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M – Scheme

e-TEXTBOOK
on

HEAT POWER ENGINEERING

for

IV Semester DME

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HEAT POWER ENGINEERING

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2) Applied Thermodynamics ,P.K. Nag, ,2nd Edition,TATA Mcgraw - Hill Publishing Company, New Delhi .

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1) Thermal Engineering ,P.LBallaney , 24th Edition ,Khanna Publishers, New Delhi.

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UNIT – I
BASICS OF THERMODYNAMICS AND THERMODYNAMIC PROCESSES OF PERFECT GASES

1.1 Introduction

Thermodynamics is a branch of science that deals with the relations between heat, and work.

Thermodynamics deals with the study of energy transformations within systems and transfer of energy across the boundaries of the system.

1.1.1 Definition and Units

1.1.2 Mass

Mass is the amount of matter contained in an object, and does not depends on gravity.

Mass is measured in grams or kilograms. Therefore it is denoted by letter 'W' and its unit is Kg.

1.1.3 Weight

Weight is the amount of mass of an object, and it's dependent upon gravity.

It is denoted by letter 'W' and *Its unit is Newton*. Mathematically,

$$W = m\,g\,.$$

Where, 'g' is the acceleration due to gravity in m/s^2.

1.1.4 Force

According to Newton's second law of motion, the applied force or impressed force is directly proportional to the rate of change of momentum. It is denoted by letter 'F'. *Its unit is Newton (N)*. Mathematically,

$$F = \text{Mass x Acceleration}$$

$$F = m\,a$$

Note:

$1N = 1kg\text{-}m/s^2$

1.1.5 Volume

Volume is defined as the ratio of mass to density. It is also defined as the space occupied by the mass. Its unit is m^3. It is denoted by '*V*'. Mathematically,

$$V = \frac{m}{\rho}$$

Note:

$1000 \text{ c.c} = 1 \text{ litre} = 0.001m^3$

1.1.6 Density

The amount of matter (mass) in a given amount of space (volume)

Density is defined as mass per unit volume. Its unit is kg/m^3. It is denoted by 'ρ'. It is also known as mass density or specific mass. Mathematically,

$$\rho = \frac{m}{v}$$

1.1.7 Specific Weight

Specific weight is defined as the weight per unit volume. Its unit is N/m^3. It is denoted by '*w*'. It is also known as weight density. Mathematically,

$$w = \frac{W}{v}$$

$$= \rho\, g \therefore (W = mg\ \&\rho = m/v)$$

1.1.8 Specific Gravity

Specific gravity is defined as the ratio of density of given liquid to the density of a standard fluid (water). It has no unit. It is denoted by 's'. It is also known as relative density. Mathematically,

$$s = \frac{\rho l}{\rho w}$$

Note:

Density of water (ρ_w) = 1000 kg/m^3.

1.1.9 Specific Volume

The volume occupied by unit mass is known as specific volume. Its unit is m^3/kg. It is denoted by 'v_s'. Mathematically,

$$v_s = \frac{V}{m}$$

Therefore, it is the reciprocal of density, i.e. v = 1/ρ.

1.1.10 Pressure

Pressure is defined as the normal force per unit area of the surface. The unit of pressure depends upon the units of force and area. Its unit is N/m^2, Pascal (Pa) and bar. It is denoted by 'p'. Mathematically,

$$p = \frac{F}{A}$$

Note:

1 bar = 1x10^5 N/m^2 = 100 kN/m^2 = 0.1 MN/ m^2.

1Pa = 1 N/m^2

1kPa= 1 kN/m^2

1.1.11 Gauge pressure and Absolute pressure

The reading of the pressure gage is known as gauge pressure, while the actual pressure is called absolute pressure. Mathematically,

For pressures above atmospheric,

Absolute pressure = Atmospheric pressure + Gauge pressure

For pressures below atmospheric, the gauge pressure will be negative. This negative pressure is known as vacuum pressure. Mathematically,

Absolute pressure = Atmospheric pressure – Vacuum pressure

Note:

$p_{abs} = p_{atm} +\ p_g$

$p_{abs} = p_{atm} - p_v$

P_{atm}= 760 mm of Hg = 1.01325 bar = 1.01325 x 10^5 N/m^2 = 101.325 kN/m^2

1.1.12 Temperature

Temperature is the amount of heat in a system.

Temperature is defined as the degree of hotness or coldness of a body. The unit of temperature measurement is degree. It is denoted by 'T'.

When the heat is added to a body and it becomes hotter, its temperature is said to rise. When a body cools down, its temperature is said to fall.

1.1.13 Absolute Temperature

Absolute zero temperature is taken as -273°C. The temperatures measured from this zero are called absolute temperatures. The absolute temperature in Celsius scale is called degree Kelvin (K). Mathematically,

Absolute Temperature in K = Temperature in °C + 273

1.1.14 Standard Temperature and Pressure (S.T.P)

The temperature and pressure of any gas, under standard atmospheric conditions, is taken as 15°C (288 K) and 760 mm of Hg respectively.

1.1.15 Normal Temperature and Pressure (N.T.P)

The conditions of temperature and pressure at 0°C (273 K) temperature and 760 mm of Hg pressure are termed as normal temperature and pressure

1.1.16 Heat

Heat is a form of energy. It can be transferred from one body to another due to the difference of temperature.

Heat is defined as the energy transferred, without transfer of mass, across the boundary of a system because of a temperature difference between the system and the surroundings. It is expressed in Joule (J). It is usually represented by 'Q'.

1.1.17 Specific Heat Capacity

It is defined as the amount of heat required to raise or lower the temperature of a unit mass of any substance through one degree. Its unit is kJ/kgK. It is denoted by 'C'.

1.1.18 Specific Heat at Capacity at Constant Volume

It is defined as the amount of heat required to raise or lower the temperature of a unit mass of any substance through one degree when the volume remains constant. Its unit is kJ/kgK. It is denoted by 'C_v'.

$$Q = m . C_v . dT$$

Where   Q = Heat transferred (kJ)

m = Mass of the gas (kg)

C_v= Specific heat capacity at constant volume (kJ/kgK)

$dT = (T_2 - T_1)$ = Temperature difference (K)

1.1.19 Specific Heat at Capacity at Constant Pressure

It is defined as the amount of heat required to raise or lower the temperature of a unit mass of any substance through one degree when the pressure remains constant. Its unit is kJ/kgK. It is denoted by 'C_p'.

$$Q = m . C_p . dT$$

Where,

Q = Heat transferred (kJ)

m = Mass of the gas (kg)

C_p= Specific heat capacity at constant pressure (kJ/kgK)

$dT = (T_2 - T_1)$ = Temperature difference (K)

Note:

$$C_p - C_v = R$$

C_p value of water = 4.189 kJ/kgK

C_p value of air = 1.005 kJ/kgK

C_v value of air = 0.718 kJ/kgK

R (Gas Constant) = 0.287 kJ/kgK

1.1.20 Work

In mechanics, Work is done when the point of application of force moves in the direction of force. Its unit is Nm or Joules. It is denoted by 'W'. Mathematically,

Work done (WD) = Force (F) x Distance moved (d)

In thermodynamics work done is defined as the energy transferred across the boundary of a system because of an intensive property difference other than temperature exists between system and surroundings.

Work done (WD) = p.d v   (d v = change in volume)

Note:

1Nm = 1 Joule

1.1.21 Power

It may be defined as the rate of doing work or work done per unit time. Its unit is Nm/s or J/s or watt. It is denoted by 'P'. Mathematically,

$$\textbf{Power} = \frac{\textit{Work done}}{\textbf{Time}}$$

Note:

1 J/s = 1 watt

1.1.22 Energy

The concept of energy was first introduced by Newton. Energy may be defined as the capacity to do work.

Types of Energy

Energy may be classified into two general types. They are,

1. Stored energy
2. Transit energy

1. Stored Energy

It is the energy possessed by a system within its boundaries.

Examples:

- Potential energy
- Kinetic energy
- Internal energy

2. Transit Energy

It is the energy possessed by a system which is capable of crossing its boundaries.

Examples:

- Heat
- Work
- Electrical energy

1. Potential energy:

It is the energy possessed by a body or a system for doing work, by virtue of its position above the ground level.

For example a body raised to some height above the ground level possesses potential energy because it can do some work by falling on earth's surface.

Let   W = weight of the body

       m- mass of the body

       z – Distance through which the body falls and (m)

       g – Acceleration due to gravity = 9.81 m / s^2

w.k.t   Force = mass (m) x Acceleration (g)

       F = m g Newton (N)

.: P. E = m g z (N m) or Joule

2.Kinetic energy:

It is the energy possessed by a body or a system, for doing work, by virtue of its mass and velocity of motion.

       m- Mass of the body

       v – Velocity of the body

When m is in kg and V is in m/s, then kinetic energy will be in N-m, as discussed below:

We know that kinetic energy.

K.E = ½ m V²

$$= kg \ x \ \frac{m2}{s2} \ = \ \frac{kg-m}{s2} x \ m = N -m = Joule$$

3.Internal energy:

It is the energy possessed by a body or a system due to its molecular arrangement and motion of the molecules. It is usually represented by U.

1.1.23 Law of Conservation of Energy:

It states that, "*The energy can neither be created nor destroyed, though it can be transformed from one form to any other form, in which the energy can exist*".

1.1.24 Thermodynamic system:

The thermodynamic system (simply known as system) may be defined as a definite area or a space where some thermodynamic process is taking place. It is a region where our attention is focussed for studying a thermodynamic process. A little observation will show that a thermodynamic system has its boundaries and anything outside the boundaries is called its surroundings as shown in fig.
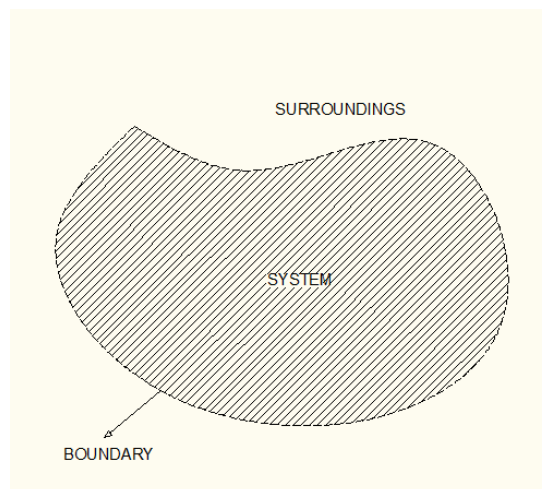
Fig 1.1

These boundaries may be fixed like that of a tank enclosing a certain mass of compressed gas, or movable like boundary of a certain volume of liquid in a pipe line.

Surroundings

Everything outside of the system which affects the behavior of the system is called surroundings.

Boundary

The system and surroundings are separated by the system boundary. It may be real or imaginary.

1.1.25 Types of thermodynamic systems:

1. *Closed system(Non flow process)*
2. *Open system (flow process)*
3. *Isolated system*

1. Closed system:

In a closed system or non-flow system, Energy crosses the system boundary in the form of heat and work, but there is no mass transfer.

Example: Gas contained in the cylinder

2. Open system:

In an open system or flow system or control volume system, both energy and mass (heat and work) cross the boundary of the system

Examples: Air compressor, flow through nozzles and turbines

3. Isolated system:

In an isolated system no mass and no energy crosses the boundary of the system.

Example: Thermal flask and insulated gas container

1.1.26 Thermodynamic Equilibrium

When mechanical equilibrium, thermal equilibrium and chemical equilibrium are satisfied, then the system is said to be in thermodynamic equilibrium.

The thermodynamic equilibrium includes three types of equilibrium. They are,

1. Thermal equilibrium
2. Mechanical equilibrium
3. Chemical equilibrium

1. Thermal Equilibrium

A system is said to be in thermal equilibrium when the temperature at all points in the system is the same.

2. Mechanical Equilibrium

A system is said to be in mechanical equilibrium when the pressure at all points in the system is the same.

3. Chemical Equilibrium

A system is said to be in chemical equilibrium, if it does not undergo any chemical reaction.

1.1.27 Properties of System:

The state of the system may be identified or described by certain observable quantities such as volume, temperature, pressure and density etc.All the quantities which identify the state of the system are called properties.

There are two sorts of property:

      1. Intensive properties

      2. Extensive properties

Intensive or Intrinsic properties:

These are the properties which are *independent of the mass* of the system.

Examples: Temperature, pressure, velocity, density, specific heat etc...

Extensive or Extrinsic properties:

These are the properties which are *dependent on the mass of the system*

Examples: volume / mass = specific volume

               Enthalpy / mass = Specific enthalpy

1.1.28 State of a system:

The state of a system when the system is in (thermodynamic equilibrium) is the *condition of the system* at any particular moment which can be identified by the statement of its properties, such as pressure, volume, temperature etc.

1.1.29 Thermodynamic process:

When a system changes its state from one equilibrium state to another equilibrium state.

During a thermodynamic process the properties of a gas (pressure, volume, temperature) changes. Energy transfer (hear or work) may also take place .
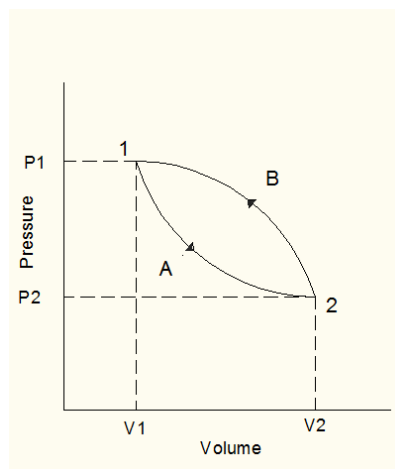

Fig 1.2

1.1.3 Process cycle:

When a process are performed on a system in such a way that the final state is identical with the initial state. This is known as cycle or cyclic process.

In fig 1-A-2 and 2-B-1 are processes whereas 1-A-2-B-1 is a thermodynamic cycle or cyclic process
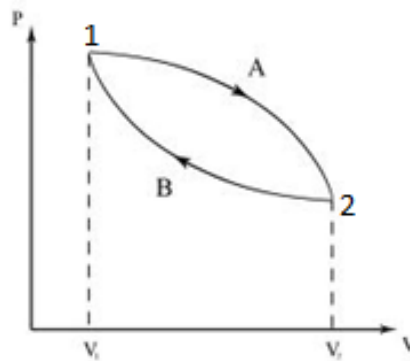

Fig 1.3

1.1.31 Point and path functions:

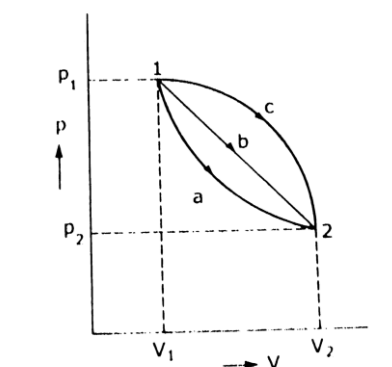The quantities which are independent on the process of path followed by the system are known as point functions.

The quantities which are dependent on the process or path followed by the system are known as path functions.

Fig 1.4

1 – 2 = point

1- a – 2 or 2 – c -1 or 1 –b- 2= path

1.1.32 LAW OF THERMODYNAMICS

1. Zeroth law of thermodynamics
2. First law of thermodynamics
3. Second law of thermodynamics

1.1.3 Zeroth Law of Thermodynamics

This law states that "when *two bodies are separately in thermal equilibrium with a third body, then they are in thermal equilibrium with each other*.
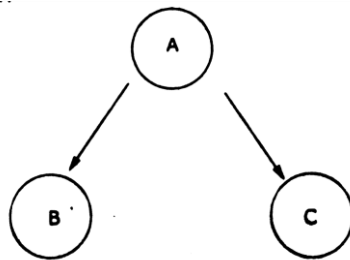

Fig 1.5

Referring the figure, systems A and B are in thermal equilibrium with each other, Systems A and C are in thermal equilibrium with each other when B and C are brought in contact B and C will not undergo any physical changes. Hence B and C are at the same temperature. (i.e) B and C will also be in thermal equilibrium with each other.

2. First Law of Thermodynamics

When a system undergoes a cyclic process, *the algebraic sum of the work transfer is proportional to the algebraic sum of the heat transfer*.

Mathematically, $\oint \delta Q = \oint \delta W$

In S.I units, the value of J is 1 Nm

.; $\int dW = \int dQ$ or

$\int dQ - \int dW = 0$

The energy can neither be created nor destroyed, though it can be transformed from one form to any other form

This law is basic for first law of thermodynamics. According to this law, when the system undergo a process, both heat transfer and work transfer takes place. The net energy transfer is stored within the system and is known as stored energy (or) total energy of the system

$$\Delta U = Q - W$$

3. Second Law of Thermodynamics

The two common statements of second law are

 (1) Kelvin plank statement

(2) Clausis statement.

Kelvin plank statement:

It is impossible to construct a heat engine working on cyclic process, whose purpose is to convert all the heat energy supplied to it into work.
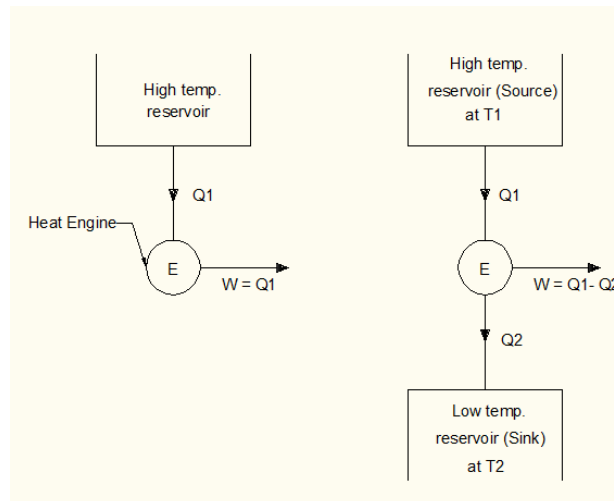

Fig 1.6   Fig 1.7

Fig 1.6   Shows a heat engine working on cyclic process it receives heat (Q) from heat source and deliver work equal to the amount of heat received. It is not possible by second law of thermodynamics.

Figure1.7 Shows the possible engine. It receives heat from heat source, a part of heat received is converted into work while the remaining heat is rejected to a lower temperature body known as sink.

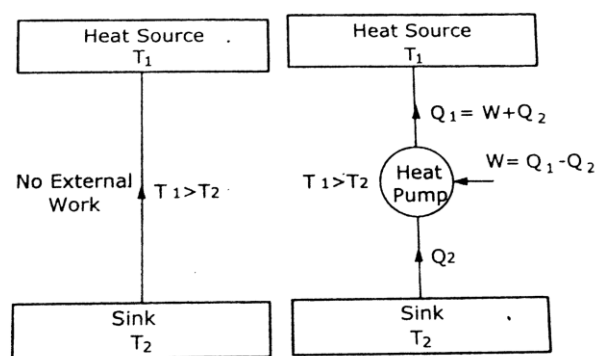**(2) Clausis statement:**

Fig 1.8   Fig 1.9

It is impossible to construct a system working on a cyclic process whose purpose is to transfer heat from a colder body to a hotter body without the aid of external work,

Fig 1.8 shows the impossible system. Here the heat source is at a temperature T1 and the heat sink at a temperature T2 if T1 > T2 then by clausis statement heat can not flow from lower temperature to higher temperature without the aid of external work.

Figurer 1.9 shows the possible system, here with the help of heat pump, heat can flow from lower temperature T2 to higher temperature T1.

1.2 Law of Perfect Gases

The physical properties of a gas are controlled by the following three variables:

1. Pressure exerted by the gas

2. Volume occupied by the gas

3. Temperature of the gas

The behaviour of a perfect gas, undergoing any change in the above mentioned variables, is governed by the following laws which have been established from the experimental results. The law of perfect gases are listed below

1. ***Boyle's law***
2. ***Charle's law***
3. ***Joule's law***
4. ***Regnault's law***
5. ***Avagardro's law***

1. Boyle's law:

➤ This law was formulated by Robert Boyle in 1662.

➤ It states "***the absolute pressure of a given mass of a perfect gas varies inversely as its volume. When the temperature remains constant***.

$$p \; \alpha \; \frac{1}{v} \quad or \quad pv = constant$$

The more useful form of the above equation is : *P1v1 = p2 v2 = p3 v3 = Constant.*

2. Charle'slaw :

➢ This law was formulated by a Frenchman Jacques A.C Charles in about 1787.

➢ It states that the volume of a given mass of a perfect gas varies directly as its absolute temperature, when the absolute pressure remains constant.

Mathematically

$$V \alpha \ T \ \text{or} \ \frac{V}{T} = \textbf{Constant}$$

Where V is the volume of the gas;

And T is the absolute temperature.

The law can also be usefully expressed as follows:

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \qquad \text{or} \qquad \frac{V_2}{V_1} = \frac{T_2}{T_1} \qquad \text{or} \qquad V_1 T_2 = V_2 T_1.$$

3. Joule's law

It states, "The change of *internal energy of a perfect gas* is directly proportional to the *change of temperature.*

Mathematically

Thus $\Delta \ u \ \alpha \ \Delta \ dT$

$\Delta \ u = m \ Cv \ (T2-T1)$

Where m – mass of the gs and

Cv – specific heat at constant volume
T₂ – Final temperature
T₁ – Initial temperature.

4. Regnault's law

It states "the two specific heats of a gas (i.e. the specific heat at constant pressure, Cp and specific heat at constant volume , Cv) do not change with the change in pressure and temperature of the gas.

That is Cp and C v of a gas always remains constant

5. Avagardro's law

It states " Equal volume of all gases, at the same temperature and pressure, contain equal number of molecules".

Mathematically **M. V = constant**

M – Molecular weight of the gas

V – specific volume of the gas

M. V is known as molar volume

1.2.1 General gas equation

In the previous section we have discussed the gas laws which give us the relation between the two variables when the third variable is constant.

But in actual practice, all the three variables i.e. pressure , volume and temperature , change simultaneously. *In order to deal with all practical cases, the Boyle's law and Charle's law are combined together, which give us a **general gas equation***

According to Boyle's law

$p \; \alpha \; \dfrac{1}{V}$ or $V \; \alpha \; \dfrac{1}{p}$ ….. (Keeping T constant)

And according to Charles law

$V \; \alpha \; T$ …… (Keeping p constant)

It is thus obvious that

$V \; \alpha \; \dfrac{1}{p}$ and T both or v $\alpha \; \dfrac{T}{p}$

$p \; V \; \alpha \; T$ or $p \; V = C \; T$

where C is a constant, whose value depends upon the mass and properties of the gas concerned. The more useful form of the general gas equation is

$\dfrac{p1V1}{T1} = \dfrac{p2V2}{T2} = \dfrac{p3V3}{T3} = Constant$

1.2.3 Characteristic gas equation

It is modified form of general gas equation. If the volume (V) in the general gas equation is taken as that of 1 kg of gas (known as its specific volume, and denoted by V_s) , then the constant C (in the general gas equation) is represented by another constant R (in the characteristic equation of gas)

Thus the general gas equation may be rewritten as

$p \; v_s = RT$

Where R is known as characteristic gas equation or simply gas constant ,

for any mass m kg of gas, the characteristic gas equation becomes

$m \; p \; v_s = m \; RT$ …. ($mv_s = V$) Specific volume = volume/mass = m³/kg

or $p \; V = m \; R \; T$

The equation p V = m R T may also be expressed in another form i.e

$p = \dfrac{m}{T} R \; T$ $= \rho R \; T$ …… ($\rho = \dfrac{m}{T}$)

1.2.4 Relationship between specific heats and gas constant

From 1 st law of thermodynamics

$\qquad$ Q = WD + du $\qquad$ (1)

Heat transfer $Q = m\, Cp\, (T_2 - T_1)$ $\qquad$ (2)

Workdone $WD = p\, (V_2 - V_1)$ $\qquad$ (3)

Change in internal energy $dU = m\, Cv\, (T_2 - T_1)$ $\qquad$ (4)

Substitute the equation (1) , (2) , (3) & (4) in (1)

$m\, Cp\, (T_2 - T_1) = p\, (V_2 - V_1) + m\, Cv\, (T_2 - T_1)$

$m\, Cp\, (T_2 - T_1) = mR\, (T_2 - T_1) + m\, Cv\, (T_2 - T_1)$ $\qquad$ Therefore $p\, v = m\, R\, T$

$m\, Cp\, (T_2 - T_1) = m\, R\, (T_2 - T_1) + Cv$

$Cp = R + Cv$ or

$Cp - Cv = R$

The above equation divided both sides by Cv

$$\frac{Cp}{Cv} = 1 + \frac{R}{Cv} \qquad \text{or} \quad \gamma = 1 + \frac{R}{Cv} \qquad\qquad \text{where } \gamma = \frac{Cp}{Cv}$$

1.2.5 Gas constant

The characteristic constant of a gas (R) is equal to the difference of its two specific heats

(ie Cp – Cv)

The unit of gas constant (R) may be obtained as discussed below

$$R = \frac{pv}{mT} = \frac{\frac{N}{m2}m3}{kgk} = \frac{N-m}{kgk} = N-m\,/\,kg\,k. = J\,/\,kg\,K. \qquad ... (\,1\,N - m = 1\,J\,)$$

The value of gas constant R is different for different gases. In S. I. units , its value for atmospheric air is taken 287 J/kg K or 0.287 kJ/kg K.

1.2.6 Universal Gas Constant

The Universal Gas Constant - R_u - appears in the ideal gas law and can be expressed as *the product between the Individual Gas Constant - R - for the particular gas - and the Molecular Weight - M_{gas} - for the gas,* and is the same for all ideal or perfect gases.

$Ru = M_{gas}\, R$

Where R_u= universal gas constant

$\qquad$ M_{gas} = molecular weight of the ideal gas or mixture of gases

The value of gas constant Ru is taken as 8314 J/kg K or 8314 kJ/kg K.

1.2.7 Internal Energy :

The energy associated with the random, disordered motion of molecules is called Internal energy.

1.2.8 Change in Internal Energy (ΔU)

In thermodynamics, the change in internal energy (dU) which depends upon the change in temperature of the system.

$$dU = m\ Cv\ (T_2 - T_1)$$

m – mass of gas in kg

Cv = specific heat at constant volume kJ/kg.k

1.2.9 Enthalpy (H) :

It is the sum of internal energy (U) and the product of pressure and volume (pv). This sum (U + p V) is known as the enthalpy of the system. It is denoted by H. It represents the *total energy stored in the system*. Thus

$$h = p\ V + U$$

The SI unit of enthalpy is the joule

1.2.10 *Change in Enthalpy:*

When a gas is heated at from p_1, V_1 & T_1 to p_2, V_2 & T_2

$$(h_2 - h_1) = dU + p\ V$$

$(h_2 - h_1) = mCv\ (T_2 - T_1) + (p_2 V_2 - p_1 V_1)$ 　　　(p V= mRT)

$$= m\ Cv\ (T_2 - T_1) + mR\ T_2 - mRT_1$$

$$= m\ Cv\ (T_2 - T_1) + mR\ (T_2 - T_1)$$

$$= m\ (T_2 - T_1)\ (Cv + R)$$

$(h_2 - h_1) = mCp\ (T_2 - T_1)$ 　　　.; $(Cp = Cv + R)$

Change in enthalpy $\Delta H = m\ Cp\ (T_2 - T_1)$

1.2.11 Entropy

Entropy is an extensive *thermodynamic property* that is the measure of a system's ***thermal energy per unit temperature***.

Entropy increases with heat addition and decreases with heat rejection.

$$dS = \frac{\delta Q}{T}$$

According to the definition of entropy

$$\delta Q = T\ .\ dS$$

1.3. Thermodynamic processes

The operation by which a system changes form one state to another is called a process. Whenever a system changes from one state to another it is accompanied by change in energy.

The following types of process are known:

1. *Constant volume process (Isochoric Process) $V_1 = V_2$*
2. *Constant pressure process (Isobaric Process) $p_1 = p_2$*
3. *Constant temperature process (Isothermal Process) $T_1 = T_2$*
4. *Hyperbolic process $p\,V = C$*
5. *Adiabatic Process $p\,V^{\gamma} = C$ (Isentropic Process) $s_1 = s_2$*
6. *Polytrophic process (General process) $p\,V^{n} = C$*
7. *Throttling process $h_1 = h_2$*
8. *Free expansion process $U_2 = U_1$*

1.3.1 Constant Volume Process (or) Isochoric Process

When a gas is heated at a constant volume, its temperature and pressure will increase. Since there is no change in its volume, therefore no work is done by the gas. All the heat supplied to the gas is stored within the gas in the form of internal energy.

Now consider m kg of a certain gas being heated at constant volume from initial state 1 to a final state 2.
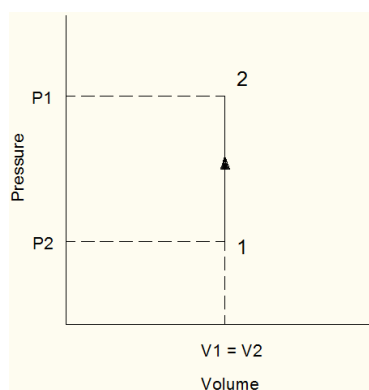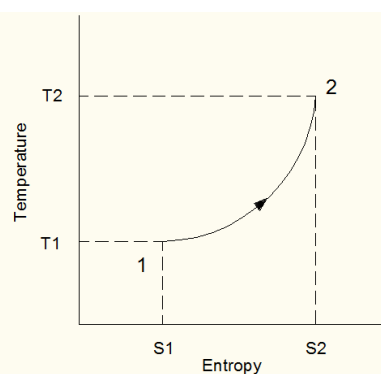

Fig 1.10 (a)   Fig 1.10 (b)

Let

p_1, V_1 and T_1 = Pressure, Volume and temperature at the initial state 1, and

p_2, V_2 and T_2 = Pressure, Volume and temperature at the final state 2.

The process is shown on the pressure-volume (p-V) and temperature – entropy (T-s) diagrams

Now let us derive the following relations for the reversible constant volume process.

Pressure-volume-temperature (*p-V*-T) relationship

We know that the general gas equation is

$$\frac{p1V1}{T1} = \frac{p2V2}{T2}$$

Since the gas is heated at constant volume, therefore $V_1 = V_2$

$$\frac{p1}{T1} = \frac{p2}{T2} \qquad or \qquad \frac{p}{T} = Constant$$

Work done by the gas

We know that $\qquad\qquad \delta w = p\ dv$

$W_{1-2} = p\ (V_2 - V_1) = 0 \qquad (V_1 = V_2)$

Change in internal energy

We know that change in internal energy,

$du = m\ cv\ dT$

$U_2 - U_1 = m\ c_v\ (T_2 - T_1)$

Heat supplied or heat transfer

We know that $\qquad\qquad dQ = dU + dW$

Since $W_{1-2} = 0$

Therefore heat supplied or heat transfer

$$Q_{1-2} = U_2 - U_1 = m\ cv\ (T_2 - T_1)$$

This shows that all the heat supplied to the gas is utilized in increasing the internal energy of the gas.

Change in enthalpy

We know that the change in enthalpy,

$$dh = dU + d(pv)$$
$$h_2 - h_1 = (U_2 - U_1) + (p_2 V_2 - p_1 V_1)$$
$$= m\, C_v\, (T_2 - T_1) + m\, R\, (T_2 - T_1)$$
$$= m(T_2 - T_1)\, (C_v + R)$$
$$\boldsymbol{= m\, C_p\, (T_2 - T_1)}$$

Change in entropy:

$$ds = \frac{dQ}{T} = \int_1^2 ds = \int_1^2 \frac{dQ}{T} = \int_1^2 mC_v \frac{dT}{T} = mC_v \int_1^2 \frac{dT}{T}$$

i.e, $S_2 - S_1 = mC_v \ln\left(\frac{T2}{T1}\right)\ kJ/K$

1.3.2 Constant Pressure process (or) Isobaric process

When a gas is heated at a constant pressure, its temperature and volume will increase. Since there is a change in its volume, therefore the heat supplied to the gas is utilized to increase the internal energy of the gas and for doing some external work. It may be noted that this process is governed by Charles' law.

Now consider m kg of a certain gas being heated at a constant pressure from an initial state 1 to a final state 2.

p_1, V_1 and T_1 = Pressure, Volume and temperature at the initial state 1, and

p_2, V_2 and T_2 = Pressure, Volume and temperature at the final state 2.

The process is shown on the pressure-volume (p-V) and temperature – entropy (T-s) diagrams
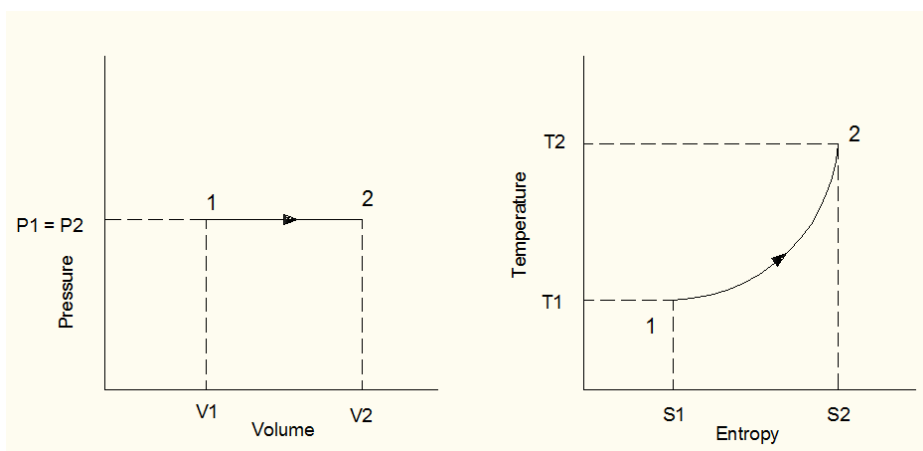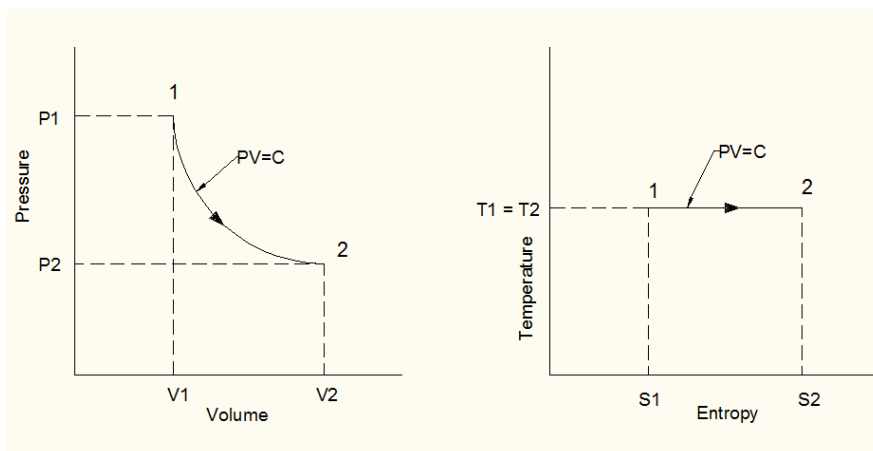

Fig 1.11 (a)   Fig 1.11 (b)

Now let us derive the following relations for the reversible constant pressure process.

Pressure – volume – Temperature (p-V-T relationship)

We know that the general gas equation is

$$\frac{p1V1}{T1} = \frac{p2V2}{T2}$$

Since the gas is heated at constant pressure, therefore $p_1 = p_2$

$$\frac{V1}{T1} = \frac{V2}{T2} \ or \ \frac{V}{T} = constant$$

Thus, the constant pressure process is governed by Charles' law.

Workdone by the gas

We know that $\delta W = p \, \delta v$

$$W = p(V2 - V1) = m \, R \, (T2 - T1)$$

Change in internal energy

$$U2 - U1 = m \, cv \, (T2 - T1)$$

Heat supplied or heat transferred

We know that $\delta Q = \delta U + \delta W$

$$Q = (U2 - U1) + W$$
$$= m \, c_v \, (T_2\text{-}T_1) + m \, R \, (T_2\text{-}T_1)$$
$$= m \, (T_2\text{-}T_1) \, (c_v\text{+}R)$$
$$= m \, c_p \, (T_2\text{-}T_1)$$

The heat supplied to the gas is utilized in increasing the internal energy of the gas and for doing some external work.

Change in enthalpy

$$h2 - h1 = m \, cp \, (T2 - T1)$$

We see that change in enthalpy is equal to the heat supplied or heat transferred.

<u>Change in entropy:</u>

$$\int_1^2 ds = \int_1^2 mCp\frac{dT}{T} = mCp\int_1^2 \frac{dT}{T}; S2 - S1 = mCp\,[\,lnT\,]_1^2$$

$S_2 - S_1 = m\ Cp\ ln\ (T2/T1)\ \ldots\ldots..kJ/K$

Note : Change in enthalpy for any reversible thermodynamic process is given by the expression

$$\Delta H = mCp\ (T2 - T1)\ \ldots\ldots\ldots kJ$$

1.3.3 Constant Temperature Process (or Isothermal Process)

A process, in which the temperature of the working substance remains constant during its expansion or compression, is called *constant temperature process* or *isothermal process.*

1. There is no change in temperature
2. There is no change in internal energy and
3. There is no change in enthalpy.

Now consider **m** kg of a certain gas being heated at constant temperature from an initial state 1 to final state 2.

Fig 1.12 (a)   Fig 1.12 (b)

Let,

p_1, V_1 and T_1 = Pressure, volume and temperature at the initial state 1 and

p_2, V_2 and T_2 = Pressure, volume and temperature at the final state 2.

Now let us derive the following relations for the reversible constant temperature process or isothermal process.

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1. *Pressure-volume-temperature (p-V-T) relationship*

We know that the general gas equation is

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2} \qquad \qquad \text{... (i)}$$

Since the gas is heated at constant temperature, therefore $T_1 = T_2$.

$\therefore \qquad \qquad p_1 V_1 = p_2 V_2 \quad \text{or} \quad p V = \text{Constant} \qquad \text{... [From equation (i)]}$

Thus, the constant temperature process or isothermal process is governed by Boyle's law.

2. *Work done by the gas*

We know that $\qquad \qquad \delta W = p \, dv$

On integrating from state 1 to state 2,

$$\int_1^2 \delta W = \int_1^2 p \, dv$$

$$W_{1\text{-}2} = \int_1^2 p \, dv \qquad \qquad \text{... (ii)}$$

Since the expansion of the gas is isothermal, i.e. $p V = C$, therefore

$$p V = p_1 V_1 \quad \text{or} \quad p = \frac{p_1 V_1}{v}$$

Substituting this value of p in equation (ii), we have

$$W_{1\text{-}2} = \int_{v_1}^{v_2} \frac{p_1 V_1}{V} \, dv = p_1 v_1 \int_{v_1}^{v_2} \frac{dV}{V}$$

$$= p_1 V_1 [log_e V]_{v_1}^{v_2} = p_1 V_1 \, log_e \left(\frac{V_2}{V_1}\right) \qquad \qquad \text{. . . (iii)}$$

(or)

$$\therefore \text{Workdone,} \quad W_{1\text{-}2} = p_1 \, v_1 \, log \left(\frac{p_1}{p_2}\right)$$

3. *Change in internal energy*

We know that change in internal energy,

$$dU = U_2 - U_1 = m \, c_v \, (T_2 - T_1)$$

Since it is a constant temperature process, i.e. $T_1 = T2$, therefore

$$dU = U_2 - U_1 = 0 \text{ or } U_1 = U_2$$

4 .Heat supplied or heat transferred

We know that heat supplied or heat transferred from state 1 to state 2.

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

This shows that all the heat supplied to the gas is equal to the workdone by the gas.

5. Change in enthalpy

We know that change in enthalpy,

$$dh = h2 - h1 = m\ c_p\ (T_2 - T_1)$$

Since it is a constant temperature process, i.e. $T_1 = T2$, therefore

$$dh = h2 - h1 = 0 \text{ or } h_1 = h_2$$

6.Change in entropy:

$$\int_1^2 ds = \int_1^2 \frac{dQ}{T} = \frac{1}{T}\int dQ \qquad (Since\ T = Constant)$$

$$= 1/T\ [Q] = \frac{Q}{T}\ i.e, S2 - S1 = \frac{p\ 1V1}{T1}\ln\frac{V2}{V1} \qquad [\ Since\ p\ 1V1\ln\frac{V2}{V1}\ and\ T = T1 = T2]$$

$$p_1V_1 = mRT_1 ;\ Since\ S_2\text{-}S_1\ \frac{mRT1}{T1}\ln(\frac{V2}{V1})$$

$$i.e,\ S_2\text{-}S_1 = mR\ ln\ (V_2/V_1)\ kJ/K$$

1.3.4 Adiabatic Process (or Isentropic Process)

A process, in which the working substance neither receives nor gives out heat to its surroundings, during its expansion or compression, is called an adiabatic process. This will happen when the working substance remains thermally insulated, so that no heat enters or leaves it during the process. It is thus obvious, that in an adiabatic or isentropic process.

1. No heat leaves or enters the gas,

2. The change in internal energy is equal to the work done.

Now consider *m* kg of a certain gas being heated adiabatically from an initial state 1 to a final state 2.

Let,

p_1, V_1 and T_1 = Pressure, volume and temperature at the initial state 1, and

p_2, V_2 and T_2 = Pressure, volume and temperature at the final state 2.

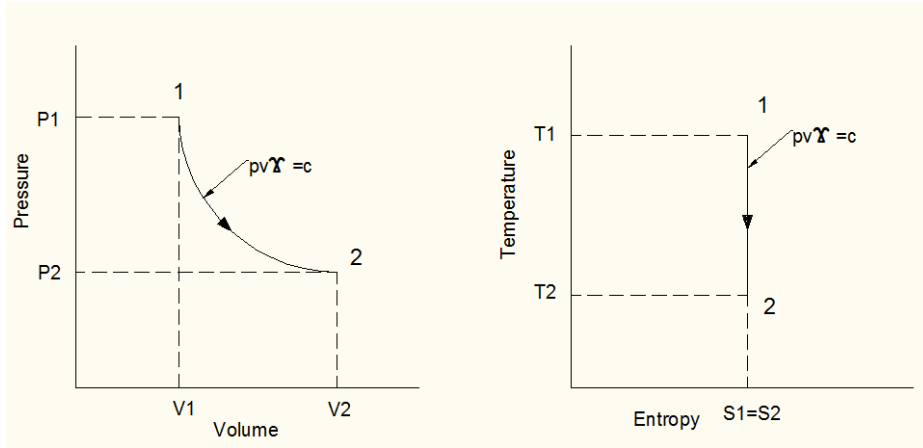<p align="center">Fig 1.13 (a)   Fig 1.13 (b)</p>

Now let us derive the following relations for a reversible adiabatic process.

1. Pressure-volume-temperature (p-V-T) relationship

We know that $\delta Q = \delta W + dU$ *(First law of thermodynamics)*   . . .(i)

Since in an adiabatic process, no heat transfer takes place, therefore

$$\delta Q = 0$$

$\therefore$ $\delta W + \delta U = 0$

or $p \, dv + m \, c_v \, dT = 0$

$\therefore$ $dT = \dfrac{-p \, dv}{mc_v}$ . . . (ii)

We know that $p \, V = m \, R \, T$

Differentiating this expression, we get

$$p \, dv + V \, dp = m \, R \, dT$$

$$dT = \frac{p \, dv + V \, dp}{m \, R} = \frac{p \, dv + V \, dp}{m(c_p - c_v)} \qquad \text{. . . (iii)}$$

$$\text{. . . } (\because R = c_p - c_v)$$

Equating equations (ii) and (iii),

$$\frac{-p \, dv}{mc_v} = \frac{p \, dv + V \, dp}{m(c_p - c_v)}$$

$$\frac{c_p - c_v}{c_v} = \frac{p \, dv + V \, dp}{-p \, dv} = -\, 1 - \frac{V \, dp}{p \, dv}$$

<p align="center">25</p>

$$\frac{c_p}{c_v} - 1 = -1 - \left(\frac{V}{dv} \times \frac{dp}{p}\right)$$

$$\gamma = -\left(\frac{V}{dv} \times \frac{dp}{p}\right) \qquad \dots \frac{c_p}{c_v} = \gamma$$

$$\gamma \times \frac{dv}{v} \times \frac{dp}{p} = 0$$

Integrating both sides,

$$\gamma \, log_e \, V + log_e \, p = \text{Constant} \qquad \text{or} \qquad log_e \, pv^\gamma = log_e \, C$$

$$pV^\gamma = C \quad \text{or} \quad p_1 V_1^\gamma = p_2 V_2^\gamma = \dots = C \qquad \dots \text{(iv)}$$

The equation (iv) may also be expressed in the following forms ;

$$\frac{p_1}{p_2} = \left(\frac{V_2}{V_1}\right)^\gamma \qquad \dots \text{(v)}$$

From the general gas equation, we now that

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2} \quad or \quad \frac{p_1}{p_2} = \frac{T_1}{T_2} \times \frac{V_2}{V_1} \qquad \dots \text{(vi)}$$

Equating equations (v) and (iv),

$$\left(\frac{V_2}{V_1}\right)^\gamma = \frac{T_1}{T_2} \times \frac{V_2}{V_1} \quad \text{or} \quad \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^\gamma \times \frac{V_1}{V_2} = \left(\frac{V_2}{V_1}\right)^\gamma \left(\frac{V_2}{V_1}\right)^{-1}$$

$$\therefore \qquad \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma - 1} \qquad \dots \text{(vii)}$$

From equation (iv), we also known that

$$\frac{V_1}{V_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \qquad \dots \text{(viii)}$$

From the general gas equation, we know that

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2} \quad \text{or} \quad \frac{V_1}{V_2} = \frac{T_1}{T_2} \times \frac{p_2}{p_1} \qquad \dots \text{(ix)}$$

Equating equations (viii) and (ix),

$$\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} = \frac{T_1}{T_2} \times \frac{p_2}{p_1} \quad \text{or} \quad \frac{T_1}{T_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \times \frac{p_1}{p_2} = \left(\frac{p_1}{p_2}\right)^{-\frac{1}{\gamma} + 1}$$

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma - 1}{\gamma}}$$

2. Workdone during adiabatic expansion

We have already discussed that workdone,

$$\delta W = p \, dv$$

On integrating from state 1 to state 2,

$$\int_1^2 \delta W = \int_1^2 p \, dv \qquad \text{or } W_{1\text{-}2} = \int_1^2 p \, dv$$

Since adiabatic expansion of the gas follows the law,

$$pv^\gamma = p_1 v_1^\gamma \qquad \text{or} \qquad p = \frac{p_1 v_1^\gamma}{v^\gamma}$$

Substituting the value of p in equation (xi),

$$W_{1\text{-}2} = \int_1^2 \frac{p_1 V_1^\gamma}{V^\gamma} \, dv = p_1 V_1^\gamma \int_1^2 V^{-\gamma} \, dV = p_1 V_1^\gamma \left[\frac{V^{-\gamma+1}}{-\gamma+1} \right]_1^2$$

$$= \frac{p_1 V_1^\gamma}{1-\gamma} \left[V_2^{1-\gamma} - V_1^{1-\gamma} \right]$$

$$= \frac{p_1 V_1^\gamma V_2^{1-\gamma} - p_1 V_1^\gamma V_1^{1-\gamma}}{1-\gamma}$$

$$= \frac{p_2 (V_2^\gamma V_2^{1-\gamma}) - p_1 (V_1^\gamma V_1^{1-\gamma})}{1-\gamma}$$

$$= \frac{p_2 V_2 - p_1 V_1}{1-\gamma}$$

$$= \frac{p_1 V_1 - p_2 V}{\gamma-1} \qquad\qquad \dots \text{(For expansion)}$$

$$\text{or}$$

$$= \frac{m \, R(T_1 - T_2)}{\gamma-1} \qquad\qquad \dots \text{(For expansion)}$$

3. Change in internal energy

We know that change in internal energy,

$$dU = U_2 - U_1 = m \, c_v \, (T_2 - T_1)$$

4. Heat supplied or heat transferred

We know that heat supplied or heat transferred in case of adiabatic process is zero, therefore

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

5. Change in enthalpy

We know that change in enthalpy,

$$dh = h2 - h1 = m\ c_p\ (T_2 - T_1)$$

6. Change in entropy:

$$ds = \frac{dQ}{T}, As\ Q = 0, \frac{dQ}{T} = 0$$

Therefore ds = 0

That is change in entropy is zero. In otherworld's, entropy remains constant. i.e, $s_2 = s_1$ because $s_2 - s_1 = 0$

1.3.5 Polytropic Process

The polytropic process is also known as the general law for the expansion and compression of gases, and is given by the relation.

$$pV^n = Constant$$

Where n is a polytropic index, which may have any value from zero to infinity, depending upon the manner, in which the expansion or compression has take place.

The various equations for polytropic process may be expressed by changing the index n for γ in the adiabatic process.

Now consider m kg of a certain gas being heated polytropically from an initial state 1 to a final state 2.
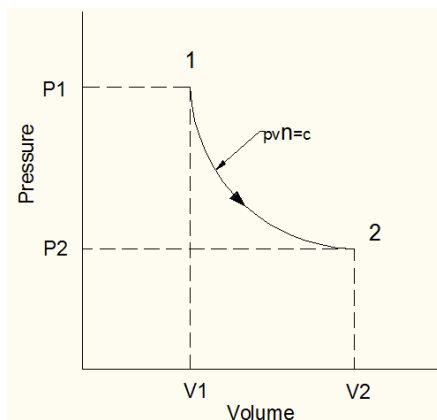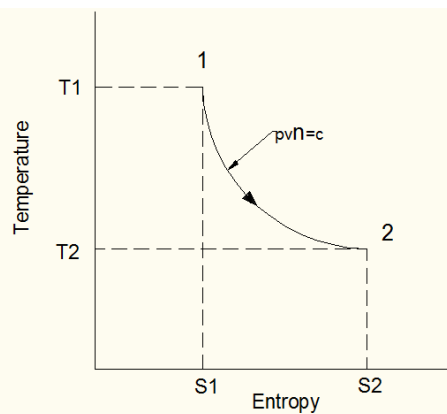
Fig 1.14 (a)   Fig 1.14 (b)

Let, p_1, V_1 and T_1=Pressure, volume and temperature at the initial state 1,

P_2, V_2 and T_2=Pressure, volume and temperature at the final state 2.

The process is shown on the p-v diagram in fig. Now let us derive the following relations for the polytropic process.

1. Pressure-Volume –temperature (p-V-T) relationship

The following relations for the polytropic process are derived in the similar way as discussed for adiabatic process.

(a) $\qquad p_1 V_1{}^n = p_2 V_2{}^n = ... = C$

(b) $\qquad \dfrac{T_1}{T_2} = \left(\dfrac{V_2}{V_1}\right)^{n-1}$

(c) $\qquad \dfrac{V_1}{V_2} = \left(\dfrac{p_2}{p_1}\right)^{\frac{1}{n}}$

(d) $\qquad \dfrac{T_1}{T_2} = \left(\dfrac{p_1}{p_2}\right)^{\frac{n-1}{n}}$

2. Work done during polytropic expansion

The equations for the work done during a polytropic process may also be expressed by changing the index n for γ in the adiabatic process.

Therefore Workdone during a polytropic process from state 1 to state 2,

$$W_{1\text{-}2} = \frac{p_1 V_1 - p_2 V_2}{n-1} = \frac{mR(T_1 - T_2)}{n-1} \qquad \text{..... (For expansion)}$$

3. Change in internal energy

We know that change in internal energy,

$\Delta U = U_2 - U_1 = mc_v (T_2 - T_1)$

4 .Heat supplied or heat transfer

We know that the heat supplied or heat transferred,

$$Q_{1\text{-}2} = w_{1\text{-}2} + \Delta U$$

$$= \frac{p_1 V_1 - p_2 V_2}{n-1} + mc_v (T_2 - T_1)$$

$$= \frac{mR(T_1 - T_2)}{n-1} + m \times \frac{R}{\gamma - 1} (T_2 - T_1) \qquad \ldots \left(* cv = \frac{R}{\gamma - 1}\right)$$

$$= mR(T1 - T2)\left[\frac{1}{n-1} - \frac{1}{\gamma-1}\right]$$

$$= mR(T1 - T2)\left[\frac{(\gamma-1)-(n-1)}{(n-1)(\gamma-1)}\right]$$

$$= mR(T1 - T2)\left[\frac{(\gamma-n)}{(n-1)(\gamma-1)}\right]$$

$$= \frac{\gamma-n}{\gamma-1} \times \frac{mR(T1-T2)}{n-1}$$

5. Change in enthalpy

We know that change in enthalpy,

$$h_2 - h_1 = m\, c_p\, (T_2 - T_1)$$

Notes: 1. The equations for heat transfer may also be expressed as:

(a) $Q_{1\text{-}2} = \frac{\gamma-n}{\gamma-1} \times \text{Work done} = \frac{\gamma-n}{\gamma-1} \times \frac{p1V1-p2V2}{(n-1)}$

(b) W.K.T.

$$Q_{1\text{-}2} = \frac{\gamma-n}{\gamma-1} \times \frac{mR(T1-T2)}{n-1} = \frac{\gamma-n}{\gamma-1} \times mc_v\,(T_1 - T_2) \qquad ..... \left(\frac{R}{\gamma-1} = cv\right)$$

$$= \frac{\gamma-n}{n-1} \times \text{change in internal energy}$$

6. Change in entropy:

$$\left[\frac{\gamma-n}{\gamma-1}\right] R.\ln\left[\frac{V2}{V1}\right] \quad \dots\dots\dots\dots\dots\dots\dots\dots \ kJ/K$$

The expression for change in entropy during a polytropic process can be derived as follows:

For polytropic process, heat transfer is given by,

$$dQ = \frac{(\gamma-n)}{\gamma-1} \times P \times dV \qquad\qquad (\text{Since } W = p \times dV)$$

Dividing both sides by T $\frac{dQ}{T} = \frac{\gamma-n}{\gamma-1} \times \frac{p}{T} \times dV$

Integrating from initial state to final state we get,

$$\int_1^2 dS = \int_1^2 \frac{\gamma-n}{\gamma-1} \times \frac{mR}{V} \times dV \qquad\qquad [\text{Since } \frac{p}{T} = \frac{mR}{V}]$$

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$$S2 - S1 = dS = \frac{\gamma - n}{\gamma - 1} \times mR \times \int_{1}^{2} \frac{dV}{V}$$

$$dS = \frac{\gamma - n}{\gamma - 1} \times mR \times [lnV]_{1}^{2} \quad or \quad dS = \frac{\gamma - n}{\gamma - 1} \times mR \left[\frac{V2}{V1}\right]$$

1.3.6 **Hyperbolic Process**

A process, in which the gas is heated or expanded in such a way that the product of its pressure and volume (i.e. p x v) remains constant, is called a hyperbolic process.

It may be noted that the hyperbolic process is governed by Boyle's law i.e. pV = c. Hence, this process is termed as hyperbolic process. It is nearly a theoretical case, and has a little importance from the subject point of view. Its practical application is isothermal process, which is discussed below.

1.3.7 **Free Expansion Process**

Free expansion occurs when a gas is allowed to expand suddenly into vacuum chamber through and orifice of the dimensions.

During this process, no heat transfer takes place and no work is done

When the both chambers are fully insulated

Therefore Q= 0 , Δ **U** = 0 and W=0

1.3.8 **Throttling process**

When a gas or vapor is expanded through a small orifice, the process called throttling process. During throttling process, no heat transfer takes place and no work is done.

Heat transfer Q = 0

Internal energy U =0

Enthalpy = 0

Entropy = 0

Problems

1. The pressure of steam inside a boiler, as measured by pressure gauge is 1 N/mm². The barometric pressure of the atmosphere is 765 mm of mercury. Find the absolute pressure of steam in N/m², kPa, bar and N/mm².

   Given:

   Gauge pressure = 1N/mm² = 1×10^6 N/m²

   Atmospheric pressure = 765 mm of Hg.

   To find: Absolute pressure of steam in N/m², kPa, bar and N/mm².

   Solution:

   We know that atmospheric pressure

   = 765 mm of Hg

   = 765 ×133.3 = $0.102\times10^{^6}$ N/m² …

   (since 1 kPa = 10^3 N/m²)

   Absolute pressure of steam

   = Atmospheric pressure + Gauge pressure

   = $(0.102\times10^6) + (1 \times 10^6)$

   Absolute pressure of steam = 1.102×10^6 N/m²

   = 1102 kPa

   = 11.02 bar

   = 1.102 N/mm²

2. In a condenser of a steam power plant, the vacuum is recorded as 700 mm of Hg. If the barometer reading is 760 mm of Hg, find the absolute pressure in the condenser in N/m², kPa, bar and N/mm².

   Given,

   Vacuum pressure = 700 mm of Hg

   Barometer reading = 760 mm of Hg

   To find: The absolute pressure in the condenser in N/m², kPa, bar and N/mm².

Solution:-

We know that absolute pressure in the condenser

= Atmospheric pressure – Vacuum Pressure

= Barometric pressure – Vacuum pressure

= 760 – 700 = 60 mm of Hg

Absolute pressure = 60 × 133.3 = 7998 N/m^2

= 7.998 kPa

= 0.07998 bar

= 0.007998 N/mm^2

3.A gas occupies a volume of 0.1 m^3 at a temperature of 20° C and a pressure of 1.5 bar. Find the final temperature of the gas, if it is compressed to a pressure of 7.5 bar and occupies a volume of 0.04 m^3.

Given,

$$V_1 = 0.1 \text{ m}^3$$
$$T_1 = 20 \text{ C} = 20 + 273 = 293 \text{ K}$$
$$p_1 = 1.5 \text{ bar} = 0.15 \times 10^6 \text{ N/m}^2$$
$$p_2 = 7.5 \text{ bar} = 0.75 \times 10^6 \text{ N/m}^2$$
$$V_2 = 0.04 \text{ m}^3$$

To find: The final temperature of the gas (T2)

Solution:-

Let, T_2 = Final temperature of the gas

We know that $\dfrac{p_1 V_1}{T_1} = \dfrac{p_2 V_2}{T_2}$

$$T2 = \frac{p_2 X V_2 X T_1}{p_1 X V_1}$$

$$\frac{0.75 \times 10^6 \times 0.04 \times 293}{0.15 \times 10^6 \times 0.1}$$

$$T_2 = 586 \text{ K}$$

$$T_2 = 586 - 273 = 313° \text{ C}$$

4. A mass of 2.25 kg of nitrogen occupying 1.5 m^3 is heated from 25°C to 200°C at a constant volume. Calculate the initial and final pressures of the gas. Take universal gas constant as 8314 J/kg mol K. The molecular mass of nitrogen is 28.

Given:-

$$m = 2.25 \text{ kg}$$
$$V_1 = 1.5 \ m^3$$
$$T_1 = 25°C = 25 + 273 = 298 \text{ K}$$
$$T_2 = 200°C = 200 + 273 = 473 \text{ K}$$
$$Ru = 8314 \text{ J/kg mol K}$$
$$M = 28$$

To find: The initial and final pressures of the gas (p_1 and p_2)

Solution:-

We know that gas constant,

$$R = \frac{Ru}{M} = \frac{8314}{28} = 297 \text{ J/kg K}$$

Initial pressure of the gas

Let p_1 = Initial pressure of the gas

We know that

$$p1V1 = mRT1$$

$$\therefore \quad p1 = \frac{mRT1}{V1}$$

$$= \frac{2.25 \times 297 \times 298}{1.5} \quad = 0.133 \times 10^6 \left(\frac{N}{m^2}\right)$$

$$p_1 = 1.33 \text{ bar}$$

Final pressure of the gas

Let p_2 = Final pressure of the gas.

Since the volume is constant, therefore

$$\frac{p1}{T1} = \frac{p2}{T2} \quad \text{or} \quad p_2 = \frac{p1T2}{T1}$$

$$= \frac{1.33 \times 473}{298}$$

$$p_2 = 2.11 \text{ bar}$$

5. One kg of a perfect gas occupies a volume of $0.85m^3$ at 15°C and at a constant pressure of 1 bar. The gas is first heated at a constant volume, and then at a constant pressure. Find the specific heat at constant volume and constant pressure of the gas. Take $\gamma = 1.4$.

Given:-

m = 1 kg;

$V = 0.85m^3$;

T = 15°C = 15 + 273 = 288K;

$p = 1 \text{ bar} = 0.1 \times 10^6 \ N/m^2$;

$\gamma = \frac{Cp}{Cv} = 1.4$

To find: The specific heat at constant volume and constant pressure of the gas.

Solution:-

Specific heat of gas at constant volume

Let   Cv = Specific heat of gas at constant volume and

R = Characteristic gas constant

We know that

$$pv = mRT$$

$$R = \frac{pV}{mT} = \frac{0.1 \times 10^6 \times 0.85}{1 \times 288} = 295 \text{ J/kg K} = 0.295 \text{ kJ/kg K}$$

We know that

$$Cv = \frac{R}{\gamma - 1} = \frac{0.295}{1.4 - 1} = 0.7375 \text{ kJ/kg K}$$

Specific heat of gas at constant pressure

We know that specific heat of gas at constant pressure

$$Cp = 1.4Cv = 1.4 \times 0.7375 = 1.0325 \text{ kJ/kg K}$$

6. One kg of ideal gas is heated from 18.3°C to 93.4°C. Assuming R = 0.264 kJ/kg K and $\gamma = 1.18$ for the gas, find: 1. Specific heats; 2. Change in internal energy and 3. Change in enthalpy.

Given:-

m = 1 kg;

T_1 = 18.3°C = 18.3 + 273 = 291.3 K;

T_2 = 93.4°C = 93.4 + 273 = 366.4 K;

R = 0.264 kJ/kg K;

$\gamma = \frac{Cp}{Cv} = 1.18$

To find: 1. Specific heats; 2. Change in internal energy and 3. Change in enthalpy.

Solution:-

Specific heats

Let   Cp = Specific heat at constant pressure and

Cv = Specific heat at constant volume

We know that

$$Cv = \frac{R}{\gamma - 1} = \frac{0.264}{1.18 - 1} = 1.47 \text{ kJ/kg K}$$

∴ $Cp = \gamma Cv = 1.18 \times 1.47 = 1.73 \text{ kJ/kg K}$

Change in internal energy

We know that change in internal energy,

$$dU = mCv (T_2 - T_1) = 1 \times 1.47(366.4 - 291.3) = 110.4 \text{ kJ}$$

Change in enthalpy

We know that change in enthalpy,

$$dh = mCp\ (\ T_2 - T_1) = 1 \times 1.73(366.4 - 291.3) = 130\ kJ.$$

7.A certain gas occupies a space of 0.3 m³ at a pressure of 2 bar and a temperature of 77° C. It is heated at a constant volume, until the pressure is 7 bar. Determine 1. Temperature at the end of the process; 2. Mass of the gas; 3. Change in internal energy;   and 4. Change in enthalpy during the process.

Assume c_p = 1.005 kJ/kg k; c_v= 0.712kJ/kg k; and R = 287 J/kg k.

Given

Initial volume V_1=0.3 m³   Initial pressure p_2 = 7bar = 0.7 x 10^6 N / m²

Initial pressure p_1 = 2bar = 0.2 x 10^6 N / m² c_p = 1.005 kJ/kg k

                                                c_v= 0.712kJ/kg k

Initial Temperature (T1) = 77° C = 350 K   R = 287 J/kg k

To find: 1. Temperature at the end of the process; 2. Mass of the gas; 3. Change in internal energy;   and 4. Change in enthalpy during the process.

1. Temperature at the end of the process

Let T_2=Temperature at the end of the process.

We know that $\dfrac{p1}{T1} = \dfrac{p2}{T2}$

$$T2 = \dfrac{p2\ \ T1}{p1}$$

$$= \dfrac{(0.7 \times 10^6 \times 350)}{0.2 \times 10^6}$$

= 1225 K   = 1225 -273   = 952℃

2. Mass of the gas

Let   m = Mass of the gas.

We know that $p_1 V_1 = m R T_1$

$$m = \frac{p1V1}{RT1}$$

$$= \frac{(0.2 \times 10^6 \times 0.3)}{287 \times 350}$$

$$= 0.597 \text{ kg}$$

3. Change in internal energy

We know that change in internal energy,

$$U2 - U1 = m \text{ cv } (T2 - T1)$$

$$= 0.597 \times 0.712(1225 - 350)$$

$$= 372 \text{KJ}$$

4. Change in enthalpy

We know that change in enthalpy,

$$h2 - h1 = m \text{ cp } (T2 - T1)$$

$$= 0.597 \times 1.005(1225 - 350)$$

$$= 525 \text{KJ}$$

8. A closed vessel contains 2 kg of carbon dioxide at temperature 20°C and pressure 0.7 bar. Heat is supplied to the vessel till the gas acquires a pressure of 1.4 bar. Calculate 1. Final Temperature; 2. Work done on or by the gas; 3. Heat added and 4. Change in internal energy. Take specific heat of the gas at constant volume as 0.657 kJ/kg K.

Given:-

$m = 2$ kg;

$T_1 = 20°C = 20 + 273 = 293K$;

$p_1 = 0.7$ bar;

$p_2 = 1.4$ bar;

cv = 0.657.

To find: 1. Final Temperature; 2. Work done on or by the gas; 3. Heat added and
4.Change in internal energy.

Solution:-

1.Final temperature

Let T_2 = Final Temperature

Since the gas is heated in a closed vessel, therefore the volume of gas
will remain constant.

We know that $\frac{p1}{T1} = \frac{p2}{T2}$

$\therefore$ $T2 = \frac{p2\ T1}{p1}$

$= \frac{1.4 \times 293}{0.7} = 586\ \text{K} = 586 - 273 = 313°\text{C}$

2. Work done on or by the gas

Since there is no change in volume, therefore work done on or by the
gas $(W1-2)$ is zero

3.Heat added

We know that heat added at constant volume,

$Q = m\ Cv\ (T2 - T1) = 2 \times 0.657\ (586 - 293)\ = 385\ \text{kJ}$

4.Change in internal energy

Let   dU = Change in internal energy

We know that Q = W + dU

Q = 385 kJ

9. The values of specific heats at constant pressure and at constant volume for an ideal gas are 0.984 kJ/kg.k and 0.728kJ/kg.k. Find the values of characteristics gas constant (R) and ratio of specific heats (γ) for the gas. If 1kg of this gas is heated at constant pressure from 25°C to 200°C , estimate the heat added, ideal work done and change in internal energy. Also calculate pressure and final volume, if the initial volume was $2m^3$.

Given,

C_p=0.984kJ/kg.k

C_v=0.728kJ/kg.k

m=1kg

T_1=25°C=25+273=298K

T_2=200°C =200+273=473K

V_1=$2m^3$

To find: 1.The heat added, ideal work done and change in internal energy.

2. Also calculate pressure and final volume, if the initial volume was $2m^3$.

Solution:

The heating of gas at constant pressure is shown in Fig.
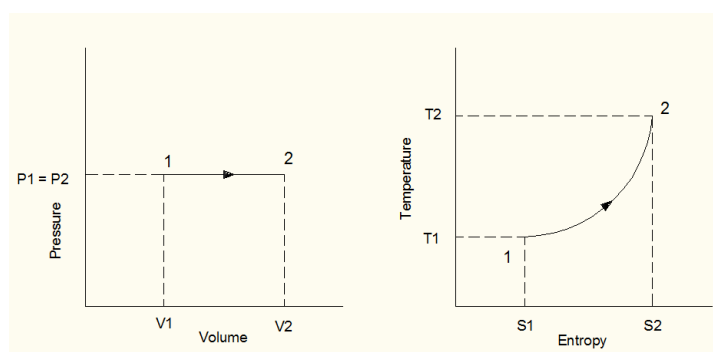
Characteristic gas constant

We know that ratio of specific heats,

$$R= C_p - C_v = 0.984 - 0.728$$

$$= 0.256\text{kJ/kg K}$$

Ratio of specific heats

We know that ratio of specific heats,

$$\gamma = Cp/Cv = 0.984/0.728 = 1.35$$

Heat added,

We know that heat added during constant pressure operation,

$$Q = mC_p(T_2 - T_1)$$

$$= 1 \times 0.984(473 - 298) = 172.2 \text{ kJ}$$

Work done,

We know that work done during constant pressure operation,

$$W = p (V_2 - V_1) = mR(T_2 - T_1)$$

$$= 1 \times 0.256(473-298) = 44.8 \text{ kJ}$$

Change in internal energy,

We know that change in internal energy,

$$U_2 - U_1 = m\ Cv(T_2 - T_1)$$

$$= 1 \times 0.728(473-298) \text{ kJ}$$

$$= 127.4 \text{ kJ}$$

Pressure and final volume of the gas if the initial volume $V_1 = 2m^3$

Let　　　　　$p_1 = p_2 =$ Pressure of the gas and

$V_2 =$ Final volume of the gas

We know that $\dfrac{p_1 V_1}{T_1} = \dfrac{p_2 V_2}{T_2}$　　　or　　　$\dfrac{V_1}{T_1} = \dfrac{V_2}{T_2}$

$$v_2 = \dfrac{V_1 T_2}{T_1} = 2 \times 473/298 = 3.17 m^3$$

We also know that

$$p_1 V_1 = mRT_1$$

$$p_1 = \dfrac{mRT_1}{V_1} = (1 \times 256 \times 298)/2 = 38140 \text{ N/m}^2$$

$$= 0.3814 \text{ bar.}$$

10. A quantity of gas has a volume of $0.14m^3$, pressures 1.5 bar and a temperature $100°C$. If the gas is compressed at a constant pressure, until its volume becomes $0.112m^3$, determine: 1.The temperature at the end of compression; 2. workdone in compressing the gas; 3. Decrease in internal energy; and 4. Heat given out by the gas.

Assume c_p=1.005kJ/Kg.k ; c_v = 0.712KJ/Kg.K ; and R=285 J/Kg.K

Given

V_1=0.14m$^{3;}$

V2=0.112m$^{3;}$

p =1.5bar=0.15×10^6N/m^2

T_1=100°C = 100+273 =373K

C_p=1.005KJ/Kg.K

C_v=0.712KJ/Kg.K

R=285J/Kg.K

To find: 1.The temperature at the end of compression; 2. Work done in compressing the gas; 3. Decrease in internal energy; and 4. Heat given out by the gas.

Solution:

The compression of gas at constant pressure is shown in Fig.
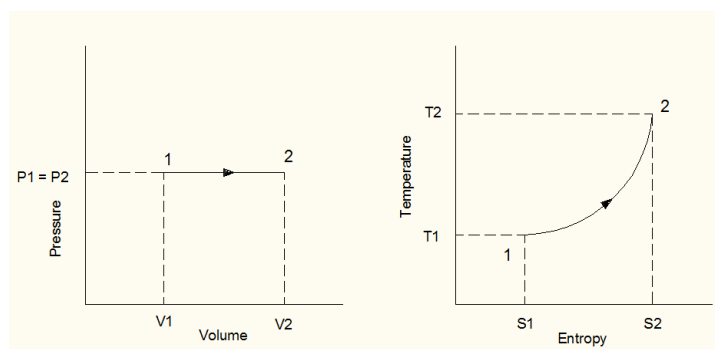
1.Temperature at the end of compression

Let T_2=Temperature at the end of compression

We know that $\frac{V1}{T1} = \frac{V2}{T2}$

$$T2 = \frac{V2\ T1}{V1} = 0.112 \times 373/0.14 = 298.4K$$

2. Work done in compressing the gas

We know that work done in compressing the gas,

$$W = p\ (V_1 - V_2)$$

$$= 0.15 \times 10^6\ (0.14 - 0.112)$$

$$= 4200J \quad = 4.2\ KJ$$

3. Decrease in Internal energy

First of all, let us find the mass of gas (m) admitted for compression.

We know that

$$p_1\ V_1 = m\ R\ T_1$$

$$m = \frac{p1V1}{RT1}$$

$$= 0.15 \times 10^6 \times 0.14\ /\ 285 \times 373$$

$$= 0.197Kg$$

$$U_1 - U_2 = m\ Cv\ (T_1 - T_2)$$

$$= 0.197 \times 0.712\ (373 - 298.4)$$

$$= 10.46\ kJ$$

4. Heat given out by the gas,

We know that heat given out by the gas,

$$Q = m\ Cp\ (T_1 - T_2)$$

$$= 0.197 \times 1.005\ (373 - 298.4)$$

$$= 14.77\ kJ$$

11. A quantity of air has a volume of 0.4 m³ at a pressure of 5 bar and a temperature of 80°C. It is expanded in a cylinder at a constant temperature to a pressure of 1 bar. Determine the amount of work done by the air during expansion.

Given:

$V_1 = 0.4$ m³

$p_1 = 5$ bar $= 5 \times 10^5$ N/m²

$T = 80°$ C

$p_2 = 1$ bar $= 1 \times 10^5$ N/m²

To find: The amount of work done by the air during expansion.

Solution:

First of all, let us find the volume of air at the end of expansion (i.e. v_2). We know that

$$p_1 V_1 = p_2 V_2 \quad \text{or} \quad v_2 = \frac{p_1 V_1}{p_2}$$

$$= \frac{5 \times 10^5 \times 0.4}{1 \times 10^5} = 2 \text{ m}^3$$

and expansion ratio, $r = \dfrac{v_2}{v_1} = \dfrac{2}{0.4} = 5$

We know that work done by the air during expansion,

$$\text{WD} \quad = p_1 V_1 \log_e r$$

$$= 5 \times 10^5 \times 0.4 \times \log_e 5$$

$$= 321887 \text{ J}$$

$$= \textbf{321.87 kJ Ans.}$$

12. 0.1 m^3 of air at a pressure of 1.5 bar is expanded isothermally to 0.5 m^3. Calculate the final pressure of the gas and heat supplied during the process.

Given:

$V_1 = 0.1 \text{ m}^3$

$p_1 = 1.5 \text{ bar} = 1.5 \text{ x } 10^5 \text{ N/m}^2$

$V_2 = 0.5 \text{ m}^3$

To find: The final pressure of the gas and heat supplied during the process.

Solution:

Final pressure of the gas

Let p_2 = Final pressure of the gas.

We know that $p_1 V_1 = p_2 V_2$

∴
$$p_2 = \frac{p_1 V_1}{V_2} = \frac{1.5 \times 10^5 \times 0.1}{0.5}$$

`
$$= 0.3 \times 10^5 \text{ N/m}^2$$

$$= \textbf{0.3 bar}$$

∴ Workdone during the process,

$$W = p_1 V_1 \log_e \frac{V_2}{V_1}$$

$$= 1.5 \text{ x } 10^5 \text{ x } 0.1 \text{ x } \log_e \frac{0.5}{0.1}$$

$$= 24115 \text{ J}$$

$$= 24.115 \text{ kJ}$$

We know that in an isothermal process, heat supplied (Q_{1-2}) is equal to the work done during the process.

∴ $Q = W = \textbf{24.115 kJ Ans.}$

13. One liter of hydrogen at 0° C is suddenly compressed to one-half of its volume. Find the change in temperature of the gas, if the ratio of two specific heats for hydrogen is 1.4.

Given:

$V_1 = 1$ litre

$T_1 = 0°\,C = 0 + 273 = 273\,K$

$V_2 = V_1/2 = ½ = 0.5$ liter

$\gamma = 1.4$

To find: The change in temperature of the gas.

Solution:

Let, $T_2 =$ Final temperature of the gas.

We know that $\dfrac{T_1}{T_2} = \left(\dfrac{V_2}{V_1}\right)^{\gamma-1} = \left(\dfrac{0.5}{1}\right)^{1.4-1} = (0.5)^{0.4} = 0.758$

$$T_2 = T1/0.758 = 273/0.758 = 360.16\,K$$

$$= 360.16 - 273$$

$$= 87.16°C \textbf{ Ans.}$$

14. The initial volume of 0.18 kg of a certain gas was 0.15m³ at a temperature of 15° C and a pressure of 1 bar. After adiabatic compression to 0.056m³, the pressure was found to be 4 bar. Find:1. Gas constant; 2. Molecular mass of the gas; 3. Ratio of specific heats; 4. Two specific heats, one at a constant pressure and the other at a constant volume; 5. Change in internal energy.

Given:

m= 0.18kg

$V_1 = 0.15$ m3

$T_1 = 15°C = 15 + 273 = 288\,K$

$p_1 = 1$ bar $= 1 \times 10^{5}$ N/m²

$V_2 = 0.056$ m3

$p_2 = 4$ bar $= 4 \times 10^{5}$ N/m²

To find: 1. Gas constant; 2. Molecular mass of the gas; 3. Ratio of specific heats; 4. Two specific heats, one at a constant pressure and the other at a constant volume; 5. Change in internal energy.

Solution:

1. Gas constant

Let,   R = Gas Constant

We know that

$$p_1 V_1 = m\ R\ T$$

$$R = \frac{p_1 V_1}{m\ T_1} = \frac{1 \times 10^5 \times 0.15}{0.18 \times 288}$$

$$= 289.4\ \text{J/kg K}$$

$$= 0.2894\ \text{KJ/kg K Ans.}$$

2. Molecular mass of the gas

We know that molecular mass of the gas,

$$M = \frac{\text{Universal gas constant } (R_u)}{\text{Charecteristic gas constant } (R)} = \frac{8314}{289.4} = 28.73\ \textbf{Ans.}$$

$$\ldots (\because R_u = 8314\ \text{J/kg K, for all gases})$$

3. Ratio of specific heats

We know that ratio of specific heats,

$$\gamma = \frac{\log\left(\frac{p_2}{p_1}\right)}{\log\left(\frac{V_1}{V_2}\right)} = \frac{\log\left(\frac{4 \times 10^5}{1 \times 10^5}\right)}{\log\left(\frac{0.15}{0.056}\right)} = \frac{\log 4}{\log 2.678} = \frac{0.6020}{0.4278}$$

$$= 1.407\ \textbf{Ans.}$$

4. specific heat at a constant volume and constant pressure

Let, C_V = specific heat at a constant volume, and

                       C_p = specific heat at a constant pressure

We know that $C_p - C_v = R$ or $1.407\ C_p - C_v = 0.2894$ $\ldots (\gamma = C_p/C_v = 1.407)$

$\therefore$ $C_v = 0.2894 / 0.407 = 0.711\ \text{KJ/kg K Ans.}$

And $\quad Cp = 1.407\ Cv = 1.407 \times 0.711 = 1$ KJ/kg K **Ans.**

5. Change in internal energy

First of all, let us find the final temperature (T_2), We know that

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{1}{4}\right)^{\frac{1.407 - 1}{1.407}} = (0.25)^{0.298} = 0.67$$

$T_2 = T_1/0.67 = 288/0.67 = 430$ K

We know that change in internal energy,

$$dU = U_2 - U_1 = m\ c_v\ (T_2 - T_1)$$

$$= 0.18 \times 0.711\ (430 - 288)\ kJ$$

$$= 18.17\ kJ\ \textbf{Ans.}$$

15. A certain quantity of air has a volume of 0.028 m^3 at a pressure of 1.25 bar and 25^{0}C. It is compressed to a volume of 0.0042m^3 according to the law pv$^{1.3}$= constant. Find the final temperature.

 Given:

V$_1$ =0.028 m^3, p$_1$ = 1.25 bar=0.125×10^{6}N/m^2; T$_1$= 25^{0}c = 25+273=298K

V$_2$ = 0.0042 m^3; n =1.3

To find: The final temperature (T2).

 Final temperature

Let, $\qquad$ T$_2$ = Final temperature.

W.K.T,

$$\frac{T1}{T2} = \left(\frac{V2}{V1}\right)^{n-1} = \left(\frac{0.0042}{0.028}\right)^{1.3-1}$$

$$= (0.15)^{0.3} = 0.566$$

T$_2$=T$_1$/0.566=298/0.566 = 526.5K-273= 253^0 C

.

16. An internal combustion engine has the following dimensions: Diameter of cylinder =550mm: Stroke = 750mm; Compression ratio 13.5. At the end of the suction stroke, the pressure is 1 bar and temperature is 316 K. The compression follow the law $pv^{1.37}$ = constant.

Determine:

1. The pressure and temperature at the end of compression;

2. The mass of the charge;

3. The Work done during compression; and

4. The heat rejected during compression

Take C_P = 0.996KJ/Kg K and C_V 0.707 KJ/Kg K.

Given

$D = 550$ mm;

$L = 750$ mm;

$r = 13.5$;

$p_1 = 1$ bar;

$T_1 = 316$ K;

$n = 1.37$;

Solution:

1. Pressure and temperature at the end of compression

Let p_2 and T_2 = Pressure and temperature at the end of compression respectively.

W.K.T,

$$p_1 V_1^n = p_2 V_2^n$$

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^n = 0.1 \times 10^6 (13.5)^{1.37}$$

$$= 3.536 \times 10^6 \text{ N/m}^2$$

We also know that

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} = \left(\frac{1}{13.5}\right)^{1.37-1} = (0.074)^{0.37} = 0.3817$$

$$T_2 = T_1 / 0.3817 = 316/ 0.3817$$

$$= 828 \text{ K}$$

2. Mass of charge

Let   m= mass of the charge.

w.k.t,   Swept volume,

$$v_1 - v_2 = \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4}(0.55)^2\, 0.75 = 0.178 m^3$$

or $v_1 - \frac{v1}{13.5} = 0.178$

$$V1 = 0.178 \times 13.5/12.5 = 0.192 m^3$$

And gas constant, $R = C_p - C_v = 0.996 - 0.707 = 0.289$ J/Kg K

W.K.T, $p_1 v_1 = mRT_1$

$$m = \frac{p1V1}{RT1} = \frac{0.1 \times 10^6 \times 0.192}{289 \times 316}$$
$$= 0.21 kg$$

3. Work done during compression

W.K.T, Work done during compression,

$$W_{1-2} = \underline{m\ R\ (T_2 - T_1)} = \frac{0.21 \times 0.289(828 - 316)}{1.37 - 1} = 84\ KJ$$

4. Heat rejected during compression

W.K.T, adiabatic index,

$\gamma = C_P/C_V = 0.0996/0.707 = 1.41$

Heat rejected during compression,

$$Q_{1-2} = \frac{\gamma - n}{\gamma - 1} \times \text{Work done during compression}$$

$$= \frac{1.41 - 1.37}{1.41 - 1} \times 84 = 8.195\ KJ$$

1.4 Steady Flow Systems:

In a thermodynamic system, the working fluid may enter the system and leave the system. If the rates of flow of mass and energy across the system are constant, then the system is known as steady flow system.

1.4.1 Control Volume:

A certain fixed region in space through which the fluid flows at a steady rate is known as control volume.

1.4.2 Steady flow energy equation:

When there is a mass transfer across the boundary, the system is called open system. Most of the engineering devices are open system, involving the flow of mass through them.

Steady flow means that the rate of flow of mass and energy are constant.

$m_1 = m_2 = m$ mass rate of flow in kg / sec.
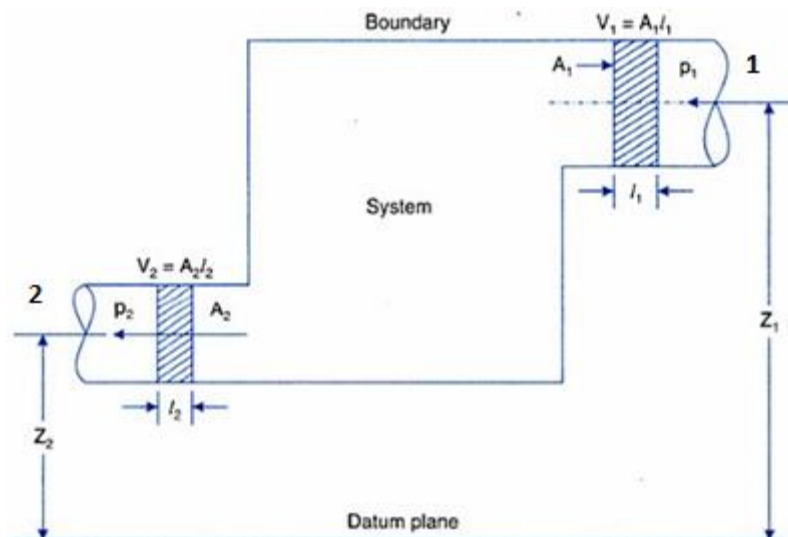
Fig 1.15

Consider a thermodynamic system in which a fluid is flowing at a steady rate, Fluid enters the system at the point 1 and leaves the system at the point 2

p_1 - pressure of working substance entering the system in N/m^2

v_1 - Specific volume of the working substance entering the system in m^3/kg

c_1 - velocity of working substance entering the system in m/s

u_1 - specific internal energy of working substance entering the system in J/ kg

Z_1 - height above datum level for inlet in meters

P_2, v_2, c_2, u_2 and Z_2 – Corresponding vales for the working substance leaving the system

Q_1 - heat supplied to the system in J/kg.

W   - Work delivered by the system in J/ kg

h_1 - Enthalpy of working substance entering the system in J/ kg

Q be the heat transfer through the control surface and W be the work transfer through the control surface

According to 1 st law of thermodynamics $Q = dU + WD$

$$Q = (U2 - U1) + WD$$

$$\mathbf{U1 + Q = U2 + WD}$$

Total energy entering the system = Total energy leaving the system

Total energy entering the system   = PE + KE + internal energy + Flow + Heat energy

$$= g.Z_1 \frac{C_1^2}{2} + u_1 + p_1v_1 + Q \quad = \quad g.Z_1 + \frac{C_1^2}{2} + h_{1} + Q$$

$$(h_1 = u_1 + p_1v_1)$$

Total energy leaving the system   *= PE + KE + internal energy + Flow + work*

$$= g.Z_2 + \frac{C_2^2}{2} + u_2 + p_2v_2 + W = g.Z_2 + \frac{C_2^2}{2} + h_{2} + W$$

$$(h_2 = u_2 + p_2v_2)$$

By the law of conservation of energy

Total energy entering the system = Total energy leaving the system

$$= g.Z_1 + \frac{C_1^2}{2} + h_1 + Q = g.Z_2 + \frac{C_2^2}{2} + h_{2} + W$$

The above equation is called steady flow energy equation.

1.4.3 Assumptions Made In the Steady Flow System Analysis

1. The mass flow rate through the control volume is constant.
2. Only potential, kinetic, flow and internal energies are taken into account and other energies such as electrical, magnetic and chemical are not considered.
3. The rate at which work and heat cross the control surface is constant.
4. The state of fluid at any point remains same at all times

1.4.4 Applications of Steady Flow Energy Equations in Engineering System

The SFEE is applied in the following engineering system

1. Steam boiler
2. Condenser
3. Nozzle
4. Steam and gas turbines
5. Reciprocating compressor
6. Rotary compressor

1. Steam boiler or generator

➢ Steam generator or boiler is a device which generates steam.
➢ In the steam generator, heat is transferred to water.
➢ The supplied water is converted into steam.
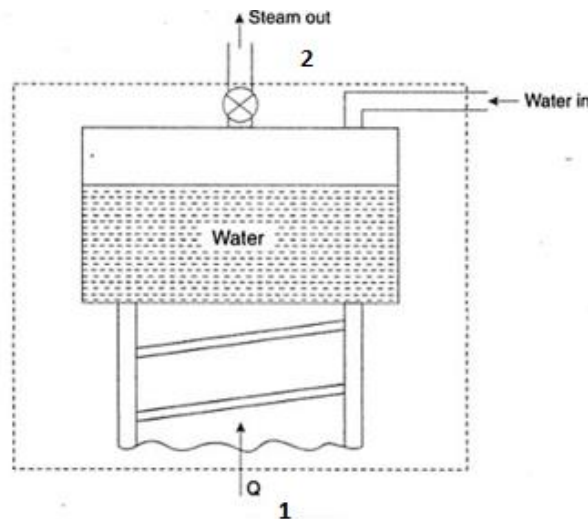
Fig 1.16

In the boiler

i. Potential energy between inlet and out let is negligible ie .$Z_1 = Z_2$

ii. Fluid velocity at inlet and outlet is negligible ie $C_1 = C_2$

iii. No mechanical work is done ie. $W=0$

Applying the SFEE to boiler, we get

$$h_1 + Q = h_2$$

Therefore $\qquad\qquad Q = h_2 - h_1$

ie. **Heat transfer** $\qquad Q = h_2 - h_1 \text{ KJ}$

2. Steam condenser

➢ In the steam condenser, the steam flows outside the tubes and cooling water flows inside the tubes. Due to the heat transfer steam condenses into water.
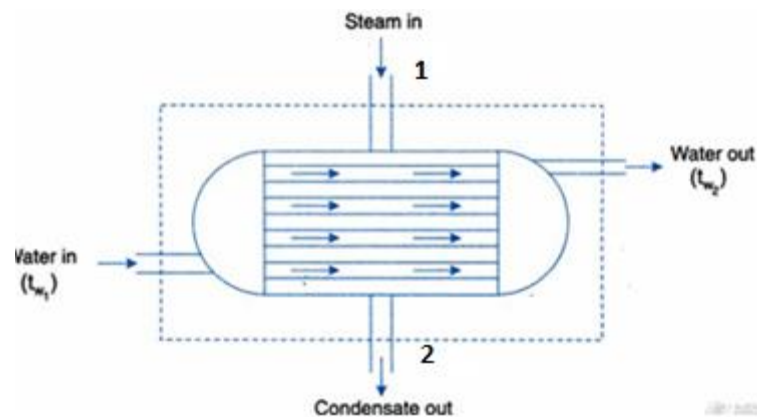
Fig 1.17

In a steam condenser

i. Potential energy between inlet and out let is negligible ie .$Z_1 = Z_2$

ii. No change in kinetic energy ie. $C_1 = C_2$

iii. No mechanical work is done ie. $W = 0$

Applying SFEE to surface condenser, we get

$$h_1 + Q = h_2$$

Therefore $\qquad\qquad\qquad Q = h_2 - h_1$

ie. Heat transfer $\qquad\qquad\qquad Q = h_2 - h_1 \text{ KJ}$

3.Nozzle

> Nozzle is a device which increases the velocity or kinetic energy on the expense of its pressure drop.
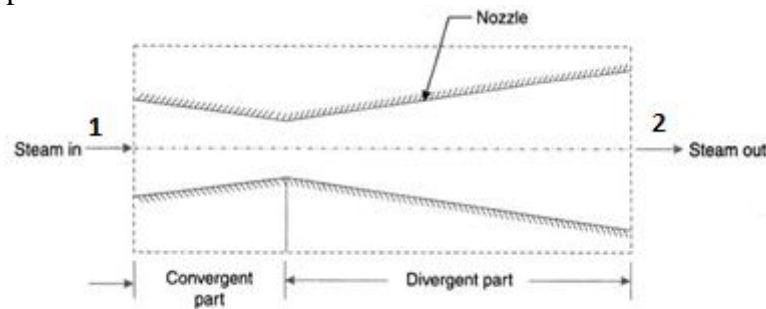
Fig 1.18

In the nozzle

i.   No change in potential energy

ii. No work done ie. W=0

iii. No heat transfer ie. Q = 0

Applying SFEE to nozzle, we get

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

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

$$C_2^2 - C_1^2 = 2\,(h_1 - h_2)$$

$$C_2^2 = 2\,(h_1 - h_2) + C_1^2$$

$$C_2 = \sqrt{2(h1 - h2) + C1^2}$$

UNIT – 2

THERMODYNAMIC AIR CYCLES AND FUELS & COMBUSTION

2.1 THERMODYNAMIC AIR CYCLES:

When air is assumed to be the working substance inside the engine cylinder, the cycle is called as an air cycle.

2.1.1 Air Standard efficiency:

The thermal efficiency of the engine which uses air as the working medium is know as 'Air standard Efficiency'. This efficiency is often called as Ideal efficiency

$$\text{Air standard } efficiency = \frac{\text{Work done}}{\text{Heat supplied}}$$

Work Done = Heat Supplied – Heat Rejected

$$= Q_s – Q_R$$

$$\text{Air standard } efficiency = 1 - \frac{Q_R}{Q_S}$$

2.1.2 Reversible and irreversible process:

A thermodynamic cycle consists of number of processes. If each process comprising the thermodynamic cycle is reversible, then the cycle is called as reversible cycle.

In a reversible cycle there should be no loss of heat due to friction, radiation.

The conditions required for reversible cycle are same as that for the reversible process. If those conditions are not satisfied, then it becomes an irreversible cycle. Hence in an irreversible cycle the original conditions are not restored.
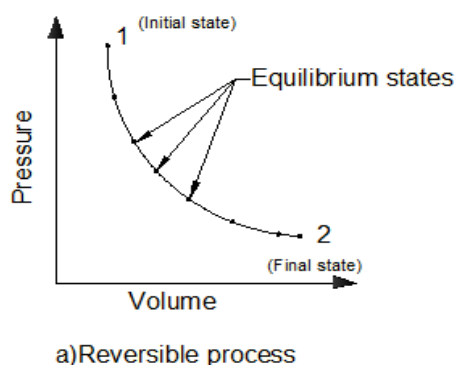

a)Reversible process   b)Irreversible process
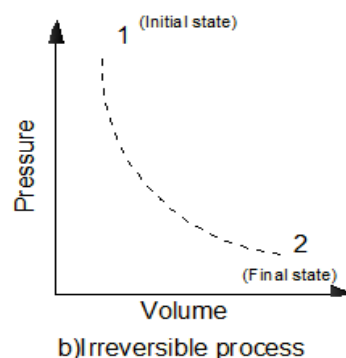
Fig 2.1(a)   Fig 2.1(b)

2.1.3 Assumptions in deriving air standard efficiency:

1. The working medium or air in the engine cylinder is a perfect gas i.e. It obeys the gas laws and has constant specific heats

2. The compression and the expansion process are adiabatic and they take place without internal friction

3. No chemical reaction takes place in the engine cylinder. Heat is supplied by bringing the hot body in contact with the engine cylinder and heat is rejected by bringing the cold body is assumed to be a closed system.

4. The mass of the working medium or air is assumed to be kept constant i.e the system is assumed to be a closed system.

2.1.4 CARNOT CYCLE

Consider 1 kg of air in a piston and cylinder arrangement as shown in fig.

Let p_1, v_1, and T_1 be the pressure, volume and temperature of the air at the beginning of the cycle
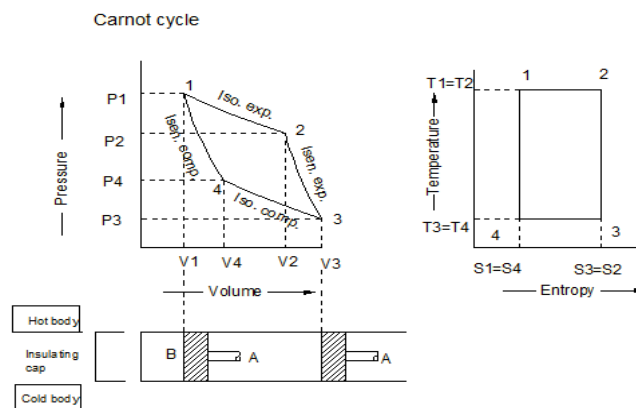

Fig 2.2 (a)   Fig 2.2 (b)

The Carnot cycle is completed with following process

Process 1-2 Isothermal expansion process

The hot body is brought in contact with the cylinder at the bottom. The air expands at constant tempt. Its volume increases from v_1 to v_2 and pressure decreases from p_1 to p_2. The process is represented by the curve pV and Ts diagram.

Heat added to the air during isothermal process is equal to the work done during the process.

$$\text{Heat supplied to the air } Q_s = p_1 V_1 \log_e \left(\frac{V_2}{V_1}\right) = RT_1 \log_e \left(\frac{V_2}{V_1}\right)$$

Process 2-3   adiabatic expansion process

The hot body is removed and an insulating cap is brought into contact with the cylinder. The air now expands adiabatically. The temperature decreases from T_2 to T_3 and pressure from p_2 to p_3.The volume of air increase from v_2 to v_3.

No heat is added or rejected by the air during this process **Q = 0**

Process 3-4   Isothermal compression process

The insulating cap is removed and the cold body is brought in contact with cylinder now. The air is compressed at constant tempt. Its volume decreases from v_3 to v_4 and pressure increase from p_3 to p_4 . Heat is rejected by the air during isothermal process is equal to work done during the process.

$$\text{Heat Rejected to the air } Q_R = p_3 V_3 \log_e \left(\frac{V3}{V4}\right) = RT_2 \log_e \left(\frac{V3}{V4}\right)$$

Process 4-1   Adiabatic compression process

The cold body is removed and an insulating cap is brought into contact with the cylinder. The air is now compressed adiabatically. The temperature increase from T_4 to T_1 and pressure from p_4 to p_1. The volume decreases from V_4 to V_1.

No heat is added or rejected by the air during this process Q = **0**

Derivation of air standard efficiency

Work Done = Heat Supplied – Heat Rejected

$$= Q_s - Q_R$$

$$= RT_1 \log_e \left(\frac{V2}{V1}\right) - RT_3 \log_e \left(\frac{V3}{V4}\right)$$

$$= RT_1 \log_e r - RT_3 \log_e r \qquad \text{where } r = \left(\frac{V2}{V1}\right) = \left(\frac{V3}{V4}\right)$$

$$= R \log_e r \, (T_1\text{-}T_3)$$

Air standard *efficiency* $= \dfrac{\text{Work done}}{\text{Heat supplied}}$

$$= \frac{R \log_e r \, (T1-T3)}{RT1 \log_e r}$$

Air standard *efficiency* $= \dfrac{(T1-T3)}{T1} = 1 - \dfrac{T3}{T1}$

Note:

T_3 – Minimum Temperature

T_1 – Maximum Temperature

2.1.5 OTTO CYCLE

These days, many gas, petrol and many of the oil engines run on this cycle. It is the also known as constant volume cycle, as the heat is received and rejected at a constant volume.

The engine conceived by Otto has air enclosed in a cylinder, whose walls are perfectly non-conductor of heat, but the bottom is a perfect conductor of heat. There is also a hot body and cold.

The ideal Otto cycle consists of two constant volume and two reversible adiabatic or isentropic processes as show on p-V and T-S diagrams in Fig. (a) and (b)
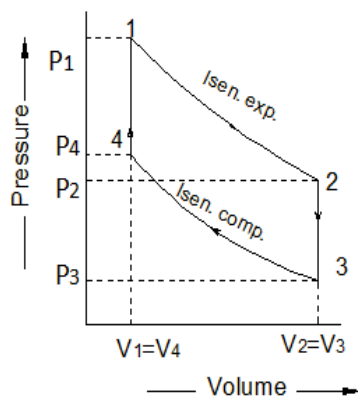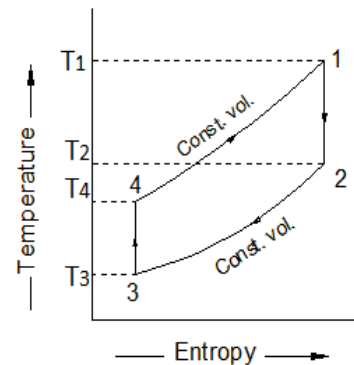
Fig 2.3 (a)   Fig 2.3 (b)

Let the engine cylinder contain m kg of air at point 1. At this point, let $p_1 T_1$ and V_1 be the pressure temperature and volume of the air. Following are the four stages of the ideal cycle.

Isentropic expansion: The air is expanded reversibly and adiabatically from initial temperature T_1 to a temperature T_2 as shown by the curve 1-2 in Fig. In this process, no heat is absorbed or rejected by the air.

Constant volume cooling: The air is cooled at constant volume from Temperature T_2 to a temperature T_3 as shown by the curve 2-3 in Fig. We know that heat rejected by the air during this process.

$$Q_{2-3} = m\ Cv\ (T_2 - T_3)$$

Isentropic compression : The air is compressed Reversibly and adiabatically from temperature T_3 to a temperature T_4 as shown in by the curve 3-4 in Fig. In this process, no heat is absorbed or rejected by the air.

Constant volume heating; the air is now heated at constant volume from Temperature T_4 to a temperature T_1 as shown by the curve 4-1 in Fig. We know that heat absorbed by the air during this process,

$$Q_{4-1} = m\ C_v\ (T_1 - T_4)$$

We know that work done = Heat absorbed − Heat rejected

$$= m\ C_v\ (T_1 - T_4) - m\ Cv\ (T_2 - T_3)$$

Therefore ideal efficiency or air standard efficiency,

$$\eta = \frac{\text{Work done}}{\text{Heat absorbed}}$$

$$= \frac{m\ Cv\ (T1 - T4) - m\ Cv\ (T2 - T3)}{m\ Cv\ (T1 - T4)}$$

$$= 1 - \frac{T2 - T3}{T1 - T4} = 1 - \frac{T3\left(\frac{T2}{T3} - 1\right)}{T4\left(\frac{T1}{T4} - 1\right)}$$

We know that for reversible adiabatic or isentropic expansion process 1-2,

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

$$r = Expansion\ ratio = \frac{V2}{V1}$$

similarly, for reversible adiabatic or isentropic compression process 3-4,

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

$$r = compression\ ratio = \frac{V3}{V4} = \frac{V2}{V1}$$

From equations (ii) and (iii), we find that

$$\frac{T2}{T1} = \frac{T3}{T4} = \left(\frac{1}{r}\right)^{\gamma - 1} = 1/(r)^{\gamma - 1}\ or\ \frac{T1}{T4} = \frac{T2}{T3}$$

Substituting the value of T_1/T_4 in equation (i),

$$\eta = 1 - \frac{T3}{T4} = 1 - \frac{T2}{T1} = 1 - 1/(r)^{\gamma - 1}$$

Note: 1. We see from equation (iv) that the efficiency of Otto cycle depends on compression ratio (r) only.

2. The efficiency increases with the increase in compression ratio (r). In actual practice, r cannot be increased beyond a value of 7 or so.

3. Compression ratio, r= Total cylinder volume / Clearance volume

$$= \text{clearance volume + stroke volume /Clearance volume}$$

$$= \frac{Vc + Vs}{Vc}$$

2.1.6 DIESEL CYCLE

This is an important cycle on which all the diesel engines work. It is also known as *constant pressure cycle* as heat is received at constant pressure.

The engine imagined by Diesel has air enclosed in the cylinder, whose walls are perfectly non – conductor of heat, but bottom is a perfect conductor of heat. Again, there is a hot body, cold body and an insulating cap, which are alternately brought in contact with the cylinder.

The ideal diesel cycle consists of two reversible adiabatic or isentropic, a constant pressure and a constant volume processes. These processes are represented on p–v and T-S diagrams as shown in figure.
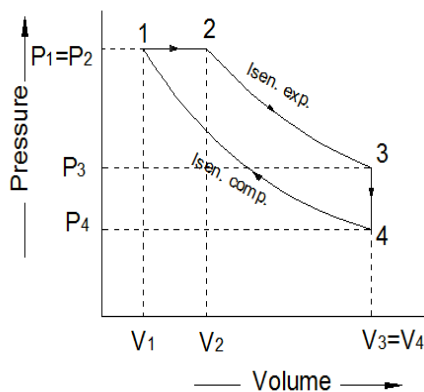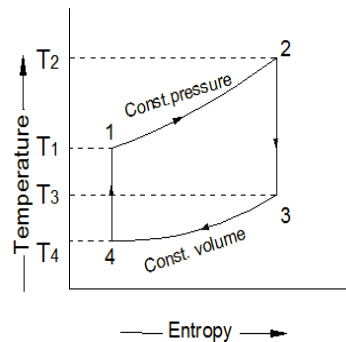

Fig 2.4 (a)   Fig 2.4 (b)

Let the engine cylinder contain m kg of air at point 1. At this point let, p_1, T1 and V_1 be the pressure, temperature and volume of the air respectively. Following are four stages of an ideal diesel cycle.

Constant pressure heating:-

The air heated at constant pressure from initial temperature T_1 to a temperature T_2 represented by the curve 1-2 in fig.

$$\therefore \qquad \text{Heat supplied to the air,}$$

$$Q_{1\text{-}2} = m\ Cp\ (T_2 - T_1)$$

Note: Since the supply of heat is cut off at point 2, therefore it is known as cut-off point.

Adiabatic expansion:-

The air is expanded adiabatically from temperature T_2 to a temperature T_3 as shown by the curve 2-3 in the fig. In this process, no heat is absorbed or rejected by the air.

Constant volume cooling:-

The air now cooled at constant volume from temperature T_3 to a temperature T_4 as shown by the curve 3-4.

$$\therefore \qquad \text{Heat rejected to the air,}$$

$$Q_{3\text{-}4} = m\ Cv\ (T_3 - T_4)$$

Adiabatic compression:-

The air is compressed adiabatically from temperature T_4 to a temperature T_1 represented by the curve 4-1. In this process, no heat is absorbed or rejected by the air.

$$\text{Work done} = \text{Heat absorbed} - \text{Heat rejected}$$

$$= m\ Cp\ (T2 - T1) - m\ Cv\ (T3 - T4)$$

$\therefore$ Air standard efficiency,

$$\eta = \frac{Work\ done}{Heat\ absorbed} = \frac{m\ Cp\ (T2 - T1) - m\ Cv\ (T3 - T4)}{m\ Cp\ (T2 - T1)}$$

$$= 1 - \frac{Cv}{Cp}\left(\frac{T3-T4}{T2-T1}\right) = 1 - \left(\frac{1}{\gamma}\right)\left(\frac{T3-T4}{T2-T1}\right) \qquad \dots\ (i)$$

Now let compression ratio,

$$r = \frac{V4}{V1}$$

Cut-off ratio $\quad \rho = \frac{V2}{V1}$

Expansion ratio $= \frac{V3}{V2} = \frac{V4}{V2}$

$$= \frac{V4}{V1} \times \frac{V1}{V2} = r \times \frac{1}{\rho} = \frac{r}{\rho}$$

We know for constant pressure heating process 1-2,

$$= \frac{V1}{T1} = \frac{V2}{T2} \qquad \text{....(Charle`s Law)}$$

$$= T2 = T1 \times \frac{V2}{V1} = T1 \times \rho \quad \text{..... (ii)}$$

Similarly, for adiabatic expansion process 2-3,

$$\frac{T3}{T2} = \left(\frac{V2}{V3}\right)^{\gamma-1} = \left(\frac{1}{r1}\right)^{\gamma-1} = \left(\frac{\rho}{r}\right)^{\gamma-1}$$

$$T3 = T2\left(\frac{\rho}{r}\right)^{\gamma-1} = T1 \times \rho\left(\frac{\rho}{r}\right)^{\gamma-1} \qquad \text{.... (iii)}$$

And for adiabatic compression process 4-1,

$$\frac{T1}{T4} = \left(\frac{v4}{v1}\right)^{\gamma-1} = (r)^{\gamma-1} \text{ or } T1 = T4(r)^{\gamma-1} \text{ .... (iv)}$$

Substituting the values of T_1 in equations (ii) and (iii),

$$T2 = T4(r)^{\gamma-1} \times \rho \qquad \text{.... (v)}$$

$$T3 = T4(r)^{\gamma-1} \times \rho\left(\frac{\rho}{r}\right)^{\gamma-1} = T4\rho^{\gamma} \qquad \text{.... (vi)}$$

Now substituting the values of T_1, T_2 and T_3 in equation (i),

$$\eta = 1 - \frac{1}{\gamma} \left[\frac{(T4\rho^{\gamma})-T4}{(T4(r)^{\gamma-1}\rho-T4(r)^{\gamma-1}}\right]$$

$$\eta = 1 - \frac{1}{(r)^{\gamma-1}}\left[(\rho^{\gamma} - 1)/\gamma(\rho - 1)\right] \qquad \text{.... (vii)}$$

2.1.7 *Comparison of Otto* and *Diesel cycle*

Otto cycle	Diesel cycle
Heat addition takes place at constant volume.	Heat addition takes place at constant pressure
Petrol engines work on this cycle.	Diesel engines work on this cycle.
At constant volume, heat rejection takes place.	In diesel cycle also the heat rejection takes place at constant volume.
Compression ratio is less. It is 7:1 to 10:1.	Compression ratio is more. It is 11:1 to 22:1.
Efficiency is less.	Efficiency is more
Adiabatic expansion takes place during the complete backward stroke of the piston.	After the heat addition is cut-off in the backward stroke, the adiabatic expansion takes place during the remaining portion of stroke

2.1.8 Comparison of ideal and actual p-v diagrams of Otto cycle

Ideal	Actual
It is assumed the inlet and exhaust valves open and close suddenly	The inlet and exhaust valves do not open and close suddenly, but they open and close gradually.
It is assumed the air – fuel mixture is ignited suddenly	The air – fuel mixture is ignited with some time lag.
It is assumed , the compression and expansion processes are frictionless adiabatic process	The compression and expansion are not frictionless adiabatic process
It is assumed combustion takes place at constant volume	Combustion does not take place at constant volume
It is assumed the specific heats of working fluid is constant	The specific heats of the working fluid are not constant but increases with temperature
The pressure during suction stroke is atmospheric pressure	The pressure during the suction stroke is less than the atmospheric pressure
The exhaust takes place at atmospheric pressure	The pressure during the exhaust stroke is above the atmospheric pressure

2.1.9 Comparison of ideal and actual p-v diagrams of Diesel cycle

Ideal	Actual
It is assumed the inlet and exhaust valves open and close suddenly	The inlet and exhaust valves do not open and close suddenly, but they open and close gradually.
It is assumed the cut off will take place suddenly	The cut off will not take place suddenly
It is assumed , the compression and expansion processes are frictionless adiabatic process	The compression and expansion are not frictionless adiabatic process
It is assumed the heat supplied at constant pressure	Combustion does not take place at constant pressure
It is assumed the specific heats of working fluid is constant	The specific heats of the working fluid are not constant but increases with temperature
The pressure during suction stroke is atmospheric pressure	The pressure during the suction stroke is less than the atmospheric pressure
The exhaust takes place at atmospheric pressure	The pressure during the exhaust stroke is above the atmospheric pressure

Problems

1. A Carnot engine, working between 650 K and 310 K, produces 150 kJ of work. Find thermal efficiency and heat added during this process.

Given:

$T_1 = 650$ K; $T_3 = 310$ K; W = 150 kJ

To find:

Thermal efficiency & heat added

Solution

Thermal efficiency

WKT the thermal efficiency

$$\eta = \frac{T_1 - T_2}{T_1} = \frac{650 - 310}{650} = 0.523 \text{ or } 52.3 \%$$

Heat added during the process

WKT the heat added during the process

$$Q_{1\text{-}2} = \frac{W}{\eta} = \frac{150}{0.523} = 286.8 \text{ KJ}$$

2. A Carnot engine operates between two reservoirs at temperature T1 and T3. The work output of the engine is 0.6 times the heat received. The difference in temperature between the source and the sink is 200⁰c. Calculate the thermal efficiency, source temperature and the sink temperature.

Given:

W= 0.6 × Heat rejected = 0.6 $Q_{3\text{-}4}$; T_1-T_3 =200⁰C

To find:

Thermal efficiency & Source and Sink temperature

Solution

Thermal efficiency

WKT the thermal efficiency

$$\eta = \frac{Work\ done}{Heat\ supplied} = \frac{Work\ done}{Workdone + Heat\ supplied}$$

$$\frac{0.6Q3-4}{0.6Q3-4+Q3-4} = \frac{0.6}{1.6} = 0.375$$

Source and Sink temperature

Let T_1 = Source temperature, and

                                       T_3 = Sink temperature.

WKT the thermal efficiency

$$0.375 = \frac{T1-T2}{T1} = \frac{200}{T1}$$

$$T_1 = 200/0.375 = 533.3\ K = 260.3^0\ C$$

$$T3 = T_1 - 200 = 260.3 - 200 = 60.3^0C$$

3. An engineer claims his engine to develop 3.75 KW. On testing, the engine consumes 0.44 kg fuel per hour having a calorific value of 42000 KJ/kg. The maximum temperature recorded in the cycle is 1400°C and minimum is 350°C. Find whether the engineer is justified in his claim.

Given:

P= 3.75 kW; Fuel consumed = 0.44 kg/h; calorific value – 42000kJ/kg

T1 = 1400^{0}C = 1400 + 273 =1673 K; T3 =350+273 = 623 K

Wkt the maximum efficiency, between two specified temperatures, is that of Carnot cycle,

$$\eta_{carnot} = \frac{T1-T3}{T1} = \frac{1673-623}{1673} = 0.627\ \%\ or\ 62.7\%$$

We also know that the heat supplied to the engine by the fuel

= Fuel consumed × Calorific value of fuel

= 0.44 × 42000 = 18480 kJ/h 5.13 kJ/s

Workdone by the engine   =3.75 KW= 3.75 kJ/s

Efficiency claimed $= \frac{Work\ done}{Heat\ supplied} = \frac{3.75}{5.13} = 0.731$ or 73.1%

Since efficiency of the engine claimed (73.1%) is more than the maximum possible efficiency (62.7%), therefore the engineer is not justified in his claim.

4. In an Otto cycle, the temperature at the beginning and end of the isentropic compression are 316K and 596K respectively. Determine the air standard efficiency and the compression ratio. Take $\gamma = 1.4$

Given:

 T3=316K   T4=596K $\gamma = 1.4$

To find:

        Air standard efficiency & compression ratio

Solution

Compression ratio

Let   r=compression ratio =v3/v4

We know that for isentropic compression 3-4 (Refer Fig. 6.9)

$$\frac{T3}{T4} = \left(\frac{v4}{v3}\right)^{\gamma-1} = \left(\frac{1}{r}\right)^{1.4-1} = 1/(r)^{0.4}$$

Therefore $(r)^{0.4} = \frac{T4}{T3}$ $or\ r = \left(\frac{T4}{T3}\right)^{\frac{1}{0.4}} = \left(\frac{596}{316}\right)^{2.5} = 4.885$

Air standard effieciency

        We knows that air standard effieciency

$$\eta = 1/(r)^{\gamma-1} = 1 - \frac{1}{(4.885)^{1.4-1}} = 1 - \left(\frac{1}{1.886}\right)$$

$$= 1\text{-}0.53$$
$$= 0.47\ \%\quad or\quad 47\%$$

5. A certain quality of air at a pressure of 1 bar and temperature 70° C is compressed reversibly and adiabatically until the pressure is 7 bar in an Otto cycle engine. 460 kJ of heat per kg of air is now added at constant volume. Determine: 1. Compression ratio of the engine, 2. Temperature at the end of compression, and 3. Temperature at the end of heat heat addition.

Take for air, C_p = 1 kJ/kg K and C_v = 0.707 kJ/kg K.

Given:

p_3 = 1 bar

T_3 = 70° C = 70 + 273= 343 K

p_4 = 7 bar

Q_{4-1} = 460 kJ

m = 1 kg

C_p = 1 kJ/kg K

C_v = 0.707 kJ/kg K

To find:

1. Compression ratio of the engine
2. Temperature at the end of compression, and
3. Temperature at the end of heat heat addition.

Solution:

We know that ratio of specific heats,

$$\gamma = \frac{c_p}{c_v} = 1/0.707 = 1.41$$

1. Compression ratio of the engine

Let   r = Compression ratio of the engine = $\frac{v_3}{v_4}$

We know that $p_3 V_3^\gamma = p_4 V_4^\gamma$

$$\frac{V_3}{V_4} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \quad \text{or} \quad r = \left(\frac{7}{1}\right)^{\frac{1}{1.41}} = (7)^{0.709} = 3.97 \textbf{ Ans.}$$

2. Temperature at the end of compression

Let T_4 = Temperature at the end of compression.

We know that $\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = \left(\frac{1}{r}\right)^{\gamma-1} = \left(\frac{1}{3.97}\right)^{1.41-1}$

$$= (0.252)^{0.41} = 0.568$$

$$T_4 = T_3/0.568 = 343 / 0.568 = 604 \text{ K}$$

$$= 331° \text{ C } \textbf{Ans.}$$

3. Temperature at the end of heat addition

Let $T_1 =$ Temperature at the end of heat addition.

We know that heat added at constant volume (Q_{4-1}),

$$460 = m \, c_v \, (T_1 - T_4) = 1 \times 0.707 \, (T_1 - 604) \text{ kJ}$$

$$T_1 - 604 = 460/0.707 = 651 \quad \text{or} \quad T_1 = 1255 \text{ K}$$
$$= 982° \text{ C } \textbf{Ans.}$$

6.In an air standard Otto cycle, the compression ratio is 7 and the compression begins at 1 bar and 313 K. The heat added is 2510 kJ/kg. Find: 1. Maximum temperature and pressure of the cycle; 2. Work done per kg of air; 3.Cyclic efficiency

Take for air, $c_v = 0.713$ kJ/kg K and R = 287 J/kg K.

Given:

$$r = \frac{V_3}{V_4} = \frac{V_2}{V_1} = 7$$

$p_3 = 1$ bar $= 1 \times 10^6$ N/m^2

$T_3 = 313$ K

$Q_{4-1} = 2510$ kJ/kg

Cv = 0.713 kJ/kg K

R = 287 J/kg K = 0.287 kJ/kg K

To find:

1. Maximum temperature and pressure of the cycle
2. Work done per kg of air
3. Cyclic efficiency

Soln:

We know that Cp – Cv = R or Cp = R + Cv = 0.287 + 0.713 = 1 kJ/kg K

And $\gamma = $ Cp/Cv =- 1 / 0.713 = 1.4

1. Maximum temperature and pressure of the cycle

Let $T_1 = $ Maximum temperature, and

$p_1 = $ Maximum pressure

We know that for isentropic compression process 3-4

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_1}\right)^{\gamma-1} = \left(\frac{1}{\gamma}\right)^{\gamma-1} = \left(\frac{1}{7}\right)^{1.4-1} = \frac{1}{2.178}$$

$$T_4 = T_3 \times 2.178 = 313 \times 2.178 = 682 \text{ K}$$

$$p_3 V_3^{\gamma} = p_4 V_4^{\gamma} \quad \text{or } p_4 = p_3 \left(\frac{V_3}{V_4}\right)^{\gamma} = 1 \times 7^{1.4} = 15.24 \text{ bar}$$

We know that heat supplied during constant volume process 4-1 ($Q_{4\text{-}1}$)

$$2510 = m\, c_v\, (T_1 - T_4) = 1 \times 0.713\, (T_1 - 682) \qquad\qquad \ldots (m = 1 \text{ kg})$$

$$T_1 - 682 = 2510/\, 0.713 = 3520 \text{ or } T_1 = 4202 \text{ K } \textbf{Ans}.$$

We also know that $\frac{p_4}{T_4} = \frac{p_1}{T_1}$ or $p_1 = p_4 \times \frac{T_1}{T_4} = 15.24 \times \frac{4202}{682} = 93.9$ bar **Ans.**

2. Work done per kg of air

We know that for isentropic expansion process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{V}{V_2}\right)^{\gamma-1} = \left(\frac{1}{\gamma}\right)^{\gamma-1} = \left(\frac{1}{7}\right)^{1.4-1} = \frac{1}{2.178}$$

$$T_2 = T_1 / 2.178 = 4202 / 2.178 = 1929.29 \text{ K}$$

And heat rejected per kg of air,

$$Q_{2\text{-}3} = m\, c_v\, (T_2 - T_3) = 1 \times 0.713\, (1929 - 313) = 1152 \text{ kJ}$$

Work done per kg of air,

$$W = \text{Heat supplied} - \text{Heat rejected}$$

$$= Q_{4\text{-}1} - Q_{2\text{-}3}$$

$$= 2510 - 1152 = 1358 \text{ kJ } \textbf{Ans.}$$

3. Cycle efficiency

We know that cycle efficiency,

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{1358}{2510} = 0.541 \text{ or } 54.1 \text{ \% } \textbf{Ans.}$$

Note: The cycle efficiency may also be calculated as follows:

We know that $\eta = 1 - \dfrac{1}{(r)^{\gamma-1}} = 1 - \dfrac{1}{(7)^{1.4-1}} = 1 - \dfrac{1}{2.178}$

$$= 0.541 \text{ or } 54.1 \text{ \% } \textbf{Ans.}$$

7. The compression ratio of an ideal air standard Diesel cycle is 15. The heat transfer is 1465 kJ/kg K , if inlet conditions are 300 K and pressure 1 bar. Find the pressure and temperature at the end of each process and determine the cycle efficiency.
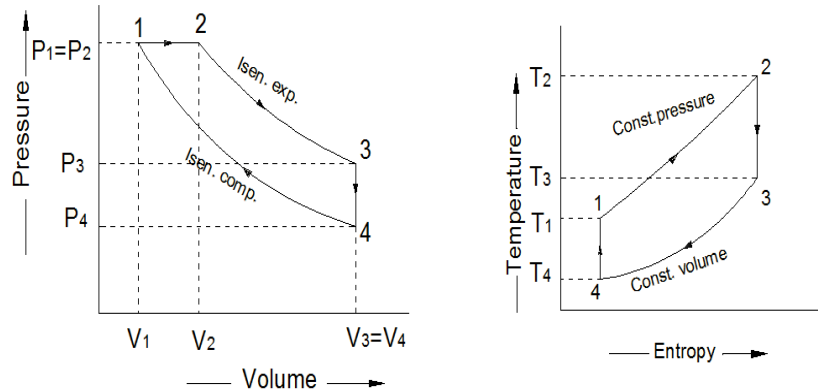
Given:-

$r = V_4/V_1 = 15$;

Q1-2 = 1475 kJ/kg K;

T$_4$ = 300 K;

p$_4$ = 1 bar = $0.1 \times 10^6 \frac{N}{m^2}$

To find:

1. Pressure and temperature at the end of each process
2. Cycle efficiency

Solution:-

Pressure and temperature at the end of each process

The p-V and T-s diagram for the Diesel cycle is shown in figure.

Let   p$_1$, p$_2$, p$_3$ = pressures at point 1, 2 and 3 respectively.

T$_1$, T$_2$, T$_3$ = Temperatures at the point 1, 2, 3 respectively.

First of all consider the isentropic process 4-1. We know that

$$p4V4^{\gamma} = p1V1^{\gamma} = p4\left(\frac{V4}{V1}\right)^{\gamma} = 1(15)^{1.4} = 44.3\ bar$$

And
$$\frac{T1}{T4} = \left(\frac{V4}{V1}\right)^{\gamma-1} = (15)^{1.4-1} = (15)^{0.4} = 2.954$$

∴
$$T1 = T4 \times 2.954 = 300 \times 2.954 = 886.2\ K$$

Now consider the constant pressure process 1-2,We know that

$$p_2 = p_1 = 44.3\ \text{bar}$$

and heat supplied per kg of air during constant pressure process 1-2 (Q_{1-2}),

$$1464 = m\ Cp(T2 - T1) = 1 \times 1(T2 - 886.2) = T2 - 886.2$$

……. (Taking Cp for air = 1kJ/kg K)

$$T_2 = 1465 + 886.2 = 2351.2\ K$$

Now consider the isentropic expansion process 2-3. First of all, let us find the volumes at points 2 and 3.

Let V_2 = Volume at point 2 and

V_3 = volume at point 3.

We know that,

$$p4V4 = mRT4\ or\ V4 = \frac{mRT4}{p4} = \frac{1 \times 287 \times 300}{0.1 \times 10^6} = 0.861 m^3$$

.. (Since R for air = 287 J/kg K)

$$V_3 = 0.861 m^3 \qquad \text{…. (Since } V_3 = V_4\text{)}$$

and
$$V_1 = V_4/15 = 0.861/15 = 0.0574 m^3 \quad \text{…. (Since V4/V1 = 15)}$$

We also know that for the constant pressure process 1-2,

$$\frac{V1}{T1} = \frac{V2}{T2}\ or\ V2 = V1 \times \left(\frac{T2}{T1}\right) = 0.0574 \times \left(\frac{2351.2}{886.2}\right) = 0.1523 m^3$$

Now
$$\frac{T2}{T3} = \left(\frac{V3}{V2}\right)^{\gamma-1} = \left(\frac{0.861}{0.1523}\right)^{1.4-1} = (5.65)^{0.4} = 2$$

$$T3 = \frac{T2}{2} = \frac{2351.4}{2} = 1175.6\ K$$

We know that $p_2 V_2^\gamma = p_3 V_3^\gamma$ or $p_3 = p_2 \left(\frac{V_2}{V_3}\right)^\gamma = 44.3 \left(\frac{0.1523}{0.861}\right)^{1.4} = 3.92\ bar$

Cycle efficiency,

We know that heat rejected per kg of air during the constant volume process 3-4,

$$Q_{3-4} = mC_v(T_3 - T_4) = 1 \times 0.712(1175.6 - 300) = 623.4 kJ$$

… (Taking Cv for air = 0.712 kJ/kg K

∴ Cycle efficiency,

$$\eta = \frac{Heat\ supplied - Heat\ rejected}{Heat\ supplied} = \frac{1465 - 623.4}{1465} = 0.5745\ or\ 57.45\%$$

8. An ideal diesel cycle has a diameter of 150 mm and stroke 200 mm. The clearance volume is 10 percent of the swept volume. Determine the compression ratio and the air standard efficiency of the engine if the cut-off takes place at 6 percent of the stroke.

Given:-

       d = 150mm;

       l = 200mm = 0.2 m;

       V_c = 10% of vs = 0.1 vs;

find:-

    1. Compression ratio
    2. Air standard efficiency

Solution:-

Compression ratio

       We know that stroke volume

$$V_s = \frac{\pi}{4} \times d^2 \times l = \frac{\pi}{4}(0.15)^2 0.2 = 3.53 \times 10^{-3}\ m^3$$

$$V_c = 0.1\ vs = 0.10.1 \times 3.53 \times 10^{-3} = 0.353 \times 10^{-3}\ m^3$$

       We know that compression ratio,

$$r = \frac{Total\ volume}{Clearance\ Volume} = \frac{Vc + Vs}{Vc} = \frac{0.353 \times 10^{-3} + 3.53 \times 10^{-3}}{0.353 \times 10^{-3}}$$

$$r = 11$$

Air standard efficiency,

Since the cut-off takes place at 6% of the stroke, therefore volume at cut-off,

$$V_2 = V_1 + 0.06vs = V_c + 0.06vs \qquad \text{…(Since } V_1 = V_c)$$

$$= 0.353 \times 10^{-3} + 0.06 \times 3.53 \times 10^{-3} = 0.565 \times 10^{-3} m^3$$

$\therefore Cut - off\ ratio,$ $\qquad \rho = \dfrac{V2}{V1} = \dfrac{V2}{Vc} = \dfrac{0.565 \times 10^{-3}}{0.353 \times 10^{-3}} = 1.6$

We know that air standard efficiency,

$$\eta = 1 - \frac{1}{(r)^\gamma}\left[\frac{\rho^{\gamma-1}}{\gamma(\rho-1)}\right] = 1 - \frac{1}{(11)^{1.4-1}}[(1.6)^{1.4} - 1/1.4(1.6-1)]$$

$$= 1 - \frac{1}{2.61} \times 1.11 = 1 - 0.4246 = 0.5753\ or\ 57.53\%$$

2.1 FUELS AND COMBUSTION

Fuel is a substance which, when burnt, i.e. on coming in contact and reacting with oxygen or air, produces heat. Thus, the substances classified as fuel must necessarily contain one or several of the combustible elements: carbon, hydrogen, sulphur, etc. In the process of combustion, the chemical energy of fuel is converted into heat energy.

To utilize the energy of fuel in most usable form, it is required to transform the fuel from its one state to another, i.e. from solid to liquid or gaseous state, liquid to gaseous state, or from its chemical energy to some other form of energy via single or many stages. In this way, the energy of fuels can be utilized more effectively and efficiently for various purposes. The fuels may be classified into the following three general types.

1. *Solid fuels*
2. *liquid fuels and*
3. *Gaseous fuels*

1. SOLID FUELS

Solid fuels are mainly classified into two categories, i.e. natural fuels, such as wood, coal, etc. and manufactured fuels, such as charcoal, coke, briquettes, etc.

The various advantages and disadvantages of solid fuels are given below:

Advantages

(a) They are easy to transport.
(b) They are convenient to store without any risk of spontaneous explosion.
(c) Their cost of production is low.
(d) They posses moderate ignition temperature.

Disadvantages

(a) Their ash content is high.

(b) Their large proportion of heat is wasted.

(c) They burn with clinker formation. (d) Their combustion operation cannot be controlled easily.

(e) Their cost of handling is high.

Wood

It consists of mainly carbon and hydrogen. It is used as a domestic fuel. The calorific value of wood varies with its kind and moisture content. The calorific value of wood is nearly 20000KJ/Kg.

Lignite or brown coal

It is an intermediate variety between peat and bituminous coal. It contains nearly 40% moisture and 60% of carbon. It has a brown fibrous structure. They are either caking or non-caking. Its calorific value is nearly 33500KJ/Kg.

Charcoal

It is prepared by the dry distillation of wood. It is a good prepared solid fuel and is used for various metallurgical processes.

Coke

It is produced when coal is strongly heated continuously in the absence of air in a closed vessel. Coke is dull back in color, porous and smokeless. It has a high carbon content and has higher calorific value. It is used as fuel in blast furnace and cupolas.

Briquetted coal

It is produced from the finely ground coal by moulding under pressure with or without a binding material such as pitch, coal tar clay or crude oil. As the briquetted coal is in the form blocks, it has the advantage of no loss of fuel through grate opening. Thus it increases the net calorific value of the fuel

2. Liquid fuels

The liquid fuels can be classified as follows : (a) Natural or crude oil, and (b) Artificial or manufactured oils.

The advantages and disadvantages of liquid fuels can be summarized as follows :

(a) They posses higher calorific value per unit mass than solid fuels.

(b) They burn without dust, ash, clinkers, etc.

(c) Their firing is easier and also fire can be extinguished easily by stopping liquid fuel supply.

(d) They are easy to transport through pipes.

(e) They can be stored indefinitely without any loss.

(f) They are clean in use and economic to handle.

Disadvantages

(a) The cost of liquid fuel is relatively much higher as compared to solid fuel.

(b) Costly special storage tanks are required for storing liquid fuels.

(c) There is a greater risk of five hazards, particularly, in case of highly inflammable and volatile liquid fuels.

(d) They give bad odour.

(e) For efficient burning of liquid fuels, specially constructed burners and spraying apparatus are required.

Petrol or gas online

It is the lightest and most volatile liquid fuel. This is used in petrol engines. It's calorific value is nearly 45000KJ/Kg.

Kerosene or paraffin oil

It is heavier and less volatile fuel than petrol. It is used as heating and lighting fuel. It is used in farm tractors and gas turbines as fuel oil.

3. Gaseous fuels

Gaseous fuels occur in nature, besides being manufactured from solid and liquid fuels.

Advantages

Gaseous fuels due to erase and flexibility of their applications, possess the following advantages over solid or liquid fuels :

(a) They can be conveyed easily through pipelines to the actual place of need, thereby eliminating manual labour in transportation.

(b) They can be lighted at ease.

(c) They have high heat contents and hence help us in having higher temperatures.

(d) They can be pre-heated by the heat of hot waste gases, thereby affecting economy in heat.

(f) They are clean in use.

(g) They do not require any special burner.

Disadvantages

(a) Very large storage tanks are needed.

(b) They are highly inflammable, so chances of fire hazards in their use is high

2.1.1 Requirement of a Good Fuel

> Should have high calorific value

> Should have low ignition temperature

> Should produce less smoke

> Should be economical

> Should be easily stored and transported

> Should burn freely if ignited

2.1.2 OCTANE VALUE OF FUELS

The octane value of knock rating of a spark ignition engine fuel gives us an idea of the tendency of the fuel to donate. To find it out, two fuels – the iso – octane and the normal heptane are selected for this purpose under modern practice. The former has little tendency towards detonation while the latter readily detonates, the actual fuel whose octane value is to be determined is first used to run a standard engine and the conditions under which detonation occurs are noted. The detonation intensity is measured by an instrument known as knock meter. Then several mixtures of iso – octane and normal heptane mixed up in different proportions are used in the same engine under the same conditions, as before. By making several trials, the mixture which gives the same intensity of detonation is found out. The percentage of iso – octane in this mixture gives the octane value of the fuel, e.g., with a mixture containing 65% of isi – octane and 35% of normal heptane, the octane value of the fuel will be 65. Higher this value, higher the anti – knock property of the fuel. The octane value of pure iso – octane and normal heptane are 100 and 0 respectively. The octane rating indicates the quality of the petrol to resist detonation.

2.1.3 DETONATION:

At present the amount of power that can be developed in the cylinder of a petrol engine is fixed by the liability of a fuel to detonate, i.e., just before the flame has completed its course across the combustion chamber and remaining unburnt charge fires throughout its mass spontaneously without external assistance.

The result is a tremendously rapid and local increase in pressure which sets up pressure waves that hit the cylinder walls with such violence that the walls emit a sound like a 'ping'. It is the ping that manifests detonation. Thus a very sudden rise of pressure during combustion accompanied by metallic hammer like sound is called **Detonation.**

The region in which detonation occurs is farthest removed from the spark plug, and is named the "detonation zone" and even with severe detonation this zone is rarely more than one quarter the clearance volume.

2.1.4 CETANE VALUE OF FUELS

The tendency of compression ignition engine fuel to knock is measured by its cetane value which may be defined as the percentage volume of normal cetane in a mixture of normal cetane and alpha methyl naphthalene which gas the same tendency to knock as the fuel under examination. Normal cetane has a good ignition quality and a short ignition lag but methyl naphthalene possesses poor ignition quality and longitudinal lag. The cetane value can be improved by the addition of dops like ethyl nitrate but because of their high cost these are not used. The cetane levels for high speed engines are about 45 – 50. In general, a high octane value implies a low cetane value.

2.1.5 DIESEL KNOCK

Just like petrol engines, a high pressure wave set up in compression ignition engines may also cause knock the phenomenon being called Diesel Knock results in a drop in power and efficiency in addition to rough running. In several cases, it may wreck the engine. A considerable amount or number of parts of the air fuel mixture may ignite at the same instant and result in a combustion knock or heavy thud rather than a pink. While it is end mixture which knocks in the spark ignition engine, it may be any fairly large part or parts of the mixture which knock in compression ignition engine. The following methods may be adopted to prevent Diesel Knock:

i. By reducing the delay period by doping e.g., adding 1% of ethyl nitrate or any nitrate so as to accelerate the combustion.

ii. By raising the compression ratio to produce a temperature much higher than that required for the spontaneous ignition of the fuel. It should be noted that the spark ignition engine would be more liable to detonate in case the compression ratio is raised as already mentioned.

iii. By increasing the turbulence of the compressed air injected, promotes homogeneous mixture by strapping the fuel from the spray.

iv. By arranging the fuel injector in such a way as to inject only a small quantity of fuel in the beginning.

v. By supercharging i.e., increasing the inlet pressure of air. An increase in the inlet pressure would increase the detonation tendency in a spark ignition engine.

vi. By raising the temperature of coolant, that of the intake air as well as cylinder head and combustion chamber. A spark ignition engine is more likely to detonate under this change.

vii. By increasing the injector pressure which encourages the atomization of fuel.

2.1.6 PRE-IGNITION:

Pre ignition is the ignition of the homogeneous mixture in the cylinder, before the timed ignition spark occurs, caused by the local overheating of the combustible mixture. For premature ignition of any local hot-spot to occur in advance of the times spark on the combustion stroke it must attain a minimum temperature of something like $700 - 800\,^{\circ}C$.

Pre ignition is initiated by some overheated projecting part such as sparking plug electrodes, exhaust valve head, metal corners in the combustion chamber, carbon deposits or protruding cylinder head gasket rim etc..,

However, Pre ignition is also caused by persistent detonating pressure shockwaves scoring away the stagnant gases which normally protect the combustion chamber walls. The resulting increased heat flow through the walls, raise the surface temperature of any protruding poorly cooled part of the chamber, and this therefore provides a focal point for Pre ignition.

The initiation of ignition and the propagation of the flame front from the heated hot spot in similar to that sparkplug is their respective instant of ignition. Thus the sparkplugs provides a times and controlled moment of ignition whereas the heated surface forming the hot

spot builds up to the ignition temperature during each compression stroke and therefore the actual instant of ignition is unpredictable.

The early ignition created by pre ignition extends the total time and the burnt gases remain in the cylinder and therefore increase the heat transfer on the chamber walls, as a result, the self-ignition temperature will occur earlier and earlier on each successive compression stroke. Consequently, the peak cylinder pressure (which normally occurs at its optimum position of 10°-15° after T.D.C) will progressively advance its position to T.D.C. where the cylinder pressure and temperature will be maximised.

2.1.7 FUEL ADDITIVES

Fuel additives are compounds formulated to enhance the quality and efficiency of the fuels used in motor vehicles. In some cases, the supplier incorporates the additive into the gasoline itself; at other times, the fuel additive is sold as a separate product that consumers may use to improve or maintain the performance of their engines. While some auto mechanics place a great deal of emphasis on using fuel additives or purchasing gasoline that is infused with the additional protectants or performance boosters, others question the effectiveness of additive products.

There are several benefits associated with the use of additives. One of the main advantages has to do with engine performance. With some fuel oil additives, the product is claimed to boost the octane level of the gasoline, providing the engine with more power from the same amount of gas.

Stages of combustion – S.I Engine
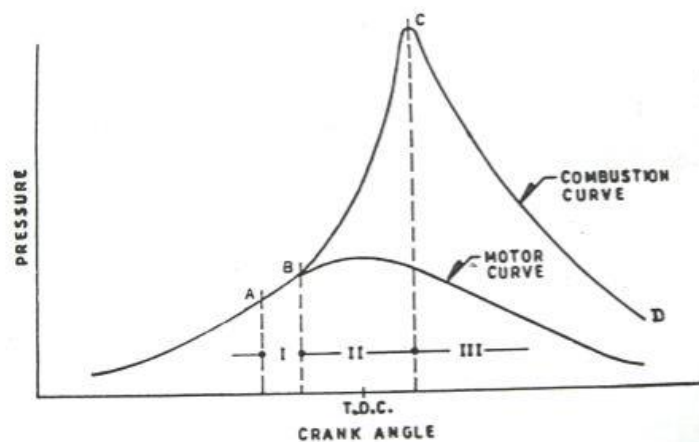
Fig 2.5 combustion phases in S.I engine

Combustion process in SI engines is divided mainly into three phases:

1. **Ignition lag or delay period-** This first phase is the duration between the occurrence of spark at spark plug and the deviation of combustion curve from motor curve (AB in figure). This period tends to be very nearly constant in time. Factors influencing this phase are:

   i. Fuel- Higher the self-ignition temperature of fuel, longer is the ignition lag.

   ii. Mixture ratio- Ignition lag increases as the relative fuel air ratio increases or decreases from 1.2

   iii. Initial temperature- Ignition lag increases rapidly with increase in temperature of the air-fuel mixture.

   iv. Flame temperature- As the temperature of the flame between the spark plug electrodes increases, the reaction time decreases and so the ignition lag decreases.

2. **Rapid combustion phase-** This second phase (BC) starts when combustion curve deviates from the motor curve and extends till the maximum pressure is reached. In this phase mean temperature of gases in the engine cylinder continues to increase beyond the maximum pressure point. The time required mainly depends on the intensity of turbulence or state of agitation of the air-fuel mixture. The duration of the phase is approximately constant in terms of the crank-angle movement.

3. **Afterburning-** This third and final phase (CD) occurs between points of maximum pressure and maximum temperature. After the flame front has reached the cylinder walls about 25% of the charge is still not completely burnt. But at this stage it is difficult for the remaining oxygen in the charge to react with the petrol vapours which slows down the rate of combustion. Simultaneously heat is liberated due to chemical interaction caused by reassociation of the combustion products throughout the expansion stroke.

However it has to be noted that no clear distinction can be made between these three phases as no abrupt change is noticed.

Stages of combustion – C.I Engine
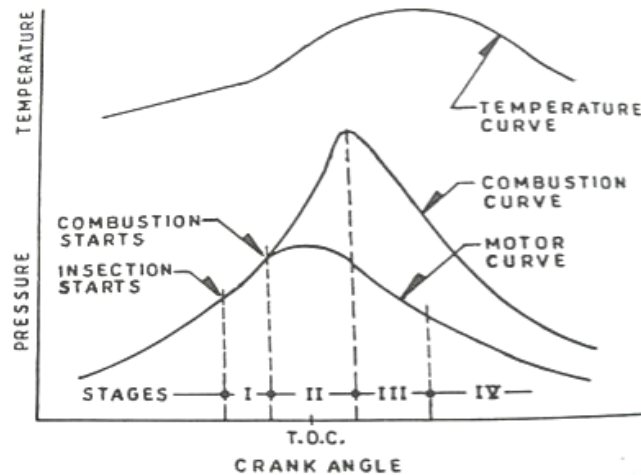
Fig 2.6

There are four stages of combustion in CI engine as follows:

1. **Ignition delay-** During this stage there is a physical delay period which is the time from beginning of injection to the attainment of chemical reaction conditions. The fuel is atomized and mixed with air and its temperature is raised. This period is followed by a chemical delay period in which pre flame reactions start and accelerate until local ignition takes place.

2. **Rapid or uncontrolled combustion-** This is second stage in which pressure rise is rapid since during delay period the fuel droplets have had time to spread themselves over a wide area and have fresh air around them. This phase extends from end of delay period to point of maximum pressure.

3. **Controlled combustion-** The very high temperature and pressure at end of second stage cause the fuel droplets injected during last stage to burn instantly and any further pressure rise can be controlled by purely mechanical means that is injection rate. This period ends at maximum cycle temperature. The heat evolved by end of this phase is 70 to 80 percent of total heat of fuel supplied.

4. **After burning-** This fourth stage may not be present in all cases but due to poor distribution of fuel particles combustion may continue in the expansion stroke. Its duration may be 70 to 80 degrees of crank travel from TDC

2.1.8 IGNITION DELAY OR IGNITION LAG

Delay period is the time between the start of injection and start of combustion. The delay period extends for about 13 degree movement of crank. This delay time decreases with increase in speed

if there is no delay, the fuel would burn at injector and there would be oxygen deficiency around the there would be oxygen deficiency around the injector, which results in incomplete combustion. If the delay period is too long, amount of fuel availability for simultaneous explosion, is too great, which results in rapid pressure rise.

The delay period should be as short as possible since long. The delay period should be as short as possible since long delay period gives more delay period gives more rapid rise in pressure and thus causes knocking. Rapid rise in pressure and thus causes knocking.

2.1.9 EFFECTS OF ENGINE VARIABLES ON IGNITION LAG:

Ignition lag (time lag between first igniting of fuel and the commencement of the main phase of combustion) is not a period of inactivity but is a chemical process. The ignition lag in terms of crank angles is 10° to 20° and in terms of time, 0.0015 sec or so.

The duration of ignition lag depends on the following factors:

1. **Fuel.** Ignition lag depends on chemical nature of fuel. The higher self-ignition temperature (S.I.T) of fuel, longer the ignition lag.
2. **Mixture ratio.** Ignition lag is the smallest for the mixture ratio which gives the maximum temperature. This mixture ratio is somewhat richer than stoichiometric ratio.
3. **Initial temperature & Pressure.** Ignition lag is reduced if the initial temperature and pressure are increased (and these can be increased by increasing the compression ratio).
4. **Turbulence.** Ignition lag is not much affected by the turbulence.

2.1.10 METHODS OF GENERATING SWIRL IN A C.I. ENGINE COMBUSTION CHAMBER

There are the basic methods of generating swirl in a C.I. engine combustion chamber, which are mentioned below:

1. By directing the flow of air during its entry to the cylinder, known as **Induction Swirl.** This method is uses in open combustion chamber.

2. By forcing the air through a tangential air passage into a separate swirl chamber during compression stroke, known as **compression swirl.** This method is used in swirl chambers.

3. By use of initial pressure rise due to partial combustion to create swirl turbulence, known as **Combustion induced swirl.** This method is used in pre combustion chambers and air cell chambers.
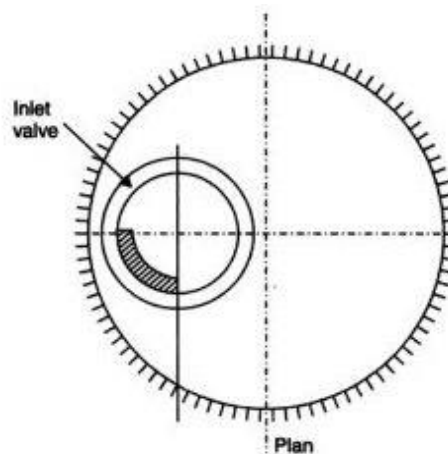
INDUCTION SWIRL

Fig 2.7 Induction swirl by masking the inlet valve

➢ In a four stroke engine induction swirl can be obtained either by careful formation of air intake passage or masking or shrouding a portion of circumference of inlet valve. The angle of mask is from 90° to 140° of the circumference.

➢ In two stroke engine, induction swirl is created by suitable inlet port forms.

➢ The induction swirl generated by air intake passage is very weak, if a masked inlet valve is used, it provides an obstruction in a passage which reduces volumetric efficiency. Therefore swirl generated is weak even with this method. With a weak swirl, a single orifice injection cannot provide a desired air fuel mixing. Therefore with induction swirl we have to use a multiple orifice injector.

COMPRESSION SWRIL

➢ The second method of generating swirl is by compression swirl in what is known as swirl chamber. A swirl chamber is a divided chamber. A divided combustion chamber is defined as one in which combustion space is divided into two or more distinct compartments, between which there are restrictions or throats small enough so that considerable pressure differences occur between them during combustion process.
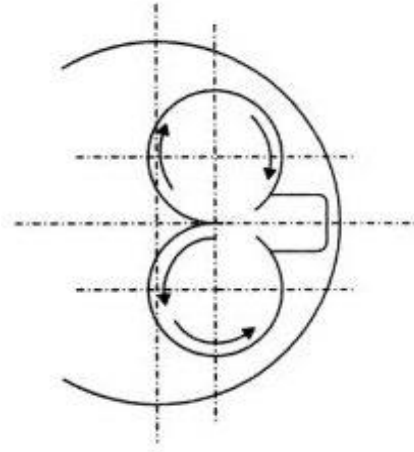
Fig 2.8

> The swirl is maximum at about 15o before T.D.C. i.e., close to the time of injection. The fuel is injected into the swirl chamber and ignition and bulk of combustion takes place therein. A considerable amount of heat is lost when products of combustion pass back through the same throat and this loss of heat is reduced by employing a heat insulated chamber. Thus, serves as a thermal regenerated receiving heat during combustion and expansion and returning the heat to air during compression stroke. However the loss of heat to surface of combustion chamber is greater than induction swirl.

> In combustion swirl, a very strong swirl which increases with speed is generated.

2.1.11 DIFFERENT TYPES OF COMBUSTION CHAMBERS

Combustion chamber is the space enclosed between the piston head and cylinder head when the piston is at top dead center position.

Depending upon the location of the spark plug, valves and type of cylinder head , the combustion chambers in SI engines are of the following shapes:

1. Spherical shape
2. I-shape
3. L-shape
4. T-shape
5. F-shape
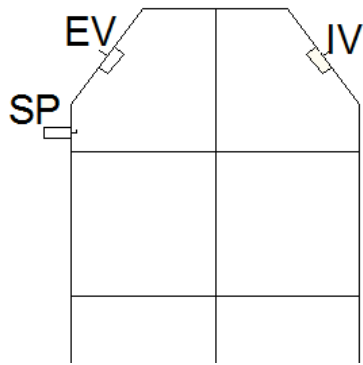
Fig 2.9 **Spherical Shape**
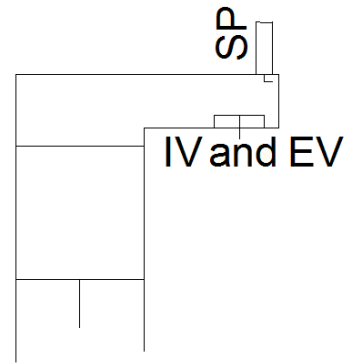
Fig 2.10 **L- Head**
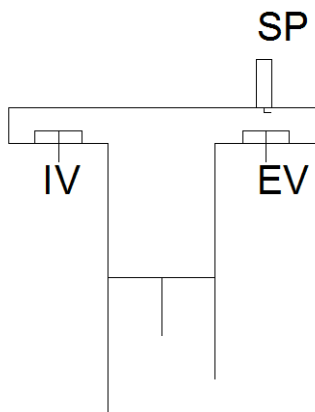
Fig 2.11**T-head**
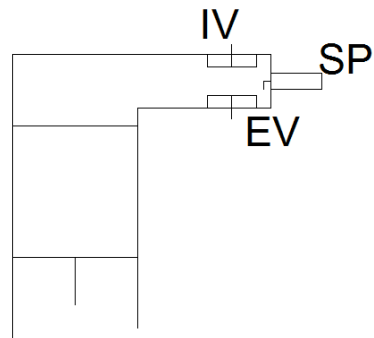
Fig 2.12 **F-head**
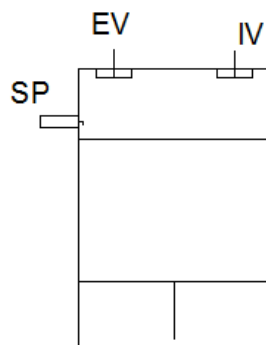
Fig 2.13 **I-head**

> In Spherical shape combustion chambers, Inlet and Exhaust valves are fitted in the cylinder head. Spark plug may be at the center or at side of the head.

> I-head combustion chambers have overhead valves. The engines with this type of combustion chamber are used in high speed vehicles and racing cars.

> T-shape combustion chamber projects around the cylinder head. The spark is fitted at the top and valves in the sides. This type of engines have good efficiency.

> F-head combustion chamber projects at one side of the cylinder head. The location of the sp plug and valves are as shown

> L-head combustion chambers use side valves.

Classification of Diesel Engine combustion Chambers are as follows,

Open combustion chamber
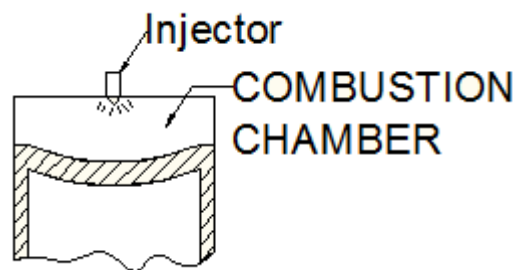
Fig 2.14

- Open combustion chamber is made like groove inside the top of the piston.

- The injector fitted at the center of the cylinder head so that it injects the fuel in the combustion chamber.

Pre-combustion chambers
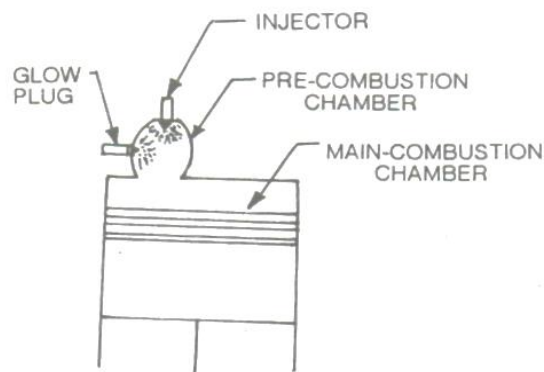
Fig 2.15

- In <u>pre-combustion chamber</u> there are two combustion chambers – one is auxiliary combustion chamber and the other is main combustion chamber.

- The auxiliary combustion chamber is smaller in size than the main combustion chamber and is called pre-combustion chamber.

- Fuel is injected into the pre-combustion chamber where it partly burns. This partly burnt fuel goes through a small orifice in the main combustion chamber, where the complete combustion takes place.

The turbulence created in the combustion chamber helps to ignite the fuel completely

Swirl combustion chambers
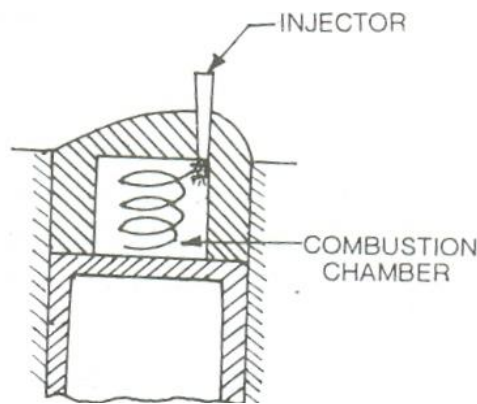
Fig 2.16

- Swirl is a circular motion which is given to the incoming air during the suction stroke. In the swirl combustion chamber (also known as turbulence combustion chamber) the air is given a swirl while coming in the cylinder.

- The fuel is injected in the swirled air so that mixing and burning of the fuel takes place completely.

Squish combustion chamber

- Squish is a flow of air which goes from the periphery to the center of the cylinder. To get a squish in the combustion chamber, a groove is made inside the piston head.
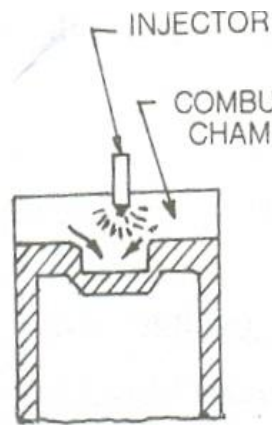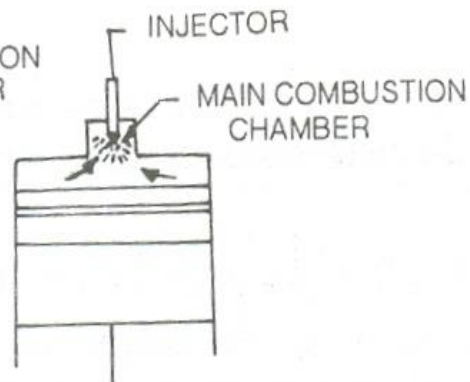
Fig 2.17 (a)   Fig 2.17 (b)

➢ During the compression stroke when the piston moves from BDC to TDC, it squeezes the air from the periphery towards the center. This action gives turbulence to the air. The fuel is injected at the center of the combustion chamber. Some times the squish combustion chamber is made inside the cylinder head.
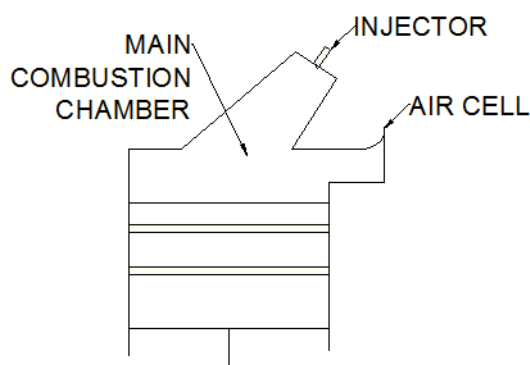
Air cell & energy cell

Fig 2.18

➢ Air cell is a small combustion chamber connected through air in a narrow passage with the main combustion chamber as shown in the sketch.

➢ During compression stroke air is compressed in the air cell also. When the fuel is injected in the main combustion chamber, it burns and the piston moves down.

➢ At this time the compressed hot air of the air cell comes in the main chamber creating a turbulence a making complete combustion. There is more heat loss due to increased surface of the cylinder.

2.1.12 COMBUSTION EQUATION OF SOLID FUELS

The chemical reaction by which elements of fuel like carbon, hydrogen, etc combine with oxygen in air at ignition temperature and release heat is called combustion.

1.   When carbon burns with sufficient quantity of oxygen, carbon dioxide is produced along with a release of Large amount of heat. This is represented by the following chemical equation:

$$C+O_2=CO_2$$
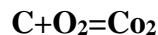
1mol+1mol=1mol   ………………… (By volume)

12kg+32kg=44kg   ………………… (By mass)

1kg+8/3 kg=11/3kg

It means that 1kg of carbon requires 8/3 kg of oxygen for its complete combustion, and produces 11/3kg of carbon dioxide gas.

2.   If sufficient oxygen is not available, then combustion of carbon is incomplete. it then produces carbon monoxide
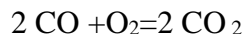
This is represented by the following chemical equation:

$$2C+O_2=2CO$$
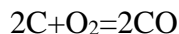
2mol+1mol=2mol   ………………… (By volume)

2X12kg+2X16kg=2X28kg   ………………… (By mass)

1kg+4/3 kg=7/3kg.

It means that 1kg of carbon requires 4/3 kg of oxygen for its complete combustion, and produces 7/3kg of carbon monoxide gas.

3.   If carbon monoxide is burnt further, it is converted into carbon dioxide. Thus,

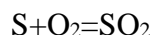$$2 CO +O_2=2 CO_2$$

2mol+1mol=2mol.   ………………… (By volume)

2X28kg+2X16kg=2X44kg   ………………… (By mass)   **1kg+4/7 kg=11/7kg.**

It means that 1kg of carbon monoxide requires 4/7kg of oxygen for its complete combustion, and produces 11/7kg of carbon dioxide gas.

4. When sulphur burns with oxygen, it produces sulphur dioxide. This is represented by the following chemical equation:
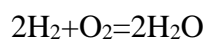
$$S+O_2=SO_2$$

1mol+1mol=1mol   ………………… (By volume)

32kg+2x16kg=64kg   ………………… (By mass)

1kg+1kg=2kg.

It means that 1kg of sulphur requires 1kg of oxygen for its complete combustion, and produces 2kg sulphur dioxide.

5. The hydrogen burns with oxygen to produce water vapor or steam.

$$2H_2+O_2=2H_2O$$

(2x2=4)+32=36 (By mass).

1kg+8kg=9kg

It means that 1kg of hydrogen requires 8kg of oxygen for its complete combustion, and produces 9kg of water vapour.

2.1.13 COMBUSTION EQUATIONS OF GASEOUS FUEL.

The gaseous fuels are generally measured by volume in m^3 and the combustion equations are generally used quantitatively in volume form.

2.1.14 COMBUSTION OF CARBON MONOXIDE:

The carbon monoxide burns with oxygen to produce carbon dioxide.

$$2\,CO+O_2=2\,CO_2$$
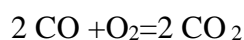
2vol+1vol=2vol

$$1m^3+0.5m^3=1m^3$$

It means that 1m of carbon monoxide requires 0.5m of oxygen to produce $1m^3$ of carbon dioxide.

2.1.15 COMBUSTION OF HYDROGEN

The hydrogen burns with oxygen to produce water vapour.

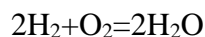$$2H_2+O_2=2H_2O$$

$$2vol+1vol=2vol$$

$$\mathbf{1m^3+0.5m^3=1m^3}$$

It means that $1m^3$ of hydrogen requires 0.5m3 oxygen for its complete combustion, and produces $1m^3$ of water vapour.

2.1.16 COMBUSTION OF METHANE

Methane burns with oxygen to produce carbon dioxide and water vapour.

$$CH_4+2O_2=CO_2+2H_2O$$
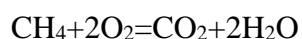
$$1vol+2vol=1vol+2vol$$

$$1m^3+2m^3=1m^3+2m^3$$

It means that $1m^3$ of methane requires $2m^3$ of oxygen to produce $1m^3$ of carbon dioxide and $2m^3$ of water vapour.

2.1.17 COMBUSTION OF ETHYLENE

Ethylene burns with oxygen to produce carbon dioxide and water vapour.

$$C_2H_4+3O_2=2O_2+2H_2O$$
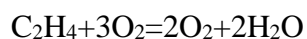
$$1vol+3vol=2vol+2vol$$

$$1m^3+3m^3=2m^3+2m^3$$

It means that $1m^3$ of ethylene requires $3m^3$ of oxygen to produce $2m^3$ of carbon dioxide and $2m^3$ of water vapour.

2.1.18 Determination of stoichiometric (or) chemically correct (or) minimum air required for the complete combustion of 1kg of solid (or) liquid fuel. The adequate supply of oxygen is very essential for the complete combustion of a fuel, and hence for obtaining maximum amount of heat from a fuel.

The theoretical or minimum mass (or volume) of oxygen required for complete combustion of 1kg of fuel may be calculated from the chemical analysis of the fuel. The mass of oxygen, required by each of the constituents of the fuel, may be calculated from the chemical equations.

Determination of stoichiometric (or) chemically correct (or)minimum air required for the complete combustion of 1kg of solid fuel

Consider 1kg of fuel,

Mass of carbon=C kg

Mass of hydrogen=H_2 kg.

Mass of sulphur=S kg

We know that 1kg of carbon requires 8/3kg of oxygen for its complete combustion.similarly,1kg of hydrogen requires 8kg of oxygen and 1kg of sulphur requires 1kg of oxygen for its complete combustion.

∴*Total oxygen required complete combustion of 1kg of fuel*

=8/3 C+8 H_2+S kg .... (i)

If some oxygen (say o_2 kg) is already present in the fuel,then total oxygen for the complete combustion of 1kg of fuel

= (8/3 C+8H_2+S) - O_2 Kg ....(ii)

It may be noted that the oxygen has to obtained from atmospheric air, which mainly consists of nitrogen and oxygen along with a small amount of carbon dioxide and negligible amounts of rare gases like argon, neon, and krypton etc. But for all calculations, the combustion of air is taken as;

Nitrogen (N_2) =77%: oxygen (O_2)=23% ....(by mass)

Nitrogen (N_2) =79%: oxygen (O_2) =21% ....(by volume)

It is thus obvious, that for obtaining 1kg of oxygen, amount of air required

=100/23=4.35Kg ....(By mass)

∴ **Theoretical or minimum air required for complete combustion of 1kg**

of fuel = 100/23 [(8/3 C+8H_2+S) -O_2] Kg

Determination of stoichiometric (or) chemically correct (or) minimum air required for the complete combustion of 1kg of gaseous fuel.

Consider $1m^3$ of a gaseous fuel,

Let   volume of cabon dioxide $=CO\ m^3$

Volume of hydrogen $=H_2\ m^3$

Volume of methane $=CH_4 m^3$

Volume of ethylene $=C_2H_4\ m^3$

We know that,

$1m^3$ of CO requires $o.5m^3$ of O_2

$1m^3$ of H_2 requires $o.5\ m^3$ of O_2

$1m^3$ of CH_4 requires $2m^3$ of O_2

$1m^3$ of C_2H_4 requires $3m^3$ of O_2

∴ Total oxygen required for complete combustion of $1m^3$ of fuel

$$=0.5CO+0.5H_2+2CH_4+3C_2H_4\ m3$$

If some oxygen (say $o_2\ m^3$) is already present in the fuel, then total oxygen required for complete combustion of $1m^3$ of fuel

$$= [0.5CO+0.5\ H_2+2CH_4+3C_2H_2]-O_2\ m^3$$

Since the oxygen present in the air is 21%by volume, therefore **theoretical or minimum volume of air required for complete combustion of $1m^3$ of fuel.**

$$=100/21[(0.5CO+0.5H_2+2CH_4+3C_2H_4)-O_2]\ m^3$$

2.1.19 EXCESS AIR

➢ Based on combustion equations, the chemically correct or stoichiometric air required for the complete combustion of 1kg of fuel is determined. But in actual practice, some air does not come in contact with the fuel supply. So a portion of fuel may be unburnt. To avoid this, excess air is supplied. This ensures complete combustion.

➢ The amount of excess air is supplied depends on the type of fuel and furnace firing conditions.

➢ Generally, 25% to 50% excess air is used.

Mass of excess air supplied = mass of minimum air *X* percentage of excess air.

∴. **Total mass of air supplied = mass of minimum air + mass of excess air**

2.1.20 PRODUCTS OF COMBUSTION

Products of combustion	Amount of products of combustion			Percentage by mass $(m/\sum m) \times 100$
	Formula	Calculation	Mass in (kg)	
CO_2	$(11/3)xC$	$(11/3)x1$	m_1	$(m_1/\sum m) \times 100$
H_2O	$9H_2$	$9x1$	m_2	$(m_2/\sum m) \times 100$
SO_2	$2xS$	$2x1$	m_3	$(m_3/\sum m) \times 100$
O_2 Excess air	0.23x(mass of excess air)	O.23x(mass of minimum air)x (% of excess air)	m_4	$(m_4/\sum m) \times 100$
N_2 Total air	0.77x(Total mass of air)	0.77x(mass of minimum air+ mass of excess air)	m_5	$(m_5/\sum m) \times 100$
			$\sum m$	100%

During combustion, the products of combustion of a fuel can be determined if the percentage of constituents of the fuel are known.

We know that,

Consider 1kg of fuel contains;   Carbon=c kg, Hydrogen=H_2 kg, Sulphur=s kg & Oxygen=O_2 kg

Consider the amount of excess air is supplied, products of combustions are calculated.

2.1.21 ANALYSIS OF EXHAUST GAS

➢ Flue gas analysis by Orsat apparatus :

➢ The flue or exhaust gases from the boiler may be analysed by means of Orsat apparatus and it is shown in figure

➢ Flask F_1 contains a solution of caustic soda (NaOH) which will absorb carbon dioxide from the sample.

➢ Flask F_2 contains pyrogallic acid and caustic soda which will absorb the oxygen from the sample .

Fig. 10.1. Orsat's apparatus.
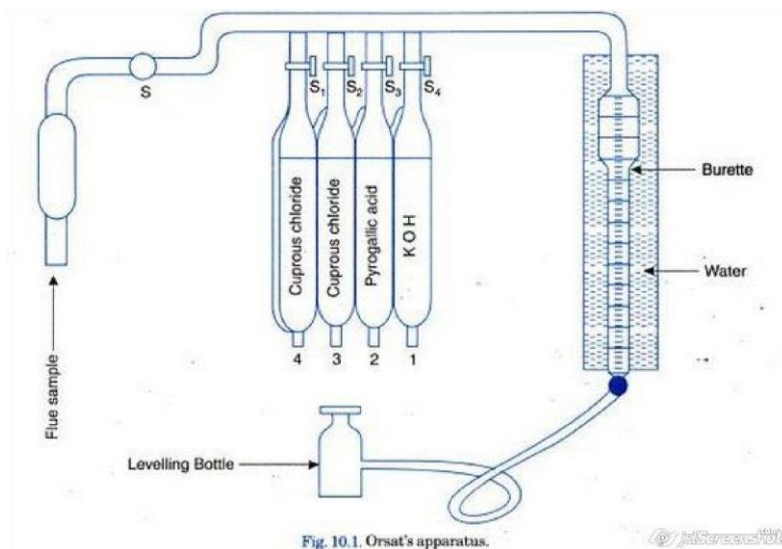
Fig 2.19

➢ Flask F_3 contains solution of cuprous chloride in hydrochloric acid which will absorb the carbon monoxide from the sample

➢ To analysis of a given sample of flue gas,

➢ 100cc of the flue gas is sucked into the measuring tube M by lowering the movable flask F_4. By adjusting the level of flask F_4, the flue gases can, inturn be forced into either of the flask F_1 or F_2 or F_3 by opening the respective cocks C_1 or C_2 or C_3.

➢ The flue gas is allowed to stand in each for about 30 secs and then sucked back into the measuring tube M. The chemicals in the three flasks absorb carbon

dioxide, oxygen and carbon monoxide and the resulting contraction in volume enable the percentage of each gas present in the sample to be read on the measuring tube M.

➢ When the percentage of carbon dioxide, oxygen and carbon monoxide are obtained the remainder of the gas is assumed to be nitrogen.

➢ Since the gas is collected over water in the measuring tube M, any steam present will be condensed. Also sulphur dioxide, if any will be absorbed.

➢ Orsat apparatus give the percentage of dry flue gas only

2.1.22 CALORIFIC VALUES OF FUELS (OR) HEATING VALUE OF FUELS

The calorific value of the solid or liquid fuel may be defined as the amount of heat given out by the complete combustion of 1kg or $1m^3$ of fuel.

It is expressed in terms of KJ/Kg for solid fuel and KJ/m^3 for liquid fuel.

Following two types of calorific values are,

1. Higher or gross calorific value [H.C.V]

2.Lower or Net calorific value [L.C.V]

UNIT – III

AIR COMPRESSORS AND GAS TURBINES

3.1 Introduction:

An air compressor, as the name indicates, is a machine to compress the air and to raise its pressure. The air compressor sucks air from the atmosphere, compresses it and then delivers the same under a high pressure to a storage vessel. From the storage vessel, it may be conveyed by the pipeline to a place where the supply of compressed air is required. Since the compression of air requires some work to be done on it, therefore a compressor must be driven by
Some prime mover.

3.1.1 Uses of compressed air

1. The compressed air is used for many purposes such as for operating pneumatic drills, riveters, road drills, paint spraying,
2. In starting and supercharging of internal combustion engines.
3. In gas turbine plants, jet engines and air motors.
4. It is also utilized in the operation of lifts, ramps pumps and a variety of other devices.
5. In industry, compressed air is used for producing blast of air in blast furnaces.

3.1.2 Classification of air compressors

The air compressor may be classified in many ways, bu the following are important from the subject point of view;

1. According to working
   a. Reciprocating compressor
   b. Rotary compressor
2. According to action
   a. Single acting
   b. Double acting

3. According to number of stages
   a. Single stage compressor
   b. Multi stage compressor

4. According to pressure limit
    a. Low pressure compressor – delivery pressure up to 1 bar
    b. Medium pressure compressor – delivery pressure is in between 1 bar and 8 bar
    c. High pressure compressor - delivery pressure is more   than 10 bar
5. According to capacity
    a. Low capacity compressors – free air delivered is less than 0.15 m³/sec
    b. High pressure compressors - free air delivered is more than 5m³/sec

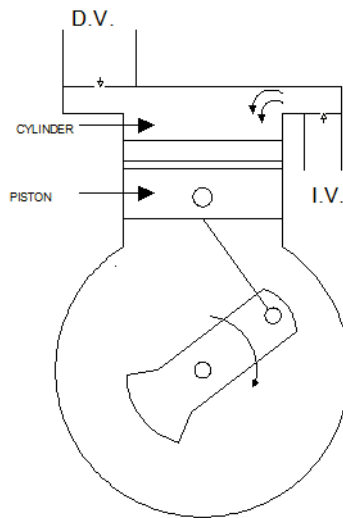### 3.1.3 WORKING OF SINGLE STAGE RECIPROCATING AIR COMPRESSOR

Fig 3.1

A single stage reciprocating air compressor, in its simplest form, consists of a cylinder, piston, inlet and discharge valve, as shown in Fig. From the geometry of the compressor, we find that when the piston moves downwards (or in other words, during outward or suction stroke), the pressure inside the cylinder falls below the atmospheric pressure. Due to this pressure difference, the inlet valve (I.V) gets opened and air is sucked into the cylinder, at inlet pressure until the piston completes the outward stroke. Now when the piston moves upwards (or in other words, during inward or delivery stroke), the pressure inside the cylinder goes on increasing till it reaches the discharge pressure. At this stage, the discharge valve (D.V) gets opened and air is delivered to the container. At the end of delivery stroke, a small quantity of air, at high pressure, is left in the clearance space. As the piston starts its suction stroke, the air contained in the clearance space

expands till its pressure falls below the atmospheric pressure. At this stage, the inlet valve gets opened as a result of which fresh air is sucked into the cylinder, and the cycle is repeated.

It may be noted that in a single acting reciprocating air compressor, the suction, compression and delivered of air takes place in two strokes of the piston or one revolution of the crankshaft.

3.1.4 Compression Processes

Fig 3.2

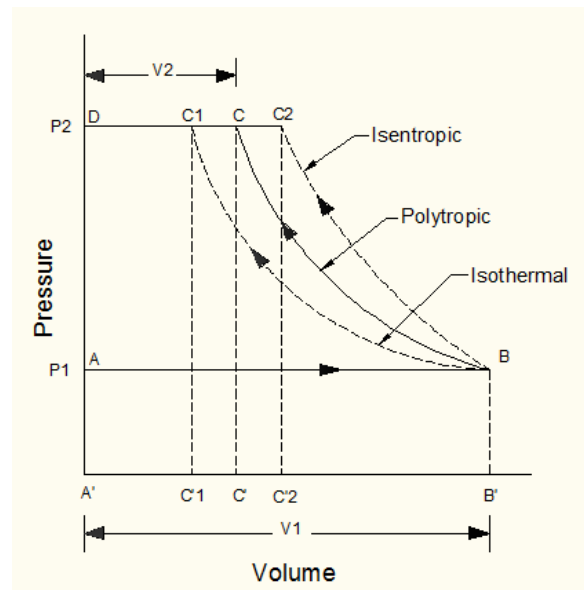The air may be compressed by the following processes.
(1) Isothermal compression,
(2) Poly tropic compression and
(3) Adiabatic compression.

Isothermal compression

During the processes work input to drive the compressor is less, because the area under the $P - V$ diagram is less when compared to other two processes. More over this isothermal process is suitable if the compressed air is to be stored.

Work done during isothermal compression

The isothermal compression and delivery of air is shown by the graphs. Now C1D represents the volume of air delivered. We know that work done by the compressor per cycle,

$$W = \text{Area A B C1 D}$$

$$= \text{Area A`DC}_1\text{C}_1\text{` + Area C}_1\text{BB`C}_1\text{`- Area A`ABB`}$$

$$= \text{work done during compression + Work done during delivery}$$

$$- \text{Work done during suction}$$

$$= p_1 V_1 \log_e\left(\frac{V1}{V2}\right) + p_2 V_2 - p_1 V_1$$

$$= p_1 V_1 \log_e\left(\frac{V1}{V2}\right) \quad \text{But } p_2 V_2 = p_1 V_1$$

Therefore $\frac{V1}{V2} = \frac{p2}{p1}$

$$= p_1 V_1 \log_e\left(\frac{p2}{p1}\right) = p_1 V_1 \log_e\left(\frac{V1}{V2}\right)$$

$$= m\ R\ T1 \log_e\left(\frac{p2}{p1}\right) = m\ R\ T1 \log_e\left(\frac{V1}{V2}\right)$$

Poly tropic process

The work input required to drive compressor during this process is more than isothermal and less than adiabatic process. This type of compression is used in Bell Coleman cycle.

Work done during polytropic compression

The polytropic compression is shown by the line BC in fig., Now CD represents the volume of air delivered i.e., V_2. We know that work done on air per cycle,

W   = Area A-B-C-D

= Area A`DCC` + Area CBB`C` - Area A`ABB`

= work done during compression + Work done during delivery

– worked done during suction

$$W = \frac{p2V2 - p1V1}{n - 1} + p2V2 - p1V1$$

$$W = \frac{p2V2 - p1V1 + (n - 1)p2V2 - (n - 1)p1V1}{n - 1}$$

$$W = \frac{p2V2 - p1V1 + np2V2 - p2V2 - np1V1 + p1V1}{n-1}$$

$$W = \frac{np2V2 - np1V1}{n-1}$$

$$W = \frac{n(p2V2 - p1V1)}{n-1}$$

$$W = \frac{n}{n-1}(p2V2 - p1V1)$$

But $p_2 V_2 = mRT_2$ and $p_1V_1 = mRT_1$

$$W = \frac{n}{n-1}(mRT2 - mRT1)$$

$$W = \frac{n}{n-1} \, mRT1 \left(\frac{mRT2}{mRT1} - 1\right)$$

$$W = \frac{n}{n-1} \, mRT1 \left(\frac{T2}{T1} - 1\right)$$

$$\frac{n-1}{n}$$

For polytropic process $\qquad \left(\frac{T2}{T1}\right) = \left(\frac{p2}{p1}\right)$

For polytropic work done on the air per cycle

$$W = \frac{n}{n-1} \, mRT1 \left(\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right)$$

Or

$$W = \frac{n}{n-1} \, p1V1 \left(\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right)$$

Isentropic or Adiabatic process

The work input required to the drive the compressor is more than the isothermal and polytropic process. This type of compression is used in I.C. engines.

Work done during isentropic compression

The isentropic compression is shown by the curve BC2 in Fig 3.2 In this case, the volume of air delivered v2 is represented by the line C2D.

The work done on the air per cycle during isentropic compression may be worked out in the similar way as polytropic compression. The polytropic index n is changed to isentropic index γ in the previous results.

$\therefore$ Work done on the air per cycle,

$\qquad$ W = $\quad$ Area 1-2-3-4-1

$\qquad$ = $\quad$ work done during compression + Workdone during delivery

$\qquad\qquad\qquad\qquad\qquad\qquad\qquad$ $-$ workedone during suction

$$W = \frac{p2V2 - p1V1}{\gamma - 1} + p2V2 - p1V1$$

$$W = \frac{p2V2 - p1V1 + (\gamma - 1)p2V2 - (\gamma - 1)p1V1}{\gamma - 1}$$

$$W = \frac{p2V2 - p1V1 + \gamma p2V2 - p2V2 - \gamma p1V1 + p1V1}{\gamma - 1}$$

$$W = \frac{\gamma p2V2 - \gamma p1V1}{\gamma - 1}$$

$$W = \frac{\gamma(p2V2 - p1V1)}{\gamma - 1}$$

$$W = \frac{\gamma}{\gamma - 1}(p2V2 - p1V1)$$

But $p_2 V_2 = mRT_2$ and $p_1V_1 = mRT_1$

$$W = \frac{\gamma}{\gamma - 1}(mRT2 - mRT1)$$

$$W = \frac{\gamma}{\gamma - 1} mRT1 \left(\frac{mRT2}{mRT1} - 1\right)$$

$$W = \frac{\gamma}{\gamma - 1} \, mRT1 \left(\frac{T2}{T1} - 1 \right)$$

For **adiabatic** process $$\left(\frac{T2}{T1} \right) = \left(\frac{p2}{p1} \right)^{\frac{\gamma - 1}{\gamma}}$$

For **adiabatic** work done on the air per cycle

$$W = \frac{\gamma}{\gamma - 1} \, mRT1 \left(\left(\frac{p2}{p1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right)$$

Or

$$W = \frac{\gamma}{\gamma - 1} \, p1V1 \left(\left(\frac{p2}{p1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right)$$

Problems

1. A single stage reciprocating air compressor is required to compress 1 kg of the air from 1 bar to 4 bar.

The initial temperature of is 27 °C. Compare the work requirement in the following cases:

1. Isothermal compression ;
2. Compression with p V $^{1.2}$ = constant ;
3. Isometric compression.

Given: m=1kg; p_1=1bar; p_2=4bar; T_1=27°C=27+273=300K; n=1.2

Solution:

1. Work required for the isothermal compression

We know that work required by the compressor,

$W = p_1 v_1 \log\left(\frac{p_2}{p_1}\right) = m R T_1 \log\left(\frac{p_2}{p_1}\right)$

=1 x 287 x 300 In (4/1) = 119230 J = 119.23 KJ Ans.

2. Work required for the polytrophic compression (i.e. PV$^{1.2}$ = constant)

We know that work required by the compressor,

$W = \frac{n}{n-1} \times m R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$

$\frac{1.2}{1.2-1} \times 1 \times 287 \times 300 \times \left[\left(\frac{4}{1}\right)^{\left(\frac{1.2-1}{1.2}\right)} - 1\right]$

=134320 J =134.32 KJ Ans.

3. Work required for the isentropic compression

We know that the work required by the compressor,

$W = \frac{\gamma}{\gamma-1} \times m R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$

$= \frac{1.4}{1.4-1} \times 1 \times 287 \times 300 \times \left[\left(\frac{4}{1}\right)^{\left(\frac{1.4-1}{1.4}\right)} - 1\right]$

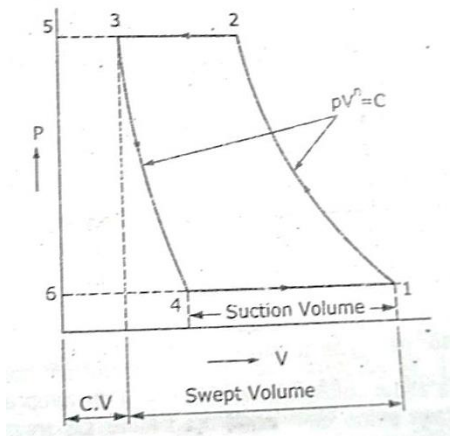= 146630 J = 146.63 KJ Ans.

3.1.5 Clearance volume and its effects

When the piston reaches the Top dead centre (TDC) in the cylinder, there is a dead space between cylinder head and the piston top. This space is known as clearance space and the volume occupied by this known as clearance volume. It's value ranges from 5% to 10% of swept volume (stroke volume).

$$\text{clearance ratio } = \frac{Clearance\ volume}{Stroke\ or\ Swept\ volume}$$

$$\text{clearance ratio } = \frac{V3}{V1 - V3}$$

Fig 3.3

3.1.6 Effects of clearance volume

1. It reduces the effective stroke volume
2. Mass of air reduced
3. Reduction in volumetric efficiency

3.1.7 Volumetric efficiency

The volumetric efficiency of a single stage reciprocating compressor is defined as the volume of gas entering the compressor per minute divided by the piston displacement of the compressor per minute.

Therefore, for a single stage compressor.

$$\text{Volumeric efficiency} = \frac{\text{Volume of gas entering compressor per minute}}{\text{piston displacement per minute}}$$

The volumetric efficiency of a multistage compressor is defined in the same way except that piston displacement is taken as that of the low pressure cylinder only. Thus for multistage compressor.

$$\text{Volumeric efficiency} = \frac{\text{volume of gas entering compressor per minute}}{\text{Piston displacement of the low pressure cylinder per minute}}$$
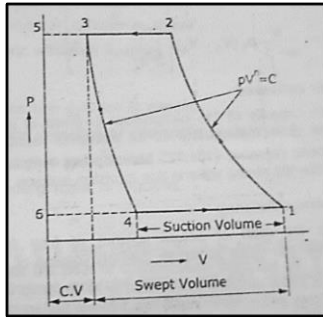
Fig 3.4

A further expression for the volumetric efficiency may derived from a consideration of the theoretical indicator diagram of fig.

Now referring fig we have

$$\text{Volumeric efficiency} = \frac{V1 - V4}{V1 - V3}$$

From the process 3-4,

$$\left(\frac{P3}{P4}\right)^{\frac{1}{n}} = \frac{V4}{v3} = \left(\frac{P2}{P1}\right)^{\frac{1}{n}} \quad \text{or} \quad v4 = v3 \times \left(\frac{P2}{P1}\right)^{\frac{1}{n}}$$

Since $p3 = p2 \ \& \ p4 = p1$

$$\text{clearance ratio} = \frac{Clearance\ volume}{Stroke\ or\ Swept\ volume} = \frac{V3}{V1-V3} = C$$

Then, volumetric efficiency $= \frac{V1-V4}{V1-V3} = \dfrac{V_1\text{-}V_3 \left(\frac{P2}{P1}\right)^{\frac{1}{n}}}{V_1\text{-}V_3} = \dfrac{V_1\text{-}V_3\left(\frac{P2}{P1}\right)^{\frac{1}{n}}}{V_3\ /\ C} = \frac{C\ V1}{V3} - C\left(\frac{p2}{p1}\right)^{1/n}$

Since $C = \frac{V3}{V1-V3}$. Therefore, $V_1 = \frac{V3}{C} + V_3 = V_3\left(\frac{1}{k} + 1\right)$

$$C = 1 + k - k(p2/P1)^{1/n}$$

$$\text{Volumetric efficiency} = \dfrac{V3\left(\frac{1}{C}+1\right)}{V_3} - C\left(\frac{p2}{p1}\right)^{1/n}$$

$$C = 1 + C - C\ (V1/V2)$$

$$C = 1 + C - C\ (p2/P1)^{1/n}$$

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3.1.8 Work done by Reciprocating Air compressor with Clearance volume:

The p-v diagram of a single stage single acting reciprocating air compressor with clearance volume (Vc). We know that during return stroke, the air is compressed by its major part i.e. compression stroke 1-2. This compression continues, till the pressure P_2 in the cylinder is sufficient to force open the delivery valve at 2. After that, no more compression takes place with the inward movement of the piston. Now during the remaining part of the compression stroke, compressed air is delivered till the piston is reaches at 3. At this stage, there will be some air (equal to clearance volume) left in the clearance space of the cylinder at pressure P_2. After the air in the clearance space will expand during some part of outward stroke of the piston i.e. expansion stroke 3-4. This expansion continues till the pressure p_1 in the cylinder is sufficient to force open the inlet valve at 4. After that the air is sucked from atmosphere during the suction stroke 4-1 at pressure P_1.

Though the compression and expansion of air may be isothermal, isentropic or polytropic, yet for all calculation purposes, it is assumed to be polytropic.

$$W = \frac{n}{n-1}\, p1V1\left(\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right) \ - \ \frac{n}{n-1}\, p1V4\left(\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right)$$

$$= \frac{n}{n-1}\, p1(V1 - V4)\left(\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right)$$

Problems

1. A single stage single acting reciprocating air compressor has the bore of 200 mm and stoke of 300 mm .It receives air at 1bar and 20°C and delivers it at 5.5bar.if the compression follows the law $pV^{1.3}=c$ and clearance volume is percent of the stoke volume, determine the power required to drive the compressor, if it runs at the 500 rpm.

Given:

D=200mm=0.2m;

L=300mm=0.3m;

p_1=1x10^{5}N/m²;

T_1=20°C=20+273=293K;

p_2=5.5 x10⁵N/m²;

n=1.3;

V_C=5%;

 N =500rpm.

To Find:

1) power required to drive the compressor runs at 500 rpm

Solution:

We know that stoke volume,

$$V_S= \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4} \times (0.2)^2 \times 0.3 = 0.00942 \; m^3.$$

Clearance volume,

$$V_C=5\% V_S =0.05 \times 0.00942=0.000471 \; m^3$$

And initial volume of air, $V_1 =V_C +V_s. = 0.000471+0.00942= 0.00989 \; m^3.$

We know that expanded clearance volume,

$$V_4 = Vc\left(\frac{p2}{p1}\right)^{\left(\frac{1}{n}\right)} = 0.00047\left(\frac{5.5}{1}\right)^{\frac{1}{1.3}} = 0.00174 \; m^3$$

And effective swept volume,

$$V_1 - V_4 = 0.00989 - 0.00174 = 0.00815 \; m^3$$

We know that work done by the compressor per cycle,

$$W = \frac{n}{n-1} \times P1\,(V1 - V4)\left[\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times 1 \times 10^5 \times 0.00815\left[\left(\frac{5.5}{1}\right)^{\frac{(1.3-1)}{1.3}} - 1\right] = 1702.32 \; N-m$$

2. Power required to drive the compressor

Since the compressor is single acting, therefore number of working strokes per minute,

$$N_W = N = 500$$

Power required to drive the compressor,

$$P = \frac{W \times Nw}{60} = \frac{1702.32 \times 500}{60} = 14186W = 14.186 \; KW \; Ans.$$

3.1.9 Multi stage compression:

In the previous articles, we have been taking into consideration the compression of air in single stage. In other words, air is sucked, compressed in the cylinder and then delivered at a higher pressure. But sometimes, the air is required at a high pressure. In such cases, either we employ a large pressure ratio (in single cylinder) or compress the air in two or more cylinders in series. It has been experienced that if we employ single stage compression for producing high pressure air (say 8 to 10 bar), it suffers the following drawbacks:

1. The size of the cylinder will be too large.
2. Due to compression, there is a rise in temperature of the air. It is difficult to reject heat from the air in the small time available during compression.
3. Sometimes, the temperature of air, at the end of compression, is too high. It may heat up at the cylinder head or burn the lubricating oil.

In order to overcome the above mentioned difficulties, two or more cylinders are provided in series with intercooling arrangement between them. Such an arrangement is known as multistage compression.

3.1.10 Merits of multistage compression:

Following are the main advantages of multistage compression over single stage compression:

1. The work done per kg of air is reduced in multistage compression with intercooler is compared to single stage compression for the same delivery pressure.
2. It improves the volumetric efficiency for the give pressure ratio.
3. The size of the two cylinders (i.e. high pressure & low pressure) may be adjusted to suit the volume and pressure of the air.
4. It reduced the leakage loss considerably.
5. It give more uniform torque, and hence a smaller size fly wheel is required.
6. It provides effective lubrication because of lower temperature range.
7. It reduces the cost of compressor.

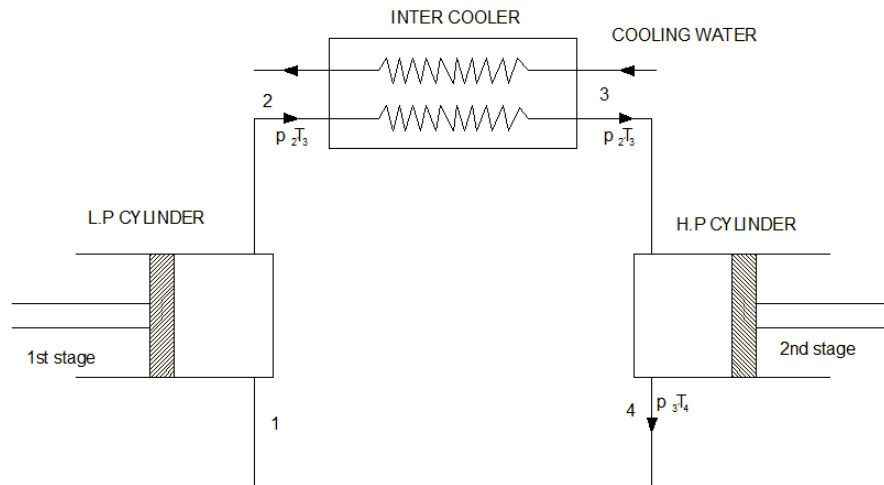### 3.1.11 Two- stage reciprocating air compressor with inter cooler :

Fig 3.5

A schematic arrangement for a two stage reciprocating air compressor with water cooled inter cooler is shown in fig.

First of all, the fresh air is sucked from the atmosphere in the low pressure (L.P) cylinder during its suction stroke at intake pressure p_1 and temperature T_1. The air, after compression in the L.P cylinder (I.e. first stage) from 1 to 2, is delivered to the inter cooler at pressure p_2 and temperature T_2. Now the air is cooled in the intercooler from 2 to 3 at constant pressure p_2 and from temperature T_2 to T_3. After that, in the air is sucked in the high pressure (H.P) cylinder during its suction stroke. Finally, the air, after further compression in the H.P cylinder (i.e. second stage) from 3 to 4, is delivered by compressor at pressure p_3 and temperature T_4.

3.1.12 Intercooling of air in a Two-stage Reciprocating Air Compressor

In the previous articles, we have discussed the working of a two-stage reciprocating air compressor with an intercooler in between the two stages. As a matter of fact, efficiency of the intercooler plays an important role in the working of a two-stage reciprocating air compressor. Following two types of intercooling are important from the subject point of view:

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Fig 3.6

Complete or perfect intercooling. When the temperature of the air leaving the intercooler (i.e. T3) is equal to the original atmospheric air temperature (i.e. T1), then the intercooling is known as complete or perfect intercooling. In this case, the point 3 lies on the isothermal curve as shown in Fig. 3.6
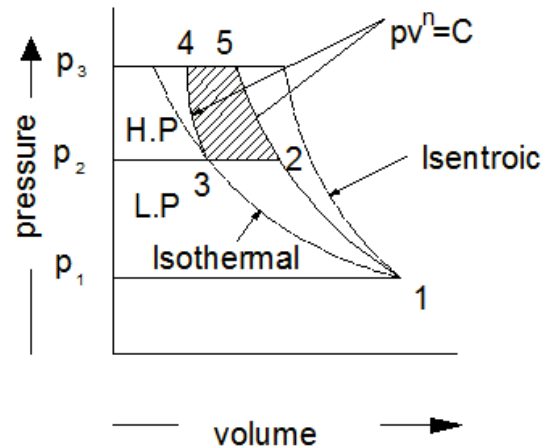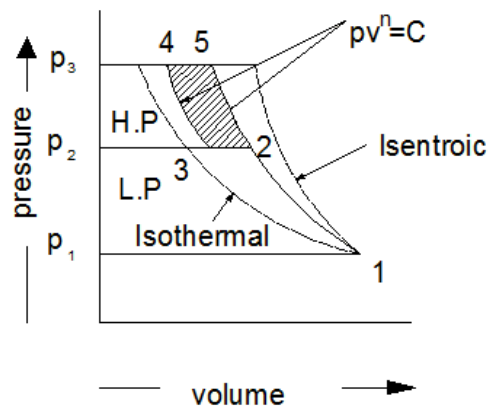

Fig 3.7

Incomplete or imperfect intercooling. When the temperature of the air leaving the intercooler (i.e. T3) is more than the original atmosphere air temperature (i.e. T1), then the intercooling is known as incomplete or imperfect intercooling. In this case, the point 3 lies on the right side of the isothermal curve as shown in fig. 3.7

3.1.13 Workdone by a Two-stage Reciprocating Air Compressor with Intercooler

Consider a two stage reciprocating air compressor with intercooler compressing air in its L.P. and H.P. cylinders.

Let,

p_1 = Pressure of air entering the L.P. cylinder,

V_1 = Volume of the L.P. cylinder,

p_2 = Pressure of air leaving the L.P. cylinder or entering the H.P. cylinder,

V_2 = Volume of H.P. cylinder,

p_3 = Pressure of air leaving the H.P. cylinder, and

N = Polytrophic index for both the cylinders.

Now we shall consider both the cases of incomplete intercooling as well as complete intercooling one by one.

1. **When the intercooling is incomplete**
    We know that work done per cycle in L.P. cyli9nder,

$$W_1 = \frac{n}{n-1} \times p1 \, V1 \left[\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1 \right] \qquad \text{.... (i)}$$

Similarly, work done per cycle in compressing air in H.P. cylinder,

$$W_2 = \frac{n}{n-1} \times p2 \, V2 \left[\left(\frac{p3}{p2}\right)^{\frac{n-1}{n}} - 1 \right] \qquad \text{.... (ii)}$$

∴ Total work done per cycle,

$$W = W_1 + W_2$$

$$W = \frac{n}{n-1} \, X \, p1V1 \left[\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1 \right] + \frac{n}{n-1} \, X \, p2V2 \left[\left(\frac{p3}{p2}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{n}{n-1} \left[\left[p1V1 \left[\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1 \right] \right] + \left[p2V2 \left[\left(\frac{p3}{p2}\right)^{\frac{n-1}{n}} - 1 \right] \right] \right] \qquad \text{.... (iii)}$$

2. When the intercooling is complete

In case of complete intercooling $p_1V_1 = p_2V_2$. Therefore substituting this value in equation (iii),

$$W = \frac{n}{(n-1)} \times p_1V_1 \left[\left(\frac{p2}{p1}\right)^{\wedge}\left(\frac{n-1}{n}\right) + \left(\frac{p3}{p2}\right)^{\wedge}\left(\frac{n-1}{n}\right) - 2\right]$$

$$= \frac{n}{(n-1)} \times m R T_1 \left[\left(\frac{p2}{p1}\right)^{\wedge}\left(\frac{n-1}{n}\right) + \left(\frac{p3}{p2}\right)^{\wedge}\left(\frac{n-1}{n}\right) - 2\right]$$

Problems - Incomplete intercooling:

1. Estimate the work done by a two stage reciprocating single acting air compressor to compress $2.8 m^3$ of air per minute at 1.05 bar and 10^0 C to the final pressure of 35 bar. The intermediate receiver cools the air to 300 c and 5.6 bar pressure. For air, take n = 1.4.

Given:

$V_1 = 2.8 \ m^3/min$	$p_1 = 1.05 \times 10^5 \ N/m^2$	$T_1 = 10 + 273 = 283 \ K$
$p_3 = 35 \times 10^5 N/m^2$	$p_2 = 5.6 \times 10^5 N/m^2$	$T_3 = 30 + 273 = 303 \ K$
$n = 1.4$		

To find:

Work done

Solution:

Let, $\qquad$ V_2 = volume of the high pressure cylinder.

WKT, $\qquad \frac{p1V1}{T1} = \frac{p2V2}{T3}$

$$V_2 = \frac{p1V1T3}{p2T1} = \frac{1.05 \times 10^5 \times 2.8 \times 303}{5.6 \times 10^5 \times 283} = 0.562 \ m^3/min$$

Work done by the compressor,

$$\frac{n}{n-1}\left[\left[p1v1\left[\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right]\right] + \left[p2v2\left[\left(\frac{p3}{p2}\right)^{\frac{n-1}{n}} - 1\right]\right]\right]$$

$$= \frac{1.4}{1.4-1}\left[1.05 \times 10^5 \times 2.8 \times \left(\frac{5.6}{1.05}\right)^{\frac{1.4-1}{1.4}} - 1\right]$$

$$+ \left[5.6 X 10^5 X 0.562 X \left(\frac{35}{5.6}\right)^{\frac{1.4-1}{1.4}} - 1 \right]$$

W = 3.5 [1.803 X 10^5 + 2.166 X 10^5]

W = 1389150 N-m

2. A two stage single acing reciprocating air compressor draws in air at a pressure of 1 bar and 17 ° C and compresses it to pressure of 60 bar. After compression in the LP. Cylinder, the air is cooled at constant pressure of 8 bar to a temperature of 37 ° C. The LP. Cylinder has a diameter of 150mm and both the cylinder have 200mm stroke. If the law of compression is pv$^{1.35}$ = C, find the power of the compressor, when it runs at 200 r.p.m. Take R = 287 J/kg K

Given:

p_1= 1 X 10^5 N/m^2 T_1 = 17+273= 290 K p_3 = 60 X 10^5 N/m^2

p_2= 8X 10^5 N/m^2 T_3 = 37+273= 310 K   D = 0.15 m

L = 0.2 m   n = 1.35   N = 200 R.P.M

R = 287 J/kg K

To find:

Power of the compressor

Solution:

WKT, Volume of LP. Cylinder,

$$V_1 = \frac{\pi}{4} X D^2 X L = \frac{\pi}{4}(0.15)^2 . 0.2 = 0.0035 \text{ m}^3$$

Let, V_2 = Volume of HP. Cylinder,

WKT, $\frac{p1V1}{T1} = \frac{p2V2}{T3}$

$$V_2 = \frac{p1V1T3}{p2T1} = \frac{1 \text{ X } 10^5 X \text{ } 0.0035 \text{ X } 310}{8 \text{ X } 10^5 \text{ X } 290} = 0.00047 \text{ m}^3$$

Work done by the compressor per stroke,

$$= \frac{n}{n-1} \left[\left[p1V1 \left[\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1 \right] \right] + \left[p2V2 \left[\left(\frac{p3}{p2}\right)^{\frac{n-1}{n}} - 1 \right] \right] \right] \quad =$$

$$= \frac{1.35}{1.35-1} \left[1 X 10^5 X 0.0035 X \left(\frac{8}{1}\right)^{\frac{1.35-1}{1.35}} - 1 \right]$$

$$+ \left[8 X 10^5 X 0.00047 X \left(\frac{60}{8}\right)^{\frac{1.35-1}{1.35}} - 1 \right]$$

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= 3.86 (250+258) = 1961 N-m

Since the compressor is single acting, therefore number of working strokes per minute,

$$N_w = N = 200$$

WKT, the power of the compressor,

$$P = \frac{W \times N_w}{60} = \frac{1961 \times 200}{60} = 6536.66 \text{ W} = 6.53 \text{ KW}$$

$$P = 6.53 \text{ KW}$$

Problems - Complete intercooling:

1..A two stage air compressor compresses air from 1 bar and 20^0C to 42 bar, if the law of compression is PV^1.35 = constant and the intercooling is complete to 20^0C. Find per kg of air, 1.)The work done in compressing, if the temperature rise of cooling water is 25^0C. Take R = 287 J/kg K, Cp = 1KJ/kg K.

Given:

$p_1 = 1$ bar $= 1 \times 10^5$ N/m², $T_1 = 20^0C = 20+273 = 293$ K, $p\,3 = 42$ bar $= 42 \times 10^5$ N/mm², n = 1.35, T3 = $20^0C = 20+273 = 293$ K, m = 1 kg, rise in temperature of cooling water = 25^0C, R = 287 J/Kg K,   Cp = 1KJ/kg K.

Solution:

W.K.T, for perfect intercooling, the intercooling pressure,

$$p\,2 = \sqrt{(p\,1.\,p\,3)} = \sqrt{(1 \times 42)} = \textbf{6.48 bar}.$$

And the volume of air admitted for compression,

$$\mathbf{V1} = \frac{mRT1}{p1} = \frac{1 \times 287 \times 293}{1 \times 10^5} = \textbf{0.84 m³/Kg of air.}$$

1) Work done in compressing the air,

$$\mathbf{W} = \frac{n}{(n-1)} \times p\,1 V1 \left[\left(\frac{p2}{p1}\right)^{\wedge} \left(\frac{n-1}{n}\right) + \frac{p3}{p2}^{\wedge} \left(\frac{n-1}{n}\right) - 2 \right]$$

$$= \frac{1.35}{(1.35-1)} \times 1 \times 10^5 \times 0.84 \times \left[\left(\frac{6.48}{1}\right)^{\wedge} \left(\frac{(1.35-1)}{1.35}\right) + \frac{42}{6.48}^{\wedge} \left(\frac{(1.35-1)}{1.35}\right) - 2 \right] \text{N-m}$$

$$= 3.24 \times 10^5 (1.62 + 1.62 - 2) = \textbf{4.039} \times \textbf{10}^5 \textbf{ N-m}$$

3.1.14 ROTARY AIR COMPRESSORS

Introduction

In a rotary air compressor, the air is entrapped between two sets of engaging surface and the pressure of air is increased by squeezing action or back flow of air.

3.1.15 Roots blower compressor
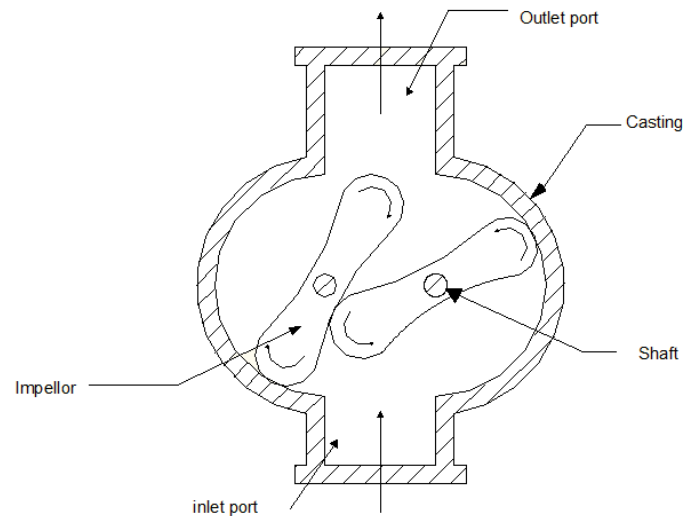
Fig 3.8

A roots blower compressor, in its simplest form, consists of two rotors with lobes rotating in an air tight casing which has inlet and outlet ports. Its action resembles with that of a gear pump. There are many designs of wheels, but they generally have two or three lobes (and sometimes even more). In all cases, their action remains the same. The lobes are so designed that they provide an air tight joint at the point of their contact.

The mechanical energy is provided to the one of the rotors from some external source, while the other gear is driven from the first. As the rotor rotate, the air, at the atmosphere pressure, is dropped in the pockets formed between the lobes and casting. The rotary motion of the lobs delivers the entrapped air into the receiver. Thus more and more flow of air into the receiver increase its pressure. Finally, the air at a higher pressure is delivered from the receiver.

It will be interesting to know that when the rotating lobe uncover the exit port, some air (under high pressure) flows back into the pockets from the receivers. It is known as backflow process. The air, which flows from the receiver to the pocket, gets mixed up with the entrapped air. The backflow of air continues, till the pressure in the pocket and receiver is equalized. Thus the pressure of air entrapped in the pocket is increased at constant volume entirely by the backflow of air the backflow process is shown in the fig. now the air is delivered to the receiver by the lobes. Finally, the air at a higher pressure is delivered from the receiver.

3.1.16 VANE BLOWER COMPRESSOR
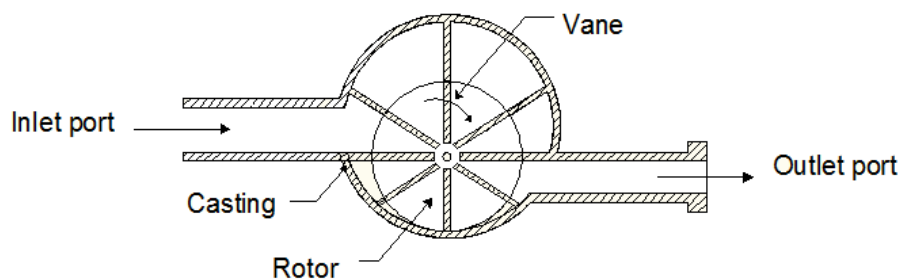
Fig 3.9

A Vane blower, in its simplest form, as shown in fig, consists of a disc rotating eccentrically in an air tight casing with inlet and outlet and outlet ports. The discs has a number of slots (generally 4 to 8) containing vanes. When the rotor rotates the disc, the vanes are pressed against the casing, due to centrifugal force, and form air tight pockets.

The mechanical energy is provided to the disc from some external source. As the disc rotates, the air is trapped in the pockets formed between the vanes and casing. First of all, the rotary motion of the vanes compresses the air. When the rotating vane uncovers the exit port, some air (under high pressure) flows back into the pocket in the same way as discussed in the case of roots blower compressor. Thus the pressure of air, entrapped in the pocket, is increased first by decreasing the volume and then by the backflow of air. Now the air is delivered to the receiver by the rotation of the vanes. Finally, the air at a high pressure is delivered from the receiver.

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3.1.17 CENTRIFUGAL COMPRESSOR

A centrifugal compressor, \in its simplest form, consists of a rotor (or impeller) to which a number of curved vanes are fitted asymmetrically. The motor rotates in an air tight volute casing with inlet and outlet points. The casing for the compressor is so designed that the kinetic energy of the air is converted into pressure energy before it leaves the casing.
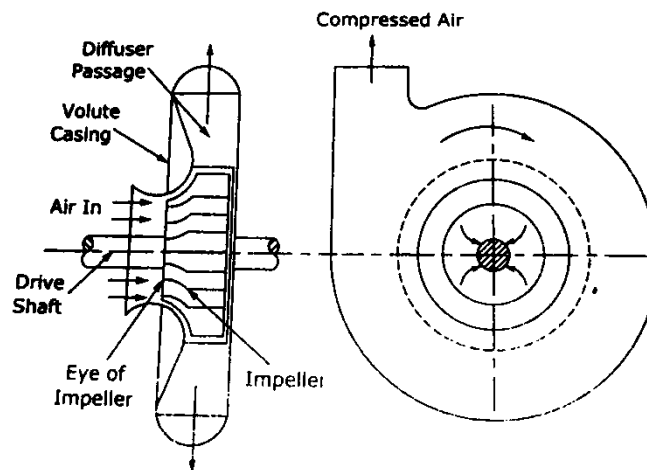
Fig 3.10

The mechanical energy is provided to the rotor from some external source. As the rotor rotates, it sucks air through its eye, increases its pressure due to centrifugal force and forces the air to flow over the diffuser. The pressure of air is further increased during its flow over the diffuser.

Finally, the air at a high pressure is delivered to the receiver, It will be interesting to know that the air enters the impeller radially and leaves the vanes axially.

3.1.18 AXIAL FLOW COMPRESSOR

An axial flow compressor, in its simplest form, consists of a number of rotating blade rows fixed to a rotating drum. The drum rotates inside an air tight casing to which are fixed stator blade rows. The blades are made of aerofoil section to reduce the loss caused by turbulence and boundary separation.
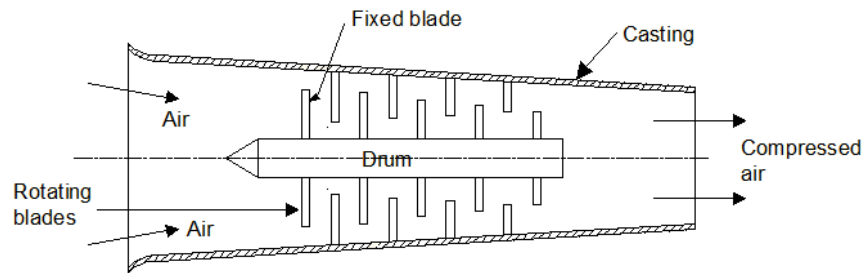
Fig 3.11

The mechanical energy is provided to the rotating shaft, which rotates the drum. The air enters from left side of the compressor. As the drum rotates, the air flow through the alternately arranged stator and rotor. As the air flows from one set of stator and rotor to another, it gets compressed. Thus successive compression of the air, in all the sets of stator and rotor, the air is delivered at a high pressure at the outlet point.

3.2 GAS TURBINES

A gas turbine is a rotary machine. It consists of three main parts – a compressor, a combustion chamber and turbine. The atmospherics air is taken into the compressor and compressed. The compressed air is heated by buming the fuel in the combustion chamber. The heated air and products of combustion is expanded in a turbine doing work output. About two – third of power produced is used to drive compressor and one third is available as useful work output. Gas turbines are used in aircraft, marine craft etc.

3.2.1 Classification

Gas turbines are classified as follows

1) Account to the path of the working fluid
   (a) Open cycle gas turbine
   (b) Closed cycle gas turbine
   (c) Semi closed cycle gas turbine
2) According to process of heat absorption
   (a) Explosion or constant volume gas turbine
   (b) Continuous combustion or constant pressure gas turbine

3.2.2 Advantages and Disadvantages of Gas turbine

Advantages :

1. Its mechanical efficiency is higher.
2. The weight of gas turbine per power developed is less .
3. The installation and running cost is less.
4. The lubrication and ignition systems are simple.
5. the exhaust to a gas turbine is free from smoke and less polluting.
6. The operation pressure is very low.
7. It is free vibrations since it develops uniform torque.
8. It can be driven at a very high speed.
9. They are suitable for air crafts.
10. No flywheel is required.
11. It has no reciprocating parts.

3.2.3 Disadvantages :

1. The starting of a gas turbine is not simpler.
2. The cooling system is complicated.
3. Operating temperature is below 900°C.

3.2.4 <u>Open Cycle gas turbine</u>

An open cycle gas turbine, in its simplest form, consists of a compressor, combustion chamber and a gas turbine which drives the generator and compressor.
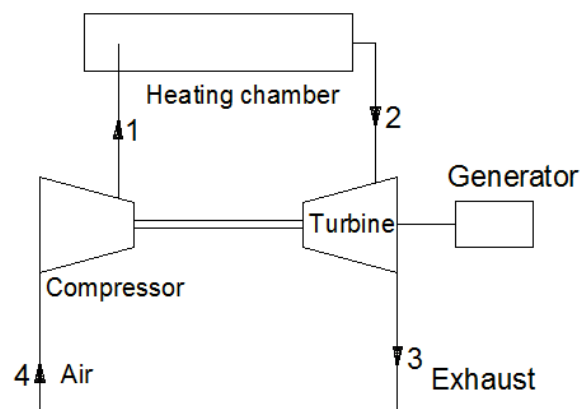

Fig 3.12

The schematic arrangement of an open cycle gas turbine is shown in Fig. In this turbine, the air is first sucked from the atmosphere and then compressed isentropically (generally in a rotary compressor) and then passed into the combustion chamber. The compressed air is heated by the combustion of fuel and the products of combustion (i.e. hot gases formed by the combustion of fuel) also get mixed up with the compressed air, thus increasing the mass of compressed air. The hot gas is then made to flow over the turbine blades (generally of reaction type). The gas, while flowing over the blades, gets expanded and finally exhausted into the atmosphere.

An open cycle gas turbine is also called a continuous combustion gas turbine as the combustion of fuel takes place continuously. This turbine also works on Joule's cycle. The relations for work done by the compressor and turbine are same as those of closed cycle gas turbine.

Merits and Demerits of Open Cycle Gas Turbine

Merits

1. Simple in construction
2. Cost is less
3. Best suitable for moving vehicle
4. Smaller in size
5. It is best suitable for air crafts since weight is less.

Demerits

1. Thermal efficiency is less
2. Combustion of fuel is internal. This dose not permit the use of any kind of fuel.
3. The turbine blades are fouled by the products of combustion.
4. Frequent cleaning of the system is necessary.

3.2.5 Methods To Improve the Efficiency

The efficiency and the power output of the simple gas turbine cycle can be improved by introducing and inter cooling.

3.2.6 Gas Turbine Cycle with Heat Generator
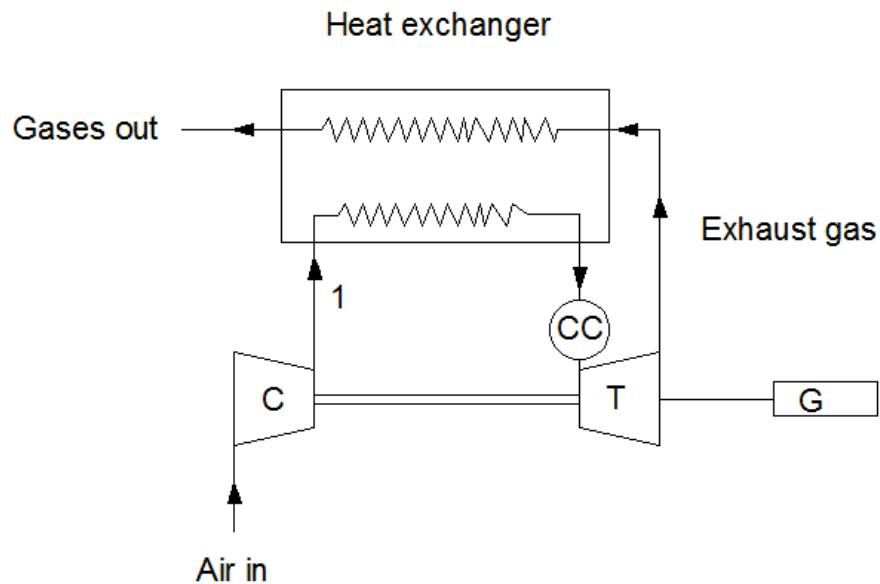
Fig 3.13

Turbine blades, and expand over them producing work at the output shaft. Finally the gases are exhausted into the atmosphere. The T-s diagram is shown in figure

Effects of Inter cooling's

1. Due to the reduction in specific volume of air, smaller compressor can be used.
2. The power requires to the compressor is reduced. Hence net work output is increased.
3. Heat supplied is more.
4. Thermal efficiency is reduced.
5. Requirement of water for the plant is more.
6. Cost of the plant is more.

3.2.7 Gas Turbine With Intercooling

We have already discussed that a major portion of the power developed by the gas turbine is utilised by the compressor. It can be reduced by compressing the air in two stages with an intercooler between the two. This improves the efficiency of the gas turbine. The schematic arrangement of a closed cycle gas turbine with an intercooler is shown in Fig.
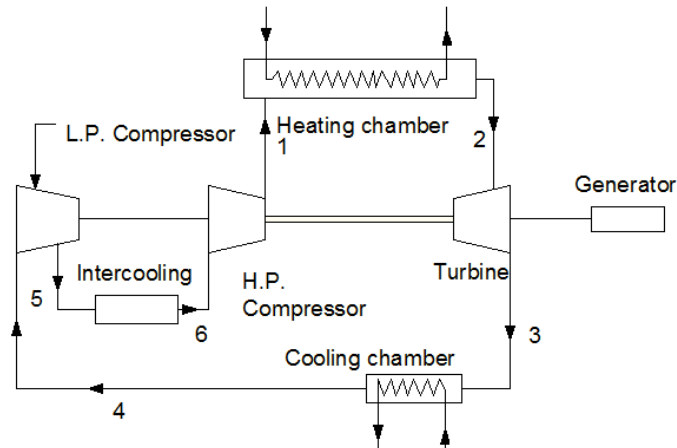
Fig 3.14

In this arrangement, first of all, the air is compressed in the first compressor, known as low pressure (L.P) compressor. We know that as a result of this compression, the pressure and temperature of the air is increased. Now the air is passed to an intercooler which reduces the temperature of the compressed air to its original temperature, but keeping the pressure constant. After that, the compressed air is once again compressed in the second compressor known as high pressure (H.P) compressor. Now the compressed air is passed through the heating chamber and then through the turbine. Finally, the air is cooled in the cooling chamber and again passed into the low pressure compressor as shown in Fig.

The process of intercooling the air in two stages of compression is shown on T-s diagram in Fig. The process 1-2 shows heating of the air in heating chamber at constant pressure. The process 2-3 shows isentropic expansion of air in the turbine. The process 3-4 shows cooling of the air in the cooling chamber at constant pressure. The process 4-5 shows compression of air in the L.P. compressor. The process 5-6 shows cooling of the air in the intercooler at constant pressure. Finally, the process 6-1 shows compression of air in the H.P. compressor.

3.2.8 Gas Turbine With Reheating

The output of the gas turbine can be considerably improved by expanding the hot air in two stages with a reheater between the two. The schematic arrangement of a closed cycle gas turbine with reheating is shown in Fig.
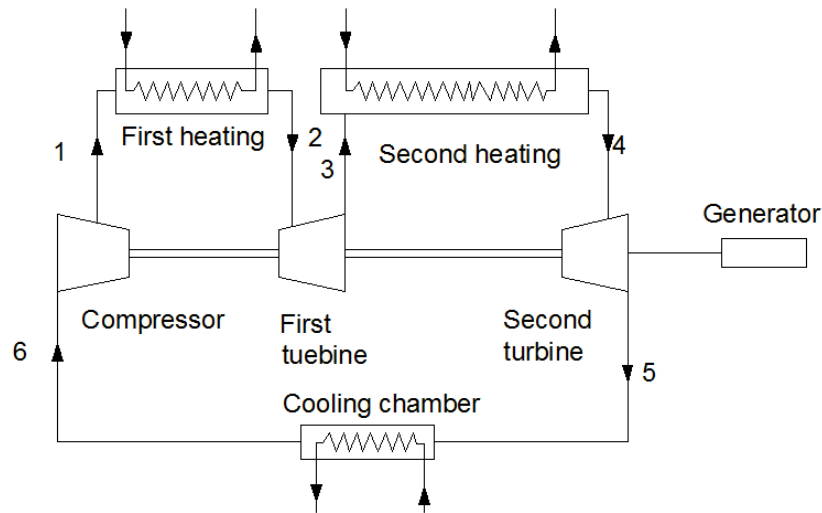
Fig 3.15

In this arrangement, the air is first compressed in the compressor, passed into the heating chamber, and then to the first turbine. The air is once again passed on to another heating chamber and then to the second turbine. Finally, the air is cooled in the cooling chamber and again passed into the compressor as shown in Fig.

The process of reheating in the two turbines is shown on T-s diagram in Fig. The process 1-2 shows heating of the air in the first heating chamber at constant pressure. The process 2-3 shows isentropic expansion of air in the first turbine. The process 3-4 shows heating of the air in the second heating chamber at constant pressure. The process 4-5 shows isentropic expansion of air in the second turbine. The process 5-6 shows cooling of the air in the intercooler at constant pressure. Finally, the process 6-1 shows compression of air in the compressor.

3.2.9 Closed Cycle gas turbines:

A closed cycle gas turbine, in its simplest form, consists of a compressor, heating chamber, gas turbine which drives the generator and compressor, and a cooling chamber.

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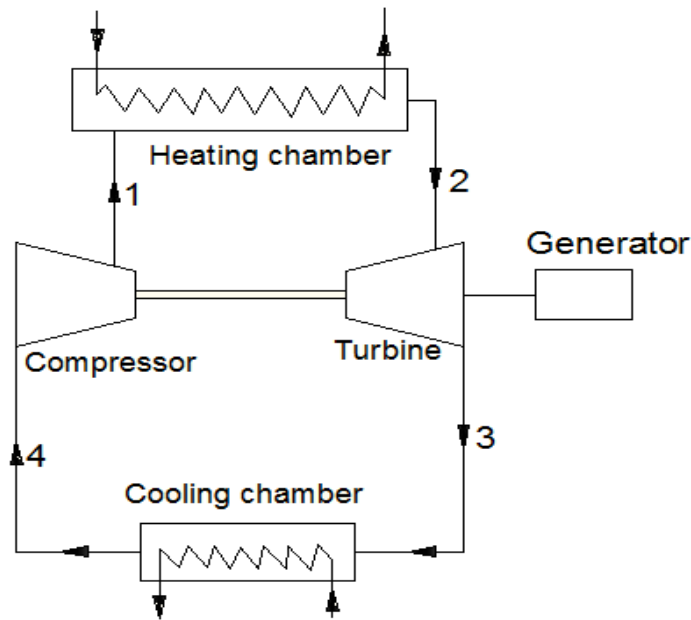
Fig 3.16

The schematic arrangement of a closed cycle gas turbine is shown in Fig. In this turbine, the air is compressed isentropically (generally in rotary compressor) and then passed into the heating chamber. The compressed air is heated with the help of some external source, and made to flow over the turbine blades (generally reaction type).The gas, while flowing over the blades, gets expanded. From the turbine, the gas is passed to the cooling chamber where it is cooled at constant pressure with the help of circulating water to its original temperature. Now the air is made to flow into the compressor again. It is thus obvious, that in a closed cycle gas turbine, the air is continuously circulated within the turbine.

Merits and Demerits of Closed Cycle Gs Turbine

Merits

1. Combustion of fuel is external. This permits the use of any kind of fuel.
2. Internal cleaning of the system is not necessary
3. The turbine blades are not fouled by the products of combustion.
4. Thermal efficiency is more

Demerits

1. Lager in size
2. A coolent is needed for precooler
3. Complicated in construction

3.2.10 Uses of Gas Turbines

The gas turbine are employed :-

1. In jet propulsion units for aircraft
2. In ships as propulsion units
3. For electric power generation
4. To run locomotives and automobiles
5. In the process of super charging I.C. engines.

3.2.11 Propulsion

Jet propulsion plant is for propulsion of air craft and other missiles by the reaction of a jet of gases which are discharged out with a high velocity from the rear side of the unit. The mass and velocity of the jet is obtained from the combustion of fuels.

All the propulsive devices are based on Network's second and third law of motion. All the devices cause a change in momentum of a mass of this accelerating mass of fluid propulsive force called thrust is produced in a direction opposite to that of mass flow, a propulsive force called thrust is produced. This trust propels the unit.

3.2.12 Types of jet propulsion units

The different types of jet propulsion plant as classified as (1) Turbo jet engine (2) Turbo propeller engine (3) Ram jet

3.2.13 Turbo jet engine

D-Diffuser
C- Compressor
S-Shaft
CC - Combustion Chamber

P- Pump
FN - Fuel Nozzle
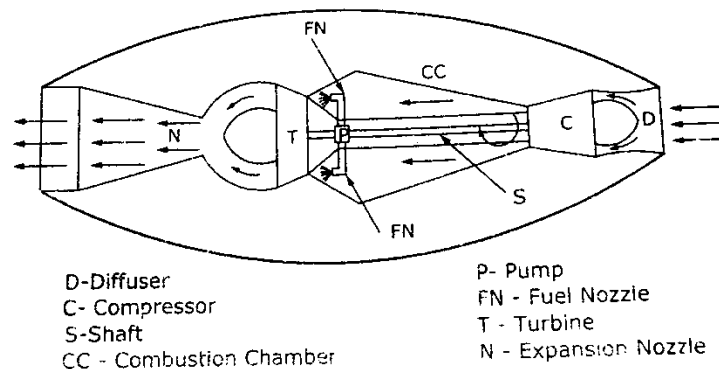T - Turbine
N - Expansion Nozzle

Fig 3.17

A brief outline of simple turbo jet unit is shown in fig. The air enters the unit at a velocity equal to that of the aircraft, through the diffuser (D) whereby the pressure of air rises above atmospheric. It is further compressed by the rotary compressor (c) (Which is usually of the axial flow type) to a pressure of about 3.5 bar and supplied to the annular combustion chamber (cc). The liquid fuel under pressure is injected and sprayed into the chamber by means of the pump (P) through a ring of fuel nozzles (FN) and is burnt at constant pressure. The products of combustion are then expanded in an axial flow turbine (T) where they lose some of their pressure thus imparting rotary motion to the turbine, the sole aim of which is to drive the compressor (C) through the shaft (S). In other words, the power developed by the turbine (T) is just nearly equal to the power required to drive the compressor (C) and the pump (P). The products of combustion after expansion through the turbine are discharged through the nozzle (N). Now, as the gases issue out from the nozzle, into the atmosphere, their pressure drops with a consequent increase in the velocity e.g., the gases emerge out with a tremendous velocity, and in turn give a thrust to the entire unit in the forward direction according to the Newton's Third law of motion. The source providing energy for the forward thrust in this case is only the high velocity jet and hence the name turbojet. The conventional jet engine is more correctly termed as turbojet unit. The air fuel ratio generally used is 60:1 so that the temperature of the gases at entrance to the turbine are kept within the desired limits. The compressor (C) may be built in several stages in order to achieve high pressure. Similarly the turbine (T) may also have a number of stages instead of single stage of expansion.

When the shaft which has to run at very high speed is supported on three bearings, there is likelihood of misalignment due to thermal expansion. This difficulty can be overcome by employing suitable means like cooling of bearings and making use of the lubricating oil of low viscosity, suitable for high speed operations. It is also quite a common practice to mount the shaft on two bearings, only.

For starting purpose an electric motor or compressed air motor is employed. In place of the annular combustion chamber shown in fig, some turbojet units are provided with a number (say 8 or 10) of separate combustion chambers spaced at equal intervals along the circumference.

Turbojet engines are generally used in high speed fighters, bombers and airlines.

3.2.14 Merits and Demerits of turbo jet

Merits

1. simple in construction.
2. Maintenance cost is less.
3. No loss of power transmission since thrust is applied directly.
4. The engine vibrations are absent.
5. Smaller frontal area.
6. Very high speeds can be achieved.
7. Low grade fuel can be used.

Demerits

1. During take off, the trust is low.
2. Fuel consumption is more.
3. The materials used are very costly.
4. Life of the units is shorter.
5. It produces more noise.

3.2.15 Turbo propeller engines

This unit is also commonly known as "Turbo-prop engine". Its basic principle is illustrated in fig. It resembles somewhat in construction to the turbo units. But the total propulsive effort or thrust in the case derived from action of the propeller (X) in addition to a small forward thrust from the reaction of the waste combustion products which are passed rear wards through the nozzle (N) of the unit at a high velocity.

X - Propeller   CC - Combustion Chamber
RG - Reduction Gear   P- Pump
D-Diffuser   FN - Fuel Nozzle
C- Compressor   T - Turbine
S-Shaft   N - Expansion Nozzle
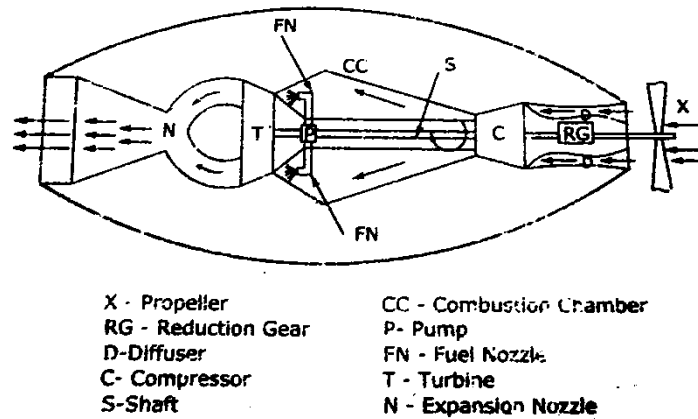
Fig 3.18

Thus the turbine (T) not only drives the compressor (C) but also impart a rotary motion to the propeller (X), so the capacity of the turbine in this case should be higher than in the case of jet propulsion unit. Now the angular velocity of the shaft (S) is extremely high and the propeller (X) cannot be run at a higher angular velocity. Therefore the R.P.M of the shaft is first reduced to a much lower value by providing a reduction gear (RG) before the power is transmitted to the propeller. The compressor (C) and the turbine (T) may be built in several stages. It is quite common to rotate the propeller (X) by power supplied from one of the two turbines (H.P. or L.P. turbine) while the other turbine drives the compressor. In this case there are two independent shafts for the two different drives. The gases after expansion through one turbine are led to the other turbine for further expansion before directed towards the nozzle (N).

Merits and Demerits of Turbo Propeller

Merits

1. Simple in construction .
2. More power is developed for take off.
3. High propulsive efficiency at low speeds (below 800 km/hr)
4. Small frontal area.
5. Less vibrations and noise.
6. High propulsive efficiency both at high and low altitudes.

Demerits

1. The propeller efficiency is reduced at high speeds.

2. Plant cost is more.

3. Occupy more space.

4. Due to the presence of reduction gears power may waste.

3.2.16 Ram jet

When the air is compressed to the required ignition pressure by the ram effect and the high velocity jet is produced without the help of any machinery, so that propulsion unit move with very high speed, then it is known as the ram jet. The ram effect is produced due the inertia of the stream of entering air.
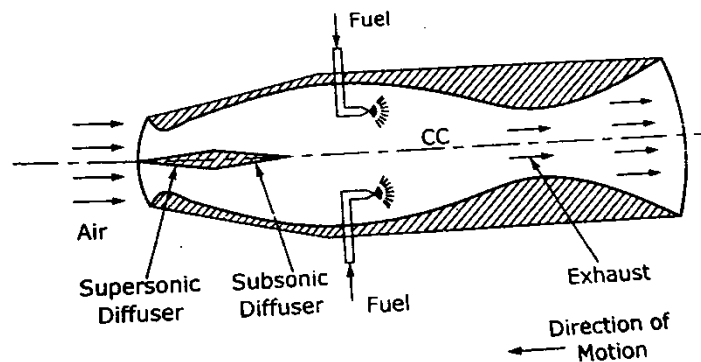
Fig 3.19

A Ram jet or Athodyd is very simple in construction and operation and operation. It does not require the compressor (and hence no need of turbine) because the compression of air takes place by ramming effect. It consist of following three major components:

1. Diffuser system.   2. Combustion chamber.   3. Exhaust nozzle.

The diffuser system consists of two parts, the supersonic diffuser and the subsonic diffuser. As the unit moves in air at supersonic velocity the entering air is first decelerated to a sonic velocity in the supersonic diffuser. This gives rise to shock waves and increase of pressure of the air. Further as the air flows through the subsonic diffuser its velocity is further reduced from the sonic value to sub sonic value (usually 02 Mach. Number). The diffuser system thus reduces the velocity of air considerably and to a value at which the combustion fuel is burnt in the presence of air raising the temperature of the products of combustion to as high a value as 2000 ^{0}C. These hot gases are then expanded through the exhaust nozzle and finally discharged to the atmosphere

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in the form of a high velocity jet imparting forward thrust to the unit. The unit is not self-operating when at rest and has therefore got to the accelerated to a flight velocity at which sufficient ram pressure is developed for combustion of fuel in the combustion chamber.

Merits and Demerits of Ram jet

Merits

1. No moving parts.
2. Simple in construction .
3. produces more thrust than turbo jet.
4. Cost is less.
5. no upper limit for flight speed.
6. Wide range of fuels can be used.

Demerits

1. It requires an external launching device.
2. The fuel consumption is more at low speeds.

For steady combustion, flame holders are necessary.

3.2.17 ROCKET PROPULSION:

Introduction: Like in a jet engine thrust in rocket is generated by a jet of exhaust gases issuing out at high velocity at the rear. A rocket should be clearly distinguished from a jet engine in that the former must carry all its supplies of oxygen in addition to fuel with it; whereas the latter is dependent upon a continuous feed of atmospheric oxygen. In a solid fuel rocket the oxygen is carried in the form of chemical which are mixed with the fuel properly. But in a liquid fuel rocket the oxygen is carried in separate fuel tanks. In other words, a rocket does not depend upon the oxygen in the atmosphere for its operation. Therefore it can function in vacuum and its thrust is independent of the environments. When a rocket is used as weapon with a warhead as pay load it is called a "missile". It should be noted that the conventional reciprocating engines, turbojets and other forms of air breathing engines have also been employed on guided missiles.

3.2.18 Rocket engines:

Fuel and oxygen burn and produces gases. They issue out of nozzle at very high velocity and produces thrust. This thrust propels the rocket engine in the forward direction. The oxygen required for burning of fuel is not taken from atmosphere. But, it is carried by the engine itself. So, rocket can operate in vacuum space.

Working:

Fuel and oxygen burn in the combustion chamber. High pressure gases are produced. They go out from nozzle at supersonic velocity. Thrust is produced. It propels the rocket in the forward direction. Missile attacks the target. This type of rocket was used by Germany in world war.

3.2.19 Application of rockets:

1.   Satellites
2.   Space crafts
3.   Signalling purpose
4.   Fireworks display
5.   Missiles
6.   Weather forecast.

UNIT 4 – FORMATION & PROPERTIE OF STEAM AND STEAM CALORIMETERS

4.1 Steam – Introduction

➢ Steam is water vapour.

➢ It is used for heating and as working medium in steam turbines and steam engines.

4.1.1 Steam Properties

➢ It can carry huge amount of heat

➢ It is produced from water which is cheap and easily available

➢ Exhaust steam can be also used for heating purpose

4.1.2 Formation of steam

➢ Let us take one kg of ice cube **under atmospheric pressure** placed in a container.

➢ It is gradually heated keeping the pressure constant.

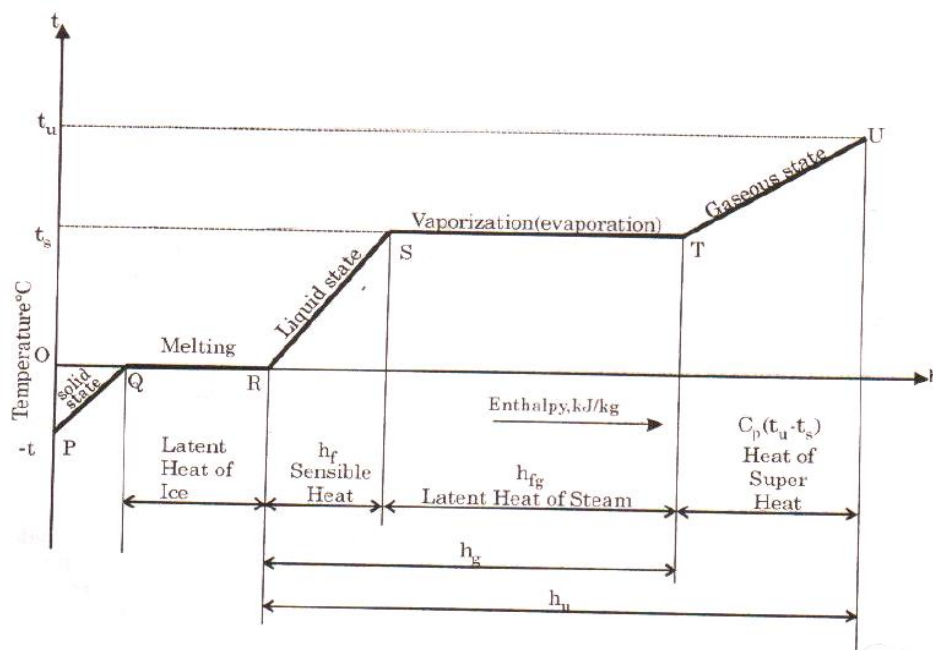➢ Figure shows the formation of steam under atmosphere pressure on *T – h diagram*.

Fig 4.1

There are five stages in the formation of steam. They are

i) Solid stage

ii) Melting stage

iii)Liquid stage

iv) Evaporation stage

v) Super heated stage

i) Solid stage

➢ In the sold stage, ice is solid state.

➢ When the temperature of ice reaches 0 ° C ice starts melting.

➢ The temperature corresponding to melting is known as *freezing point or melting point of ice.*

➢ Solid stage is shown by the line PQ on the T – h diagram in the figure

ii) Melting stage

➢ The heat supplied during this stage *converts ice into water* at constant temperature (0° C).The supplied during this stage is known as *latent heat of fusion or latent heat*.

➢ Melting stage is shown by a horizontal line QR on the T-h diagram in the figure.

iii) Liquid stage

➢ *Further addition of heat rises the temperature of water, till the water beings to boil. The temperature at which the water beings to boil to known as **saturation temperature**.*

➢ The heat supplied during this stage is known as liquid enthalpy or sensible heat of water . It is denoted by h_f. Liquid stage is shown by the line RS on the T-h diagram.

iv) Evaporation stage

➢ In this stage water is changed into steam at saturation temperature.

➢ It is presented by horizontal line ST on the T – h diagram.

➢ The point S is the beginning of the evaporation process and it is all water at saturation temperature.

➢ The point T is the end of evaporation process and it is all dry.

➢ The steam produced in between S and T contains droplets of water in suspension and this steam is known as wet steam.

➢ The heat supplied during stage is known as specific enthalpy of evaporation or latent heat of evaporation. It is denoted by h_{fg}.

iv) Super heating stage

Further heating of dry steam causes the temperature rise and produces super heated steam.

This is represented by the TU on the T – h diagram. The heat added during stage is known as heat of super heat or super heat enthalpy.

The difference between the temperature of superheated steam and saturation temperature is known as degree of superheat (T_{sup} - t_s)

4.1.3 Saturation temperature (t_s)

➤ The temperature at which the phase change takes place at a given pressure is known as saturation temperature.

➤ For atmosphere pressure (1.01325 bar) the saturation is 100°C

4.1.4 Enthalpy of water (h_f)

➤ It defined as the amount of heat required to rise the temperature of one kg water °C to the saturation temperature.

➤ It is denoted by h_f. It is expressed in kJ/ kg.

4.1.5 Enthalpy of evaporation (h_{fg})

➤ It defined as the amount of heat required to convert one kg of water at saturation temperature into dry steam completely.

➤ It is denoted by h_{fg}. It expressed in kJ/kg

4.1. 6 Conditions of steam

Steam occurs in following conditions

1. Wet steam
2. Dry steam or dry saturated steam and
3. Super-heated steam

1. Wet steam

➤ When the steam contains moisture or particles of water in suspension, it is said to be wet steam.

➤ It means that the evaporation of water is not complete, and the whole of the latent heat has not been absorbed.

2. Dry steam or dry saturated steam

➤ When the wet steam is further heated, and it does not contain any suspended particles of water, it is known as dry saturated steam.

➤ The dry saturated steam has absorbed its full latent heat and behaves practically, in the same way as a perfect gas.

3. Superheated steam

When the dry steam is further heated at constant pressure, thus raising its temperature, it is said to be superheated steam. Since the pressure is constant, therefore the volume of superheated steam increases. It may be noted that the volume of one kg of superheated steam is considerably greater than the volume of one kg of dry saturated steam at the same pressure.

In actual practice, the superheated steam is produced in a separate apparatus known as superheater, so that it is out of contact with water from which it is formed.

4.1.7 Dryness fraction (x)

It is the ratio of the mass of dry steam to the mass of wet steam containing it

It is denoted by x or q

Dryness fraction $= \dfrac{\text{mass of dry steam contained in wet steam }(mg)}{\text{mass of wet steam }(mw)}$

Let

m_g - mass of dry steam contained in the wet steam in kg

m_f - mass of water particles present in the wet steam in kg

m_w - mass of wet steam in kg ($m_g + m_f$)

then dryness fraction $= x = \dfrac{mg}{mw} = \dfrac{mg}{mg+mf}$

NOTE:

1. When $x < 1$: The steam is wet steam

2. When $x = 1$: The steam is dry and saturated steam.

3. When $x > 1$: The steam is superheated steam

4.1.8 Specific Enthalpy of Wet steam:

It is the total amount of heat received by 1 kg of water from 0 °C at constant pressure to convert it to wet steam. The enthalpy of wet steam is given by:

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

Where x is the dryness fraction of steam

4.1.9 Specific Enthalpy of Dry steam:

It is the total amount of heat received by 1 kg of water from 0 °C at constant pressure to convert it to dry steam. The enthalpy of dry steam is given by:

Enthalpy or total heat of steam = Sensible heat + Latent heat

It is denoted by h_g and its value for dry saturated steam may be read direct from the steam tables.

We know that in case of dry steam, $x = 1$.

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

4.1.10 Specific Enthalpy of Superheated steam:

If we further add heat to the dry steam, its temperature increases while pressure remaining constant. The increase in temperature shows the superheat stage of the steam. Thus, the total heat required for the steam to be superheated is :

h_{sup} = Total heat for the dry steam + Heat for superheated steam

$$= h_f + h_{fg} + c_p\,(t_{sup} - t) = h_g + c_p\,(t_{sup} - t)$$

Where C_p = Mean specific heat at constant pressure for superheated steam

T_{sup} = Temperature of the superheated steam, and

T = saturation temperature at the given constant pressure.

4.1.11 Advantages of superheated steam

1. The superheated steam contains more heat contents, and hence its capacity to do work is also increased without increasing its pressure.

2. The superheating is done in a superheater, which obtains heat from waste furnace gases. These gases would have otherwise passed, uselessly, through the chimney.

3. The high temperature of the superheated steam results in an increase of thermal efficiency. Since the superheated steam is at a high temperature than that corresponding to its pressure, therefore it can be considerably cooled during expansion in an engine cylinder. This is done before the temperature of superheated steam falls below that at which it condenses and, thereby, becomes wet. It is thus obvious, that heat losses due to condensation steam on cylinder walls, walls, etc,. are avoided to a great extent.

4.1.12 p-V diagram
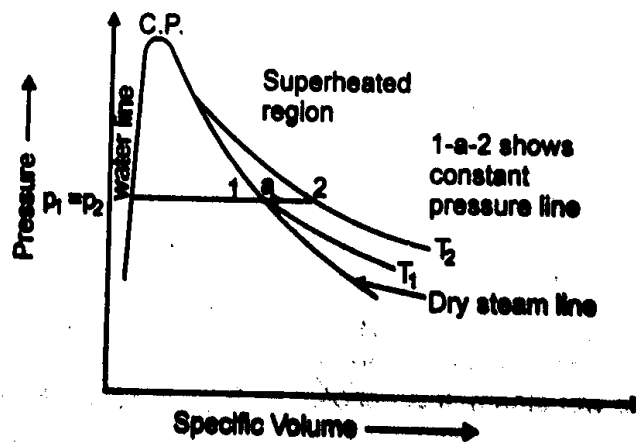
Fig 4.2

➢ In this diagram, the pressure is plotted in y axis and the specific volume is plotted in x-axis. The figure shows the pressure volume diagram of steam.

➢ The curve on the left marked as water line shows the relationship between the pressure and specific volume of water.

➢ In the figure, the curve on the right marked is dry steam line shows the relationship between the pressure and specific volume of dry saturated steam.

➢ The water line and dry steam line meet at a point, which is the critical point .The pressure (221.2 bar) corresponding to the critical point is called to critical pressure.

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➤ The temperature (374.15°C) corresponding to the critical point is called the critical temperature. At the critical pressure the water is changed into dry steam directly. The area between these curves is called wet region. Here the steam is wet steam. The area to the right of the dry steam line is known as superheated region.

4.1.13 Temperature – Enthalpy (T-h) diagram
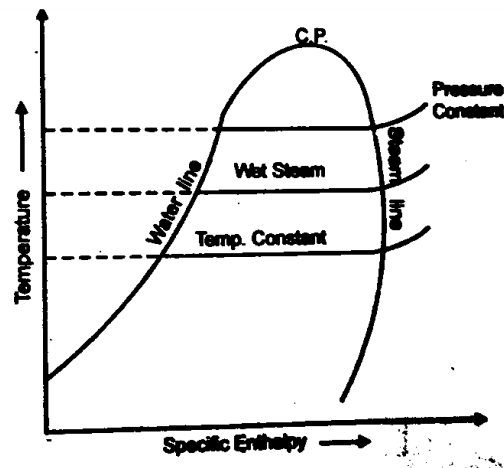
Fig 4.3

➤ In this diagram, the absolute temperature plotted in y axis and the specific enthalpy is plotted in x-axis .

➤ The curve on the left marked as water line shows the relationship between the temperature and specific enthalpy of water

➤ The curve on the right marked as dry steam line shows the relationship between the temperature and the specific enthalpy of dry steam.

➤ During evaporation, the temperature and pressure lines are one and the same. In the superheated region, the pressure lines are shown by curved lines. The water line and dry steam line meet at a point known as critical point. At the critical point the water is changed into steam directly.

4.1.14 Temperature – Entropy (T-s) diagram

If we plot the saturation temperature of water and steam corresponding to the various absolute pressures against the entropies at those saturation temperatures we get a graph. It is called temperature entropy diagram which is of great use in solving numerical problems on steam as it considerably reduces the time and labor in making calculations.
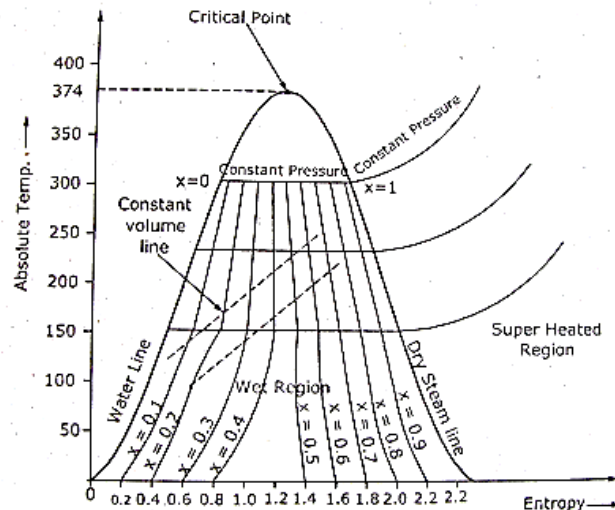

Fig 4.4

The value of the various quantities can be read directly from the diagram. In the figure the curve on the left marked *Water line* shows the relationship between the entropy and temperature before the steam is formed. The "Dry steam line" which is also called "Steam boundary curve" shows the relationship between the entropy and temperature when all the water has been converted into steam. The area in between these two lines or curves is called the "Wet Region" The area to the right of the dry steam lines is known as the "Region of Superheat" which contains lines of constant pressure; four of which are shown in the figure. The lines of constant dryness (for x = 0.1, 0.2, 0.3, etc., upto 0.9) have also been shown. The lines of constant volume and lines of constant total heat can also be drawn but are not shown in the figure in order to avoid congestion. Corresponding to a given temperature the horizontal distance between two points on the two curves gives the change in entropy, during evaporation, $\left(i.e. \dfrac{hfg}{T}\right)$. It can be noted from the figure that the water line and steam line are converging with the increase in temperature. In fact these are parts of the same curve. The dotted curve joining the two curves shown in the figure is simply a smooth conjectural curve which forms continuation of the two curves and drawn in such a manner as to be tangent to the line t = 374.15℃.This temperature at which the distinction between the entropy of water and that of dry steam disappears is called the "critical temperature". The pressure (221.2 bar)

corresponding to this temperature is known as critical pressure. This point is very unstable and is unaffected by the presence of gases in the water. Since the horizontal distance between the two curves at the critical temperature vanishes, the latent heat absorbed is zero. Hence at this temperature the water is converted into dry steam without the absorption of any latent heat.
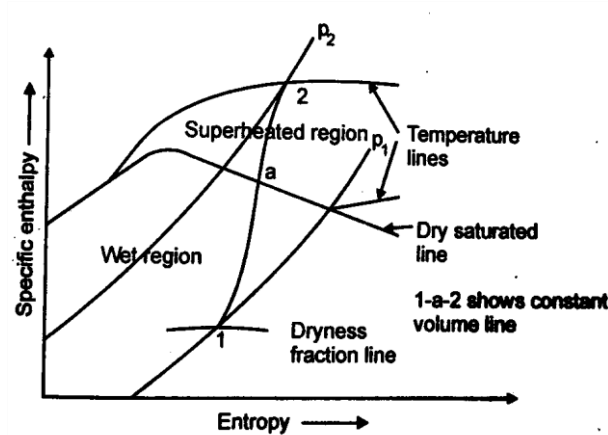
4.1.15 Enthalpy – Entropy (h-s) diagram

Fig 4.5

In the mollier diagram which is also very commonly used by engineers, the vertical represents the total heat while the base represents the entropy. A mollier diagram for steam is shown in figure in which the region above and below the saturation line represent the superheated and wet conditions of steam respectively. The lines of constant dryness fraction is shown in the wet steam region while the lines of constant temperature in the region of superheat. It should be noted that the lines of constant pressure are straight in the wet steam region but curved in the region of superheat. In order to facilitate taking readings, horizontal as well as vertical lines are drawn at small intervals. An adiabatic process will be represented by a vertical line on the Mollier diagram while expansion during which the total heat remains constant will be represented by a horizontal line. One of the outstanding advantages of this diagram is that the drop in the total heat during an adiabatic expansion can be directly read from it.

4.1.16 Pressure – Enthalpy (p-h) diagram
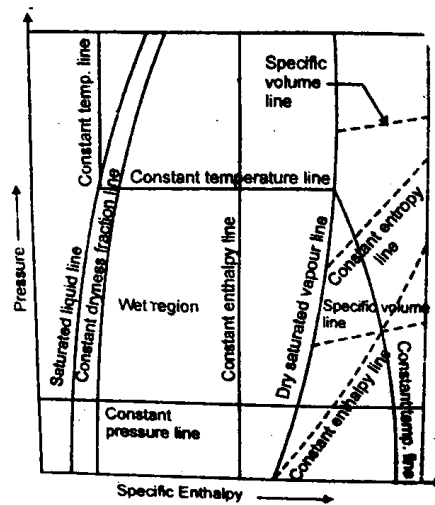
Fig 4.6

➢ This diagram is used an refrigeration work. In a p-h diagram the pressure is represented in y-axis and the specific enthalpy, represented in x-axis.

➢ The diagram is divided into three regions namely 1. Liquid region, 2. Wet region and 3. Superheated region.

➢ The lines of constant dryness fraction, pressure, entropy, temperature and specific enthalpy have also been shown.

4.1.17 Critical conditions

➢ From T – h diagram shown in figure, when the pressure and saturation temperature increases, the latent heat of evaporation decreases and it becomes zero at a point C where water and dry seam lines meet.

➢ The point C is called as the critical point where liquid and vapour phases merge.

Fig 4.7

> The temperature corresponding to critical point is ***known as critical temperature*** and the pressure is ***known as critical pressure***.

For steam,

> The critical temperature is 374 .15 °C
> The pressure 212.2 bar.

4.1.18 Specific volume of steam:

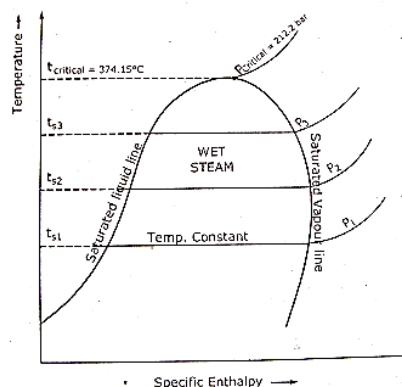It is the volume occupied by the steam per unit mass at a given temperature and pressure, and is expressed in $\frac{m^3}{kg}$. It is the reciprocal of density of steam in $\frac{kg}{m^3}$. It may be noted that the value of specific volume decreases with the increase in pressure.

4.1.19 Specific volume of Wet steam:-

Consider 1 kg of wet steam of dryness fraction x. We know that this steam will have x of dry steam and (1- x) kg of water. Let v_f be the volume of 1 kg of water, then Volume of 1 kg of wet steam

$$= x \, v_g + (1 - \, V_f)$$

Since v_f is very small as compared to v_g, therefore the expression $(1 - \, V_f)$ may be neglected.

Therefore, Volume of 1 kg of wet steam $=$ $x \, v_g \, m^3$

or specific volume of wet steam, $\qquad V_{wet} = \qquad x \, v_g \, m^3/kg$.

4.1.20 Specific volume of Dry steam:-

We know that in case of dry steam, the mass of water in suspension is zero and dryness fraction is unity. Therefore specific volume of dry steam,

$$V = V_g \, m^3/kg$$

4.1.21 Specific volume of Superheated steam:-

when the dry steam is further heated under a constant pressure, there is an increase in volume with the rise in temperature. The superheated steam behaves more or less like a perfect gas. Therefore, according to Charle's law,

$$\frac{vsup}{Tsup} = \frac{vg}{T} \; or \; vsup = \frac{(vgTsuP)}{T}$$

Where, $\quad v_{sup} \quad = \quad$ Specific volume of superheated steam,

$\qquad V_g \quad = \quad$ Specific volume of dry steam at the pressure

$\qquad$ of steam formation.

$\qquad T_{sup} \quad = \quad$ Absolute temperature of superheated steam.

$\qquad T \qquad = \quad$ Absolute saturation temperature at the

$\qquad$ pressure of steam formation.

4.1.22 Density of steam

The mass of *one m³ of steam at given pressure and temperature* is known as density of steam.

Density of wet steam ρ_{wet} $= \dfrac{1}{Vwet}$ $= \dfrac{1}{x.Vg}$ kg / m³

Density of dry steam ρ_{dry} $= \dfrac{1}{Vdry}$ $= \dfrac{1}{Vg}$ kg / m³

Density of superheated steam ρ_{sup} $= \dfrac{1}{Vsup}$ $= \dfrac{Ts}{Tsup .Vg}$ kg / m³

4.1.23 External work of during evaporation

We have already discussed that whenever water at boiling temperature is heated at a constant pressure, it gets converted into steam after absorbing the latent heat. This latent heat is utilized in the in the following two ways:

1.   In overcoming the internal molecule resistance of water changing it's the state from the saturated water to dry saturated steam,

2.   In overcoming the external resistance to the movement of the piston due to increase in volume during expression.

The first effect is called internal work or internal latent heat, as the change takes place with in the body of the steam, and represents the energy stored in the steam. The second effect is called external work of evaporation, and represents the energy which has been taken out of the steam.

Let   p = Pressure on the piston in bar = $p \times 10^5 \, N/m^2$,

v_f = Volume of water in m³ at pressure p, and

v_g = Volume of steam in m³ at pressure p.

We know that work done during evaporation,

W = Pressure × Change in volume

$= p \times 10^5 \, (v_g - v_f) \, J = 100 \, p \, (v_g - v_f)$ KJ

At low pressures, the volume of the water (V_f) is very small, as compared to the volume of steam (V_g). Therefore neglecting the value of V_f in the above equation,

W $= 100 \, p \, v_g$ KJ

If the steam is completely dry, and has dryness fraction of x, then work done,

W $= 100 \, p \, x \, v_g$ KJ

If the steam is superheated, then work done,

W $= 100 \, p \, v_{sup}$ KJ

4.1.24 Internal latent heat (or) Internal work of evaporation

The heat supplied during evaporation converts water into seam by overcoming intermolecular resistance of water. This energy is stored in steam. This stored energy is called internal latent heat or internal work of evaporation.

Internal work of evaporation = latent heat of evaporation - External work of evaporation

Internal work of evaporation for wet steam = $x \cdot h_{fg} - E.W_{wet}$

$= x \cdot h_{fg} - 100 \, p \, v_{wet}$ kJ/kg

Internal work of evaporation for dry steam = $h_{fg} - E.W_{dry}$

$= h_{fg} - 100 \, p \, v_{g}$ kJ/kg

4.1.25 Inernal energy of steam (u)

It is the actual heat energy stored in steam, above the freezing point of water. The internal energy may be calculated by subtracting the external work done during evaporation from the enthalpy or total heat of steam. Mathematically,

Internal energy of steam = Enthalpy – External work during evaporation

The expressions of internal energy per kg of steam (u) for wet steam, dry saturated steam and superheated steam are given below:

(a) For wet steam

$u = h - 100 \, p \, x \, v_{g} = h_{f} + x \, h_{fg} - 100 \, p \, x \, v_{g}$ KJ/kg K.

(b) For dry saturated steam

$u = h_{g} - 100 \, p \, v_{g} = h_{f} + h_{fg} - 100 \, p \, v_{g}$ KJ/kg K.

(c) For superheated steam,

$u = h_{sup} - 100 \, p \, v_{sup} = [h_{g} + c_{p} \, (t_{sup} - t_{S})] - 100 \, p \, v_{sup}$ KJ/kg K.

4.1.26 Entropy of steam

Entropy is an imaginary thermal property that increses with addition of heat and decreses with the removal of heat.

It is denoted as s. If *dQ be a small amount of heat added* at an *absolute temperature T* and if ds be the small increase in enropy, then

$ds = \dfrac{dQ}{T}$. KJ/kg.k

dQ = Heat added or removed

T = Absolute temperature

4.1.27 Specific entropy of water (s_f)

Consider 1 kg of water being heated at a constant pressure from freezing temperature (i.e. 0°C or 273 K) to the boiling temperature. Now consider an instant, when the absolute temperature of water is T K.

Let, for a small rise in temperature of dT, the heat absorbed by 1 kg of water is δq. Then

$$\delta q = \text{Mass} \times \text{Sp. Heat of water} \times \text{Rise in temperature.}$$

$$= 1 \times c_w \times dT = c_w \, dT$$

Where, c_w = Specific heat of water.

The value of c_w may be taken as 4.2 KJ/Kg K.

We know that increase in entropy for rise in temperature dT,

$$ds = \delta q/T = \frac{c_w dT}{T}$$

The total increase in entropy of water from freezing point to boiling point, may be obtained by integrating the above expression within the limits 273 K and T K.

$$\int_0^s ds = \int_{273}^T \frac{c_w dT}{T}$$

$$s_f = c_w \log_e \left[\frac{T}{273}\right]$$

4.1.28 Specific entropy of evaporation (s_{fg})

Specific entropy of evaporation is the change in entropy during evaporation of one kg of water into dry saturated steam at constant pressure. It is denoted as s_{fg}.

Heat added during evaporation = h_{fg}

Since the temperature is constant at T_s during evaporation

$$\therefore s_{fg} = h_{fg} / T_s$$

4.1.29 Specific entropy of wet steam (s_{wet})

The entropy of wet and dry steam, above the freezing point of water, is the entropy of water plus the entropy during evaporation. Mathematically, increase in entropy

$$= s_f + \frac{x \, h_{fg}}{T} = s_f + x \, s_{fg} \qquad \text{.....(For wet steam)}$$

4.1.30 Specific entropy of dry saturated steam (s_{dry})

$$= s_f + \frac{h_{fg}}{T} = s_f + s_{fg} = s_g \qquad \text{.....(For dry steam)}$$

4.1.31 Specific entropy of super heated steam (s_{sup})

We know that during superheating, the heat is supplied at a constant pressure, and the temperature of dry steam (T) increases to the temperature of superheated steam (T_{sup}). For a small rise in temperature dT, the heat absorbed.

$$\delta q = cp \; dT \; per \; kg \; of \; steam$$

$$\frac{\delta q}{T} = \frac{cpdT}{T}$$

$$\therefore \quad ds = \frac{cpdT}{T} \qquad\qquad ....\frac{\delta q}{T} = ds$$

Where, c_p = Specific heat of superheated steam at constant pressure. Its value may be taken between 1.67 KJ/Kg K to 2.5 KJ/Kg K.

The total increase in entropy from T K to T_{sup} K may be obtained by integrating the above expression between the proper limits.

$$\int_{sg}^{ssup} ds = \; cp \int_{T}^{Tsup} \frac{dT}{T}$$

or

$$s_{sup} - s_g = c_p \; \log_e \left[\frac{Tsup}{T}\right]$$

where ($s_{sup} - s_g$) is the increase in entropy.

4.1.32 Steam tables

The properties of dry saturated steam like its temperature of formation (saturation tempearature), sensible heat, latent heat of vaporisation, enthalpy or total heat, specific volume, entropy etc,. vary with pressure and can be found with experiments only. These propertis have been carefully determined, and made available in a tabular form known as steam tables. It may be noted, that there is a slight difference in figures, quoted by various authors, and even in their various editions of a book. But in this book we shall use the standard properties which are internationally recognised.

There are two important steam tables, one in terms of absolute pressure and other in terms of temperature. An extract of the properties of dry saturated steam is shown in the following two tables:

Absolute pressure (p) in bar	Temperature(t) in ℃	Specific Volume in $\frac{m^3}{kg}$		Specific enthalpy in KJ/Kg			Specific Entropy in KJ/ Kg K		
		Water (vf)	Steam (vg)	Water	Evaporation (h$_{fg}$)	Steam (h$_g$)	Water (s$_f$)	Evaporation(s$_{fg}$)	Steam (s$_g$)
0.010	6.983	0.001000	129.21	29.3	2485.1	2514.4	0.106	8.871	8.977
0.015	13.04	0.001001	87.982	54.7	2470.8	2525.5	0.196	8.634	8.830
0.2	60.09	0.001017	7.649	251.5	2358.4	2609.9	0.832	7.077	7.909

Table 4.8 (a)

Temperature(t) in ℃	Absolute pressure (p) in bar	Specific Volume in $\frac{m^3}{kg}$		Specific enthalpy in KJ/Kg			Specific Entropy in KJ/ Kg K		
		Water (vf)	Steam	Water	Evaporation (h$_{fg}$)	Steam	Water	Evaporation(s$_{fg}$)	Steam (s$_g$)
0	0.00611	0.001000	206.31	0.0	2501.6	2501.6	0.000	9.158	9.158
5	0.00872	0.001000	147.16	21.0	2489.7	2510.7	0.076	8.951	9.027
10	0.01227	0.001000	106.43	42.0	2477.9	2519.9	0.151	8.751	8.902

Table 4.8 (b)

4.1.33 Mollier Chart

It is also known as enthalpy – entropy chart. In the mollier chart, enthalpy is the ordinate and entropy is the abscissa. From the steam tables, both enthalpy and entropy of water and dry saturated seam, for any pressure are obtained and on plotting these enthalpes against entropies,
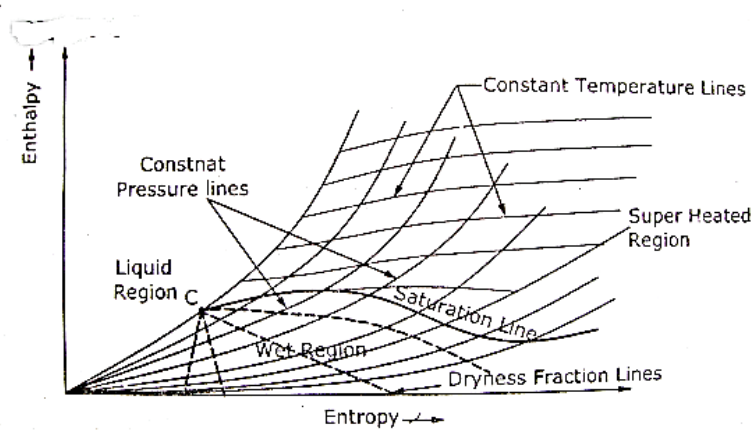
Fig 4.9

Liquid line and the dry saturated line are obtained mollier for steam is shown in figure 4.9. In which the regions above and below the saturation line represents the super heated and wet conditions of steam respectively.the lines constant dryness fraction are shown in the wet region. The lines of constant temperature are shown in the super heated region. The lines of constant pressure are straing in the wet region and curved in the super heated region. If any region, constant enthalpy (throttling) process will be represented by a horizontal straight line. The reversible adiabatic process or isentropic process will be represented by a vertical line. One of the most outstanding advantages of the diagram is that enthalpy drop during an adiabatic expansion can be directly read from it.

4.2 Expansion processes of steam

The various methods of expansion of steam are:

1. Constant volume process (Isochoric process)

2. Constant pressure process (Isobaric process)

3. Constant temperature process (Isothermal process)

4. Hyperbolic process

5. Adiabatic process (Isentropic process)

6. Polytropic process

7. Throttling process

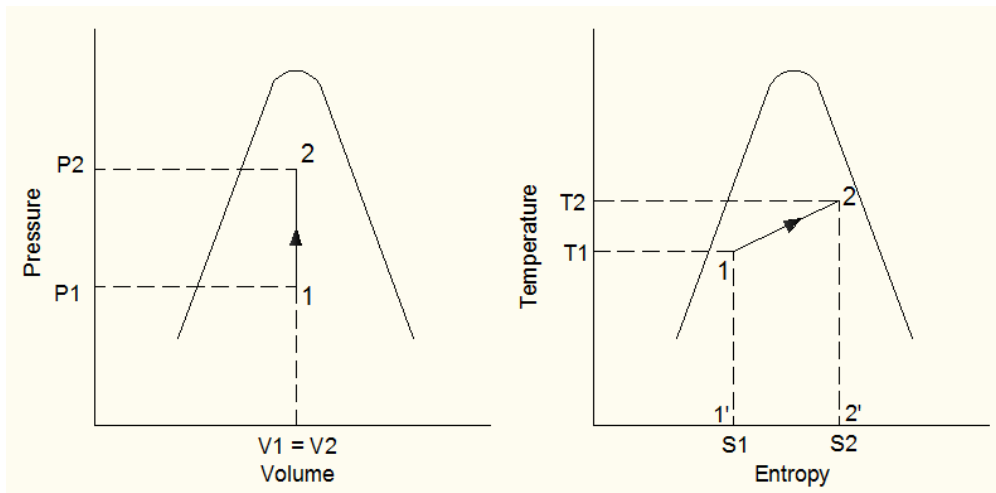## 4.2.1 Constant volume process (Isochoric process)

Fig 4.10 (a)   Fig 4.10 (b)

The heating or cooling of a steam in a closed vessel is an example for constant volume process. In this process, the volume or mass of the steam before and after the process is constant. It may be noted that, in this process, no work is done.

Now consider 1kg of wet steam being heated at constant volume from an initial state 1 to the final state 2.

Let, p_1 = Initial pressure of the wet steam in bar,

V_{g1} = Specific volume of the dry saturated steam in m³/kg corresponding to initial pressure P_1 (from steam tables)

x_1 = Initial dryness fraction of the wet steam, and

p_2, V_{g2}, x_2 = Corresponding values for final condition of steam.

1.   *Condition of steam:*

We know that initial volume of 1 kg of wet steam,

$$V_1 = x_1 \, V_{g1}$$

Final volume of 1kg of superheated steam,

$$V_2 = \frac{Tsup}{Ts} \text{ x } V_{g2}$$

$$V_1 = V_2$$

$$x_1 \, V_{g1} = \frac{Tsup}{Ts} \text{ x } V_{g2}$$

2.   *Workdone during the process:*

We know that workdone during the process,

$W_{1\text{-}2}$ = Pressure x Change in volume

$W_{1\text{-}2}$ = p. dV

$W_{1\text{-}2}$ = p . (V_2- V_1) kJ/kg

Since there is no chance in volume, therefore

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

3.   *Change in internal energy of steam:*

We know that initial internal energy of steam,

Initial internal energy of steam,

$$U_1 = h_1 - 100P_1V_1$$

Final internal energy of steam,

$$U_2 = h_2 - 100P_2V_2$$

$$du = U_2 - U_1 \text{ kJ/kg}$$

4.   *Heat absorbed or heat transferred:*

According to First law of thermodynamics,

$$dQ = dU + W$$

wkt, $W = 0$

$$dQ = dU \text{ kJ/kg}$$

Thus we find that during a constant volume heating process, the heat absobed or heat transferred by the steam is equal to the change in internal energy of steam.

4.2.2 Constant Pressure process (Isobaric process)
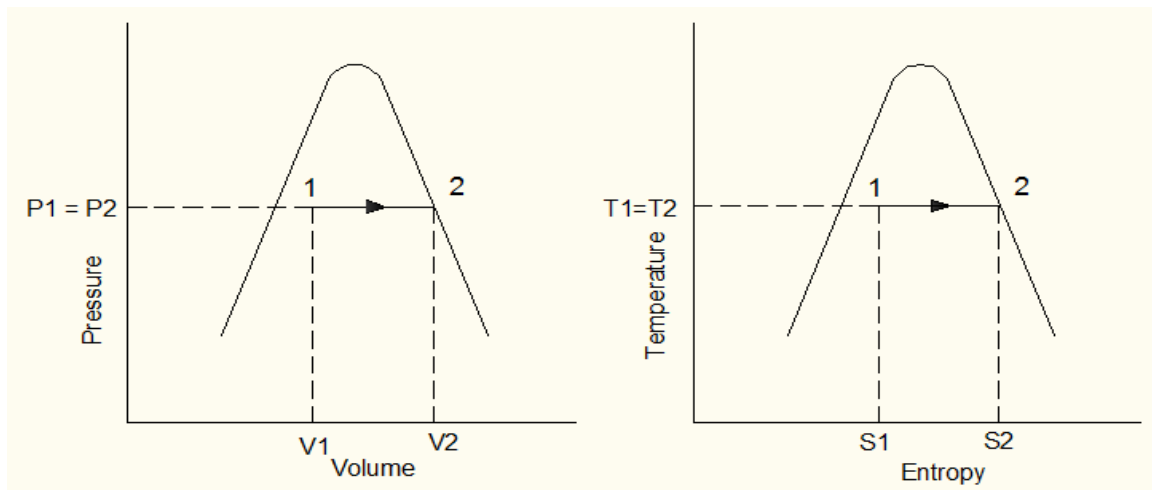

Fig 4.11 (a)   Fig 4.11 (b)

The generation of steam in steam boilers is an example of constant pressure. In this process, the pressure of steam before after the process is constant.

Now consider 1kg of wet steam being heated at constant pressure from an initial state 1 to the final state 2.

p = Constant perssure

x_1 = Initial dryness fraction of steam

x_2 = Final dryness fraction of steam

1.　*Condition of steam:*

We know that initial volume of 1 kg of wet steam,

$$V_1 = x_1 V_{g1}$$

Final volume of 1kg of wet steam,

$$V_2 = x_2 V_{g2}$$

$$V_1 = V_2$$

$$x_1 V_{g1} = x_2 V_{g2}$$

2.　*Workdone during the process:*

We know that workdone during the process,

$$W_{1-2} = \text{Pressure x Change in volume}$$

$$W_{1-2} = 100 \, p \, . \, dV$$

$$W_{1-2} = 100 \, p \, . \, (V_2 - V_1) \, kJ/kg$$

3.　*Change in internal energy of steam:*

We know that initial internal energy of steam,

Initial internal energy of steam,

$$U_1 = h_1 - 100 \, p_1 V_1$$

Final internal energy of steam,

$$U_2 = h_2 - 100 \, p_2 V_2$$

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

4.　*Heat absorbed or heat transferred:*

According to First law of thermodynamics,

$$dQ = dU + W$$

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

$$dQ = (h_2 - h_1) \, kJ/kg$$

Thus we find that during a constant pressure heating process, the heat absobed or heat transferred by the steam is equal to the change in enthalpy of steam.

4.2.3 Constant Temperature process (Isothermal process)
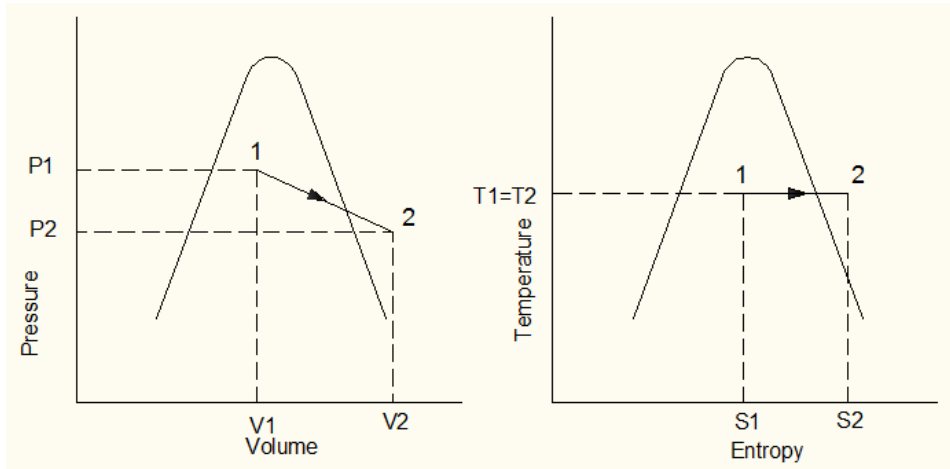

Fig 4.12 (a)   Fig 4.12 (b)

The heating of wet steam at constant temperature till it becomes dry saturated is similar to that of heating at a constant pressure. Thus for an thus the isothermal process is the same expressions for the workdone, change in internal energy and heat absorbed.

Now consider 1kg of wet steam being heated at constant temperature from an initial state 1 to the final state 2.

1.   *Condition of steam:*

We know that initial volume of 1 kg of wet steam,

$$V_1 = x_1 \, V_{g1}$$

Final volume of 1kg of superheated steam,

$$V_2 = \frac{Tsup}{Ts} \times V_{g2}$$

2.   *Workdone during the process:*

i)For non flow energy process

According to First law of thermodynamics,

$$dQ = dU + W$$

$$W = dQ - dU$$

$$W = T\,(S_2\text{-}S_1) - [\,(h_2\text{-}h_1)\text{-}100(p_2V_2\text{-}\,p_1V_1)\,]$$

ii)For steady flow energy process

$$dQ = dh + W$$

$$W = dQ - dh$$

$$W = T\,(S_2\text{-}S_1) - (h_2\text{-}h_1) \text{ kJ/kg}$$

3. *Change in internal energy of steam:*

We know that initial internal energy of steam,

Initial internal energy of steam,

$$U_1 = h_1 - 100\ p_1 V_1$$

Final internal energy of steam,

$$U_2 = h_2 - 100\ p_2 V_2$$

$$dU = U_2 - U_1\ \text{kJ/kg}$$

$$dU = (h_2 - h_1) - 100\ (p_2 V_2 - p_1 V_1)$$

4. *Heat absorbed or heat transferred:*

$$dS = \frac{dQ}{T}$$

$$dQ = T.dS$$

$$dQ = T.\ (S_2 - S_1)\ \text{kJ/kg}$$

4.2.4 Hyperbolic process

Fig 4.13 (a)   Fig 4.13 (b)

Now consider 1kg of wet steam being heated hyperbolically from an initial state 1 to the final state 2.

Let, p_1 = Initial pressure of the wet steam in bar,

V_{g1} = Specific volume of the dry saturated steam in m^3/kg corresponding to initial pressure p_1 (from steam tables)

x_1 = Initial dryness fraction of the wet steam, and

P_2, V_{g2}, x_2 = Corresponding values for final condition of steam.

$$p_1 V_1 = C$$

$$p_2 V_2 = C$$

1.   *Condition of steam:*

We know that initial volume of 1 kg of wet steam,

$$V_1 = x_1 \, V_{g1}$$

Final volume of 1kg of wet steam,

$$V_2 = x_2 \, V_{g2}$$

$$p_1 V_1 = p_2 V_2$$

$$p_1 \, x_1 V_{g1} = p_2 \, x_2 V_{g2}$$

2.   *Workdone during the process:*

Wkt, $p \, V = C$

$$W = 100 \, p_1 V_1 \, \log_e \left(\frac{V2}{V1}\right)$$

$$W = 100 \, p_1 V_1 \, \log_e \left(\frac{P1}{P2}\right) \text{kJ/kg}$$

3.   *Change in internal energy of steam:*

We know that initial internal energy of steam, Initial internal energy of steam,

$$U_1 = h_1 \text{-} 100 \, p_1 V_1$$

Final internal energy of steam,

$$U_2 = h_2 \text{-} 100 \, p_2 V_2$$

$$dU = U_2 - U_1 \text{ kJ/kg}$$

$$dU = (h_2 \text{-} h_1) - 100 \, (p_2 V_2 - p_1 V_1)$$

$$dU = (h_2 \text{-} h_1) \text{ kJ/kg}$$

4.   *Heat absorbed or heat transferred:*

According to First law of thermodynamics,

$$dQ = dU + W$$

$$dQ = (h_2 \text{-} h_1) + 100 \, p_1 V_1 \, \log_e \left(\frac{V2}{V1}\right)$$

4.2.5 Adiabatic process (Isentropic process)
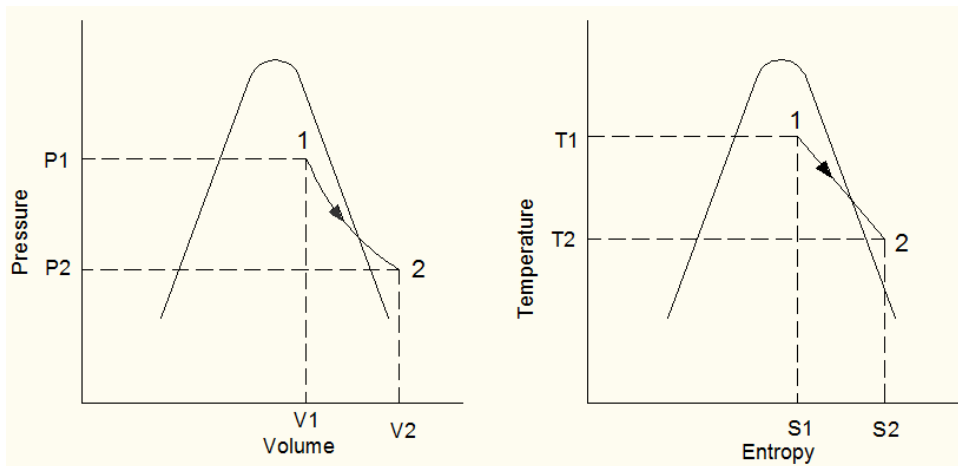

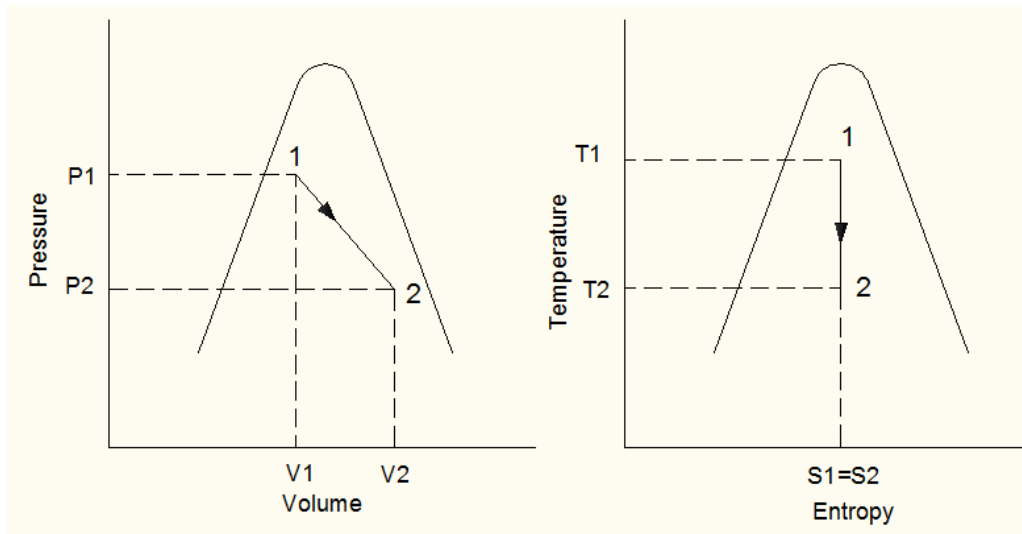Fig 4.14 (a)   Fig 4.14 (b)

Now consider 1kg of wet steam being heated Isentropically from an initial state 1 to the final state 2.

Let, p_1 = Initial pressure of the wet steam in bar,

V_{g1} = Specific volume of the dry saturated steam in m^3/kg corresponding to initial pressure p_1 (from steam tables)

x_1 = Initial dryness fraction of the wet steam, and

p_2, V_{g2}, x_2 = Corresponding values for final condition of steam.

1.   *Condition of steam:*

We know that initial volume of 1 kg of wet steam,

$$V_1 = x_1 \, V_{g1}$$

Final volume of 1kg of dry steam,

$$V_2 = V_{g2}$$

2.   *Workdone during the process:*

i) For non flow energy process

According to First law of thermodynamics,

$$dQ = dU + W$$

$$0 = (U_2 - U_1) + W$$

$$W = U_2 - U_1$$

ii)For steady flow energy process

$$dQ = dh + W$$

$$0 = (h_2 - h_1) + W$$

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$$W = h_2 - h_1$$

3.   Change in internal energy of steam:

We know that initial internal energy of steam,

Initial internal energy of steam,

$$U_1 = h_1 - 100\, p_1 V_1$$

Final internal energy of steam,

$$U_2 = h_2 - 100\, p_2 V_2$$

$$dU = U_2 - U_1 \text{ kJ/kg}$$

$$dU = (h_2 - h_1) - 100\,(p_2 V_2 - p_1 V_1)$$

4.   Heat absorbed or heat transferred:

Since there is no heat is added or rejected during an adiabatic process, Therefore,

$$dQ = 0$$

4.2.6 Polytropic process
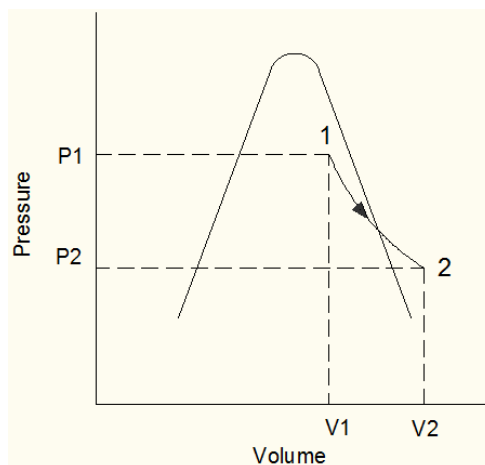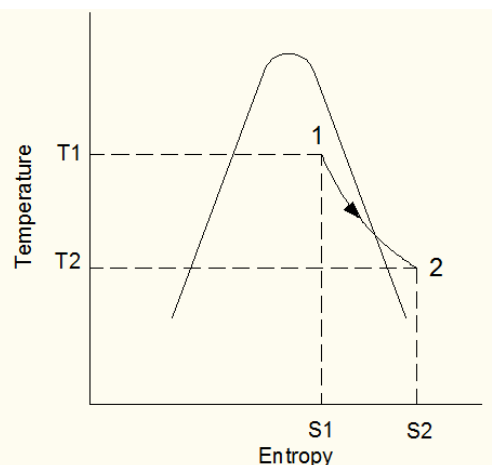
Fig 4.15 (a)   Fig 4.15 (b)

Now consider 1kg of wet steam being heated Polytropically from an initial state 1 to the final state 2.

Let, p_1 = Initial pressure of the wet steam in bar,

V_{g1} = Specific volume of the dry saturated steam in m^3/kg corresponding to initial pressure P_1 (from steam tables)

x_1 = Initial dryness fraction of the wet steam, and

p_2, V_{g2}, x_2 = Corresponding values for final condition of steam.

n = Polytropic index

1. *Condition of steam:*

We know that initial volume of 1 kg of wet steam,

$$V_1 = x_1 \, V_{g1}$$

Final volume of 1kg of dry steam,

$$V_2 = V_{g2}$$

2. *Workdone during the process:*

Wkt, $W = 100 \, p \, (dV)$

$W = 100 \, p \, (V_2 - V_1)$

$W = \dfrac{100 \, P_1 V_1 - 100 \, P_2 V_2}{n-1}$

3. *Change in internal energy of steam:*

We know that initial internal energy of steam,

Initial internal energy of steam,

$$U_1 = h_1 - 100 \, p_1 V_1$$

Final internal energy of steam,

$$U_2 = h_2 - 100 \, p_2 V_2$$

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

$dU = (h_2 - h_1) - 100 \, (p_2 V_2 - p_1 V_1)$

4. *Heat absorbed or heat transferred:*

According to First law of thermodynamics,

$dQ = dU + W$

$dQ = (h_2 - h_1) - 100 \, (p_2 V_2 - p_1 V_1) + \left[\dfrac{100 p_1 V_1 - 100 p_2 V_2}{n-1}\right]$

4.2.7 Throttling process

Steam is said to be throttled when it passes through a restricted opening such as a narrow aperture or a slightly opened valve. The leakage of fluid through a crack in the vessel is an example of throttling.

It may be noted that during a throttling process:

      i)   No heat is supplied or rejected,

      ii)   No work is done,

      iii)   No change in internal energy,

iv)   The enthalpy or total heat of the fluid remains constant.

Now consider 1kg of wet steam being throttled through a narrow aperture from an initial state 1 to the final state 2.

1.   Condition of steam:

We know that initial enthalpy of 1 kg of wet steam,

$$h_1 = h_{wet}$$

$$h_1 = h_f + \boldsymbol{x}\, h_{fg}$$

Final enthalpy of 1kg of dry steam,

$$h_2 = h_{sup}$$

$$h_2 = \frac{Tsup}{Ts} \times V_g$$

2.   Workdone = 0

3.   Change in internal energy = 0

4.   Heat absorbed or heat transferred = 0

3.1 Steam Calorimeters

Steam calorimeters are used to measure the dryness fraction of wet steam. The types of calorimeter used for the determination of dryness fraction are:

1. Bucket calorimeter (barrel calorimeter)
2. Separating calorimeter
3. Throttling calorimeter
4. Combined separating and throttling calorimeter

3.2 Bucket Calorimeter

➢ Bucket calorimeters are used to measure the dryness fraction of wet steam.

➢ It consists of a copper calorimeter placed on wooden block and provided with a wooden cover.

➢ The water equivalent of the copper calorimeter is stamped on it.

➢ It contains quality of cold water and is surrounded by an outer vessel thus providing an air space in between, which acts as an insulator.

➢ A thermometer which is inserted through one of the holes in the wooden cover is provided to measure the temperature.

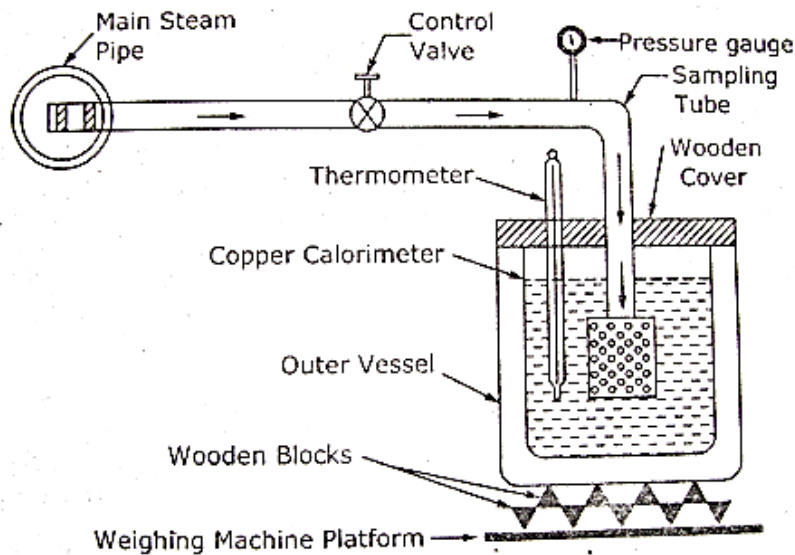➢ The whole assembly is placed on the platform of weighing machine.

Fig 4.16

The steam from the main steam pipe enters through the sampling tube via the control valve and flows into the cold water through fine exit holes of the ring provide as shown.

The steam gets condensed and the temperature of cold after rises.

The quantity of steam condensed can be known from the difference of readings of the weighing machine before and after the condensation of steam.

The temperature of water is also taken before and after blowing the steam. The pressure at which steam enters the calorimeter is indicated by the pressure gauge.

Let

p	-	Pressure of steam in the main seam pipe
t_s	-	Saturation temperature corresponding to the pressure p.
x	-	Dryness fraction of steam in the main steam pipe
h_{fg}	-	Enthalpy of evaporation (latent heat) of steam
m	-	Mass of steam condensed (difference in mass before and after the sampled steam supplied to the vessel)
M_1	-	Mass of cold water in the vessel before steam supply
m_c	-	Mass of calorimeter
Cp_c	-	Specific heat of copper calorimeter
M_2	-	Water equivalent of calorimeter $\quad = \dfrac{mc\,Cpc}{Cpw}$
M	-	Mass of cold water + water equivalent of calorimeter $\quad = M1 + M2$
t_1	-	Initial temperature of water at commencement
t_2	-	Final temperature of water after steam has condensed

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C_{pw} - Specific heat of water.

From law of conservation of energy

Heat lost by the wet steam = heat gained by the calorimeter and water

$$m\,[c_{pw}\,(t_s-t_2) + x\,h_{fg}] = m_c\,C_{pc}\,(t_2-t_1) + M_1\,C_{pw}\,(t_2-t_1)$$

$$= M_2\,C_{pw}\,(t_2-t_1) + M_1\,C_{pw}\,(t_2-t_1) \qquad (.:m_cC_{pc}) = M_2C_{pw})$$

$$= (M_1+M_2)\,C_{pw}\,(t_2-t_1)$$

$$= M\,C_{pw}\,(t_2-t_1)$$

$$C_{pw}\,(t_s-t_2) + x\,h_{fg} = \frac{M}{m}\,C_{pw}\,(t_2-t_1)$$

$$x\,h_{fg} = \frac{M}{m}\,C_{pw}\,(t_2-t_1) - C_{pw}\,(t_s - t_2)$$

$$.: x = \frac{\dfrac{M}{m}\,c_{pw}\,(t_2- t_1) - c_{pw}\,(t_s - t_2)}{h_{fg}}$$

3.3 Combined separating and throttling calorimeter
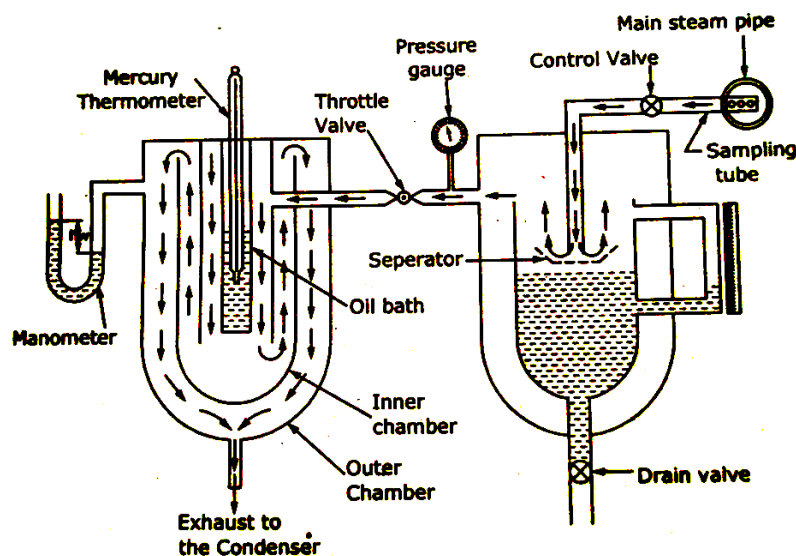
Fig 4.17

A very successful method of measuring the dryness fraction of steam is by a combined separating and throttling calorimeter as shown in figure.

In this calorimeter, the wet steam is first passed through a separating calorimeter. A part of water is removed by the separating calorimeter owing to quick change of direction of flow. The resulting semi dry steam is then passed through the throttling calorimeter where super heating takes place without change of enthalpy. The temperature and pressure of steam after

throtting are measured by using a thermometer and pressure gauge respectively. The calorimeter is well insulated to prevent any loss.

p_1 -   Pressure of wet steam

x -   Dryness fraction of the set steam

h_{f1} -   Specific enthalpy of water at pressure p1

h_{fg1} -   Specific enthalpy of evaporation at pressure p1

x_1 -   Dryness fraction of steam considering separating calorimeter or apparent dryness fraction of steam

x_2 -   Dryness fraction of steam entering the throttling calorimeter

M   -   Mass of steam passed from the separating calorimeter to throttling calorimeter

m   -   Mass of water particles collected in the separating calorimeter

p_2 -   pressure of steam after throttling

t_{sup2} -   Temperature of super heated steam

t_{s2} -   Saturation temperature of steam pressure p2

C_{ps} -   specific heat of super heated steam.

In the separating calorimeter

The apparent of dryness fraction of steam $x_1 = \dfrac{M}{M + m}$

In the throttling calorimeter

Enthalpy of steam entering the throttling calorimeter = Enthalpy of steam leaving the calorimeter.

$$h_{f1} + x_2\, h_{fg1} = h_{g2} + C_{ps}\,(t_{sup2} - t_{s2})$$

$$\therefore x_2 = \frac{h_{g2} + C_{ps}\,(t_{sup\,2} - t_{s2}) - h_{f1}}{h_{fg1}}$$

Then $x = x_1 \cdot x_2$

Problems

1. Steam enters an engine at a pressure of 12 bar with a 67°C of superheat. It is exhausted at a pressure of 0.15 bar and 0.95 dry. Find the drop in enthalpy of the steam.

Given:

$p1 = 12$ bar;

Degree of super heat ($t_{sup} - ts$) = 67°C;

$p2 = 0.15$ bar;

$x = 0.95.$°C .

To find: The drop in enthalpy of the steam

Solution:

From steam tables, corresponding to a pressure of 12 bar, we find that

$$h_f = 798.4 \text{ KJ/Kg} ; \quad h_{fg} = 1984.3 \text{ KJ/kg}$$

We know that enthalpy or total heat of 1 kg of superheated steam,

$$h_{sup} = h_f + h_{fg} + c_p \, (t_{sup} - t_s) = 798.4 + 1984.3 + 2 \times 67$$

$$= 2916.7 \text{ KJ/Kg} \qquad \text{….. (Taking } c_p = 2 \text{ KJ/Kg K)}$$

Similarly, from steam tables, corresponding to a pressure of 0.15 bar, we find that

$$h_f = 226 \text{ KJ/Kg}; \quad h_{fg} = 2373.2 \text{ KJ/Kg}$$

We know that enthalpy or total heat of 1 kg of wet steam,

$$h = h_f + x \, h_{fg} = 226 + 0.95 \times 1984.3 = 2111 \text{ KJ/Kg}$$

Therefore, Drop in Enthalpy of steam,

$$h_{sup} - h = 2916.7 - 2111 = 805.7 \text{ KJ/Kg}.$$

2. Determine the volume of 1 kg of superheated steam at a pressure of 20 bar and a temperature of 300°C .

Given:

$p = 20$ bar;

$T_{sup} = 300°C = 300 + 273 = 573$ K;

To find: The volume of 1 kg of superheated steam.

Solution:

Let V_{sup} = Volume of 1 kg of superheated steam.

From steam tables, corresponding to a pressure of 20 bar, we find that

$$T = 212.4°C \text{ and } V_g = 0.1\frac{m^3}{kg}$$

$$T = 212.4 + 273 = 485.4 \text{ K}$$

We know that the superheated steam obeys Charle's law

$$\frac{Vg}{T} = \frac{vsup}{Tsup}$$

$$V_{sup} = \frac{Vg}{T} \times Tsup = \frac{0.1}{485.4} \times 573 = 0.118m^3$$

3. Calculate the internal energy of 1 kg of steam at a pressure of 10 bar when the steam is (a) 0.9 dry; and (b) dry saturated. The volume of water may be neglected.

Given:

p = 10 bar;

x = 0.9.

To find: The internal energy of steam.

Solution:

From steam tables, corresponding to a pressure of 10 bar, we find that

h_f = 762.6 KJ/kg; h_{fg} = 2013.6 KJ/kg and V_g = 0.1943 $\frac{m^3}{kg}$

(a) Internal energy of 1 kg of steam when it is 0.9 dry (i.e $x = 0.9$)

We know that internal energy for wet steam

$$u = h_f + x\ h_{fg} - 100\ p\ x\ V_g$$

$$= 762.6 + 0.9 \times 2013.6 - 100 \times 10 \times 0.9 \times 0.1943 \text{ KJ}$$

$$= 2574.8 - 174.8 = 2400 \text{ KJ.}$$

(b) Internal energy of 1 kg of steam when it is dry saturated steam,

$$u = h_f + x\ h_{fg} - 100\ p\ V_g$$

$$= 762.6 + 2013.6 - 100 \times 10 \times 0.1943$$

$$= 2776.2 - 194.3 = 2581.9 \text{ KJ.}$$

4. Find the external work done during evaporation per kg of steam at a pressure of 15 bar when the steam is (a) 90% dry and (b) dry saturated.

Given:

   p = 15 bar;

To find: The external work done.

Solution:

(a) Work done when the steam is 90% dry (i.e. $x = 0.9$)

   We know that work done,

   $$w = 100 \text{ p } x \text{ V}_\text{g} = 100 \times 15 \times 0.9 \times 0.1317 = 177.8 \text{ KJ/kg}$$

(b) Workdone when the steam is dry saturated,

   $$w = 100 \text{ p V}_\text{g} = 100 \times 15 \times 0.1317 = 197.5 \text{ KJ/kg.}$$

5. Calculate the enthalpy of 1 kg of steam at a pressure of 8 bar and dryness fraction of 0.8. How much heat would be required to raise 2 kg of this steam from water at 20℃

Given:

   p = 8 bar;

   $x = 0.8$.

Solution:

Enthalpy of one kg of steam

   From steam tables, corresponding to a pressure of 8 bar, we find that

   $$h_\text{f} = 720.9 \text{ KJ/kg and } h_\text{fg} = 2046.5 \text{ KJ/kg}$$

   We know that enthalpy of 1 kg of steam,

   $$h = h_\text{f} + x \text{ } h_\text{fg} = 720.9 + 0.8 \times 2046.5 = 2358.1 \text{ KJ}$$

Heat required to raise 2 kg of this steam from water at 20℃

We have calculated above the enthalpy or total heat required to raise 1 kg of steam from water at 0°C. Since the water, in this case, is already at 20°C, therefore

Heat already in water $= 4.2 \times 20 = 84$ KJ

Heat required per kg of steam

$= 2358.1 - 84 = 2274.1$ KJ

and heat required for 2 kg of steam

$= 2 \times 2274.1 = 4548.2$ KJ.

6. Determine the entropy per kg of superheated steam at a pressure of 20 bar and a temperature of 250°C. Assume c_p for superheated steam as 2.2 KJ/kg K.

Given:

p = 20 bar;

$T_{sup} = 250°C = 250 + 273 = 523$ K;

$c_p = 2.2$ KJ/Kg K.

To find: The entropy per kg of superheated steam.

Solution:

From steam tables, corresponding to a pressure of 20 bar, we find that

T = 212.4°C = 212.4 + 273 = 485.4 K; $\qquad$ $s_g = 6.337$ KJ/Kg K

We know that entropy per kg of steam,

$$s_{sup} = s_g + c_p \log_e \left(\frac{Tsup}{T}\right) = 6.337 + 2.2 \log_e \left(\frac{523}{485.4}\right)$$

$$= 6.5 \text{ KJ/Kg K.}$$

7. Find the entropy of 1 kg of dry saturated steam at a pressure 5.2 bar. The boiling point of water at this pressure is given as 152.6°C and its total heat at this temperature is 2110 KJ/Kg K.

Given:

p = 5.3 bar;

T = 152.6°C = 152.6 + 273 = 425.6 K;

h_{fg} = 2110 KJ/Kg K

To find: The entropy of 1 kg of dry saturated steam.

Solution:

We know that entropy of 1 kg of dry saturated steam,

From steam tables, corresponding to a pressure of 5.2 bar

$$s_g = s_f + s_{fg}$$

$$= 6.82 \text{ KJ/Kg K.}$$

8. Calculate the entropy of 1 kg of wet steam with dryness fraction of 0.9 at a pressure of 8.4 bar.

Given:

x = 0.9;

p = 8.4 bar.

To find: The entropy of 1 kg of wet steam.

Solution:

From steam tables, corresponding to a pressure of 8.4 bar, we find that

s_f = 2.066 KJ/ Kg K; s_{fg} = 4.577 KJ/ Kg K.

We know that entropy of 1 kg of wet steam,

$$s = s_f + x \ s_{fg} = 2.066 + 0.9 \times 4.577 = 2.066 + 4.12 = 6.186 \text{ KJ/Kg.}$$

9. A closed vessel of 0.2 m^3 contains steam at a pressure of 10 bar and a temperature of 250° C. The vessel is cooled till the pressure of steam in the vessel falls to 3.5 bar. Find the final temperature, final dryness fraction, and change in internal energy, heat transferred and change in entropy.

Given: V_1= 0.2 m^3; P_1 = 10bar; T_{sup} = 250°c = 250+273=523K; p_2 = 3.5bar

To find: T_2, dU, dQ, x_2, dS

Solution: From steam table at pressure 10 bar

T_1 = 179.9^{o}C = 179.9+273 = 452.9 K, V_{g1} = 0.1943 m^3/kg

At pressure 3.5 bar

V_{g2} = 0.524 m^3/kg, h_{f2} = 584.3 kJ/kg, h_{fg2} = 2147.3 kJ/kg

S_{f2} = 1.727 kJ/kgK, S_{fg2} = 5.212 kJ/kgK

Final temperature of steam:

Let, T_2 = Final temperature of steam

$$\frac{Vsup}{Tsup} = \frac{Vg1}{T1}$$

V_{sup} = V_{g1} X $\frac{Tsup}{T1}$ = 0.1943 X $\frac{523}{452.9}$ = 0.2244 m^3/kg

$$m = \frac{Total\ volume\ of\ steam\ in\ the\ vessel}{Specific\ volume\ of\ steam} = \frac{V1}{Vsup}$$

$$= \frac{0.2}{0.2244} = 0.89\ kg$$

Final volume V_2 = m V_{g2} = 0.89 X 0.524 = 0.466 m^3

From steam table pressure 3.5 bar T_2 = 138.9^{o}C

Final dryness fraction of steam:

V_{sup} = $x_2 V_{g2}$

$X_2 = \frac{Vsup}{Vg2} = \frac{0.2244}{0.524} = 0.43$

x_2 = 0.43

Change in internal energy:

From steam table pressure 10 bar & 250°C

$h_{sup} = $ 2943 kJ/kg

$U_1 = h_{sup} - 100 \, p_1 V_{sup}$

= 2943 – 100 X 10 X 0.2244 = 2718.6 kJ/kg

$U_2 = h_2 - 100 \, p_2 V_2$

= $h_{f2} + x_2 \, h_{fg2} - 100 \, P_2 \, x_2 V_{g2}$

= 584.3 + 0.43 X 2147.3 – 100 X 3.5 X 0.43 X 0.524

= 1507.6 – 79 = 1428.6 kJ/kg

change in internal energy $dU = U_2 - U_1 = 1428.6 - 2718.6 = -1290$ kJ/kg

Total change in internal energy = m $(U_2 - U_1) = 0.89$ X (-1290)

$$dU = -1148 \text{ kJ}$$

'-'sign indicates decrease in internal energy.

Heat transferred

$Q_{1-2} = $ m $(U_2 - U_1) = -1148$ kJ

'-'sign indicates heat is rejected from the steam.

Change in entropy

From steam table pressure 10 bar & 250°C

Initial entropy of SH steam $S_1 = 6.926$ kJ/kg K

Final entropy of wet steam, $S_2 = S_{f2} + x_2 S_{fg2}$

$$= 1.727 + 0.43 \text{ X } 5.212 = 3.97 \text{ kJ/kg K}$$

$$dS = S_2 - S_1 = 3.97 - 6.926 = -2.956 \text{ kJ/kg K}$$

Total dS = m $(S_2 - S_1) = 0.89$ x (-2.956) = -2.63 kJ/K

'-'sign indicates decrease in entropy.

10.One kg of steam at pressure of 17.5 bar and dryness 0.95 is heated at a constant pressure, until it is completely dry. Determine Increase in volume, Quantity of heat supplied, Change in entropy.

Given: P= 17.5 bar; $x_1 = 0.95$; $x_2 = 1$

To find: dV, dS, dQ

Solution:

From steam table pressure 17.5 bar

$h_{f1} = h_{f2} = 878.2$ kJ/kg; $h_{fg1} = h_{fg2} = 1915.9$ kJ/kg; $V_g = 0.1134$ m³/kg

$S_{f1} = S_{f2} = 2.384$ kJ/kg K $S_{fg1} = S_{fg2} = 4.001$ kJ/kg K

Increase in volume

$V_1 = x_1 V_g = 0.95$ X $0.1134 = 0.1077$ m³

$V_2 = V_g = 0.1134$ m³/kg

Increase in volume = $V_2 - V_1 = 0.1134 - 0.1077 = 0.0057$ m³ Ans.

Quantity of heat supplied

$h_1 = h_{f1} + x_1 h_{fg1} = 878.2 + 0.95$ x $1915.9 = 2698.3$ kJ/kg

$h_2 = h_{f2} + x_2 h_{fg2} = 878.2 + 1915.9 = 2794.1$ kJ/kg

Since the heat is supplied at constant pressure, therefore

Heat supplied = change in enthalpy of steam

= $h_2 - h_1 = 2794.1 - 2698.3 = 95.8$ kJ/kg Ans.

Change in entropy

$S_1 = S_{f1} + x_1 S_{fg1} = 2.384 + 0.95$ x $4.001 = 6.185$ kJ/kg K

$S_2 = S_{f2} + x_2 S_{fg2} = 2.384 + 4.001 = 6.385$ kJ/kg K

dS = $S_2 - S_1 = 6.385 - 6.185 = 0.2$ kJ/kg K Ans.

11.Steam at a pressure of 5.4 bar and 0.8 dry expands in a cylinder reversibly and isothermally to a pressure of 1 bar. Find, Final condition of steam, Change in internal energy, change in entropy, heat transferred and work done per kg of steam.

Given: P_1 = 5.4 bar x_1 = 0.8 bar p_2 = 1bar

To find: condition of steam, dU, dS, dQ, W

Solution:

From steam table at pressure 5.4 bar

$\quad\quad$ T = 154.8^{o}C = 154.8+273= 427.8 K; h_{f1} = 652.8 kJ/kg; h_{fg1} = 2098 kJ/kg;

$\quad\quad V_{g1}$ = 0.348 m^3/kg; S_{f1} = 1.89 kJ/kg; S_{fg1} = 4.903 kJ/kg K

$\quad\quad\quad\quad$ At pressure 5.4 bar $\quad$ h_{f2} = 417.5 kJ/kg; h_{fg2} = 2258 kJ/kg K

Final condition of steam

From steam table at pressure 5.4 bar, the saturation temperature, T = 154.8^{o}C

Change in internal energy

$\quad\quad$ U_1 = h_1 − 100 p_1V_1 = h_{f1} + x_1 h_{fg1} − 100 p_1 x_1 V_{g1}

$\quad\quad\quad$ = 652.8 + 0.8 x 2098 − 100 x 5.4 x 0.8 x 0.348 kJ/kg

$\quad\quad\quad$ = 2331.3 − 150.3 = 2181 kJ/kg

$\quad\quad$ U_2 = h_2 − 100 p_2V_2 = h_{sup} − 100 p_2V_{sup}

$\quad\quad$ U_2 = 2785.8 − 100 x 1 x 1.96 = 2589.8 kJ/kg

$\quad\quad$ dU = U_2 - U_1 = 2589.8 − 2181 = 408.8 kJ/kg

Change in entropy

S_1 = S_{f1} + x_1 S_{fg1} = 1.89 + 0.8 x 4.903 = 5.81 kJ/kg K

$\quad\quad$ Final entropy of steam is from steam tables at pressure 1 bar & 154.8^{o}C

$\quad\quad$ S_2 = 7.635 kJ/kg K

$\quad\quad$ dS = S_2 - S_1 = 7.635 − 5.81 = 1.825 kJ/kg K Ans.

Heat Transferred

$$Q_{1-2} = T \, dS = 427.8 \times 1.825 = 781 \text{ kJ/kg K}$$

Work done per kg of steam

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

$$= 781 - 408.8 = 372.2 \text{ kJ/kg}$$

12.Steam at pressure of 10 bar and 0.9 dry expand to atmospheric pressure hyperbolically. Find Work done; Change in internal energy; Change in enthalpy; Heat absorbed, Specific heat of steam at constant pressure = 2 kJ/kg K

Given: $p_1 = 10$ bar $x_1 = 0.9$ bar; $p_2 = 1.013$bar; $C_p = 2$ kJ/kg K

To find: condition of steam, dU, dh, dQ, W

Solution:

From steam table at pressure 10 bar $h_{f1} = 762.6$ kJ/kg; $h_{fg1} = 2013.6$ kJ/kg;

$$V_{g1} = 0.1943 \text{ m}^3/\text{kg} \quad T_1 = 179.9°C = 179.9 + 273 = 452.9 \text{ K}$$

From steam table at pressure 1.013 bar $h_{f2} = 419$ kJ/kg; $h_{fg2} = 2257$ kJ/kg;

$$V_{g2} = 1.673 \text{ m}^3/\text{kg} \quad T_2 = 100°C = 100 + 273 = 373 \text{ K}$$

Work done

$$W_{1-2} = 100 \, p_1 V_1 \log_e\left(\frac{V2}{V1}\right) = 100 \times 10 \times 0.1943 \times \log_e\left(\frac{1.673}{0.1943}\right)$$

$$W_{1-2} = 418.32 \text{ kJ/kg}$$

Change in enthalpy

$$p_1 V_1 = p_2 V_2$$

$$p_1 \, x_1 V_{g1} = p_2 \, x_2 V_{g2}$$

$$x_2 = \frac{P1x \, 1Vg1}{P2Vg2} = \frac{10 \times 0.9 \times 0.1943}{1.013 \times 1.673} = 1.032$$

$p_1 \, x_1 V_{g1} = p_2 \, V_{sup}$

$$V_{sup} = \frac{P1x \, 1Vg1}{P2} = \frac{10 \times 0.9 \times 0.1943}{1.013} = 1.726 \text{ m}^3/\text{kg}$$

$$\frac{Vsup}{Tsup} = \frac{Vg2}{T2} = T_{sup} = 373 \text{ x } \frac{1.726}{1.673} = 385 \text{ K}$$

$$h_1 = h_{f1} + x_1 h_{fg1} = 762.6 + 0.9 \text{ x } 2013.6 = 2574.8 \text{ kJ/kg}$$

$$h_2 = h_{sup} = h_{f2} + h_{fg2} + C_p (T_{sup} - T_2)$$

$$= 419 + 2257 + 2(385\text{-}373) = 2700 \text{ kJ/kg}$$

$$dH = h_2 - h_1 = 2700 - 2574.8 = 125.2 \text{ kJ/kg}$$

Change in internal energy

$$dU = dh = 125.2 \text{ kJ/kg}$$

Heat absorbed

$$Q_{1\text{-}2} = dU + W_{1\text{-}2} = 125.2 + 400 = 525.2 \text{ kJ/kg}$$

13.Steam from an initial pressure of 7 bar and 200°C is expanded isentropically to a pressure of 1 bar. Calculation the final condition of steam, change in internal energy, work done, the value of n, if the expansion follows the law $PV^n = C$.

Given: P_1 = 7 bar; T_{sup} = 200°C = 200+273= 473 K; p_2 = 1 bar

To find: Final condition, dU, W, n

Solution:

From steam table at pressure 7 bar & 200°C

$$S_{sup} = 6.886 \text{ kJ/kg K}; h_{sup} = 2844 \text{ kJ/kg}; V_{sup} = 0.3 \text{ m}^3/\text{kg}$$

From steam table at pressure 1 bar

$$S_{f2} = 1.303 \text{ kJ/kg}; S_{fg2} = 6.057 \text{ kJ/kg K}; h_{f2} = 417.5 \text{ kJ/kg};$$

$$h_{fg2} = 2258 \text{ kJ/kg}; V_{g2} = 1.694 \text{ m}^3/\text{kg}$$

Final condition of steam

$$x_2 = \text{Final dryness fraction of steam}$$

$$S_{sup} = S_2 = S_{f2} + x_2 S_{fg2}$$

$$6.886 = 1.303 + x_2 \text{ x } 6.057$$

$$x_2 = 0.92$$

Change in internal energy

$$U_1 = h_1 - 100p_1V_1 = h_{sup} - 100\ p_1V_{sup}$$

$$= 2844 - 100 \times 7 \times 0.3 = 2634\ kJ/kg$$

$$U_2 = h_2 - 100\ p_2V_2 = h_{f2} + x_2\ h_{fg2} - 100\ p_2\ x_2V_{g2}$$

$$= 417.5 + 0.92 \times 2258 - 100 \times 1 \times 0.92 \times 1.694$$

$$= 2495 - 156 = 2339\ kJ/kg$$

Change in internal energy, $dU = U_2 - U_1 = 2339 - 2634 = -295\ kJ/kg$

'-'Sign indicates decrease in internal energy.

Work done during the process

$$W_{1-2} = -dU = -(-295) = 295\ kJ/kg$$

Value of n

$$p_1V_1^n = p_2V_2^n\ or\ p_1\ (V_{sup})^n = p_2\ (x_2V_{g2})^n$$

Taking log on both sides,

$$\log p_1 + n \log (V_{sup}) = Log\ p_2 + n \log (x_2V_{g2})$$

$$n\ [\ \log(V_{sup}) - \log (x_2V_{g2})] = \log p_2 - \log p_1$$

$$n = \frac{\log p2 - \log p1}{\log(Vsup) - \log(x\ 2Vg2)} = \frac{\log(1\times10^5) - \log(7\times10^5)}{\log(0.3) - \log(0.92\times1.694)}$$

$$= \frac{5 - 5.845}{-0.523 - 0.1927} = \frac{-0.845}{-0.7157} = 1.18$$

14.Steam of dryness fraction 0.96 is expanded in a cylinder according to $PV^{1.02}$ = Constant. The pressure at the beginning of expansion is 20 bar, and is continued till the pressure reduces to 1 bar. Determine the final condition of steam, change in internal energy, work done, heat exchange that occurs between the steam and cylinder walls per kg.

Given: $x_1 = 0.96$; n = 1.02; $P_1 = 20$ bar; $P_2 = 1$ bar

To find: Final condition, dU, W, dQ

Solution:

From steam table at pressure 20 bar $h_{f1} = 908.6$ kJ/kg; $h_{fg1} = 1888.7$ kJ/kg;

$$V_{g1} = 0.0995 \text{ m}^3/\text{kg}$$

From steam table at pressure 1 bar $h_{f1} = 417.5$ kJ/kg; $h_{fg1} = 2258$ kJ/kg;

$V_{g1} = 1.694$ m³/kg; $T_2 = 99.6°C = 99.6 + 273 = 372.6$ K

Final condition of steam

$x_2 =$ Final dryness fraction of steam

$p_1 V_1^n = p_2 V_2^n$ or $p_1 (X_1 V_{g1})^n = p_2 (X_2 V_{g2})^n$

$$x_2 = \frac{X1Vg1}{Vg2} \left(\frac{P1}{P2}\right)^{\frac{1}{n}} = \frac{0.96 \times 0.0995}{1.694} \left[\frac{20}{1}\right]^{\frac{1}{1.02}} = 1.062$$

We know that $p_1 V_1^n = p_2 V_2^n$ or $p_1 (x_1 V_{g1})^n = p_2 (V_{sup})^n$

$$V_{sup} = x_1 V_{g1} \left(\frac{p1}{p2}\right)^{\frac{1}{n}} = 0.96 \text{ X } 0.095 \left[\frac{20}{1}\right]^{\frac{1}{1.02}} = 1.8 \text{ m}^3/\text{kg}$$

$$\frac{Vsup}{Tsup} = \frac{Vg2}{T2} \qquad T_2 = 372.6 \text{ X } \frac{1.8}{1.694} = 396 \text{ K}$$

Work done during the process

$$W_{1-2} = \frac{100 \ (P1V1 - p2V2)}{n-1} = \frac{100 \ (p1 \ x \ 1Vg1 - p2Vsup)}{n-1}$$

$$= \frac{100 \ (20 \times 0.96 \times 0.0995 - 1 \times 1.8)}{1.02 - 1} = 552 \text{ kJ/kg}$$

Change in internal energy

$U_1 = h_1 - 100 \ p_1 V_1 = h_{f1} + x_1 h_{fg1} - 100 \ p_1 X_1 V_{g1}$

$= 908.6 + 0.96 \text{ X } 1888.7 - 100 \text{ X } 20 \text{ X } 0.96 \text{ X } 0.0995$

$= 2721.8 - 191 = 2530.8$ kJ/kg

$U_2 = h_2 - 100 \ p_2 V_2 = h_{sup} - 100 \ p_2 V_{sup}$

$= [h_{f2} + h_{fg2} + C_p (T_{sup} - T_2)] - 100 \ P_2 V_{sup}$

$= [417.5 + 2258 + 2(396 - 372.6)] - 100 \text{ X } 1 \text{ X} 1.8$

$= 2722.3 - 180 = 2542.3$ kJ/kg

$dU = U_2-U_1 = 2542.3 - 2530 = 12.3$ kJ/kg

Heat exchange between the steam and cylinder walls

$Q_{1-2} = dU + W_{1-2} = 12.3 + 552 = 564.3$ kJ/kg

15.Steam at 10bar and 0.9 dry is throttled to a pressure of 2 bar. Using steam tables only, evaluate the final dryness fraction or degree of superheat. Estimate change of entropy during this process.

Given: $x_1 = 0.9$; $p_1 = 10$ bar; $p_2 = 2$ bar

To find: Final dryness fraction, dS

Solution:

From steam table at pressure 10 bar $h_{f1} = 762.6$ kJ/kg; $h_{fg1} = 2013.6$ kJ/kg;

$S_{f1} = 2.138$ kJ/kg; $S_{fg1} = 4.445$ kJ/kg K

From steam table at pressure 2 bar $h_{f2} = 504.7$ kJ/kg; $h_{fg2} = 2201.6$ kJ/kg;

$S_{f2} = 1.53$ kJ/kg; $S_{fg2} = 5.597$ kJ/kg K

We know that throttling process,

$h_{f1} + x_1 h_{fg1} = h_{f2} + x_2 h_{fg2}$

$762.6 + 0.9 \times 2013.6 = 504.7 + x_2 \times 2201.6$

$x_2 = 0.94$

Change in entropy

$S_1 = S_{f1} + x_1 S_{fg1} = 2.138 + 0.9 \times 4.445 = 6.14$ kJ/kg K

$S_2 = S_{f2} + x_2 S_{fg2} = 1.53 + 0.94 \times 5.597 = 6.79$ kJ/kg K

$dS = S_2 - S_1 = 6.79 - 6.14 = 0.65$ kJ/kg K

UNIT V - STEAM BOILES AND PERFORMANCE OF BOILERS

5.1 Steam Boiler:

➢ Steam boiler is *equipment which generates steam by transferring heat produced of fuel to water.* Heat energy released by burning of fuel is transferred to water in the boiler drum. This converts water into steam as the required pressure and temperature

➢ Boiler (or) steam generator is a high quality steel made closed vessel

➢ Steam is generated by heating the water.

➢ When the fuel is burnt in the boiler chemical energy of fuel is converted into heat energy which is used to convert the water into steam

5.1.1 Classification Of Boilers:

Boilers may be classified as

1) According to the flow of water and hot gases.

  a) Fire tube boiler b) water tube boiler

2) According to the position of furnace

    a) Internal fired boiler b) Externally Fired boiler

3) According the fuel used

  a) Coal fire boiler b) Oil fire boiler

4) According to the method of circulation of water and steam

    a) Natural circulation boiler b) Forced circulation boiler

5) According to the use or service

    a) Stationary boiler b) Mobile boiler.

6) According to the position of shell

    a) Vertical boiler b) Horizontal boiler

7) According the number of tubes

  a) Single tube boiler b) Multi tube boiler

8) According the pressure develop

    a) Low pressure boiler b) High pressure boiler

Fire Tube Boiler:

➢ In fire tube boiler the *fuel hot gases pass through tubes that are surrounded by water*

➢ The water is heated up and convert into steam

➢ The exhaust gas are sent to atmosphere to chimney

➢ *Examples: Cornish boiler, locomotive boiler, Lancashire boiler, vertical boiler, Cochran boiler*
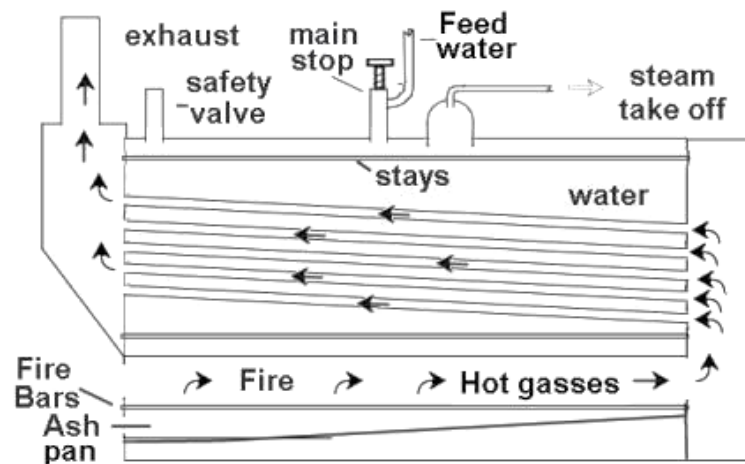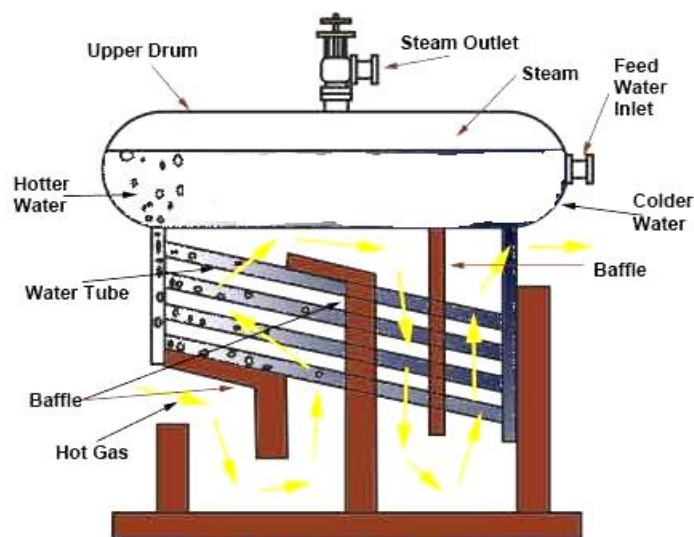

Fig 5.1

Water Tube Boiler:

Fig 5.2

➢ In water tube boilers, *water is circulated through number of tubes* and hot flue gases flow over these tubes

➢ The water is converted into steam

➢ *Examples are: Babcock and Wilcox boiler, Lamont boiler, velox boiler, loffer boiler and BHEL boiler*

Position Of Boiler Or Method Of Firing

 A) *Internally fired boilers:*
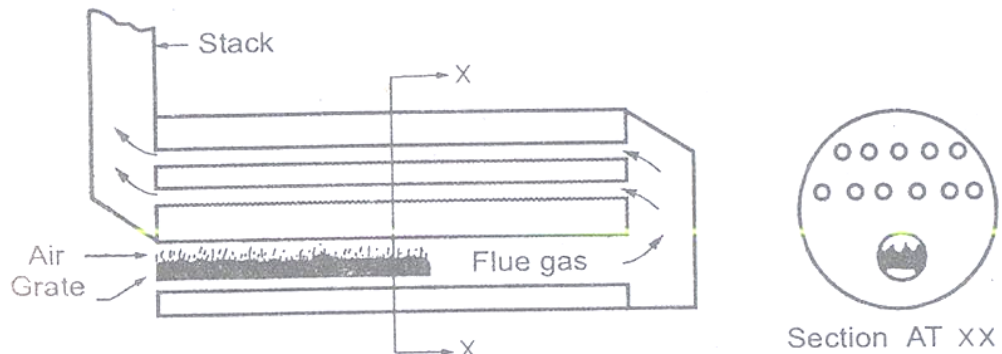
Fig 5.3

➢ The *furnace grate is providing inside the boiler shell.*

➢ Examples; Lancashire boilers, locomotive boilers and scotch boilers.

B) *Externally fired boilers:*
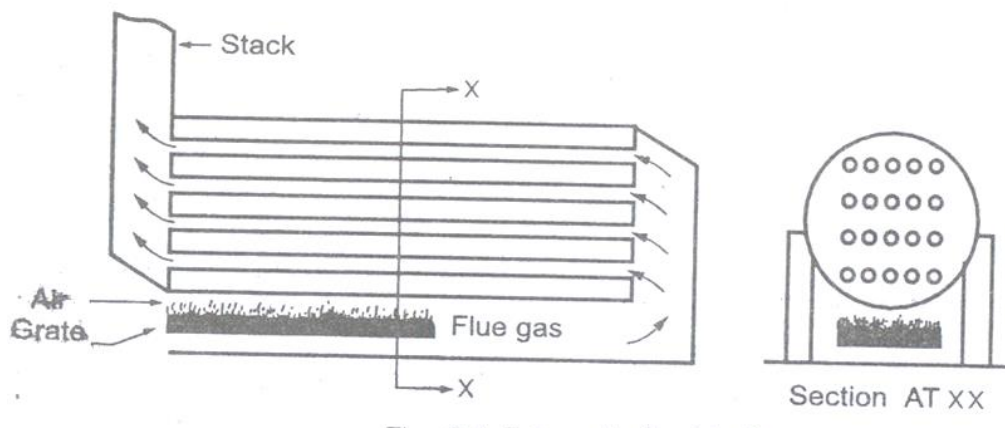
Fig 5.4

➢ In externally fired boilers the furnace gate is *provided outside.*

Method of circulation

i) Natural circulation:

➢ The water is circulated by the *natural convection* currents caused by the application of the heat.

➢ Examples: Lancashire boilers, Locomotive boilers, Babcock and Wilcox boiler.

ii) Forced circulation

➢ The water is circulated by *mass of a pump*.

➢ Examples: Lamont boiler, velox boiler etc.

Pressure developed

i) Low pressure boilers.

➢ Low pressure boilers produce steam at a pressure *lower than* the 80 bar

➢ Examples are: Cochran boiler, Cornish boiler, Lancashire boilers, and locomotive boilers.

ii) High pressure boilers.

➢ High pressure boilers produce *steam at a pressure greater than 80 bar.*

➢ Examples**: Lamont boiler,** Benson boilers, loffer boiler and **BHEL BOILERS**.

Use or service

i) Stationary boilers:

➢ The boilers are which are *stationary* are called as stationary boilers

➢ Examples: boilers used in power plants, chemical and textile industries.

ii) Mobile boilers:

➢ The boilers are which are *not stationary* or moving are called as mobile boilers.

➢ Examples: locomotive boiler, marine boilers.

Nature of draught

i) Nature draught;

➢ The fuel in the furnace burns with *natural circulation* of air.

ii) Forced draught:

➢ The fuel in the furnace burns by means of *air forced* by a fan.

5.1.2 Advantages of High Pressure Boilers

1. Scale formation is eliminated since water is circulated with high velocity through the tubes.

2. High pressure and high temperature steam can be obtained. This increases efficiency of the plant.

3. Efficiency of the plant is increased up to 42%

4. It has less number of or complete elimination of drums.

5. Less floor space is reduced.

6. Less foundation cost due to lighter weight of tubes.

7. Gas and air leakages are reduced.

8. Increased rate of production

9. Rapid start from cold is possible

10. Steam can be raised quickly to meet variable load requirements.

5.1.3 Comparison of Fire Tube And Water Tube Boilers

Fire Tube Boilers	Water Tube Boilers
Hot flue gas passes through tubes which are immersed in water	Water is circulated through tubes that are surrounded by hot flue gas
The operation pressure is limited to 25 bar	The operation pressure is more than 80 bar
Pressure range is limited to 80 bar	This can be generated steam at a pressure of 250 bar
Rigid and simple construction	Construction is complicated
Less skill is efficient required for efficient and economic working	More skill is efficient required for efficient and economic working
Not suitable for large power plants	Suitable for large power plants
Ordinary water is required	Treated water can be used
Moe floor area is required for the given power	Less floor area is required for a given power
Raising of steam is slow	Raising of steam is more rapid
In case of explosion it cause heavy damage	No serious explosion occurs even if one tube fails
Chance of bursting is less	Chance of bursting is more.

High Pressure Boiler

> When the operation pressure of boiler is high is called high pressure boiler.
> Steam generating units that generates steam of high pressure and temperature of about 80 bar.

High pressure boilers are usually referred for this application

> Lamont boiler - 130 bar at 540^0 C (45 – 50 tonnes / hour)
> BHEL boiler - 180 bar at 600 ^{0}C (130 tonnes / hour)

5.1.4 Lamont boiler:

This is a modern high pressure water tube steam boiler working on a forced circulation. The circulation is maintained by a centrifugal pump, driven by a steam turbine, using steam from the boiler. The forced circulation causes the feed water to circulate through the water walls and drums equal to ten times the mass of steam evaporated. This prevents the tubes from being overheated.
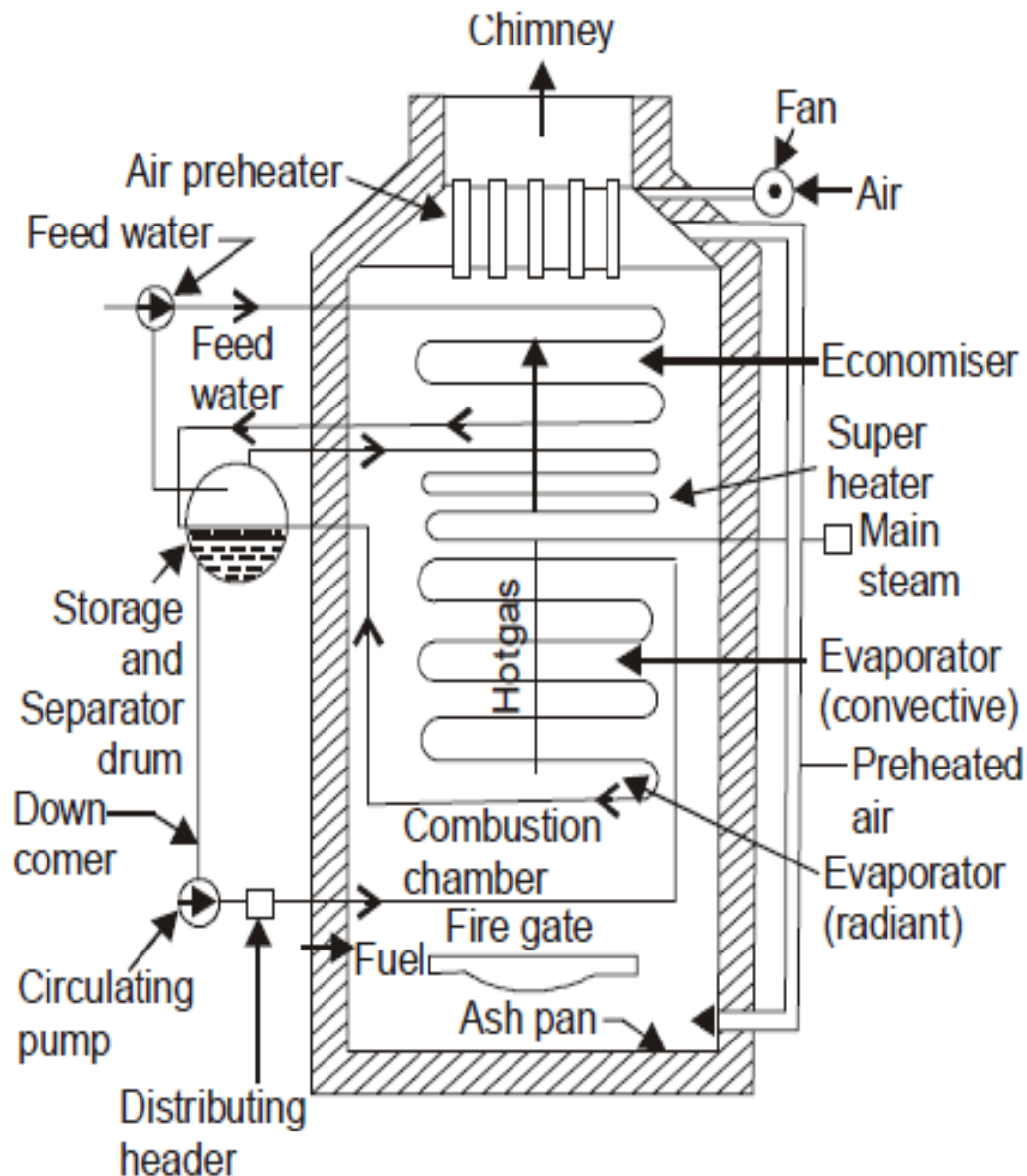
Fig 5.5

A diagrammatic sketch of La-Mont steam boiler is shown In Fig. The feed water passes through the economizer to an Evaporating drum. It is then drawn to the circulating pump through The tube. The pump delivers the feed to the headers, at a pressure above the drum pressure. The header distributes water through nozzles into the generating tubes acting in parallel. The water and steam from these tubes passes into the drum. The steam in the drum is then drawn through the superheater.

Advantages

1. Light in weight
2. Less number of drums are required
3. Quick starting in even cold conditions
4. High evaporation rate
5. Absence of scaling trouble in the pipes.

Disadvantages

1. Salts sediments are deposited on the inner surfaces of water tubes.

5.1.5 BHEL Boiler

- ❖ It is a high pressure boiler
- ❖ *Bharath Heavy Electrical* limited high pressure boiler as shown in figure.

Construction

It consists of

1. Boiler drum
2. Radiant super heaters
3. Convective super heaters,
4. Air pre heaters
5. Economiser
6. Forced draught fan
7. Induced draught fan,
8. Electrostatic precipator, and

Working

The working of this boiler is given in two circuits

1. *Water and steam circuit*
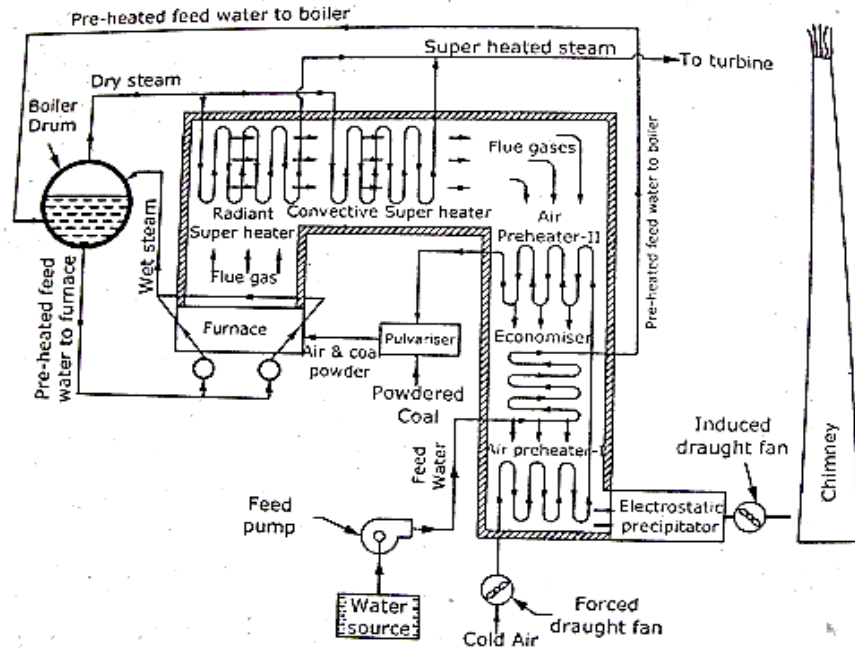2. *Air and the gas circuit*

Fig 5.6

Working

The working of this boiler is given in two circuits

Water and steam circuit:

Feed water is pumped to the economiser. In the economiser feed water is heated by the flue gases. The preheated water enters the boiler drum. The water is circulated through number of pipes in the furnace, where the fuel is burnt. Water is converted into steam and mixture of water and steam flows back into the boiler drum. In the drum dry steam is separated from wet steam. This dry steam is then passed through the radiant superheater and convection superheater and becomes superheated steam. This superheated steam is supplied to the turbine.

Air and the gas circuit;

Forced draught fan forces the air into air pre heaters I and II, air absorbs heat from flue gases in the air pre heater. This hot air enters the pulversier and carries the powdered coal wit it to the furnace. In the furnace burning of fuel takes place and heat energy is released.

The hot flue gases from the furnace pass over the radiant super heater and convective super heater to super heat the steam. Then it passes through air pre heater II (to be pre heat the air). Then the flue gases pass through an electrostatic precipator. In the electrostatic precipitator dust and ash are separated from flue gas. The cleaner flue gas passes through induced draught fan and is exhausted to the atmosphere through chimney"

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5.1.6 Boiler mountings

❖ The boilers are equipped with some components which are primilarly intended for the _**safety of the boiler and for complete control of steam**_ generation process. These components are known as boiler mounting.

❖ Mounting are important part of the boiler

❖ Without Mountings, boilers should not be operated.

Functions are

❖ They are for the safety of the boiler

❖ They are for the control of the process of steam generation.

Mountings are

1. Safety valve
2. Water level indicator
3. Fusible plug
4. Pressure gauge
5. Steam stop valve
6. Feed check valve
7. Blow off cock
8. Man holes

1. Safety valve

These are the devices attached to the steam chest for preventing explosions due to excessive internal pressure of the steam. A steam boiler is, usually, provided with two safety valves. These are directly placed on the boiler. In brief, the function of safety valve is to below off the steam when the pressure of steam inside the boiler exceeds the working pressure. The following are the four type of safety valves:

1. Dead weight safety valve
2. Lever safety valve
3. Spring loaded safety valve
4. High steam and low water safety valve

It may be noted that the first three types of the safety valves are usually employed with stationary boilers, but the fourth type is mainly used for locomotive and marine boiler.
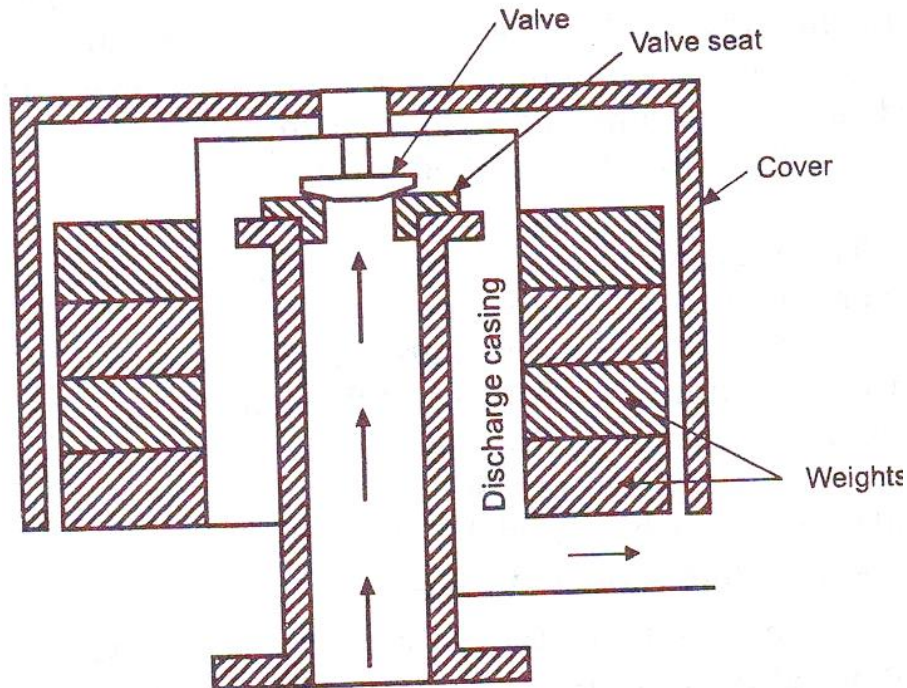
i). Dead weight safety valve

Fig 5.7

A dead weight safety valve, used for stationary boilers, is shown in the fig. the valve is made of gun metal, and rests on its gun metal seat. It is fixed to the top of a steel pipe. This pipe is bolted to the mounting block, riveted to the top of the shell. Both the valve and the pipe are covered by a case which contains weights. These weights keep the valve on its seat under normal working pressure. The case hangs freely over the valve to which it is secured by means of a nut.

When the pressure of steam exceeds the normal pressure, the valves as well as the case (along with the weights) are lifted up from its seat. This enables the steam to escape through the discharge pipe, which carries the steam outside the boiler house.

The lift of the valve is controlled by studs. The head of the studs projects into the interior of casting. The center of gravity of dead weight safety valve is considerably below the valve which ensures that the load hangs vertically.

The dead weight safety valve has the advantage that it cannot be readily tempered because any added weight must be equal to the increased pressure of steam on the valve. The only disadvantage of these valves, is the heavy load which these valves carry.
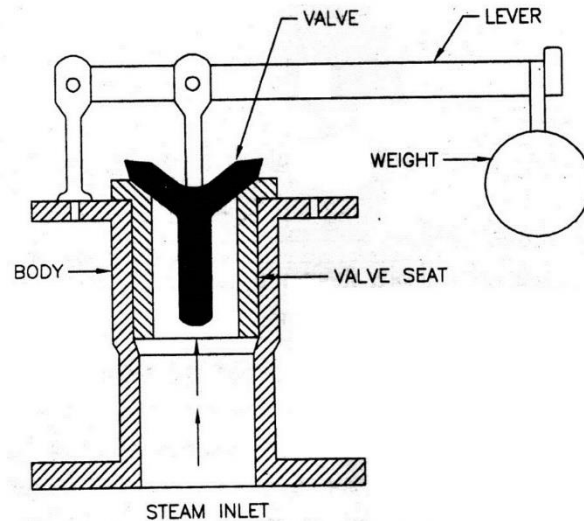
ii). Lever safety valve

Fig 5.8

A lever safety valve used on stream boiler shown in fig. It serves the purpose of maintaining constant safety pressure inside the steam boiler. If the pressure inside the boiler exceed the designed limit, the valve lifts from its seat and blows off the steam pressure automatically.

A lever safety valve consist of a valve body with a flange fixed to the steam boiler. The bronze valve seat is screw to the body, and the valve is also made of bronze. It may be noted that by using the valve and seat of the same material, rusting is considerable reduced. The thrust on the valve is transmitted by the suit. The guide keeps the lever in a vertical plane. The load is properly adjusted at the other end of the lever.

When the pressure of steam exceeds the safe limits, the upward thrust of steam raise the valve from its seat. This allows the steam to escape till the pressure falls back to its normal valve. The valve then returns back to its original position.

iii). spring loaded safety valve

A spring loaded safety valve is mainly used for locomotives and marine boilers. It is loaded with spring instead of weights. The spring is made of round or square spring steel rod in helical form. The spring may be in tension or compression, as the steam pressure acts along the axis of the spring. In actual practice, the spring is placed in compression.
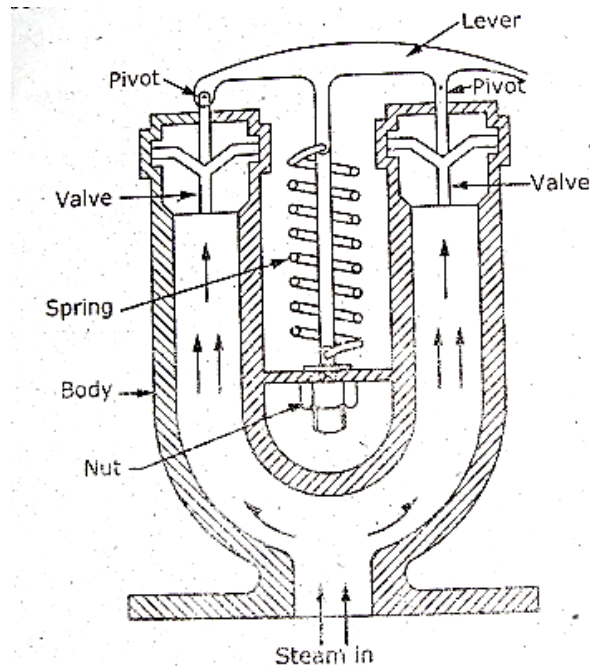
Fig 5.9

A Rams bottom spring loaded safety valve is shown in Fig. It is usually, fitted to locomotives. It consists of a cast iron body connected to the top of a boiler. It has two separate valves of the small size. These valves have their seating in the upper ends of two hollow valve chests. These valve chests are united by a bridge and a base. The base is bolted to a mounting block on the top of a boiler over the fire box.

The valves are held down by means of a spring and a lever. The lever has two pivots at E and F. The pivot E is joined by a pin to the lever, while the pivot F is forged on the lever. These pivots rest on the centres of the valves. The upper end of the spring is hooked to the arm H, while the lower end to the shackle, which is secured to the bridge by a nut. The spring has two safety links, one behind the other, or one on either side of the lever connected by pins at their ends. The lower pin passes through the shackle while the upper one passes through slot in arm H of the lever. The lever has an extension, which projects into the driver's cabin. By pulling or raising the lever, the driver can release the pressure form either valve separately.

iv). High steam and low water safety valve

These valves are placed at the top of Cornish and Lancashire boilers only. It is a combination of two valves, one of which is the lever safety valve which blows off steam when the working pressure of steam exceeds. The second valve operates by blowing off steam when the water level becomes too low.
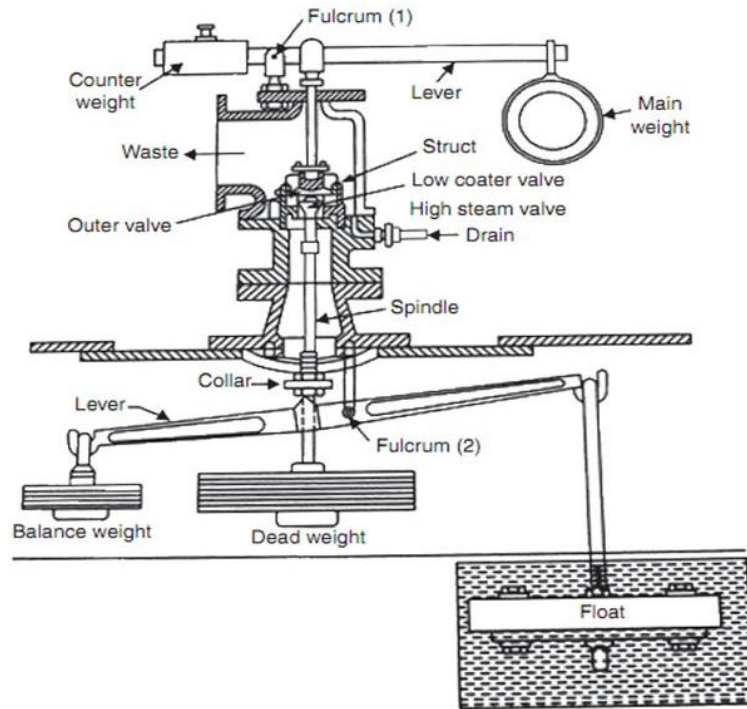
Fig 5.10

A best known combination of high steam low water safety valve is shown in Fig. It consists of a main valve (known as lever safety valve) and rests on its seat. In the centre of the main valve, a seat for a hemispherical valve is formed for low water operation. This valve is loaded directly by the dead weights attached to the valve by a long rod. There is a lever J-K, which has its fulcrum at K. The lever has a weight E suspended at the end K. When it is fully immersed in water, it is balanced by a weight F at the other end J of the lever.

When the water lever falls, the weight E comes out of water and the weight F will not be sufficient to balance weight E. Therefore weight E comes down. There are two projections on the lever to the left of the fulcrum which comes in contact with a collar attached to the rod. When weight E comes down, the hemispherical valve is lifted up and the steam escapes with a loud noise, which warns the operator. A drain pipe is provided to carry water, which is deposited in the valve casing.

2. Water level indicator

Fig 5.11

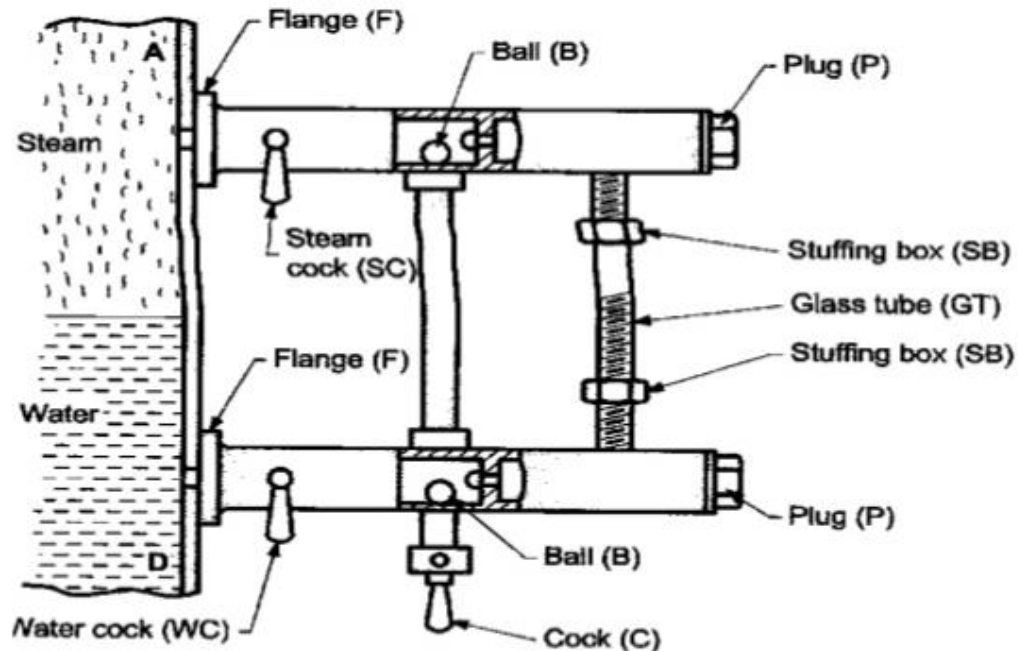It is an important fitting, which indicates the water level inside the boiler to an observer. It is a safety device, upon which the correct working of the boiler depends. This fittings may be seen in the front of the boiler, and are generally two in number.

A water level indicator, mostly employed in the steam is shown in the fig. it consists of three cocks and a glass tube. Steam cock SC keeps the glass tube in connection with the steam space. Water cock WC puts the glass tube in the connection with the water level in the boiler. Drain cock C is used at frequent interval to ascertain that the steam and water cock are clear.

In the working of the steam boiler and for a proper functioning of the water level indicator, the steam and water cocks are opened and the drain cock is closed. In this case, the handles are placed in a vertical position as shown in fig. the rectangular passage at the end of the glass tube contains two balls.

In case the glass tube is broken, the two balls are carried along its passage to the ends of the glass tube. It is thus obvious, that water and steam will not escape out. The glass tube can be easily replaced by closing the steam and water cocks and opening the drain cock.

When the steam boiler is not working, the bolts may be removed for cleaned. The glass tube is kept free from leaking by means of conical ring and the gland nut.

3. Fusible plug
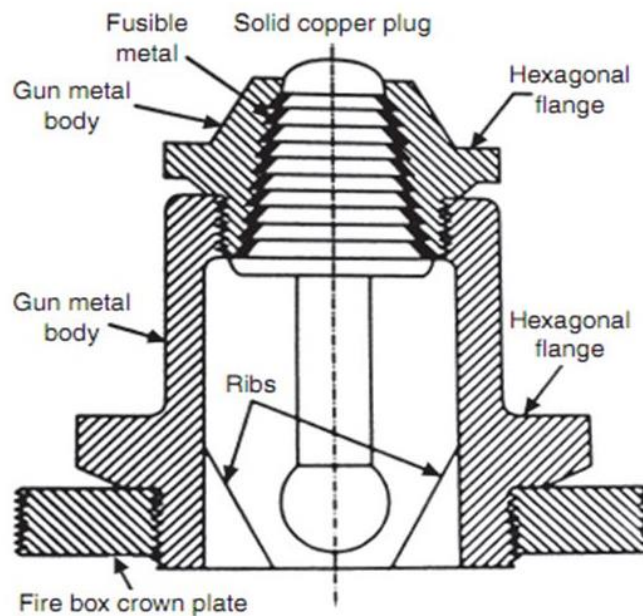
Fig 5.12

It is fitted to the crown plate of the furnace or the fire box. Its object is to put off the fire in the furnace of the boiler when the laver of water in the boiler falls to an unsafe limit, and thus avoids the explosion which may take place due to overheating of the furnace plate.

A fusible plug consists of a hollow gun metal plug P1 as shown in Fig. It is screwed to the furnace crown. A second hollow gun metal plug P2 is screwed to the first plug. There is also a third hollow gun metal plug P3 are grooved so that when the fusible metal is poured into the plug,P2 and P3 are locked together. A hexagonal flange is provided on plug P1 to take a spanner for fixing or removing the plug P1. There is a hexagonal flange on plug P2 for fixing or removing it. The fusible metal is protected from fire by the flange of the lower end of plug P2. There is also a contact at the top between P2 and P3 so that the fusible metal is completely enclosed. The fusible plugs must be kept in a good condition and replaced annually. A fusible plug must not be refilled with anything except fusible metal.

4. Pressure gauge
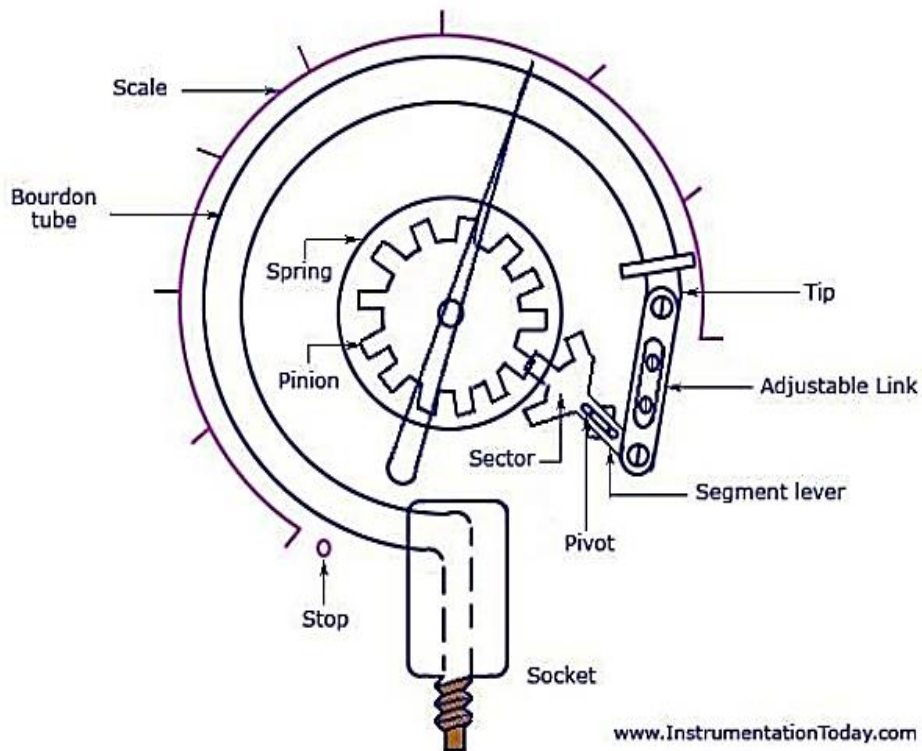
Fig 5.13

A pressure gauge is used to measure the pressure of the steam inside the steam boiler. It is fixed in front of the steam boiler. The pressure gauges generally used are of Bourden type.

A Bourden pressure gauge, in its simplest form, consist of an elliptical elastic tube ABC bent into an arc of a circle as shown in the fig. this bent up tube is called Bourden's tube.

One end of the tube gauge is fixed and connected to the steam space in the boiler. The other end is connected to a sector through a link. the steam, under pressure, flows into the tube. As a result of this increased pressure, the Bourden's tube tends to straighten itself. Since the tube encased in a circular curve, therefore it tends to become circular instead of straight. With the help of a simple pinion and sector arrangement, the elastic- deformation of the Bourden's tube rotates the pointer. This pointer moves over a calibrated scale, which directly gives the gauge pressure.

5. Steam stop valve
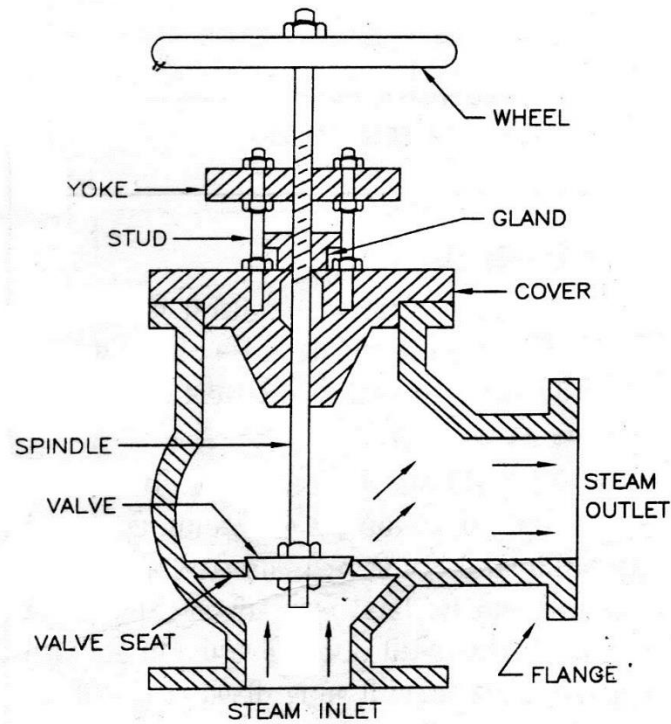
Fig 5.14

It is the largest valve on the steam boiler. It is, usually, fitted to the highest part of the shell by means of a flange as shown in Fig. The principal functions of a stop valve are :

1. To control the flow of steam from the boiler to the main steam pipe.

2. To shut off the steam completely when required.

The body of the stop valve is made of cast iron or cast steel. The valve, valve seat and the nut through which the valve spindle works, are made of brass or gun metal.

The spindle passes through a grand and stuffing box. The spindle is rotated by means of a hand wheel. The upper portion of the spindle is screwed and made to pass through a nut in a cross head carried by two pillars. The pillars are screwed in the cover of the body as shown in Fig. The boiler pressure acts under the valve, so that the valve must be closed against the pressure. The valve is, generally, fastened to the spindle which lifts it up.

A non-return valve is, sometimes, fitted near the stop valve to prevent the accidental admission of steam from other boiler. This happens when a number of boiler are connected to the same pipe, and when one is empty and under repair.

6. Feed check valve
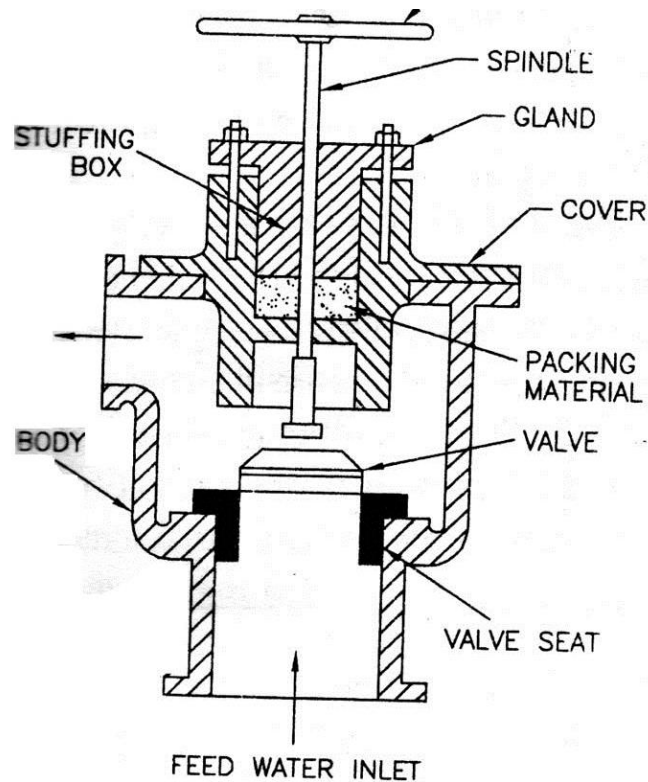
Fig 5.15

It is a non-return valve, fitted to a screwed spindle to regulate the lift. Its function is to regulate the supply of water, which is pumped into the boiler, by the feed pump. This valve must have its spindle lifted before the pump is started. It is fitted to the shell slightly below the normal water level of the boiler.

A feed check valve for marine boilers is shown in Fig. It consists of a valve whose lift is controlled by a spindle and hand wheel. The body of the valve is made of brass casting and except spindle is made of metal. A flange is bolted to the end of boiler at a point from which perforated pipe leads the feed water. This pipe distributes the water in the boiler uniformly.

7. Blow off cock
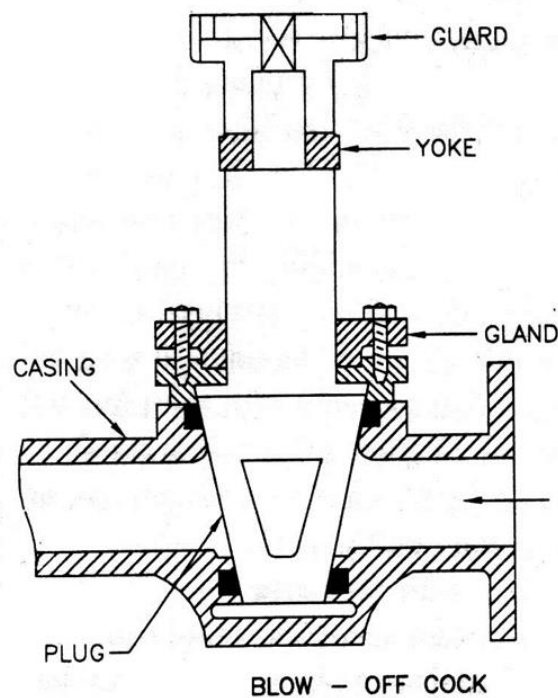
Fig 5.16

The principal functions of a blow-off cock are :

1. To empty the boiler whenever required.

2. To discharge the mud, scale or sediments which are accumulated at the bottom of the boiler.

The blow-off cock, as shown in Fig., is fitted to the bottom of the boiler drum and consists of a conical plug fitted to the body casing. The casing is packed, with asbestos packing, in grooves round the top and bottom of the plug. The asbestos packing is made tight and plug bears on the packing. It may be noted that the cocks in this way keep the grip better under high pressure and easily operated than unpacked.

The shank of plug passes through a gland and stuffing box in the cover. The plug is held down by a yoke and two stud bolts (not shown in figure). The yoke forms a guard on it. There are two vertical slots on the inside of a guard for the box spanner to be used for operating cock.

8. Man holes

Function

- ❖ The function of man hole is to allow a man to enter inside the boiler for cleaning and inspecting of interior parts of the boiler.
- ❖ Man holes are provided on the boiler shell at any convenient position such that a man can enter through the hole into the boiler.
- ❖ Man holes are usually made in oval shape .They are large in size (40 x 30 cm) and the opening is closed by a cover called man hole door.

5.1.7 Boiler accessories

- ❖ To increase the boiler efficiency and proper working of boiler.
- ❖ To help for smooth and proper working of boiler.
- ❖ To prevent the energy losses.

The following accessories are generally used with the boilers

1. Feed pump
2. Injector
3. Economise
4. Super heater
5. Air preheater
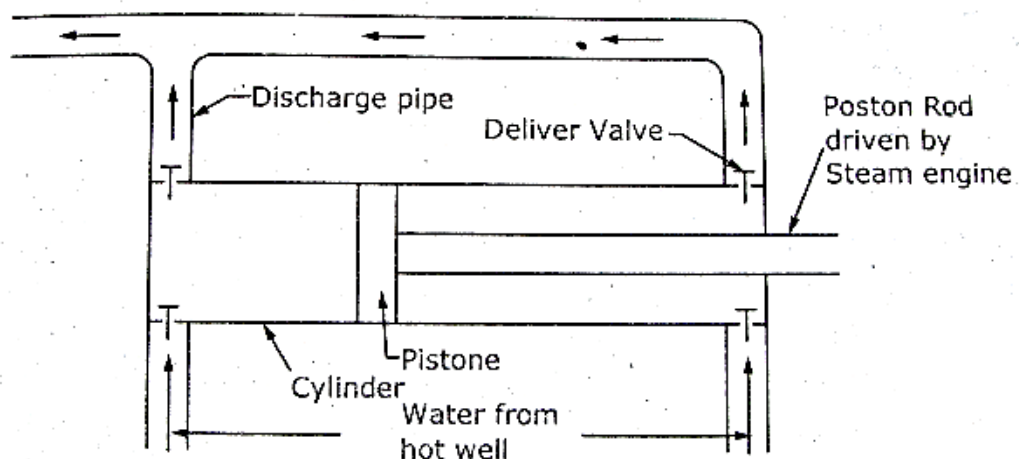6. Steam trap
7. Steam separator

1. Feed pump

Fig 5.17

We know that water, in a boiler, is continuously converted into steam, which is used by the engine. Thus we need a feed pump to deliver water to the boiler.

The pressure of steam inside a boiler is high. So the pressure of feed water has to be increased proportionately before it is made to enter the boiler. Generally, the pressure of feed water is 20% more than that in the boiler.

A feed pump may be of centrifugal type or reciprocating type. But a double acting reciprocating pump is commonly used as a feed pump these days. The reciprocating pumps are run by the steam from the same boiler in which water is to be fed. These pumps may be classified as simplex, duplex and triplex pumps according to the number of pump cylinders. The common type of pump used is a duplex feed pump, as shown in fig. This pump has two sets of suction and delivery valves for forward and backward stroke. The two pumps work alternately so as to ensure continuous supply of feed water.

2. Injector

Function

- ❖ The main function of injector is to feed water to the boiler with the help of steam jet.
- ❖ They are used in vertical and locomotive boilers.
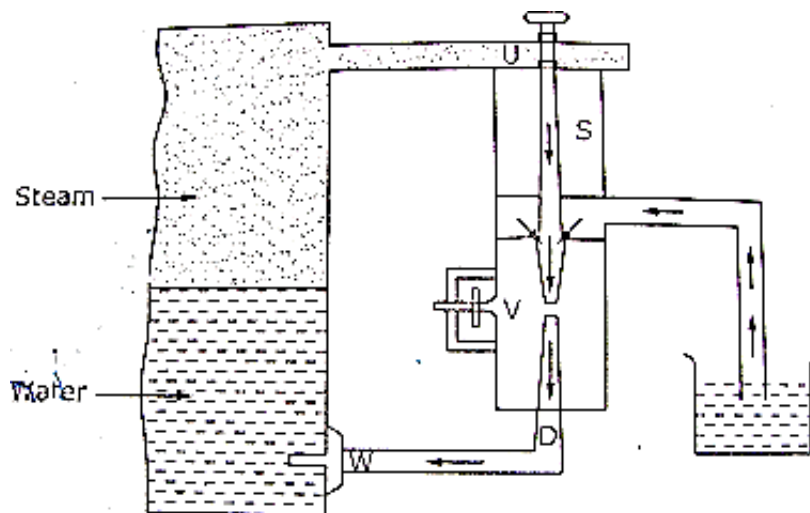
Fig 5.18

It consists of convergent steam nozzle inside a mixing cone and a diverting tube. A mixing chamber is provided with over flow pipe in between mixing cone and divergent tube. The angular space between the convergent nozzle and mixing cone may be altered by turning the hand wheel by doing so, the amount of water supplied through the suction pipe can be adjusted.

When the steam is allowed to flow through the injector. The pressure in the mixing chamber is more than atmospheric. Water under condensed steam pass out through the

overflow pipe. As soon as the steam nozzle is brought into its correct position. Jet action is established.

The high velocity steam creates partial vacuum in the mixing cone, further as steam condensed during the mixing. Vacuum is increased. Hence water rushes into the mixing chamber with high velocity from the suction pipe. When the water flows through the divergent cone. The pressure of water increases and it is fed to the boiler.

3. Air pre heater
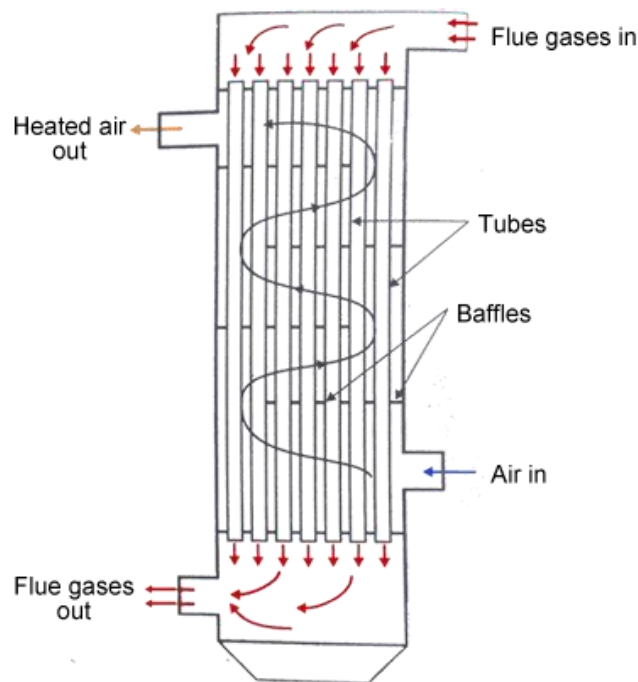
Fig 5.19

An air preheater is used to recover heat from the exhaust flue gases. It is installed between the economizer and the chimney. The air required for the purpose of combustion is drawn through the air preheater where its temperature is raised. It is then passed through ducts to the furnace. The air is passed through the tubes of the heater the tubes of the heater internally while the hot flue gases are passed over the outside of the tubes.

The following advantages are obtained by using an airpreheater:-

1. The preheated air gives highter furnace temperature which results in more that transfer to the water and thus increases the evaporative capacity per kg of fuel.
2. There is an increase of about 2% in the boiler efficiency for each 35-45°C rise in temperature of air.
3. It results in better combustion with less soot, smoke and ash.
4. It enables a low grade fuel to be burnt with less excess air.

4.Economiser

The function of the economiser is to pre heat the water and delivers to boiler drum.
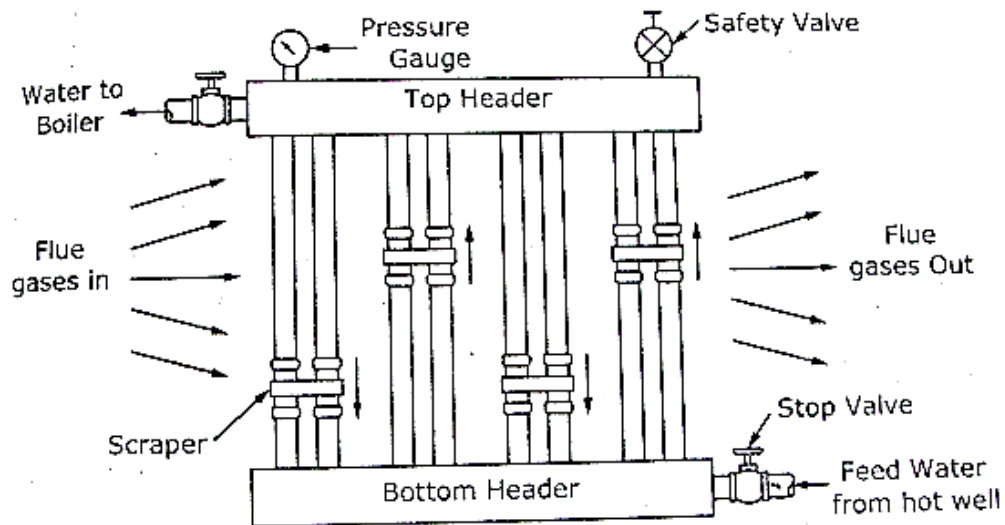
Fig 5.20

The feed water is pumped to the bottom header and this water reaches the upper header through the vertical tubes. From the upper header water leads to the boiler drum. When the water flows upwards inside the tube it gains heat from the hot flue gases which are made to pass over the external surface of the tubes. This preheated feed water which is supplied to the boiler drum.

To keep the surface of the tubes clean from soot and ash deposits, scrappers are provided in the tubes.

The action of adjacent pairs of scrapper is in opposite direction. The deposits of soot are collected in the soot chamber which is situated below the bottom header. The soot from the soot chamber is removed periodically

5. Super heater

Super heaters are used in the boiler to increase the temperature of seam above the saturation point. The super heaters are super heat the steam or dry steam. The steam is heated in the super heater at constant pressure.

Line sketch of super heater is shown in figure. It consists of U shaped steel tubes. They are connected to heaters. The arrangement is placed on the path of the flue gases. The temperature of flue gases at his region is not less than $500\,^{\circ}$C.
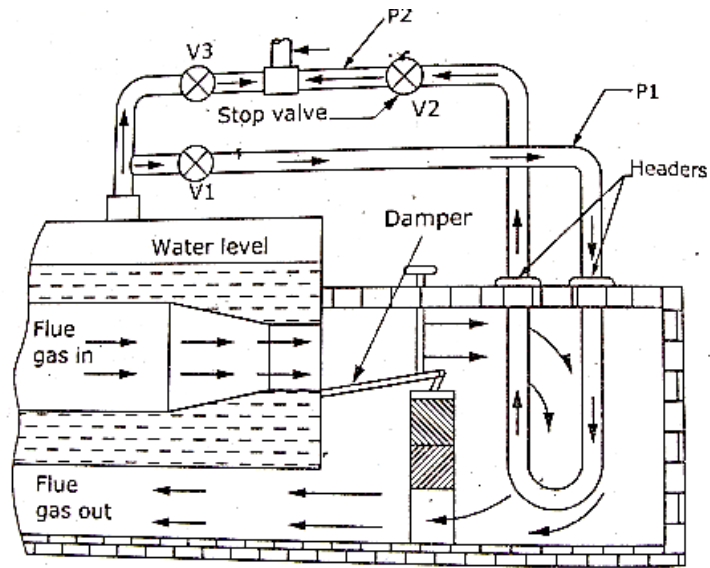
Fig 5.21

In large boilers, the super heaters may be placed in a separate independently fired furnace. The seam enters the headers of super heater from the boiler drum and flow through U tubes. During the flow, it absorbs the heat from the flue gases through steel tubes. This it becomes superheated steam. Then it is supplied to the steam engine or turbine.

6. Steam trap

The function of steam trap is to collect the water and discharge automatically without allowing the steam to escape.
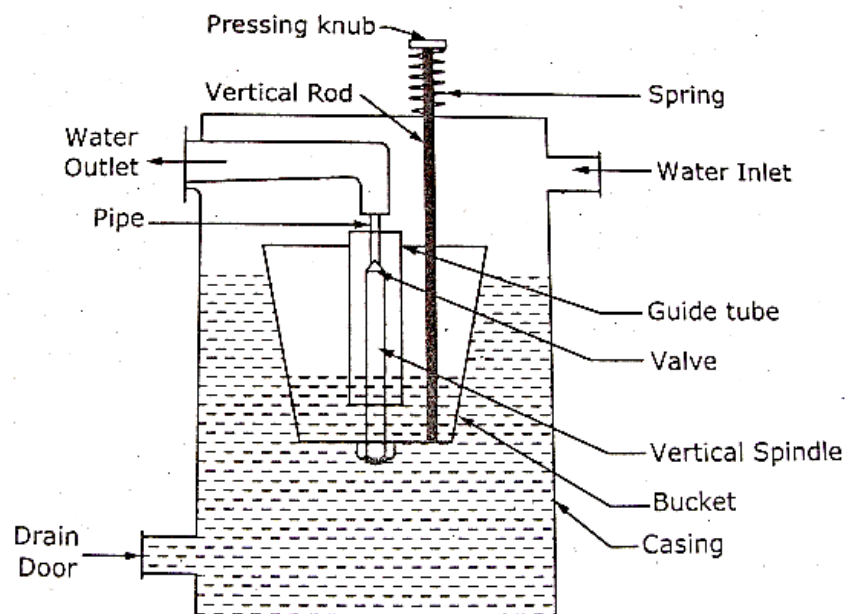
Fig 5.22

When the steam flow in the pipe line water from due to parallel condensation. This water is removed by the steam trap. The function of steam trap is to collect the water and discharge automatically without allowing the steam to escape. Figure show the line diagram of a steam trap.

It consists of a casting and a bucket with valve, the casting contains sufficient water to float a bucket. The bucket carries a vertical spindle. A valve floated to the top of the spindle is in the outlet pipe. A guide tube guides the spindle.

The trap is so located that the condenser in the steam pipe flows by gravity to it. When the water rises high enough in the casing. It overflows into the bucket, now the bucket sinks down and the valve opens. The water in the bucket is then forced out through the guide tube by the pressure of seam acting on it. The water passing through the pipe leaves the trap through the outlet. When the bucket is nearly empty, it again float and the valve close automatically.

7. Steam separator

The function of the steam separator is to separate water particles from steam before it is supplied to a steam engine or turbine.

It is fitted as close as possible to the engine or turbine in the main steam pipe line leading from the boiler to the engine or turbine.
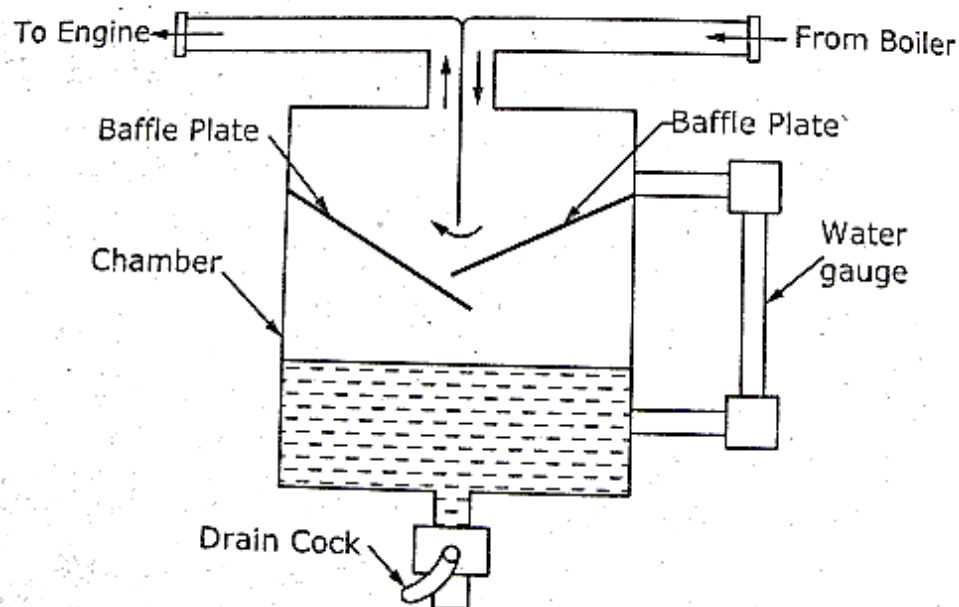
Fig 5.23

A line diagram of seam separator is shown in the figure. It consists of a cylindrical vessel with baffle plates. A water gauge is fitted with the separator. The steam coming from the boiler strikes the baffle plates. It is then suddenly deflected up. So the direction of its flow is changed.

The striking particles of water, due to their greater weight and inertia, fall to the bottom of the separator. The water gauge indicates the water level in the separator. Then the steam free from water particles, goes to the engine or turbine. A drain cock provided at the bottom of the separator to drain away the water, when it necessary.

5.1.8 Comparison of boiler mountings and accessories

Boiler Mountings	Boiler Accessories
Mountings are fitted for safety purpose and to control the process of steam generator	Accessories are fitted to increase the efficiency of plant and to help for the smooth and proper working of boiler.
Mountings are integral part of the boiler	Accessories are not integral part of the boiler
A boiler should not be operated without mountings	A boiler can run without accessories
Mountings are usually fitted on the boiler shell	Accessories are usually installed outside the boiler shell.
Examples: Safety valve, water level indicator, fusible plug, pressure gauge, etc	Examples : Economiser, super heater, Air pre heater etc.

5.1.9 Feed water Treatment

Natural water contains solid, liquid, and gaseous impurities. This natural water cannot be used for the generation for steam in the boilers. The various impurities present in the natural water should be removed before it is supplied to the boiler. Hence impure water is treated to remove suspended solids, dissolved solids and dissolved gases. ***The treatment methods are as given below.***

1. **Internal Treatment**
2. **External Treatment**

1. Internal Treatment:

> *The impurities are removed by treating the water in the boiler itself during evaporation and it is known as internal treatment.*

> In this method, chemical are added to the water in the boiler to precipitate the impurities. These can be removed in the form sludge (mud) or salts

The common internal treatments given to the boiler feed water are

    a) Sodium carbonate (soda ash) treatment

    b) Phosphate treatment

    c) Colloidal treatment

    d) Blow down

a)Sodium carbonate treatment:

> Sodium carbonate is added to the boiler water.

> It reacts with sulphates of calcium and magnesium in the boiler water and produces with calcium and magnesium bicarbonates as sludge which can be removed easily.

b)Phosphate treatment:

> Sodium phosphate is added in the boiler water.

> It reacts with calcium sulphate and precipitates tricalcium phosphate which can be easily removed.

c)Colloidal treatment

> Colloidal materials such as starch (gruels used to stiffen clothes), tannins (a kind of binding substance), lines are added to the boiler water.

> They absorb the inorganic sledges formed. They prevent precipitating particles from sticking to each other or to the boiler drum surfaces.

d)Blow down:

> Draining off some quantity of water in the boiler through the bottom drain of the boiler known as blowing down.

> The water thus discharge is known as blow down.

> By blowing down, the concentration of dissolved solids (which causes foaming and priming) can be reduced

2. External Treatment:

> Removal of impurities present in water before it is supplied to the boiler, is known as external treatment.

> The different external treatments given to the boiler feed water are as follows

a) Mechanical treatment

b) Thermal treatment

c) Chemical treatment

d) Demineralization

a) Mechanical treatment

➤ Suspended matter can be removed by this treatment. it includes *sedimentation, coagulation and filtration*,

➤ During Sedimentation, in valves allowing the water to stand quietly for some time. In this way the way the solid matter settlements down and is removed periodically.

➤ During coagulation, some coagulates like aluminium sulphate, sodium sulphate or ferrous sulphates are added to remove minute colloidal suspensions.

➤ During filtration, the water is passed through the filter to remove the suspended matter.

b) Thermal treatment

➤ In this treatment, dissolved gases (oxygen, carbondioixde etc) in the water are removed.

➤ This is done by heating the water to about 110 C with subsequent agitation while heating.

c) Chemical treatment

➤ In this treatment, some chemicals are added to water and this causes precipitation of impurities.

➤ The various chemicals used are sodium carbonate (soda ash), sodium phosphate, sodium hydroxide (Lime) etc. in the external treatment.

➤ The chemicals are added to the raw water in the tank.

➤ The external treatment is suitable for high pressures boiler.

d) Demineralization

➤ It is used to remove mineral contents in water.

➤ In this process raw water passes through the hydrogen zeolite exchangers where caution is removed.

➤ Then this water flows to an ion exchanger and degasified and finally passes through the silicon absorber. The water thus obtained is soft water.

5.1.10 Starting boiler from cold condition

The following steps are to be followed for starting the boiler from cold condition

1. All the joints and fittings are checked

2. Boiler mountings are checked for their proper functioning

3. The stokers, controls, etc., are checked for their proper functioning.

4. The boiler is filled with water upto the specified level. The drain and blow off cocks are opened and water is allowed to escape for cleaning purpose. Then the valves are closed again.

5. The boiler is again refilled with the treated water slightly above the normal operating level.

6. The pressure gauge is connected by opening the steam valve

7. The steam is raised to the predetermined pressure by boiling the treated water

8. The steam valves are opened to allow steam into super heater. Chemicals are introduced into the boiler with the feed water.

9. Upto blow down, the water is heated till the half of the working pressure is reached. Then the pressure is raised to the working pressure.

10. Maintain drum water level at normal

5.1.11 Safety precautions in boiler operation

The following safety precautions should be followed during boiler operation

1. Workers should be provided with safety devices to avoid cuts,burns,falts and injury

2. Constant watch on the boiler mountings is necessary

3. The fire extinguishing equipments should be kept in ready condition for use at any time

4. All the necessary mounting should be fitted in the boiler and they should be checked for proper functioning.

5. Guards, steps, floor and railings should be kept clean.

5.1.12 Clauses of Indian boiler act.

To ensure boiler safety some standard rules and regulations have been enforced through a law known as Indian boiler act, 1923. The law has to be followed by all boiler owners.

This law has the following provisions.

Boiler Registration:

➢ An owner cannot use a boiler unit it has been registered with the chief inspector of boilers.

➢ The chief inspector will determine the maximum working pressure by a test. He will issue a certificate to this effect. It will authorise the owner to run the boiler a present not exceeding the maximum limit.

➢ Boiler should be operate by a person holding competency certificate (boiler operate certificate)

Renewable of Certificate

➢ The registration should be renewed on the expiry of the period for which it was granted.

➢ The due date (for which it was certified expires

➢ The registration should also be renewed in the case of any accident in the boiler.

Transfer of boiler

➢ When a boiler transferred from one state to another state, it must be noted in the register.

➢ The owner of boiler should apply to the cheif inspector of the boiler of the state in which the boiler is to be installed for the regisation of transfer and obtain a fresh certificate of registration

Accident Report

➢ In the case of accident, the owner has to report to the chief inspector within 24 hours giving full details of nature and cause of accident.

➢ All details about the accident and injuries caused to the boiler or any persons should be given.

➢ The chief inspector os boiler will inspect the accident site and decide whether the boiler can be reused.

Boiler Repairs

➢ Any repair of the boiler should be carried out after obtaining sanction from the chief inspector of boiler.

Alternation and Renewals

➢ Any alternation in the boiler structure and additions or renewals should be made only obtaining permission from the chief inspector of boiler.

Penalties

➢ Any boiler owner violating the law is libel for prosecution and punishment with fine.

5.2. Performance of Boiler

Introduction:

The performance of steam boiler is measured in terms of its evaporative capacity. However, the evaporative capacities of two boilers cannot be compared unless both the boilers have the same feed water temperature, working pressure, fuel and the final condition of steam. In actual practice, the feed water temperature and working pressure varies considerably. It is thus obvious, that the comparison of two boilers becomes difficult unless some standard feed temperature and working pressure is adopted.

The feed temperature usually adopted is 100° C and the working pressure as normal atmospheric pressure, *i.e.* 1.01325 bar. It is assumed that the boiler is supplied with water at the boiling temperature (100° C) corresponding to the atmospheric pressure.

5.2.1. Actual Evaporation

Actual evaporation is defined as the quantity of water evaporated per kg of fuel used.

It is denoted by the letter M_a. It is expressed in kg / kg of fuel.

$$m_a = \frac{The\ quantity\ of\ water\ evaporated/\ hr\ of\ steam\ generated/hr}{The\ quantity\ of\ fuel\ used\ /\ hr}$$

$$m_a = \frac{mw}{mf} = \frac{ms}{mf} \ \ \textbf{kg / kg of fuel}$$

m_w = mass of water evaporated (kg/hr)

m_s = mass of steam generated (kg/hr)

m_f= Mass of fuel used (kg/hr)

Actual evaporation is also known as steam production ratio or steam production rate

5.2.2 Equivalent Evaporation

It is the amount of water evaporated from feed water at 100 ˚C and formed into dry and saturated steam at 100 ˚C at normal atmospheric pressure. It is usually written as "*from and at 100 ˚C*"

Let m_a be the actual evaporation per kg of fuel

         h   be the specific enthalpy of steam produced.

         h_w be the specific enthalpy of feed water supplied

Heat required to evaporate 1 kg of water = (h- h_w)

Total Heat required to evaporate m_a kg of water = m_a (h- h_w)

Evaporation of one kg of water at100 ˚C into dry saturated steam at 100 ˚C requires 2257 kJ/kg of heat

Equivalent evaporation of boiler from and at 100 ˚C $\quad$ $\mathbf{m_e} = \dfrac{ma\ (h-hw)}{2257}$

Note:

h_{f1}= Enthalpy or sensible heat of feed water in kJ/kg of steam corresponding to t_1˚C (from Steam tables)

h = Enthalpy or total heat of steam in KJ/kg of steam corresponding to a given working pressure (from steam tables)

$\quad$ h=h_f + x. h_{fg} $\qquad$ (for wet steam)

$\quad$ = h_f+h_{fg} = h_g $\qquad$ (for dry steam)

$\quad$ =h_g+C_p(t_{sup} − t) $\qquad$ (for superheated steam)

5.2.3 Factor of Evaporation:

$\qquad$ It is ratio of the heat required to produce one kg of steam in the boiler under standard condition to the heat required to evaporate one kg of water from and at 100 ˚C. It is denoted by the letter F.

$\qquad$ $\mathbf{F=} \dfrac{(h-hw)}{2257}$

$\quad$ h $\qquad$ be the specific enthalpy of steam produced.

$\quad$ h_w $\qquad$ be the specific enthalpy of feed water supplied

Relationship between m_a, m_e and F

We know $\qquad$ $m_e = \dfrac{ma\ (h-hw)}{2257}$

$\qquad$ $m_{e} = m_a \times F$

5.2.4 Boiler Efficiency

Boiler efficiency of a boiler is defined as the ratios of heat energy utilize to generate steam In the boiler under actual operation condition to the heat energy released by complete combustion of fuel during the same time. It is also as thermal efficiency of a boiler.

Thus boiler efficiency $= \dfrac{\text{Heat energy utilized in the boiler per hour}}{\text{Heat supplied to the boiler by the fuel per hour}}$

$$= \frac{ms(\,h - hw\,)}{mf \; X \; C.V.} X \, 100 = \frac{ma(\,h - hw\,)}{C.V.} X \, 100$$

Where m_s - amount of water evaporator per hour

      h   - heat enthalpy of steam generated in KJ / kg

      h_w -specific enthalpy of feed water in kJ/kg

      C.V.   - calorific valve of the fuel in kJ/kg

      m_f - mass of fuel burnt in kJ/kg

      m_a -Actual evaporation in kg /kg burnt of fuel

If the boiler is fitted with economizer and super heater then the efficiency of the unit is called overall efficiency of the boiler plant.

5.2.5 Economiser efficiency

It is defined as "the ratio between the heat gained by the feed water in the economiser and the heat in the flue gases entering the economiser measured above the temperature of air."

Economiser efficiency $= \dfrac{\text{heat absorbed by the feed water in the economiser}}{\text{heat in the flue gases entering the economiser}}$

Economiser efficiency $= \dfrac{ms \; cpw \; (\,t2 - t1\,)}{mg \; cp \; (tg - ta)} \, x100$

$$= \frac{ms \; (hw2 - hw1\,)}{mg \; cp \; (tg - ta)} \, x \, 100$$

m_s - actual evaporation of steam per hour

C_{pw} - specific heat capacity of water (2.1 KJ/Kg K)

t_1 - Temperature of feed water entering economiser (°C)

t_2 - Temperature of feed water leaving economiser (°C)

h_{w1} - specific enthalpy of feed water entering the economizer

h_{w2} - specific enthalpy of feed water leaving the economizer

m_g - Mass of flue gases passing economise (kJ/hr or Kg/ kg of fuel)

C_p - specific heat of flue gases (if not given assume = 1.005 KJ/kg k)

t_s - Temperature of flue gases entering economizer (°C)

t_a - Temperature of boiler room or temperature of air supplied to the boiler (°C)

The percentage of heat absorbed in the economizer

$$= \frac{m_s\,(h_{w2}-h_{w1})}{m_f \times CV} \times 100$$

5.2.6 Super heater efficiency

It is also known as effective of super heater. It is denoted as the ratio of heat absorbed by the steam in the super heater per hour to the heat supplied in flue gases per hour.

Super heater efficiency $= \dfrac{\text{heat absorbed in super heater per hr}}{\text{heat supplied in flue gases per hour}} \times 100$

$$= \frac{m_s\,(h_{sup}-h_{wet})}{m_g\,c_p\,(t_{f2}-t_{f1})} \times 100$$

m_s -actual evaporation of steam per hour

m_g -mass flue gases per hour

h_{sup} -specific enthalpy of super-heated steam in kJ/kg

h_{wet} -specific enthalpy of wet steam in kJ/kg

C_p -specific heat of flue gases in kJ/kg.k

t_{f1} -temperature of flue gases entering the superheater.

t_{f2} -temperature of flue gases leaving the superheater

The percentage of heat supplied to the super heater.

$$= \frac{ms\,(hsup - hwet\,)}{mf \times CV)} \times 100$$

5.2.7 Factors affecting boiler efficiency

Factors affecting the boiler efficiency are given below:
1. Calorific valve of the fuel used
2. Feed water temperature
3. Condition of steam produced
4. Amount of steam produced per kg of fuel

5.2.8 Boiler Trial:

The main objects of a boiler trial are:
1. To determine the generating capacity of the boiler.
2. To determine the thermal efficiency of the boiler when working at a definite pressure
3. To prepare heat balance sheet for the boiler.

5.2.9 Heat Losses in a Boiler

The portion of the heat of combustion of fuel which is not used for the production of steam in a boiler is called as heat losses. The various heat losses in a boiler are:
1. Heat lost in hot flue gases
2. Heat lost due to hydrogen present in the fuel
3. Heat lost due to unburnt fuel
4. Heat lost in evaporation the moisture in the fuel
5. Heat lost due to incomplete combustion
6. Heat carried away by excess air
7. Heat lost due to external radiation

5.2.10 Heat Balance Sheet

a) Heat supplied (Q_s)

It is the heat supplied by burning of fuel

Heat supplied per kg of fuel = C.V

b) Heat utilized (Q_b)

i) Formation of steam

A part of heat is utilized in steam formation.

Heat utilized in formation of steam = Q_b = ma (h - h_w)

Where

m_a - actual evaporation kg/kg of fuel

h   -specific enthalpy of steam

h_w -specific enthalpy of feed water

Percentage of heat utilized in steam formation or

Boiler efficiency = $\frac{Qb}{Qs}$ X 100   = $\frac{ma(h-hw)}{C.V.}$ X 100

1) Heat lost in dry flue gases (Q_g)

Heat lost in dry flue gases

$Q_g = m_g\, C_{pg}\, (t_g - t_a)$ kJ / kg of fuel

m_g - mass of flue gases in kg /kg of fuel

C_{pg} -specific heat dry flue gases

t_g - temperature of the flue gases in °C

t_a -boiler room temperature or temperature of ambient air in °C

% of heat lost due to flue gases = $\frac{Qg}{Qs}$ X 100   = $\frac{Qg}{C.V.}$ X 100

Note: if m_g is given in kg/hr, then

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$$M_g = \frac{mg \text{ in kg per hr}}{mf \text{ in kg per hr}} = \frac{mg}{mf} \; kg/kg \; of \; fuel$$

Where m_f - mass of fuel used in kg / hr

2) Heat lost due to unburnt coal (Q_u)

The heat lost due to unburntcoal $= Q_u = m_u \, CV_u$ kJ/kg of fuel

m_u - mass of unburnt coal in kg / kg of fuel

CV_u -calorific value of unburnt coal in KJ/kg

% of heat lost due to unburnt coal $= \frac{Qu}{Qs} \times 100 \;\; = \frac{Qu}{C.V} \times 100$

Note:

If m_u is given in kg/hr, then

$$m_u = = \frac{Mu \text{ in kg per hr}}{Mf \text{ in kg per hr}} = \frac{mu}{mf} \; kg/kg \; of \; fuel$$

Where m_f -mass of fuel used in kg / hr

3) Heat lost due to moisture present in the fuel (Q_m):

The moisture present in the fuel is converted into superheated steam during the combustion of the fuel. The amount of heat required to convert the moisture in the fuel into superheated steam is not available for steam generation and hence it is a loss. It is assumed that the moisture in the fuel is converted into, superheated steam at atmospheric pressure of 1.01325 bar and temperature 100 ° C from boiler room temperature.

Heat lost due to moisture in the fuel

$Q_m = m \, (h_{sup} - h_a) = m[(2676 + c_p \, (t_g - 100) - h_a]$

% of heat lost due to moisture = $\frac{Qm}{Qs} \times 100 \;\; = \frac{Qm}{C.V} \times 100$

Where

m   - mass of moisture in the fuel I kg/kg of fuel

C_p - specific heat of steam

t_g - temperature of flue gas leaving the chimney in ° C

4) Heat lost due to hydrogen present in the fuel:

Let H_2 be the mass of hydrogen present in the fuel in kg/kg of fuel, then the mass of steam

formed = $9 H_2$

Heat lost due to hydrogen present

$Q_{H2} = 9 H_2 [2676 + c_p (t_g - 100) - h_a]$

% of heat lost due to H_2 present in the fuel $= = \dfrac{QH2}{Qs}$ x 100 $= \dfrac{QH2}{C.V}$X100

C_p -specific heat of steam

h_a -specific enthalpy of water at boiler room temperature

c) **Heat carried away by excess air Q** air

     This is the heat loss due to excess air supplied for burning of the fuel. It is denoted by

     Q air

     Q air$=m_{air}.C_{pa} (t_{g-t_a})$ kg /kg of fuel

M_{air} - mass of excess air per kg of fuel

C_{pa} -specific heat of air

t_g -Temperature of flue gases in ° C

t_a -Room temperature in ° C

% of heat lost due to excess air $= \dfrac{Qair}{Qs}$ x 100 $= \dfrac{Qair}{C.V}$ X 100

d) **Heat lost due to unaccounted or Heat lost due to external radiation (Q_r)**

     There is no direct method for finding the heat lost due to radiation. This loss is

calculated by subtracting the heat utilized in raising steam and heat losses from the heat

supplied by the fuel. It is also known as unaccounted heal losses.

$$Q_r = Q_s - [Q_g + Q_u + Q_m + Q_{H2} + Q_{air} + ...]$$

Heat balance sheet or energy balance of a boiler:

A heat balance sheet of a boiler is a statement that shows the distribution of heat through the boiler plant. It indicates all losses due to flue gases, excess air, moisture and hydrogen in the fuel. Unburnt fuel, external radiation etc. the typical heat balance sheet for a boiler trail per kg of fuel will be shown as below

Heat supplied	KJ	%	Heat consumed	KJ	%
Heat supplied by one kg of fuel (C.V.)		100	Heat utilized in raising steam		
			Heat lost in dry fuel gases		
			Heat lost due to unburnt fuel		
			Heat lost in evaporation the moisture in the fuel		
			Heat lost due to hydrogen present in the fuel		
			Heat lost due to excess air		
			Heat lost due to unaccounted		
	CV	100		CV	100

Problems

1.A boiler evaporates 8.2 kg of water per kg of coal fired from feed water at 40°C when working at 10 bar abs. Determine the equivalent evaporation, from and at 100°C per kg coal fired, (i) if the steam produced is 0.95 dry, (ii) If the steam produced is dry saturated, (iii) if the steam produced is superheated to 240°C.

Given data:-

$m_a = 8.2$;

$p = 10$ bar abs;

$x = 0.95$ dry;

$t_{sup} = 240°C.$

To find:

i) $m_{e\,wet}$

ii) $m_{e\,dry}$

iii) $m_{e\,sup}$

Solution:-

At 10 bar the following information is written down from the steam tables,

$h_g = 2776.2$ KJ/kg; $h_f = 762.61$ KJ/kg;

$h_{fg} = 2013.6$ KJ/kg; $h_{sup} = 2920.6$ KJ/kg.

Therefore,

(i) $m_e = \dfrac{ma\ (hwet - hw)}{2257} = \dfrac{(hf + x\ hfg) - hw}{2257} \times m_a$

$= \dfrac{(762.1 - 0.95 \times 2013.6) - 40 \times 4.1868}{2257} \times 8.2 = \dfrac{2508}{2257} \times 8.2$

$= 9.112\ kg\ per\ kg\ of\ coal\ fired.$

(ii) $m_e = \dfrac{hg - 40 \times 4.1868}{2257} \times m_a = \dfrac{2776.2 - 40 \times 4.1868}{2257} \times 8.2$

$= 9.478$ kg per kg of coal fired.

(iii) $m_e = \dfrac{hsup - 40 \times 4.1868}{2257} \times 8.2 = \dfrac{2920.6 - 40 \times 4.1868}{2257} \times 8.2$

$= 10$ kg per kg of coal fired.

2. A boiler evaporates 3.6 kg of water per kg of coal into dry saturated steam at 10 bar. The temperature of feed water is 32 °C. Find the equivalent evaporation "from and at 100° C" as well as the factor of evaporation.

Given:

ma = 3.6 kg/kg of coal

p = 10 bar

t_1 = 32 °C

To find:

1. Equivalent evaporation "from and at 100° C".

2. Factor of evaporation.

Solution:

Equivalent evaporation "from and at 100° C":

From steam tables, corresponding to a feed water temperature of 32 °C, we find that

$$h_{f1 =} 134 \text{ kJ/kg}$$

and corresponding to a steam pressure of 10 bar, we find that

$$h = h_g = 2776.2 \text{ kJ/kg} \qquad \text{… (For dry saturated steam)}$$

We know that equivalent evaporation 'from and at 100° C'.

$$m_e = \frac{m_a \ (h - h_w)}{2257} = \frac{3.6 \ (2776.2 - 134)}{2257} = \textbf{4.2 kg/kg of coal. Ans.}$$

Factor of evaporation:

We know that factor of evaporation

$$= \frac{(h - h_f)}{2257} = \frac{2276.2 - 134}{2257} = \textbf{1.17 Ans.}$$

3. The following observations were made in a boiler trial:

Coal used 250 kg of calorific value 29800kJ/kg, water evaporated 2000kg, steam pressure 11.5 bar, dryness fraction of steam 0.95 and feed water temperature 34° C.

Calculate the equivalent evaporation "from and at 100° C" per kg of coal and efficiency of the boiler.

Given:

$$m_f = 250 \text{ kg}$$

$$C = 29800 \text{ kJ/kg}$$

$$m_s = 2000 \text{ kg}$$

$p = 11.5$ bar

$x = 0.95$

$t_1 = 34°$ C

To find:

1. Equivalent evaporation "from and at 100° C".

2. Efficiency of the boiler.

Solution:

Equivalent evaporation 'from and at 100° C':

From steam tables, corresponding to a feed water temperature of 34° C, we find that

$$h_{f1} = 142.2 \text{ kJ/kg}$$

and corresponding to a steam pressure of 11.5 bar, we find that

$$h_f = 790 \text{ kJ/kg}; \ h_{fg} = 1991.4 \text{ kJ/kg}$$

We know that enthalpy or total heat of steam,

$$h = h_f + x \, h_{fg} = 790 + (0.95 \times 1991.4) = 2681.8 \text{ kJ/kg}$$

and mass of water evaporated per kg of coal

$$\text{ma} = \frac{m_s}{m_f} = \frac{2000}{250} = 8 \text{ kg/kg of coal.}$$

∴ Equivalent evaporation 'from and at 100° C'

$$m_e = \frac{\text{ma} \, (h - h_{f1})}{2257} = \frac{8 \, (2681.8 - 142.4)}{2257} = \textbf{9 kg/kg of coal. Ans.}$$

Efficiency of the boiler:

We know that efficiency of the boiler,

$$\eta = \frac{\text{ma} \, (h - h_f)}{CV} = \frac{8 \, (2681.8 - 142.4)}{29800} = \textbf{0.682} \ \textbf{or} \ \textbf{68.2 \% Ans.}$$

4.A boiler generates 8.5 kg of steam per kg of coal burnt at a pressure of 12.5 bar gauge from feed water having absolute tempt of 350 K. The efficiency of the boiler is 70 % and factor of evaporation 1.17, specific heat of super heated steam at constant pressure is 2.2 kJ/kg K. Calculate

i) Degree of super heat

ii C.V. of coal in kJ/kg

iii) Equivalent evaporation from and at 100 2592.03 -117.3)/ per kg of coal

Given data

Actual Evaporation m_a	= 8.5 kg/kg of coal.
Pressure of steam P	= 12.5 bar gauge
Feed water tempt t_f	= 3050 K = 77 °C
Efficiency of the boiler	= 70 %
F.O. Evaporation F	= 1.17
Specific Heat of S.H. steam Cp	= 2.2 KJ/Kg.k

To find

i) Degree of super heat

ii C.V. of coal in kJ/kg

iii) Equivalent evaporation from and at 100 2592.03 -117.3)/ per kg of coal

Solution

Specific enthalpy of feed water at 77 °C , h_w = 322.2 kJ/kg

Assuming atmospheric pressure is 1 bar , then

Absolute pressure of steam = 12.5 + 1 = 13.5 bar abs

But, Factor of evaporation = 1.17

$(h - h_w) / 2257 = 1.17$

$(h - 322.3) / 2257 = 1.17$

h - 322.3 = 2640.69

h = 2962.99

From steam tables at 13.5 bar abs

Specific enthalpy of dry steam h_g = 2786.7 KJ/kg

Saturation temperature t_s = 193.3° C

Specific enthalpy of S.H steam at 13.5 bar

h= h_g + C_{ps} (t_{sup} – t_s) = 2786.7 +2.2 x degree of super heat

2962.99 = 2786.7 + 2.2 x degree of super heat

2.2 x degree of super heat = 2962.99 – 2786.7 = 176.29

Degree of super heat = 179.29 / 2.2 = 80.132 °C

Let C.V be the calorific value of the fuel, then the boiler efficiency

$ma \frac{h-hw}{C.v}$ = 0.7

8.5 x 2640.69 / C.V = 0.7

C.V =8.5 x 2640.69/ 0.7

= 32065.52 kJ/kg

Equivalent evaporation from and at 100 ° C per kg of coal

= ma x factor of evaporation

= 8.5 x 1.17

= 9.945 kg

5.The feed water to a boiler enters an economiser at 32°C and leaves at 120 °C being fed into the boiler at this temperature, the steam leaves the boiler 0.95 dry at 20 bar and passes through a superheater where its temperature is raised to 250 °C without change of pressure. The steam output is 8.2 kg/kg of coal burned and calorific value of the coal is 28000 kJ/kg. Calculate the energy picked up per kilogram of water and steam (a) Economiser (b) the boiler (c) the superheater percentage of the energy supplied by the coal.

Take specific heat capacity of superheated steam = 2.093kJ/kg.K

Given

Economiser inlet temperature $t_1 = 32°C$

Economiser outlet temperature $t_2 = 120 °C$

$x = 0.95 \ dry$; $p = 20$ bar $t_{sup} = 250°C$

Steam / kg $= \dfrac{ms}{mf} = m_a = 8.2$ kg/ kg

To find

Heat in economiser

Heat in boiler

Heat in superheater

Solution

From steam stables at Temperature 30°C & 120 °C

$h_{w1} = 134$ Kj

$h_{w2} = 503.$ Kj

Heat transfer in the economiser / kg $= m_a \, c_{pw} \, (h_{w2} - h_{w1})$

$$= 8.2 \times 4.18 \, (120 - 32) = \underline{3016.288 \text{ Kj}}$$

Heat transfer in boiler $= m_a \, (h_{wet} - h_w)$

$$= m_a (h_f + x._{hfg} - h_w)$$

$$= 8.2 \times (908.6 + 0.95 \times 1888.7 - 503.7) = \underline{18033.153 \text{ Kj}}$$

Heat transfer in super heater $= m_a \, (h_{sup} - h_{wet})$

$$= m_a \, (h_g + c_{ps} \, (t_{sup} - t_s) - h_{wet})$$

$$8.2 \times (2797.2 + 2.093 \, (250 - 212.4) - 2702.865 = \underline{1418.846 \text{ Kj}}$$

Percentage of heat taken by economiser

$$= \frac{Heat \ transfered}{CV}$$

$$= \frac{3016.288}{28000} \text{ x } 100 = 10.77 \ \%$$

Percentage of heat taken by super heater

$$= \frac{1418.846}{28000} \text{ x } 100 = 5.06 \ \%$$

Percentage of heat taken by boiler

$$= \frac{18033.153}{28000} \times 100 = 64.4\ \%$$

6. The following data's are provided for two boilers

	Steam pressure	Quality of steam	Evaporation rate
Boiler 1	8 bar	0.9	8.5 kg/kg of coal
Boiler 2	20 bar	Superheated to 300 ° C	7 kg / kg of coal

Both the boilers are supplied with feed water at 40 °C. Determine which has the comparatively higher utilization rate per kg of coal fired and which has higher thermal efficiency.

Given data

Boiler 1

Dryness fraction x = 0.9

Feed water tempt t_f =40 °C

Actual evaporation m_a = 8.5 kg/kg of coal fired

From steam tables at pressure 8 bar

Specific enthalpy of water h_f = 720.9 kJ/kg

Specific enthalpy of evaporation h_{fg}=2046.5 kJ/kg

Specific enthalpy of wet steam at 8 bar and 0.9 dry

h= $h_f + x$ .h_{fg} 720.0 +0.9x2046.5   = 2562.75 KJ/kg

Specific enthalpy of water at 40 ° C, h_w = 167.5 KJ/kg

Equivalent evaporation from and at 100 °C

$m_e1 = m_a$ (h - h_w) / 2257   = 8.5 (2562.75 - 167.5) / 2257

                       = 9.021 kg/kg of coal fired.

Boiler 2

Feed water tempt t_f =40 °C

Actual evaporation m_a= 8.5 kg/kg of coal fired

From steam table, specific enthalpy of super heated steam at 20 bar and 300 °C

h= 3025 KJ/kg

Equivalent evaporation from and at 100 °C

m_e2= m_a (h - h_w) / 2257   = 7 (3025-167.5) / 2257

                                   =8.86 kg/kg of coal fired

$m_e1>m_e2$, therefore the boiler 1 has better utilisation rate than boiler 2.

Boiler efficiency is given by

$$= \frac{ma(h-hw)}{C.V.} X\ 100$$

For a constant calorific value

The value of m_a (h - h_w) for first boiler   = 20359.625 kJ/kg

The value of m_a (h - h_w) for second boiler   =20002.5 kJ/kg

The value of m_a (h - h_w) is more for the boiler 1.

Therefore the boiler 1 has higher thermal efficiency than boiler 2.

7.In a boiler, the following observations were made:

Pressure of steam	=10bar
Steam condensed	=540kg/hr
Fuel used	=65kg/hr
Moisture in fuel	=2% by mass
Mass of dry flue gases	=9 kg/kg of fuel
Lower calorific value of fuel	=32000kJ/Kg
Temperature of the flue gases	=325^{0}C

Temperature of boiler house $=28^0C$

Feed water temperature $=50^0C$

Mean specific heat of flue gases $=1KJ/kg\ k$

Dryness fraction of steam $=0.95$

Draw up a heat balance sheet for the boiler.

Given:

P=10 bar, m=540kg/h, m_f=65kg/h, m_m= 0.02 kg/kg of fuel, m_g=9kg/kg of fuel, C =32000

kJ/kg, t_g =325^{0}C, t_a =28^{0}C, t_1= 50^{0}C, c_{pg} =1KJ/kg k, $x = 0.95$

First of all, let us find the heat supplied by 1 kg of fuel. Since the moisture in fuel is

0.02kg, therefore heat supplied by 1 kg of fuel

$$(1- 0.02)\ 32000 =31600\ kJ$$

1. **Heat utilized in raising steam per kg of fuel**

   We know that the mass of water actually evaporated per kg of fuel,

   $$m_a= m_S/m_f = 540/65 =8.31\ kg$$

   From steam table, corresponding to a feed water temperature of 50^{0}c, we find that

   $$h_{f1}=209.3\ KJ/kg$$

   And corresponding to a steam pressure of 10 bar, we find that

   $$h_f= 762.6\ kJ/Kg;\ h_{fg}= 2013.6\ kJ/kg$$

   Heat utilized in raising steam per kg of fuel

   $$=m_a\ (h-h_{f1}) =m_a\ (h_f+ x\ h_{fg}-h_{f1})$$

   $$=8.31\ (762.6+0.95\times2013.6 - 209.3) = 20495\ kJ$$

2. **Heat carried away by dry flue gas**

   We know that heat carried away by dry flue gas

   $$= m_g\ c_{pg}\ (t_g - t_a) = 9\times1(325 - 28)\ 2673\ kJ$$

3. **Heat carried away by moisture in fuel per kg of fuel**

From steam tables, corresponding to a temperature of 28^0c, we find that

ha = 117.3kJ/kg

We know that heat carried away by moisture in fuel

$$= m_m[(2676 + c_p (t_g - 100) - h_a]$$

$$= 0.02 [2676+2.1 (325-100)-117.3] \qquad (cp=2.1 \ kJ/kg \ k)$$

$$= 60.6 \ kJ$$

4. Heat lost by radiation etc.

We know that heat lost by radiation etc. (by difference)

$$=31360 - (20495+2673+60.6) = 8131.4kJ$$

Now complete heat balance sheet per kg of fuel is given below:

Heat supplied	kJ	Heat expenditure	kJ	%
Heat supplied by 1 kg of fuel	31360	1. Heat utilized in raising steam	20495	65.35
		2. Heat carried away by dry flue gases	2673	8.53
		3. Heat carried away by moisture in fuel	60.6	0.19
		4. Heat lost by radiation etc. (by difference)	8134.4	25.93
Total	31360	Total	31360	100