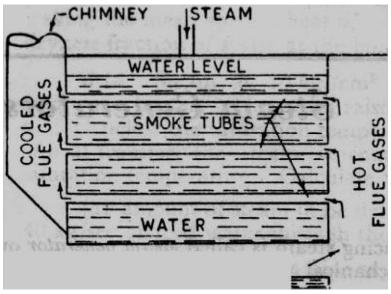
**BOILERS**

### 3.7 Classification or boilers

Boilers are mainly classified according to the following;

1. Relative position of hot gases and water.

   a) *Fire tube boiler:* The hot gases pass through the tubes that are surrounded by water. The products of combustion leaving the furnace are passed through fire (smoke) tubes which are arranged within the water space. The heat energy of the flue gas is transferred to water which is converted into steam. The spent gases are then discharged to atmosphere through chimney. The construction is as shown in fig.1.1.
   *Examples: Locomotive Boilers, Lancashire Boiler, and Cochran Boilers*.




Fire tube boiler

   b) *Water Tube Boilers*: The tubes contain water and hot gases produced by combustion of fuel flows outside. A bank of water tubes (tubes containing water) is connected with steam-water drum through two sets of headers. The hot flue gases from the furnace are made to flow around the water tubes a sufficient number of times. The gases thus give up their heat to an appreciable extent, get cooled and are discharged to the stack, The steam formed separates from water in the drum and gets accumulated in the steam space. Schematic arrangement is shown in the fig. 2. *Examples: Babcock and Wilcox, Stirling Boiler, Yarrow boiler*

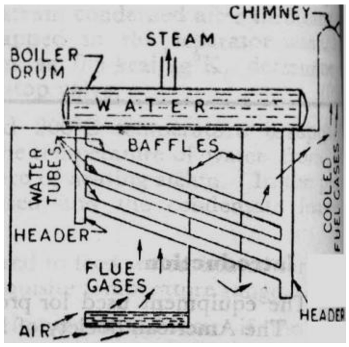



Fig. 2.2 Water tube boiler

2. Method of firing.

   (a) *Internally fired boiler*: The furnace region (space in which combustion of fuel takes place) is provided inside the boiler shell and is completely surrounded by water cooled surfaces. The Examples: Cochran, Lancashire, Locomotive and Scotch boilers.

   (b) *Externally fired boiler*: The furnace region is provided outside or built under the boiler. The externally fired boiler has the advantage that its furnace region is simple to construct and can be easily enlarged.

   Examples: Babcock and Wilcox boiler.

3. Pressure of steam:

   a) Boilers producing steam at a pressure of 80 bar and above are called *high pressure boilers.*

   Examples: Babcock and Wilcox Boiler, Lamont Boiler, Velox Boiler and Benson Boiler etc.

   b) The boilers which produce steam at pressures lower than 80 bar are called *low pressure boilers*. Examples are Cochran, Cornish, Lancashire and Locomotive boilers.

4. Method of circulation of water:

   a) In forced circulation type of boilers, the circulation of water is done by a forced pump.

   Examples: Velox, Lamont boiler, Benson boiler etc.

   b) In natural circulation type of boilers, circulation of water in the boiler takes place due to natural convection currents produced by the application of heat.

   Examples: Lancashire, Babcock and Wilcox boilers etc.,

5. Nature of service to be performed:

   a) Boilers which are used with stationary plants are classified as *land boilers*.

b) Boilers which can be readily dismantled and easily carried from one site to another are called *portable boilers*.

c) Marine and Locomotive boilers belong to another category called *mobile boilers*.

6. Number of fire tubes available

a) If a boiler contains single fire tube, then it is called *single fire tube boiler*.

Example: Cornish and Simple Vertical boiler

b) If a boiler contains more than one fire tube, then it is called *multi-tube boiler*.

7. Nature of draught.

a) When the fuel burns in the furnace of the boiler with the natural circulation of air, the draft is named as natural draught.

b) In artificial draught, the air is forced by means of a forced fan.

8. Heat source:

The heat energy utilized for the conversion of fluid into a vapour may be derived from:

- combustion of solid, liquid or gaseous fuel
- electrical and nuclear energy
- hot waste gases of other chemical reactions.

9. Fluid used:

The boilers are classified as steam boiler water as fluid, mercury boilers using mercury as fluid, and the which are used for heating special chemicals.

10. Material of construction of boiler shell:

Depending upon material used for the construction of boiler shell, we classify the boiler into cast iron boilers and steel boilers.

a) Power boilers are usually fabricated from steel plates.

b) Low pressure heating boilers are built either of cast iron or steel.

**3.8 Comparison between Fire tube and Water tube boilers**

| S. No | Particulars | Fire tube boiler | Water tube boiler |
|---|---|---|---|
| 1 | Position of water and hot gases | Hot gases flow inside the tubes and water outside the tubes. | Water flows inside the tubes and hot gases outside the tubes. |
| 2 | Mode of firing | Generally internally fired | Externally fired |
| 3 | Operating pressure | Limited to 16 bar | Can work under as high pressure as 100 bar. |
| 4 | Rate of steam production | Lower | Higher |
| 5 | Suitability | Not suitable for large power plants | Suitable for large power plants |
| 6 | Risk on bursting | Involves lesser risk on explosion due to lower pressure | Involves more risk on explosion due to higher pressure |
| 7 | Floor area | For a given power it occupies more floor area | For a given power it occupies less floor area |
| 8 | Construction | Difficult | Simple |
| 9 | Shell diameter | Large for same power | Small for same power |
| 10 | Treatment of water | Not so necessary | More necessary |
| 11 | Accessibility of various parts | Not so easily accessible for cleaning, repair and inspection | More accessible |
| 12 | Requirement of skill | Require less skill for efficient | Require more skill and |

| | | | |
|---|---|---|---|
| | | and economic working | careful attention |

## 3.9 Fire Tube Boilers

These boilers essentially consist of or more fire tubes passing through a cylindrical shell which may be horizontal or vertical and is filled to at least half of its volume with water,

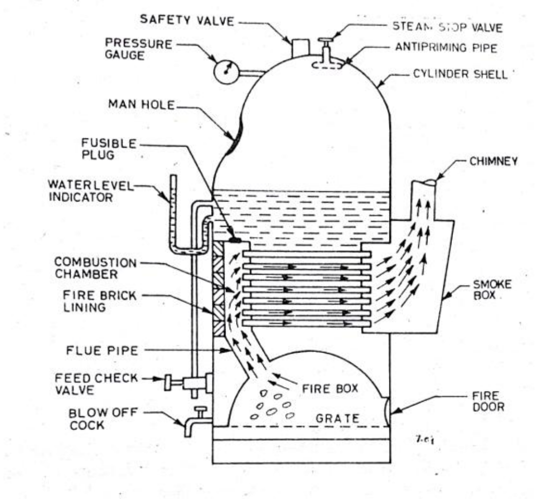
**a. Cochran boiler**

It is one of the best types of vertical multi-tubular boiler and has a number of horizontal fire tubes.

*Construction*: It consists of a cylindrical shell with a dome shaped top where the space is provided for steam. The furnace is one piece and seamless. Its crown has a hemispherical shape and thus provides maximum volume of space. A large number of fire tubes which are usually 75 – 100 mm in diameter are supported between two end covers. A number of hand-holes are provided around the outer shell for cleaning purposes.

*Specifications*:

| | | |
|---|---|---|
| Shell diameter | : | 2.75 m |
| Height | : | 5.79 m |
| Working pressure | : | 6.5 bar (max. 15 bar) |
| Steam capacity | : | 3500 kg/hr (max. 4000 kg/hr) |
| Heating surface | : | 120 $m^2$ |
| Efficiency | : | 70 – 75% |



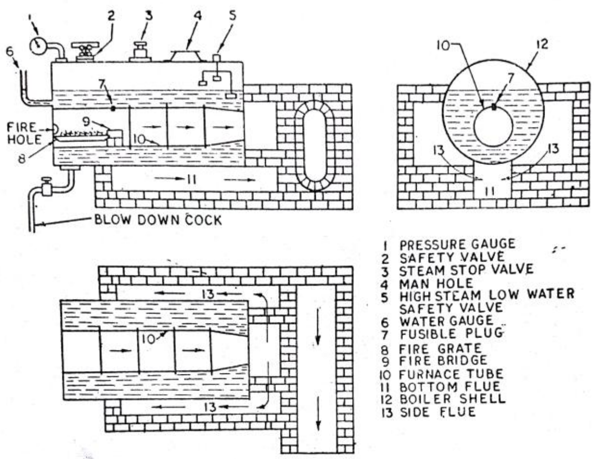

Cochran Boiler

*Working*: The fuel is burnt on the grate and ash is collected and disposed of from ash pit. The gases of combustion produced by burning of fuel enter the combustion chamber through the fire tube and strike against fire brick lining which directs them to pass through number of horizontal tubes, being surrounded by water. After heat transfer to the water, the gases escape to the atmosphere through smoke box and chimney.

b. **Cornish Boiler**:

*Specification:*

| | | |
|---|---|---|
| No. of flue tubes | : | One |
| Diameter of the shell | : | 1.25 – 1.75 m |
| Length of the shell | : | 4 – 7 m |
| Pressure of the steam | : | 10.5 bar |
| Steam capacity | : | 6500 kg/hr |




Cornish boiler and its brickwork setting.

***Construction & working***: It consists of a cylindrical shell with flat ends through which passes a smaller flue tube containing the furnace. The products of combustion pass from the fire grate forward over the brickwork bridge to the end of the furnace tube; they then return by the two side flues to the front end of the boiler, and again pass to the back end of a flue along the bottom of the boiler to the chimney.

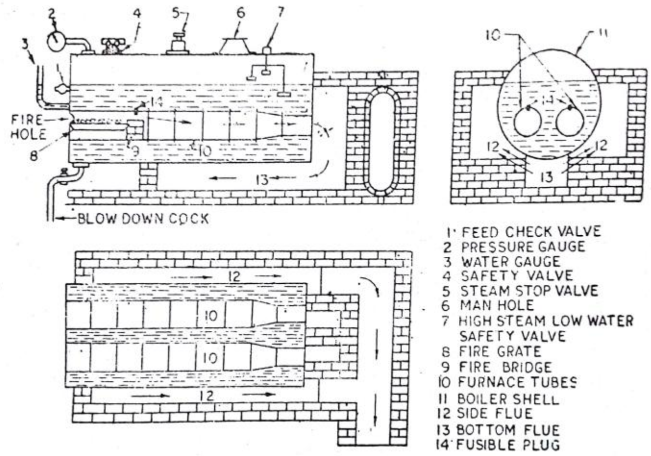
*Advantage*:

The sediment contained in the water falls to the bottom, where the plates are not brought into contact with the hottest portion of the furnace gases.

c. **Lancashire Boiler**:

*Specification:*

| | | |
|---|---|---|
| Shell diameter | : | 2 – 3 m |
| Length of the shell | : | 7 – 9 m |
| Max. Working pressure | : | 16 bar |
| Steam capacity | : | 9000 |
| kg/hr | | |
| Heating surface | : | 120 $m^2$ |
| Efficiency | : | 50 – 70% |



Lancashire boiler and its brick-work setting.

*Construction*: It consists if a cylindrical shell inside which two large tubes are placed. The shell is constructed with several rings of cylindrical form and it is placed horizontally over a brick work which forms several channels for the flow of hot gases. These two tubes are also constructed with several rings of cylindrical form. They pass from one end to the other and are covered with water. The furnace is placed at the front end of each tube and they are known as furnace tubes. The coal is introduced through the fire hole into the grate.

*Working*: The combustion products from the grate pass upto the back end of the furnace tubes and then in the downward direction. Thereafter they move through the bottom channel or bottom flue upto the
front end of the boiler where they are divided and pass upto the side flues. Now they move along side flues and come to the chimney flue from where they are released into the atmosphere through chimney. To control the flow of hot gases to the chimney, dampers are provided. This results in the control of flow of air to the grate.

*Advantages*:

- It is reliable.
- Simplicity of design
- Ease of operation
- Less operating and maintenance costs

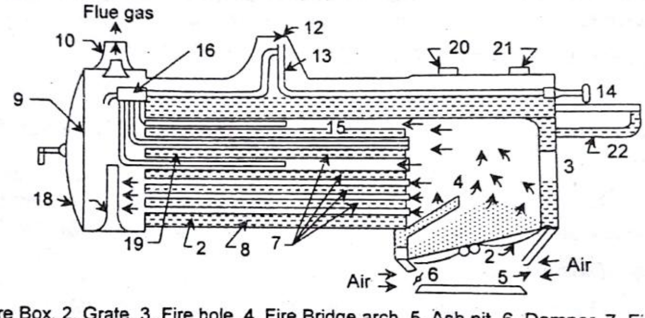
*Applications*:

- It is used where large reserve of water and steam are needed
- Sugar mills and textile mills where along with the power steam and steam for the process work is also needed.

d. **Locomotive Boiler:**

It is compact and its capacity for steam production is high for its size as raise large quantity of steam rapidly.

*Construction*: It consists of a cylindrical barrel with a rectangular fire box at one end and a smoke box at the other end. The coal is introduced through the fire hole into the grate which is placed at the bottom of the fire box. The hot gases which are generated due to burning of the coal are



1. Fire Box, 2. Grate, 3. Fire hole, 4. Fire Bridge arch, 5. Ash pit, 6. Damper, 7. Fire tubes, 8. Barrel or shell, 9. Smoke Box, 10. Chimney (short), 11, Exhaust steam pipe, 12. Steam Dome, 13. Regulator, 14. Lever, 15. Superheater tubes, 16. Superheater header, 17. Superheater exit pipe, 18. Smoke box door, 19. Feed check valve, 20 Safety valve, 21. Whistle, 22. Water gauge,

Locomotive Boiler.

deflected by an arch of fire bricks, so that the walls of the fire box may be heated properly. The fire box is entirely surrounded by water except for the fire hole and the ash pit which is below the fire box which is fitted with dampers at its front and back ends. The dampers control the flow of air to the grate.

*Working*: The hot gases pass from the fire box to the smoke box through a series of fire tubes and then they are discharged into the atmosphere through the chimney. The fire tubes are placed inside the barrel. The superheater tubes are placed inside the fire tubes in larger diameter. The heat of the hot gases is transmitted into the water through the heating surface of the fire tubes. The steam generated is collected over the water surface.

A dome shaped chamber known as steam dome is fitted on the upper part of the barrel, from where the stream flows through a steam pipe into the chamber. The flow of steam is regulated by means of regulator. From the chamber, it passes through the superheater tubes and returns to the superheated steam chamber.

*Merits*:

- High steam capacity
- Low cost of construction
- Portability
- Low installation cost.
- Compact.

*Demerits*:

- There are chances to corrosion and scale formation in the water legs due to the accumulation of sediments and the mud particles.
- It is difficult to clean some water spaces.
- Large flat surfaces need bracing.
- It cannot carry high overloads without being damaged by overheating.

## 3.10 Water Tube Boilers:

Water tube boilers are classified as:

a. Horizontal straight tube Ex. Babcock and Wilcox Boiler
   i. Longitudinal drum
   ii. Cross drum
b. Bent tube
   i. Two drum
   ii. Four drum
   iii. Low head three drum

a. **Babcock and Wilcox Boiler**:

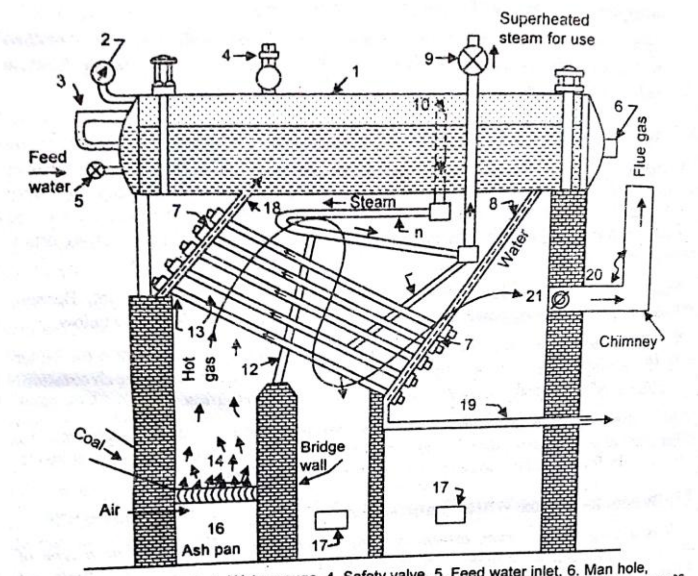
It is horizontal straight water tube boiler and may be designed for stationary or marine purposes.

These are exclusively used when pressure above 10 bar and capacity in excess of 7000 kg of steam per hour is required.

*Specifications:*

| | | |
|---|---|---|
| Diameter of the drum | : | 1.22 – 1.83 m |
| Length of the drum | : | 6.096 – 9.144 m |
| Size of water tubes | : | 7.62 – 10.16 cm |
| Size of superheater tubes | : | 3.84 – 5.71 cm |
| Working pressure | : | 40 bar (max) |
| Steaming capacity | : | 40,000 kg/hr |
| Efficiency | : | 60 – 80% |

***Construction*:** It consists of a high pressure drum mounted at the top. For each end of the drum, the connections are made with the upper header and downtake header. A large number of water tubes connect the uptake and downtake headers. The water tubes are inclined at $5^0 - 15^0$ to promote water circulation. The headers have a serpentine form (sinusoidal). This arranges the water tubes such that they are staggered and this exposes the complete heating surface to the hot flue gases.



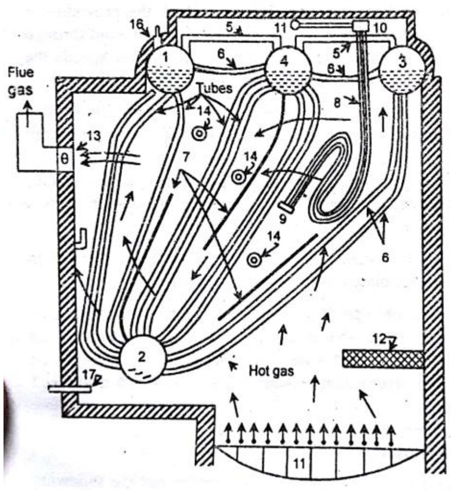

1. Drum, 2. Pressure gauge, 3. Water gauge, 4. Safety valve, 5. Feed water inlet, 6. Man hole, 7. Heaters, 8. Down comer, 9. None- return valve, 10. Anti priming pipe, 11. Superheater, 12. Baffles, 13. Water tubes, 14. Fire grate, 15. Fire door, 16. Ash pan, 17. Clean out doors, 18. Riser, 19. Blow-off pipe, 20. Chimney, 21. Damper.

. Babcock Wilcox Boiler.

*Working*: Below the uptake header, the furnace of the boiler is arranged. The coal is fed to the chain grate stoker through the fire door. The chain speed is so adjusted that by the time, the coal reaches the other end of the grate, its combustion has been completed. The hot gases are forced to move upwards between the tubes by baffle plates provided. The water from the drum flows through the inclined tubes via downtake header and goes back into the shell in the form of water and steam via uptake header. The steam gets collected in the steam space of the drum. The steam then enters through the antipriming pipe and flows in the superheater tubes where it is further heated and is finally taken out through the main stop valve. The superheaters remain flooded until the steam reaches the working pressure. The superheater is then drained and steam is allowed to enter in it for superheating purposes.

b. **Stirling Boiler**: it is an example of bent tube boiler. The main elements of bent tube boiler are essentially drum or drums and headers connected by bent tubes. These are used for large central power stations. They have steaming capacities of as high as 50,000 kg/hr ad pressure as high as 60 bar.

*Construction*: It consists of two upper drums known as steam drums and a lower drum known as mud drum or water drum. The steam drum is connected to mud drum by the bank of bent tubes. For cleaning operation, a man hole at one end of each drum is provided. The feed water from the economizer is delivered to the steam drum-1 which is fitted with a baffle. The baffle deflects the water to move downwards into the drum.




1, 2, 3. Steam drums,
4. Mud drum,
5. Equalizing tubes,
6. Water header,
7. Baffles,
8. Superheat tubes,
9. Inlet super heater,
10. Outlet superheater tubes,
11. Grate,
12. Fire brick arch,
13. Fire door,
14. Cleaning doors,
15. Steam stop valve,
16. Safety valve,
17. Blow-off cock.

Stirling Bent Tube Boiler.

The water flows from the drum-1 to the mud drum through the rearmost water tubes at the back side. The baffle provided at the mud drum deflects the pure water to move upward to the drum-1 through the remaining half of water tubes at the back. The water also flows from it to the drum-2 through the water tubes which are just over the furnace. So they attain a higher temperature than the remaining portion of the boiler and a major portion of evaporation takes place in these tubes.

The steam is taken from the drum-1 through a steam pipe and then it passes through the super heater tubes where the steam is superheated. Finally the steam moves to the stop valve from where the steam can be supplied for further use.

### 3.11 High pressure Boilers

The modern high pressure boilers employed for power generation are for steam capacities 30 to 650 tonnes/hr and above with a pressure upto 160 bar and maximum steam temperature of about 540$^0$C.

Water tube boilers are generally preferred for high pressure and high output whereas shell boilers for low pressure and lower output. The salient characteristic of high pressures boilers are :

i. *Method of circulation of water*: The circulation of water through the boiler may be natural circulation due to density difference or forced circulation. In all modern high pressure boiler plants, the water circulation is maintained with the help of pump which forces the water through the boiler plant. The use of natural circulation is limited to use of sub-critical boilers.

ii. *Type of tubing*: In most of the high pressure boilers, the water circulated through the tubes and their external surfaces are exposed to the flue gases. In water tube boilers, if the flow takes place through one continuous tube, the large pressure drop takes place due to friction. This is considerably reduced by arranging the flow to pass through parallel system of tubing.

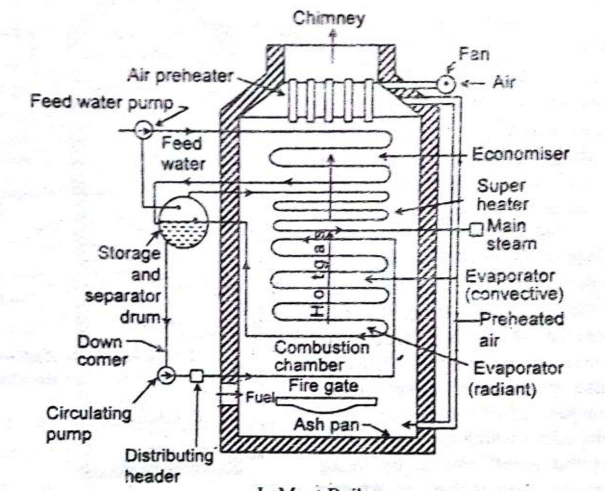
iii. *Improved method of heating*: The following improved arrangement of heat devices may by incorporated:
   a. heating water by mixing superheated steam. This mixing phenomenon gives higher heat transfer coefficient
   b. saving of heat by evaporation of water above critical pressure of the steam.
   c. Overall heat transfer coefficient can be increased by increasing the water velocity inside the tube and increasing the gas velocity above sonic velocity.

### 3.12 Lamont Boiler:

This works on a forced circulation and the circulation is maintained by a centrifugal pump, which is driven by a steam turbine using steam from the boiler.

*Working*: The feed water passes through the economizer to the drum from which it is drawn to the circulating pump. The circulating pump delivers the feed water to the tube evaporating section which in turn sends a mixture of steam and water to the drum. The steam in the drum is then drawn through the superheater. The superheated steam obtained is supplied to the steam turbine for generating work output.

These boilers have been built to generate 45 to 80 tonnes of superheated steam at a pressure of 130 bar and at a temperature of 500$^0$C.

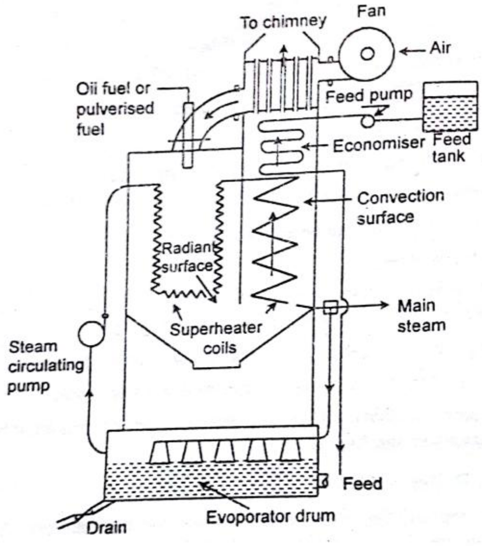



LaMont Boiler

## 2.13 Loefler Boiler:

***Principle***: Evaporating the feed water by means of superheated steam from the superheater, the hot gases from the furnace being primarily used for superheating purposes. It makes use of pump for the circulation of feed water.

***Working***: The high pressure feed pump draws the water through economizer or feed water heater and deliver it into the evaporating drum. The steam circulating pump draws saturated steam from the evaporating drum and passes it through radiant and convective superheaters where steam is heated to required temperatures. From the superheater about one-third of the superheated steam passes to the turbine and the remaining two-thirds passing through the water in the evaporating drum in order to evaporate feed water.



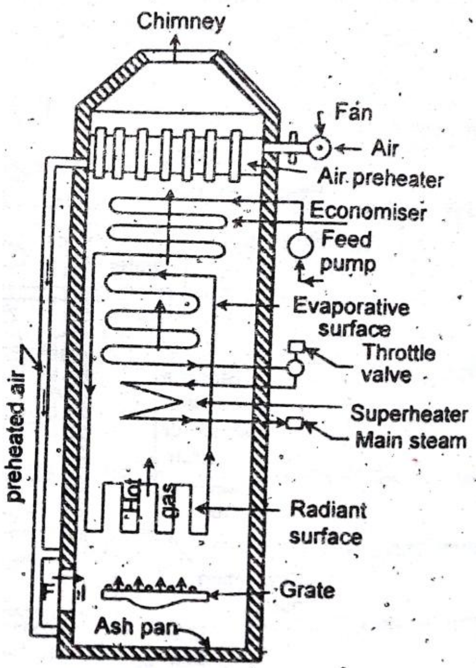
**Loeffler Boiler**

This boiler can carry higher salt concentration than any other type and is more compact than indirectly heated boilers having natural circulation.

## 2.14 Benson Boiler

*Principle*: It eliminates the latent heat of water by first compressing the feed water to a pressure of 235 bar, it is then above the critical pressure and its latent heat is thus zero.

**Working**: This boiler does not use any drum. The feed water after circulating through the economizer tubes flows through the radiant parallel tube section to evaporate partly. The steam water mixture produced thus moves to the transit section where this mixture is converted to steam. The steam is now passed through the convection superheater and finally supplied to the turbine. Boilers of 150 tonnes/hr steam capacity, with maximum pressure of 500 atm and temperature of $650^0$C are in use.



Benson Boiler

**3.15 Boiler Mountings and Accessories:** For efficient operation and maintenance of safety, the boiler equipped with two categories of components and elements.

First categories include the fittings which are primarily indicated for the safety of the boiler and for complete control the process of steam generation. These units are called *mountings*. The mounting from an integral part of the boiler and are mounted on the body of the boiler itself. The following mountings are usually installed on the boiler.

1. Two safety valve
2. Two water level indicators
3. Pressure gauge
4. Fusible plug
5. Steam stop valve
6. Feed check valve

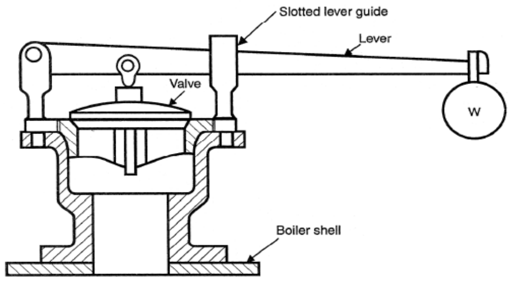
7. Blow-of cock
8. Man and mud hole

Second categories include the components which are installed to increase the efficiency of the steam power plants and help in the power working of the boiler unit. These fitting are called *boiler accessories*. The following accessories are given below.

1. Air pre-heater
2. Economiser
3. Super heater
4. Feed pump and
5. Injector

**Boiler Mountings:**

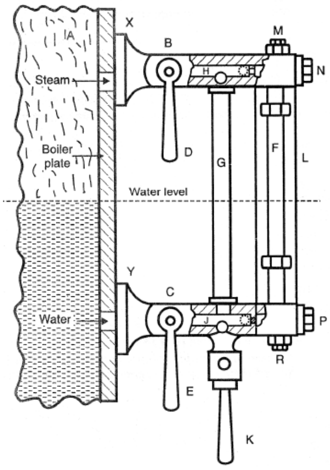
1. **Safety valve**: Safety valves are located on the top of the boiler. They guard the boiler against the excessive high pressure of steam inside the drum. If the pressure of steam in the boiler drum exceeds the working pressure then the safety valve allows blow-off the excess quantity of steam to atmosphere. Thus the pressure of steam in the drum falls. The escape of steam makes a audio noise to warn the boiler attendant. There are four types of safety valve. 1. Dead weight safety valve. 2. Spring loaded safety valve 3. Lever loaded safety valve 4. High steam and low water safety valve.



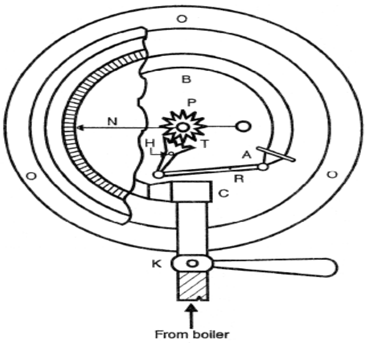
Lever Safety Valve

2. **Water level indicators:** Water level indicator is located in front of boiler in such a position that the level of water can easily be seen by attendant. Two water level indicators are used on all boilers.

| | |
|---|---|
| $A$ = End plate of boiler | $H$ and $J$ = Two balls |
| $B$ and $C$ = Hollow gun metal castings | $K$ = Drain cock |
| $D$ and $E$ = Cocks | $L$ = Guard glass |
| $F$ = Gauge glass | $M, N, P, R$ = Screwed caps |
| $G$ = Hollow metal column | $X, Y$ = Flanges |



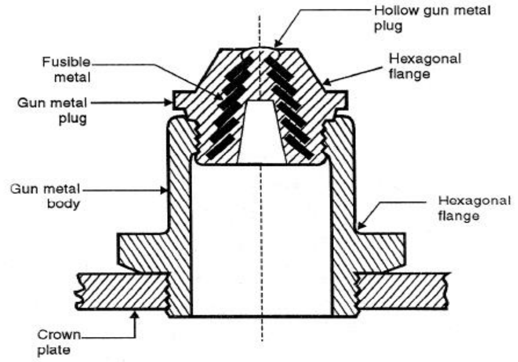
Water level indicator



Pressure Gauge

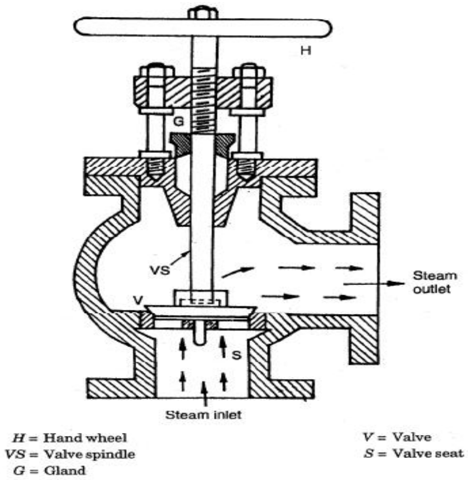
3. **Pressure gauge:** A pressure gauge is fitted in front of boiler in such a position that the operator can conveniently read it. It reads the pressure of steam in the boiler and is connected to steam space by a siphon tube. The most commonly, the Bourdon pressure gauge is used.

4. **Fusible plug:** It is very important safety device, which protects the fire tube boiler against overheating. It is located just above the furnace in the boiler. It consists of gun metal plug fixed in a gun metal body with fusible molten metal. During the normal boiler operation, the fusible plug is covered by water and its temperature does not rise to its melting state. But when the water level falls too low in the boiler, it uncovers the fusible plug. The furnace gases heat up the plug and fusible metal of plug melts, the inner plug falls down The water and steam then rush through the hole and extinguish the fire before any major damage occurs to the boiler due to overheating.



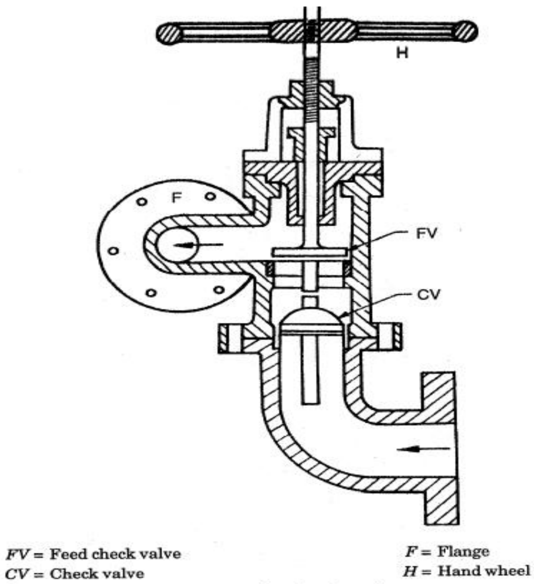
Fusible Plug

5. **Steam stop valve:** The steam stop valve is
located on the highest part of the steam space. It regulates the steam supply to use. The steam stop valve can be operated manually or automatically.



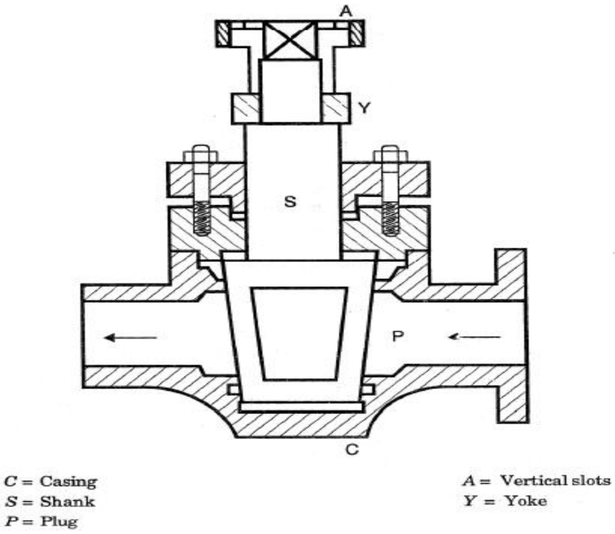
Steam Stop Valve

6. **Feed check valve:** The feed check valve is fitted to the boiler, slightly below the working level in the boiler. It is used to supply high pressure feed water to boiler. It also prevents the returning of feed water from the boiler if feed pump fails to work.



Feed Check Valve

6. **Blow-of cock:** The function of blow-off cock is to discharge mud and other sediments deposited in the bottom most part of the water space in the boiler, while boiler is in operation. It can also be used to drain-off boiler water. Hence it is mounted
at the lowest part of the boiler. When it is open, water under the pressure rushes out, thus carrying sediments and mud.

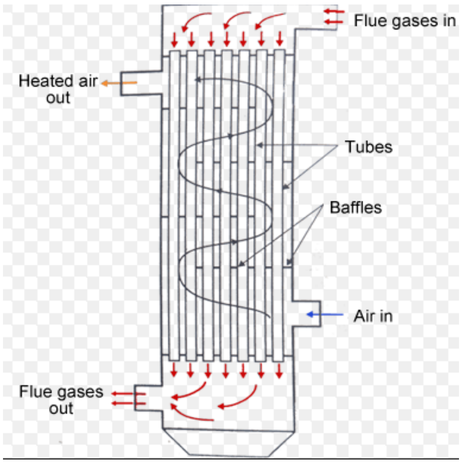



Blow off Cock

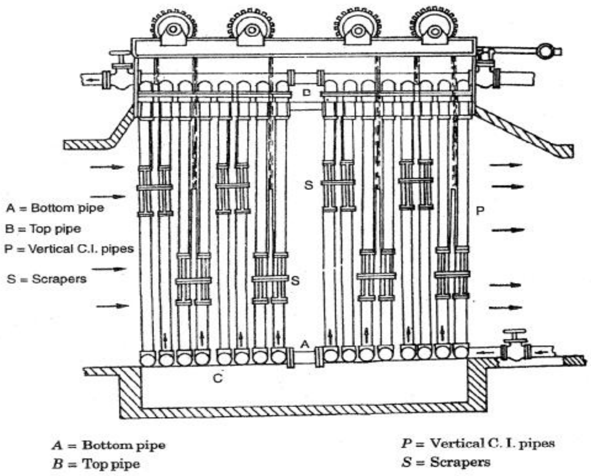
7. **Man and mud hole:** These are door to allow men to enter inside the boiler for the inspection and repair

**Boiler Accessories:**

1. **Air pre-heater:** The function of an air pre-heater is similar to that of an economizer. It recovers some portion of the waste heat of hot flue gases going to chimney, and transfers same to the fresh air before it enters the combustion chamber. Due to preheating of air, the furnace temperature increases. It results in rapid combustion of fuel with less soot, smoke and ash. The high furnace temperature can permit low grade fuel with less atmospheric pollution. The air pre-heater is placed between economizer and chimney.

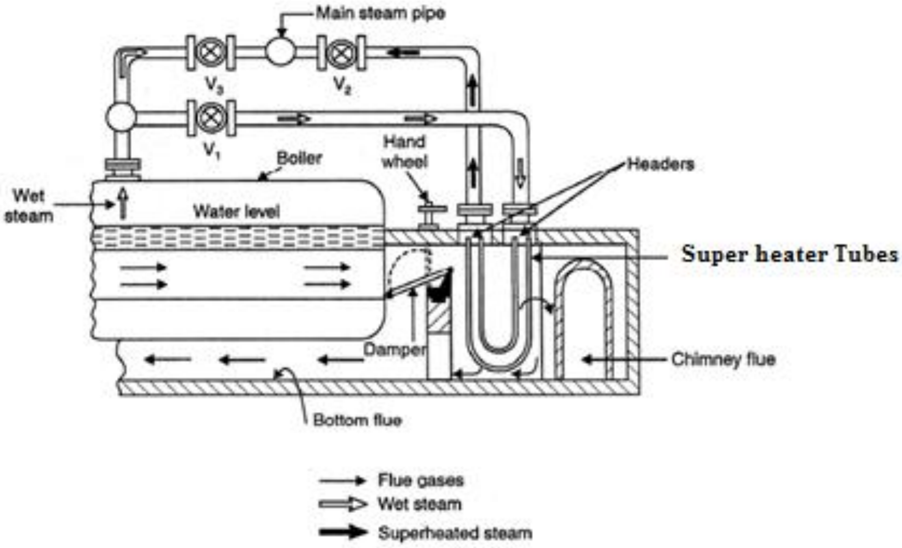



2. **Economizer:** An economizer is a heat exchanger, used for heating the feed water before it enters the boiler. The economizer recovers some of waste heat of hot flue gases going to chimney. It helps in improving the boiler efficiency. It is placed in the path of flue gases at the rear end of the boiler just before air pre-heater.




Economizer

3. **Super heater:** It is a heat exchanger in which heat of combustion products is used to dry the wet steam, pressure remains constant, its volume and temperature increase. Basically, a super heater consists of a set of small diameter U tubes in which steam flows and takes up the heat from hot flue gases.




Super Heater

## `FLOW THROUGH NOZZLES

**Objective:**

- To design steam nozzle for the given pressure drop and understand the concept of supersaturated flow in steam nozzles.

**Syllabus:**

Steam nozzles, function of nozzle, Nozzle Applications and Types, Flow through nozzles, thermodynamic analysis, Calculation of Velocity of nozzle at exit, Ideal and actual expansion in nozzle, velocity coefficient, Condition for maximum discharge, critical pressure ratio, Criteria to decide nozzle shape, Super saturated flow. its effects, Degree of super saturation and degree of Under cooling, Wilson line

**Outcomes:**

Students will be able to

- Perform the thermodynamic analysis of a steam nozzle
- Narrate the effect of friction on nozzle efficiency
- Explain the concept of supersaturated flow in nozzles

**Introduction:**

- It is a device for increasing the velocity of a steadily flowing fluid with the expense of the pressure of the flowing fluid.
- The fluid enters the nozzle with high pressure and relatively low velocity.
- As it flows through the nozzle it expands to a lower pressure and in the process, enthalpy of the fluid decreases and pressure drops; Simultaneously the flow of fluid is accelerated from the entrance to the exit of the nozzle.

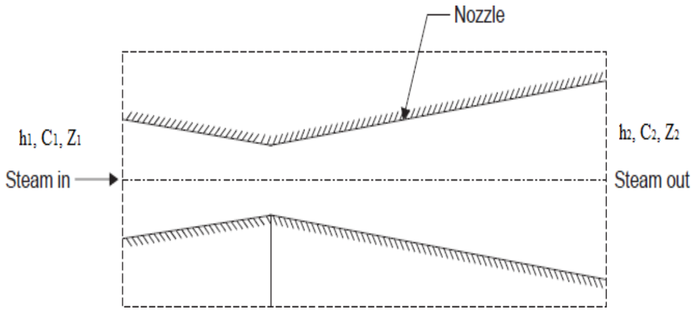



Fig. 3.1 Steam Nozzle

- Fig. 3.1 represents the steam nozzle with inlet and outlet of steam. The terms $h_1$, $C_1$, $Z_1$ and $h_2$, C2, $Z_2$ represents the enthalpy and velocity of steam and elevations at the inlet and outlet respectively.
- Applying the steady flow energy equation to the nozzle,

$$h_1 + \frac{C_1^2}{2} + Z_1 + \frac{dQ}{dm} = h_2 + \frac{C_2^2}{2} + Z_2 + \frac{dW}{dm} \qquad \text{Eq. (3.1)}$$

Assuming that (i) the change in potential energy is negligible, (ii) the nozzle is perfectly insulated (i.e. adiabatic conditions are maintained), (iii) inlet velocity is very less than velocity at the exit of the nozzle, and (iv) no work transfer, the equation 3.1 modifies to

$$C_2 = \sqrt{2(h_1 - h_2)} \qquad \text{Eq. (3.2)}$$

where the enthalpy is in Joules. This equation tells that the drop in enthalpy is contributing for the increase of kinetic energy of the steam at the exit of the nozzle.

➢ **Functions of Nozzle:**

- It transforms a portion of energy of steam (obtained from steam generating unit) into kinetic energy.
- In impulse turbine, it directs the steam jet of high velocity against the blades, which are free to move in order to convert kinetic energy into shaft work. In reaction turbines the nozzles which are free to move, discharge high velocity steam. The reactive force against the nozzle produce the motion and work is obtained.

**Applications of nozzles:**

1. In turbomachines (steam turbines), the high pressure stream of fluid is converted to high velocity stream before passing over the curved blades to produce mechanical work.
2. In rocket motors and jet propulsion, the thrust produced by the jet provides the propulsive effort.
3. In flow measurement, the differential pressure drop is correlated to the velocity to find the discharge.
4. Injectors for pumping feed water into the boiler.
5. The ejectors for removing the air from the condensers.
6. Artificial fountains.

**Types of Nozzles:**

There are three types of nozzles:

i. Converging nozzle

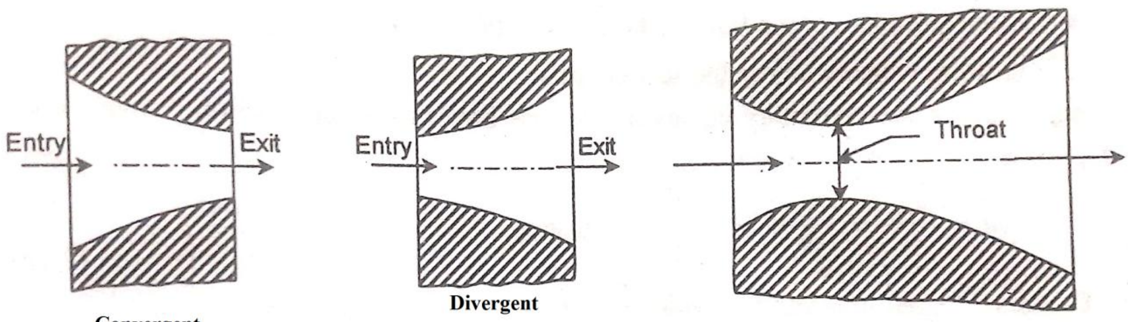
ii. Diverging nozzle

iii. Converging-diverging nozzle

**Converging nozzle:** The cross-section of the nozzle decreases continuously from the entrance to the exit.

**Diverging nozzle:** The cross-section of the nozzle decreases continuously from the entrance to the exit.

**Converging-Diverging nozzle:** The cross-section of the nozzle first decreases and then increases.



Convergent Divergent convergent - divergent

**Flow through Nozzles:**

**Assumptions:**

1. Thermodynamic and mechanical properties change only in the direction of flow.
2. The process of out flow is stable, i.e. the velocity of the fluid anywhere along the cross-section of the nozzle does not change in the course of time.
3. There are no abrupt changes in the cross-section in the nozzle.
4. The fluid flowing through the nozzle in an ideal and the flow is isentropic in behavior.

➢ **Continuity Equation**

Let m – mass flow rate of the fluid

ρ – density of the fluid

C – velocity of flow

A – Cross section area of the nozzle at any section

Using continuity equation, we have $m = \rho AC$

Since mass flow rate (m) is taken as constant, $\rho AC = \text{constant}$

$or \quad \log \rho + \log A + \log C = \log \text{constant}$

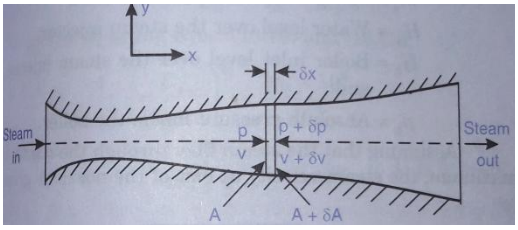
$$or \quad \frac{d\rho}{\rho}+\frac{dA}{A}+\frac{dC}{C}=0$$

For incompressible flow, there is no change in density. Therefore, $\frac{dA}{A}=-\frac{dC}{C}$ (3.3)

- ✓ ***This equation (3.3) tells that a small fractional decrease in area implies a corresponding equal fractional increase in velocity and vice versa. Thus a fluid would accelerate in contracting nozzle and decelerate in an expanding nozzle.***

- **General relationship between area, velocity and pressure in nozzle flow**

Figure shows a nozzle in which a steady and isentropic flow is taking place. Let us consider two transverse plane sections at a distance δx apart. Assuming that the nozzle is running full and the velocity is uniform across cross-section.




$$m=\frac{A*C}{V}=\frac{(A+\delta A)(C+\delta C)}{(V+\delta V)}$$

$$\frac{(V+\delta V)}{V}=\frac{(A+\delta A)(C+\delta C)}{(A*C)}\Rightarrow 1+\frac{\delta V}{V}=\left(\frac{A+\delta A}{A}\right)\left(\frac{C+\delta C}{C}\right)$$

$$or, \quad 1+\frac{\delta V}{V}=\left(1+\frac{\delta A}{A}\right)\left(1+\frac{\delta C}{C}\right)$$

$$or, \quad \frac{\delta A}{A}+\frac{\delta C}{C}-\frac{\delta V}{V}=0$$

and in limits, $\frac{dA}{A}+\frac{dC}{C}-\frac{dV}{V}=0$ (3.4)

Since the flow is isentropic, $PV^{\gamma}=\text{constant}$

$\therefore \quad \log P+\gamma \log V=\log \text{constant}$

On differentiating and dividing the equation by PV, we get

$$\therefore \quad \frac{dP}{P}+\gamma\frac{dV}{V}=0\Rightarrow \quad \frac{dV}{V}=-\frac{1}{\gamma}\frac{dP}{P} \qquad (3.5)$$

From steady flow energy equation, when applied to a nozzle, by considering the isentropic process, and neglecting the changes in potential energy, no work transfer, between two sections one may get

Change in enthalpy in a nozzle = change in kinetic energy

$$h_1 - h_2 = \frac{C_2^2 - C_1^2}{2} \Rightarrow \int_1^2 CdC = -\int_1^2 dh \Rightarrow CdC = -dh \tag{3.6}$$

Also since it is an open system, in which expansion is taking place,

$$dh = VdP \tag{3.7}$$

From Eq. (3.6) and (3.7) modifies to

$$CdC = -VdP \Rightarrow \frac{CdC}{C^2} = \frac{-VdP}{C^2} \Rightarrow \frac{dC}{C} = \frac{-VdP}{C^2} \tag{3.8}$$

Substituting the expression from eq. (3.5) and (3.8) in eq. (3.4), we get

$$\frac{dA}{A} - \frac{VdP}{C^2} - \left(-\frac{1}{\gamma}\frac{dP}{P}\right) = 0$$

$$\frac{dA}{A} = \frac{VdP}{C^2} - \left(\frac{1}{\gamma}\frac{dP}{P}\right) = \frac{1}{\gamma}\frac{dP}{P}\left(\frac{VdP}{C^2}\frac{P\gamma}{dP} - 1\right)$$

$$\frac{dA}{A} = \frac{1}{\gamma}\frac{dP}{P}\left(\frac{VP\gamma}{C^2} - 1\right) \tag{3.9}$$

Sonic velocity,

$$C_s^2 = \gamma RT = \gamma PV$$

$$\frac{dA}{A} = \frac{1}{\gamma}\frac{dP}{P}\left(\frac{C_s^2}{C^2} - 1\right)$$

***The ratio of velocity C to local sonic velocity '$C_s$' is known the Mach number (M).***

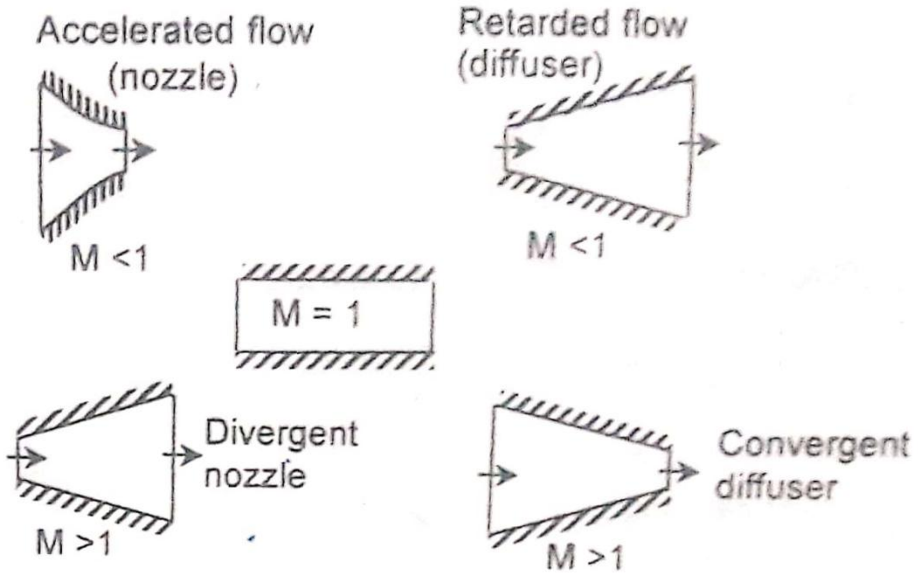
$$\frac{dA}{A} = \frac{1}{\gamma}\frac{dP}{P}\left(\frac{1 - M^2}{M^2}\right) \tag{3.10}$$

**Case: 1**: In an accelerated flow, dP/P is negative i.e. the pressure decreases along the flow direction.

- If C < Cs, M<1. Then dA/A must be negative. This corresponds to the convergent part of the nozzle.

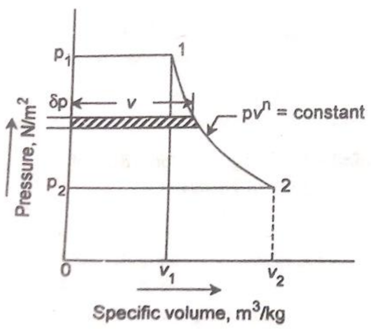
  i. As C approaches the value of Cs (sonic velocity), (i.e. Mach number=1), then dA/A=0 and the throat of the nozzle is reached.

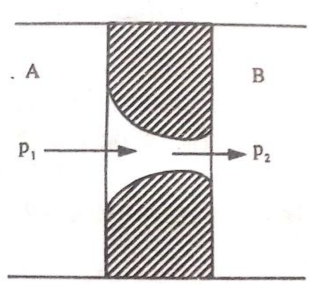
ii. If C > Cs, M>1. Then dA/A must be positive. This corresponds to the divergent part of the nozzle.




**Mass of Discharge through the nozzle**

Consider an adiabatic and frictionless flow of steam through a nozzle a shown in fig.




Let $P_1$ = inlet pressure of steam

$v_1$ = specific volume of steam at $P_1$

$C_1$ = inlet velocity of steam

$P_2$ = inlet pressure of steam

$v_2$ = specific volume of steam at $P_2$

$C_2$ = inlet velocity of steam

$A_2$ = cross-sectional area of nozzle at any section (say throat )

From continuity equation, mass flow rate of steam in kg/sec through the cross-sectional area $A_2$ and pressure $P_2$ is given as

$$m = \frac{A_2.C_2}{v_2}$$

From steady flow energy equation, when applied to a nozzle, by considering the isentropic process, and neglecting the changes in potential energy, no work transfer, between two sections one may get

Change in enthalpy in a nozzle = change in kinetic energy

$$h_1 - h_2 = \frac{C_2^2 - C_1^2}{2} \tag{3.11}$$

Also since it is an open system, in which expansion is taking place,

$$dh = VdP \tag{3.12}$$

From Eq. (3.11) and (3.12), we get

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

$$dh = \int_1^2 vdP$$

$$\frac{C_2^2 - C_1^2}{2} = -dh = \int_2^1 vdP \tag{3.13}$$

Adiabatic and frictionless flow of steam through the nozzle may approximately be represented by

$$Pv^n = c \Rightarrow v = \left(\frac{c}{P}\right)^{1/n}$$

Substituting for ‘v’ in eq. (3.13) and on simplification, we get

$$\frac{C_2^2 - C_1^2}{2} = \frac{n}{n-1}\left(P_1 v_1 - P_2 v_2\right)$$

If the expansion of the steam is from rest i.e. $C_1$=0 and then the above equation modifies to

$$C_2^2 = \frac{2n}{n-1}\left(P_1 v_1 - P_2 v_2\right) \Rightarrow C_2^2 = \frac{2n}{n-1} P_1 v_1 \left(1 - \frac{P_2 v_2}{P_1 v_1}\right)$$

Also

$$P_1 v_1^n = P_2 v_2^n \Rightarrow \frac{v_2}{v_1} = \left(\frac{P_2}{P_1}\right)^{1/n}$$

On simplification, Finally velocity at the outlet is given as

$$C_2 = \sqrt{\frac{2n}{n-1} P_1 v_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{n-1/n}\right)} \tag{3.14}$$

Substituting the values of $C_2$ and $v_2$ in continuity equation gives the mass flow rate as

$$m=\frac{A_2}{v_1\left(\frac{P_2}{P_1}\right)^{1/n}}\sqrt{\frac{2n}{n-1}P_1v_1\left(1-\left(\frac{P_2}{P_1}\right)^{n-1/n}\right)}$$

On simplification, we get Mass flow rate of the steam in kg/sec is given as

$$m=A_2\sqrt{\frac{2n}{n-1}\frac{P_1}{v_1}\left(\left(\frac{P_2}{P_1}\right)^{2/n}-\left(\frac{P_2}{P_1}\right)^{n+1/n}\right)} \tag{3.15}$$

- **Throat pressure for Maximum discharge (or) Existence of a Critical Pressure in Nozzle flow or Choked flow**

The pressure at which the area is minimum and the discharge per unit area is maximum is termed as critical pressure. The smallest cross-sectional area of the nozzle is known as throat.

Now mass flow rate per unit area at the throat is given as

$$\frac{m}{A_2}=\sqrt{\frac{2n}{n-1}\frac{P_1}{v_1}\left(\left(\frac{P_2}{P_1}\right)^{2/n}-\left(\frac{P_2}{P_1}\right)^{n+1/n}\right)} \tag{3.16}$$

In this equation, mass flow rate per unit area is a function of $P_2/P_1$. There will be only one value of ratio $P_2/P_1$ which will produce maximum discharge for the nozzle.

The maximum discharge can be found by differentiating eq. (3.16) with $P_2/P_1$ and equating to zero.

$$\frac{d}{d(P_2/P_1)}\left[\sqrt{\frac{2n}{n-1}\frac{P_1}{v_1}\left(\left(\frac{P_2}{P_1}\right)^{2/n}-\left(\frac{P_2}{P_1}\right)^{n+1/n}\right)}\right]=0$$

Let $P_2/P_1$ = r. Then On simplification, we get

$$r=\left(\frac{P_2}{P_1}\right)=\left(\frac{P_c}{P_1}\right)=\left(\frac{2}{n+1}\right)^{n/n-1} \tag{3.17}$$

Eq. (3.17) is called critical pressure ratio and depends on the value of ‘n’.

**Case 1:** For Saturated Steam, n = 1.135.

The critical pressure ratio is r = 0.5774 = 0.58 (approximately).

**Case 2:** For superheated steam, n = 1.3

The critical pressure ratio is r = 0.5457 = 0.546 (approximately).

**Case 3:** For wet steam, n = 1.035+0.1x.

- **Critical temperature ratio**

The ratio of the temperature at the section of the nozzle where the velocity is sonic to the inlet temperature is called the critical temperature ratio.

$$\therefore \ Critical \ \ temperature \ \ ratio = \frac{T_2}{T_1} = \frac{T_c}{T_1} = \left(\frac{P_2}{P_1}\right)^{n-1/n} = \left(\left(\frac{2}{n+1}\right)^{n/n-1}\right)^{n-1/n} = \left(\frac{2}{n+1}\right) \quad (3.18)$$

- **Note worthy points:**

  - ✓ For a convergent nozzle, maximum mass flow through is obtained when the pressure ratio across the nozzle is the critical pressure ratio.
  - ✓ When a nozzle operates with the maximum mass flow rate, it is said to be choked. A correctly designed convergent-divergent nozzle is always choked i.e. it must have critical pressure at the throat in isentropic flow with zero approaching velocity.
  - ✓ For a convergent – divergent nozzle with sonic velocity at the throat, the cross-sectional area of the nozzle at the throat fixes the maximum mass flow through the nozzle for fixed inlet conditions.

- **Nozzle efficiency and Coefficient of Discharge**

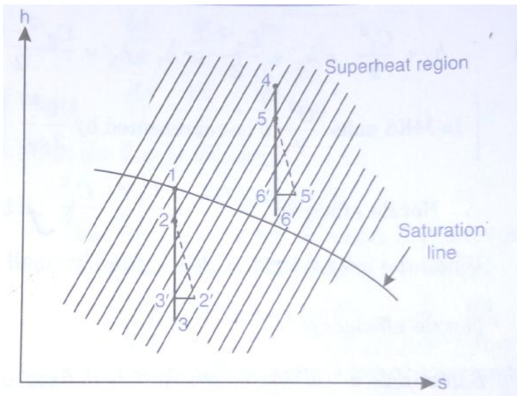
When the steam flows through a nozzle, the final velocity of steam for a given pressure drop is reduced due to the following reasons:

  - ✓ The friction between the nozzle surface and steam;
  - ✓ The internal friction of steam itself and The shock losses

These friction losses occur between the throat and exit in convergent – divergent nozzle. These losses entail the following effects:

  - ✓ The expansion is no more isentropic and enthalpy drop is reduced;
  - ✓ The final dryness fraction is steam is increased as the kinetic energy gets converted into heat due to friction and is absorbed by the steam.

- ✓ The specific volume of the steam is increased as the steam becomes more dry due to this frictional heating.




*Definition: **Nozzle efficiency*** is defined as the ratio of actual enthalpy drop to enthalpy drop in isentropic expansion process. In other words, it is the ratio of actual gain in kinetic energy to the gain in kinetic energy in isentropic expansion process.

$$\eta_{nozzle} = \frac{\left(h_1 - h_2^{'}\right)_{actual}}{\left(h_1 - h_3\right)_{isentropic}} = \frac{C_2^{2'} - C_1^2}{C_2^2 - C_1^2} \tag{3.19}$$

If $C_1$ is small compared to $C_2$, then nozzle efficiency is given by

$$\eta_{nozzle} = \frac{C_2^{2'}}{C_2^2} = K_n^2$$

Where $K_n$ – velocity coefficient of the nozzle. Thus velocity coefficient is square root of nozzle efficiency, when the inlet velocity is assumed to be negligible.

*Definition*: ***Coefficient of discharge*** of a nozzle may be defined as the ratio of actual mass flow rate to the mass flow rate due to isentropic expansion.

$$Coefficient\ \ of\ \ disch\arg e, c_d = \frac{m_{actual}}{m_{isentropic}} \tag{3.20}$$

**Note**: *If the coefficient of discharge and nozzle efficiency is given, then the throat area should be calculated based on the coefficient of discharge and nozzle efficiency should be used to calculate exit area.*

- **Supersaturated flow in Nozzles**

  - ✓ During the ***expansion of initially dry saturated steam*** in a nozzle, due to the effect of friction, the actual discharge is about 3 to 5% less than theoretical value but the measured value is found to be 1 to 3% greater than the theoretical value.

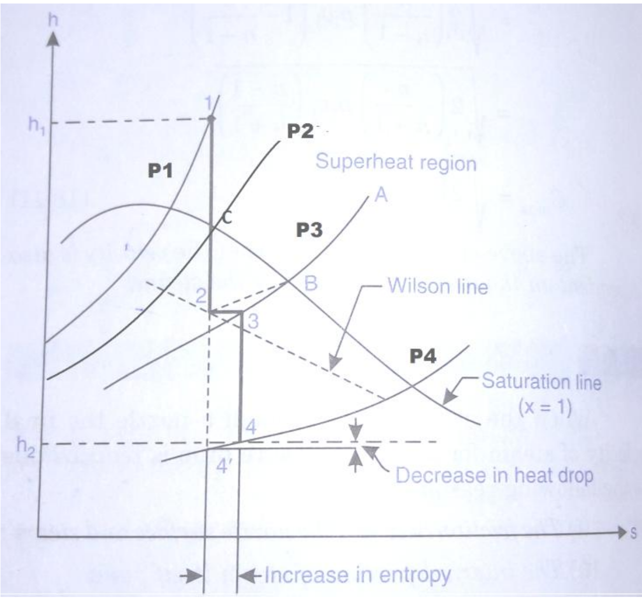
- ✓ During the ***expansion of superheated steam in a nozzle***, the increase in discharge is just sufficient to compensate the losses due to friction, so the measured value is more or less in agreement with the theoretical one.
- ✓ The increase in measured discharge to the theoretically calculated discharge is due to the time lag in the condensation of steam and thus the steam remains in dry state instead of wet; this phenomenon is called "**supersaturation**"**.**

There are three causes involved in supersaturation.

(a) In commercial steam generation process, tiny dust particles are always present and the condensation starts surrounding them. When such particles are absent, then the condensation process is considerably delayed and the temperature continues to fall. This phenomenon is known as supersaturation. When a certain degree of supersaturation is reached, it appears that the presence of foreign particles is no longer necessary and that the equilibrium can then be attained completely.

(b) A certain time interval is essential for the collection of molecules to form droplets. The flow of the steam through a short convergent part of the nozzle is with a high velocity and of the order of one-ten thousandth of a second, which may be quite inadequate for the condensation of steam and the condensation is delayed and therefore, the flow is supersaturated.

(c) Water drops are not formed when steam and water are in thermal equilibrium. At any given temperature, the vapor pressure which is proportional to the no. of vapor molecules per unit volume must be greater surround the tiny droplet. As a result, the molecules of a very small water droplet will go on evaporating into a space that is saturated and the space will therefore



become supersaturated with water vapor.

The supersaturation phenomenon is shown in the fig. The initial condition of the steam is at the point 1 in superheated region on the pressure line P1. Process 1-C shows the isentropic expansion of the steam in thermal equilibrium up to saturation. Due to above mentioned 3 reasons, the condensation of steam will not start at point C resulting in delay of equilibrium between vapor and liquid phase. The vapor continues to expand in dry state instead of wet state. This process continues until the density of superheated steam is about 8 times the density of saturated vapor of the same pressure. When this limit is reached at point 2, the steam will suddenly condense at constant pressure and constant enthalpy to its normal state, as shown by the horizontal line 2-3. The limit of supersaturation can be represented by a line known as Wilson line on the Mollier chart as shown in fig.

The process C-2 represents the expansion under supersaturation condition, which is not in equilibrium. The zone between the Wilson line and the dry saturated line is called the supersaturated zone and the flow through this zone is called the supersaturated flow. This is also called undercooled because at any pressure P2 and P3 i.e. within the supersaturated zone, the temperature of the vapor is always less than saturation temperature corresponding to that pressure.

Process 2-4' represents the isentropic expansion of steam when there is no supersaturation. Because of supersaturation, the expansion process follow the path 3-4, which is isentropic but in thermal equilibrium.

It is assumed that the supersaturated vapor behaves like a superheated steam and the index of steam is 1.3. So the equation for the expansion of superheated steam is

$$\frac{T_2}{T_1} = \left(\frac{P_3}{P_1}\right)^{n-1/n} \qquad (3.21)$$

From eq. (3.21) the actual temperature ($T_3$) of the supersaturated steam can be calculated.

Velocity at the throat for the supersaturated flow is given by eq. (3.14).

$$C_2 = \sqrt{\frac{2n}{n-1} P_1 v_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{n-1/n}\right)}$$

where the subscripts 1 and 2 refers to entry and throat conditions.

At critical pressure ratio

$$r = \left(\frac{P_2}{P_1}\right) = \left(\frac{P_c}{P_1}\right) = \left(\frac{2}{n+1}\right)^{n/n-1}$$

For superheated steam, n = 1.3. The critical pressure ratio is r = 0.5457 = 0.546.

For maximum flow conditions,

$$\frac{m}{A_2} = \sqrt{\frac{2n}{n-1}\frac{P_1}{v_1}\left(\left(\frac{P_2}{P_1}\right)^{2/n} - \left(\frac{P_2}{P_1}\right)^{n+1/n}\right)} = 0.66726\sqrt{\frac{P_1}{v_1}} \tag{3.23}$$

**Effects of super saturation**

- ✓ The heat drop is reduced below that for thermal equilibrium as a consequence the exit velocity of the steam is reduced considerably.
- ✓ There is an increase in the entropy and specific volume of the steam.
- ✓ The dryness fraction of the steam increases.
- ✓ Since the condensation does not takes place during supersaturation, the temperature at which it occurs will be less than the saturation temperature corresponding to the pressure. Therefore, the density of steam will be more than that for the equilibrium conditions which gives the increase in the mass of the steam discharged.

## *5.1) Definition of Steam Condenser:*

"Condenser is a closed vessel in which exhaust steam from steam turbine is condensed by cooling water, and vacuum is maintained, resulting in an increase in work done and thermal efficiency of steam turbine plant and use of condensate as feed water to the boiler."

## *5.2) Advantages of a Condenser in a steam power plant:*

i. Improved work done and efficieny due to low pressure of the condense.

ii. Recovery of condensate to be fed to the boiler as a high quality feed water for reuse.

iii. Reduced steam consumption for the same power output due to increased work done.

iv. Reduced thermal stresses due to high pressure of feed water entering to boiler.

v. Economy in water softening plant as only make up water is to be treated instead of full feed water.

## *5.3) Requirements of a Steam Condensing Plant:*

The principal requirements of a condensing plant as shown in fig 5.2 are as follows

❖ *Condenser:*

It is a closed vessel in which steam is condensed .The steam gives up latent heat to coolant (which is water) during the process of condensation.

❖ *Condensate pump*:

It is a pump, which extract condensate (I.e. condensed steam) from the condenser to the hot-well.

❖ *Hot well*:

It is a sump between the condenser and boiler, which receives condensate pumped by the condensate pump.

❖ *Boiler Feed water pump*:

It is a pump, which pumps the condensate from hot well to boiler. This is done by increasing the pressure of condensate above the boiler pressure.

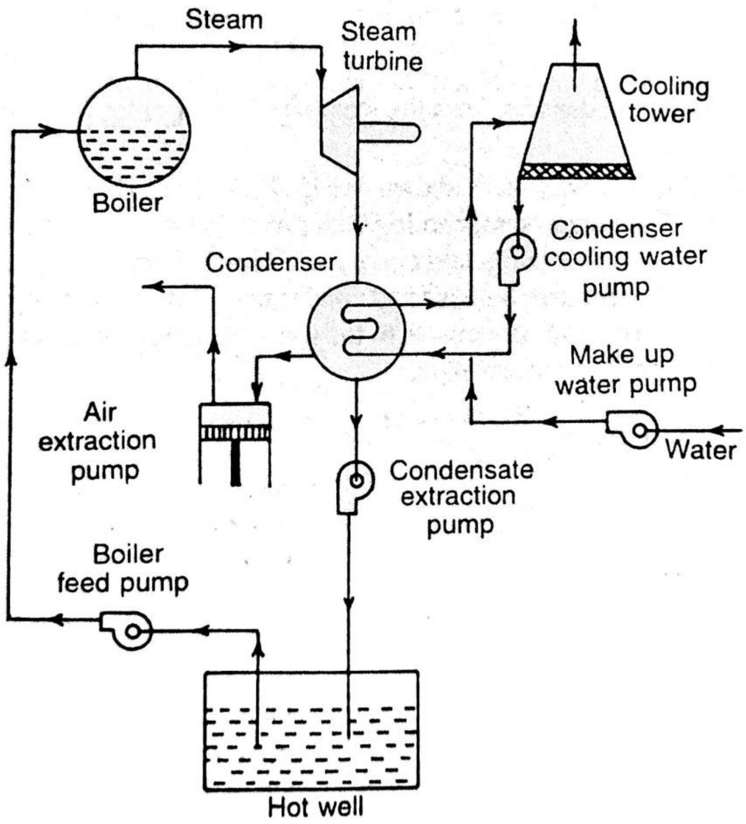



*Fig. 5.2 Requirements of steam condensing plant*

❖ *Air Extraction pump*:

The function of air extraction pump is to remove air and other non-condensable gases from the condenser.

❖ *Cooling Tower:*

A cooling Tower or a spray pumps to re-cool the circulating water of the condenser which is heated in the condenser due to condensation of steam. Cooling Tower is essential where there is a scarcity of water.

❖ *Cooling water circulation pump:*

It is a pump, which circulates the cooling water through the condenser.

❖ *Relief valve*:

The purpose of Relief valve is to relieve the steam from the condenser when the condenser is not in working order. Using this, the plant becomes a non-condensing unit.

## *5.4) Classification of Steam Condensers:*

Condensers can be broadly classified on the basis of type of heat exchange i.e. direct or indirect contact condensers.

(i) Direct contact type or mixing type or Jet condenser

(ii) Indirect contact type or Non-mixing type or Surface condenser
(iii) Evaporative condenser

***Jet condenser:*** Jet condensers have direct contact between steam and cooling fluid thereby causing contamination of condensate. Jet condensers are divided into:

a) *Parallel flow jet condenser*: In this condenser, the steam and cooling water flow are in the same direction.
b) *Counter flow jet condenser*: In this condenser, the steam flows in opposite direction to the cooling water.

***Surface condenser:*** Surface condensers have indirect heat exchange through metal interface and the two fluids do not come in direct contact to each other.

The surface condensers are classified according to:

a) Number of water passes- single pass and multi pass condensers
b) Direction of condensate flow and tube arrangement- Down flow type and Central Flow type.

***Evaporative condenser:*** Evaporative condensers use evaporation of water for heat extraction and is well suited for dry weather so that evaporation is not difficult.

## 5.4.1) Jet Condensers:

### *a) Parallel flow Jet Condenser*:

In parallel flow jet condensers, both the steam and water enter at the top and the mixture is removed from the bottom.

The principle of this condenser is shown in fig. The exhaust steam is condensed when it mixes up with water. The condensate, cooling water and air flow downwards and are removed by two separate pumps known as air pump and condensate pump. Sometimes a single pump known as the wet air pump is also used to remove both air and condensate. But the former gives a greater vacuum. The condensate pump delivers the condensate to the hot well, from where surplus water flows to the cooling water tank through an overflow pipe.

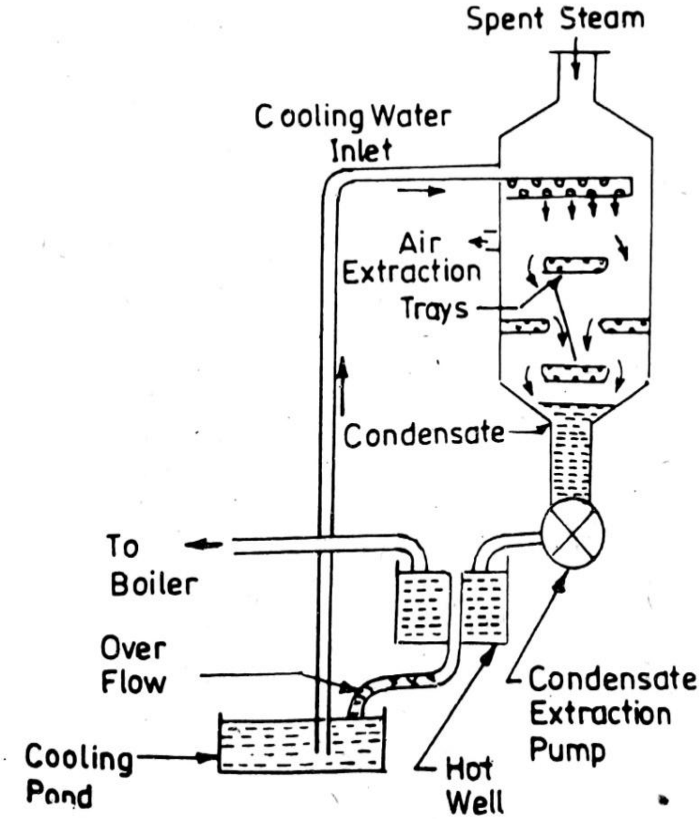


Fig. 18.4. Low Level Jet Condenser (Parallel flow).

### *b) Counter flow jet condenser:*

The vacuum is created by the air pump, placed at the top of the condenser shell. This draws the supply of cooling water, which falls in a larger number of jets, through perforated conical plate as shown in fig.5.4. The falling water is caught in the trays, from which it escapes in a second series of jets and meets the exhaust steam entering at the bottom. The rapid condensation occurs and the condensate and cooling water descends through a vertical pipe to the condensate pump, which delivers it to the hot well.

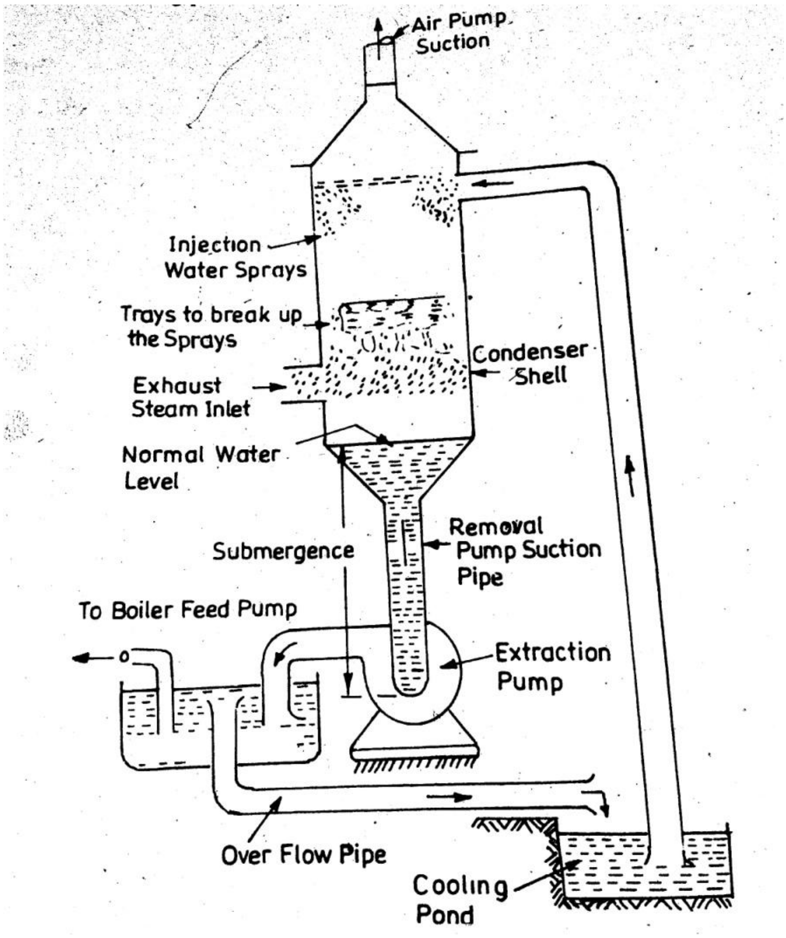



*Fig. 5.4 counter flow Jet condenser*

## *c) Barometric or high level jet condensers:*

These condensers are provided at a high level with a long vertical discharge pipe as shown in fig.5.5. In high level jet condensers, exhaust steam enters at the bottom, flows upwards and meets the down coming cooling water in the same way as that of low level jet condenser. The vacuum is created by the air pump, placed at the top of the condenser shell. The condensate and cooling water descends through a vertical pipe to the hot well without the aid of any pump. The surplus water from the hot well flows to the cooling water tank through an overflow pipe.

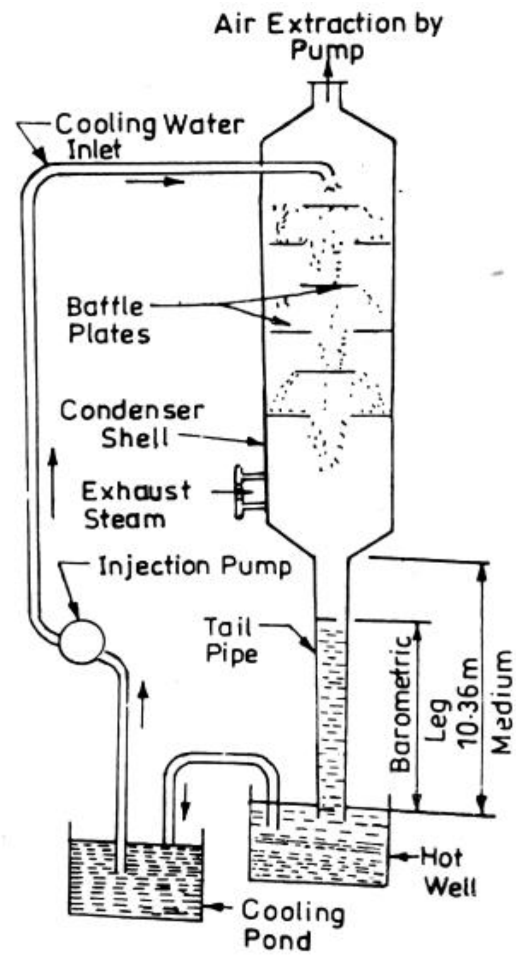



*Fig. 5.5 High level or barometric jet condenser*

## *d)Ejector condenser:*

In ejector condensers, the steam and water mix up while passing through a series of metal cones. Water enters at the top through a number of guide cones. The exhaust steam enters the condenser through non return valve arrangement. The steam and air then passes through the hollow truncated cones.

After this it is dragged into the diverging cones where its kinetic energy is partly transformed to pressure energy. The condensate and cooling water is then discharged to the hot well as shown in fig.5.6

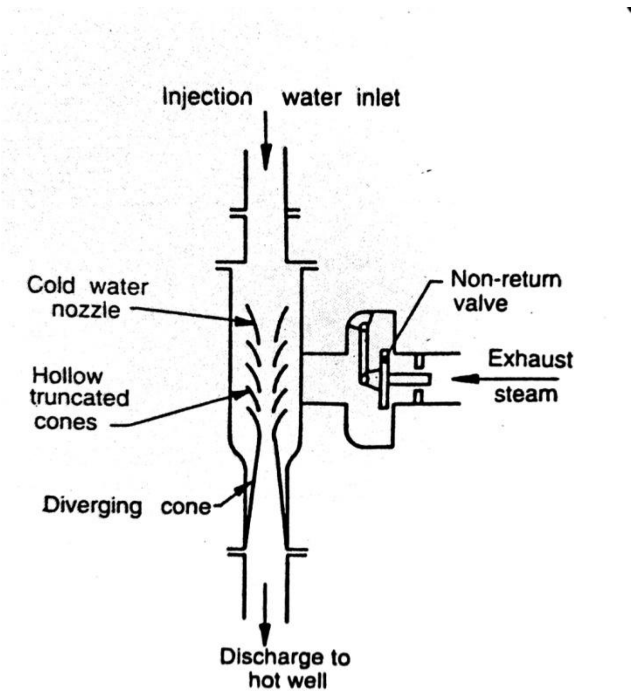



Fig 5.6 Ejector condenser

## 5.4.2) Surface condensers:

A surface condenser has a great advantage over the jet condensers, as the condensate does not mix up with the cooling water. As a result of this, whole condensate can be reused in the boiler. This type of condenser is essential in ships which can carry only a limited quantity of fresh water for the boilers. It is also widely used in land installations, where inferior water is available or the better quantity of water for feed is to be used economically.

Fig shows a longitudinal section of a two pass surface condenser. It consists of a horizontal cast iron cylindrical vessel packed with tubes, through which the cooling water flows. The ends of the condenser are cut off by vertical perforated type plates into which water tubes are fixed. This is done in such a manner that the leakage of water into the centre condensing space is prevented. The water tubes pass horizontally through the main condensing space for the steam. The steam enters at the top and is forced to flow downwards over the tubes due to the suction of the extraction pump at the bottom.

The cooling water flows in one direction through the lower half of the tubes and return to opposite direction through the upper half as shown in fig.

## **Types of surface condensers**:

The surface condensers may be further classified on the basis of the direction of flow of the condensate, the arrangement of tubing system and the position of the extraction pump, into following four types:

a) Down flow surface condensers
b) Central flow surface condenser
c) Regenerative surface condenser

### *a) Down flow surface condenser:*

In down flow surface condensers, the exhaust steam enters at the top and flow down wards over the tubes due to force of gravity as well as suction of the extraction pump fitted at the bottom. The condensate is collected at the bottom and then pumped by the extraction pump. The dry air pump suction pipe, which is provided near the bottom, is covered by a baffle so as to prevent the entry of condensed steam into it as shown in fig.5.7.

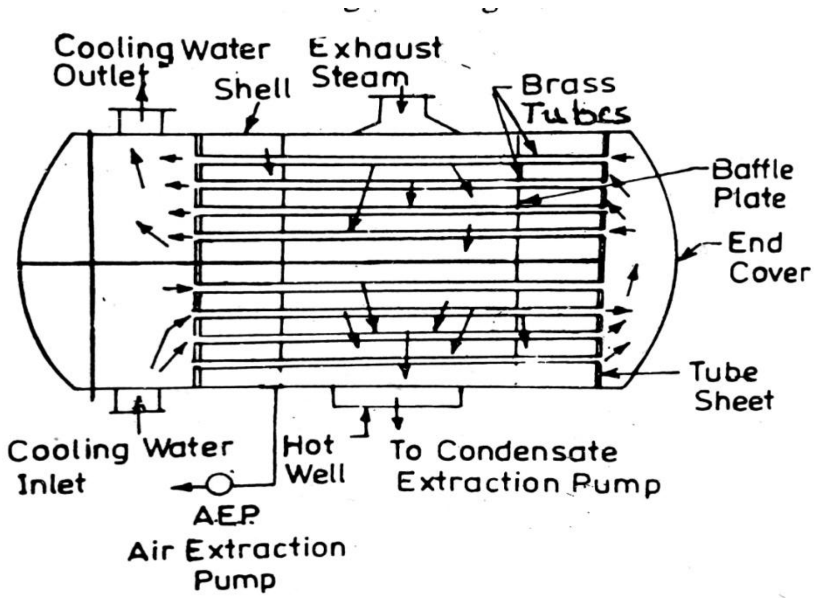



*Fig. 5.7 Two pass Down flow surface condenser*

As the steam flows perpendicular to the direction of flow of cooling water, this is also called a cross surface condenser.

## *b) Central flow surface condenser:*

In central flow surface condensers, the exhaust steam enters at the top and flow down wards. The suction pipe of the air extraction pump is placed in the centre of the tube nest as shown in fig. This causes the steam to flow radially inwards over the tubes towards the suction pipe. The condensate is collected at the bottom and then pumped by the extraction pump as shown in fig.5.8.

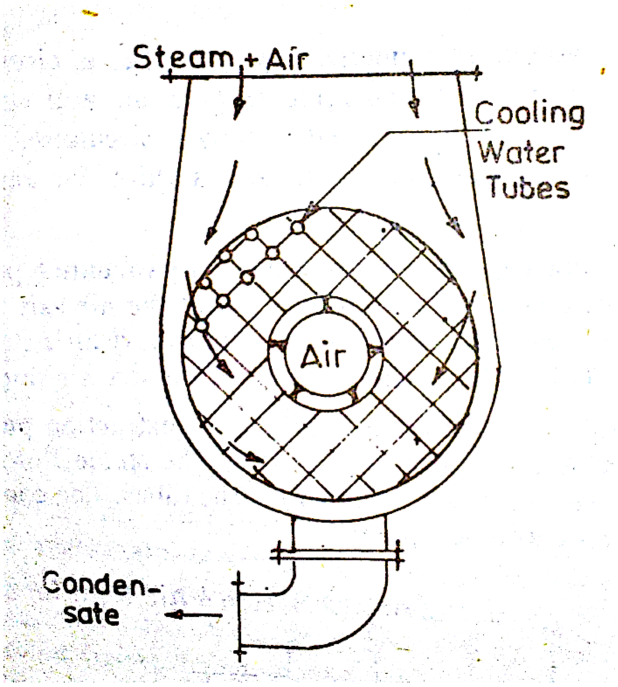



*Fig. 5.8 Central flow surface condenser*

The central flow surface condenser is an improvement over the down flow type as the steam is directed radially inwards by a volute casing around the tube nest. It thus, gives an access to the whole periphery of the tubes.

### *c) Regenerative surface condenser:*

In regenerative surface condenser, the condensate is heated by a regenerative method. The condensate after leaving the tubes is passes through the exhaust steam from the engine or turbine. It thus, raises the temperature for use as feed water for the boiler.

## 5.4.3) Evaporative condenser:

The steam to be condensed enters at the top of a series of pipes outside of which a film of cold water is falling. At the same time, a current of air circulates over the water film, causing rapid evaporation of some of the cooling water. As a result of this, the steam circulating inside the pipe is condensed.

The remaining cooling water is collected at an increased temperature and is reused. Its original temperature is restored by the addition of the requisite quantity of cold water.

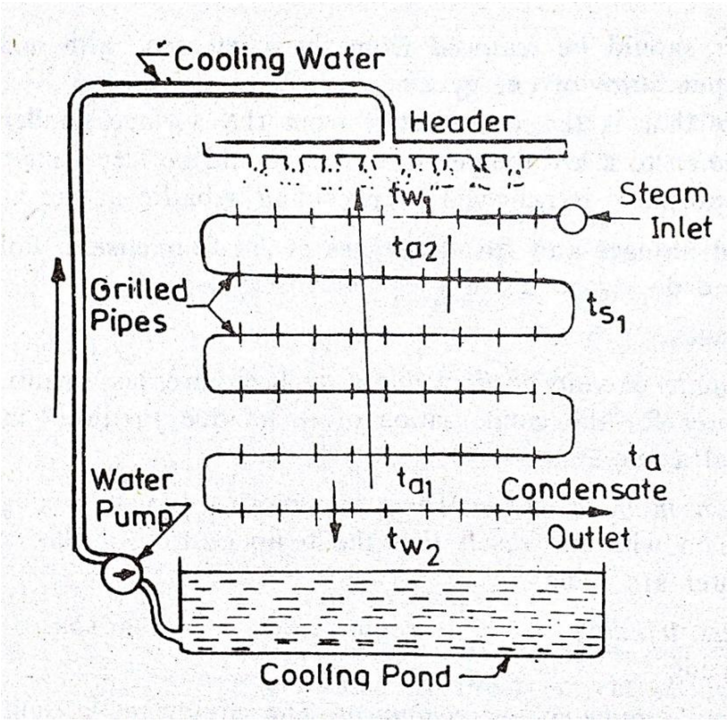



*Fig. 5.9 Evaporative Condenser*

The evaporative condensers are provided when the circulating water is to be used again and again. These condensers consist of sheets of gilled piping, which is bent backwards and forward and placed in a vertical plane as shown in fig.5.9.

## 5.5) Sources of Air Leakage:

The performance of condenser is adversely affected by the presence of air in the condenser. The sources of air in the condenser are due to following:

- Leakage through packing glands and joints.
- Leakage through condenser accessories, such as atmospheric relief valve etc.
- Air associated with exhaust steam may also liberate at low pressure.
- In the jet condenser, the dissolved air in the cooling water liberates at low pressure.

It is to be noted that in a well designed and properly maintained surface condenser, *the amount of air leakage is about 5kg per 10,000 kg of steam condensed in a steam turbine plant and 15 kg per 10,000 kg of steam in a steam engine plant.*

## 5.6) The Effects of presence of air in a condenser:

- The pressure in the condenser is increase; this reduces the work done by the engine or turbine.
- Partial pressure of steam and temperature are reduced. The steam tables tell us that at lower pressure, the latent heat of steam is more. In order to remove this great quantity of heat, more cooling water has to be supplied and thus under-cooling of condensate is likely to be more severe resulting in lower overall efficiency.

- The presence of air reduces the rate of condensation of steam since the abstraction of heat by the circulating cooling water is partly from the steam and partly from the air.
- The rate of heat transfer from the vapour is reduced due to poor thermal conductivity of air. Thus, the surface area of the tubes has to be increased for a given condenser duty.
- An air extraction pump is needed to remove air; still some quantity of steam escapes with the air even after shielding of the air extraction section. This reduces the amount of condensate. Moreover, the condensate is under-cooled, with the result that more heat has to be supplied to the feed water in the boiler.

## 5.7) Methods to detect Air leakage in Condensers:

The most important reason for air in the condenser is due to leakage of air. The following method is adopted to check whether there is a leakage in the condenser or not.

The plant is run until the pressure and temperature conditions are steady in the condenser. At this stage the steam supply from the engine is shut-off, and the air and condenser extraction pumps are simultaneously closed down bringing about complete isolation of the condenser. The vacuum gauge and thermometer readings are recorded. If there is any leakage, the vacuum gauge readings will fall after sometimes.

The source of air leakage can be checked by the following methods-

- By passing a candle flame over possible openings, large leakages can be detected when the condenser is under vacuum.
- By putting the condenser under air pressure, Its effect on soap water is observed at the points where infiltration is possible.
- Under operating conditions, peppermint oil is spread on joint from where the leakage is suspected, a Check is made of the peppermint odour in the air ejector discharge.

Thus it is most important to check all the air leakages and to remove any air that may be in the condenser. In practice of course, it is impossible to remove all the air. Therefore, it is continuously removed by the air pump suction, which sucks it from the condenser, compresses it to a little above atmospheric pressure so that it is forced out.

## 5.8) Comparison of Jet condensers and surface condensers

| S.No. | Jet Condensers | Surface Condensers |
|---|---|---|
| 1 | Cooling water and steam are mixed up. | Cooling water and steam are not mixed up. |
| 2 | Not Suitable for high capacity plants | Suitable for high capacity plants |
| 3 | Condensate is wasted. | Condensate is re-used. |
| 4 | It requires less quantity of circulating cooling water | It requires a large quantity of circulating cooling water. |
| 5 | Condensing plant is simple and economical. | Condensing plant is complicated and expensive. |
| 6 | Maintenance cost is low. | Maintenance cost is high. |
| 7 | More power is required for air pump. | Less power is required for air pump. |
| 8 | High power is required for water pumping. | Less power is required for water pumping. |

## 5.10) Advantages and disadvantages of Jet condensers:

### *Advantages:*

The following are the advantages and disadvantages of jet condensers**:**

1) *Smaller quantity of circulating water*: It requires less quantity of circulating water for the condensation of steam due to direct mixing of water and steam.
2) *More intimate mixing*: There is a more thorough mixing of water and steam with the result that the temperature of the condensate and water are same.
3) *Less building space*: It requires a less building space due to direct mixing.
4) *Simple and low cost equipment*: The equipment is simple and low in cost.
5) *Absence of pumps*: Barometer and Jet condensers do not require cooling water pumps.

### *Disadvantages:*

1) *Wastage of condensate***:** The condensate is mixed with water and where pure water is not available .It cannot be used as feed water, thus it is wasted.
2) *Pure cooling water*: If the condensate is to be used as feed water, the cooling water should be pure and free from harmful impurities.
3) *Costly pipe*: In the barometric condenser, long pipe is used which increases the cost of the condenser**.**
4) *Engine flood*: In the low level jet condenser, there is always a possibility of the water rushing into the engine, if condensate extraction pump is fails.
5) *Reduced vacuum*: There is reduced vacuum on account of leakage in the long exhaust pipe of the barometric condenser and liberation of air dissolved in cooling water.
6) *High power extraction pump*: The air extraction pump needs almost double the amount of power required by the surface condenser.

## 5.12) Dalton's Law of Partial pressure*:*

Dalton's law of partial pressure is very helpful for the analytical treatment of problems dealing with a mixture of gases or vapor. This law states that.-

*"The total pressure exerted in a container having a mixture of gases or gas and vapor is equal to the sum of the partial pressures of the individual constituents at the common temperature."*

Suppose there is a mixture of air and steam in condenser.

Let

$p_a$ = partial pressure of air at temperature, t $^0$C

$p_s$ = partial pressure of steam (water vapor) or saturation pressure of steam at temperature, t $^o$C.

p= total pressure in condenser.

t= temperature of mixture in condenser.

Then according to Dalton's law

$p = p_a + p_s$

Or

$p_a = p - p_s$

## 5.13) Vacuum Efficiency:

"Vacuum efficiency is defined as the ratio of actual vacuum as recorded by vacuum gauge to the ideal vacuum, when the air is absent."

But ideal vacuum = Barometric pressure- Absolute pressure of steam

$$\eta_{vacuum} = \frac{Actual\ vaccum\ recorded}{Aarometric\ pressure - Absolute\ pressure}$$

Thus vacuum efficiency is measure of the degree of perfection to maintain a desired vacuum in the condenser. The value of vacuum efficiency depends up on the effectiveness of air cooling and the rate at which the air removed by the air pump. The vacuum efficiency is generally around 98% and 99%.

## 5.14) Condenser Efficiency:

"The condenser Efficiency is defined as the ratio of actual rise in temperature cooling water to the maximum possible rise in temperature of the cooling water."

Let $t_s$ = Saturation temperature corresponding to condenser pressure in $^0$C

$t_{w1}$ and $t_{w2}$ = Inlet and outlet water temperature of cooling water in $^0$C

$$\eta_{condense} = \frac{t_{w1} - t_{w2}}{t_s - t_{w1}}$$

Condenser efficiency generally varies from 75% to 85%.

## 5.15) Mass of circulating water required:

The function of the circulating water in a condenser is to absorb heat from the steam and thereby to condense it. In surface condenser, the temperature of the condensate and exit water is not the same while in jet condenser it is the same. To design a condenser, it is essential to calculate the quantity of circulating water necessary for a certain capacity of steam to be condensed. Fig shows an energy exchange in a surface condenser.

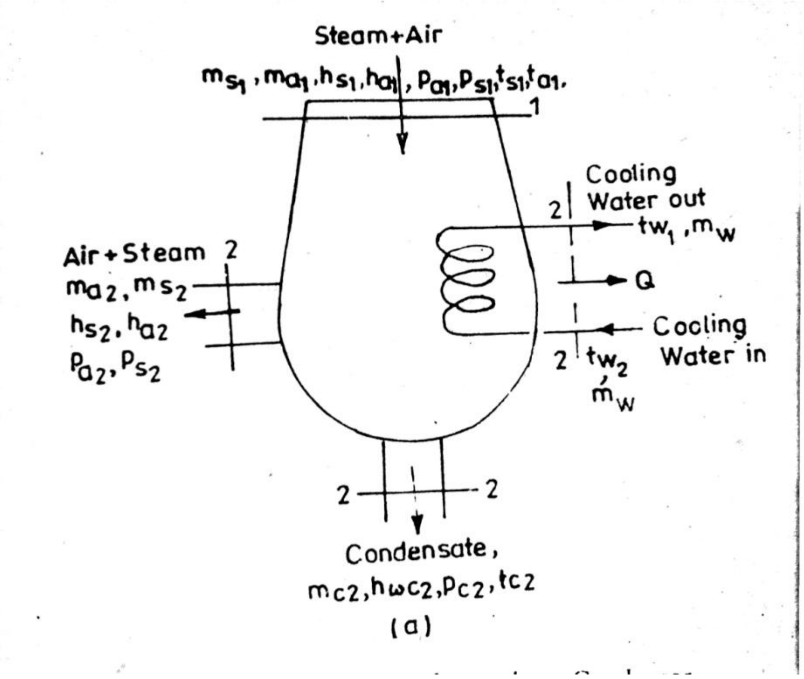



Fig. 5.12 Heat Exchange in Condenser

Let suffixes 1 and 2 refer to inlet and outlet conditions and s, w, a and c refers to steam, water, air and condensate respectively.

Let

Q = heat given up by the steam, kJ

$m_w$ = mass of circulating water, kg/h

$m_{c2}$ = mass of condensate at outlet, kg/h

$t_{c2}$ = temperature of the condensate at outlet, $^0$C

$h_{wc2}$ = sensible heat of condensate at outlet, kJ/kg

$t_{c1}$ = temperature of exhaust steam of inlet, $^0$C

$h_{w1}$ = sensible heat or enthalpy of water at $t_{s1}$ $^0$C, kJ/kg

$L_1$ = latent heat at $t_{s1}$ $^0$C, kJ/kg

$x_1$ = dryness fraction of steam

$t_{w1}$, $t_{w2}$ = inlet and outlet temperature of cooling water in $^0$C

$m_{a1}$, $m_{a2}$ = mass of inlet and extracted air, kg/h

$m_{s1}$, $m_{s2}$ = mass of entering and extracted steam, kg/h

$h_{s1}$, $h_{s2}$ = enthalpy of entering and extracted steam, kJ/kg

$h_{a1}$, $h_{a2}$ = enthalpy of entering and extracted air, kJ/kg

Neglecting kinetic and potential energy, energy entering is equal to energy leaving.

$$(m_{s1} \times h_{s1} + m_{a1} \times h_{a1}) = Q + (m_{s2} \times h_{s2} + m_{a2} \times h_{a2}) + m_{c2} \times h_{wc2}$$

$$Q = (m_{s1} \times h_{s1} + m_{a1} \times h_{a1}) - (m_{s2} \times h_{s2} + m_{a2} \times h_{a2}) - m_{c2} \times h_{wc2}$$

If all the air is extracted, then

$$m_{a1} = m_{a2}$$

$h_{a1} = h_{a2}$ may be assumed for simplicity

If there is no extraction or steam by the air pump, then

$m_{s2} = 0$, thus $m_{s1} = m_{c2} + m_{s2} = m_{c2}$

Therefore,

$$Q = m_{c2} (h_{s1} - h_{wc2})$$

But, $h_{s1} = h_{w1} + x_1.L_1$

$$Q = m_{c2} (h_{w1} + xL_1 - h_{wc2})$$

Heat received by cooling water, $= m_w.C_{pw} (t_{w2} - t_{w1})$

Heat given up by steam = heat received by cooling water

$$m_{c2} (h_{w1} + xL_1 - h_{wc2}) = m_w.C_{pw} (t_{w2} - t_{w1})$$

$$m_w = \frac{m_{c2}(h_{s1} - h_{wc2})}{C_{pw}(t_{w2} - t_{w1})}$$

For jet condenser, $t_{w2} = t_{c2}$

$$m_w = \frac{m_{c2}(h_{s1} - h_{wc2})}{C_{pw}(t_{c2} - t_{w1})}$$

# GAS TURBINES AND JET PROPULSION

**Classification of Gas turbines:**

**(A) On the basis of combustion process**

*(i) Continuous combustion or constant pressure type:* The cycle working on this principle is called Joule or Brayton cycle.

(ii) *The explosion or constant volume type:* The cycle working on this principle is known as Atkinson cycle.

**(B) On the basis of the action of expanding gases similar to steam turbine**

*(i) Impulse turbine and (ii) Impulse reaction turbine.*

**(C) On the basis of path of working substance**

*(i) Open cycle gas turbine (*working fluid enters from atmosphere and exhaust to Atmosphere.)

*(ii) Closed cycle gas turbine (*working fluid is confined within the plant), and

*(iii) Semi-closed cycle (*part of the working fluid is confined within the plant and another part flows from and to the atmosphere*)*

**(D) On the basis of direction of flow**

*(i) Axial flow and (ii) Radial Flow*

**Applications of Gas turbines**

(i) Locomotive propulsion

(ii) Central power stations

(a) Utility plants (b) combined cycle and cogeneration plants (c) stand by plants for hydro installations.

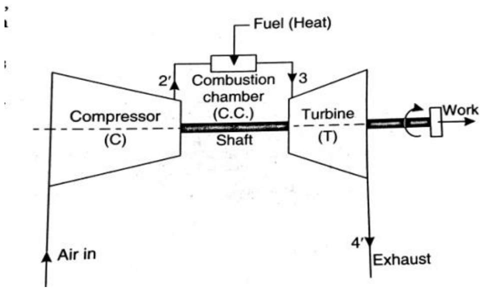
(iii) Industrial gas turbines: In Natural or crude oil pumping in the form of driver of compressors or pumps

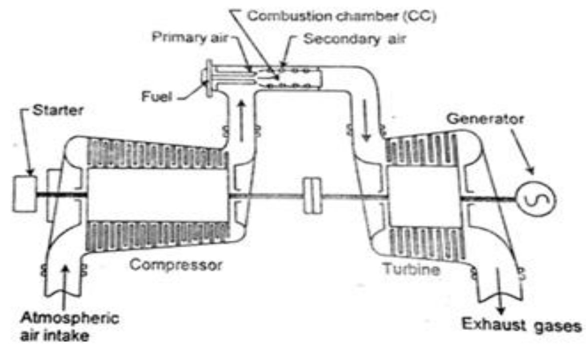
(iv) Space applications: Turbo jet, turbo prop, ram jet, pulse jet, rocket

(v) Marine applications.

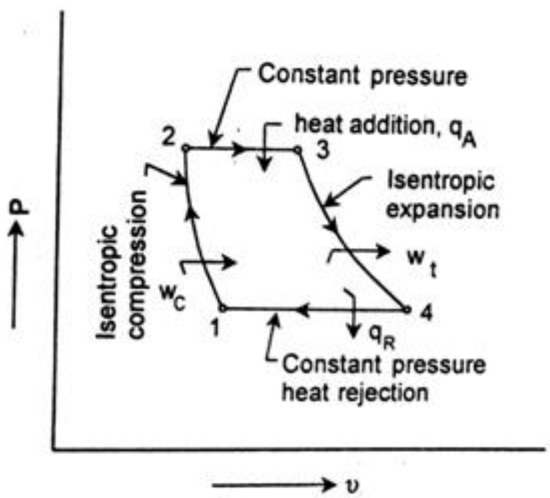
***(i) Simple open cycle Gas Turbine:(constant pressure heat addition)***

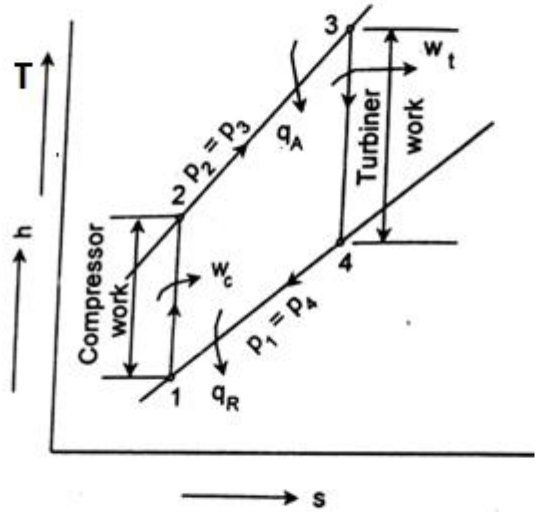
Atmosphere air is compressed from p1 to a high pressure p2 in the compressor and delivered to the burner or combustion chamber (CC) where fuel is injected and burned. The combustion process occurs nearly at constant pressure. Due to combustion heat is added to the working fluid and temperature of working fluid rises from T2 to T3. The products of combustion from the combustion chamber are expanded in the turbine from p2 to atmosphere pressure p1, and then discharges into atmosphere. The turbine and compressor are mechanically coupled, so the network is equal to the difference between the work done by the turbine and work consumed by compressor.




*P-V and h-s (T-s) diagrams:*




Heat supplied= $h_3 - h_2 = c_p\left(T_3 - T_2\right)$

Heat rejected= $h_4 - h_1 = c_p\left(T_4 - T_1\right)$

Net work=Heat supplied –Heat rejected= $c_p\left[\left(T_3 - T_2\right) - \left(T_4 - T_1\right)\right]$

Work done by turbine= $h_3 - h_4 = c_p\left(T_3 - T_4\right)$

Work consumed by compressor= $h_2 - h_1 = c_p\left(T_2 - T_1\right)$

Thermal Efficiency = net work/Heat supplied

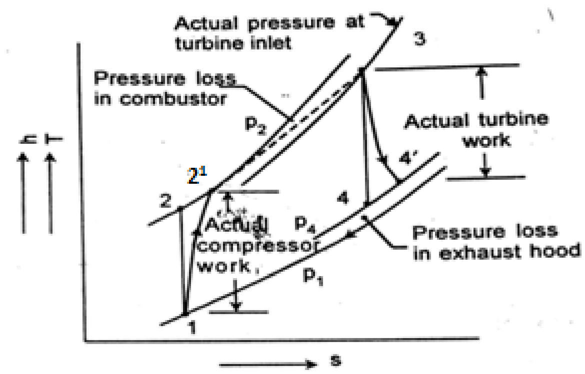
$$= \frac{c_p\left[\left(T_3 - T_2\right) - \left(T_4 - T_1\right)\right]}{c_p\left(T_3 - T_2\right)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

We know that $\frac{T_2}{T_1} = \left[\frac{p_2}{p_1}\right]^{\frac{\gamma-1}{\gamma}}$ && $\frac{T_3}{T_4} = \left[\frac{p_3}{p_4}\right]^{\frac{\gamma-1}{\gamma}}$

$$\eta_{th} = 1 - \frac{T_1}{T_2}$$

$$\eta_{th} = 1 - \frac{1}{\left[\frac{p_2}{p_1}\right]^{\frac{\gamma-1}{\gamma}}} = 1 - \frac{1}{\left[r_p\right]^{\frac{\gamma-1}{\gamma}}}$$

**Actual cycle:**




1 – 2 is isentropic compression. 1 – 2′ is actual compression.

3 – 4 is isentropic expansion. 3 – 4′ is actual expansion.

$\eta_c$ =Isentropic compression work/Actual compression work

$$\eta_c = \frac{T_2 - T_1}{T_2^1 - T_1}$$

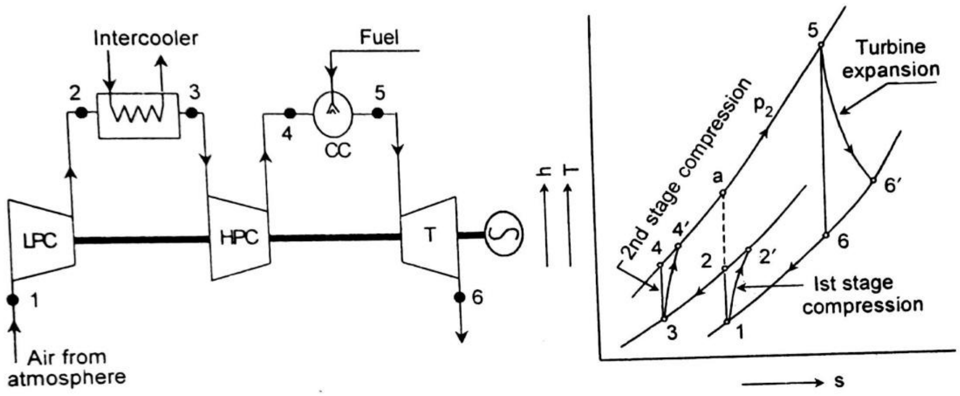
Turbine Efficiency=Actual turbine work/Isentropic Turbine work

Actual turbine work= $h_3 - h_4^1 = c_{pg}\left(T_3 - T_4^1\right)$

Turbine efficiency= $\eta_t = \frac{T_3 - T_4^1}{T_3 - T_4}$

**Methods for improving the efficiency of Gas turbine:**

(i) **Intercooling**




**Theoretically**

- Work input with Inter cooling $= c_p\left[\left(T_2 - T_1\right) - \left(T_4 - T_3\right)\right]$
- Work ratio=Net work/Gross work out put.

  = (Work of Expansion-work of compression)/Work of expansion

  Work of expansion, i.e turbine work = $h_5 - h_6 = c_p\left(T_5 - T_6\right)$

  Compressor work = $(h_2 - h_1) - (h_4 - h_3)$

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

So thermal efficiency = $\dfrac{\left[\left(T_5 - T_6\right) - \left(T_4 - T_3\right) - \left(T_2 - T_1\right)\right]}{T_5 - T_4}$

For minimum compression work in the compressor or maximum work out put in this cycle, only the compressor is responsible. And here $T_3 = T_1$.

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

$$c_p T_1\left\{\left(\frac{p_x}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 + \left(\frac{p_2}{p_x}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\}$$

Let $c_p T_1 = A$ and $\dfrac{\gamma - 1}{\gamma} = a$

$$w_c = \left[A\left(\frac{p_x}{p_1}\right)^a - 2 + \left(\frac{p_2}{p_x}\right)^a\right]$$

For minimum work, differentiate $w_c$ wrt $p_x$ and equal to zero.

$$ap_x^{a-1}p_1^{-a} - ap_x^{-a-1}p_2^a = 0$$

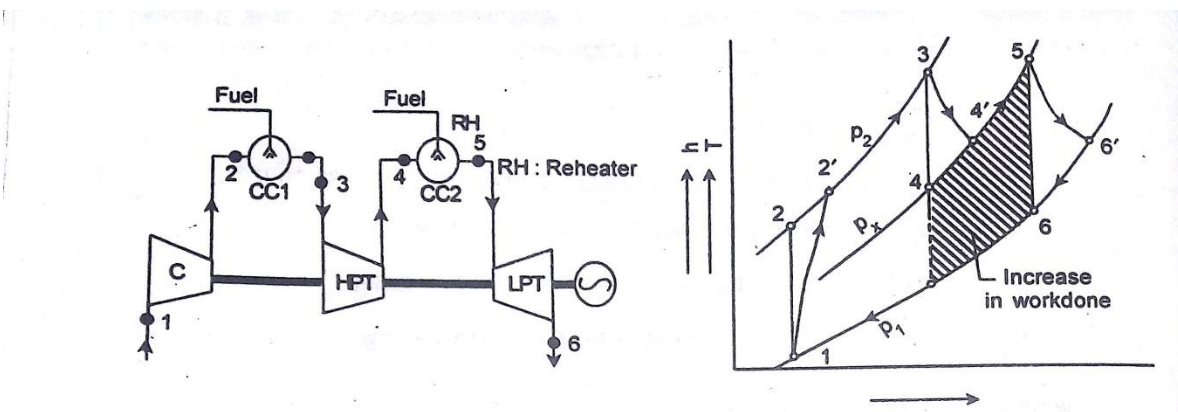
$$\frac{p_x^{a-1}}{p_1^a} = \frac{p_2^a}{p_x^{a+1}}$$

$$p_x = \sqrt{p_1 p_2}$$

- Heat supplied with Inter cooling= $c_p\left(T_5 - T_4\right)$
- Heat supplied without Inter cooling= $c_p\left(T_5 - T_a\right)$

  Thus heat supplied when intercooling is used is greater than woth no intercooling. Although the network out put is increased by intercooling, it is found that the increase in heat to be supplied causes the thermal efficiency to decrease.

**Reheating:**




By reheating or adding heat to the gases after they have passed through a part of the rows of the turbine blading, a further increase in work done is obtained. In reheating, the gas temperature, which has dropped due to expansion is brought back to approximately the initial temperature for the expansion in the next stage.

Theoritically,

Net work = $(h_3 - h_4) + (h_5 - h_6) - (h_2 - h_1)$

Heat supplied= $(h_3 - h_2) + (h_5 - h_4)$

$$\eta_{th} = \frac{(h_3 - h_4) + (h_5 - h_6) - (h_2 - h_1)}{(h_3 - h_2) + (h_5 - h_4)}$$

For maximum work output, there must ve an optimum pressure at which reheating should be done. As we know, the compressor work $W_c$ is not effected by reheating, so for the maximum output, we have to find the condition where turbine work $W_t$ is maximum.

$$W_t = c_p(T_3 - T_4) + c_p(T_5 - T_6)$$

$$= \mathrm{c}_p\left[T_3\left\{1-\left(\frac{p_x}{p_2}\right)^{\frac{\gamma-1}{\gamma}}\right\}+T_5\left\{1-\left(\frac{p_1}{p_x}\right)^{\frac{\gamma-1}{\gamma}}\right\}\right]$$

Here $T_3$= and $c_p$, $T_3$, $p_1$,$p_2$ are constants. The only variable is $p_x$.

Let $c_pT_3 = A$ and $\frac{\gamma-1}{\gamma} = a$

Then $w_t = \left[A\left(\frac{p_x}{p_2}\right)^a + 2 - \left(\frac{p_1}{p_x}\right)^a\right]$

For maximum work, differentiate the above equation with respect to $p_x$ and equate to zero.

$$ap_x^{a-1}p_2^{-a} - ap_x^{-a-1}p_1^a = 0$$

$$\frac{p_x^{a-1}}{p_x^a} = \frac{p_1^a}{p_x^{a+1}}$$

or

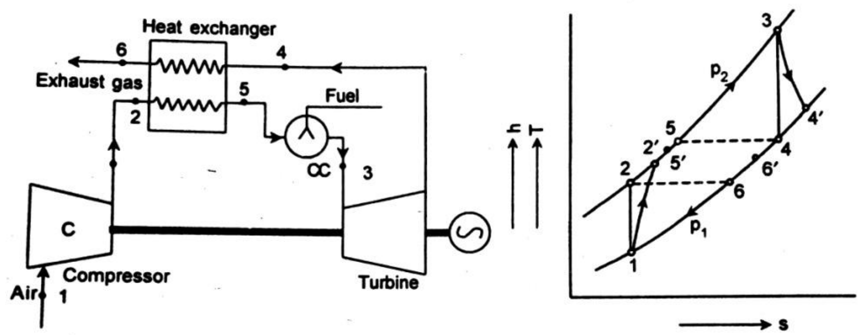
$$p_x = \sqrt{p_1p_2}$$

or

$$\boxed{\left(\frac{p_x}{p_1}\right) = \left(\frac{p_2}{p_x}\right)}$$

**Open cycle Gas turbine with regeneration:**

In this method, the air delivered by the compressor passes through a heat exchanger utilizing the gases exhausted from the turbine. The heated air then passes into the combustion chamber and part of it is employed to burn the fuel. Since some heat is added already to the air in the heat exchanger itself, so the same turbine gas inlet temperature is achieved with lower fuel consumption. Hence the thermal efficiency is accordingly higher.

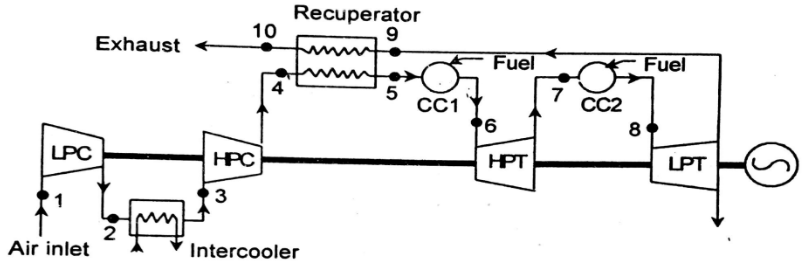



$$W_{net} = (h_3 - h_4) + (h_2 - h_1) = c_p(T_3 - T_4) + c_p(T_2 - T_1)$$

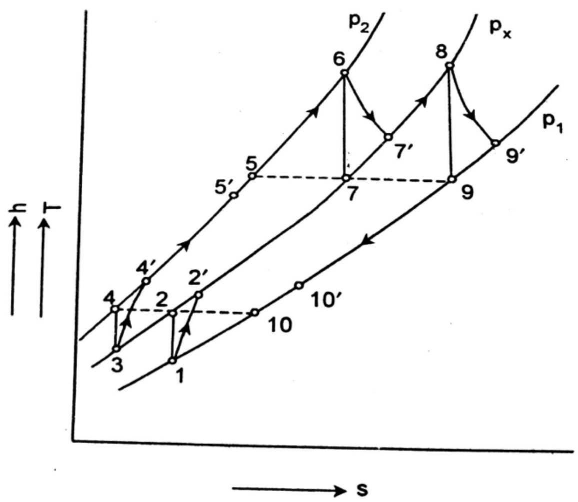
$$\text{Heat supplied} = h_3 - h_5 = c_p(T_3 - T_5)$$

$$\text{Thermal Efficiency} = \eta_{th} = \frac{c_p(T_3 - T_4) + c_p(T_2 - T_1)}{c_p(T_3 - T_5)}$$

**Open cycle gas turbine with Inter cooling, reheat and regeneration:**

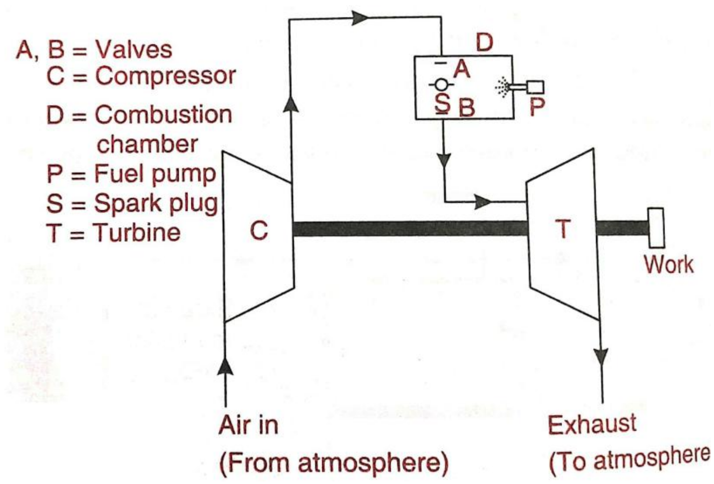



OGTC with Intercooling, Reheat and Regeneration. (Complete Cycle)




**Constant volume combustion gas turbine cycle**

In a constant volume combustion turbine, the compressed air from an air compressor C is admitted into the combustion chamber D through the valve A. when the valve A is closed, the fuel is admitted into the chamber by means of a fuel pump P. Then the mixture is ignited by means of a spark plug S. The combustion takes place at constant volume. The valve B opens and the hot gases flow to the turbine T, and finally they discharged, into atmosphere. The energy of the hot gases is thereby converted into mechanical energy. For continuous running of the turbine these operations are repeated.




**Jet Propulsion:**

The principle of jet propulsion involves imparting momentum to a mass of fluid in such a manner that the reaction of imparted momentum provides a propulsive force. It may be achieved by expanding the gas, which is at high temperture and pressure through a nozzle due to which a high velocity jet of hot gases is produced that gives a propulsive force.

The propulsion systems may be classified as

1. Air stream jet engines.

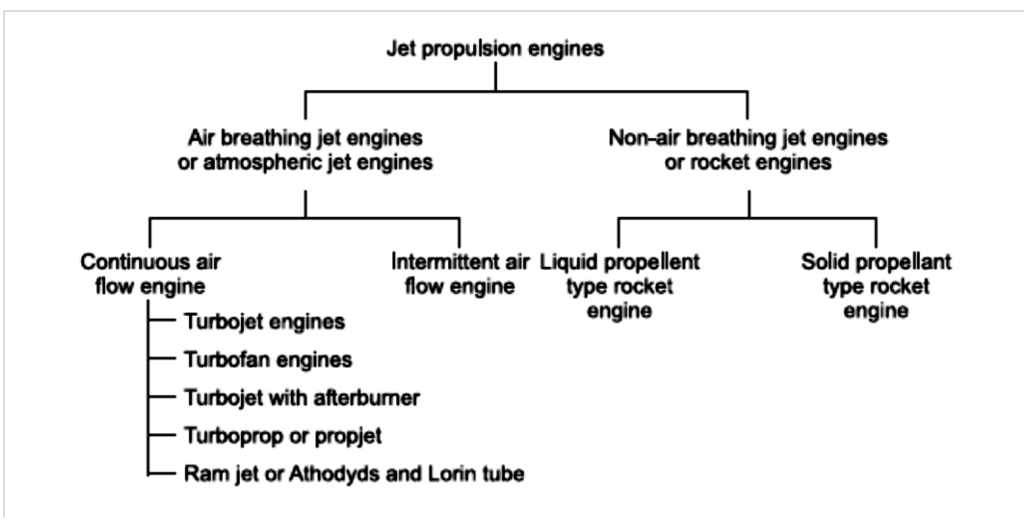
(a) Steady combustion systems; continuous air flow

(i) Turbo Jet (ii) Turbo-prop (iii) Ram jet

(b) Intermittent combustion systems; Intermittent flow

(i) Pulse jet or flying bomb.

2. Self- contained rocket Engines:



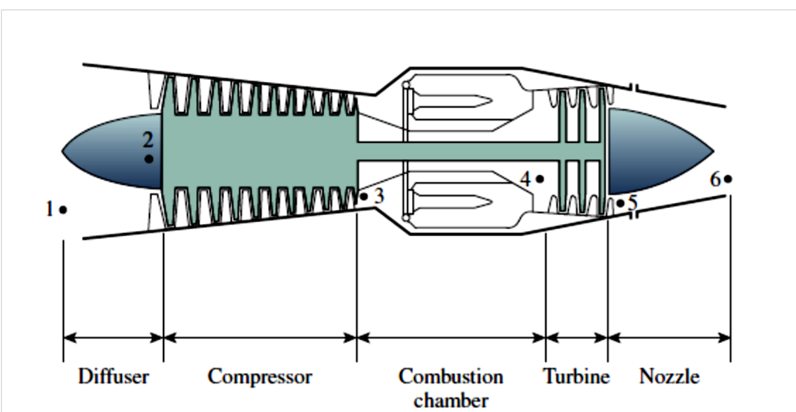

(i) Liquid

Propellent (ii) Solid Propellent

In air stream jet engines the oxygen necessary for the combustion is taken from the surrounding atmosphere where as in rocket engine the fuel and the oxidiser are contained in the body of the unit which is to be propelled.

(i) **Turbo Jet** :

- It consists of a diffuser at entrance which shows down the air and part of the kinetic energy of the air stream is converted into pressure. This type of compression is called as Ram compression.
- The air is further compressed to a pressure of 3 to 4 bar in a rotory compressor.
- The compressed air is then enters the combustion chamber where the fuel is added.
- The hot gases then enters the gas turbine where the partial expansion takes place. The power produced is just sufficient to drive the compressor.
- The exhaust gases from the gas turbine which are at higher pressure than atmosphere are expended in a nozzle and a very high velocity jet is produced which provides a forward motion to the air craft by the jet reaction.



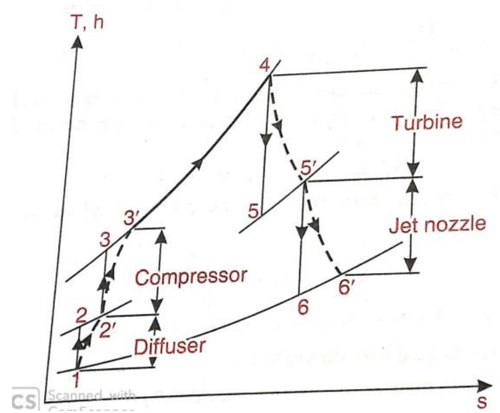




Fig. **Turbo jet**

Most suited to the air crafts travelling above 800km/h.

**Advantages:**

**1.** Construction is much simpler.
2. Engine vibrations absent.
3. Much higher speeds are possible.
4. Power supply is uninterrupted and smooth.
5. Weight to power ratio is superior.
6. Rate of climb is higher.
7. Radio intereference much less.
8. Smaller frontal area.

9. Fuel can be burnt over large range of mixture strength.

**Disadvantages:**

**1.** Less efficient.
2. Life of the unit is comparitively shorter.
3. The turbo jet becomes rapidly inefficient below 550km/h.
4. More noisy.
5. Materials required are quite expensive.
6. Required longer strip since length of take-off is too much.
7. At take-off the thrust is low, this effect is over come by boosting.

**Basic cycle for Turbo-Jet Engine:**

(i) Diffuser:

Applying Steady flow energy Equation:

$$\frac{c_a^2}{2}+h_1+Q_{1-2}=\frac{c_2^2}{2}+h_2+W_{1-2}$$ where $C_a(=C_1)$ = velocity of entering air from atmosphere.

In an Ideal diffuser $C_2=0, Q_{1-2}=0 \,\&\, W_{1-2}=0.$

Enthalpy at state 2 is $h_2=h_1+\frac{C_a^2}{2}$

$$T_2=T_1+\frac{C_a^2}{2C_p}$$

Diffuser efficiency $\eta_d=\frac{T_2-T_1}{T_2^1-T_1}$

**Compressor:**

$$W_c=h_3-h_2=c_p(T_3-T_2)$$

$$\eta_c=\frac{h_3-h_2}{h_3^1-h_2}$$

Combustion Chamber:

Ideal heat supplied per kg = $Q=h_4-h_3=c_p(T_4-T_3)$

Actual Heat supplied = $Q_a=c_{pg}(1+\frac{m_f}{m_a})h_4-h_3^1$

**Turbine:**

$$W_t=\left(h_4-h_5\right)=c_p(T_4-T_5)$$

Actual turbine work=$W_t=\left(h_4-h_5^1\right)=c_p(T_4-T_5^1)$

**Jet Nozzle:**

$$h_5^1 + \frac{c_5^{1^2}}{2} = \frac{c_6^{1^2}}{2} + h^1{}_6 + \cancelto{=0}{W_{1-2}}$$

$$h_5^1 = h_6^1 + \frac{c_6^{1^2}}{2}$$

**Thrust:**

Thrust is the force produced due to change of momentum.

Absolute velocity of gases leaving the air craft= $c_j - c_a$

Absolute velocity of air entering the air craft=0;

Mass of products leaving the nozzle per kg of air=$1+\frac{m_f}{m_a}$

$c_j$ is the velocity of jet relative to the exit nozzle.

Change in momentum = $\left(1+\frac{m_f}{m_a}\right)\left(c_j - c_a\right)$

hence Thrust = $T = \left(1+\frac{m_f}{m_a}\right)\left(c_j - c_a\right) N / kg.air / s$

**Thrust Power:**

It is defined as the rate at which work must be developed by the engine if the air craft is to be kept moving at a constant velocity $C_a$ against friction force or drag.

$\therefore$ Thrust Power = Forward thrust x Speed of the air craft.

T.P = $T = \left(1+\frac{m_f}{m_a}\right)\left(c_j - c_a\right)c_a$ W/kg of air.

$=\left(c_j - c_a\right)c_a$ W/kg of air if mass of fuel is neglected.

**Propulsive Power:**

The energy required to change the momentum of the mass flow of gas represents the propulsive power. It is expressed as the difference between the rate of kinetic energies of the entering air and exit gases.

$$P.P = \frac{\left(1+\frac{m_f}{m_a}\right)C_j^2}{2} - \frac{C_a^2}{2} W / kg$$

$=\frac{C_j^2}{2} - \frac{C_a^2}{2} W / kg$ neglecting the mass of the fuel.

**Propulsive efficiency:**

The ratio of thrust power to propulsive power is called propulsive efficiency.

$$\eta_{prop.} = \frac{Thrust\ power}{propulsive\ power} = \frac{\left(1+\frac{m_f}{m_a}\right)\left(c_j - c_a\right)c_a}{\frac{\left(1+\frac{m_f}{m_a}\right)C_j^2}{2} - \frac{C_a^2}{2}}$$

Neglecting the mass of the fuel,

$$\eta_{prop.} = \frac{2c_a}{c_j + c_a}$$

**Thermal Efficiency:**

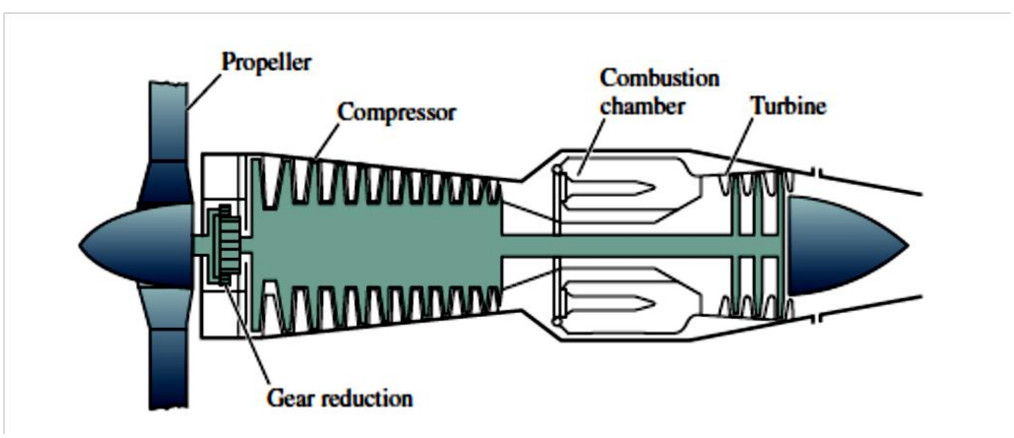
It is defined as the ratio of propulsive work and the energy released by combustion of the fuel.

$$\eta_{th.} = \frac{propulsive\ power}{heat\ released\ by\ \ combustion\ of\ fuel}$$

$$\eta_{th.} = \frac{\left(1+\frac{m_f}{m_a}\right)C_j^2 - C_a^2}{2\frac{m_f}{m_a} \times C.V}$$

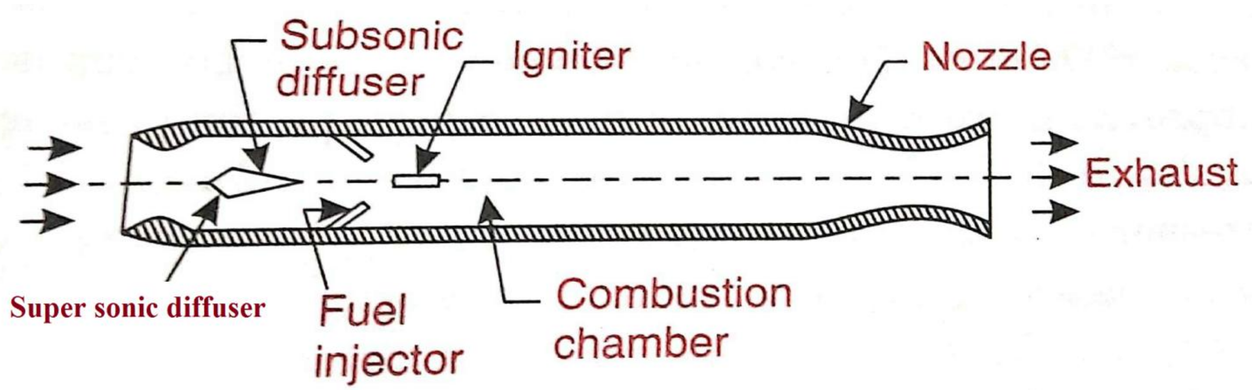
Overall efficiency is given by $\eta_o = \eta_{th.} * \eta_{prop.}$

## <u>Turbo-Prop:</u>



- Here the expansion of gases taken place partly in turbine (80%) and partly in nozzle(20%). The power developed by the turbine is consumed in running the compressor and the propeller. The propeller and jet produced by the nozzle give forward motion to the air craft.
- The turbo-prop entails the advantages of turbo jet and propeller power for take- off and high propulsion efficiency at speeds below 600kmph.
- The overall efficiency of the turbo-prop is improved by providing the diffuser before the compressor.

**RAM JET**




- Also called athodyd, lorin tube or flying stove. It has the capability to fly at supersonic speeds. Ramjet engine is the simplest of jet engines having no moving parts. Ramjet is a typically shaped duct open at both ends with air being compressed merely due to forward motion of engine. Fuel is subsequently added for combustion and thus high pressure, high temperature gases exit from exhaust nozzle.
- High pressure air is continuously available as engines keeps on moving forward. These ramjets are extensively used for propulsion in number of high speed aircrafts.
- The ram jet engine consists of a diffuser, combustion chamber and nozzle.
- The pressure rise in the engine is provided by the ram effect of the incoming high-speed air being rammed against a barrier. Therefore, a ramjet engine needs to be brought to a sufficiently high speed by an external source before it can be fired.
- The ramjet performs best in aircraft flying above Mach 2 or 3 (two or three times the speed of sound). In a ramjet, the air is slowed down to about Mach 0.2, fuel is added to the air and burned at this low velocity, and the combustion gases are expended and accelerated in a nozzle.

**Advantages of ramjet engine are:**
(*i*) It has no moving parts and hence ramjet are better balanced.
(*ii*) It yields greater thrust per unit mass as compared to any other propulsion engine at supersonic speed.
(*iii*) It is much simpler in construction and light in weight.
(*iv*) It yields much greater thrust per unit frontal area at supersonic speeds. Best performance can be had at 1700km/hr to 2200 km/hr speed range.
(*v*) Variety of fuels can be used in ramjet.
(*vi*) These are ideal propulsion device for aircraft missiles.

**Disadvantages of ramjet engine are:**

(*i*) Forward motion is very much necessary to realize ram compression.
(*ii*) Ram pressure ratio increases gradually.
(*iii*) Ramjets are unable to work at low flight speeds.

## PULSE JET ENGINE

- It is quite similar to ramjet engine except the difference that pulse jet employs a non-return type mechanical valve of V-type for preventing flow of hot gases through diffuser. Pulse jet engine has diffuser section in which ram compression occurs and after diffuser section a grid of non-return valves is put for maintaining intermittent flow of compressed air.
- In combustion section fuel is atomized during injection and burnt using spark plug or igniters. Combustion of air and fuel results in combustion products having high pressure and temperature. Due to high pressure of combustion product non-return valve remains closed and causes flow through tail pipe so as to produce thrust.
- When combustion gases move out from combustion chamber then the pressure lowers down in this side as compared to pressure available in diffuser section. Due to this pressure differential the flow of compressed air again occurs into the combustion chamber which again burns and flows out from exit nozzle. Thus there occur the processes of suction, combustion and exhaust one after the other.
- Pulse jet engine tube shown in Fig. has temperature varying continuously from inlet to exit. This pulse jet engine has advantages of being cheap compared to turbojet engine and produces static thrust. Pulse jet engine produces thrust more than drag at lower speed compared to ramjet. Pulse jet engines have disadvantages of noise, maintenance in view of mechanical operation of valves, high fuel consumption rate and vibration etc.

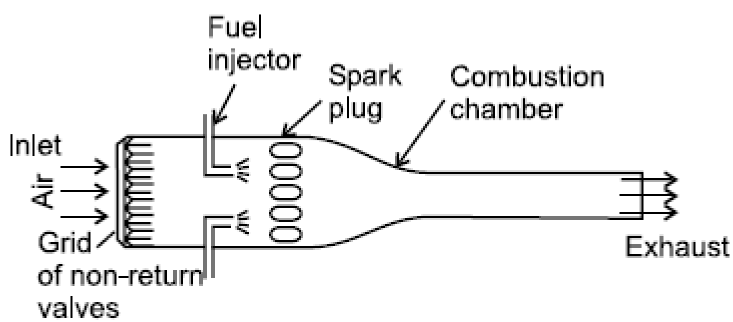



***Fig.** Pulse jet engine*

## PRINCIPLE OF ROCKET PROPULSION

- Rocket engines are non-air breathing engines and carry their own oxidiser for burning of fuel. Rocket propulsion is realized by the thrust produced by combustion products leaving exit nozzle. It has injection system for fuel and oxidizer followed by combustion chamber and exit nozzle as shown in Fig.

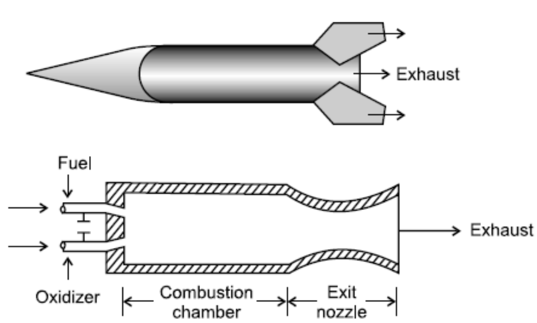



***Fig.** Layout for rocket engine*

- In rocket engines the combustion products get discharged from the exit nozzle with supersonic velocity and thus have very high kinetic energy. Rocket gets desired thrust by the reaction available from the nozzle stream. Thrust is available due to change of momentum and pressure with which jet comes out.

  Net thrust $T = m_p^{'} C_e + A_e \left( p_e - p_a \right)$

  where is $m_p^{'}$ is the mass flow rate of propellant, jet exit velocity $C_e$, area of exit nozzle $A_e$, pressure of exit jet $p_e$ and atmospheric pressure is $p_a$.

- Above expression shows that for maximizing thrust exit velocity should be maximized, pressure difference at exit ($p_e - p_a$) should be maximized. Thus rocket would get maximum thrust when atmospheric pressure is not there *i.e.*, $p_a = 0$, which means maximum thrust would be available in vacuum. Thrust could also be given in terms of rocket performance parameter called effective jet velocity.

  $$T = m_p^{'} C_e + A_e \left( p_e - p_a \right) = m_p^{'} C_{ej}$$

  Here $C_{ej}$ is effective jet velocity which could be given as, $C_{ej} = C_e + \frac{A_e}{m_p^{'}} \left( p_e - p_a \right)$

  Specific impulse can be given as $I_{sp} = \frac{T}{m_p^{'}} = C_{ej}$

  i.e., Specific impulse is the thrust produced per unit mass flow rate of propellant. **Thrust power** in case of rocket engine can be given as;

  $TP = T.C_a = m_p^{'} C_{ej} C_a$ , here $C_a$ is the velocity with which rocket moves forward.

  **Propulsive efficiency:** For rocket engine

  $$\eta_{prop} = \frac{TP}{TP + Lossofkineticenergy}$$

  $$\eta_{prop} = \frac{2C_a / C_{ej}}{1 + \left( C_a / C_{ej} \right)^2}$$

# Learning Material

## 1. Necessity of Refrigeration

i) Perishable food items such as vegetables, fruits, beverages, poultry products, etc. can be preserved for longer duration at low temperatures without losing taste, since the bacteria cannot sustain at low temperatures.

ii) Development of certain scientific equipment and their operation under controlled environment needs proper refrigeration and air conditioning.

iii) Industries such as spinning mills require maintaining proper temperature and humidity to avoid wastage of threads due to breakage.

iv) Photographic materials show excellent prints when the environment is maintained at required temperature.

v) Air conditioning of work spaces such as workshops, offices, etc. increases the efficiency of workers.

vi) Air conditioning of theatres, commercial shops, etc. attracts the customers and improves the business.

vii) Air conditioning of operation theatres, scanning centers, etc. is necessary for proper functioning of equipment.

viii) Finally, air conditioning is used to provide the comfort to human beings in all seasons.

All the above things require refrigeration for maintaining required temperature and humidity.

## 2. Applications of Refrigeration

i) Ice manufacturing

ii) Cooling and storage of perishable food, drinks and medicines.

iii) Food processing, preservation and distribution.

iv) Comfort air-conditioning of hospitals, hotels, theatres and residences etc.

v) Air conditioning of industries such as textiles, printing, manufacturing, photographic, etc.

vi) Air conditioning of computer centers, CNC Machines, etc.

vii) Liquefying gases and vapours in chemical and pharmaceutical industries.

viii) Manufacturing and treatments of metals.

ix) Oil refining and manufacturing of synthetic rubber.

x) Cooling of concrete in big buildings and dams.

The refrigeration has also wide applications in rockets, aircrafts and sub-marine ships.

## 3. Unit of Refrigeration

The unit of refrigeration is ton refrigeration and is denoted by TR.

It is equivalent to the production of cold at the rate at which heat is to be removed from 1 tonne (1000 kg) of water at $0^oC$ to freeze it to ice at $0^o$ C in one day or 24 hours. Thus,

$$1\,TR = \frac{1000\,kg\ X\ 80\,k\,cal\,/\,kg}{24\,hr\ X\ 60\,\min/\,hr} = 55.55\,kcal/min$$

$$\approx 50\,\frac{kcal}{min} \text{ or } 210 \text{ kJ/min or } 3.5 \text{ kW.}$$

**Note**: latent heat of fusion of ice = 80 kcal/kg.

- A refrigerator or an air-conditioner of 1 TR can extract 210 kJ of heat in a minute or 3.5 kJ of heat in a second from a low temperature space.

### 5. Coefficient of Performance (COP)

- It is defined as the ratio of desired effect to the work input.
- The desired effect is the refrigerating effect or cooling effect produced at a low temperature space in the case of refrigerator. Therefore,

$$(COP)_R = \frac{Refrigerating\ effect}{Work\ input} = \frac{q_0}{w}$$

- The desired effect is the heating effect or the quantity of heat supplied to a high temperature space in the case of heat pump. Therefore,

$$(COP)_{HP} = \frac{Heating\ effect}{Work\ input} = \frac{q_k}{w} = \frac{q_0+w}{w} = \frac{q_0}{w} + 1$$

$$= (COP)_R + 1$$

- Therefore, the COP of a heat pump is one greater than the COP of a refrigerator.
- Since the COP of a heat pump is always greater than unity, running the heat pump is quite economical over electric resistance heater for heating applications.

### 6. Types of Refrigeration Systems

i) Air cycle refrigeration system

ii) Vapour compression refrigeration system

iii) Vapour absorption refrigeration system

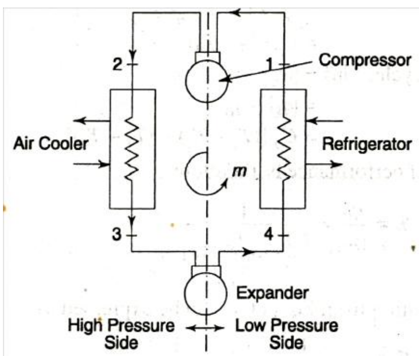
iv) Steam Jet refrigeration system

v) Thermoelectric refrigeration system

## 7. Air Cycle Refrigeration

### 7.1 Air Cycle Refrigeration System

The schematic diagram of Air Cycle refrigeration system is shown in figure . The air cycle refrigeration system works on Bell Coleman or reversed Brayton cycle. The air is compressed in the compressor during the isentropic compression process 1-2. The compressed air is cooled at constant pressure in the cooler during the process 2-3, wherein it rejects heat to the coolant. The compressed and cooled air is expanded in the expander during the isentropic expansion process 3-4 so that the air leaves the expander at a low temperature. The chilled air at state point 4 passes through the refrigerator and extracts heat during the constant pressure heating process 4-1 from the cooled space.



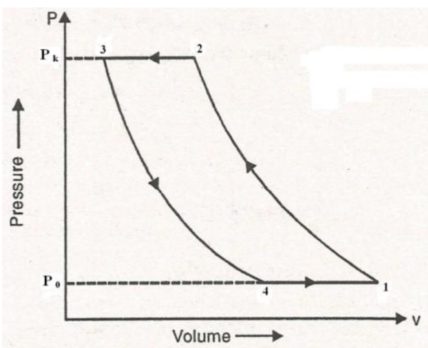
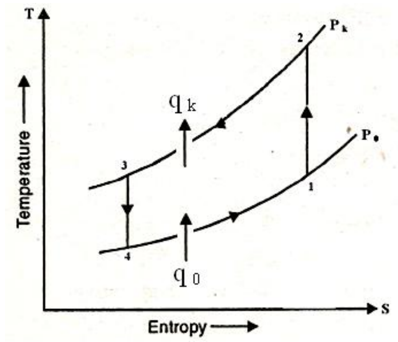



***Figure : Bell Coleman Cycle***




The Bell Coleman cycle consists of the following four processes as shown in figure :

Process 1-2: Isentropic compression of air in the compressor from low pressure to high pressure

Process 2-3: Cooling of air at constant pressure in the cooler

Process 3-4: Isentropic expansion of air in the expander from high pressure to low pressure

Process 4-1: Heating of air (extraction of heat from low temperature space) at constant pressure in the refrigerator

- Comparison of reversed Carnot cycle and reversed Brayton cycle is given in figure. From figure , it is observed that the average temperature of heat extraction is low and the average temperature of heat rejection is high in reversed Brayton cycle as compared to reversed Carnot cycle. Hence, the COP of reversed Brayton cycle is much lower as compared to that of Carnot cycle.

### 7.1.1 Analysis of Air Cycle Refrigeration

The relationship between the various temperatures of the cycle is given by the isentropic relations applied to both compression and expansion processes, viz.,

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = \left(\frac{p_k}{p_0}\right)^{\frac{\gamma-1}{\gamma}}$$

Also, assuming air to be a perfect gas, we have per unit mass of air circulated:

Refrigerating effect, $q_0 = C_p(T_1 - T_4)$

Heat rejected, $q_k = C_p(T_2 - T_3)$

Compressor work, $w_c = \frac{\gamma}{\gamma-1}(p_2v_2 - p_1v_1)$

$= C_p(T_2 - T_1)$

Expander work, $w_E = \frac{\gamma}{\gamma-1}(p_3v_3 - p_4v_4)$

$= C_p(T_3 - T_4)$

Net work of the cycle, $w = w_c - w_E$

$$= q_k - q_0$$

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

The coefficient of performance is, therefore,

$$(COP)_R = \frac{q_0}{w} = \frac{1}{\left(\frac{T_2 - T_3}{T_1 - T_4}\right) - 1}$$

$$(COP)_R = \frac{1}{\left(\frac{p_k}{p_0}\right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{r_p^{\frac{\gamma-1}{\gamma}} - 1}$$

### 7.2.2 Polytropic and Multistage Compression

The polytropic compression with cooling would reduce the net work of the cycle by reducing the average temperature of the compression process and the value of the compression index from $\gamma$ to n. Then, the expression for compressor work becomes

$$w_c = \frac{n}{n-1}(p_2 v_2 - p_1 v_1)$$

$$= \frac{n}{n-1}\frac{\gamma-1}{\gamma} C_p (T_2 - T_1)$$

$$w_e = \frac{n}{n-1}(p_3 v_3 - p_4 v_4)$$

$$= \frac{n}{n-1}\frac{\gamma-1}{\gamma} C_p (T_3 - T_4)$$

The net work is

$$w = w_c - w_e$$

$$= C_p\left[\frac{n}{n-1}\frac{\gamma-1}{\gamma}(T_2 - T_1) - (T_3 - T_4)\right]$$ and the COP is

$$(COP)_R = \frac{T_1 - T_4}{\frac{n}{n-1}\frac{\gamma-1}{\gamma}(T_2 - T_1) - (T_3 - T_4)}$$

# <u>Vapour Compression Refrigeration System</u>

## 7.2 Vapour Compression Refrigeration System

The vapour compressor refrigeration system, as shown in figure, consists of a compressor, a condenser, an expansion device and an evaporator.

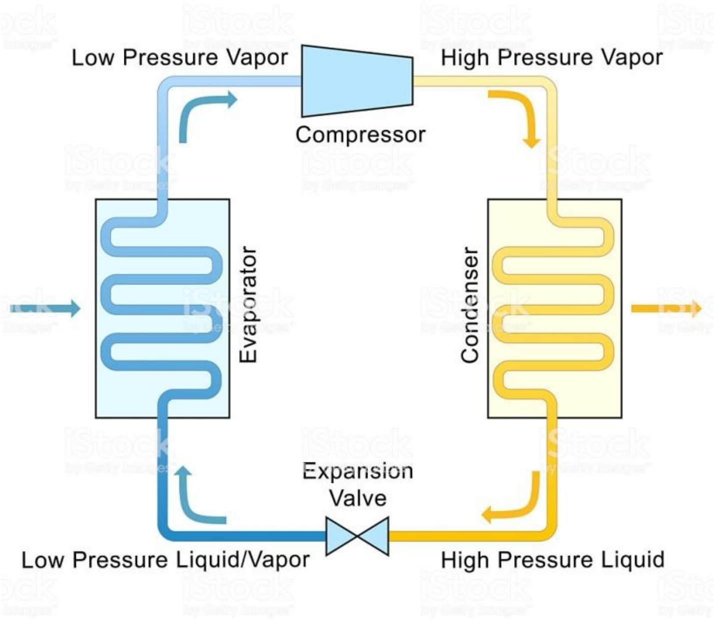
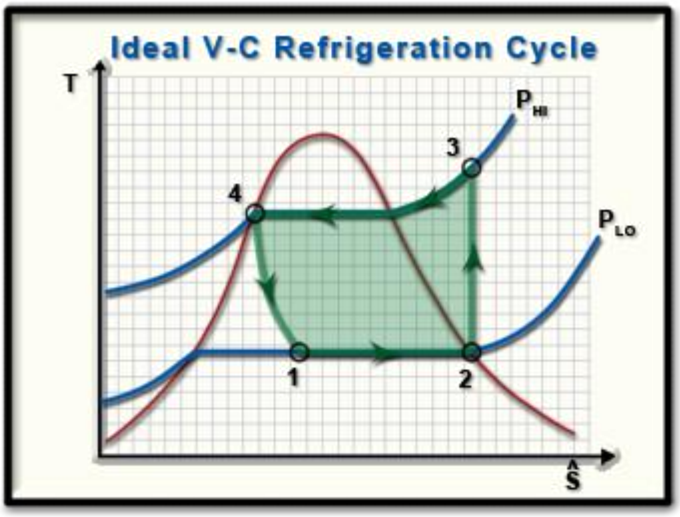


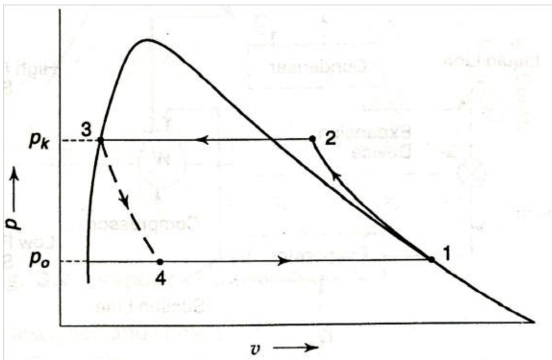

***Figure : Schematic Diagram of Vapour Compression Refrigeration System***

The refrigerant vapour formed in the evaporator at low pressure are continuously sucked by the compressor and compressed and discharged into the high pressure condenser. The refrigerant vapour is condensed in the condenser by the rejecting heat to the coolant circulated through it. The liquid refrigerant is throttled in the expansion device from condenser pressure to evaporator pressure. Upon throttling, some liquid flashes in to vapour and temperature is decreased. The liquid vapour mixture of refrigerant flows through the evaporator coil and absorbs heat energy from the refrigerated space at low temperature and becomes vapour.

### 7.2.1 Vapour Compression cycle

The cycle with dry compression of vapour and throttling of liquid is named as the vapour compression cycle and because of its high coefficient of performance it is most widely used in commercial refrigeration systems. A complete vapour compression cycle is shown on the T-s, p-v,and on the p-h diagrams.



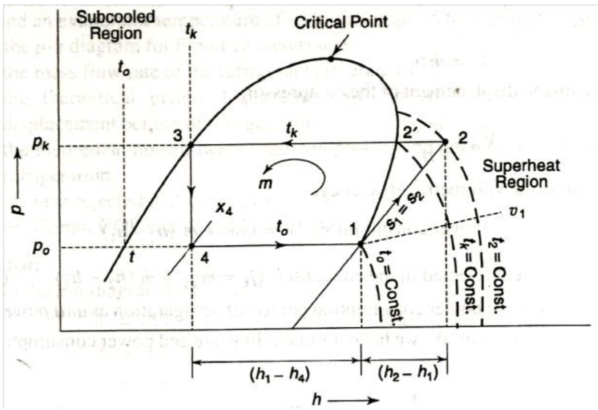




***Figure: Vapour Compression Cycle on p – h diagram***

### 7.2.2 Analysis of vapour compression cycle:

Refrigerating effect, $q_0 = area\ 1 - 4 - d - e$

$= h_1 - h_4$

Heat rejected, $q_k = area\ 2 - 2' - 3 - c - e$

$= h_2 - h_3$

Work input to the compressor, $w = q_k - q_0$

$= h_2 - h_1$

$(COP)_{Refrigerator} = \frac{Refrigerating\ effect}{work\ input} = \frac{q_0}{w} = \frac{h_1-h_4}{h_2-h_1}$

## 7.3 Vapour Absorption System

### 7.3.1 Simple Vapour Aqua Ammonia Vapour Absorption Refrigeration System

In the vapour-absorption system, the function of the compressor is accomplished in a three-step process by the use of the absorber, pump and generator or re-boiler as follows:

(i) Absorber: Absorption of the refrigerant vapour by its weak or poor solution in a suitable absorbent or adsorbent, forming a strong or rich solution of the refrigerant in the absorbent / adsorbent.

(ii) Pump: Pumping of the rich solution raising its pressure to the condenser pressure.

(iii) Generator: Distillation of the vapour from the rich solution leaving the poor solution or recycling.

A simple vapour-absorption system, therefore, consists of a condenser, an expansion device and an evaporator as in the vapour-compression system, and in addition, an absorber, a pump, a generator and a pressure-reducing valve to replace the compressor. The schematic representation of the system is shown in figure in which various components of the system are arranged according to their pressures and temperatures.

The refrigerating effect is shown as $Q_0$ at temperature $T_0$ and the heat rejected in the condenser as $Q_c$ at temperature $T_c$=$T_k$ of the environment. The compressor work is replaced by the heat supplied in the generator $Q_h$ plus pump work $Q_p$. Cooling must be done in the absorber to remove the latent heat of the refrigerant vapour as its changes into the liquid state by absorption by the weak solution. Let this heat rejected in the absorber be $Q_A$ at absorber temperature $T_A$=$T_k$. Then the energy balance of the system is

$$Q_0 + Q_p + Q_h = Q_c + Q_A = Q_k$$

Where $Q_k$ represents the net heat rejected to the environment

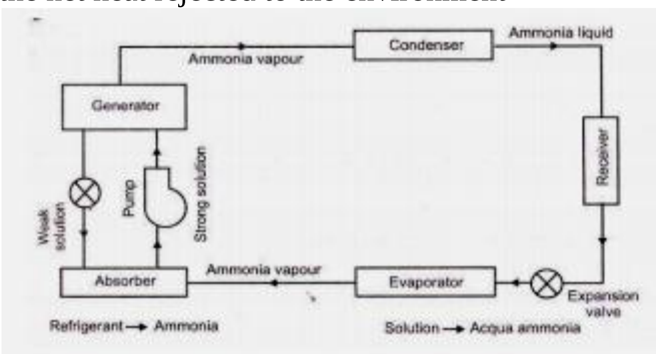



***Figure: Schematic Representation of Simple Aqua Ammonia Vapour Absorption System***

The pump work $Q_p = -\int vd\,p$, is very small compared to the compressor work in the vapour compression system, as the specific volume $v$ of the liquid is extremely small compared to that of the vapour $\left(v_f \ll v_g\right)$. The energy consumption of the system is mainly in the generator in the form of heat supplied $Q_h$.

The overall coefficient of performance is expressed as

$$\zeta = \frac{Refrigerating\ effect}{Energy\ \mathrm{supplied}}$$

$$= \frac{Q_0}{Q_h+Q_p} = \frac{Q_0}{Q_k-Q_0}$$

$\cong Q_0/Q_h$ neglecting pump work which is small compared to $Q_k$.

### 7.3.2 Desirable characteristics of Refrigerant – Absorbent Solutions:

Some of the desirable characteristics of a refrigerant-absorbent pair for an absorption system are

- low viscosity to minimize pump work,
- low freezing point and
- good chemical and thermal stability.

In addition to the above, two main thermodynamic requirements of the mixture are:

i) Solubility requirement: A strong solution, highly rich in the refrigerant, is formed in the absorber by the absorption of the refrigerant vapour.

ii) Boiling points requirement: There should be a large difference in the normal boiling points of the two substances, at least 200°C, so that the absorbent exerts negligible vapour pressure at the generator temperature. Thus, almost absorbent-free refrigerant is boiled off from the generator and the absorbent alone returns to the absorber.

### 7.3.4 Common Refrigerant – Absorbent Systems

The two commonly used pairs of refrigerant – absorbent systems are;

i) Refrigerant $NH_3$ and absorbent $H_2O$,

ii) Refrigerant $H_2O$ and adsorbent $LiBr_2$.

- In the ammonia-water system, ammonia is the refrigerant and water is the absorbent. From the point of view of the solubility requirement, it is satisfactory. But the difference in their boiling points is only 138°C. Hence the vapor leaving the generator contains some amount of water vapor.
- Thus, the ammonia-water system is not suitable from the point of view of the boiling point requirement.
- In the water-lithium bromide system, water is the refrigerant and lithium bromide is the adsorbent. Hence the mixture is used only in air-conditioning applications since water freezes at 0°C.
- The mixture is satisfactory from the point of view of the solubility requirement. Since lithium bromide is a salt, it exerts no vapour pressure. So the vapour leaving the generator is a pure refrigerant. The mixture, therefore satisfies the boiling point requirement also.
- However, it is corrosive and the plant works under high vacuum, both in condenser and evaporator. Hence, a purge unit is used.

### 7.3.5 Practical (Two Shell) Water-Lithium Bromide Vapour Absorption Chiller

A two shell water - Lithium Bromide absorption system is shown in figure. The generator and condenser are housed in the single-cylindrical vessel 1 and the flash evaporator and absorber in another similar vessel 6. Water is boiled off from vessel 1 by the steam coils 4. The vapour is condensed in the condenser over the cooling water coils 2 and collected in the tray 3. The condensate is flashed through expansion valve 5 into the vessel 6. The refrigerant water is collected in the tray 8. Chilled water is circulated by the pump 12 and is returned to the system at 13.

The strong brine from vessel 1 flows by gravity through the heat exchanger 7 and the pressure reducing valve 9 to the vessel 6. The flashed water vapour filling the space in 6 is absorbed by this solution. The absorber heat is removed by the cooling coils 10. Again, there is separate cooling water line for the absorber. The weak salt solution leaving the absorber is then returned to the generator by the pump 11 through the liquid-liquid regenerative heat exchanger 7.

The chilled water, used as a secondary refrigerant, and refrigerant water are kept separate.

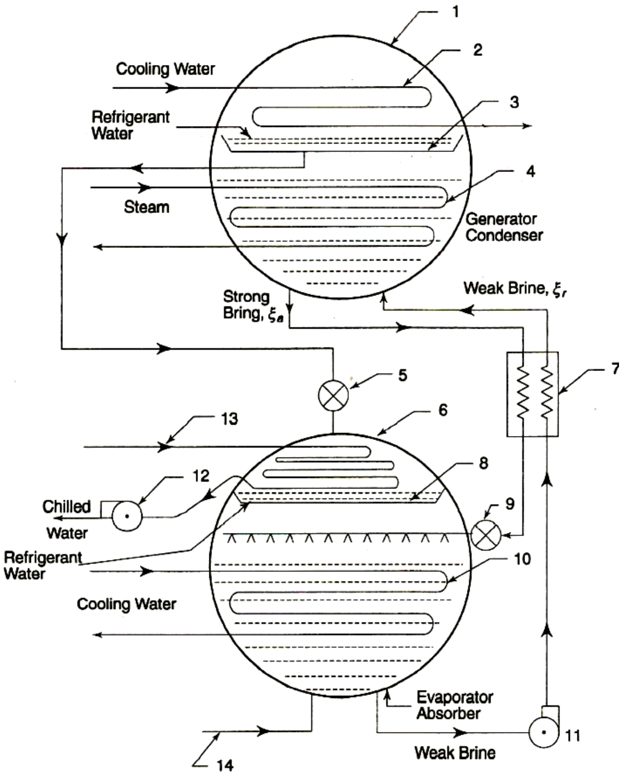



***Figure :Single-Effect Water-Lithium Bromide Absorption Chiller***

Both the vessels 1 and 6 are maintained under high vacuum, vessel 1 corresponding to the condensing temperature (e.g. 55.3 mm Hg pressure at 40$^o$C) and vessel corresponding to the flashed refrigerant water temperature (e.g. 4.9 mm Hg pressure at 1$^o$C). To remove air and other non-condensable gasses that may enter the sealed system through pump glands, a two-stage purge unit is provided. To avoid corrosion, the temperature in the boiler should not be higher than 120$^o$C. The overall COP of the system is reported to be approximately 0.7. The lithium bromide-water system is thus found to be more suitable in applications involving low heat-source temperatures such as are obtained with low-pressure (1 to 8 bar) or even exhaust (say 0.4 bar) steam, waste heat, solar energy, etc. Absorption chillers are available in capacities from 100 TR upwards up to 7500 TR.

### 7.3.6 Comparison of vapour absorption and vapour compression refrigeration systems

| Sl.No. | Vapour Absorption System | Vapour Compression System |
|---|---|---|
| 1 | The compressor is replaced with absorber, solution pump and generator. | Compressor is the heart of the system |
| 2 | It is primarily heat operated machine and hence utilizes low grade heat energy. | It requires high grade work energy as input. |
| 3 | The Coefficient of Performance is low. | The Coefficient of Performance is high |
| 4 | The performance of the system does not get affected by load variation due to good part load characteristics. | Poor part load characteristics and the performance decreases with decrease in load. |

## 8.0 Psychrometry

The subject which deals with the behavior of moist air is known as psychrometry

### 8.1.1 Working Substance in Air Conditioning

Moist Air is a mixture to two gases. One of these is dry air which itself is a mixture of a number of gases and the other is water vapour which may exist in a saturated or superheated state.

Thus, moist air consists of two parts: one, comprising dry air, considered as the fixed part, and the other, solely of water vapour, considered as the variable part.

Both dry air and water vapour can be considered as perfect gases since both exist in the atmosphere at low pressures. Hence, perfect gas laws can be applied to them individually.

### 8.1.2 Dalton's Law of Partial Pressures

Consider a homogeneous mixture of non-reaching ideal gases 1, 2, etc. at temperature *T*, pressure *p* and occupying volume *V* as shown in figure. Let the number of moles of individual gases be $n_1$, $n_2$, ...etc., and their respective masses be $m_1$, $m_2$,.....*etc*. Then we have for total number of moles *n* and total mass *m*

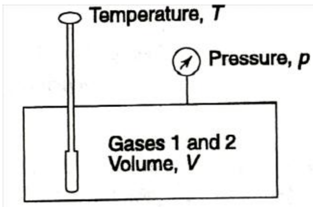



***Figure Gas Mixture***

In the Dalton's model, each gas is conceived of as existing separately at the temperature T and total volume V of the mixture as show in figure .

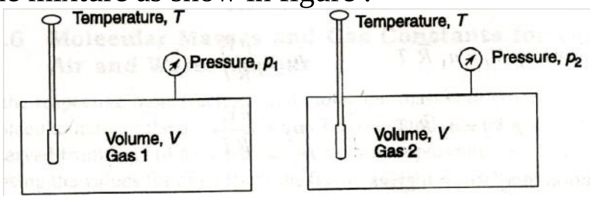


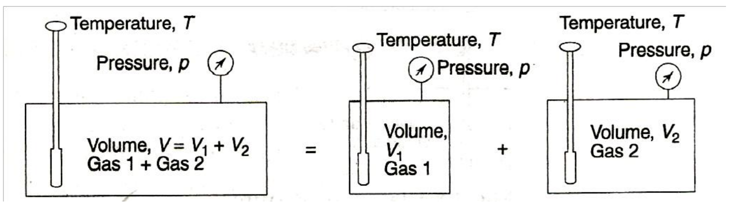

***Figure: Figure Illustrating Dalton's Model***

If one were to measure the pressures exerted by individual gases, they would be found to be $p_1$, $p_2$,....etc., viz., less than the total pressure *p* of the mixture. There are referred to as partial pressures.

**Thus, for a mixture of ideal gases, the total pressure p is equal to the sum of the partial pressures. This is known as the Dalton's law of partial pressures.**

### 8.1.3 Amagat Law of Partial Volumes

In the Amagar model, each component gas is considered as existing separately at the total pressure p and temperature T of the mixture as shown in figure.



Let the volumes of individual gases under these conditions be $V_1$, $V_2$,.....etc. These are referred to as partial volumes.

**The total volume is equal to the sum of the partial volumes. This is known as the Amagat Law of partial volumes.**

The ratios $V_1/V$, $V_2/V$, ..... etc. are referred to as volume fraction.

$$V = V_1 + V_2 \text{ or } V = \Sigma_{\text{i}}\, V_i$$

### 8.1.4 Mole Fractions of Component Gases

It is seen from the above section that

$$\frac{p_1}{p} = \frac{V_1}{V} = \frac{n_1}{n} = y_1$$

**Thus, the ratio of partial pressure to total pressure, and volume fraction are equal to the mole fraction $y_{\text{i}}$ of the gas. It also shows that both Dalton's law and Amagat law are equivalent.**

### 8.1.5 Molecular Mass of Mixture

Since $m = m_1 + m_2$, and $m = Mn, m_1 = M_1\, n_1, m_2 = M_2 n_2$, we have

$$Mn = M_1 n_1 + m_2\, n_2$$

Thus $M = y_1 M_1 + y_2 M_2$ or $M = \Sigma_{\text{i}}\, y_i\, M_i$

Where, M represents the molecular mass of the mixture. Note that $n_1 = \frac{m_1}{M_1}, n_2 = m_2/M_2$ etc. Similarly, for the mixture $n=m/M$.

### 8.1.6 Psychrometric Properties

The properties of moist air are called psychrometric properties.

#### 8.1.7.1 Specific Humidity or Humidity Ratio

- Specific or absolute humidity or humidity ratio or moisture content denoted by the symbol 'w' is defined as the ratio of the mass of water vapour (w.v.) to the mass of dry air (d.a.) in a given volume of the mixture.

Let $m_v$ and $m_a$ are the masses of water vapour and dry air present in a certain volume V of moist air at pressure P and temperature T.

Thus

$$w = \frac{m_v}{m_a} = \frac{V/v_v}{V/v_a} = \frac{v_a}{v_v}$$

Where the subscripts $a$ and $v$ refer to dry air and water vapour respectively.

Now
$$p_a v_a = \frac{\bar{R}}{M_a} T \qquad p_a V = m_a \frac{\bar{R}}{M_a} T$$

$$p_a v_a = \frac{\bar{R}}{M_v} T \qquad p_v V = m_v \frac{\bar{R}}{M_v} T$$

$$w = \frac{M_v\, p_v}{M_a\, p_a} = \frac{18.016}{28.966} \frac{p_v}{p_a} = 0.622\ \frac{p_v}{p_a}$$

The units of $w$ are kg of water vapour per kg of dry air.

Also, since $p$ denotes the actual total atmospheric pressure, then from Dalton's law

$$p = p_a + p_v$$

so that $$w = 0.622\frac{p_v}{p - p_v}$$

Considering that the total atmospheric pressure remains constant at a particular locality, we can see that

$$w = f(p_v)$$

- Thus, the specific humidity is a function of the partial pressure of water vapour only.

It may be noted that since $p_v$ is very small compared to barometric pressure $p$, the denominator in Eq remains more or less constant, i.e.,

$$p_a = p - p_v \approx p$$

- Hence, $w$ is approximately a linear function of $p_v$.

### 8.1.7.2 Dew Point Temperature

The water vapour existing at temperature $T$ of the mixture and partial pressure $p_v$ of the vapour in the mixture is normally in a superheated state. Moist air containing moisture in such a state is considered as unsaturated air.

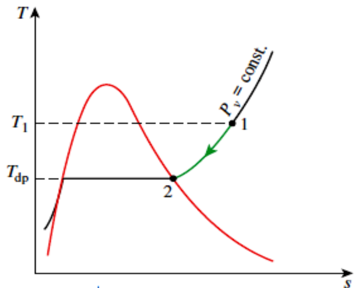



**Figure:2**

Constant-presssure cooling of moist air and the dew-point temperature on the *T-s* diagram of water.

***Figure: Thermodynamic State of Water Vapour in Moist Air***

If a sample of such unsaturated moist air containing superheated water vapour is cooled (at constant pressure), the mixture will eventually reach the saturation temperature $t_d$ of water vapour corresponding to its partial pressure $p_v$, at which point the first drop of dew will be formed, i.e., the water vapour in the mixture will start condensing. This temperature $t_d$ is called the dew point temperature (DPT).

- Dew Point Temperature is the temperature to which moist air must be cooled at constant pressure before condensation of moisture takes place.
- Moisture can be removed from humid air by bringing the air in contact with a cold surface or cooling coil whose temperature is below its dew point temperature.
- It is seen that the dew point temperature can be found by knowing, from the steam tables, the saturation temperature $t_d$ at the partial pressure $p_v$ of the water vapour.

#### 8.1.7.3 Degree of Saturation

Figure . again shows the superheated thermodynamic state 1 of water vapour in unsaturated moist air representing the control volume V in figure. The water vapour exists at the dry bulb temperature T of the mixture and partial pressure $p_v$.

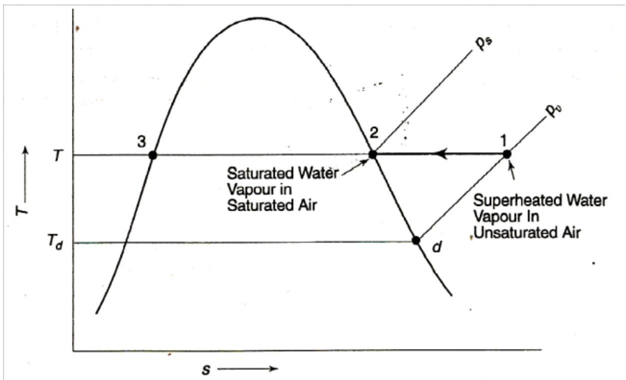



***Figure: An Imaginary Isothermal Process Representing Change of State of Water Vapour in Unsaturated Air to that of Saturated Air at the Same Temperature***

Consider now that more water vapour is added in this control volume V at temperature T itself. The partial pressure $p_v$ will go on increasing due to the addition of water vapour until it reaches a value $p_s$ corresponding to state 2 in figure , after which it cannot increase further as $p_s$ is the saturation pressure or maximum possible pressure of water at temperature T. The thermodynamic state of water vapour is now saturated at point 2. The air containing moisture in such a state is called saturated air. In this state the air is holding the maximum amount of water vapour at temperature T of the mixture. The maximum possible specific humidity, $w_s$ at temperature T is thus

$$w_s = 0.622 \frac{p_s}{p - p_s}$$

- The ratio of the actual specific humidity $w$ to the specific humidity $w_s$ of saturated air at temperature T is termed as the degree of saturation denoted by the symbol $\mu$. Thus

$$\mu = \frac{w}{w_s} = \frac{p_v}{p_s}\left[\frac{1 - p_s/p}{1 - p_v/p}\right]$$

- The degree of saturation is a measure of the capacity of air to absorb moisture.

#### 8.1.7.4 Relative Humidity

- Relative humidity denoted by the symbol $\phi$ or RH is defined as the ratio of the mass of water vapour $m_v$ in a certain volume of moist air at a given temperature to the mass of water vapour $m_{v_s}$ in the same volume of saturated air at the same temperature.

Thus, referring to Figure again, if $v_v$ and $v_s$ are the specific volumes of water vapour in the actual moist air and saturated air respectively at temperature T and in volume V, viz., at points 1 and 2 respectively, we see that

$$\phi = \frac{m_v}{m_{v_s}} = \frac{p_v V/\bar{R}T}{p_s V/\bar{R}T} = \frac{p_v}{p_s}$$

Also, $$\phi = \frac{V/v_v}{V/v_s} = \frac{v_s}{v_v}$$

Using the perfect-gas relationship between points 1 and 2, viz.,

$$p_1 v_1 = p_2 v_2 \text{ or } p_v v_v = p_s v_s$$

we have $$\phi = \frac{p_v}{p_s} = \frac{v_s}{v_v}$$

- Thus, relative humidity turns out to be ratio of partial pressures of water vapour in a certain unsaturated moist air at a given temperature T to the partial pressure of water vapour in saturated air at the same temperature T. It is usually measured in percentage.
- When $p_v$ is equal to $p_s$, $\phi$ is equal to unity, and the air is saturated and is considered to have 100 percent RH.

$$w = 0.622\phi \frac{p_s}{p_a}$$

$$\phi = \frac{w}{0.622}\frac{p_a}{p_s}$$

$$\mu = \phi\left[\frac{1-p_s/p}{1-p_v/p}\right]$$

$$\phi = \frac{\mu}{1-(1-\mu)p_s/p}$$

### 8.1.7.5 Enthalpy of Moist Air

According to Gibb's law, the enthalpy of a mixture of perfect gases is obtained by the summation of the enthalpies of the constituents. Thus the enthalpy of moist air h is equal to the sum of the enthalpies of dry air and associated water vapour, i.e.,

$$h = h_a + wh_v$$

per kg of dry air, where $h_a$ is the enthalpy of the dry air part and $wh_v$ is the enthalpy of the water vapour part.

Considering the change in enthalpy of a perfect gas as a function of temperature only, the enthalpy of the dry air part, above a datum of 0°C, is expressed as $h_a = c_{p_a}\, t = 1.005\, t$ kJ/kg.

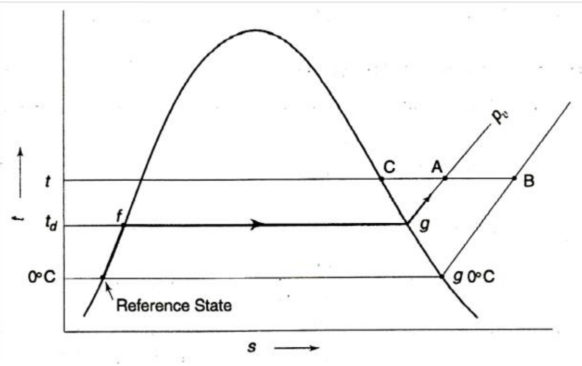



*Figure: Evaluation of Enthalpy of Water Vapour Part*

At low pressures for an ideal gas, the enthalpy is a function of temperature only.

$$h_A = h_B = (h_g)_{0^oC} + C_{p_v}(t-0)$$

The latent heat of vaporization of water at 0°C is 2500 kJ/kg.

Thus the enthalpy of moist air

$$h = 1.005t + w(2500 + 1.88t) \text{ kJ/kg d.a.}$$

#### 8.1.7.6 Dry Bulb Temperature (DBT) and Wet Bulb Temperature (WBT)

A psychrometer comprises of a dry bulb thermometer and a wet bulb thermometer.

- The dry bulb thermometer is directly exposed to the air and measures the actual temperature of air. Such a temperature is called Dry Bulb Temperature (DBT or $t$).
- The bulb of the wet bulb thermometer is covered by wick thoroughly wetted by water as shown in figure . The temperature which is measured by the wick-covered bulb of such a thermometer indicates the temperature of liquid-water in the wick and is called the wet bulb temperature (WBT or $t'$).

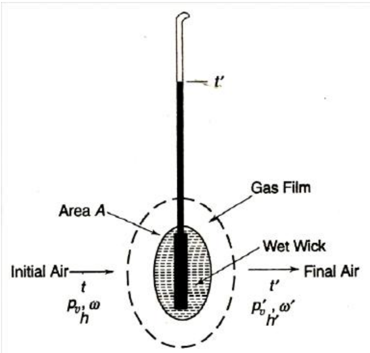



***Figure: Flow of Air Over the Wick-Covered Bulb of a Wet Bulb Thermometer***

The change of state of water vapor in air flowing over a wet bulb thermometer is shown in figure. below.

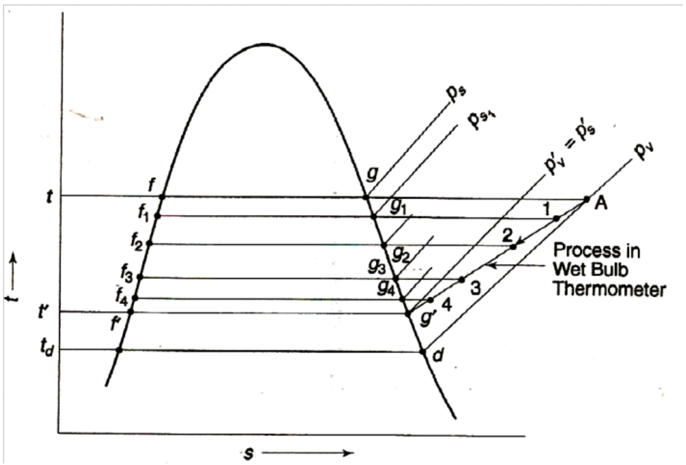



***Figure: Change of State of Water Vapour in Air Flowing Over a Wet Bulb Thermometer***

- The difference between the dry bulb and wet bulb temperatures is called wet bulb depression (WBD). Thus, WBD= $(t - t^{'})$.
- If the ambient air is saturated, viz., the RH is 100 percent, then there will be no evaporation of water on the bulb and hence WBT and DBT will be equal. The wet bulb depression will be zero. Thus WBT is an indirect measure of the dryness of air.

The wet bulb temperature is essentially not a thermodynamic property. It is the temperature of equilibrium reached by het transfer from air to water in the wick due to the temperature difference $(t - t')$ causing the evaporation of water and the consequent diffusion of water vapour into air due to the partial pressure difference $(p_v' - p_v)$, where $p_v'$ is the saturation water vapour pressure at temperature $t'$.

### 8.1.8 Measurement of Psychrometric Properties

- There is no convenient way of measuring $w, \mu\ or\ \phi$. They are properties which have to be calculated.
- The measurable properties are dry bulb, wet bulb and dew point temperatures.
- The dry bulb temperature is measured by putting ordinary bulb thermometer in a stream of air.
- The dew point temperature is measured by cooling a bulb in a stream of air until the first dew appears on the bulb.
- The wet bulb temperature is measured by rotating the wick-covered bulb of a thermometer at 160 to 660 rpm in air.
- From the dew point temperature, the saturation pressure of water can be obtained from the steam table which in turn is equal to the actual partial pressure of water vapour in the air. It is generally difficult to accurately measure the dew point temperature.
- The wet bulb temperature is easily measured with the help of a psychrometer. The wet bulb temperature, as stated earlier, is not a thermodynamic property. Therefore, no analytical expression can be derived to relate WBT with $p_v$ or the thermodynamic properties. Empirical relations exist of obtain the value of $p_v$ in terms of $t'$. One of such relations **is Carrier Equation:**

$$\boldsymbol{p_v = p_v' - \frac{(p_v - p_v')(t - t')(1.8)}{2800 - 1.3\ (1.8t + 32)}}$$

**Where, $p_v$= Partial pressure of water vapour**

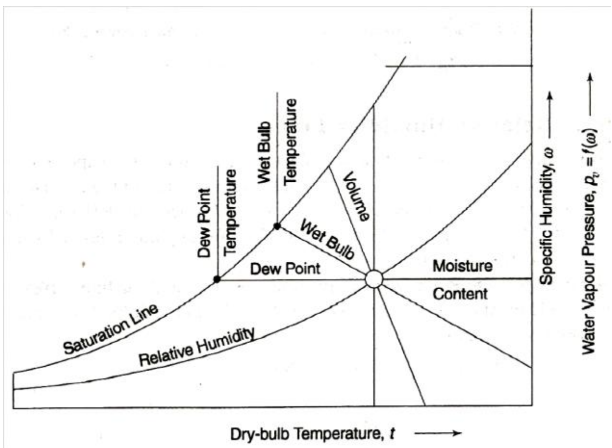
$p_v'$= Partial pressure of water vapor at wet bulb temperature $t'$

$t$ = Dry bulb temperature in $^oC$

$t'$ = Wet bulb temperature in $^oC$

### 4.1.9 Psychrometric Chart

All data essential for the complete thermodynamic and psychrometric analysis of air-conditioning processes can be summarized in a psychrometric chart. The chart which is most commonly used is the $w - t$ chart, i.e. a chart which has specific humidity or water vapour pressure along the ordinate and the dry bulb temperature along the abscissa. The chart is normally constructed for a standard atmospheric pressure of 760 mm Hg or 1.01325 bar, corresponding to the pressure at the mean sea level. A typical layout of this chart is shown in figure .



## 8.2 Psychrometric or Air-Conditioning Processes

Four basic thermodynamic processes and four combinations of processes by which the state of moist air can be altered are shown in figure. They are:

i) Sensible heating – process OA
ii) Sensible cooling – process OB
iii) Humidification – process OC
iv) Dehumidification – process OD
v) Heating and humidification – process OE
vi) Cooling and dehumidification – process OF
vii) Cooling and humidification – process OG
viii) Heating and dehumidification – process OH.

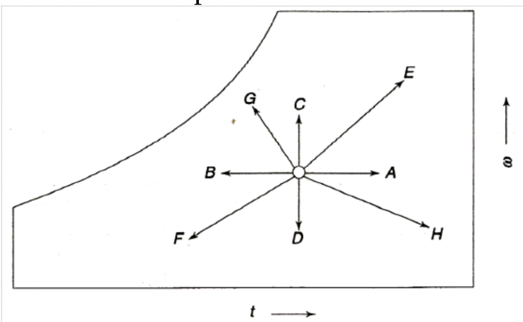



***Figure: Basic Psychrometric Process***

- The first two processes, viz., sensible heating and cooling, involve only a change in the dry bulb temperature, whereas the processes of humidifying and dehumidifying involve a change in the specific humidity.
- Thus, when the state of the air moves from O to A or to B, there is no change in the moisture content of the air; if the state changes from O to C or to D, the DBT remains constant. However, most practical moisture-transfer processes involve a change in temperature as well.
- The last four fundamental processes listed above involve both changes in temperature as well as in humidity.

### 8.2.1 Sensible Heat process – Heating or Cooling

When the state of moist air is altered along the $w$=constant line such as AB in figure, the heat has to be transferred which goes to change the temperature of the air. The heat transfer, is given by

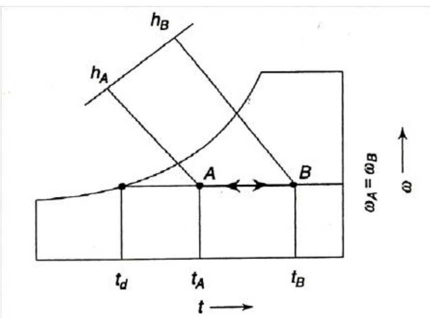



***Figure: Sensible Heat Process***

$$Q_s = m_a(h_B - h_A)$$

$$= m_a C_{p_a}(t_B - t_A) + m_a\, w C_{p_v}(t_B - t_A)$$
$$= m_a(1.005 + 1.88\, w)(t_B - t_A)$$

- If a building to be air conditioned receives or loses heat due to transmission or other reasons, it is supposed to have sensible heat load.
- Heat gain in buildings will require the conditioning of air to lower temperatures, causing a cooling load on the air-conditioning equipment. However, heat loss in buildings will require the heating of air causing a heating load on the equipment.

### 8.2.2 Latent Heat Process – Humidification or Dehumidification

When the state of air is altered along the t = constant line, such as BC in figure, moisture in the form of vapour has to be transferred to change the humidity ratio of the air. This transfer of moisture is given by

$$G = m_a(w_c - w_g)$$

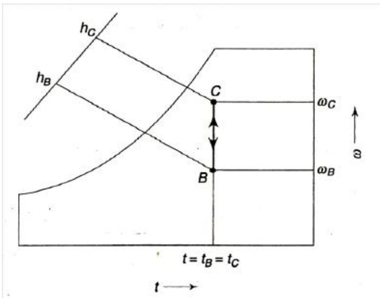



***Figure: Latent Heat Process***

Because of this change in the humidity ratio, there is also a change in the specific enthalpy of the air given by $(h_C - h_B)$ as shown in figure. In air-conditioning practice, this change in enthalpy due to the change in the humidity ratio is considered to cause a latent-heat transfer given by

$$Q_L = m_a(h_C - h_B)$$
$$= m_a h_{fg_0}(w_C - w_B)$$
$$= G h_{fg_0} = 2500\, G$$

- Accordingly, if a building gains or losses moisture, it is supposed to have a latent-heat load. A loss of moisture will require the condensation of moisture for the dehumidification of air in the conditioning apparatus, and hence a cooling load. On the other hand, a gain of moisture will necessitate the evaporation of water for the humidification of air in the apparatus and hence a heating load.

### 8.2.3 Total Heat Process

Consider now a change in the state of air along the path AC as shown in figure. This involves both a change in temperature as well as in the humidity ratio.

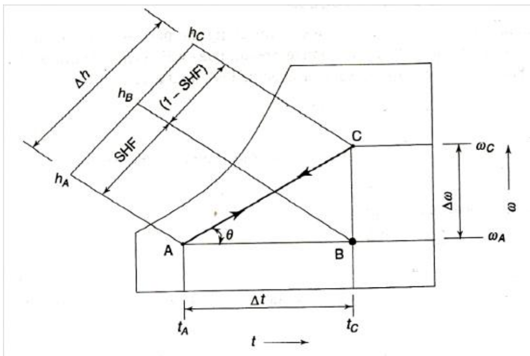



***Figure: Total Heat Process***

The change in temperature causes a sensible heat load given by

$$Q_S = m_a(h_B - h_A)$$

and a latent heat load given by

$$Q_L = m_a(h_C - h_B)$$

Adding above Eqs. we obtain an expression for total heat load as

$$Q = Q_S + Q_L$$

$$= m_a(h_C - h_A)$$

**Sensible Heat Factor (SHF)**

The ratio of the sensible heat transfer to the total heat transfer is termed as the sensible heat factor. Thus,

$$SHF = \frac{Q_S}{Q_S + Q_L} = \frac{Q_S}{Q_L}$$

we obtain

$$SHF = \frac{h_B - h_A}{(h_B - h_A) + (h_C - h_B)} = \frac{h_B - h_A}{h_C - h_A}$$

- The sensible heat transfer taking place along AB is proportional to SHF and the latent heat transfer along BC is proportional to 1 – SHF. The process line AC is called the sensible heat factor line or process or condition line.
- It is obvious that a sensible heat factor of unity corresponds to no latent heat transfer and the SHF line is horizontal on the psychrometric chart.
- However, a zero SHF line is vertical on the psychrometric chart and implies no sensible heat transfer.
- An SHF of 0.75 to 0.8 is quite common in air-conditioning practice in a normal dry climate. A lower value of SHF, such as 0.65, implies a high latent heat load, which is quite common in a humid climate.

**8.2.4 Bypass Factor**

Figure shows the process that the moist air undergoes while flowing over a surface. The air enters at 1 and leaves at 2 when the surface is maintained at S. The state of the contacted air is that of saturated air at the temperature of the surface. The un-contacted air remains at the entering state. Thus the end state of the air after mixing of contacted and un-contacted air will be at state point 2 as shown in figure.

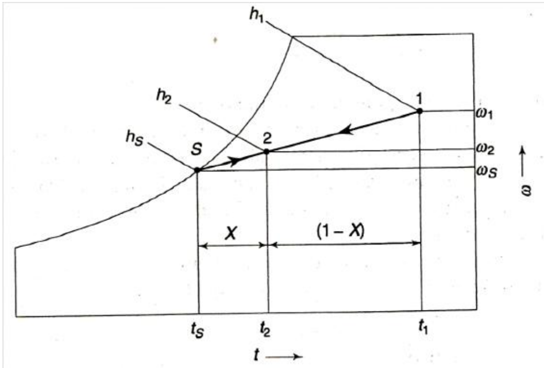



***Figure: Bypass Factor and Leaving Air State***

Thus one can define a bypass factor (BPF) of the apparatus representing the fraction of "un-contacted" air in terms of the states 1, 2 and S, as

$$X = \frac{t_2 - t_s}{t_1 - t_s} = \frac{w_2 - w_s}{w_1 - w_s} = \frac{h_2 - h_s}{h_1 - h_s}$$

Conversely, one can define a contact factor (1-X) representing a fraction of the contacted air.

### 8.3 Mixing Process

Let us consider the adiabatic mixing of different quantities of air in two different states at constant pressure as shown in figure . Let subscripts 1 and 2 refer to the two steams of air, and let $m_a$ refer to the mass of dry air in the steam.

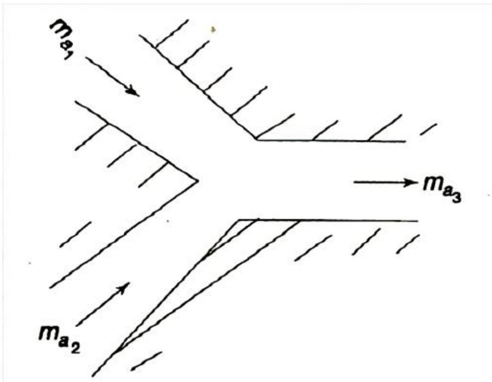


***Fig: Adiabatic Mixing of Air Streams***

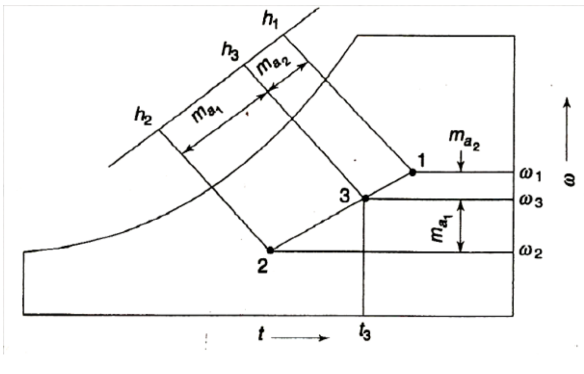



***Figure: Mixing Process on Psychrometric Chart***

Then by moisture balance, we have for the specific humidity of the mixture.

$$m_{a_3}\, w_3 = m_{a_1} w_1 + m_{a_2} w_2$$

$$w_3 = \frac{m_{a_1} w_1 + m_{a_2} w_2}{m_{a_3}}$$

Where by dry air mass balance, the mass of dry air in the mixture

$$m_{a_3} = m_{a_1} + m_{a_2}$$

Also, by energy balance, we similarly get the expression for the enthalpies and temperatures as

$$h_3 = \frac{m_{a_1} h_1 + m_{a_2} h_2}{m_{a_3}}$$

$$t_3 = \frac{m_{a_1} t_1 + m_{a_2} t_2}{m_{a_3}}$$

**Summer air conditioning systems:**

In most of the places the summer season is hot and humid. Hence, in order to provide comfortable conditions to the occupants during summer, it is required to supply cold and dry air to the occupied space. This requires systems wherein the hot and humid air can be cooled to temperatures lower than the dew point temperature, so that the water vapour in air can be removed by condensation, and the resulting cold and dehumidified air supplied to the conditioned space in required quantity for providing thermal comfort. Thus it can be seen that a typical summer air conditioning system requires a refrigeration system that reduces the temperature of the air to temperatures much lower than the surroundings. Of course, in some areas such as deserts, the summer is hot and dry. Air conditioning systems for these hot and dry climates also require cooling of air below the ambient temperatures, however, instead of removing water vapour it may be required to add water to the air supplied to the conditioned space.

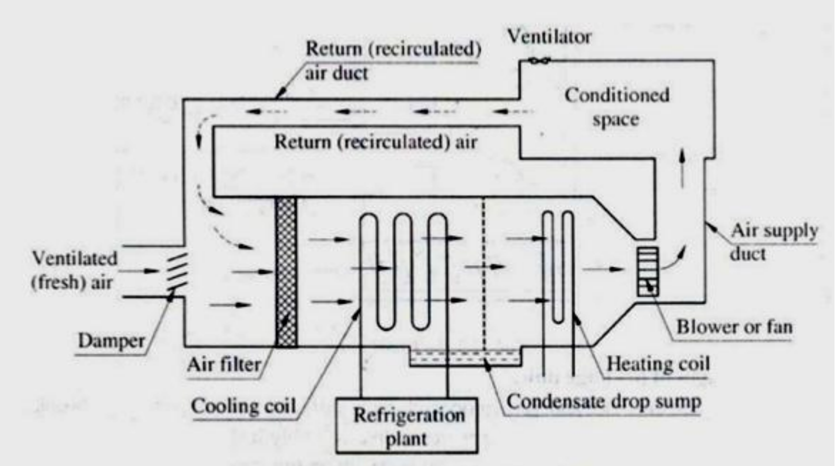
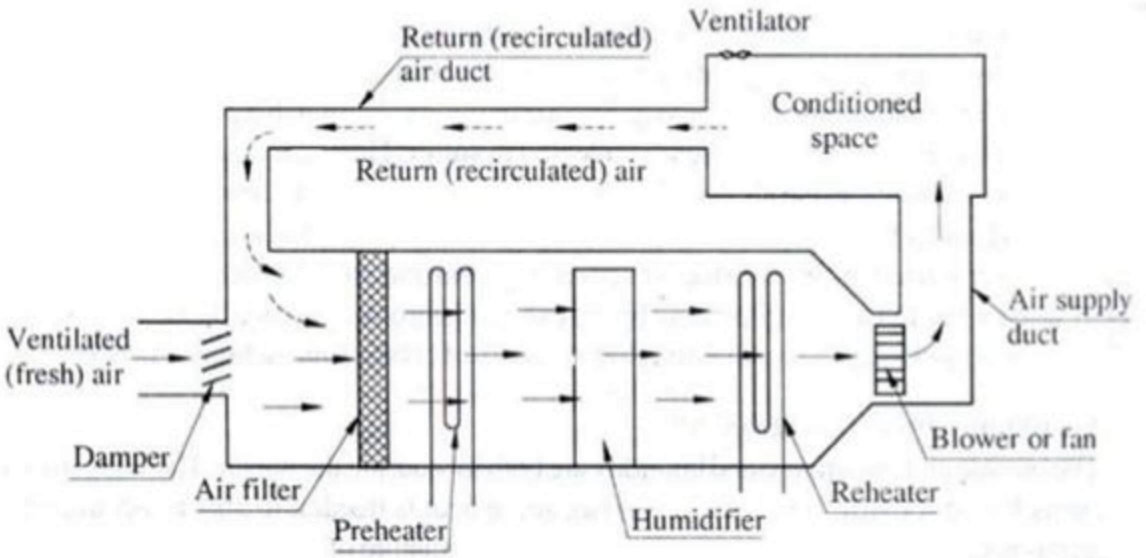



*Fig. Summer air conditioning system*

**Winter Air Conditioning System**

In winter air conditioning, the air is heated which is generally followed by humidification. The schematic arrangement of the system is shown in Fig. The outside air flows through a damper and mixes up with the re-circulated air (which is obtained from the conditioned space). The mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a preheat coil in order to prevent the possible freezing of water and to control the evaporation of water in the humidifier. After that, the air is made to pass through a reheat coil to bring the air to the designed dry bulb temperature. Now, the conditioned air is supplied to the conditioned space by a fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilators. The remaining part of the used air (Known as re-circulated air) is again conditioned as shown in Fig. The outside air is sucked and made to mix with re-circulated air, in order to make up for the loss of conditioned (or used) air through exhaust fans or ventilation from the conditioned space.




*Winter air conditioning system*