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THERMAL ENGINEERING - II

Rakesh Kumar Mahanta

-: Testing & Performance of I.C Engine :-

An Internal combustion Engine is a machine that converts chemical energy (fuel) into mechanical energy. The combustion of fuel takes place inside a combustion chamber (cylinder), releases its chemical energy in the form of heat, which is converted into mechanical energy with the help of a reciprocating piston & crank mechanism.

An engine is selected for a particular application on the basis of its power output and rated speed. Other factors include capital cost and operational cost. Therefore, certain measurements and calculations are required to judge the performance of an I.C Engine. These are

(1) Indicated power
(2) Brake Power
(3) Frictional Power
(4) Fuel Consumption.
(5) Air consumption.
(6) Brake thermal Efficiency
(7) Indicated thermal efficiency.
(8) Mechanical Efficiency.
(9) Air fuel ratio

Indicated Power (I.P) :-

(It is the power devloped inside the engine cylinder) It is defined as the rate of workdone on the piston by burning of charge inside the cylinder. It is evaluated from an indicated diagram obtained from an engine.

$$\boxed{\text{Indicated Power (I.P)} = \frac{P_m L A n K}{60000} \text{ K.W}}$$

where, P_m = mean effective pressure in N/m^2
A = cross-sectional area of piston in m^2
L = stroke length in m
n = no. of working stroke made by the piston per minute
K = no. of cylinders.

If N = No. of revolution per minutes (rpm) of the engine

then $n = N$ (for 2-stroke engine, as power stroke in every revolution)

$= \frac{N}{2}$ (for 4-stroke engine, as power stroke in every 2 revolution of crank shaft)

In one working stroke, work done in the cylinder

Work done $= p_m \times L \times A$ Joule

Work done per minute $= p_m \times A \times L \times n$ Joule/min

Work done per second (Power) $= \frac{p_m \times A \times L \times n}{60}$ Joule/sec (Watt)

$$\boxed{I.P = \frac{p_m A L n k}{60000} \text{ K.W}}$$

k = no. of cylinders = 1, 2, 4, 6, 8, 12, 16

Actual Mean Effective Pressure

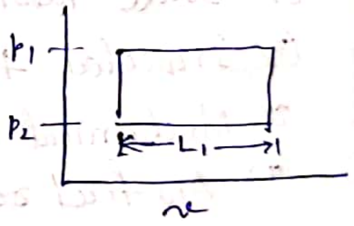
p_m :- Mean effective pressure can be measured from the indicator diagram.

$$p_m = \frac{\text{Area of the indicator diagram}}{\text{Base or Length of indicator diagram}} \times \text{spring scale}$$

$$= \frac{A_1}{L_1} \times S_1$$

$A_1 = (p_1 - p_2) L_1$

$A_1 = p_m \times L_1$



Theoritical mean effective pressure in otto cycle

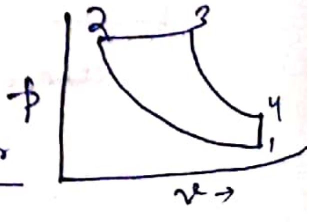
$$p_m = \frac{\text{Net work done during cycle}}{\text{stroke volume}} = \frac{\text{Area } 1\text{-}2\text{-}3\text{-}4\text{-}1}{V_1 - V_2}$$

$$= \frac{\frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1}}{V_1 - V_2} = \frac{(p_3 V_3 - p_4 V_4) - (p_2 V_2 - p_1 V_1)}{(\gamma - 1)(V_1 - V_2)}$$

$$\boxed{p_m = \frac{(p_3 V_3 - p_4 V_4) - (p_2 V_2 - p_1 V_1)}{(\gamma - 1)(V_1 - V_2)}}$$

Theortical mean effective pressure in Diesel cycle

$$p_m = \frac{\text{Area } 1\text{-}2\text{-}3\text{-}4\text{-}1}{V_1 - V_2} = \frac{\text{Area under 2-3} + \text{Area under 3-4} - \text{Area under 2-1}}{V_1 - V_2}$$



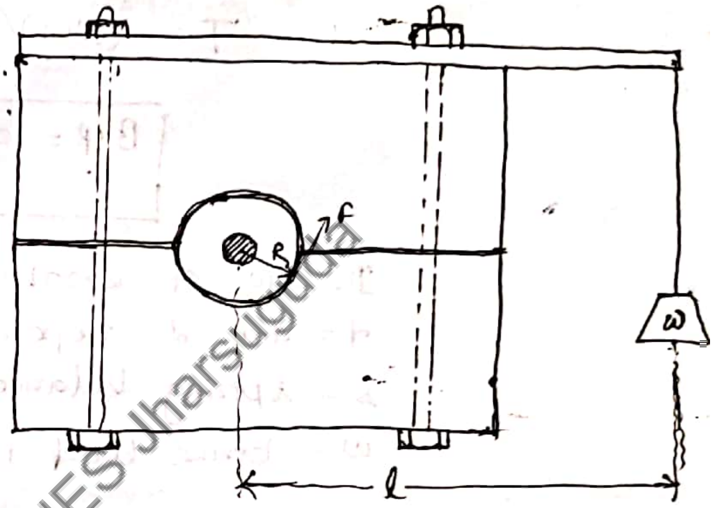
$$\boxed{p_m = \frac{p_2 (V_3 - V_2) + \frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1}}{V_1 - V_2}}$$

Brake Power (B.P) :-

The net power available at the crank shaft for external use is called brake power. If it is expressed in H.P then it is called brake horse power (B.H.P). Brake power is measured by dynamometer. Dynamometers are (i) Prony Brake dynamometer
(ii) Rope Brake dynamometer.

Prony Brake Dynamometer :-

Torque (T) = ωL



Frictional torque = torque due to available power.

i.e $F \times R = \omega \times l$

F = frictional resistance
R = Radius of pulley.
ω = load applied
l = length of brake arm

Now, work lost in friction per revolution $= F \times 2\pi R$ Joule/rev.
If N = rpm of the pulley then, work lost in one minute

$$= F \times 2\pi R \times N \quad \frac{\text{Joule}}{\text{rev}} \times \frac{\text{rev}}{\text{min}}$$

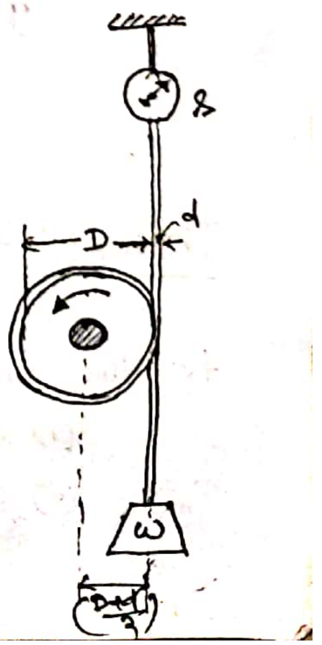
So Brake Power = work lost per second

$$= \frac{F \times 2\pi R N}{60} \quad \frac{\text{Joule}}{\text{sec.}}$$

$$= \frac{\omega l}{R} \times \frac{2\pi R N}{60} \quad \text{Watt}$$

$$\boxed{B.P = \frac{2\pi N T}{60000} \text{ kWatt}} \quad (\because T = \omega \times l)$$

Rope Brake Dynamometer :.



Total effective braking load = (ω − S) N

This effective load act at the mean radius $= \frac{D+d}{2}$

Work done (lost) against the motion of the crankshaft by the effective load $= 2\pi(\omega - S) \times \frac{(D+d)}{2}$ $\frac{\text{Joule}}{\text{rev.}}$

Work lost in one minute $= \pi(W-S)(D+d)N \ \frac{\text{Joule}}{\text{min}}$

where N = no. of revolution per minute of crank shaft

So Brake power = Work lost per second

$$= \frac{\pi(W-S)(D+d)N}{60} \ \frac{\text{Joule}}{\text{sec}} \ (\text{Watt})$$

$$\boxed{B.P = \frac{\pi(W-S)(D+d)N}{60000} \ \text{K.Watt}}$$

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

$$\boxed{B.P = \frac{2\pi N T}{60000} \ \text{KWatt}}$$

D = dia. of brake drum. in m.
d = dia. of rope. in m.
S = spring balance reading
W = brake load in N.

Fractional Power (F.P)

When the power is transmitted from the engine cylinder to the crank shaft some power is lost due to friction in the mechanical parts within the engine. Such power loss is called frictional power.

$$\boxed{F.P = I.P - B.P}$$

Mechanical Efficiency

It is the ratio of Brake power to the Indicated power

$$\eta_{mech} = \frac{B.P}{I.P}.$$

Thermal Efficiency :-

It is the ratio of power developed by the engine to the heat supplied. It may be

(i) Indicated thermal Efficiency
(ii) Brake thermal Efficiency.

Indicated thermal Efficiency:- Indicated thermal efficiency is based on Indicated Power. It is the ratio of Indicated power to the heat supplied.

I.P = indicated power in KWatt = $IP \times 3600 \ \frac{kJ}{hr}$

Heat supplied = $\dot{m}_f \times CV \quad \frac{KJ}{hr}$

$\dot{m}_f$ = mass of fuel consumed per hours in $\frac{Kg}{hr}$

C.V = calorific value of fuel in $\frac{KJ}{Kg}$

C.V :- It is the amount of heat liberated by combustion of one kg of fuel.

$$\boxed{\eta_{Indicated} = \frac{I.P \times 3600}{\dot{m}_f \times C.V}}$$

$$= \frac{3600}{\left(\frac{\dot{m}_f}{I.P}\right) \times Cv}$$

$\frac{\dot{m}_f}{I.P}$ is called specific fuel consumption per KWhr on indicated power basis.

$$\frac{\dot{m}_f}{I.P} \left(\frac{Kg}{KWhr}\right)$$

It is also known as Indicated specific fuel consumption (I.S.F.C)

Brake thermal Efficiency:- Brake thermal efficiency is based on Brake power. It is the ratio of Brake power to the heat supplied.

$$\boxed{\eta_{Brake} = \frac{B.P \times 3600}{\dot{m}_f \times CV}}$$

$$= \frac{3600}{\left(\frac{\dot{m}_f}{BP}\right) \times CV}$$

$\frac{\dot{m}_f}{B.P}$ is called Brake specific fuel consumption (B.S.F.C) in $\left(\frac{Kg}{KWhr}\right)$

Specific fuel consumption (sfc) :-

It is defined as the ratio of mass of the fuel consumed per hour per unit power out put. It is a parameter which decides the economical power production from an engine.

indicated specific fuel consumption (isfc) :- The specific fuel consumption based on Indicated power is called indicated specific fuel consumption (isfc). It is the ratio of mass of fuel consumed ($\dot{m}_f$) in Kg/hr to the Indicated power (I.P) in KW.

$$isfc = \frac{\dot{m}_f}{I.P} \frac{(Kg/hr)}{K.W} = \frac{\dot{m}_f}{I.P} (Kg/KWhr)$$

Brake specific fuel consumption (bsfc) :- The specific fuel consumption based on Brake power is called brake specific fuel consumption (bsfc). It is the ratio of mass of the fuel consumed ($\dot{m}_f$) in Kg/hr to the Brake power (B.P) in K.W

$$bsfc = \frac{\dot{m}_f}{BP} \frac{(Kg/hr)}{KW} = \frac{\dot{m}_f}{BP} \; Kg/KWhr$$

Fuel consumption

$$\dot{m}_f = \frac{V_{fuel} \times \rho_{fuel} \times 3600}{\Delta t} \; (Kg/hr)$$

where, V_{fuel} = Volume of fuel in m^3 used in time Δt.
ρ_{fuel} = Density of fuel
Δt = time in second.
$\dot{m}_f$ = mass of fuel consumed in (Kg/hr)

Air Consumption :-

Air consumption rate of an engine can be calculated by means of an orifice meter installed in air box

$$\boxed{\dot m_a = \rho_{air} \times \dot V_a}$$

$\dot m_a$: mass flow rate of air

ρ_{air} : density of air

$\dot V_a$ = Volume flow rate of air in m^3/s = $\frac{\pi}{4} d_o^2 C_d \sqrt{2gh_a}$.

d_o = dia of orifice
C_d = co-efficient of discharge
h_a - head of air .

Air fuel Ratio (A/F)

It is the ratio between the mass of air and mass of fuel supplied to the engine. It is expressed as

$$A/F = \frac{\dot m_a \text{ (mass flow rate of air)}}{m_f \text{ (mass flow rate of fuel)}}.$$

→ Theoritically correct (stoichiometric) air fuel ratio is 15. But combustion of air fuel mixture can takes place in A/F ratio ranges from 12 to 19 for petrol and 20 to 60 in diesel engine .

Relative Efficiency :-

The efficiency obtained from theoritical cycle is called air standard efficiency & the efficiency obtained during the trial of actual engine is called thermal efficiency. The ratio of thermal efficiency to the air standard efficiency is called relative efficiency.

→ Relative efficiency for most of the engines varies from 75 to 95%.

$$\eta_{relative} = \frac{\text{Brake thermal efficiency}}{\text{air standard efficiency}} \quad \text{(based on Brake power)}$$

$$= \frac{\text{Indicated thermal efficiency}}{\text{air standard efficiency}} \quad \text{(based on Indicated power)}$$

It is defined as the ratio of the mass of the actual charge inducted into the cylinder to the mass of the charge corresponding to the swept volume

$$\eta_{vol} = \frac{\text{Actual mass flow rate of the charge}}{\text{Density} \times \text{Swept volume per second.}}$$

$$= \frac{\dot{m}_a \,(kg/s)}{\rho_a\left(\frac{\pi}{4}d^2 L\right)\frac{n}{60}}$$

where, ρ_a = Density of inlet charge

d = bore , L = stroke

n = no. of effective suction strokes per cycle per minute

$= \frac{N}{2}$ (4-S engine)

$= N$ (2-S engine)

→ The volumetric efficiency can also be defined as the ratio of the volume of the charge inducted in the cylinder, measured at NTP to the swept volume ~~of the~~ cylinder.

$$\eta_{vol} = \frac{V_{act}}{V_s}$$

Problem (24.18 M. Rathore)

The following data and results refers to a test on a single cylin
two-stroke cycle engine

indicated mean effective pr = 550 KPa.

cylinder diameter = 21 cm

piston stroke = 28 cm

engine speed = 360 rpm

brake torque = 628 N-m

fuel consumption = 8.16 kg/hr

Calorific value of fuel = 42700 kJ/kg

Calculate

AIR COMPRESSOR (Reciprocating)

An air compressor is a machine which takes in atmospheric air, compresses it with the help of some mechanical energy and delivers it at higher pressure. It is also called air pump. An air compressor increases pressure of air by decreasing the specific volume of the using mechanical mean.

Uses of compressed air:-

Compressed air has wide applications in industries as well as in commercial equipments. It is used in

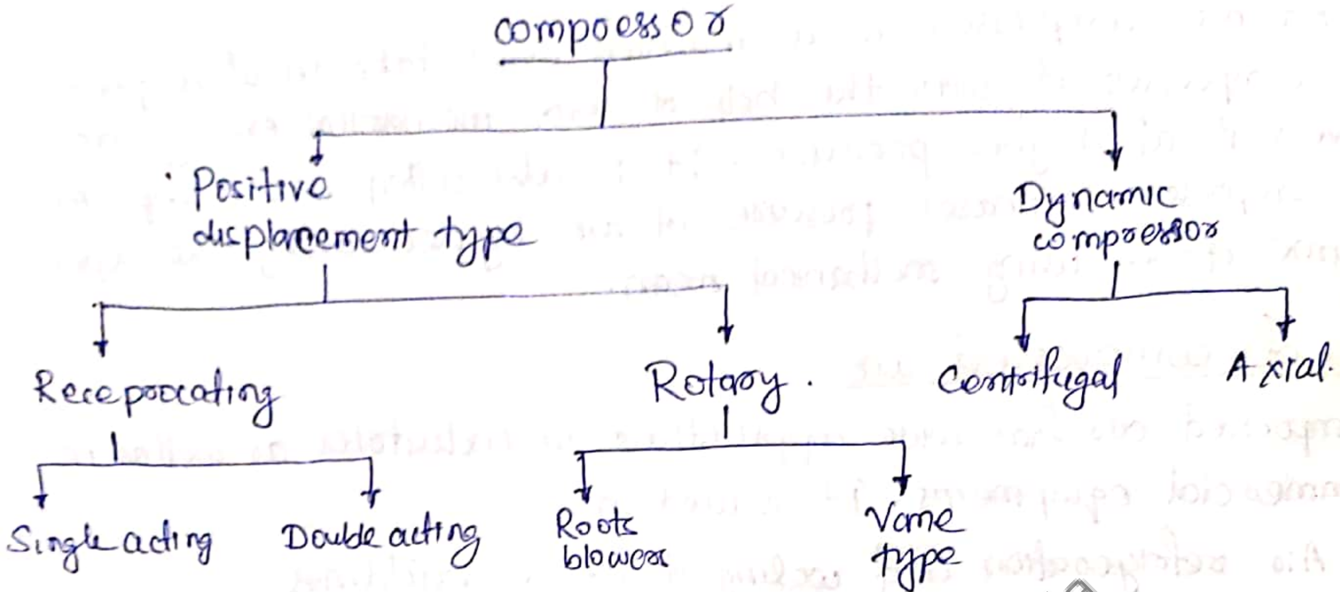
1. Air refrigeration and cooling of large buildings.
2. Driving pneumatic tools in shops like drills, rivetters, screw drivers.
3. Driving air motors in mines, where electric motors and IC engines can not be used because of fire risks due to the presence of inflamable gases.
4. Cleaning purposes.
5. Blast furnaces.
6. Spray painting and spraying fuel in diesel engines.
7. Hard excavation work, tunneling, ~~boring~~ boring, mining etc.
8. Starting heavy duty diesel engines.
9. Operating heavy duty air brakes in buses, trucks and trains etc.
10. Inflating automobile and aircraft tyres.
11. Supercharging Internal combustion engines.
12. Conveying solid and powder materials in pipe lines.
13. Process industries
14. Operating lifts hoists crane and to operate pumps etc.
15. Pump sets for oil and gas transmission line.
16. Automobile suspension system.

Classification of compressor:

compressors are mainly classified as

(i) reciprocating air compressor (ii) Rotary air compressor

Air compressor can be broadly clasified as



→ A reciprocating air compressor is used to produce high pressure gas. It uses a piston cylinder arrangement for compression of air. It handles a low mass of gas and a high pressure ratio

→ The rotary compressor are used for low and medium pressures. They usually consist of bladed wheel or impeller that pins inside a circular housing. They handle a large mass of gas.

Reciprocating Air compressor Terminology :-

① Single-Acting compressor :- In ~~sp~~ single acting compressor suction, compression and delivery of a gas takes place only in one side of the piston during a cycle of one revolution of crank shaft.

② Double acting compressor :- is a compressor in which suction, compression and delivery of gas takes place on both sides of the piston and two cycles take place during one revolution of crank shaft.

③ Single stage compressor :- is a compressor in which the compression of gas to final delivery pressure is carried out in one cylinder only.

④ Multi-stage compressor :- is a compressor in which the compression of gas to the final pressure is carried out in more than one cylinder in series.

5. **Pressure ratio** :- It is defined as the ratio of absolute discharge pressure to the absolute suction pressure

6. **Free air** :- Free air is the air that exists under atmospheric condition.

7. **Compression displacement volume (swept volume)** :- It is the volume when piston travels a stroke. It is given as

$$V = \frac{\pi}{4} d^2 L$$

d = bore
L = stroke

8. **Induction volume rate or volume flow rate** :- Volume flow rate into the compressor is expressed in m^3/s and is given as

$\dot{V}$ = Volume inducted per cycle × No. of suction per ~~volume~~ revolution × rpm.

$$\dot{V} = \frac{\pi}{4} \times d^2 \times L \times \frac{N}{60} \quad \text{(single acting)}$$

$$= \frac{\pi}{4} \times d^2 \times L \times \left(\frac{2N}{60}\right) \quad \text{(double acting)}$$

9. **Capacity of a compressor** :- It is the actual quantity of air delivered per unit time at atmospheric condition.

10. **Free air delivery (FAD)** :- It is the discharge volume of the compressor corresponding to ambient condition.

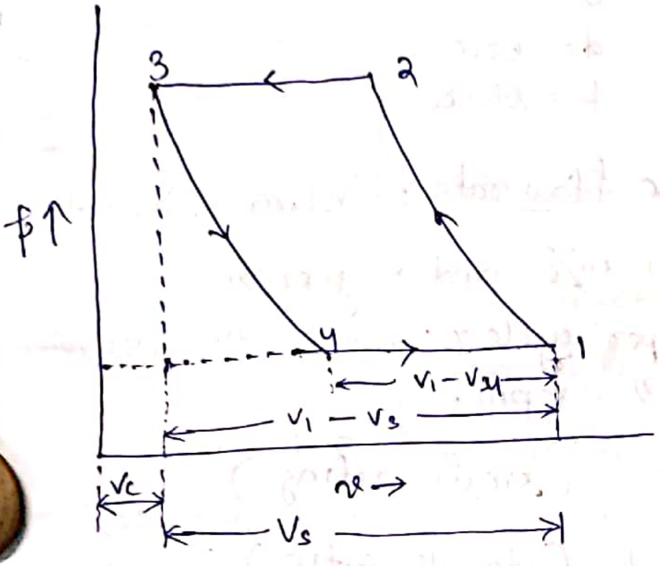
Free air delivers is the amount of compressed air delivered at the outlet of a compressor, convert back to the actual inlet (free air) condition before it was compressed. In other words, it is the amount of free air draw in at atmospheric condition by the compressor, compressed and delivered at a specific pressure.

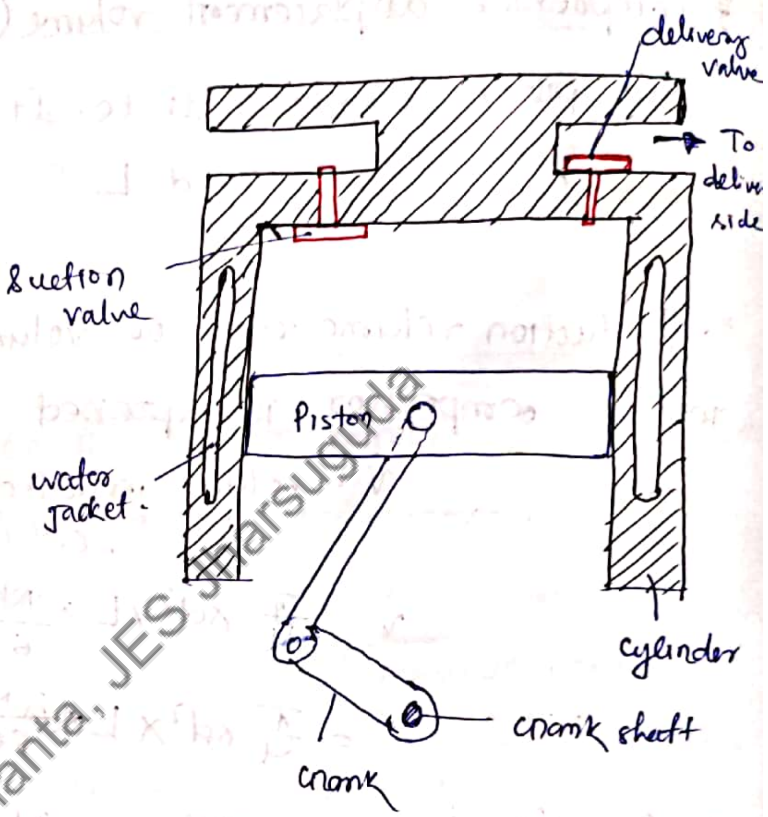
11. **Piston Speed** :- It is the linear speed of the piston measured in m/min. It is expressed as $V_{piston} = 2LN$

Reciprocating compressor :-

Reciprocating compressors are used to increase the pressure of gas, vapour, or steam by the reciprocating action of the piston inside the cylinder.

Working :-





→ The sectional ~~ve~~ view of a single single acting compressor is shown in the figure. It consists of a piston, cylinder with cooling arrange-ment, connecting rod, crank, inlet and delivery valve.

→ The crank shaft is connected to a prime mover shaft, which may be a electric motor or an I.C engine. When the prime mover runs, the crank shaft rotates and the piston reciprocates inside the cylinder

→ As the piston moves downward (from TDC to BDC) the pressure inside the cylinder decreases. When the pressure falls below the atmospheric pressure the suction valve opens and the free air from atmosphere enters into the cylinder till the piston reaches the BDC.

→ Now the piston starts moving from BDC to TDC which stops creating vaccum pressure and closes the inlet valve. The further upward movement of the piston increases the pressure of the gas gradually. When the pressure inside the cylinder exceeds the pressure on the delivery side, the delivery valve opens and the high pressure air is discharged to the receiver.

→ At the TDC position some high pressure air is left in the clearance space. This high pressure air expands as the piston moves downward. When the pressure inside the cylinder falls below the atmospheric pressure air is again inducted into the cylinder and the cycle is repeated.

p-V Diagram :-

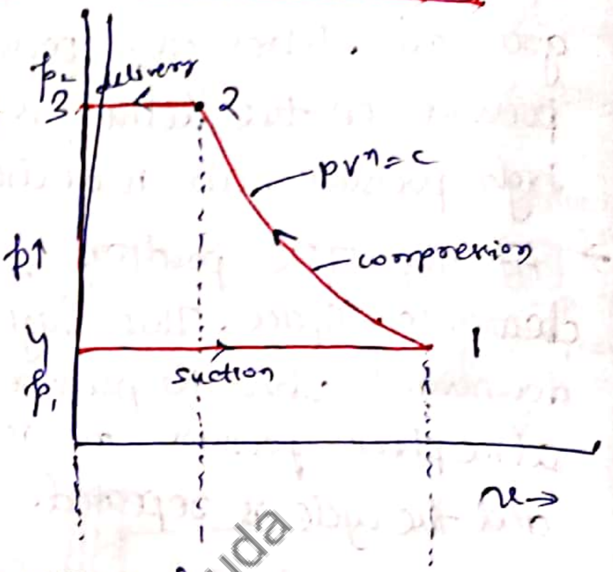
(1) Process 1-3 : represents the compression of air during which the pressure increases from suction pressure 'p_1' to delivery pressure 'p_2'

(ii) Process 2-3 represents the delivery of high pressure air at constant pressure 'p_2'.

(iii) Process 3-4 : Some high pressure air is left in the clearance space which expands as the piston moves downward. The expansion of the left gas is represented by the process 3-4. During this process the delivery pressure 'p_2' drops to inlet pressure 'p_1'.

(iv) Process 4-1 represents the suction of air into cylinder at pressure 'p_1'.

Work done Without Clearance (Indicated work for a single acting compressor without clearance)



The theoretical p-V diagram for a single stage single acting reciprocating air compressor without clearance is shown in the figure. The net work done in the cycle is equal to the area enclosed by the curve (cycle) on pv diagram.

Process 1-2 – represent polytropic compression of air during which pressure increases from p_1 to delivery pressure p_2

Process 2-3 – represents delivery of high pressure air at constant pressure p_2

Process 3-4 – represent pressure drop inside the ~~cylined~~ cylinder from delivery pr. p_2 to suction pressure p_1

Process- 4-1 - represents the suction of air at const. pr. p_1

Work done per cycle = Area under the cycle

= Area of 1-2-3-4-1

= Area under 1-2 + Area under 2-3 − Area under 4-1

$$= \frac{p_2 v_2 - p_1 v_1}{n-1} + p_2 v_2 - p_1 v_1$$

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

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

$$\boxed{W_{\text{per cycle}} = \frac{n}{n-1}\left(p_2 v_2 - p_1 v_1\right)}$$

$$W/_{cycle} = \frac{n}{n-1}(mRT_2 - T_1 mR)$$

$$\boxed{W_{per\,cycle} = \frac{n}{n-1} mR(T_2 - T_1)}$$

$$W = \frac{n}{n-1}(p_2v_2 - p_1v_1) \quad \text{for single action}$$

$$= 2 \times \frac{n}{n-1}(p_2v_2 - p_1v_1) \quad \text{for double action}$$

$$W = \frac{n}{n-1} p_1v_1\left[\frac{p_2}{p_1} \times \frac{v_2}{v_1} - 1\right]$$

We know that $p_1 v_1^n = p_2 v_2^n$

$$\Rightarrow p_2v_2^n = p_1v_1^n \Rightarrow \frac{v_2^n}{v_1^n} = \frac{p_1}{p_2}$$

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

So. $$W = \frac{n}{n-1} p_1v_1\left[\left(\frac{p_2}{p_1}\right) \times \left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}} - 1\right]$$

$$\boxed{W = \frac{n}{n-1} p_1v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}$$

$$\boxed{W = \frac{n}{n-1} mRT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}$$

v_1 = volume inducted

Power and Mechanical Efficiency

Indicated Power (I.P): The work done on air per unit time is called indicated power input to the compressor. The power required by an air compressor, running N rpm is given by

Indicated power (I.P) = Work input per cycle × No. of cycles per unit time.

$$I.P = \frac{W_{in} N K}{60} \text{ KW}$$

W_{in} = work input in KJ/cycle

N = rpm.

K = no. of suction for revolution.

From an indicated diagram, it is calculated as

I.P = Indicated mean effective pressure × swept vol. rate

$$= \frac{p_{mi} L A N K}{60} \text{ KW}$$

p_{mi} = indicated mean effective pr. (KPa or KN/m²)

L = stroke length in m

A = cross sectional area $= \frac{\pi}{4} \times d^2$ in m²

N = rpm

K = no. of suction per revolution of crank shaft

= 1 for single acting compressor.

= 2 for double acting compressor

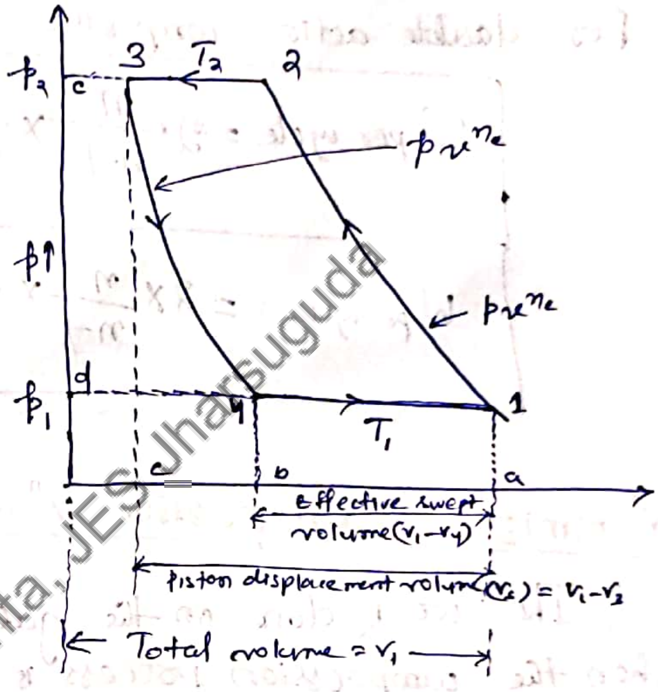
(2) Brake Power (BP) :- ~~The~~ The actual power brake (power or shaft power) input to the compressor is more than the indicated power because some work is required to overcome the irreversibilities and mechanical frictional effects.

Brake power = Indicated power + Frictional power.

(3) Mechanical Efficiency (η_{mech}) :- The mechanical efficiency of the compressor is given by

$$\eta_{mech} = \frac{\text{Indicated Power}}{\text{Brake Power}}.$$

Work done with Clearance Volume :-



① Process 1-2 represents polytropic compression of air (pv^{n_c}) during which pressure increases from p_1 to p_2

② Process 2-3 represents delivery of compressed air at const. pr. p_2

③ Process 3-4 represents expansion of remained compressed air in clearance vol. to suction pressure p_1

④ Process 4-1 represents the suction of air at constant pressure p_1

Work done

Work done per cycle = Area of 1-2-3-4-1

= Area of (1-2-c-d-1) − Area of (4-3-c-d-4)

$$= \frac{n_c}{n_c-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n_c-1}{n_c}} - 1\right] - \frac{n_e}{n_e-1} p_4 v_4 \left[\left(\frac{p_3}{p_4}\right)^{\frac{n_e-1}{n_e}} - 1\right]$$

$$\boxed{W_{per\ cycle} = \frac{n_c}{n_c-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n_c-1}{n_c}} - 1\right] - \frac{n_e}{n_e-1} p_1 v_4 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n_e-1}{n_e}} - 1\right]}$$

If indices of compression and expansion are same then

i.e $n_c = n_e = n$

$$W_{per\ cycle} = \frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] - \frac{n}{n-1} p_1 v_4 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\boxed{W_{per\ cycle} = \frac{n}{n-1} p_1 (v_1 - v_4) \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}$$

$$W \text{ per cycle} = \frac{n}{n-1} p_1 (v_1 - v_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

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

where v_a = actual volume inducted per cycle = $V_1 - V_4$

Clearance has no effect on work done.

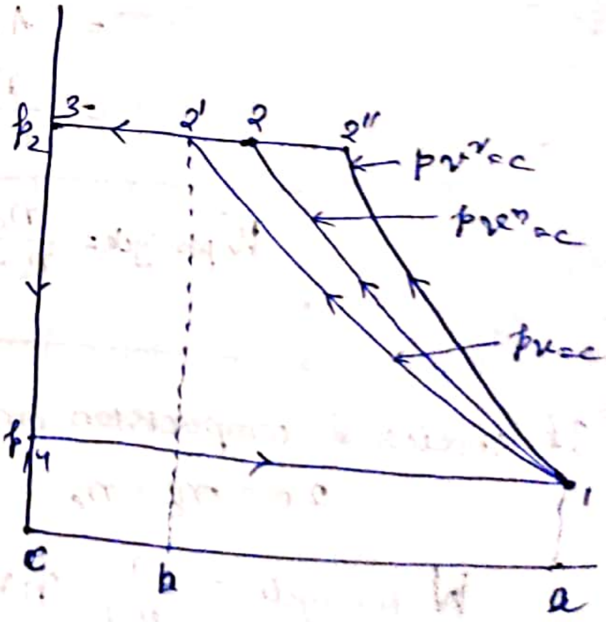
For double acting compressor

$$W \text{ per cycle} = 2 \times \frac{n}{n-1} \times p_1 (V_1 - v_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$W \text{ per cycle} = 2 \times \frac{n}{n-1} \times p_1 v_a \times \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Minimizing Compressor Work

The work done on the gas for compression can be minimized when the compression process is executed in an internally reversible manner. i.e by minimizing the irreversibilities. The other way of reducing the compression work is to keep the specific volume of gas as small as possible during compression process. It is achieved by keep the gas temp. as low as possible during compression



→ For better understanding of the effect of cooling compression process, we consider three types of compression:

(i) Isothermal compression

(ii) Polytropic compression

(iii) Isentropic compression.

a) Isothermal work done & Isothermal efficiency

from the p-v diagram, Isothermal work done

W_{iso} = Area of 1-2'-3-4-1

= Area under 1-2' + Area under 2'-3 - Area under 4-1

$$= -\int_{v_1}^{v_2'} p\,dv + p_2 v_2 - p_1 v_1$$

$$= p_1 v_1 \ln \frac{v_1}{v_2'} + p_2 v_2 - p_1 v_1$$

but $p_2 v_2' = p_1 v_1 \Rightarrow \frac{v_1}{v_2'} = \frac{p_2}{p_1}$

$$W_{iso} \text{ per cycle} = p_1 v_1 \ln \frac{v_1}{v_2'} = p_1 v_1 \ln\left(\frac{p_2}{p_1}\right)$$

$$\boxed{W_{iso} \text{ per cycle} = p_1 v_1 \ln\left(\frac{p_2}{p_1}\right) = mRT_1 \ln\left(\frac{p_2}{p_1}\right)}$$

$$\boxed{\text{Isothermal efficiency} = \frac{\text{Isothermal work done}}{\text{polytropic work done (actual)}} = \frac{\text{Isothermal power}}{\text{Shaft power}}}$$

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

$$\boxed{\eta_{isothermal} = \frac{\ln\left(p_2/p_1\right)}{\frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}}$$

compressor efficiency :- It is the ratio of isothermal work input to indicated work

$$\boxed{\eta_{compressor} = \frac{\text{Isothermal work input}}{\text{Indicated work input}} = \frac{\text{Isothermal power}}{\text{Indicated power}}}$$

(b) Isentropic work done & Adiabatic efficiency

From pv diagram, Isentropic work done

$$W_{isentropic} = \text{Area of } 1-2''-3-4-1$$
$$= \frac{\gamma}{\gamma-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

Adiabatic efficiency :- The adiabatic efficiency of an air compressor is defined as the ratio of isentropic work input to actual work input.

$$\eta_{adiabatic} = \frac{\text{Isentropic work input}}{\text{Actual work input (polytropic work)}}$$

Method of cooling the compressor. (or Method of improving isothermal efficiency) :-

Work done on the compressor will be reduced if the compressor approaches the isothermal compression process. Therefore following methods are adopted to maintain the temperature constant during the compression inorder to reduce the work done.

(1) Water spray.

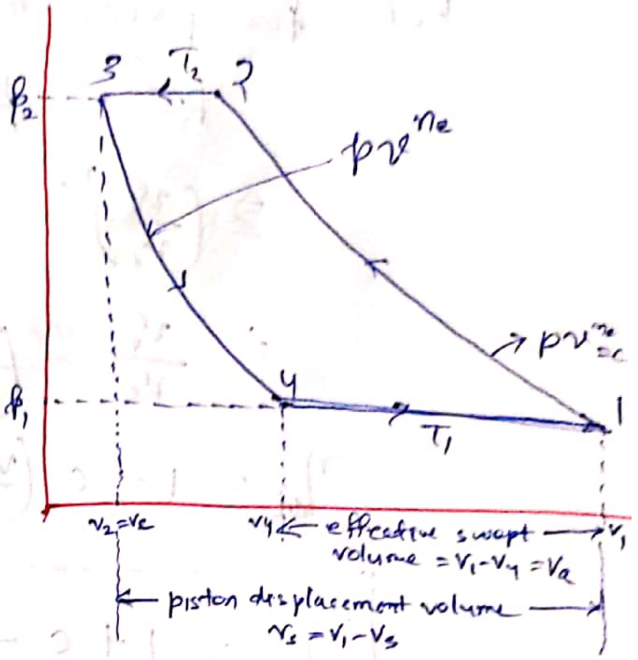
② Water jacketing

③ External fins

④ Intercooler

(5) By suitable cylinder proportion

Volumetric Efficiency :-



Actual volume sucked in the cylinder during the suction stroke is always less than the swept volume. It is due to

(i) Resistance offered by inlet valve to incoming air

(ii) temp. of incoming air

(iii) back pr. of residual gases left in the clearance volume.

Defn :- The volumetric efficiency of air compressor is defined as the ratio of actual volume of air sucked in to the compressor, measured at atmospheric pressure and temperature to piston displacement volume.

In terms of mass ratio, it is defined as the ratio of actual mass of air sucked per stroke to the mass of air corresponding to piston displacement volume at atmospheric condition.

$$\therefore \eta_{volumetric} = \frac{\text{Actual mass sucked}}{\text{mass corresponding to swept vol. at atmospheric pr. \& temp.}} = \frac{\text{free air delivered per stroke}}{\text{swept volume}}$$

$$= \frac{\text{Effective swept volume}}{\text{piston displacement volume}}$$

from p-v diagram,

$$\text{stroke volume} = V_c = V_1 - V_3 = V_1 - V_c$$

$$\text{actual volume of air inducted} = V_a = V_1 - V_4$$

$$\text{Let clearance ratio} = \frac{V_c}{V_s} = c$$

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

$$= \frac{V_s - (V_4 - V_3)}{V_s} = 1 - \frac{1}{V_s} \times V_3\left(\frac{V_4}{V_3} - 1\right)$$

$$= 1 - \frac{V_c}{V_s} \times \left(\frac{V_4}{V_s} - 1\right) = 1 - c\left(\frac{V_4}{V_s} - 1\right)$$

but for process 3-4

$$p_2 v_3^{n_e} = p_4 v_4^{n_e}$$

$$\Rightarrow \left(\frac{v_4}{v_3}\right)^{n_e} = \left(\frac{p_3}{p_4}\right) = \left(\frac{p_2}{p_1}\right)$$

$$\Rightarrow \frac{v_4}{v_3} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}}$$

$$\therefore \eta_{vol} = 1 - c\left(\frac{V_4}{V_3} - 1\right) = 1 - c\left[\left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}} - 1\right]$$

$$\boxed{\eta_{vol} = 1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}}}$$

Problem-1

A single stage reciprocating air compressor takes in 1.4 kg of air per minute at 1 bar and 17°C and delivers at 6 bar. Assuming compression process follows the law $pv^{1.35} = \text{const}$; Calculate the indicated power input to compressor.

Solⁿ Data given - A single stage reciprocating air compressor

mass flow of air $(\dot{m}_a) = 1.4\ kg/min$

inlet pr. $(p_1) = 1\ bar$

inlet temp $(T_1) = 17°C = 290K$

delivery pr. $(p_2) = 6\ bar$

polytropic index $(n) = 1.35$ & $pv^{1.35} = c$

We know that work input to compressor

$$W_{in} = \frac{n}{n-1} \times \dot{m}_a R(T_2 - T_1) = \cancel{\frac{n}{n-1} \dot{m}_a R T_1 \left(\frac{p_2}{p_1}\right)}$$

$$= \frac{n}{n-1} \dot{m}_a R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\therefore W_{in} = \frac{1.35}{1.35-1} \times 1.4 \times 0.287 \times \left[\left(\frac{6}{1}\right)^{\frac{0.35}{1.35}} - 1\right] \quad \frac{Kg}{min} \times \frac{KJ}{Kg\,K} \times K$$

$$= 265.72 \ KJ/min.$$

$$\therefore \text{Indicated Power } (I.P) = \frac{W_{in}}{60} = \frac{265.73}{60} \ KW$$

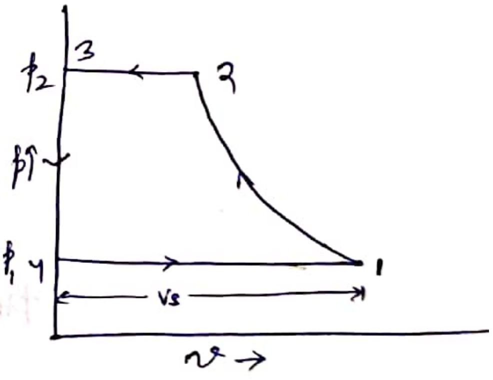
$$I.P = 4.43 \ KW$$

Problem-2

A single acting-single cylinder reciprocating air compressor has a cylinder diameter of 200 mm and a stroke of 300mm. Air enters the cylinder at 1 bar, 27°C. It is then compressed polytropically to 8 bar according to the law $pv^{1.3}$ = const. If the speed of the compressor is 250 rpm, calculate the mass of air compressed per minute and the power required in KW for driving the compressor

Soln Data given - A single acting ~~cy~~ single cylinder reciprocating air compressor

cylinder dia. $(d) = 200\,mm = 0.2\,m$
stroke $(L) = 300\,mm = 0.3\,m$
suction pr. $(p_1) = 1\,bar = 100\,KPa$
delivery pr. $(p_2) = 8\,bar = 800\,KPa$
suction temp $(T_1) = 27°C = 300\,K$
speed $(N) = 250\,rpm$
$n = 1.3$



The swept volume of the cylinder per cycle $(V_s) = V_1 = \frac{\pi}{4}d^2L$

$$V_1 = \frac{\pi}{4}\times(0.2)^2\times 0.3 = 9.424\times10^{-3}\ m^3$$

Using perfect gas equation to point '1'.

$$p_1 v_1 = m_a R T_1 \Rightarrow m_a = \frac{p_1 V_1}{R T_1} = \frac{100\,KPa \times 9.424\times10^{-3}\,m^3}{0.287\,\frac{KJ}{Kg\,K} \times 300\,K}$$

$$m_a = 0.0109 \ Kg/cycle.$$

Mass flow rate of air $(\dot{m}_a)$ = mass of air per cycle × no. of suction/min

$$= 0.0109 \times 250$$

$$= 2.74 \ Kg/min$$

Work input to compressor $(W_{in}) = \frac{n}{n-1} \times \dot{m}_a R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$

$$W_{in} = \frac{1.3}{1.3-1} \times 2.74 \times 0.287 \times 300 \times \left[(8)^{\frac{0.3}{1.3}} - 1\right]$$

$$W_{in} = 629.56 \text{ KJ/min}$$

$$I.P = \frac{W_{in}}{60} = \frac{629.56}{60} = 10.49 \frac{KJ}{sec} = 10.49 \text{ KW}$$

Problem-3

A single-cylinder, double-acting, reciprocating air compressor receives air at 1 bar, 17°C compresses it to 6 bar according to the law, $pv^{1.25}$ = constant. The cylinder diameter is 300 mm. The average piston speed is 150 m/min at 100 rpm. Calculate the power required in KW for driving the compressor. Neglect clearance.

Soln Data given – A double acting. ~~sig~~ single cylinder reciprocating air compressor

cylinder diameter $(d) = 300 \text{ mm} = 0.3 \text{ m}$

$(p_1) = 1 \text{ bar} = 100 \text{ kPa}$

$(p_2) = 6 \text{ bar} = 600 \text{ kPa}.$

$T_1 = 17°C = 290 \text{ K}$

$n = 1.25 \qquad N = 100 \text{ rpm}$

$K = 2$ (double acting)

Piston speed $(V_{piston}) = 150$ m/min

Piston speed $= 2LN \Rightarrow 150 = 2 \times L \times 100$

$\Rightarrow L = 0.75 \text{ m}$

Swept volume $(V_s) = V_1 = \frac{\pi}{4} \times d^2 \times L = \frac{\pi}{4} \times 0.3^2 \times 0.75$

$V_1 = 0.053 \text{ m}^3$

$W_{in} = \frac{n}{n-1} \times p_1 V_1 \cdot \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \times 2$ (for double acting cylinder

$$= \frac{1.25}{1.25-1} \times 100 \times 0.053 \times \left[\left(\frac{6}{1}\right)^{\frac{1.25-1}{1.25}} - 1\right] \times 2$$

$= 11.42 \text{ KJ/cycle} \times 2$

$= 22.84 \text{ KJ/cycle}$

Win per second = Win per cycle x speed(N))

$$I.P = 22.84 \times \frac{200}{60} \text{ KW}$$

$$I.P = 88.1 \text{ KW}$$

Properties of Pure Substance (Steam)

→ A pure substance is a substance of constant chemical composition. It is a one-component system. It may exist one or more phases.

→ A pure substance may be defined as a substance which is chemically homogeneous and has a fixed chemical composition.
Example :- Water, Nitrogen, helium, carbon dioxide etc.

→ A mixture may also be a pure substance. For example, dry air is mainly a mixture of nitrogen and oxygen.

Gas :- A gas refers to a substance that has a single defined thermodynamic state at room temperature.
Ex- Air, Nitrogen, Helium etc.

It is one of the state of matter which ~~ocuy~~ occupy the available space irrespective of the ~~sp~~ shape and volume.

→ Gases can be compressed easily. In normal conditions gas molecules are spread out as due to low intermolecular attraction. This is why gases can be compressed with minimum effort.

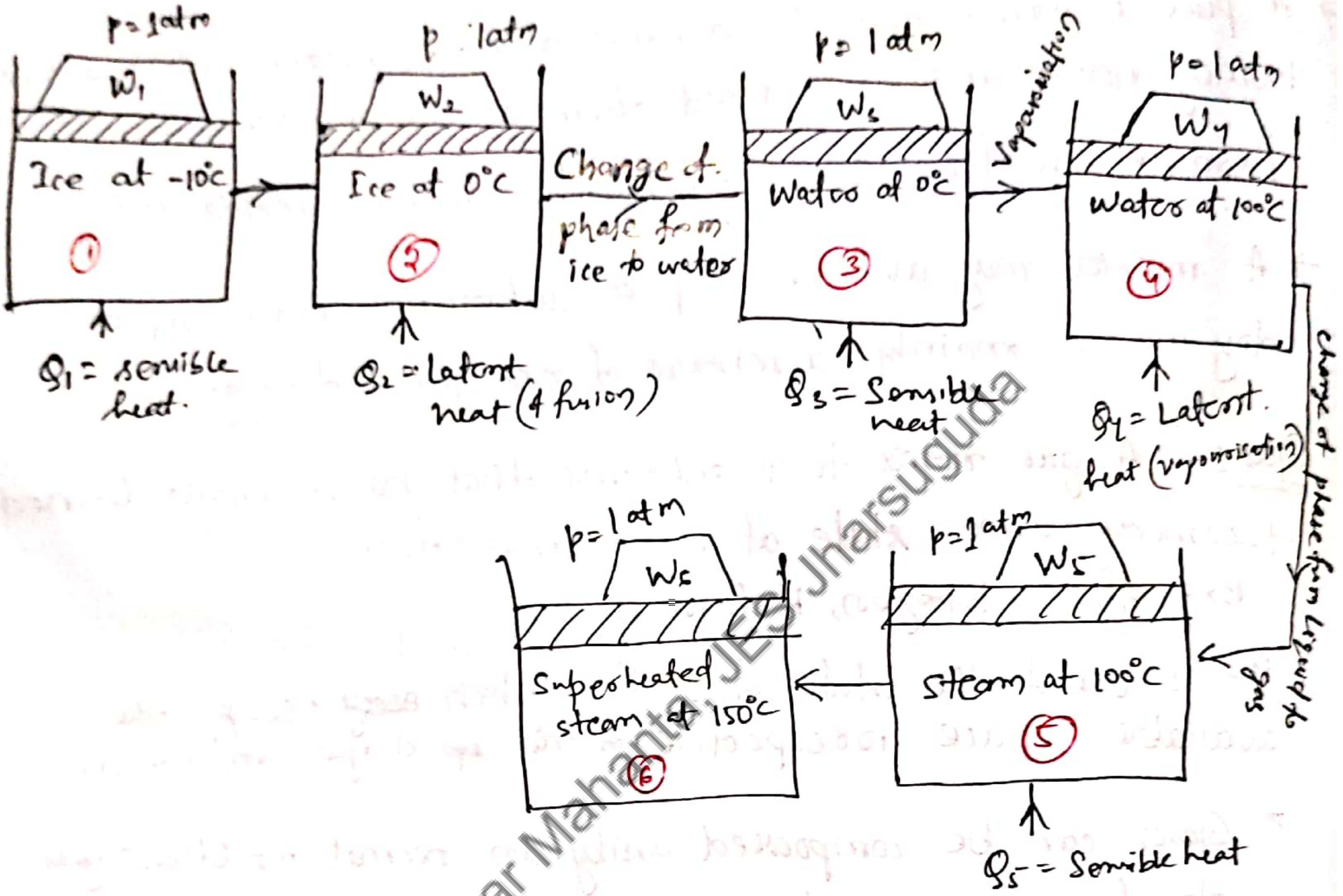
Vapour :- Vapours refers to a substance that is a mixture of two phases at room temperature namely gaseous and liquid phase. Ex- Water vapour. Mercury vapour.

→ Vapour is formed by boiling a liquid and the process is known as evaporation. It is a transition state from liquid phase to gas.

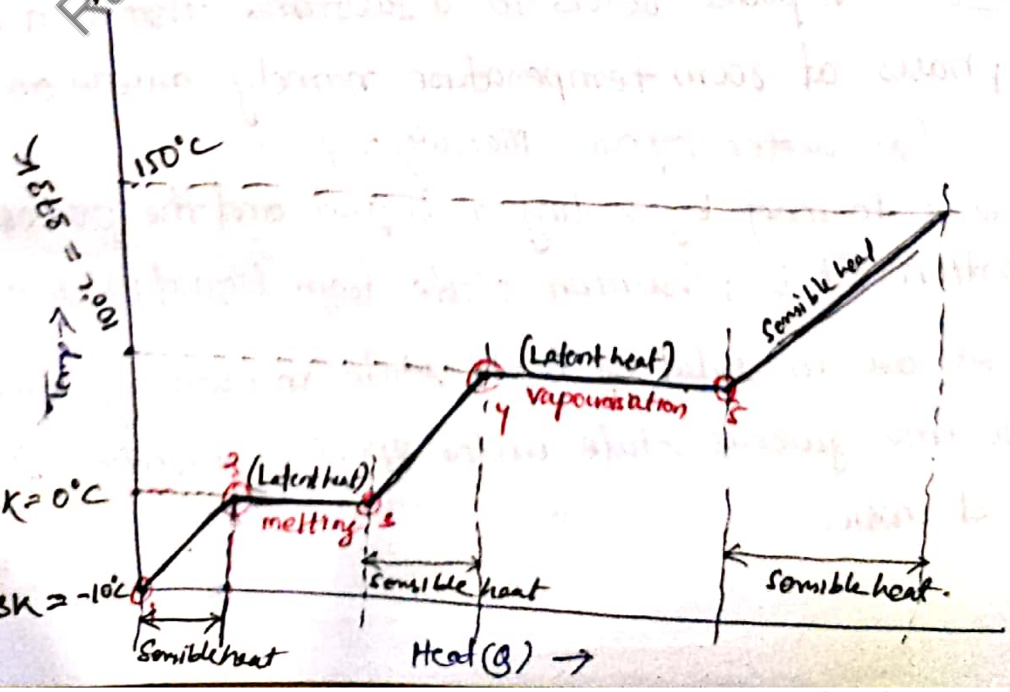
→ Vapours are in solid or liquid state in ordinary conditions, but change into gaseous state under specific conditions. It is not a state of matter

Formation or Generation of steam:-

Let us consider 1 kg of ice at constant pressure inside a vessel. Now it is heated by keeping the pressure constant. Let ice at −10°C is heated upto steam at 150°C at ~~temp~~. 1 atm.



ice at −10°C ① $\xrightarrow{\text{sensible heat}}$ ice at 0°C ② $\xrightarrow{\text{Latent heat of fusion}}$ water at 0°C ③ $\xrightarrow{\text{sensible heat}}$ Water at 100°C ④ $\xrightarrow{\text{Latent heat of vaporisation}}$ steam at 100°C ⑤ $\xrightarrow{\text{sensible heat}}$ steam at 150°C ⑥



state ① to ② :- Decrease in temp of ice from -10°C to 0°C by absorbing sensible heat at 1 atm pressure.

state ② to ③ :- Ice at 0°C starts melting and converted to water at 0°C by absorbing Latent heat (of fusion) at 1 atm pr.

state ③ to ④ :- water at 0°C receives heat (sensible) and starts boiling at 100°C. During this process vaporisation of water occurs at const pr. 1 atm.

state ④ to ⑤ :- Water at 100°C converted into steam at 100°C by absorbing Latent heat (of vaporisation). at ~~on~~ 1 atm pr.

state ⑤ to ⑥ :- Dry steam receives sensible heat and temp increases from 100°C. to 150°C at 1 atm pr. This steam is called as superheated steam.

Terms related to Steam

Saturation Temperature :- Phase change occurs at constant pressure and temperature. A pure liquid at a given pressure will transform into vapour only at a particular temperature, known as saturation temperature at a particular pressure.

Saturation Pressure :- Similarly if the temperature is fixed the liquid will boil only at a particular pressure called saturation pressure.

(or) Saturation temperature (Tsat) :- It is the temperature at which a pure substance starts to evaporate at a given pressure.

(or) Saturation Pressure (Psat) :- It is the pressure at which a pure substance starts to evaporate at a given temperature.

Steam :- It is the gaseous form of water.

Sensible heat :- It is the quantity of heat supplied to increase the temperature of a substance.

Latent heat :- It is the quantity of heat required to change the phase of a substance i.e from solid to liquid or from liquid to gas.

Vaporisation :- It is the process that involves change of phase from liquid to vapour, when latent heat of vaporisation is supplied to saturated water.

Evaporation :- It is the process of vapour generation only at free surface of the liquid.

Boiling :- It is the phenomenon of vapour formation in the whole mass of liquid, when heat is supplied.

Saturated Water :- It is the temperature of water is equal to its saturation temperature at a given pressure is called saturated water.

Compressed Liquid :- Liquid existing at a temperature lower than saturation temperature is called compressed liquid or sub-cooled liquid

$$\text{Degree of subcooling} = T_{sat} - T_i$$

where T_i = initial temp. of the liquid.

Steam :- It is the gaseous phase of water.

Dry Saturated steam :- If the steam does not contain any water particle is called dry saturated steam.

Wet Steam :- If the steam contain some water particle then it is called wet steam. The quality of steam is expressed by a factor known as dryness fraction.

Superheated Steam :- If the steam is heated above its saturation temperature at corresponding pressure, it is called superheated steam. and the temperature is called superheated temperature.

(degree of superheat = $t_{sup} - t_{sat}$)

Dryness fraction :- (x) The quality of the steam is express by a factor, called as dryness fraction. It is defined as the ratio of mass of dry vapour to the total mass of the steam.

Mathematically,
$$x = \frac{\text{mass of dry vapour}}{\text{mass of water particle} + \text{mass of dry vapour}}$$

$$x = \frac{m_g}{m_f + m_g}$$

- → The value of x varies from '0' to '1'
- → The value of x at saturated liquid line is zero '0' & the value of x at saturated vapour line is one '1'.
- → If dryness fraction is 'x' then.

mass of dry vapour = 'x' kg &
mass of water particle = (1−x) kg.

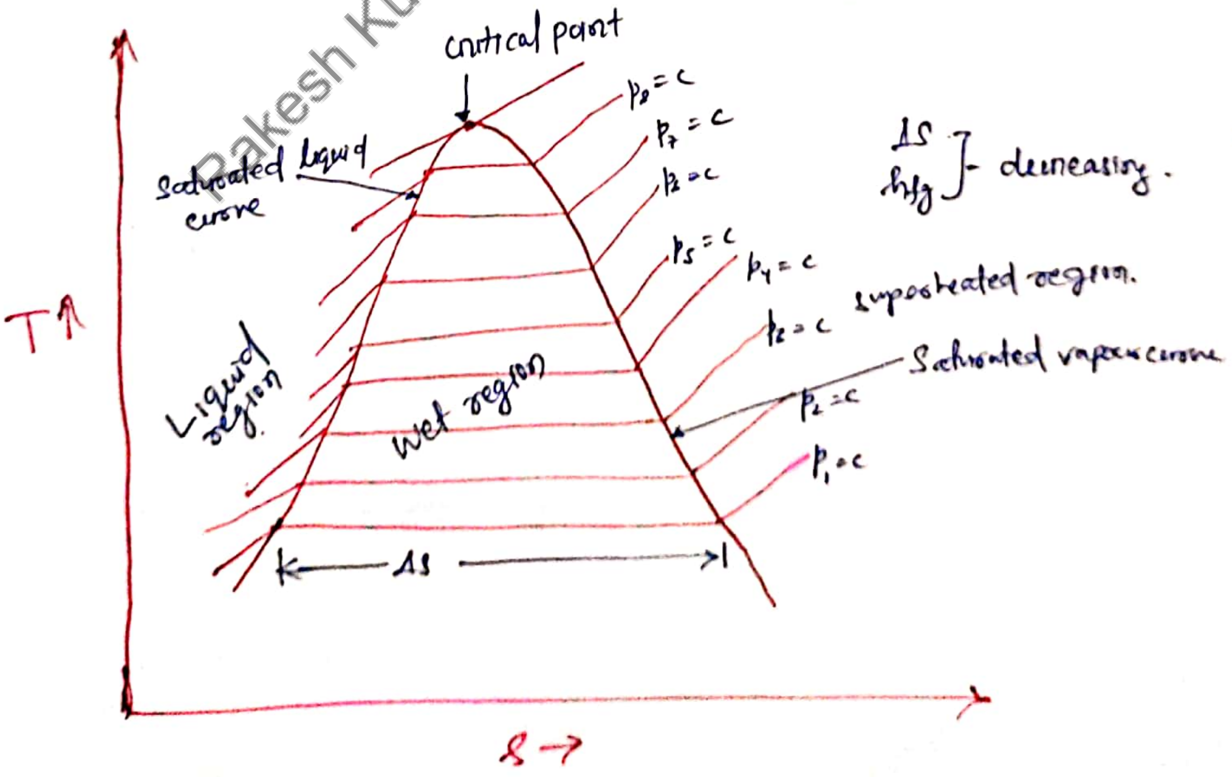
Critical Point, Critical pressure & temp. :-

As the pressure increases the enthalpy of vaporisation decreases. If the pressure is gradually increasing a point will be reached at which no heat is required to convert the water into steam. Such point is called critical point. The corresponding pressure and temperature are called critical pressure and critical temperature.

$$P_{critical} = 225.65\ kgf/cm^2 = 221.2\ bar$$

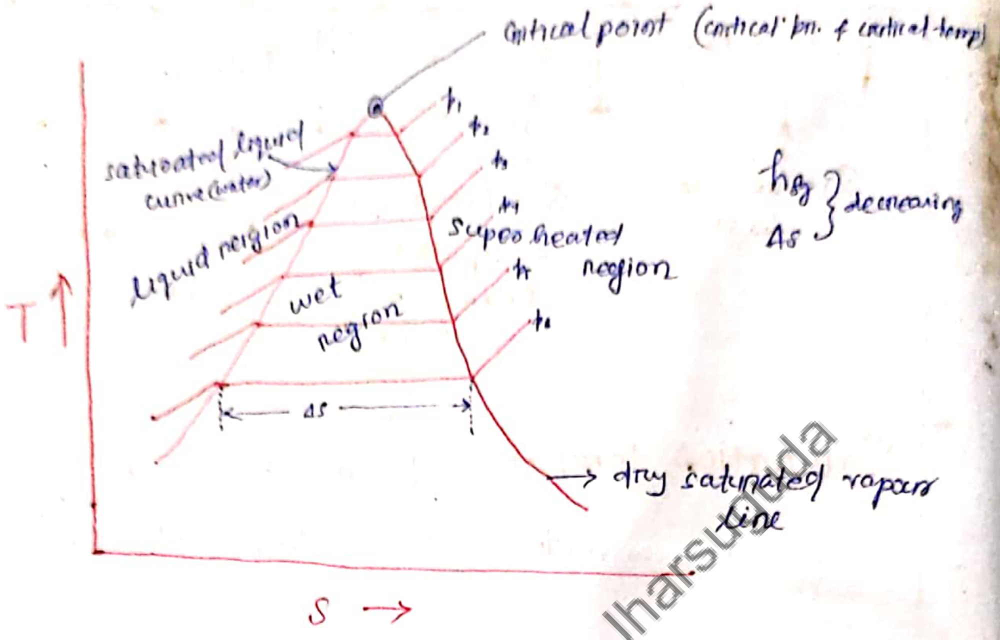
$$T_{critical} = 374.14^\circ C$$

At critical point it is impossible to liquify the vapour.



Saturated water :

If the temperature of water is equal to its saturation temperature at corresponding pressure it is called saturated water.



Critical point, critical pressure & temp :-

As the pressure increases the enthalpy of vaporisation decreases. If the pressure is gradually increasing a point will be reached at which no heat is required to convert the water into steam. Such point is called critical point. The corresponding pressure and temperature is called critical pressure and critical temperature respectively.

$$P_{cr} = 225.65 \ \text{Kgf/cm}^2$$

$$T_{cr} = 374.14^\circ C$$

At critical point it is impossible to liquify the vapour.

Dry saturated steam :-

If the steam does not contain any water particle it is called dry and saturated steam

Superheated steam: If the steam is heated above its saturation temperature at corresponding pressure it is called superheated steam & the temperature is called superheated temperature.

Wet steam: If the steam contain some water particles it is called wet steam. The quality of steam is expressed by a factor known as dryness fraction:

Dryness fraction (x): It is defined as the ratio of mass of dry vapour to the total mass of the steam.

$$x = \frac{\text{mass of the dry vapour}}{\text{mass of the water particle} + \text{mass of dry vapour}}$$

Example For 1 kg of wet steam if $x = 0.8$ find mass of the water particle.

Ans

Mass of dry vapour $= 0.8 \times 1\text{ kg} = 0.8\text{ kg}$

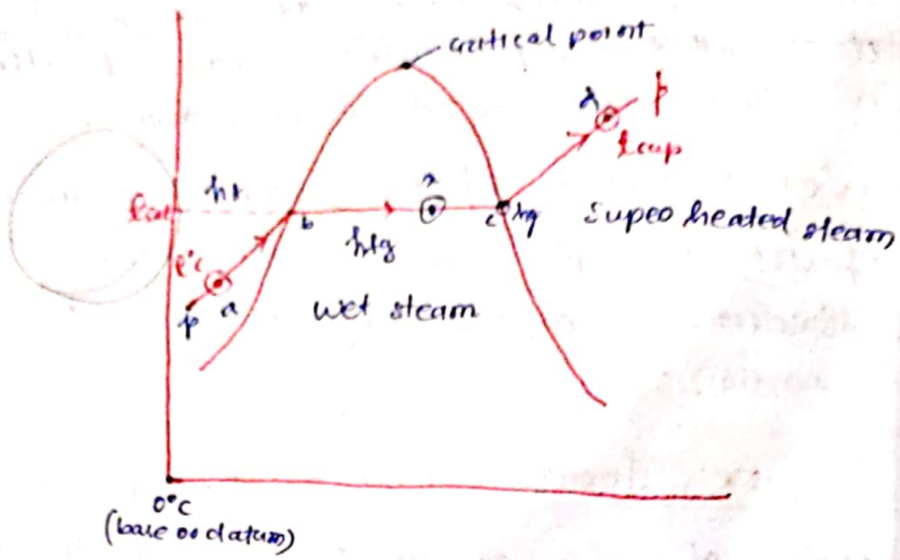
Mass of water vapour $= 1 - 0.8 = 0.2\text{ kg}$.

If dryness fraction is 'x' then,

Mass of dry vapour $= x$ kg &

mass of water particles $= (1 - x)$ kg

Enthalpy of 1 kg of steam



$$Q = \Delta h = h_2 - h_1 = m c_{pw} (t_2 - t_1)$$

$$= 1 kg \times 1\ kcal/kg°C\ (t - 0)°C$$

$$= t\ kcal/kg$$

$$\boxed{C_{pw} = 1\ kcal/kg°K = 4.18\ KJ/kg°K}$$

$$h_2 = t\ kcal/kg = 4.18\ t\ KJ/kg$$

If temperature of water is t°C the enthalpy 1kg of water will be t kcal/kg or 4.18 KJ/kg (S.I)

Enthalpy of saturated water

① Enthalpy of saturated water $= t_{sat}\ kcal/kg$

$$= 4.18 \times t_{sat}\ KJ/kg$$

$$= h_f$$

b - c → Q_{bc}

$$Q_{bc} = h_{fg} \text{ or } L = h_c - h_b$$
$$= h_c - h_f$$

$$\boxed{h_c = h_f + h_{fg}}$$

② Enthalpy dry of saturated steam (h_g) = $h_f + h_{fg}$

③ Enthalpy of wet steam having dryness fraction x

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

④ Enthalpy of super heated steam having temp. of t_{sup}

$$h_{sup} = h_g + C_{ps}(t_{sup} - t_{sat})$$

C_{ps} = sp. heat of super heated steam
= 0.5 to 0.55 Kcal/kg°C ≅ 2.1 KJ/kg°C

$$Q_{cd} = m C_{ps} \cdot (t_{sup} - t_{sat})$$
$$\Rightarrow h_d - h_c = C_{ps}(t_{sup} - t_{sat})$$
$$\Rightarrow h_d - h_g = C_{ps}(t_{sup} - t_{sat})$$

$t_{sup} - t_{sat}$ = degree of super heat

Q = Find the enthalpy of 3 kg of steam available at 10 bar under following conditions

(a) Steam is wet with dryness fraction 0.8

(b) Steam is dry saturated

(c) temp. of steam is 200°C.

Soln

(a) From steam table

At pressure 10 bar h_f = 763 KJ/kg

$h_{fg} = 2013.5$ kJ/kg

$h_g = 2776.5$ kJ/kg

$t_{sat} = 179.97$

(a) Enthalpy of 1kg of wet steam $= h_f + x h_{fg}$

$= 763 + 0.8 \times 2013.5$

$= 2373.8$ KJ/kg

Enthalpy of 3kg of steam $= 3 \times 2373.8$

$= 7121.4$ KJ/kg

(b) Enthalpy of 3 kg of dry saturated steam $= 3 \times h_g$

$= 3 \times 2776.5 = 8329.5$ KJ/kg

(c) Enthalpy of 3 kg of super heated steam

$= 3 \left[h_g + C_{ps} \cdot (t_{sup} - t_{sat})\right]$

$= 3 \left[h_g + 2.1(200 - 179.97)\right]$

$= 3 \times \left[2776.5 + 2.1(200 - 179.97)\right]$

$= 8455.689$ KJ/kg

at 10 bar · 200°C $h_g = 2827.9$

Q 1kg Steam is available at 12 bar if the degree of super heat is 50 °C find the enthalpy of steam.

$t_{sup} - t_{sat} = 50°C$

$\Rightarrow t_{sup} = 50 + 188 = 238°C$

Enthalpy $= h_g + C_{ps} \cdot (t_{sup} - t_{sat})$

$= 2782.8 + 2.1 \times 50$

$= 2887.8$ KJ/kg

Find the enthalpy of 1 kg of dry saturated steam at pressure 20.2 bar

Soln

	20 bar	21 bar
h_g	2797.0	2798.2

$h_{g\,21\,bar} - h_{g\,20\,bar} = 1.2$

$\therefore$ 0.2 bar = $1.2 \times 0.2 = 0.24$ KJ/kg

$\therefore$ h_g at 20.2 bar = 2797.24 ~~bar~~ KJ/kg

Find the enthalpy of 1kg of steam at pr. 10.2 bar under following conditions.

(a) Steam is dry saturated

(b) Dryness fraction of steam is 0.75

Soln

Enthalpy of dry saturated steam =

h_g =

at 10 bar = 2776.5

at 11 bar – 2779.7

for 1 bar – 3.2

for 0.2 bar = $3.2 \times 0.2 = 0.64$

For 10.2 bar = 2776.5 + 0.64

$$\boxed{h_g = 2777.14 \text{ KJ/kg}}$$

(b) Enthalpy of wet steam

$= h_f + x\, h_{fg}$

	10 bar	11 bar
h_f =	763.	781.3

1 bar = 18.3

0.2 bar = 0.2 × 18.3 = 3.66

10.2 bar = 763 + 3.66 = 769.66 KJ/kg

	10 bars	11 bars
h_{fg} =	2013.5	1998.4

1 bar = 15.1.9

0.2 bar = 15.1 × 0.2 = 3.02

h_{fg} at 10.2 bar = 2013.5 − 3.03

= 2010.48

Enthalpy = $h_f + x \times 2010.48$

= 769.66 + 0.75 × 2010.48

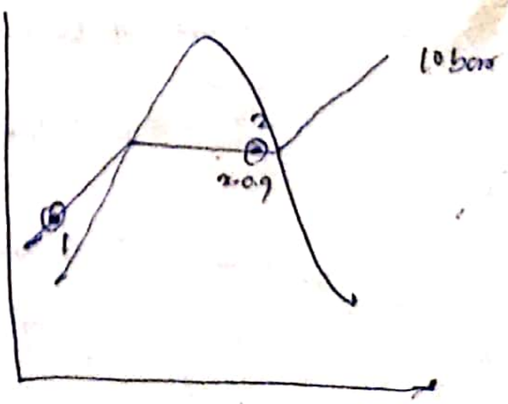
2 kg of water is available at 10 bar and 60°C. Find the amount of heat required to convert it into 0.9 dry steam

Soln

For 1 kg of steam

$Q = h_2 - h_1$

$h_1 = 4.18 \times 60$ KJ/kg



$h_2 = h_f + x h_{fg}$

$= 763 + 0.9 \times 2013.5 = 2574.7$ KJ/kg

$Q = 2[h_2 - h_1]$ KJ

$Q = 2 \ [2574.7 - 250.8]$

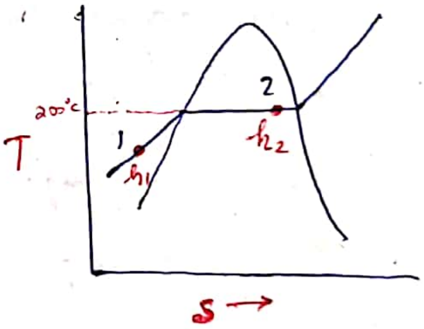
$= 4647.8$ KJ/kg

Q Temp. of a sample steam having 0.9 as dryness fraction is 200°C feed water temp is 20°C. If steam is generated under constant pressure. Find
(1) sp. enthalpy of steam
(2) Heat required to generate 100 kg of steam
Assume sp. heat of water 4.1868 KJ/kg°K

Ans Data given

Dryness fraction = 0.9

Saturation temp. = 200°C



Specific enthalpy of steam

$h_2 = h_f + x h_{fg}$

$= 852.479 + 0.9 \times 1938.488$

$= 2597.1182$ KJ/kg

Heat required to generate 100kg of steam

$Q = m(h_2 - h_1)$

Sp. enthalpy of water at 20°C $= 20 \times 4.1868$

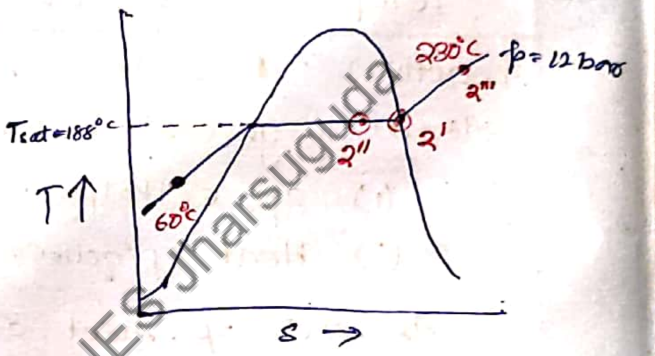
$= 83.736$ KJ/kg

∴ Heat required $Q = m(h_2 - h_1) = 100(2597.1182 - 83.736)$

$Q = 251338.22$ KJ

② Steam is being generated in a boiler at a pn. of 12 bar from water at 60°C. Find the heat required to generate 10 kg of steam when
① Steam is dry saturated
② Dryness fraction of steam is 0.9
③ Temp. of steam is 230°C
Take sp. heat of water as 4.2 KJ/kg°K

Solⁿ Data given



Sp. enthalpy of water at 60°C $(h_1) = 60 \times 4.2$

$= 252$ KJ/kg

From steam table at pn. 12 bar

$h_f = 798.7$ KJ/kg

$h_{fg} = 1984.1$ KJ/kg

$h_g = 2782.8$ KJ/kg

(I) Sp. enthalpy of dry saturated steam

$h_2' = h_g = 1984.1$ KJ/kg

∴ Heat supplied $= m(h_2' - h_1)$

$= 10\,(\overset{2782.8}{\cancel{1984.1}} - 252)$

$= \overset{25308}{17321}$ KJ

(II) Sp. enthalpy steam at dryness fraction 0.9

$h_2'' = h_f + x\,h_{fg}$

$h_2'' = 798.7 + 0.9 \times 1984.1$

$= 2584.39$ KJ/kg

∴ Heat supplied $Q = (h_2'' - h_1)\, m$

$= (2584.39 - 252)\, 10$

$= 23323.9$ KJ

(iii) From steam table at 12 bar & 230°C sp. enthalp.

At	200° C	250°C
h_2'''	2815.9	2935.0

For 50°C increase in enthalpy = 119.1

For 30°C increase in enthalp $= \frac{119.1}{50} \times 30$

$= 71.46$ KJ/kg

∴ $h_2''' = 2815.9 + 71.46$

$= 2887.36$ KJ/kg

∴ Heat required $= m \cdot (2887.36 - 252)$

$= 26353.6$ KJ.

(or) $h_2'' = h_g + C_{ps} (t_{sup} - t_{sat})$

$= 2782.8 + 2.1 (230 - 188)$

$= 2871$ KJ/kg

Heat required (Q) $= (h_2''' - h_1) \times m$

$= (2871 - 252)\, 10$

$= 26190$ KJ

Q-3 5 kg of 0.9 dry steam is passing through a pipe line at const. pressure of 12 bar. If 2000 KJ of heat is rejected from the steam through the wall of the pipe. What will be the quality of steam

Soln

Mass of steam = 5 kg

dryness fraction = 0.9

pressure = 12 bar

Heat rejected = 2000 KJ

Sp. enthalpy of steam at the inlet of the pipe

$h_1 = h_f + x h_{fg}$

$= 798.7 + 0.9 \times 1984.1$

$= 2584.39$ KJ/kg

Total heat $= m \times h_1$

$H_1 = 5 \times 2584.39$

$= 12921.95$ KJ

Heat rejected = 2000 KJ

∴ Total heat at out let of the pipe

$H_2 = 12921.95 - 2000$

$= 10921.95$ KJ

Specific heat at out let of the pipe

$h_2 = \frac{H_2}{5} = \frac{10921.95}{5}$

$= 2184.39$ KJ/kg

Again $h_2 = (h_f + x_2 \cdot h_{fg})$

$x_2 h_{fg} = 2184.39 - 798.7$

$= 1385.69$

$$x_2 = \frac{1385.69}{1984.1} = 0.69$$

The quality of steam is wet of drynessfractio

$= 0.69$

No-4 10 Kg of 0.9 dry steam is passing through a pipe under constant pressure of 10 bar. If 3000 KJ of heat supplied to the steam, what will be the quality steam at out let

Soln Data given

mass of the steam = 10 Kg

dryness fraction = 0.9

pressure = 10 bar

heat supplied = 3000 KJ

Specific enthalpy of steam at the inlet of the pipe

$h_1 = h_f + x h_{fg}$

$= 763 + 0.9 \times 2013.5$

$= 2575.15$ KJ/kg

Total heat at the inlet of the pipe

$H_1 = m \times h_1 = 10 \times 2575.15$

$= 25751.5$ KJ

Heat supplied = 3000 KJ

∴ Total heat at the out let of the pipe

$H_2 = H_1 + 3000 = 25751.5 + 3000$

$= 28751.5$ KJ

Sp. enthalpy of the steam at the exit of the pipe

$h_2 = \frac{H_2}{10} = \frac{28751.5}{10} = 2875.15$ KJ/kg

Again,

$$h_2 = (h_f + x_2 \times h_{fg})$$

$$\Rightarrow x_2 \times h_{fg} = h_2 - h_f$$

$$\Rightarrow x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{2875.15 - 763}{2013.5}$$

$$= 1.048$$

Hence the steam is super heated.

At 10 bar $t_{sat} = 179.97$

$$\therefore h_2 = h_g + C_{ps}(t_{sup} - t_{sat})$$

$$\Rightarrow C_{ps}(t_{sup} - t_{sat}) = h_2 - h_g = 2875.15 - 2776.5$$

$$\Rightarrow t_{sup} - t_{sat} = \frac{98.65}{2.1} = 46.976$$

$$\Rightarrow t_{sup} = 46.976 + 179.97$$

$$\boxed{t_{sup} = 226.946\ °C}$$

Specific volume of steam (v) m^3/kg

① Specific volume of saturated water = v_f

② Specific volume of dry saturated steam = v_g

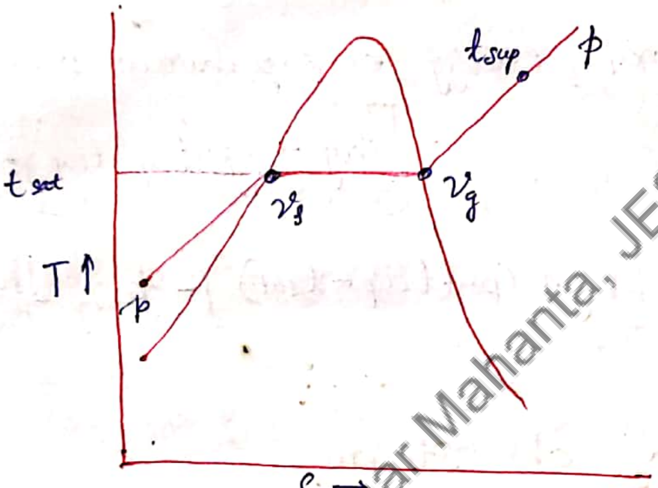
③ Specific volume of wet steam (dryness fraction = x)

$$= (1-x)v_f + x v_g$$

$$\simeq x v_g$$

④ Specific volume of superheated steam

$$v_{sup} = v_g \frac{T_{sup}}{T_{sat}}$$



Super heated steam behaves as perfect gas & obeys the gas law

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}$$

$$\Rightarrow \frac{v_1}{T_1} = \frac{v_2}{T_2} \Rightarrow \boxed{\frac{v_g}{T_{sat}} = \frac{v_2}{T_{sup}}}$$

Specific internal energy of Steam (u) :- kJ/kg

Specific internal energy (u) = $h - pv$

Unit S.I system

$$u = h - pv$$

$$\frac{kJ}{kg} = \frac{kJ}{kg} \cdot \frac{kJ}{1000kg}$$

M.K.S

$$\frac{Kcal}{kg} = \frac{Kcal}{kg} \cdot \frac{Kgf}{m^2} \times \frac{m^3}{kg}$$

$\text{Kgf m/Kg} \rightarrow \frac{1}{427}\ \text{Kcal/kg}$

$1\ \text{kcal} = 427\ \text{kgf m}$
$= J$

① Specific internal energy of saturated water

$$= h_f - p v_f$$

② Specific internal energy of dry saturated steam

$$= h_g - p v_g$$

③ Specific internal energy of wet steam (dryness fraction = x)

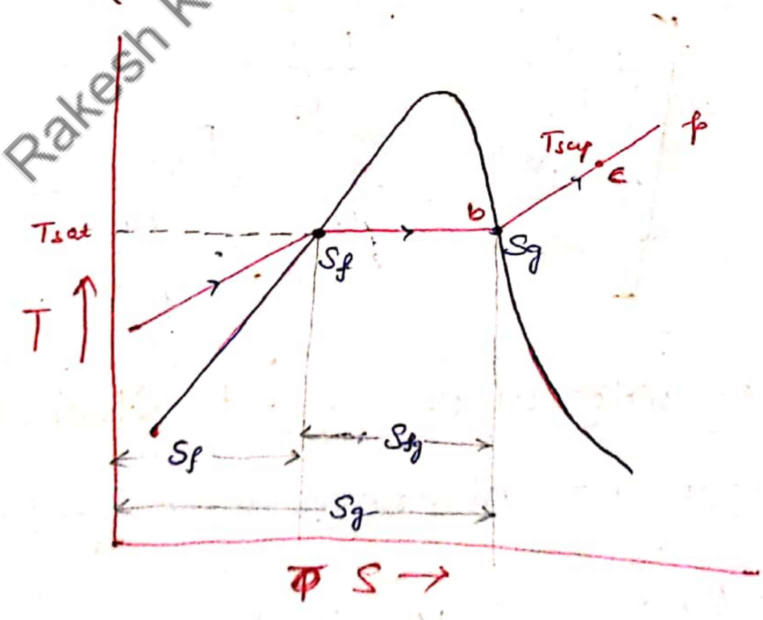
$$= (h_f + x h_{fg}) - p x v_g$$

④ Specific internal energy of sup heated steam

$$= \left[h_g + C_{ps}(t_{sup} - t_{sat})\right] - p v_g \frac{T_{sup}}{T_{sat}}$$

$$= \left[h_g + C_{ps}(t_{sup} - t_{sat})\right] - p\, v_g \frac{T_{sup}}{T_{sat}}$$

Specific entropy of steam



① Specific entropy of saturated water

s_f = (from steam table)

$$\Delta s = m C_p \ln \frac{T_2}{T_1}$$

$$\Rightarrow s_f - 0 = 1 \times C_{pw} \ln \frac{T_{sat}}{273}$$

$$\boxed{s_f = C_{pw} \ln \frac{T_{sat}}{273}}$$

$C_{pw} = 1 \text{ Kcal/kg}^\circ K$
$= 4.2 \text{ kJ/kg}^\circ K$

② * $\Delta s = \int \frac{dQ}{T_{sat}} = \frac{1}{T_{sat}} \int dQ = \frac{1}{T_{sat}} Q$

$$= \frac{h_{fg}}{T_{sat}} = s_{fg}$$

② Specific entropy of dry saturated steam $= s_g =$

$$s_g = s_f + s_{fg}$$
$$= s_f + \frac{h_{fg}}{T_{sat}}$$

③ Specific entropy of wet steam $= s_f + \frac{x h_{fg}}{T_{sat}}$

$$= s_f + x s_{fg}$$

④ * $\Delta s_{bc} = m C_{ps} \ln \frac{T_{sup}}{T_{sat}}$

$s_c - s_b = C_{ps} \ln \frac{T_{sup}}{T_{sat}}$ $m = 1$

$$s_c = s_b + C_{ps} \ln \frac{T_{sup}}{T_{sat}}$$
$$= s_g + C_{ps} \ln \frac{T_{sup}}{T_{sat}}$$

(4) Specific entropy of superheated steam

$$= s_g + C_{ps} \ln \frac{T_{sup}}{T_{sat}}$$

① Specific entropy of saturated water = s_f

② Specific entropy of dry saturated steam = s_g

$$s_g = s_f + s_{fg}$$
$$= s_f + \frac{h_{fg}}{T_{sat}}$$

③ Specific entropy of wet steam = $s_f + \frac{x h_{fg}}{T_{sat}}$

$$= s_f + x s_{fg}$$

④ Specific entropy of superheated steam

$$= s_g + C_{ps} \ln \frac{T_{sup}}{T_{sat}}$$

Problem

Find the sp. entropy & sp. internal energy of steam at 10 bar pn. under following condition.

(i) Steam is 0.9 dry

(ii) Steam is superheated having temp. of 250°C

Soln

Data given

pressure = 10 bar

(i) dryness fraction of steam = 0.9

ⓐ Specific entropy wet steam = $s_f + x s_{fg}$

from steam table $s_f = 2.1392$

$s_{fg} = 4.4432$

∴ Sp. entropy wet steam = 2.1392 + 0.9 × 4.4432

= 6.138 kJ/kg°K

ⓑ Sp. internal energy of wet steam

$$= (h_f + x h_{fg}) - x p v_g$$

from steam table $h_f = 763$ KJ/kg

$h_{fg} = 2013.5$ KJ/kg

$p = 10$ bar $= 10 \times 10^5$ N/m²

$v_g = 0.1923$ m³/kg

∴ Sp internal energy of wet steam

$$= (763 + 0.9 \times 2013.5) - \frac{0.9 \times 10^6 \times 0.1933}{1000}$$

$$= 2575.15 - 173.97 \text{ KJ/kg}$$

$$= 2401.18 \text{ KJ/kg}$$

(ii) ⓐ from steam table sp. entropy of super heated steam having temp. 250°C is

$$s_g = 6.926 \text{ KJ/kg°K}$$

ⓑ Sp. internal energy of super heated steam

$$u = h - p.v$$

from steam table $h =$ [illegible]

$= 2942.6$ KJ/kg

$v = 0.2327$ m³/kg

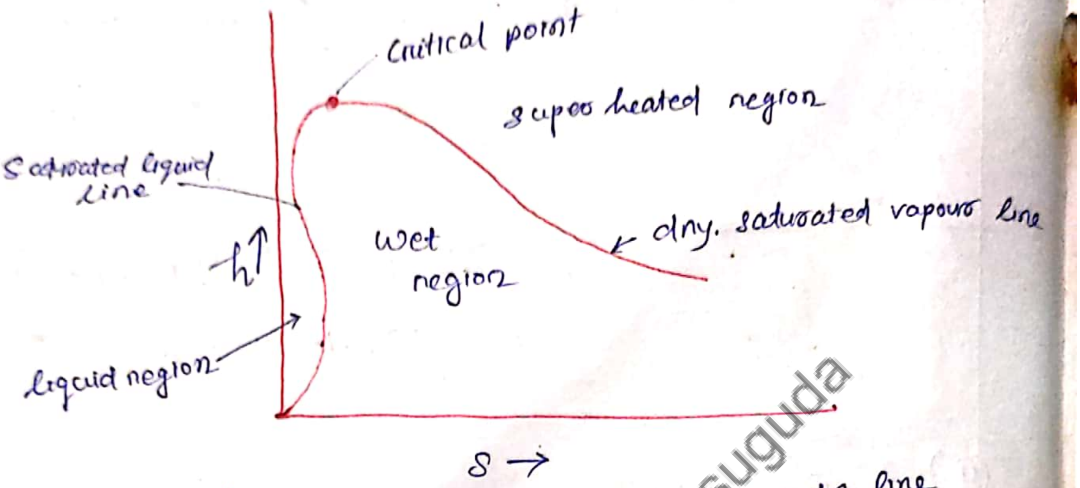
$p = 10$ bar $= 10^6$ N/m²

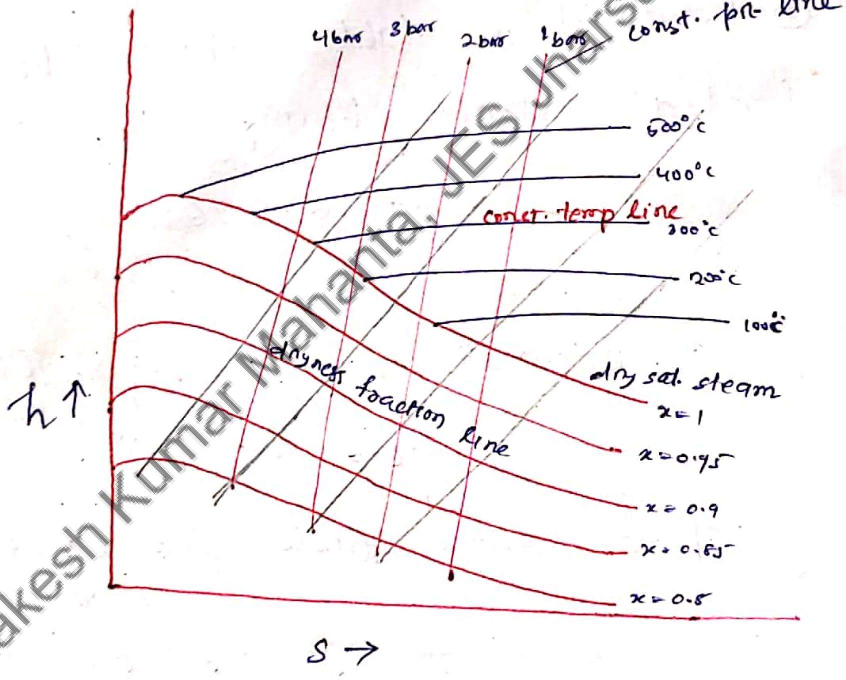
$$\therefore u = 2942.6 - \frac{10^6 \times 0.2327}{1000}$$

$$= 2709.9 \text{ KJ/kg}$$

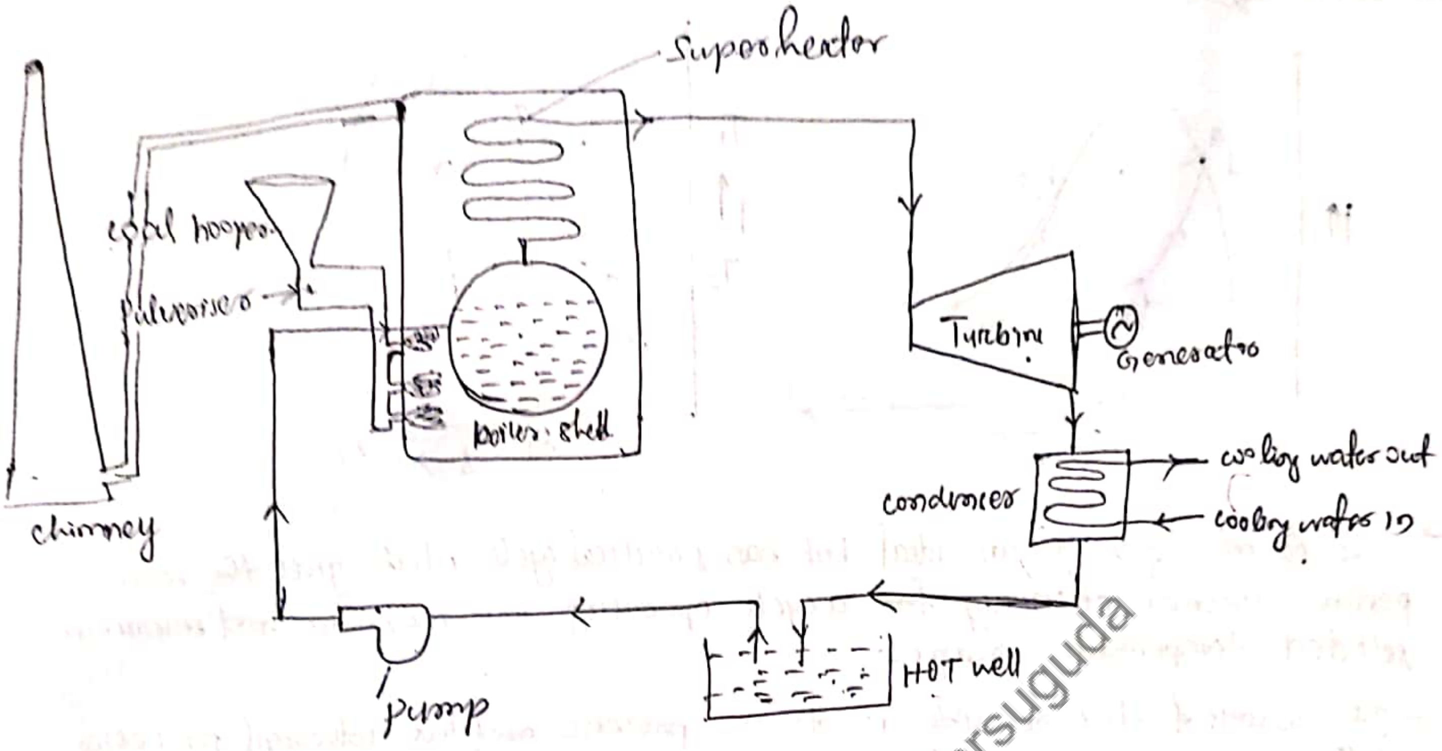
Mollier Chart :-

It is a graph of specific enthalpy and specific entropy (or) It is called enthalpy entropy chart.





VAPOUR POWER CYCLE



→ A steam power plant continuously converts energy stored fossil fuels (coal, oil, natural gas) into shaft work and ultimately into electricity. The working fluid is water which sometimes in liquid phase and sometimes in vapour phase during its cycle of operations.

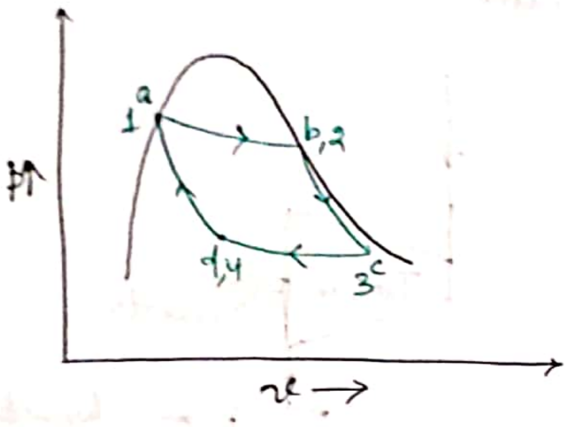
→ Energy released by burning fuels is transferred to the water inside the boiler to generate steam at a very high pressure and temperature.

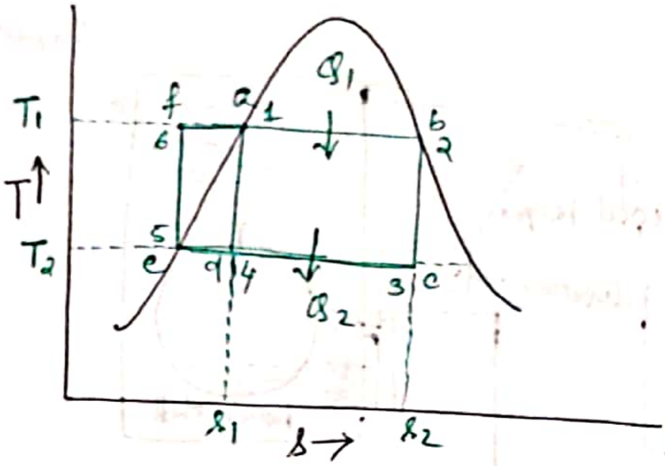
→ The high pr. and high temp. steam is then allowed to expand in the turbine to a very low pressure to produce shaft work

→ The low pr. steam leaving from the turbine is then condensed inside the condenser where cooling water is circulated.

→ The condensed water is collected in a (tank) reservior or Hot well and then fed to the boiler by using a feed pump.

Carnot Cycle :-





→ The Carnot cycle is an ideal but non practical cycle which gives the maximum possible thermal efficiency for a cycle operating on maximum and minimum selected temperature ranges.

→ It consists of two reversible isentropic processes and two isothermal processes. Different processes are as follows.

(i) The process 1-2 represents reversible isothermal expansion during which heat is supplied into the system at constant temp. 'T_1'.

(ii) The process 2-3 represent reversible isentropic expansion of the vapour during which temp. falls from T_1 to T_2.

(iii) Process 3-4 represents, reversible isothermal compression during which heat is rejected from the system.

(iv) Process 4-1 represents isentropic compression of vapour during which temp. increases from T_2 to T_1

Heat supplied (Q_1) = area under 1-2 = $T_1(s_2 - s_1)$

Heat rejected (Q_2) = area under (3-4 = $T_2(s_2 - s_1)$

Net work done $(W) = Q_1 - Q_2 = (T_1 - T_2)(s_2 - s_1)$

Carnot efficiency $\eta_{carnot} = \dfrac{\text{Net work done}}{\text{heat supplied}} = \dfrac{Q_1 - Q_2}{Q_1}$

$$= \frac{T_1(s_2 - s_1) - T_2(s_2 - s_1)}{T_1(s_2 - s_1)}$$

$$\boxed{\eta_{carnot} = \frac{T_1 - T_2}{T_1}}$$

Problem :- A system working on Carnot cycle, dry saturated steam expands from 40 bar to 1 bar. Find the work done, heat supplied & carnot efficiency.

Soln :

boiler pr. $(p_1) = 40$ bar

condenser pr. $(p_2) = 1$ bar.

$T_1 =$ saturation temp. at 40 bar $= 250.4^\circ C = 523.4\ K$

$T_2 =$ saturation temp. at 1 bar $= 99.63^\circ C = 372.63\ K$

$s_2 = s_g|_{40\ bar} = 6.0701\ kJ/kg\ K$

$s_1 = s_f|_{1\ bar} = 1.3026\ kJ/kg\ K$

Work done $(w) = Q_1 - Q_2 = (T_1 - T_2)(s_2 - s_1)$

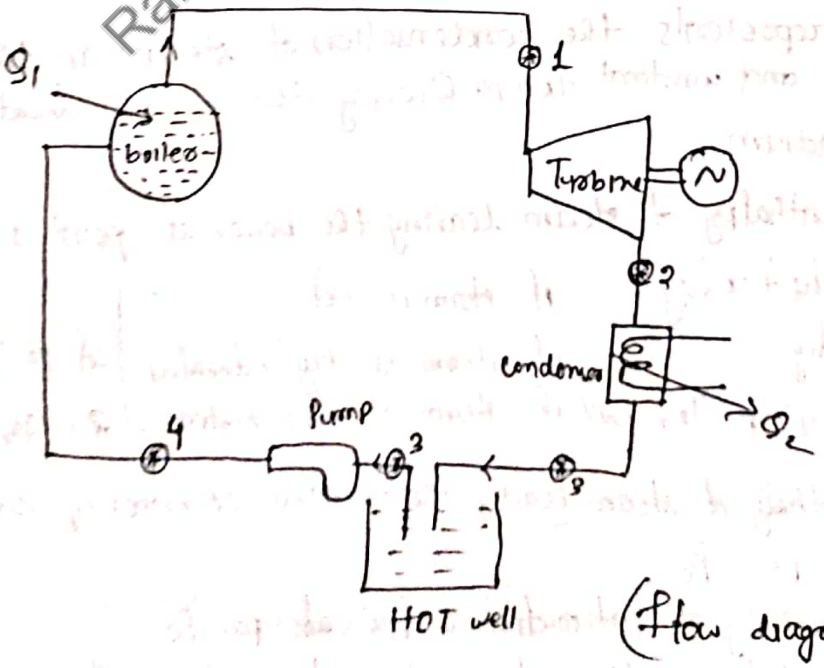
$= (250.4 - 99.63)(6.0701 - 1.3026)$

$= 718.8\ kJ/kg$

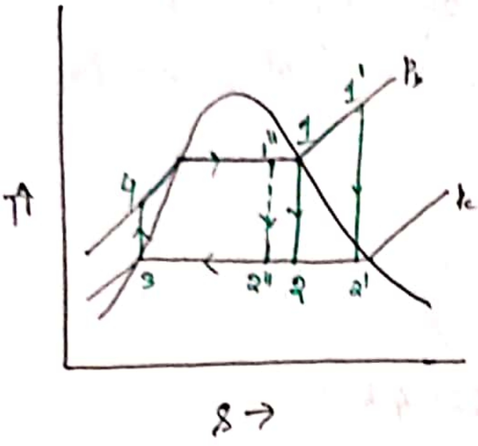
Heat supplied $(Q_1) = T_1(s_2 - s_1) = 523.4(6.0701 - 1.3026) = 2495.3095\ kJ/kg$

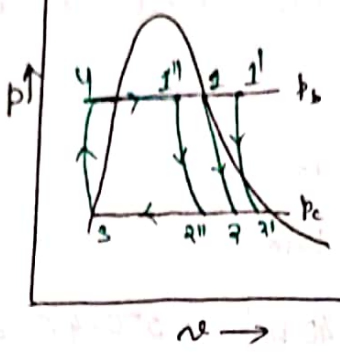
Carnot efficiency $= \frac{T_1 - T_2}{T_1} = \frac{523.4 - 372.63}{523.4} = 0.288 = 28.8\%$.

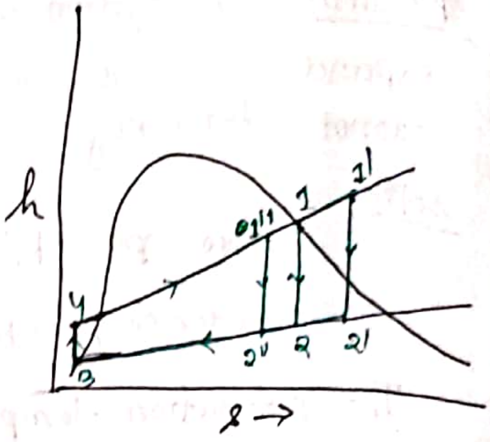
RANKINE CYCLE :-



(Flow diagram of steam power plant)







→ The process 3-4 represents the pressurisation of feed water in the feed pump during which the pressure increases from condenser pr. (P_c) to boiler pr. (P_b). The pressurisation of water is treated as isentropic process. During the process there is a slight rise in temp. of water & the vol. of water is assumed to be constant.

→ The process 4-1 represents the heat supplied in the boiler at constant pr (P_b). During the process ~~4-1~~ 3-4 the temp. of the water increases upto its saturation temp. at pr. (P_b). During this process the high pr. water is 1st heated upto saturation temp & then vaporised at const. pr. (P_b). At point '1' the steam leaves the boiler. The condition of the steam at point '1' may be wet, dry saturated or superheated.

→ The process 1-2 represents the isentropic expansion of the steam in the turbine during which the pressure falls from boiler pr. (P_b) to condenser pr. (P_c). Due to expansion of the steam work is produced

→ The process 2-3 represents the condensation of steam in the condenser at constant pressure and constant temp. During this process heat is rejected to the cooling medium.

h_1 = enthalpy of steam leaving the boiler at point '1'

$= h_f + x h_{fg}$ if steam is wet

$= h_g$ if steam is dry saturated

$= h_g + C_{ps}(T_{sup} - T_{sat})$ if steam is superheated

} at pr. P_b

2.1 KJ/kg K = C_{ps}

h_2 = Enthalpy of steam leaving the turbine or entering the condenser at pr. P_c

h_3 = Enthalpy of saturated water at pr. 'P_c'

h_4 = enthalpy of the water entering the boiler or leaving the pump

(h_4 can't be calulated)

W_T = Turbine work / work done by the turbine

W_P = Pump work

Q_1 = Heat supplied in the boiler

Q_2 = Heat rejected in the condenser.

Applying steady flow energy equation to different processes

(i) For Turbine

$$h_1 = h_2 + W_T$$

$$\Rightarrow \boxed{W_T = h_1 - h_2}$$

(ii) For condenser

$$h_2 = Q_2 + h_3$$

$$\Rightarrow \boxed{Q_2 = h_2 - h_3}$$

(iii) For pump

$$h_4 = h_3 + W_P$$

$$\boxed{W_P = h_4 - h_3}$$

(iv) For Boiler

$$h_1 = h_4 + Q_1$$

$$\Rightarrow \boxed{Q_1 = h_1 - h_4}$$

Now efficiency of the cycle

$$\eta = \frac{Q_1 - Q_2}{Q_1} = \frac{(h_1 - h_4) - (h_2 - h_3)}{h_1 - h_4} = \frac{h_1 - h_4 - h_2 + h_3}{h_1 - h_4}$$

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

$$\boxed{\eta = \frac{W_T - W_P}{Q_1} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}}$$

For the process 3-4, $dh = Tds + vdp$

As process 34 is isentropic, so $ds = 0$

$$\Rightarrow dh = vdp$$

$$\Rightarrow \int_3^4 dh = \int_3^4 vdp$$

$$\Rightarrow [h_4 - h_3] = v(p_4 - p_3)$$

$$\Rightarrow h_4 - h_3 = v(p_b - p_c) = W_p$$

$$\Rightarrow h_4 = W_p + h_3$$

As pump work is very very less we can neglect pump work

$$\boxed{\eta = \frac{h_1 - h_2}{h_1 - h_4}}$$

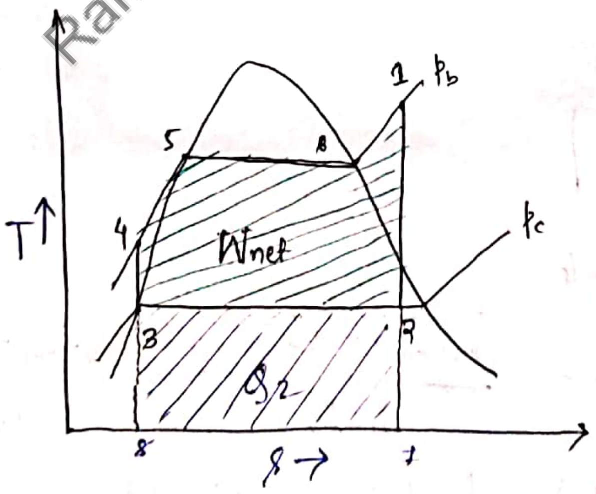
or

$$\eta = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4} = \frac{W_T - W_P}{Q_1} = \frac{W_T - W_p}{h_1 - h_4}$$

$$= \frac{(h_1 - h_2) - W_p}{h_1 - (h_3 + W_p)} = \frac{(h_1 - h_2) - W_p}{h_1 - h_3 - W_p}$$

Neglecting Pump work

$$\boxed{\eta = \frac{h_1 - h_2}{h_1 - h_3}}$$



Steam rate :- The capacity of a steam power plant is often expressed in terms of steam rate, which is defined as the rate of steam flow (kg/hr) required to produce unit shaft output (i.e. 1 kW)

$$\text{steam rate} = \frac{m_s}{m_s(W_T - W_P)} \qquad (m_s = \text{mass flow rate in kg/s})$$

$$= \frac{m_s \ \text{kg/s}}{m_s \times (W_T - W_P) \ \text{kg/s} \times \text{kJ/kg}}$$

$$= \frac{1}{1\,(W_T - W_P)} \times \frac{\text{kg}}{\text{kW.s}}$$

$$= \frac{1}{W_T - W_P} \times \frac{\text{kg}}{\frac{\text{kW}\cdot\text{h}}{3600}}$$

$$\boxed{\text{steam rate} = \frac{3600}{W_T - W_P} \ \frac{\text{kg}}{\text{kWh}}}$$

Power (P) Power devloped by the cycle

$$\boxed{P = m_s \times (W_T - W_P) \ \text{kW}}$$

Heat rate :- The cycle efficiency is some time expressed alternatively as heat rate, which is the rate of heat input (Q_1) required to produce unit work output (1kW).

$$\text{Heat rate} = \frac{m_s \times Q_1}{m_s \times (W_T - W_P)}$$

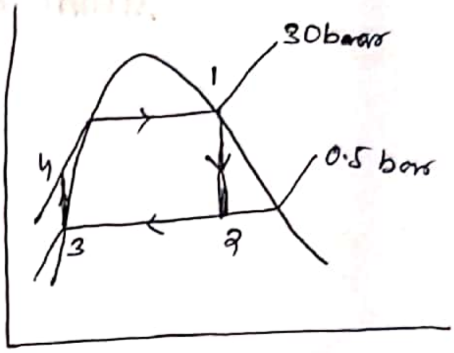
$$\boxed{\text{Heat rate} = \frac{3600 \times Q_1}{W_T - W_P} = \frac{3600}{\eta_{cycle}} \ \frac{\text{kJ}}{\text{kWhr}}}$$

Back work Ratio :- It is defined as the ratio betn pump work to the turbine work

$$\boxed{\text{Back work ratio} = \frac{W_P}{W_T}}$$

Q-1 In a power plant working on Rankine cycle dry saturated steam at boiler pr. of 30 bar expands to 0.5 bar in the turbine. Find the Rankine efficiency neglecting pump work

Soln Data given



boiler pr. $(P_b) = 30$ bar

condenser pr. $(P_c) = 0.5$ bar

$h_1 = h_g\big|_{30bar} = 2804.4$ KJ/kg

$s_1 = s_g\big|_{30bar} = 6.1840$ KJ/kg K

$h_3 = h_f\big|_{0.5bar} = 340.6$ KJ/kg $\quad s_f\big|_{0.5bar} = 1.0915$ KJ/kgK $\quad s_{fg}\big|_{0.5bar} = 6.5035$ KJ/kg K

$h_{fg}\big|_{0.5} = 2305.4$ KJ/kg

Process 1-2 isentropic process i.e $s_1 = s_2$

$\Rightarrow s_1 = s_f\big|_{0.5bar} + x\, s_{fg}\big|_{0.5bar}$

$\Rightarrow 6.184 = 1.0915 + 6.5035 \times x$

$\Rightarrow \boxed{x = 0.78}$

$h_2 = h_f\big|_{0.5bar} + x\, h_{fg}\big|_{0.5bar} = 340.6 + 0.78 \times 2305.4$

$= 2138.8$ KJ/kg

$$\eta_{Rankine} = \frac{h_1 - h_2}{h_1 - h_3} = \frac{2804.4 - 2138.8}{2804.4 - 340.6} = 0.27 = 27\% \quad \text{Ans}$$

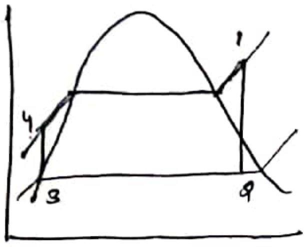
Q A power plant working on Rankine cycle starts at pr. 30 bar and 300°C expands isentropically to 0.7 bar in the turbine. Find the Rankine efficiency. (i) neglecting pump work (ii) considering pump work

Soln

Data given, $P_b = 30$ bar , $P_c = 0.7$ bar

$T_{sup.}$ at 30 bar $= 300°C = 573$K

T_{sat} at 30 bar $= 233.85°C = 506.85$ K



At 30 bar $\quad T_{sat} = 233.85°C$

$h_g = 2802.4$ KJ/kg $\quad s_g = 6.1840$ KJ/kg K

At 0·7 bar

$h_f = 163.3$ kJ/kg , $h_{fg} = 2409.2$ kJ/kg

$s_{fg} = 7.7607$ kJ/kgK , $s_f = 0.5592$ kJ/kgK $v_f = 0.00103$ m³/kg

Now $h_1 = h_g|_{30\,bar} + C_{ps}(T_{sup} - T_{sat})$

$= 28024 + 2.1\,(300 - 233.85)$

$$\boxed{h_1 = 2941.315 \text{ kJ/kg}}$$

Again $s_1 = s_2$

$\Rightarrow s_g|_{30\,bar} + C_{ps}\ln\left(\frac{T_{sup}}{T_{sat}}\right) = s_f|_{0.7\,bar} + x\, s_{fg}|_{0.7\,bar}$

$\Rightarrow 6.184 + 2.1\ln\left(\frac{573}{506.85}\right) = 0.5592 + x \times 7.7607$

$\Rightarrow 6.442 - 0.5592 = x \times 7.7607$

$\Rightarrow \frac{5.8828}{7.7607} = x \qquad \Rightarrow \boxed{x = 0.75}$

$h_2 = h_f|_{0.7\,bar} + x\, h_{fg}|_{0.7\,bar} = 163.3 + 0.75 \times 2409.2$

$\boxed{h_2 = 1970.2}$ $\qquad h_3 = h_f|_{0.7\,bar} = 163.3$ kJ/kg

$W_p = v_f(P_b - P_c) = 0.00103\,(30 - 0.7) \times \frac{10^5}{1000}$ kJ/kg

$= 3.0179$ kJ/kg

(i) Efficiency of plant considering pump work

$$\eta_{rankine} = \frac{h_1 - h_2 - W_p}{h_1 - h_3 - W_p} = \frac{2941.315 - 1970.2 - 3.0179}{2941.315 - 163.3 - 3.0179}$$

$$\boxed{\eta_{rankine} = 0.348 = 34.8\%}$$ Ans

(ii) Efficiency of plant without considering pump work

$$\boxed{\eta_{rankine} = 0.3495 = 34.95\%} = \frac{h_1 - h_2}{h_1 - h_3}$$

Q. Determine the efficiency of Rankine cycle using steam as working fluid in which the condenser pr is 10 KPa. The boiler pr. is 2MPa. The steam leaves the boiler as saturated vapours.

Soln Data given, Boiler pr. $(P_b) = 2 MPa = 20$ bar

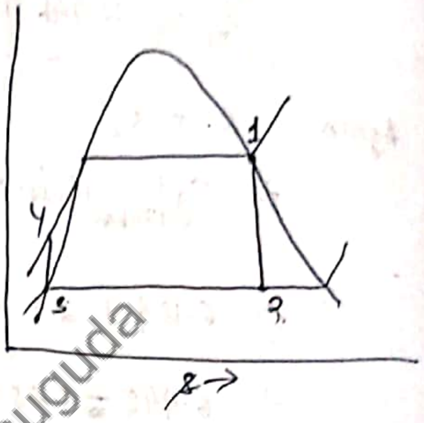
Condenser pr. $(P_c) = 10 KPa = 0.1$ bar

From steam table

At 20 bar.

$h_g = 2799.5$ KJ/kg $= h_1$

$s_g = 6.3409$ KJ/kgK $= s_1$



At 0.1 bar

$h_f = 191.83$ KJ/kg $= h_3$

$h_{fg} = 2392.8$ KJ/kg

$s_f = 0.6493$ KJ/kgK

$s_{fg} = 7.5009$ KJ/kgK

Process 1-2 is isentropic i.e. $s_1 = s_2$

$\Rightarrow 6.3409 = s_f|_{0.1bar} + x\, s_{fg}|_{0.1bar} = 0.6493 + x \times 7.5009$

$\Rightarrow \boxed{x = 0.758}$

$h_2 = h_f|_{0.1bar} + x\, h_{fg}|_{0.1bar} = 191.83 + 0.758 \times 2392.8$

$\boxed{h_2 = 2005.57 \text{ KJ/kg}}$

$W_p = v_f|_{0.1bar} \times (P_b - P_c) = 0.001010\,(20 - 0.1) \times 100$

$= 2.0099$ KJ/kg

$$\eta_{Rankine} = \frac{h_1 - h_2 - W_p}{h_1 - h_3 - W_p} = \frac{2799.5 - 2005.57 - 2.0099}{2799.5 - 191.83 - 2.0099}$$

$\boxed{\eta_{Rankine} = 0.3039 = 30.39\%}$ Ans. (with pump work)

$$\eta_{Rankine} = \frac{h_1 - h_2}{h_1 - h_3} = \frac{2799.5 - 2005.57}{2799.5 - 191.83}$$

$\boxed{\eta_{Rankine} = 0.3044 = 30.44\%}$ Ans. (neglecting pump work)

Characteristics of an Ideal Working fluid in Vapour Cycle :-

(1) The fluid should have high critical temp so that the saturation pr. of the maximum temp is relatively low. It should have large enthalpy of evaporation at that pr.

(2) The saturation pr. at the temp of heat rejection should be above atmospheric pr. so as to avoid the necessity of maintaining vaccuum in the condenser.

(3) The specific heat of liquid should be small so that little heat transfer is required to raise the liquid to the boiling point.

(4) The saturated vapour line of the T-s diagram should be steep, very close to the turbine expansion process (isentropic process) so that excessive moisture doesn't appear ~~after~~ ~~expan~~ during the expansion process. Presence of moisture in the turbine leads to turbine blade erosion

(5) The freezing point of the fluid should be below the room temp. so that it doesn't get solidified while flowing through pipelines.

(6) The fluid should be chemically stable and should not contaminate the materials of construction at any temperature.

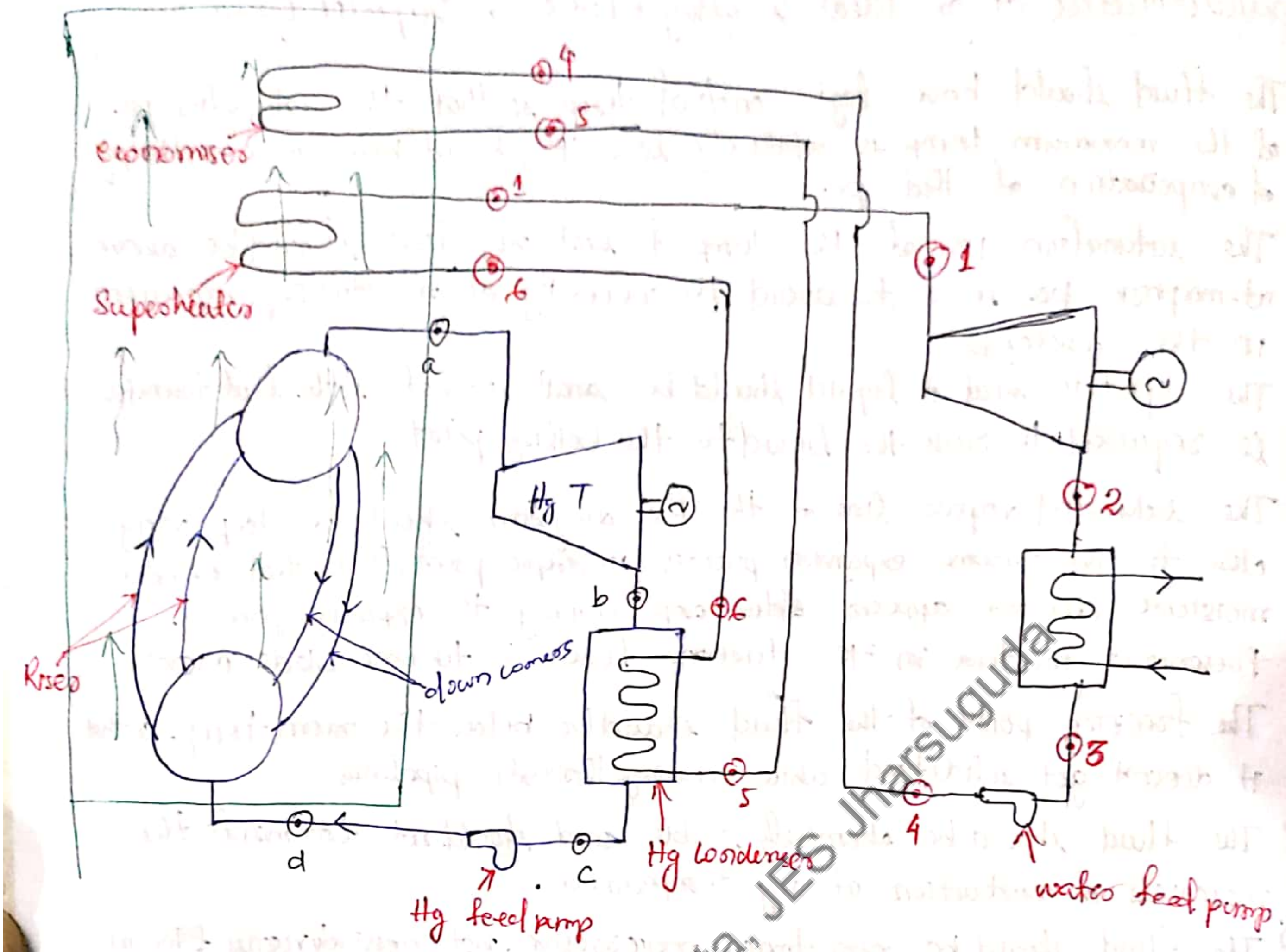
(7) The fluid should be non toxic, noncorrosive, not excessively viscous & low in cost.

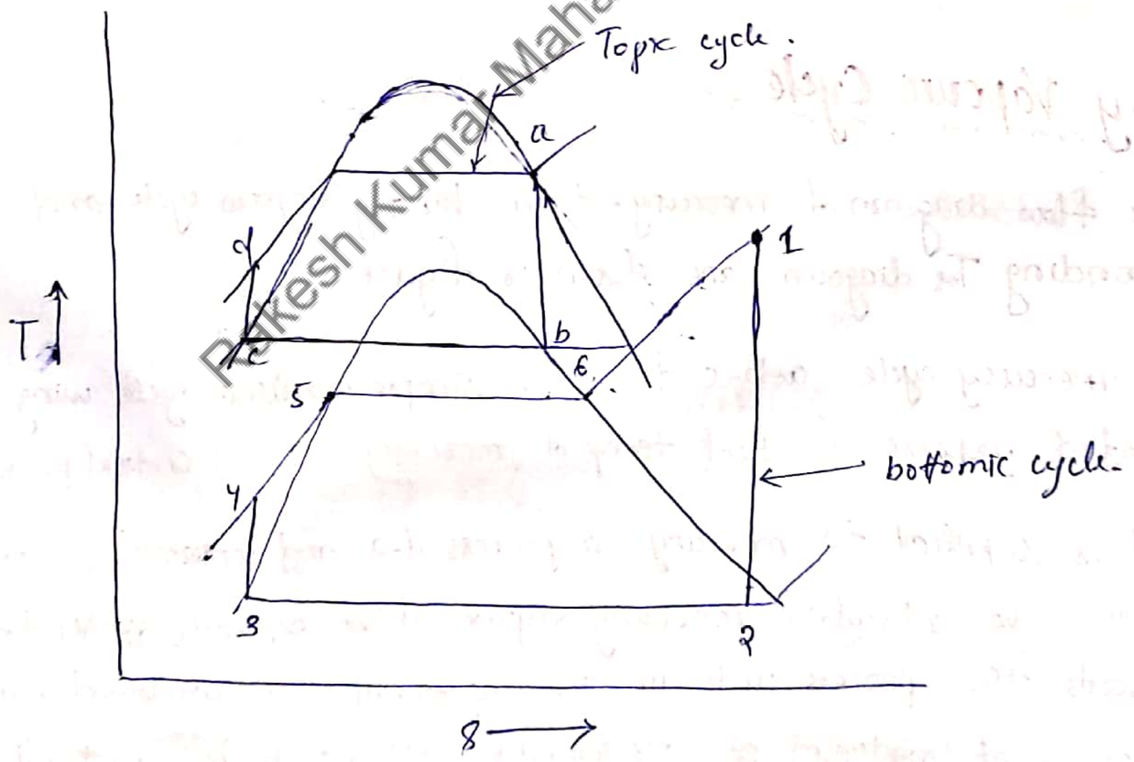
Binary Vapour Cycle :-

The flow diagram of mercury-steam binary vapour cycle and corresponding T-s diagram are shown in figure

→ The mercury cycle a-b-c-d is a simple rankine cycle using saturated vapour. (Critical temp. of mercury 1750K & critical pr. 172 MPa)

→ Heat is supplied to mercury in process d-a and converted to saturated vapour. The saturated mercury vapour then expands in the turbine represents the process a-b in T-s diagram and then condensed in the condenser at condenser pr. represented by the process b-c in T-s diagram. The Hg feed pump then pump the Hg to boiler pr. represented by the process cd in T-s diagram & hence completes the cycle.


Economiser
Superheater
Riser
down comers
Hg T
Hg condenser
Hg feed pump
water feed pump
a
b
c
d
1
2
3
4
5
6
4
5
1
6


Topic cycle.
bottomic cycle.
T
s
a
b
c
d
e
1
2
3
4
5

→ The heat rejected by the mercury during condensation is transfered to boil the water ~~from~~ to form saturated vapour (process 5-6). The saturated vapour is superheated in the furnace (process 6-1) by the superheater. The superheated steam expands in the turbine (process 1-2) and is then condensed (process 2-3). The feed water is then pumped (process 3-4) & heated till it is saturated liquid in the economiser (process 4-5)

⇒ Let 'm' be the flow rate of mercury in the mercury cycle for per kg of steam circulating in the steam cycle. Then for 1 kg of steam

$$\text{Heat supplied } (Q_1) = m(h_a - h_d) + (h_5 - h_4) + (h_1 - h_6)$$

$$\text{Heat rejected } (Q_2) = 1(h_2 - h_3)$$

$$\text{Turbine work } (W_T) = m(h_a - h_b) + 1(h_1 - h_2)$$

$$\text{Pump work } (W_P) = m(h_d - h_c) + 1(h_4 - h_3)$$

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

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

Considering the energy balance of Hg condenser

$$m(h_b - h_c) = h_6 - h_5.$$

$$\text{Steam rate} = \frac{3600}{W_{net}} \text{ kg/kWhr}$$

-: HEAT TRANSFER :-

The subject dealing with the rate at which heat flow occurs is called heat transfer. It is observed from our experience that heat flows from the region at high temp. to the region at low temperature. The region may be solid, liquid or gaseous state.

Modes of Heat Transfer:

There are three modes of heat transfer
(i) conduction
(ii) convection
(iii) radiation

(i) Conduction :- Conduction is the flow of heat energy in a substance due to exchange of energy betn molecules having more energy and molecules having less energy. The heat transfer medium may be solid, liquid or gas does not matter. Conduction occurs due to Lattice vibration (in solids) or motion of free electrons (for metalic solids) or due to collision of molecules (in liquids or gases)

(ii) Convection :- It is the combination effect of conduction & fluid motion. If there is no fluid motion, then the heat transfer between a solid and its adjacent fluid is pure conduction. When a temperature difference produces a density difference, which results in mass movement, the process is called free convection or natural convection. When the mass motion of the fluid is caused by an external device like a pump, compressor or blower or fan, the process is called as forced convection.

(iii) Radiation :- The third mode of heat transfer is thermal radiation. All the surfaces at finite temperature emit energy in the form of electromagnetic waves (or photons) as a result of the change in electron configuration of the atoms or molecules. This mode of heat transfer does not require the presence of material medium. The energy transferred by radiation is fastest and it can also travel in vacuum.

Fourier Law of Heat conduction :

The heat flux ($\frac{\dot{Q}}{A}$) by conduction in a particular direction is directly proportional to the gradiation of temperature in that direction

$$\frac{\dot{Q}}{A} \propto -\frac{\partial T}{\partial n}$$

$$\Rightarrow \boxed{\frac{\dot{Q}}{A} = -K\frac{\partial T}{\partial n}}$$

$\frac{\dot{Q}}{A}$ = heat flux by conduction in $\frac{W}{m^2}$

K = Thermal conductivity - $\frac{W}{mK}$

The -ve sign is being used because heat flows from a high to low temp region & the slope $\partial t/\partial n$ is negative

→ In cartesian co-ordinate Fourier law will be

$$\left.\frac{\dot{Q}}{A}\right)_x = -K\frac{dT}{dx}$$

$$\left.\frac{\dot{Q}}{A}\right)_y = -K\frac{dT}{dy}$$

$$\left.\frac{\dot{Q}}{A}\right)_z = -K\frac{dT}{dz}$$

→ In cylindrical co-ordinate Fourier law will be

$$\left.\frac{\dot{Q}}{A}\right]_r = -K\frac{\partial T}{dr}$$

$$\left.\frac{\dot{Q}}{A}\right]_\theta = -K\frac{dT}{d\theta}$$

$$\left.\frac{\dot{Q}}{A}\right|_z = -K\frac{dT}{dz}$$

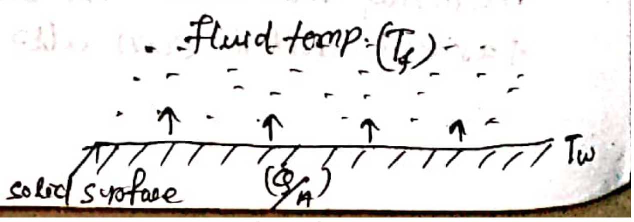
⇒ Generally for solids and liquids $K = f(T)$
for gases & vapours $K = f(p, T)$

Newton's Law of Cooling :-

It states that heat flux by convection from the wall to the fluid or vice versa is proportional to the temperature difference $(T_w - T_f)$

$$\frac{\dot{Q}}{A} \propto (T_w - T_f)$$

$$\Rightarrow \boxed{\frac{\dot{Q}}{A} = h(T_w - T_f)}$$



$\left(\frac{\dot{Q}}{A}\right)$ = heat flux by convection → $\frac{W}{m^2}$

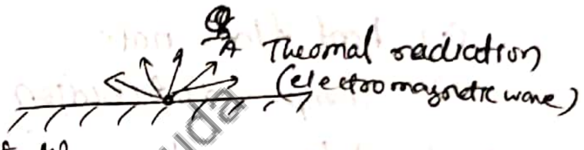
h = heat transfer co-efficient → $\frac{W}{m^2 K}$ (it is always a +ve quantity)

Stiffan-Boltzmann Law:

It states that the rate of radiation heat transfer per unit area (heat flux) from a black surface is directly proportional to fourth power of the absolute temp. of the surface and is given by

$$\frac{\dot{Q}}{A} \propto (T^4)$$

$$\Rightarrow \boxed{\frac{\dot{Q}}{A} = \sigma T_s^4}$$



σ = Stefan Boltzmann constant = $5.67 \times 10^{-8} \frac{W}{m^2 K}$

$\frac{\dot{Q}}{A}$ = Thermal radiation heat flux $\left(\frac{W}{m^2}\right)$

T_s = absolute temp of the surface in K.

The heat flux emitted by ~~the~~ a real surface is less than that of a black surface & is given by $\boxed{\frac{\dot{Q}}{A} = \sigma \varepsilon T_s^4}$

where ε = a radiative property of the surface called emissivity.

Thermal conductivity:- It is the property of the material and is defined as the ability of the material to conduct heat through it. It can also be defined as the rate of heat transfer through a unit thickness of material per unit area per unit temperature difference.

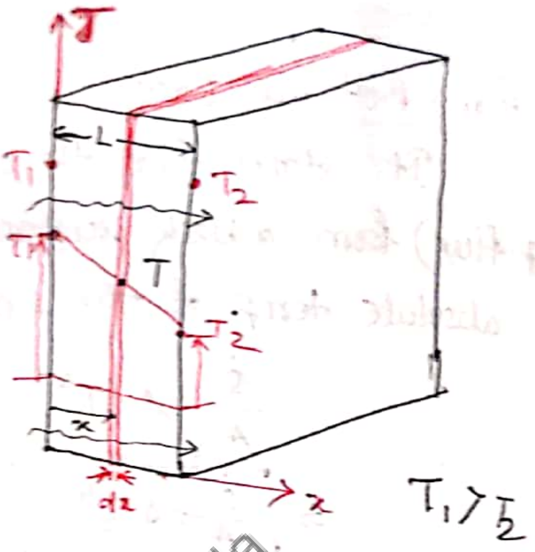
→ A Large value of thermal conductivity indicates that the material is a good conductor & low value indicates that material is a poor heat conductor.

Thermal conductivity (k) → $\frac{W}{mK}$

One-Dimensional Steady State Situations

(1) Infinite slab / Plane wall :-

Let us consider a plane wall of thickness L. Its left face at $x=0$, is at temp. T_1, and right face at $x=L$, is at temp. T_2. The wall has thermal conductivity K



We have to find out
(1) heat flow rate
(2) Temp. distribution.

steady state so temp doesn't change with time in direction (x)
so $T = f(x)$

(i) Applying Fourier Law

$$\frac{\dot{Q}}{A} = -K\frac{dT}{dx} \qquad \frac{\dot{Q}}{A} = \text{const.}$$

$$\Rightarrow \frac{\dot{Q}}{A}\int_0^L dx = -K\int_{T_1}^{T_2} dT$$

$$\Rightarrow \frac{\dot{Q}}{A}[x]_0^L = -K[T]_{T_1}^{T_2}$$

$$\Rightarrow \frac{\dot{Q}}{A}\times L = K(T_1-T_2)$$

$$\Rightarrow \boxed{\frac{\dot{Q}}{A} = \frac{K}{L}(T_1-T_2)} \longrightarrow (1)$$

(ii) Temp. distribution - Let us consider a small elemental area of thickness dx at a distance 'x'. Let the temp. at the strip is T

so,
$$\frac{\dot{Q}}{A}\int_0^x dx = -K\int_{T_1}^{T} dT$$

$$\Rightarrow \frac{\dot{Q}}{A}\times x = K(T_1 - T)$$

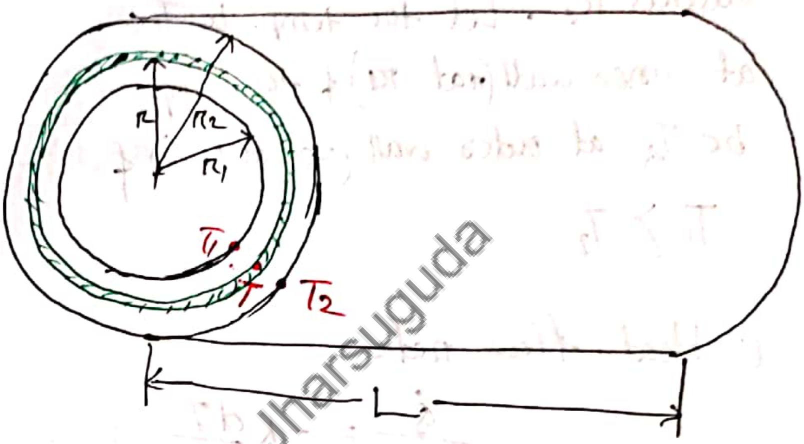
$$\Rightarrow \frac{\dot{Q}}{A} = \frac{K}{x}(T_1 - T)$$

$$\Rightarrow \frac{K}{L}(T_1 - T_2) = \frac{K}{x}(T_1 - T) \qquad (\text{from eq}^n\ 1)$$

$$\Rightarrow \boxed{\frac{T_1 - T}{T_1 - T_2} = \frac{x}{L}} \quad \text{—— (ii)}$$

(ii) Hollow Cylinder :-

Consider a hollow cylinder of inner and outer radii r_1 & r_2 respectively & of Length L. The temp is T_1 at inner wall (i.e at r_1) & T_2 at outer wall (i.e at r_2) & $T_1 > T_2$



(i) The radial heat flow rate , $T = f(r)$

$$\frac{\dot{Q}}{A} = -K\frac{dT}{dr}$$

$$\Rightarrow \frac{\dot{Q}}{2\pi r L} = -K\frac{dT}{dr} \Rightarrow \frac{\dot{Q}}{2\pi L}\int_{r_1}^{r_2}\frac{dr}{r} = -K\int_{T_1}^{T_2} dT$$

$$\Rightarrow \frac{\dot{Q}}{2\pi L}\ln\left(\frac{r_2}{r_1}\right) = K(T_1 - T_2)$$

$$\Rightarrow \boxed{\dot{Q} = \frac{2\pi L K (T_1 - T_2)}{\ln(r_2/r_1)}}$$

(ii) Temp. distribution :- Let us consider a small elemental ring of thickness dr at a radial distance r. Let the temp at the strip is T

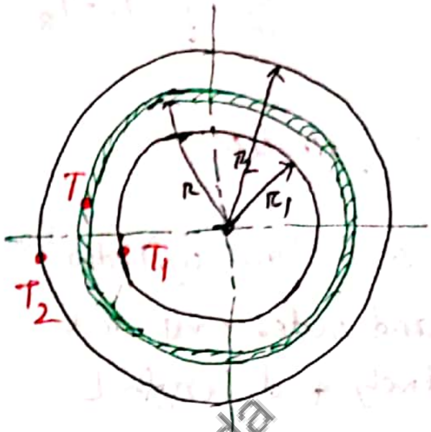
So
$$\frac{\dot{Q}}{2\pi L}\int_{r_1}^{r}\frac{dr}{r} = -K\int_{T_1}^{T} dT$$

$$\Rightarrow \frac{\dot{Q}}{2\pi L}\ln(r/r_1) = K(T_1 - T)$$

$$\Rightarrow \frac{\cancel{2\pi L}\,\cancel{K}(T_1 - T_2)}{\ln(r_2/r_1) \times \cancel{2\pi L}} \times \ln(r/r_1) = \cancel{K}(T_1 - T)$$

$$\Rightarrow \boxed{\frac{T_1 - T}{T_1 - T_2} = \frac{\ln(r/r_1)}{\ln r_2/r_1}}$$

(iii) Hollow Sphere :-



Consider a hollow sphere of internal radius r_1 & outer radius r_2. Let the temp. be T_1 at inner wall (i.e at r_1) & the temp be T_2 at outer wall (i.e at r_2) & $T_1 > T_2$

(i) Heat flow rate

$$\frac{\dot{Q}}{A} = -k\frac{dT}{dr}$$

$$\Rightarrow \frac{\dot{Q}}{4\pi r^2} = -k\frac{dT}{dr} \quad \Rightarrow \frac{\dot{Q}}{4\pi r^2}\,dr = -k\,dT$$

Integrating

$$\Rightarrow \frac{\dot{Q}}{4\pi}\int_{r_1}^{r_2}\frac{dr}{r^2} = -k\int_{T_1}^{T_2} dT$$

$$\Rightarrow \frac{\dot{Q}}{4\pi}\left[-\frac{1}{r}\right]_{r_1}^{r_2} = k(T_1 - T_2)$$

$$\Rightarrow \frac{\dot{Q}}{4\pi}\left(\frac{1}{r_1} - \frac{1}{r_2}\right) = k(T_1 - T_2)$$

$$\Rightarrow \frac{\dot{Q}}{4\pi}\left(\frac{r_2 - r_1}{r_1 r_2}\right) = k(T_1 - T_2)$$

$$\Rightarrow \boxed{\dot{Q} = \frac{4\pi r_1 r_2 k(T_1 - T_2)}{r_2 - r_1}}$$

(ii) Temperature distribution :- Let us consider a small elemental area at a radial distance r

so, $$\frac{\dot{Q}}{4\pi}\int_{r_1}^{r}\frac{dr}{r} = -k\int_{T_1}^{T} dT$$

$$\Rightarrow \frac{\dot{Q}}{4\pi}\left(\frac{1}{r_1}-\frac{1}{r}\right) = K(T_1-T)$$

$$\Rightarrow \frac{\dot{Q}}{4\pi}\left(\frac{r-r_1}{r_1 r}\right) = K(T_1-T)$$

$$\Rightarrow \frac{\cancel{4\pi}\, r_1 r_2 \cancel{K}(T_1-T_2)}{\cancel{4\pi}\,(r_2-r_1)} \times \frac{(r-r_1)}{r_1 r} = \cancel{K}(T_1-T)$$

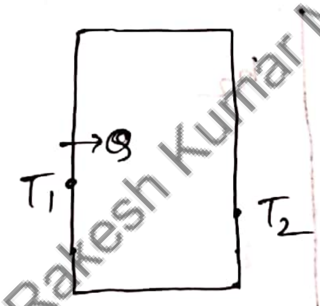
$$\Rightarrow \frac{T_1-T}{T_1-T_2} = \frac{r_1 r_2 (r-r_1)}{r_1 r (r_2-r_1)}$$

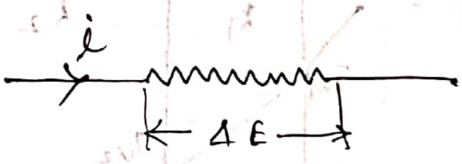
$$\Rightarrow \boxed{\frac{T_1-T}{T_1-T_2} = \frac{r_2(r-r_1)}{r(r_2-r_1)}}$$

Combined modes of Heat transfer & Thermal Resistance

The three modes of heat transfer have been explained separately earlier. In actual practice, it is very rare that only one mode is involved in transfer of heat energy instead combined modes are involed.

Thermal resistance





electrical resistance $R = \frac{\Delta E}{i}$

Thermal resistance

$$R_{th} = \frac{T_1 - T_2}{Q}$$

⇒ For infinite slab

$$Q = \frac{KA}{b}(T_1 - T_2)$$

$$\boxed{R_{th} = \frac{T_1 - T_2}{Q} = \frac{b}{KA}}$$

⇒ For heat flowing from a surface to fluid (convective heat transfer)

$$\boxed{R_{th} = \frac{1}{hA}}$$

⇒ For infinite hollow cylinder

$$\dot{Q} = \frac{2\pi L K (T_1 - T_2)}{\ln(r_2/r_1)} \Rightarrow \frac{\ln(r_2/r_1)}{2\pi L K} = \frac{T_1 - T_2}{Q}$$

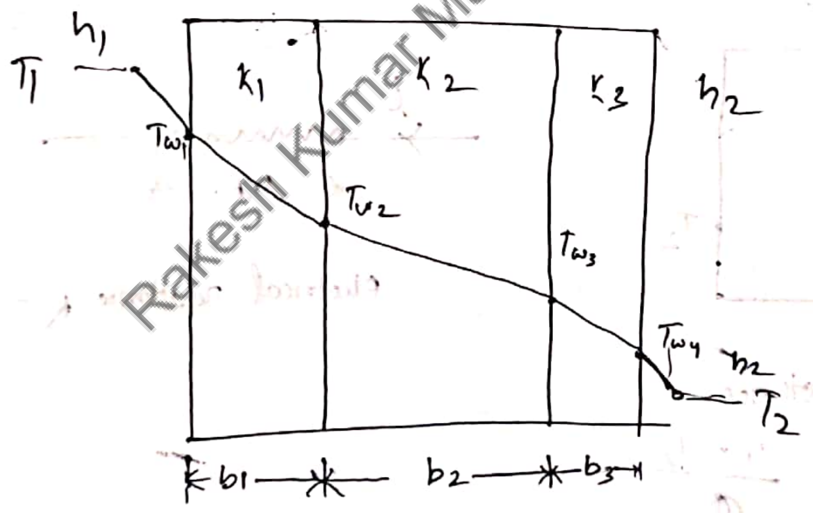
$$\Rightarrow \boxed{R_{th} = \frac{\ln(r_2/r_1)}{2\pi K L}}$$

⇒ For infinite hollow sphere

$$\dot{Q} = \frac{4\pi r_1 r_2 K (T_1 - T_2)}{r_2 - r_1} \Rightarrow \frac{r_2 - r_1}{4\pi r_1 r_2 K} = \frac{T_1 - T_2}{\dot{Q}}$$

$$\Rightarrow \boxed{R_{th} = \frac{r_2 - r_1}{4\pi r_1 r_2 K}}$$

Thermal Resistance of an Infinite composite slab :-



$$\frac{Q}{A} = h_1 (T_1 - T_{w_1}) \Rightarrow T_1 - T_{w_1} = \frac{Q}{A} \times \frac{1}{h_1}$$

$$\frac{Q}{A} = \frac{K_1}{b_1}(T_{w_1} - T_{w_2}) \Rightarrow T_{w_1} - T_{w_2} = \frac{Q}{A} \times \frac{b_1}{K_1}$$

$$\frac{Q}{A} = \frac{K_2}{b_2}(T_{w_2} - T_{w_3}) \Rightarrow T_{w_2} - T_{w_3} = \frac{Q}{A} \times \frac{b_2}{K_2}$$

$$\frac{Q}{A} = \frac{K_3}{b_3}(T_{w_3} - T_{w_4}) \Rightarrow T_{w_3} - T_{w_4} = \frac{Q}{A} \times \frac{b_3}{K_3}$$

$$Q/A = h_2 (T_{w_4} - T_2) \Rightarrow T_{w_4} - T_2 = \frac{Q}{A} \times \frac{1}{h_2}$$

$$\Rightarrow T_1 - T_2 = \frac{Q}{A}\left(\frac{1}{h_1} + \frac{b_1}{K_1} + \frac{b_2}{K_2} + \frac{b_3}{K_3} + \frac{1}{h_2}\right)$$

$$\Rightarrow \frac{T_1 - T_2}{Q} = \frac{1}{A}\left[\frac{1}{h_1} + \frac{b_1}{K_1} + \frac{b_2}{K_2} + \frac{b_3}{K_3} + \frac{1}{h_2}\right]$$

$$\Rightarrow \boxed{R_{th} = \frac{1}{A}\left(\frac{1}{h_1} + \frac{b_1}{K_1} + \frac{b_2}{K_2} + \frac{b_3}{K_3} + \frac{1}{h_2}\right)}$$

Overall heat transfer co-efficient

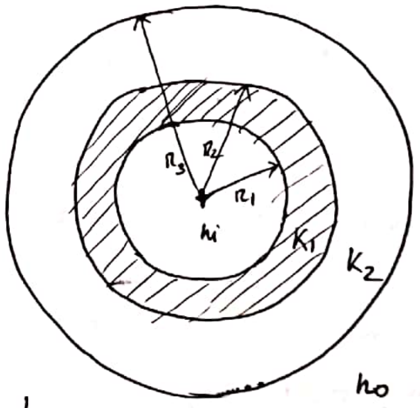
$$Q = UA(T_1 - T_2)$$

for infinite composite slab

$$U = \frac{1}{\frac{1}{h_1} + \frac{b_1}{K_1} + \frac{b_2}{K_2} + \frac{b_3}{K_3} + \frac{1}{h_2}}$$

Thermal resistance fo ~~an~~ long Hollow composite cylinder :-

$$\dot{Q} = \frac{2\pi L K(T_1 - T_2)}{\ln(r_2/r_1)}$$



$$\frac{Q}{A_i} = h_i(T_i - T_1) \Rightarrow (T_i - T_1) = \frac{Q}{A_i} \times \frac{1}{h_i}$$

$$\dot{Q} = \frac{2\pi L K_1(T_1 - T_2)}{\ln(r_2/r_1)} \Rightarrow T_1 - T_2 = \frac{Q \cdot \ln(r_2/r_1)}{2\pi L K_1}$$

$$\dot{Q} = \frac{2\pi L K_2(T_2 - T_3)}{\ln(r_3/r_2)} \Rightarrow T_2 - T_3 = \frac{Q \cdot \ln(r_3/r_2)}{2\pi L K_2}$$

$$\frac{\dot{Q}}{A_o} = h_o(T_3 - T_o) \Rightarrow T_3 - T_o = \frac{Q}{A_o} \times \frac{1}{h_o}$$

$$\Rightarrow T_1 - T_o = Q\left(\frac{1}{A_i h_i} + \frac{\ln(r_2/r_1)}{2\pi L K_1} + \frac{\ln(r_3/r_2)}{2\pi K_2 L} + \frac{1}{A_o h_o}\right)$$

$$\Rightarrow \frac{T_i - T_o}{Q} = \frac{1}{2\pi r_1 L h_i} + \frac{\ln(r_1/r_2)}{2\pi L K_1} + \frac{\ln(r_2/r_3)}{2\pi L K_2} + \frac{1}{2\pi r_3 L h_o}$$

$$\Rightarrow R_{ts} = \frac{1}{2\pi r_1 L h_i} + \frac{\ln(r_1/r_2)}{2\pi L K_1} + \frac{\ln(r_2/r_3)}{2\pi L K_2} + \frac{1}{2\pi r_3 L h_o}$$

Overall heat transfer co-effrient

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

Principle of Convection :-

Previously we have discussed the heat transfer by conduction & have considered the convection only to the extent of boundary conditions. As discussed earlier, the convection is the mode of heat transfer, in which energy is transferred betn the surface & the moving fluid.

→ The convection heat transfer comprises of two mechanisms. First is the transfer of energy due to the random molecular motion (diffusion) and second is, the energy transfer by bulk or macroscopic motion of fluid.

→ The molecules of the fluid are moving collectively or as aggregates & thus carry energy from a high temp region to a low temp. region

→ Convection heat transfer may be classified according to the nature of fluid flow as (i) free or natural convection
(ii) forced convection.

Free convection :- when the fluid is heated, the density gradients are devloped and ~~reds~~ results into buoyancy force which induces convection. Such situation is refferred as natural or free convection.

→ In free convection, the fluid motion's set up by buoyancy forces are much smaller than those associated with forced convection.

→ There are many situations, where the heat is transferred by free convection to the surroundings air. Ex- Heat transfer from a heater to heat room, heat transfer from pipes. electric transformers etc.

Forced convection :- Convection heat transfer is complecated since it involves fluid motion as well as heat conduction. The fluid motion enhances the heat transfer i.e higher the velocity higher the heat transfer rate.

Physical Significance of the convection : Dimensionless Parameters

1. Reynolds Number :- It is the ratio of inertia forces to viscous forces in the velocity boundary layer. It is used in forced convection and approximated as

$$Re = \frac{\text{Inertia forces}}{\text{Viscous forces}} = \frac{\rho u_\infty L_c}{\mu} = \frac{u_\infty L_c}{\nu}$$

where L_c = characteristics length of the geometry.
= distance from the leading edge, in the flow direction for a flat plate and d for a cylinder or sphere (m)
u_∞ = free stream velocity. (~~kg/m~~ m/s)
ρ = fluid density (kg/m³)
μ = dynamic viscosity (Ns/m²)
ν = kinematic viscosity (m²/s)

2. Grashof Number :- It is the ratio of buoyancy forces to the viscous forces acting in the fluid. It is used in free convection & its role is same as that of Reynolds no. in forced convection. It characterises the type of boundary layer developed in natural convection heat transfer.

$$Gr = \frac{\text{Buoyancy forces}}{\text{Viscous forces}} = \frac{g\Delta\rho V}{\rho\nu^2} = \frac{g\beta\Delta T L_c^3}{\nu^2}$$