Department of Mechanical Engineering
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LECTURE NOTES

Name of the Subject: Power station Engineering

Semester: Sixth **Year:** Third

Name of the Faculty: Deepak Kumar Hota

05/08/08

Source of Energy

The source of energy is defined as the quantity or substance from which a continuous supply of energy is possible. Broadly it is divided into

(1) Renewable source of energy
(2) Non-renewable " "

Renewable source of energy

fill up again

Solar, bio-mass, water, wind

→ Renewable energy is generally defined as energy that is collected from resources which are naturally replenished on a human timescale, such as sunlight, wind, rain, tides, waves, & geothermal heat

Non-renewable or finite Resource.

fossil fuels, agricultural waste, fire wood

→ These are the resources that are consumed much faster than nature can create them.

→ fossil fuels (such as coal, petroleum & natural gas), types of nuclear power (uranium)..

Solar energy

(Haryana)

Wind power plant

Kanyakumari

Arabian Sea.

Wave energy (Tidal Power Plant)
→ Gujarat set to develop India's 1st

Thermal power plant

06/08/08

Requirement for Site Selection for Power Plant

(I) The site should not be prone to ~~aob~~ earthquan or any geographical disorder.

(II) The site should be located at an optimal distance from the regions, where raw materials are available.

(III) Adequate transfer facilities basically road & rail service should be available.

(IV) To dispose the waste materials large water bodies like lakes, rivers & oceans should be preferred.

(V) The site should be farther from the socities or residential areas to avoid pollution.

(VI) Availability of skilled human resource should be given priority

Layout of a Thermal Power Plant

A thermal power plant basically consists of four devices.
(I) Boiler (II) Turbine (III) Condenser (IV) Heat Pum

- Apart from this other accessories like evaporator, super heater, reheater, economiser, Air reheater & electric precipitator can be assembled to increase the plant efficiency.
- The thermal power plant works on Rankine cycle.

Brief description of each devices

Coal conveyor belt

Coal from the coal storage transfer to the boiler end by means of a conveyor belt

Stoker

By this device the coal from the conveyor belt will be taken to the coal grinding m/c.

Coal Grinding machine

Before the fuel (coal) is supplied to the furnace, it should be grinded to the fine particles for smooth combustion. & this crushing is done with the help of coal grinding m/c or. pulveriser.

This ~~polvarciser~~ pulveriser is of two types.

(I) Tube & mill ~~polvariser~~ pulveriser

(II) Ball & ring ~~polvariser~~ pulveriser.

Boiler :-

The working substance in the thermal power plant is water. The feed water will enter into the boiler, where by gaining the latent heat of vapourisation from the combustion of the fuel inside the furnace, the change of phase from liquid to vapour is occured. The water is heated to a predesigned temperature. The boilers are classified into two types.

(I) Water tube boiler

(II) fire tube boiler.

Water tube boiler

In this kind of boiler the feed water is allowed to pass through the tube (horizontal, vertical or inclined) & the hot gases or flue gases pass over the tubes carrying waters. Due to the convective heat transfer the water is converted to steam & collected by a container at the top side of the boiler called dome.

→ Here generally the counter flow arrangement is preferred for effective heat transfer. As more area of the water tubes are exposed to the hot gases, the heat transfer rate is more in comparision to the fire tube boilers. Also any variable demand ~~in~~ of the steam supply can be easily achieved by this kind of boiler. The working pressure is somewhere around 200 kg/cm^2. . More explosive

Fire Tube Boiler

As the name indicates, the hot gases is allowed to pass through the tubes, which are surrounded by the feed water. As the water vol. is more in comparision to the hot gases. The heat transfer rate is relatively low. The working pressure ~~is~~ of these boilers is very low, due to ~~so~~ safety consideration somewhere around 15 to 20 kg/cm^2.

Evaporator

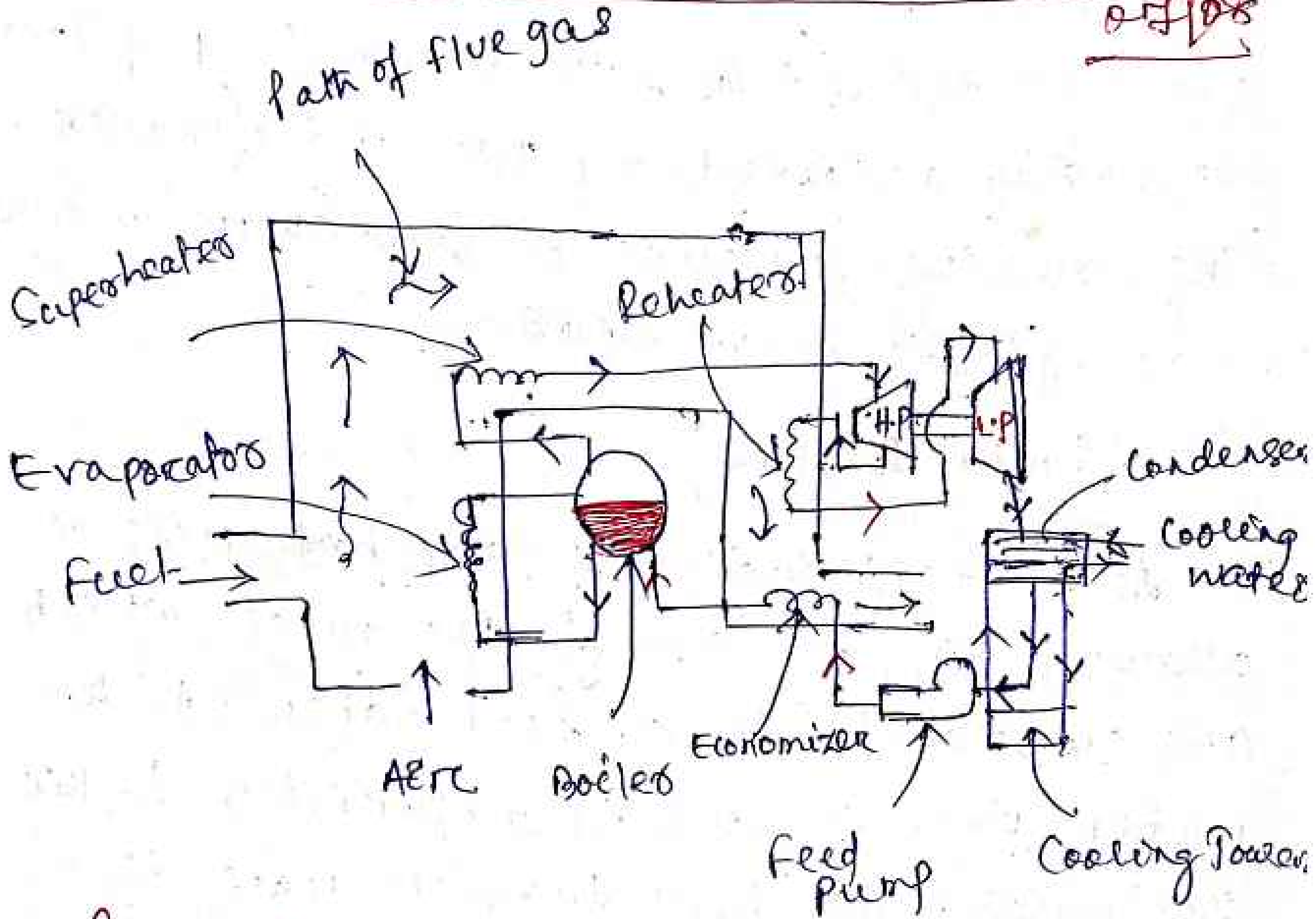
Now-a-days all the modern boilers are attached with an evaporator, so that ~~the steam before entering the turbine can be superheat~~

some part of the feed water can be ~~pto~~ bled (partwise extraction) & allowed to pass through the evaporator located inside the furnaces to minimize the heat supply to the cycle (i.e. Rankine cycle).

Superheater

Layout of a Power Plant

07/08



Superheater

Boilers or Steam generator

The uses of steam

(I) utilized in Power plant

(II) Chemical & textile industries

(III) For the heating purpose

Classification of boilers

(1) Depending on the orientation of axis of boiler.

- vertical
- horizontal
- Inclined

(2) Fire tube or water tube

Fire tube ↓
Cochran
Lankashire
Cornish
Locomotive boiler

Water tube ↓ Externally fired
Babcock & Wilcox
Stirling boiler

(3) Externally fired or, Internally fired

If the furnace is integral to boiler then it is called internally fired boiler.

(4) Depending on the type of flow of water

→ Forced flow → ex:- Velox, Lamont, Benson
→ Natural flow → Lankashire.

(5) Depending upon the working pressure.

→ High pressure. (≥ 80 bar) eg. → Babcock & Wilcox, velox, Lamont, Benson
→ Low pr. (< 80 bar) (fire tube)
↓ Cochran, cornish, Lancashire ↓ Lankashire

(6) Depending upon the transportability

→ Stationary
→ Portable (or, Mobile)
↑ Temporary uses

locomotive boiler & the portable boiler.

(7) Depending on the number of tubes (only for fire tube boiler)

- Single tubular
- Multi tubular — Lankashire, Cochran.

Difference bet^n. fire tube & water tube boiler.

Criteria	Fire tube	Water tube
(1) Pos^n. of water & hot gases	Hot gases will flow through the tubes & water will flow outside	Water will flow inside & hot gas outside
(2) Mode of firing	Internally fired	externally fired
(3) Operating pr.	Low operating pr. (15 bar)	High operating pr. More than 80 bar
(4) Rate of prod^n. of steam	Less	More
(5) Floor area	More	Less
(6) Construction	difficult	Simple
(7) Transportation	difficult	easy
(8) accessibility of various part	~~easy~~ ,,	Simple
(9) Requirement of skilled labour	Less skilled	More skilled
(10) The Rate of product^n. of steam for same power	More	Less

09/08/08

Selection of Boiler

(i) On the working pressure.
(ii) Quality of the steam & the rate of ~~heat~~ steam
(iii) production.

→ Floor area or space available
→ Availability of the fuel.
→ ~~Acce~~
→ Accessibility & Maintenance
→ Weight of the boiler & transportability

Requirement of a good boiler

(i) It should produce max^m amount of steam in the reqd. rate & quality ~~to~~ of min^m expense of energy. It should be good in handling, accessability, ~~an~~ transportability & maintenance.

(ii) Less prone

(iii) operating parameters pr. & temp.

(iv) Less skill ~~is~~ to be reqd. to operate it.

Boiler Parts

(i) Boiler Shell

It is made up of 3 to 4 ~~steam pl~~ steel plates when to give a cylindrical shape by welding & riveting. It is covered at two ends by two flat plate.

(i) Grate

This is the platform upon which the fuel in the furnace is burnt. Cast iron bars are arranged parallely to form the grate surface. These are so arranged as the air for combustion of fuel can easily pass over the fuel. The ash produced by the combust? is collected over the ashpit.

Furnace

This is a refractory heat wall designed or constructed in a manner so that effective combustion of the fuel & the smooth draught (flow of air) can be obtained.

Setting :- The main fun. of the setting is to confined the heat upto the boiler.

Water & Steam space :-

The part of the shell occupied by the feed water is called as water space & the rest available for the steam is called as steam space.

Boiler Mountains :- The devices like pressure gauge, steam stop valve, safety valv water level box, heat check valve, blow off cock, etc. are called as boiler mountains without which the boiler can't be operated safely.

The Boiler Accesorcies

Equipments like economizers, superheater, preheater, evaporator, electrostatic precipitator, etc. are called as boiler accesorcies. which are used to increase the eff. of boiler

Smoke box :- This is a rectangular box type constrn, which is integral to the furnace for the passage of hot flue gases.

Foming :- This is the formn. of water bubble at the surface of the boiler due to the high surface tension of water.

Scaling :- Deposition of impurities or sediments inside boiler surface due to hardness of water.

Blowing off :- Removal of mud, impurities from the boiler is called as blowing off & this is done by opening the blow off cock or valve

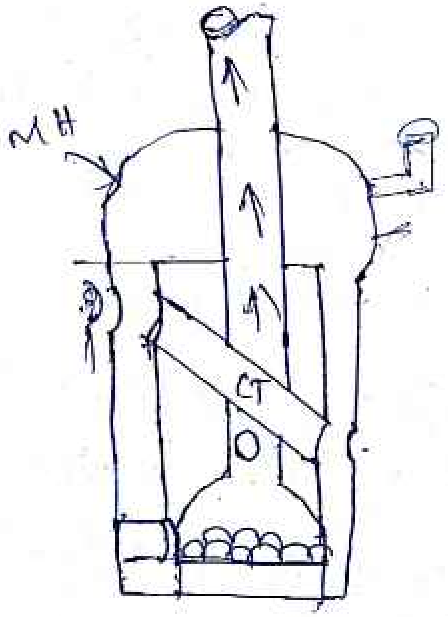
Lagging :- Insulation of external surface of boiler & combustn. chamber by covering with a layer of asbestos or magnesia is called as lagging.

Refractory :- The thermal insulating brick used for the lining of furnace & combustn. chamber is called as refractory.

Constrⁿ & working principle of different kinds of boiler.

Fire Tube boiler -

Simple vertical fire Tube boiler



G → Grate
AP → Ash pit
FB → Fire box
C → Chimney
MH → Man hole
HH → Hand hole
FD → Fire Dome
PG → Pr. Gauge
SV → Saftey valve
SSV → Steam stock valve
WLG → Water level gauge
CT → Cross tubes

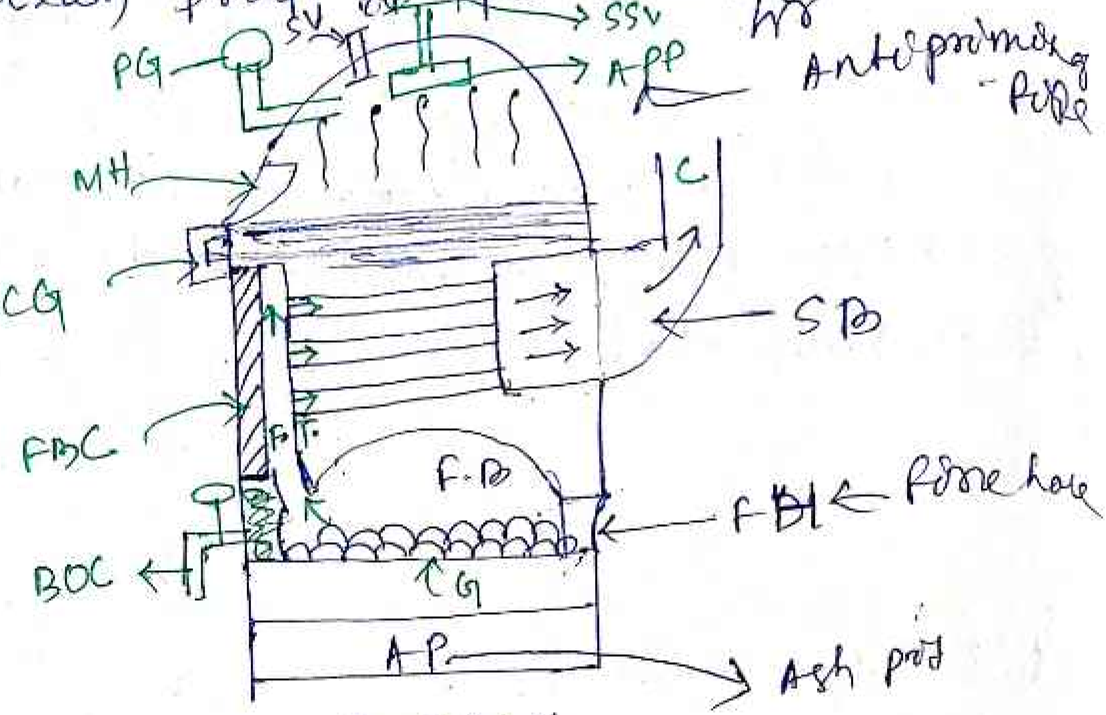
Working principle

The fuel is entered into the furnace which is placed inside the boiler through the fire dome. & it is burnt over the grate surface & the ashes are collected over the ash pit. The combustⁿ gases will pass through the fire tube (FT), which is completely surrounded by water. & for the effective heat transfer to occur cross tube are attached. After the HT betⁿ the water & the hot gases steam is burnt & collected in the upper part of the boiler. On the boiler shell the boiler mountings such as pr. gauge, safety valve, SSV, WLG. are attached for the safe operⁿ. Many no. of hand holes are designed across the cylinder shell for the cleaning of cross tubes & the manhole are constructed on the top part for the cleaning of interior part of the boiler & ext. ~~cock~~ of the combustⁿ. chamber.

11/08

Cochran Boiler

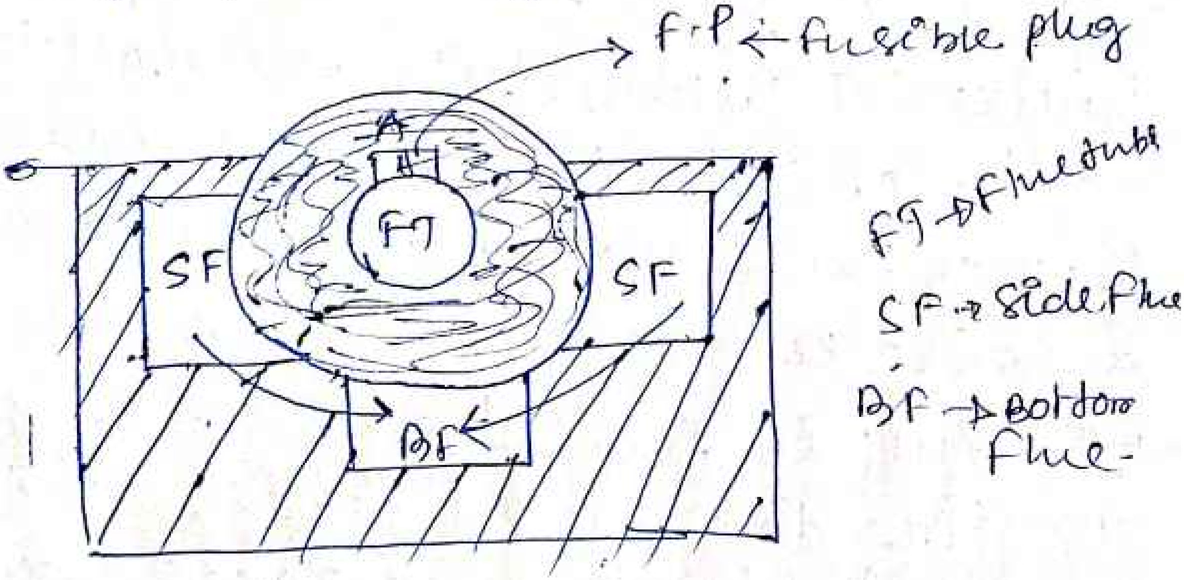
It is one of the best multi tube fire tube boiler. It consists of a cylindrical shape, which is seamless. (throughout a similar material constrⁿ.) constrⁿ. having a dome ~~set~~ shape steam space which gives maxⁿ. space requirement for the steam prodⁿ. The fuel is supplied ~~through~~ to the fire door to the (grate), where it is burnt & ~~blastes~~ the ashes get deposited over the ashpeed. The hot flue gas after the combustⁿ. in the fire blocks pass through the fire tube having its left portion lined up by fire brick lining, whose main purpose is to guide the flue gases through a number of horizontal fire tubes to the smoke box before exiting the chimney. The boiler mountings such as pr. gauge, steam stock valve, safety valve, water level gauge, globe of cock, etc. are attached for the safe operⁿ. Its working pressure is betⁿ. 6.5 to 15 bar & the rate of steam prodⁿ. is upto 3500 kg/hr.



COCHRAN

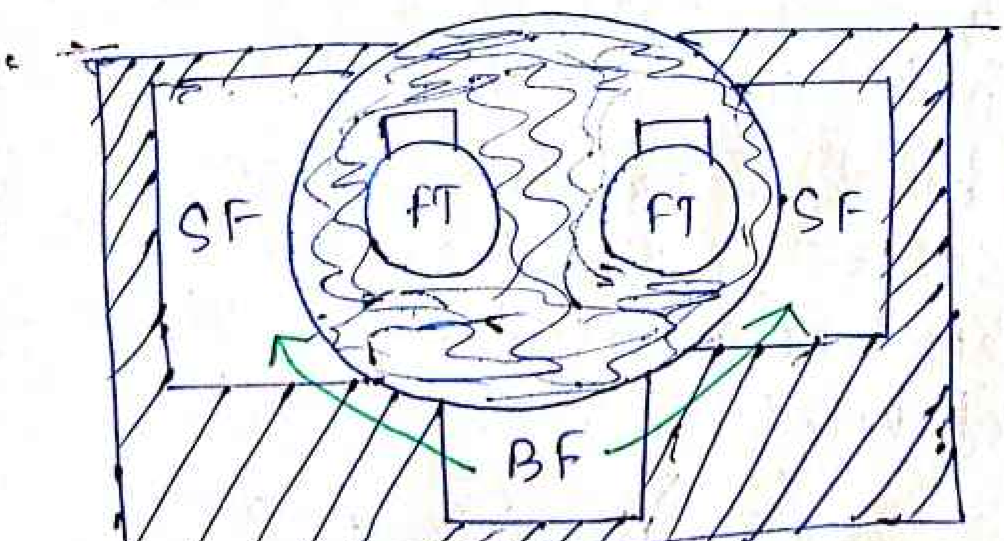
CORNISH BOILER

It consists of a cylindrical shell of two flats surfaces at the ends. which is kept horizontally over the constrⁿ. of fire bricks. The shell contains a single fire tube carrying a furnace which is at the front end of the fire tube. The fuel or the coal after combustⁿ. on the grate the hot gases so produced passes through the fire tube to the back end of the boiler, where by the modification inside the boiler hot gas travel through the two side flues upto the front end. The hot gases will pass through the bottom flue before exiting through the chimney. The boiler mountings such as high steam water level gauge, safety valve, The main adv. of this kind of boiler is the sediments which are deposited at the bottom of boiler don't come in direct contact with the hot gases carrying more heat as due to the provision of bottom flue through which the hot gases travel in the last part inside the boiler. The working pr. is about 10.5 bar & steam prodⁿ. rate 6500 kg/hr.



(CORNISH)

LANCASHIRE



(LANCASHIRE)

working pr. is @ upto 16 bar under the prod^n of steam is 9000 $\frac{kg}{hr}$.

LOCOMOTIVE BOILER

This is the best transportable boiler for high steam rate prod^n & can be operated upto a pr. of 17 bar. It consists of ~~upto~~ a horizontal cylindrical barrel, which carrying many no. of fire tubes placed horizontally. The shell or barrell has a fire box at one end & a smoke box at @ the other end.

→ The fuel is entered ~~at the grate~~ through the fire door to the grate, where the combust^n of fire takes place & the fire brick arch is used to circulate the hot gases effectively inside the fire box & dampers are provided to guide the flue gases through the fire tubes. The steam is collected in the steam dome where a regulator is there to control the steam prod^n. Also the superheaters are attached inside the larger diamt. flue tubes.

Steam prod^n rate → 3000 $\frac{kg}{hr}$.

@ working pr. → 100 bar

Scotch

This type of boiler is preferable as it is more compact & portable for a steam prodⁿ rate of 1000 kg/hr. It consists of a cylindrical shell, which is vertical & confining 3 to 4 steel made furnaces. Each furnace has its own fire box in which the combustⁿ of fuel takes place & the hot gases are allowed to pass through a bunch of horizontal fire tubes to the smoke box.

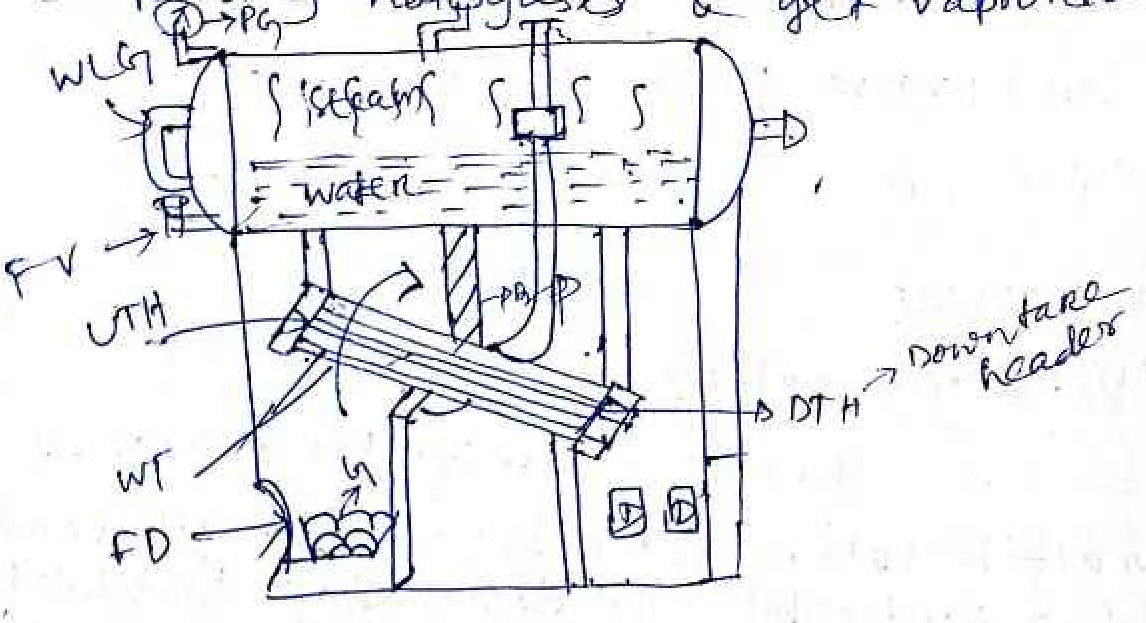
The boiler mountings are blow off valve, pr. gauge safety valve, steam ~~cock~~ stop valve, etc.

Water Tube Boilers

(1) ~~Backup~~

(1) Babcock-Wilcox

This is a horizontal inclined multitubular water tube boiler. It consists of a horizontal cylindrical shape to which uptake header & downtake header are attached with the help of the constrⁿ ~~subtached~~ suspended from the boiler shape. To these headers many no. of inclined straight water tubes are attached. Each water tube is supplied with a handhole for the cleaning & maintenance of the tools. Through the feed valve feed water pumps into the boiler to level can be measured by the help of water level gauge. The feed water passes through down take header. Then uptake header & finally enter into the shell in wet condn. The fuel is supplied through the fire door & is burnt over the plate & the hot gases after the combustn. are circulated by the special constr. of the fire brick lining. Also baffles are attached to each fire brick lining to guide the circulation of hot gases. The water coming from the shell passes through the down take header & during its circulation inside the water tube it absorbs the heat from the over passing hot gases & get vaporized



When we start moving from down take header start to move uptake header & then the steam is collected at the steam stage uptake header.

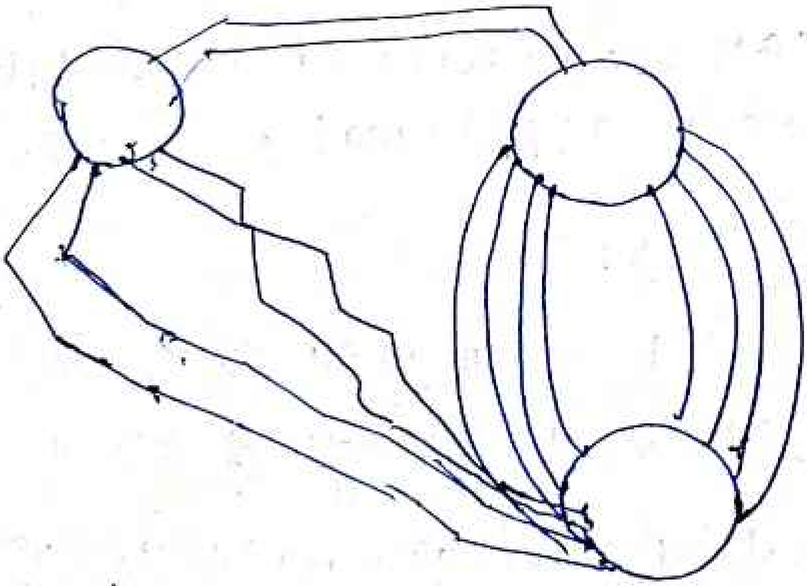
The steam from the boiler shell is ~~circulated~~ supplied in the superheater through the anti priming pipe to the super heater & then the steam is supplied to the turbine through the steam valve.

Boiler mounting

feed wall,

working pr. of this boiler is about 80 to 100 bar & the steam generⁿ rate is upto 30000 kg/hr.

STIRLING BOILER



- Bent tube
- Drums & heades are connected
- For large central power station
- Steaming capacity more than 50000 kg/h
- & pr. as high as 60 bar

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(i) The Type of Water Circulation

As in the natural circulation, convective heat transfer co-efficient is very very less, which prevents effective heat transfer to occur betn the water tube & hot gas. That's why generally forced circulation is used to design high pr. boiler. This can be achieved by supplying the feed water through a feedwater pump.

(ii) The type of Tubic

If the feed water will pass through a single tube, then the length of the tube to meet the desired quality of steam will be very large, which will cause large pr. drop across the pr. tube. To overcome this problem a high pr. boiler multi tube evaporating system is preferred.

(iii) The Type of Heating

(i) We know if water is heated at different pr., then the latent heat of vapourization decreases with increase in pr.. That's why if we can heat the water closed to or above to the critical point it will minimized the amt. of heat reqd. for the prodn of steam. The high pr. boiler uses steam heating above (super critical) or below (sub critical) critical pt. heating approach.

(II) To increase the heat transfer co-eff. of convection, it is found that the best way is to heat the feed water by using some part of the superheated steam.

(III) The heat transfer co-eff. of convection largely depends on velocity of the fluid. In designing high pr. boiler by the use of feed water pump & steam circulating pump, the vel. of the water or steam can be increase appreciatively & to increase the vel. of air or hot gases use of compressor or gas turbine arrangement is reqd.

(iv) Generally the vel. of hot gases is in the range of sonic or super-sonic.

ADV. of High pressure Boiler

(I) Due to forced circulation, effective heat transfer betn the water & the hot gases is achieved.

(II) Reduction in scale deposition & over heating due to high vel. of the water & effective tubic arrangement.

(III) No path of the tubes is subjected to thermal or differential stages due to effective distribution of heat transfer area, which prevent heating of air or water from the pipelines.

(I.V) Due to compactness the floor area occupied by this kind of boiler is very less.

(v) High steam prodn. rate (upto 50 to 60 tonnes of air) can be achieved.

(vi) Due to pressurized combustion the heat genera rate is increased which helps in high steam prodn. rate.

(vii) Operating pr. above 160 bar can also be achieved

At critical pt. of v
Pr.

221·21 bar

0·00375 $\frac{m^3}{kg}$ → critical sp. vol. of water.

14/08

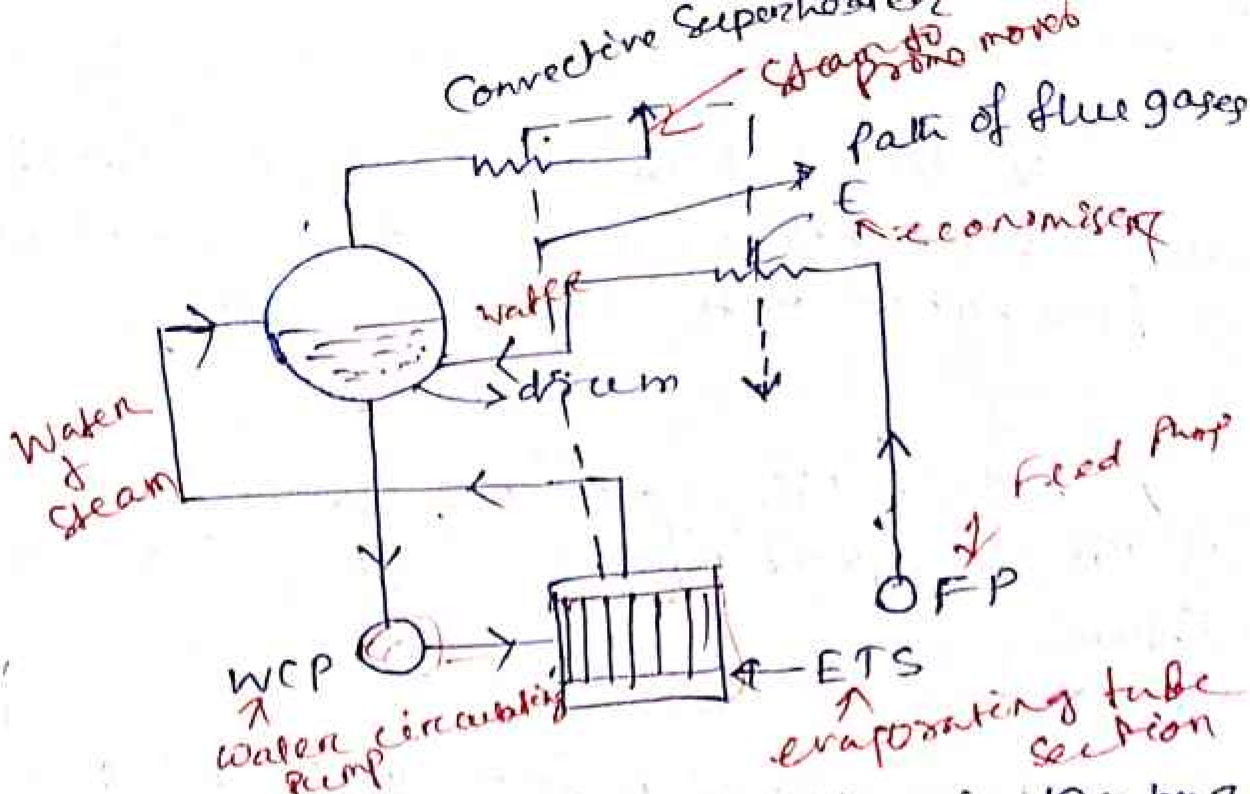
LAMONT BOILER

Principle :-

→ It increases the convective HT co-eff. by utilizing a centrifugal pump for the forced circulation of water

Operation

Feed water from the feed pump comes to the drum by passing through the economizer. The water from the drum is extracted by the water circulating pump which delivers it to the evaporating tube section, where due to the HT the water is converted to steam & again supplied to the steam space of the drum. The saturated steam from the drum passes through the convective superheater & then to the primary mover.



The working pr. of the Lamont boiler is 130 bar.

- Rate of steam prodn is 45 to 50 tonnes per hr.
- The maxm temp upto which it is superheated to 500°C

Disadv. of Lamont Boiler

- The Deposition of sediments and impurities or salt at the bottom of the drum as well as in some parts of the evaporating tube system prevents the effective HT to occur.
- As due to density difference of the liquid phase & vapour phase of the water, there is an every possibility of bubble formn in the wall of the tubes. which also prevents the ~~eff~~ effective HT.

LOEFFLER BOILER

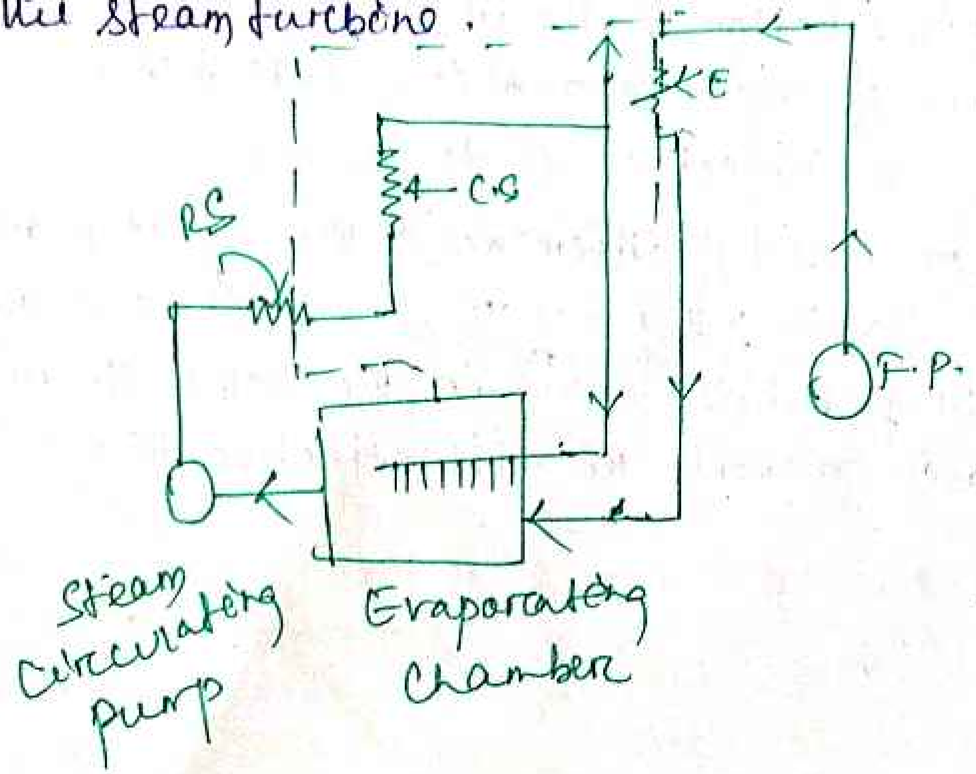
The disadv. of Lamont boiler i.e. deposition of salt & sediments is was overcome by not allowing a feed water into the boiler drum.

Principle of Heating of Steam

The noble principle is utilized in heating of the feed water to the help of superheated steam.

OPERATION.

The feed water from the feed pump & after passing through the economizer is supplied to the evaporating chamber. A steam circulating pump is supplies the feed water first to a radiant superheater & then by passing through a convective superheater 2/3rd of the superheated steam is supplied to the evaporating chamber. & only 1/3rd of the steam is utilized for driving the steam turbine.



Working pr. → Upto 150 bar
Steam prod^n. rate → Very high 120 - 170 tonnes/hr
Temp. → ~~1300 120 to 170~~ 540°C.

BENSON BOILER

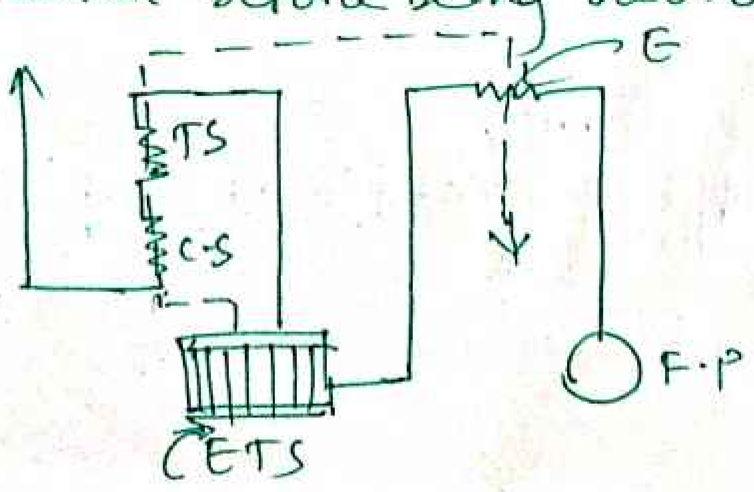
The 2nd disadv. of Lamont boiler is overcome by Benson Boiler

Principle

It utilizes the heating of water above or at critical state (221.21 bar). As no wet region is available during heating, this type of boiler doesn't reqd. any drum.

Operation

Feedwater at a very high pr. (equal to or above critical state) comes to the evaporating steam system by passing through the economizer & then for partial evaporation of the water occur in the steam section & it is again supplied to the transit section. Where all of them are converted to saturated steam & finally it will pass through the convective superheater before being delivered to the turbine.



Working pr. → 220 to 230 bar

Rate of steam prodⁿ → 150 to 160 tonnes/hr.

Maxᵐ temp. → About 600°C

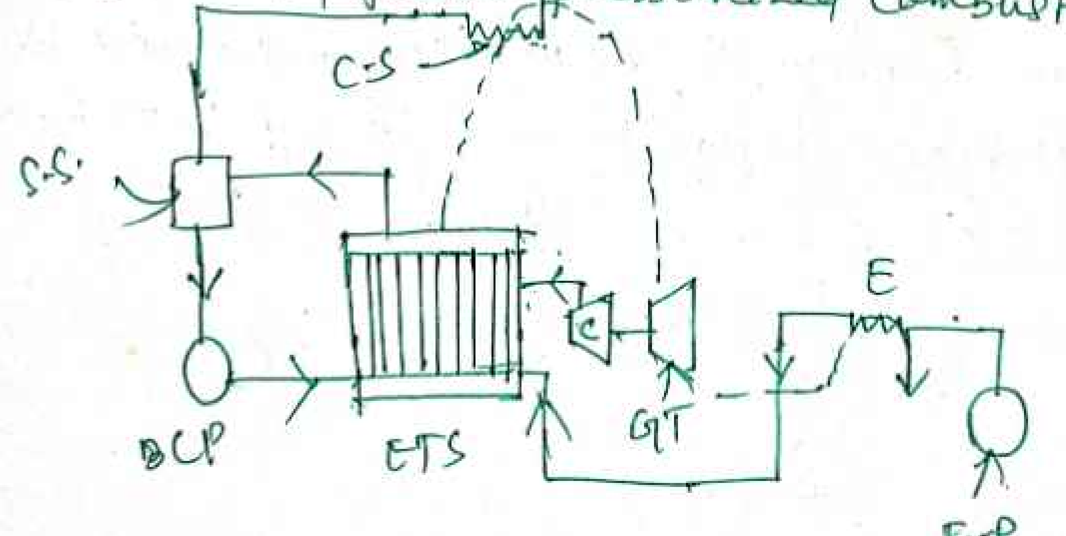
Velox Boiler

The principle of Heating or evaporating steam

It utilizes pressurized combustion of the fuel due to which the vel. of hot gases is increased above sonic velocity.

Operation

Feed water from the feed pump is supplied to the evaporating section by passing through the economizer. By the help of boiler circulating pump the ~~feed water~~ steam formed in the evaporating tube system is supplied to the steam separator & then it passes through the convective superheater before it delivered to the turbine. A gas turbine is used to drive the axial compressor which is used for the pressurized combustion.



19/08

Performance of Boiler

Evaporative capacity of Boiler

(I) kg of steam produced/hr.

(II) kg of steam produced/hr / sq. mtr of surface area.

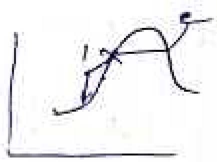
(III) kg of steam / kg of fuel fired

(IV) Equivalent evaporation.

Equivalent evaporation :- It is defined as the mass of steam that can be produced by utilizing the same heat that is actually consumed in the boiler assuming the evaporation to occur at 100°C.

$$m_a \text{ kg} = \frac{h_2 - h_1}{2257 \frac{kJ}{kg}}$$

→ At 1 atm & 100°C



Boiler efficiency

$$\eta_{boiler} = \frac{m_a \times (h_2 - h_1)}{m_f \times C}$$

$$= \frac{\text{heat actually utilised in generation of steam}}{\text{the heat supplied by the fuel}}$$

✓ Equivalent evaporation may be defined as the amount of water evaporated from water at 100°C to dry & saturated steam at 100°C.

Factors for the heat losses in a boiler

(1) Variable factors

(2) Fixed factors

Fixed factors:

→ Boiler Design → Built in losses

→ Heat recovering equipments (it includes economiser, superheaters, air preheaters & feed water heater.)

Variable factors

→ Air flow rate
→ Excess air fluctuations
→ The cond^n. of fuel
→ Humidity & temp. of the combustion air
→ Combustion
→ Incomplete combustion.

Types of heat losses for boilers

1→ Heat loss to the flue gases.
2→ Heat loss due to incomplete combustion.
3→ The heat lost due to convection & Radiation
4→ Heat loss due to unburnt fuel.

(1)

Let m_g = Mass of hot gas generated after the combust

T_g = Temp. of ~~out exit~~ the flue gas

T_a = Ambient temp.

C_{pg} = Sp. heat of the flue gases

Heat lossed $= m_g C_{pg} (T_g - T_a)$

m_2 = Mass of the moisture produced during combustⁿ.

$m_2 (h_{fg})_{Tg}$

→ (2) CO_2 → In the flue gases
⇒ complete combustⁿ.
If in the combustⁿ process, we get CO, ~~then it is~~ in the hot gases, then it is incomplete combustⁿ.

CO → 10,120 KJ/kg

CO_2 → 33800 KJ/kg.

Net amt. of heat lossed due to incomplete combustion = CO − CO_2

→ (4)

If m_f = Mass of the unburnt fuel.

Heat lossed to the unburnt fuel = $m_f \times C$.

→ (3)

$(m_f \times C) - (1+2+4)$ [Criteria] − Amt. of heat utilized for generating steam

Factor of evaporation

It is defined as the ratio of heat received by 1 kg of water under working conditions to that received by 1 kg of water evaporated from & at 100°C. It is denoted by Fe.

h_{f1} = sp. enthalpy of water at a given feed temp. & $Fe = \frac{h - h_{f1}}{2257}$

21/08/08

Q:- A boiler produces 2000 kg of dry & saturated steam per hr. at 10 bar & feed water is heated by an economizer to a temp. of 110°C. 225 kg of coal of a calorific value of 30100 $\frac{kJ}{kg}$ are fired per hour. If 10% of coal remains unburnt. Find thermal efficiency of

(i) The boiler (ii) The boiler & the grate combined

Sol^n:- Given Datas

$\dot{m}_a = 2000 kg/hr.$

Pressure = 10 bar

Quality = Dry saturated

Inlet cond^n. temp. = 110°C

Actual amt. of coal utilized per hour to generate 2000 kg of steam per hour

$= 0.9 \times 225 = 202.5 kg$

Mass of the steam generated per hour by the combustion of 1 kg of fuel

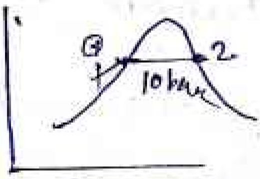
$= \frac{2000}{202.5} = 9.87 kg$

↑ Actual mass of the fuel burnt

(i) Thermal efficiency of the boiler

i.e. $\eta_{th} = \frac{\text{Act. amt. of heat used to generate 1 kg of steam}}{\text{Calorific value}}$

$$\Rightarrow \eta_{th} = \frac{(h_2 - h_1)\frac{kJ}{kg} \times 9.87\,kg}{C}$$



(ii) Mass of steam produced per kg of fuel supply to the grate

$$= \frac{2000}{225} = 8.88\ kg$$

$$\therefore \eta_{th} = \frac{\frac{2000}{225}(2776.2 - 459.8)}{30100}$$

$$h_2 = h_g|_{10\,bar} = 2776.2\ \frac{kJ}{kg}$$

$$h_1 = h_f|_{110°C} = 459.8\ \frac{kJ}{kg}$$

$$\therefore \eta_{th} = \frac{2316.4 \times 9.87}{30100}$$

26/08/08

Draught in Power Plant

Def^n :- The small pr. difference that causes the flow of a gas is called as draught.

1. Natural Draught
2. Artificial Draught
   - Mechanical
     - forced
     - balanced
     - Induced
   - Steam jet draught
     - Induced
     - Forced

~~Natural Draught~~

Requirement of Draught

→ Basically the draught have mainly two objective

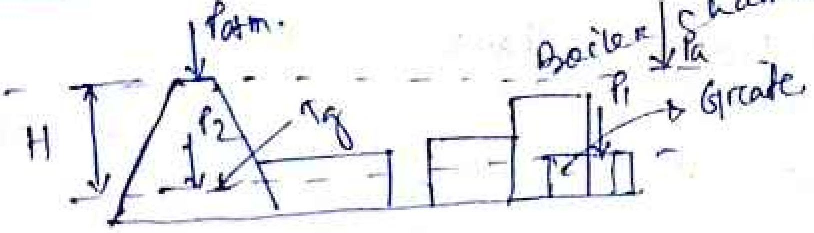
(1) To force the air so that it can pass over the grate, boiler settings or any boiler accessories.

(2) It should carry away the products of combustion before throwing them to a safe height.

Natural Draught

(1) Natural draught is produced by the help of chimney. The draught produced by the chimney is due to the density difference betn the column of hot gases inside the chimney & cold air outside.

Analysis of Natural Draught



$$P_1 = P_a + \rho_a g H$$

$$P_2 = P_a + \rho_g g H \qquad \rho_g = \text{Mass density of hot gases.}$$

$$\Rightarrow P_2 - P_1 = gH(\rho_g - \rho_a)$$

The Natural draught value of any Power Plant is upto 12 mm of water now-a-days.

Height of the Chimney & Diameter

Assumption

(1) After the combustion process the pr. & temp. of the hot gases as well as the air is same.

P, T_a $\quad m_a + 1 =$ Mass of hot gas
$m_a =$ Mass of air

$$PV = m_a R_a T_a$$

$$\Rightarrow P = \frac{m_a}{V} R_a T_a$$

$$\Rightarrow \rho_a = \frac{P}{R_a T_a} = \frac{101.325}{0.287} \times \left(\frac{1}{T_a}\right)$$

$$= \frac{K}{T_a} \quad \left(K = \frac{P}{R_a}\right)$$

$$\frac{m_a + 1}{m_a} = \frac{\text{Mass of hot gas}}{\text{Mass of air}}$$

$$\frac{\rho_g}{\rho_a} = \left(\frac{m_a + 1}{m_a}\right)$$

$$\Rightarrow \rho_g = \rho_a \left(\frac{m_a + 1}{m_a}\right) \times (\rho_a)_{T_g}$$

$$= \left(\frac{m_a + 1}{m_a}\right) \times \frac{K}{T_g}$$

The natural draught value

$$= (\rho_g - \rho_a) g H$$

$$= \left[\left(\frac{m_a + 1}{m_a} \times \frac{K}{T_g}\right) + 1 - \rho_a\right] gH$$

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$$\Delta P = \rho_g - \rho_a$$

$$\Rightarrow \cancel{K_g+1}\left[\left(\frac{1+m_a}{m_a}\right)\frac{1}{T_g} - \frac{1}{T_a}\right] = \rho_g g H_1$$

$$= \left(\frac{m_a+1}{m_a}\right) \times \frac{K}{T_g} \times g H_1$$

~~$\Rightarrow H_1 -$~~ m

$$\Delta P = \rho_a - \rho_g$$

$$K_g+1\left[-\left(\frac{1+m_a}{m_a}\right)\times\frac{1}{T_g} + \frac{1}{T_a}\right] = \rho_a g H_1$$

$$= \left(\frac{m_a+1}{m_a}\right)\frac{K}{T_g} \times g H_1$$

$$\Rightarrow H_1 = \left(\frac{m_a}{1+m_a}\right) \times T_g H\left[\left(\frac{m_a}{1+m_a}\right)\frac{T_g}{T_a} - 1\right]$$

$$\Rightarrow H_1 = H\left[\left(\frac{m_a}{1+m_a}\right)\frac{T_g}{T_a} - 1\right]$$

Chimney Draught

vel. of hot gases $= C = \sqrt{2gH_1}$

$$\dot{m}_g = \rho_g \times A \times C$$

$$A = \frac{\pi}{4}D^2$$

for $\frac{dm_g}{dT_g} = 0$

$\Rightarrow \frac{T_g}{T_a} = 2\left(\frac{m_a + 1}{m_a}\right)$

$$\boxed{(H_1)_{max} = H}$$

Draught Losses

(I) Draught loss due to friction along the flue gas path & boiler settings.

(II) Draught loss occurs due to bends provided along the flue gas path to the accessories.

(III) Due to friction loss in the headers of the boiler accessories

Artificial Draught

Mechanical Draught

Forced Draught :-

Whenever the pos^n of blower (fan or) near the boiler base the pr. over the fuel bed is reduced below that of atms.

Induced Draught :-

Pos^n. of the fan or blower is placed at the base of ~~boiler~~ the Chimney.

Balanced Draught :-

Combination of forced & induced

Factor of evaporation

It is defined as the amount of heat utilised to produce 1kg of steam to the amount of heat required ~~1kg of~~ to produce one kg. of steam.

Difference bet^n. forced & Induced draught

(i) In the forced draught to drive the fans water cooled bearings are not required.

(ii) The fan size & the power reqd. in case of forced draught is $\frac{1}{5}$th & $\frac{1}{7}$th respectively in comparision to the induced draught.

(iii) The leaking of air into the boiler is reduced or minimized by using forced draught.

(iv) While cleaning, the internal surface of the boiler chamber by opening the fire door doesn't affect, the combustion hot gas leakage.

ADv. ~~of~~ Mechanical Draught over Natural Draught

(i) The circulation of air & hot gases is effectively increase by the Mechanical draught

(ii) Boiler efficiency is increased.

(iii) Efficiency of the power plant also increases.

(iv) Evaporative capacity of the boiler increases

(v) Chimney height is reduced.

(vi) No grate fuels can be used.

(vii) Steam Jet Draught

The steam jet draught

02/11/08

Module : 2

Nozzle :- Steam nozzle may be defined as a passage of varying C/S, through which heat energy of steam is converted to K.E.

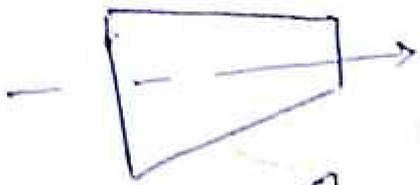
K.E. increase with the expense of pr. drop.

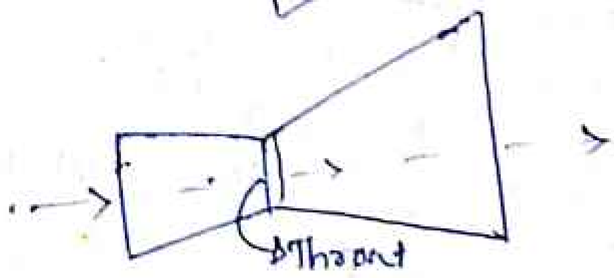
Types of Nozzle

(I) Convergent

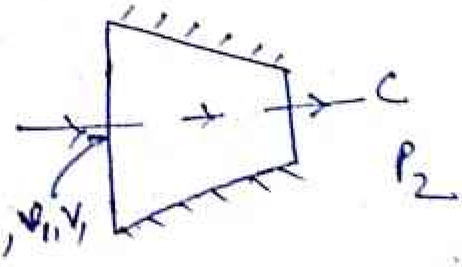
(II) Convergent & Divergent

(III)





The Exit velocity through the Nozzle



$$\frac{V_1^2}{2} \times 10^{-3} + h_1 = \frac{C^2}{2} \times 10^{-3} + h_2$$

$$\Rightarrow \frac{C^2}{2000} = h_d = h_1 - h_2$$ (Neglecting V_1 in comparison to C)

$$\Rightarrow C = \sqrt{2000 (h_d)} = 44.7 \sqrt{h_d} \text{ m/s}$$

Calculation of Discharge through the Nozzle & Cond^n for max^m discharge-

Let P_1 is the pr. at inlet to the nozzle

v_1 is the sp. vol. at inlet to the nozzle

C is the velocity at throat

P_2 is the pr. at the throat area

v_2 is the sp. vol. at the throat

$\frac{C^2}{2}$ = enthalpy drop (J/kg)

$= \Delta h$

$Tds = \delta Q = dh - vdP = 0$

$\Rightarrow dh = vdP$

$\Rightarrow \int_1^2 dh = \int_{P_1}^{P_2} v dP$

Assuming the expansion process ~~assuming~~ i.e. the flow through the nozzle the polytropic eqn. Pv^n = Const.

$n = 1.3$, if superheated

$= 1.135$, if the steam is saturated

Zenner's equation

$$n = 1.035 + x \times 0.1$$ for wet steam

$x \rightarrow$ Dryness fraction

$pv^n = \text{const.} = K$

$\Rightarrow v^n = \frac{K}{P}$

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

Enthalpy drop $= \int_{P_1}^{P_2} \left(\frac{K}{P}\right)^{1/n} dP$

$= K^{1/n} \int_{P_1}^{P_2} P^{-1/n} dP$

$= K^{1/n} \times \left[\frac{P^{\frac{n-1}{n}}}{\left(\frac{n-1}{n}\right)}\right]_{P_1}^{P_2}$

$= \frac{n}{n-1} \times K^{\frac{1}{n}} \times \left[P_2^{\frac{n-1}{n}} - P_1^{\frac{n-1}{n}}\right]$

$= \frac{n}{n-1} \times \left[(P_2 v_2^n)^{1/n} \cdot P_2^{1-1/n} - (P_1 v_1^n)^{1/n} \cdot P_1^{1-1/n}\right]$

$= \frac{n}{n-1}\left[P_2^{1/n} \cdot v_2 \cdot P_2^{1-1/n} - P_1^{1/n} \cdot v_1 \cdot P_1^{1-1/n}\right]$

Gain in K.E. = Adiabatic heat drop
= work done during Rankine cycle.

$\Rightarrow h_2 - h_1 = \frac{n}{n-1}\left[P_2 v_2 - P_1 v_1\right]$

$\Rightarrow \frac{C^2}{2} = \frac{n}{n-1}\left[P_1 v_1 - P_2 v_2\right]$

$= \frac{n}{n-1} \times P_1 v_1 \left[1 - \frac{P_2 v_2}{P_1 v_1}\right]$

We have $p_1 v_1^n = p_2 v_2^n$

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

$$\therefore \frac{c^2}{2} = \frac{n}{n-1} \times p_1 v_1 \times \left[1 - \left(\frac{p_2}{p_1}\right) \times \left(\frac{p_2}{p_1}\right)^{-1/n}\right]$$

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

$$\Rightarrow c = \sqrt{2 \times \frac{n}{n-1} \times p_1 v_1 \times \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

If m is the mass of Steam discharged on $\frac{kg}{s}$

$$m = \frac{A_2 c}{v_2} = A_2 c \rho_2$$

$$= m^2 \times \frac{m}{s} \times \frac{kg}{m^3}$$

$$= \frac{kg}{s}$$

$$\dot{m} = \rho_2 \times (A_2 \times c)$$

$$= \frac{A_2 \times c}{v_2}$$

$$= \frac{A_2}{v_1 \left(\frac{p_1}{p_2}\right)^{1/n}} \times c$$

$$= \frac{A_2}{v_1 \left(\frac{p_1}{p_2}\right)^{1/n}} \times \sqrt{2 \times \frac{n}{n-1} \times p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

~~Dividing~~ Multiplying $\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}}$ both on N & D

$$= \frac{A_2}{v_1} \times \sqrt{\frac{2 \times \frac{n}{n-1} \times p_1 v_1 \times \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}{\left(\frac{p_1}{p_2}\right)^{2/n}}}$$

$$= \frac{A_2}{v_1} \times \sqrt{2 \times \frac{n}{n-1} \times p_1 v_1 \times \left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]}$$

The discharge is max^m, when $\frac{P_2}{P_1}$ is optimum. Differentiating m w.r.t. $\frac{P_2}{P_1}$ & equating to zero.

$$\therefore \frac{dm}{d\left(\frac{P_2}{P_1}\right)} = 0$$

$$\Rightarrow \frac{2}{n} \times \left(\frac{P_2}{P_1}\right)^{\frac{2}{n}-1} - \frac{n+1}{n} \times \left(\frac{P_2}{P_1}\right)^{1/n} = 0$$

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

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

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

Taking inverse of both the sides

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

$$\Rightarrow \boxed{\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}}$$

This is the critical pressure at which the discharge of nozzle is max^m.

Maximum Discharge Through The Nozzle

Only putting the value of $P_2/P_1 = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$

$$\dot{m}_{max} = \frac{A_2}{v_1}\sqrt{2\times\frac{n}{n-1}\times P_1 v_1\times\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

$$= \frac{A_2}{v_1}\sqrt{\frac{2n}{n-1}\times P_1 v_1\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$= A_2\sqrt{\frac{2n}{n-1}\times\left(\frac{P_1}{v_1}\right)\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}-\frac{n+1}{n-1}} - 1\right]}$$

$$= A_2\sqrt{\frac{2n}{n-1}\times\left(\frac{P_1}{v_1}\right)\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{1-n}{n-1}} - 1\right]}$$

$$= A_2\sqrt{\frac{2n}{n-1}\times\left(\frac{P_1}{v_1}\right)\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{n+1}{2}\right) - 1\right]}$$

$$= A_2\sqrt{\frac{2n}{n-1}\times\left(\frac{P_1}{v_1}\right)\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\times\left(\frac{n-1}{2}\right)}$$

$$(\dot{m})_{max} = A_2\sqrt{n\times\left(\frac{P_1}{v_1}\right)\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}$$

We know,

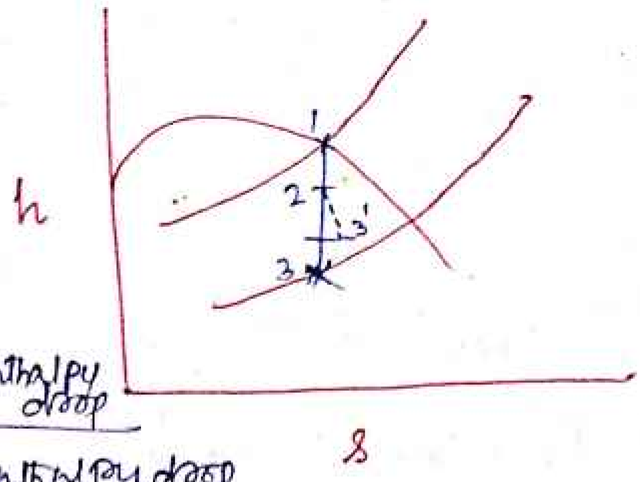
$$C = \sqrt{\frac{2n}{n-1}\times P_1 v_1\times\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

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

$$= \sqrt{\frac{2n}{n-1}\times P_1 v_1\times\frac{n-1}{n+1}}$$

$$C_{max} = \sqrt{\frac{2n}{n+1}\times P_1 v_1}$$

Nozzle Efficiency :-



$C = 44.7 \times \sqrt{h_1 - h_3}$

Again $h_1 - h_3 > h_1 - h_{3'}$

$$\eta_{NOZZLE} = \frac{\text{Actual enthalpy drop}}{\text{Ideal enthalpy drop}}$$

$$\eta_N = \frac{h_1 - h_{3'}}{h_1 - h_3}$$

$$C' = 44.7 \times \sqrt{h_1 - h_{3'}}$$

~~$\therefore \eta_N = \frac{c'^2}{c^2}$~~

$$\therefore \eta_N = \frac{c'^2}{c^2}$$

c'^2 ← Actual exit vel.

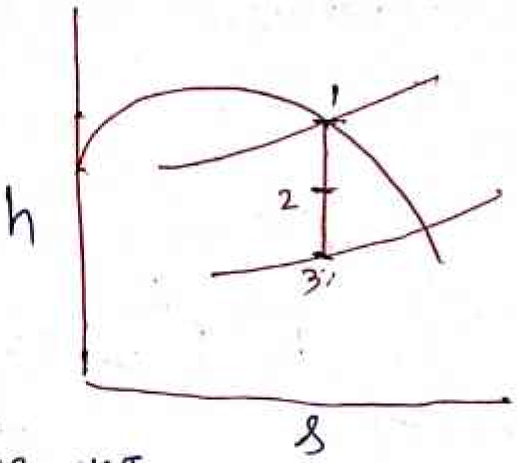
c^2 ← Ideal velocity

Super Saturated flow or, Metastable flow Through The Nozzle

Q:- Dry & Saturated Steam at a pr. of 11 bars enters a convergent & divergent nozzle & leaves at a pr. of 2 bars. If the flow is adiabatic & frictionless, Determine

(i) The exit vel. of steam

(ii) Ratio of C/S at the exit & at the throat



$h_1 = h_g|_{11\,bar} = 2780\ \frac{kJ}{kg}$.

Let P_2 = Throat pressure

$n = 1.135$ as dry saturated

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

$P_2 = 6.35$ bar

corresponding to 6.35 bar on Mollier Diagram

$x_2 = 0.96$

$h_2 = 2679\ \frac{kJ}{kg}$, $P_3 = 2$ bar

from Mollier chart

$s_2 = 6.5,\ P_3 = 2 \Rightarrow x_3 = 0.9,\ h_3 = 2480\ \frac{kJ}{kg}$

$C_3 = 44.7\sqrt{h_1 - h_3}$

$C_3 = ?$ m/s

Let Vel. at the throat $= C_2$

$\therefore C_2 = 44.7\sqrt{h_1 - h_2}$

$\dot{m}$ (at the throat) $= \rho_2 A_2 C_2$

$= \frac{1}{v_2} A_2 C_2$

At $P_2 = 6.35$ bar

$v_g = 0.297 \frac{m^3}{kg}$

$v_2 = x_2 v_g$ v_f is neglected at lower pr. in comparision to v_g

At the exit

$\dot{m} = \rho_3 A_3 C_3$

$= \frac{A_3 C_3}{v_3}$

At $P_3 = 2$ bar, $v_g = 0.885 \frac{m^3}{kg}$

$x_3 = 0.9$

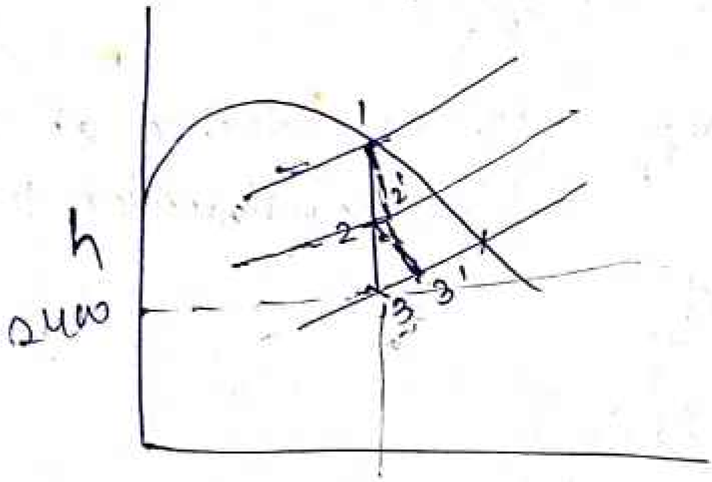
$v_3 = x_3 v_g$

$\therefore \rho_2 A_2 C_2 = \rho_3 A_3 C_3$

$\Rightarrow \frac{A_2}{A_3} = ?$

08/IX/08

Q:- The nozzle of a steam turbine is supplied with dry saturated steam at a pr. of 9 bar. The pr. at the outlet is 1 bar. The nozzle throat area is having a diamt. of 2.5 mm. Assuming nozzle efficiency as 90%. find quantity of steam used per hour. & the velocity at the throat & that at the exit.



When decimal
⇒ M.D.

$P_1 = 9$ bar , $P_3 = 1$ bar

$h_1 = h_g|_{9 bar} = 2770 \frac{kJ}{kg}$ $\quad \frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$, $n = 1.135$

From Mollier Diagram
$h_2 = 2670 \frac{kJ}{kg}$ $\quad P_2 = .$

$C_2 = 44.7\sqrt{h_1 - h_2}$

$= 44.7 \times 10$

$= 447$

$\frac{C_2'}{C_2} = \sqrt{\eta_N}$ $\quad \eta_N = \frac{h_1 - h_2'}{h_1 - h_2} = 0.9$

$\Rightarrow \frac{2770 - h_2'}{2770 - 2670} = 0.9$

$\Rightarrow h_2' = 2780 \frac{kJ}{kg}$

P & h are known

⇒ Locate 2' in pr. line.

⇒ Locate Dryness fraction

$x_2' = 0.96$

By Interpolation

$v_{g_2'} = ?$

$\dot{m} = A_2 C_2' \rho_2$

$= \dfrac{A_2 C_2'}{v_{g_2'}}$

$x_{3'} \times v_{g_{3'}}$

$h_3 = 2400 \ \frac{kJ}{kg}$

Energy supplied by the steam to the wheel/sec.

$= \dfrac{\dot{m} C_2'^2}{2} = 4$

∴ useful work = $\eta_{Turbine} \times 4$

i.e. Power $= 0.35 \times 4$.

Steam Turbine

Classification

Broadily classified in 3 types.

(I) Impulse

(II) Reaction

(III) Impulse & Reaction

(1) Depending on the no. of stage

→ Single

→ Multi

(2) Depending on the type of flow

→ Axial

→ Radial

→ ~~Tangential~~

(3) Depending on no. of cylinders

→ Single

→ Multi

- Single shaft
- Multi shaft.

(4) Depending on the type of governing
↓
To regulate the flow

(I) Throttle govern Steam turbine.
(II) Nozzle govern Steam turbine.
(III) By-Pass governing Steam turbine.

(5) Depending on the heat drop
→ condensing steam turbines with regenerator
→ Back Pressure Turbine
(6) → Topping Turbine

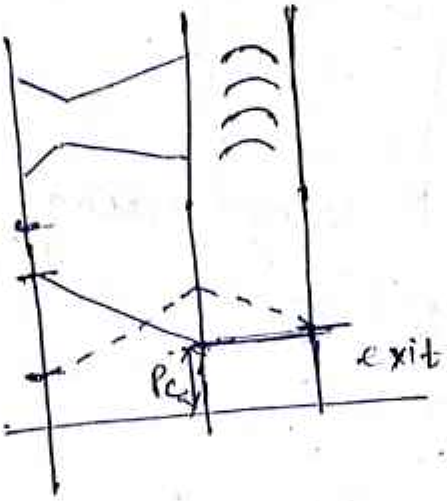
(6) Depending on the condn. of steam at turbine inlet

→ if working pr. < 2 atm. ⇒ low pr. Turbine
if working pr is betn. 2 atm & 40 atm ⇒ Medm. pr.
if working pr. is > 40 & < 170 atm
⇒ high pr.
if working pr. 170 & steam temp. > 550°C (very high) (Super critical)

Depending on the Indl. Appln.

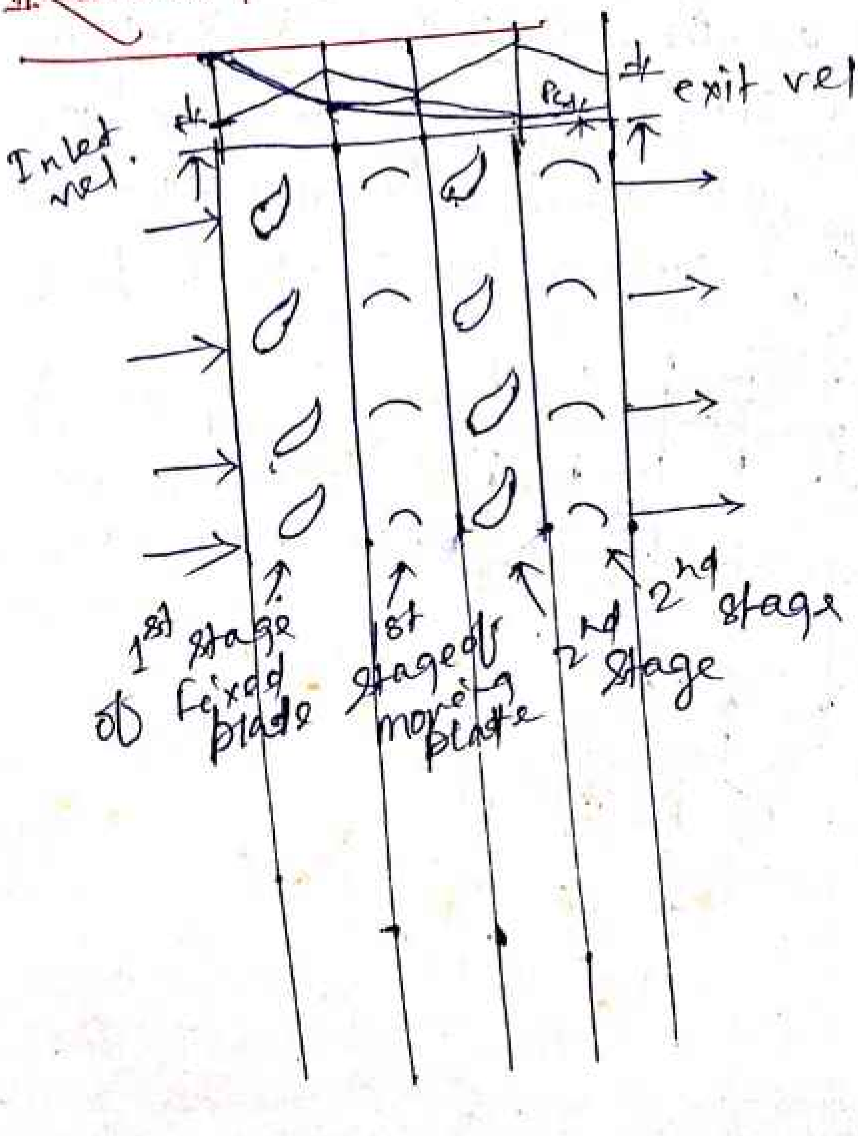
→ Stationary
→ Non-Stationary

Working Principle Pr. & vel. diagram for a simple impulse turbine.



ex:- De Laval → simple impulse turbine.

Reaction Turbine



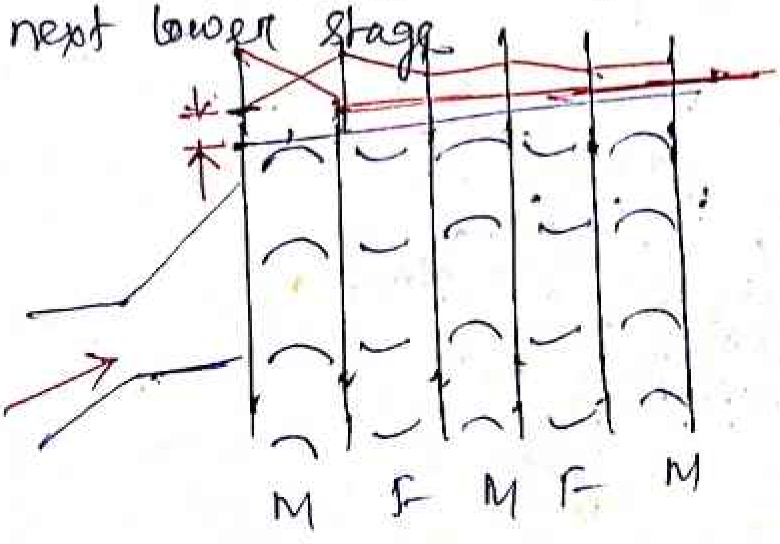
Various Ways to reduce the Velocity of the Steam for Simple Impulse Turbine.

The ways of reducing velocity

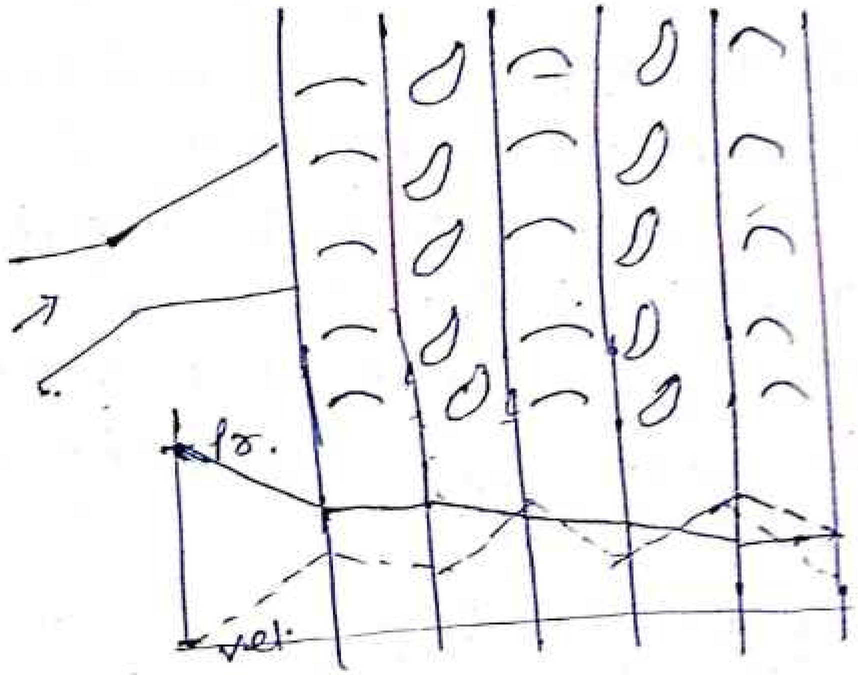
1. Velocity compounding
2. Pressure compounding
3. Comb? of Pr. & vel? compounding.
4. Reaction turbine

Velocity Compounding

In this process after the expansion in the nozzle the steam is allowed to pass over the moving blades, where a part of the K.E. is absorbed & then it is allowed to pass over the fixed ~~plates~~ blades where the pr. remains constant as well as the velocity. The fixed ~~plates~~ blades only guide the steam to the next lower stage.



Pressure Compounding



Vel. & Pr. Compounding both

In this case pr. remains const. at the begining of each state & when the steam will pass over the moving blades the vel. will decrease & when it will pass over the fixed blades its vel. ~~&~~ pr as well as pr. will same

The Work Done On The Moving Blades

Let us assume

C_1 → is the absolute vel. of the steam at which it strikes the moving blade.

C_{w_1} → The whirl vel. or the tangential component of C_1.

C_{f_1} → The flow vel. of C_1 at the inlet of the blade. This is also known as Axial vel. of C_1

C_{r_1} → Relative vel. of the steam w.r.t the blade vel. at the entrance.

C_{bl} → The blade vel. horizontal to the tangential component of C_1.

~~C_0 → At the exit o C_0~~

C_0 → Absolute velocity of the steam at the blade exit.

C_{w_0} → The whirl vel. at the exit of blade.

C_{f_0} → The flow vel. at the exit.

C_{r_0} → Relative vel. of the steam w.r.t the blade at the exit.

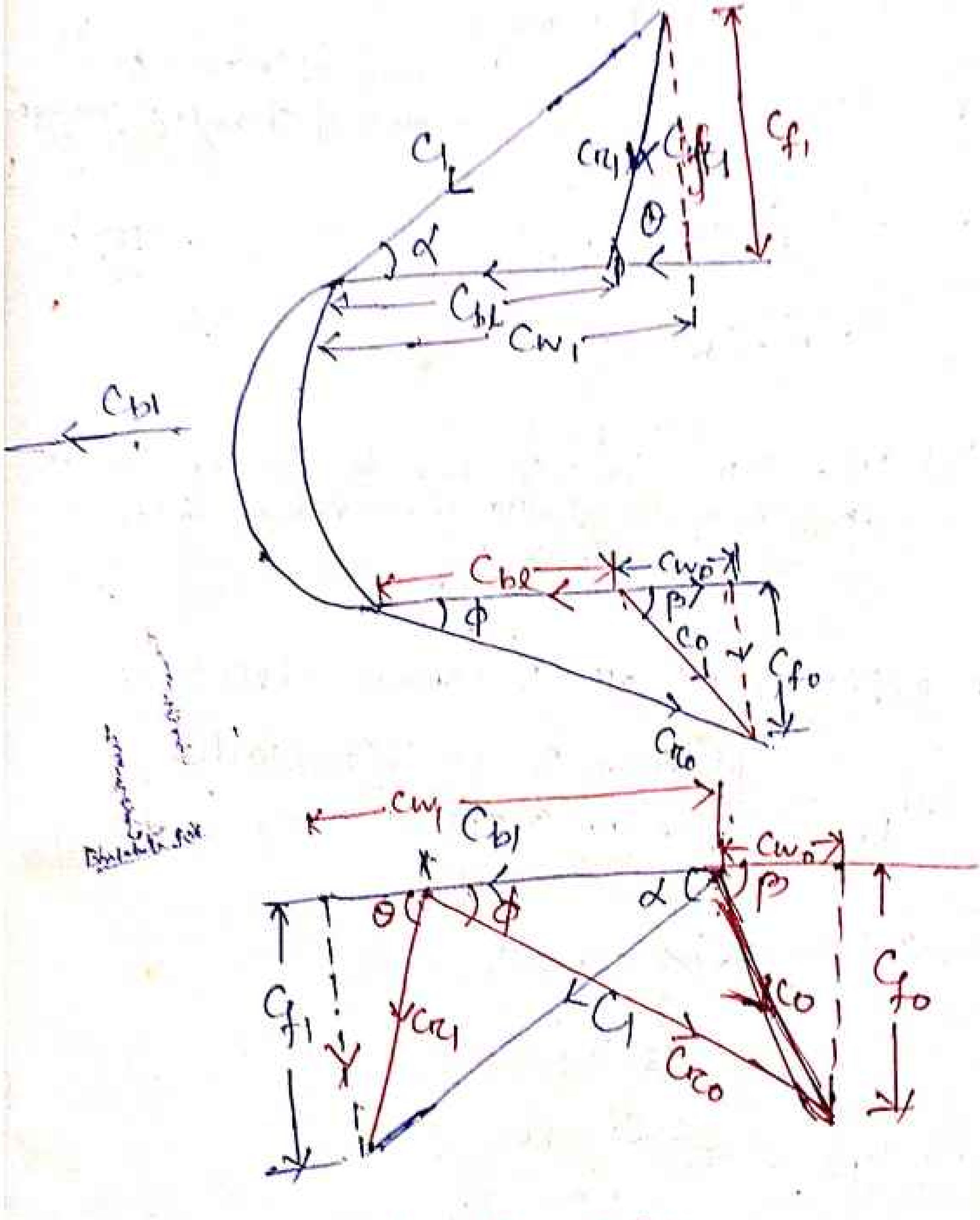
C_{bl} → The blade vel. at the exit.

- Let 'α' (nozzle angle) be the angle at which the steam strikes the blade. (w.r.t. horizontal)

θ → this is the angle of the blade at the entrance.

φ → The exit angle of the blade. (or the relative angle)

β → Vel. at which the steam exit from the blades.



$C_{w_1} + C_{w_0} = C_w$

13/12/08

Calculation of workdone in a moving blade of a single stage simple impulse turbine.

Only the tangential component of the steam velocity is responsible for doing work in the blade.

Force acting on the blade $= \dot{F}$

$F =$ Mass of steam $\times$ Acc^n
$=$ Mass of steam/sec $\times$ Change of vel

$$\therefore \dot{F} = \dot{m}_s (C_{w_1} - C_{w_0})$$

$$= \dot{m}_s (C_{w_1} + C_{w_0}) \quad (\because \text{As } C_{w_0} \text{ is opposite dir}^n)$$

$$= \dot{m}_s C_w$$

Total work done on the blade per sec is

$\dot{w} =$ force $\times$ Displace travelled / sec.

$$\dot{w} = \dot{m}_s \times C_w \times C_{bl}$$

Blade efficiency or Diagram efficiency

$$\eta_{bl} = \frac{\text{Net W.D. on the blades}}{\text{Energy supplied to the blades}}$$

$$= \frac{2 \times \dot{m}_s \times C_w \times C_{bl}}{\dot{m}_s \times C_1^2}$$

$$\Rightarrow \eta_{bl} = \frac{2 C_w C_{bl}}{C_1^2}$$

Stage Efficiency

Ratio of W.D. on the blades to the net energy supplied to the stage.

$$\eta_{stage} = \frac{\dot{m}_s \times C_w \times C_{bl}}{(h_1 - h_2)\dot{m}_s}$$

$$= \frac{C_w C_{bl}}{h_1 - h_2}$$

If h_1 & h_2 be the total heats before & after expansion through the nozzle, then $h_1 - h_2$ is the heat drop through a stage of fixed blade ring & moving blades ring.

Nozzle Efficiency

$$\eta_N = \frac{\text{Actual enthalpy drop} \leftarrow \frac{C_1^2}{2}}{\text{Isentropic enthalpy drop}}$$

$$= \frac{C_1^2}{2 \times (h_1 - h_2)}$$

Energy converted to heat by blade friction
= loss of K.E. during flow over blades
$= \dot{m}_s (C_{r1}^2 - C_{r0}^2)$

$$\therefore \eta_{stage} = \eta_{bl} \times \eta_N$$

Net Axial Thrust On the blades

Net Axial Thrust $= C_{f1} - C_{f0}$ → Differences betⁿ the vel. of flow at entrance & exit

= Mass of steam × axial accⁿ

$= \dot{m}_s (C_{f1} - C_{f0})$

Velocity Co-eff.

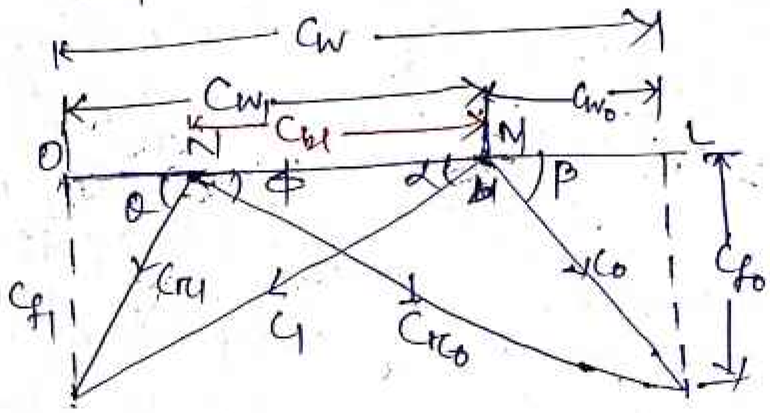
The ratio of C_{r0} to C_{r1} is called as velocity co-eff.

$$K = \frac{C_{r0}}{C_{r1}} < 1$$

Generally K lies in betⁿ 0.8 to 0.9

Derivation of the optimal value of the ratio of blade velocity to steam velocity (speed ratio) for max^m blade efficiency or max^m work done.

$\rho = \frac{C_{bl}}{C_1}$ ← speed ratio



$$C_W = OL = C_{W_1} + C_{W_0}$$

$$= ON + NL$$

$$= C_{r_1}\cos\theta + C_{r_0}\cos\phi$$

θ, φ are blade angles

$$= C_{r_1}\cos\theta \left(1 + \frac{C_{r_0}}{C_{r_1}}\frac{\cos\phi}{\cos\theta}\right) \qquad \frac{\cos\phi}{\cos\theta} = Z$$

$$= C_{r_1}\cos\theta\,(1 + KZ)$$

$$C_{r_1}\cos\theta = ON = OM - NM$$

$$= C_1\cos\alpha - C_{bl}$$

$$\therefore C_W = (C_1\cos\alpha - C_{bl})(1 + KZ)$$

$$\therefore \eta_{bl} = \frac{2 \times C_{bl} \times (C_1\cos\alpha - C_{bl})(1 + KZ)}{C_1^2}$$

$$= \left[\frac{2C_{bl}C_1\cos\alpha}{C_1^2} - \frac{2C_{bl}^2}{C_1^2}\right](1 + KZ)$$

$\Rightarrow \eta_{bl} = 2(\rho\cos\alpha - \rho^2)(1+KZ)$

Differentiating w.r.t. ρ and equating to zero.

$$\frac{d\eta_{bl}}{d\rho} = 0$$

$\Rightarrow \cos\alpha - 2\rho = 0$

$\Rightarrow 2\rho = \cos\alpha$

$$\Rightarrow \boxed{\rho = \frac{\cos\alpha}{2}}$$

$$(\eta_{bl})_{max} = 2\times\left(\frac{\cos^2\alpha}{2} - \frac{\cos^2\alpha}{4}\right)(1+KZ)$$

$$= 2\times\frac{\cos^2\alpha}{4}\times(1+KZ)$$

$$= \frac{\cos^2\alpha}{2}\times(1+KZ)$$

Assuming no friction & equiangle blades i.e. $\theta = \phi$, $K=1$, $Z=1$

$$\therefore \boxed{(\eta_{bl})_{max} = \cos^2\alpha}$$

Max^m Work Done in the blade per kg

$$W/kg = C_{bl}\times C_w$$

$$= (C_1\cos\alpha - C_{bl})(1+KZ)\times C_{bl}$$

for $K=1$, $Z=1$

$$\Rightarrow W/kg = 2(C_1\cos\alpha - C_{bl})\times C_{bl}$$

$\Rightarrow W/kg = 2\left(\frac{2C_{bl}}{\cos\alpha} \times \cos\alpha - C_{bl}\right) \times C_{bl}$

$[\because \rho = \frac{C_{bl}}{C_1} \Rightarrow C_1 = \frac{C_{bl}}{\rho}]$

$= 2C_{bl}^2$

15/IX/08

Q:- Steam with absolute vel. of 300 m/s is supplied through a nozzle to a single stage impulse turbine. The nozzle angle is 35°. The mean diamt. of the blade rotor is 1m & it has speed of 1000 rpm for zero axial thrust find suitable blade angle. Take the vel. co-efficient as 0.9. The mass flow rate to be 10 kg/s. What is the power developed.

Soln:- Given Datas:

Abs. vel. = 300 m/s = C_1

Nozzle angle = 35°C = α

Mean Diamt. of blade rotor = 1m = D

speed = 1000 rpm = N

Zero axial thrust $\Rightarrow C_{f_1} - C_{f_0} = 0 \Rightarrow C_{f_1} = C_{f_0}$

vel. co-eff. = 0.9 = K = $\frac{C_{r_0}}{C_{r_1}} \Rightarrow C_{r_0} = 0.9 C_{r_1}$

Mass flowrate = 10 kg/s = $\dot{m}_s$.

$C_{bl} = \frac{\pi D N}{60}$ m/s

$= 52.35$ m/s

$\simeq 50$

$\simeq 5$ cm

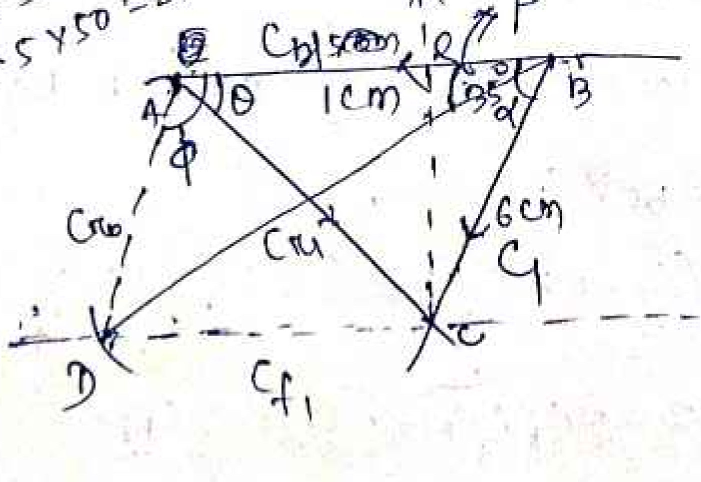
1 cm = 50 m/s

$C_1 = 300$ m/s

$\frac{300}{50} = 6$ cm

$\theta = 40°$

$C_{r_1} = AC = 5.5$ cm $= 5.5 \times 50 = 27.5$ m/s



$C_{r_0} = C_{r_1} \times 0.9$

$= 27.5 \times 0.9$

$= \frac{27.5 \times 0.9}{50}$

$= 4.9$ cm.

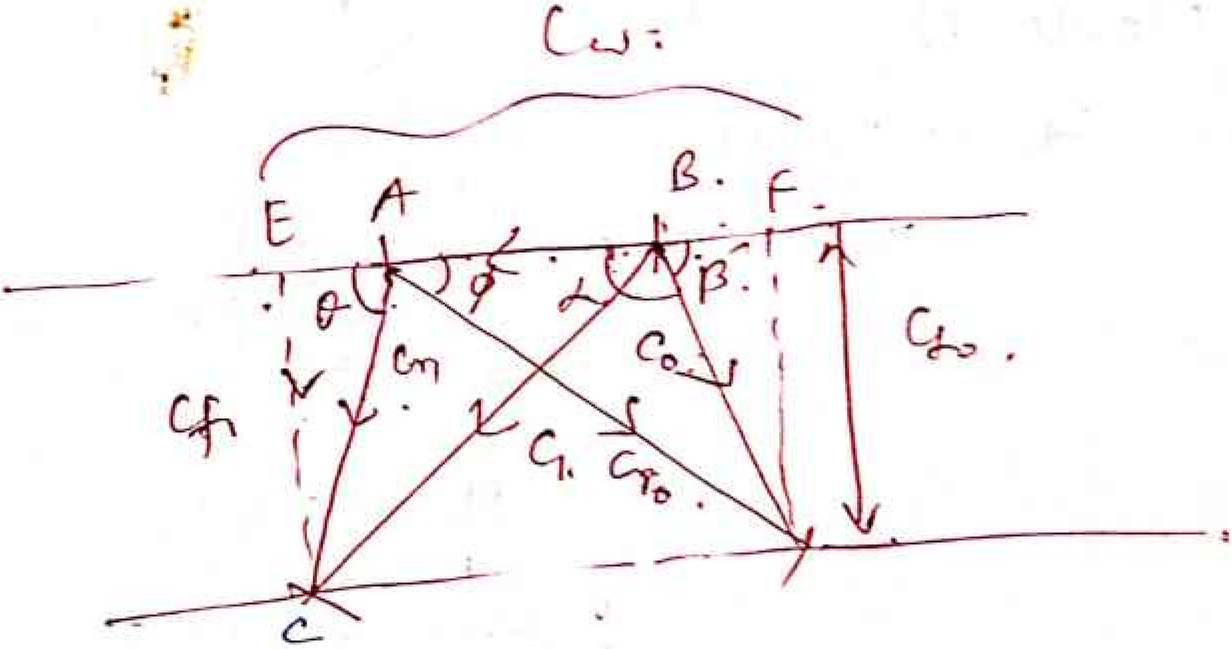
Blade angle = θ, ϕ

$\theta = 40^\circ$, $\phi = 49^\circ$

Power Developed = ($\dot{W}$)

$\dot{W} = \dot{m}_s \times C_w \times C_{bl} = \frac{kg}{s} \times \frac{m}{s} \times \frac{m}{s}$

$ef = C_w = \_$



$AE = C_{r_1}$

Q:- For an impulse turbine single stage the mean diamt. of blade is 1 m & the speed is 3000 rpm. The nozzle angle is 18°. The ratio of blade speed to steam speed is 0.42 & the ratio of the relative vel. at outlet from the blade to that at inlet is 0.85. The difference betn. the outlet & inlet blade angle is 5°. The difference in steam flow is 10 kg/sec. Dercive (i) tangential thrust in the blade.

(ii) Axial thrust

(iii) Power developed

(iv) Blade eff.

Soln.:- Given Datas;

$$\text{Mean Diamt} = 1\,m = D$$

$$N = 3000\,rpm.$$

$$\alpha = 18°$$

$$\frac{C_{bl}}{C_1} = 0.42 = f$$

$$K = \frac{C_{r0}}{C_{ri}} = 0.85$$

$$\phi - \theta = 5°$$

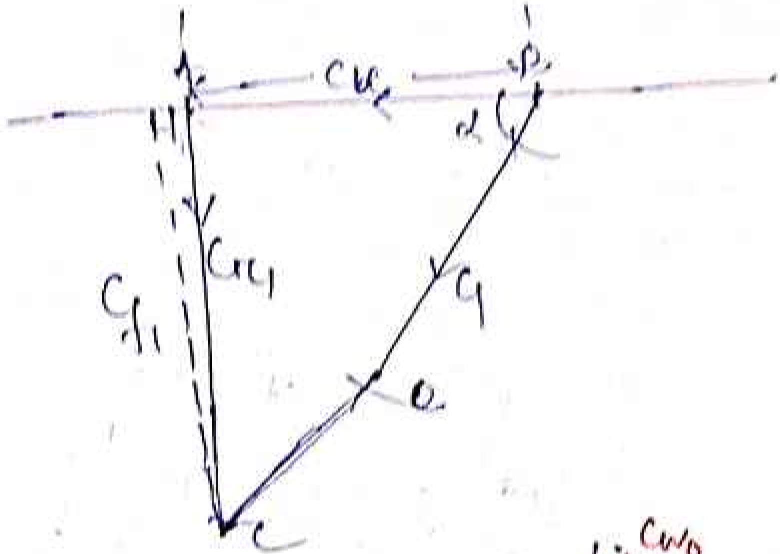
$$\dot{m} = 10\,\frac{kg}{sec.}$$

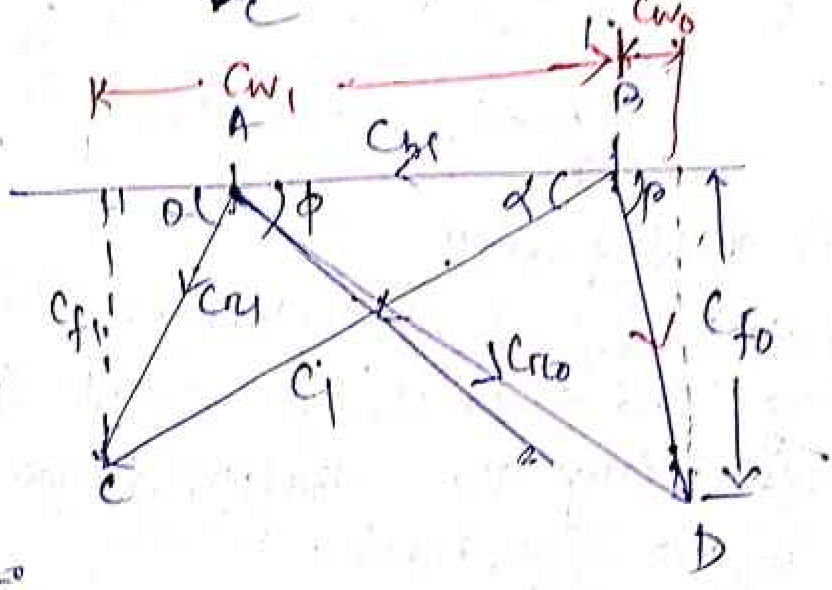
$$C_{bl} = \frac{\pi D N}{60} = \frac{3.14 \times 1 \times 3000}{60}$$

$$\Rightarrow C_{bl} = 157\ m/s$$

$$\frac{C_{bl}}{C_1} = 0.42$$

$$\Rightarrow C_1 = \frac{157}{0.42} = 373.8 \text{ m/s}$$





$\phi - \theta = 5°$

$$\frac{C_{r0}}{C_{r1}} = 0.85$$

(I) Tangential thrust $= \dot{m}_s (C_{w_1} + C_{w_0})$

(II) Axial thrust $= \dot{m}_s (C_{f_1} - C_{f_0})$

(III) Power Developed $= \dot{m}_s (C_{w_1} + C_{w_0}) C_{bl}$

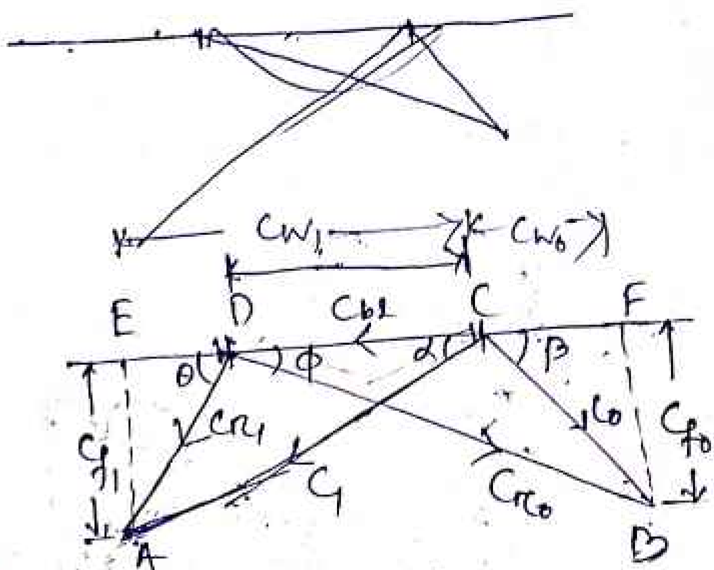
(IV) Blade eff. $= \dfrac{2 C_w C_{bl}}{C_1^2}$

MOD: III

Reaction Turbine

20/IX/08

The velocity Diagram of the moving blades in the reaction turbine



$C_{r_0} > C_{r_1}$

Degree of Reaction

The ratio of enthalpy drop in the moving blades to the total enthalpy drop (considering the enthalpy drops in both in fixed & moving blades)

i.e. $R_d = \dfrac{\Delta h_m}{(\Delta h)_f + (\Delta h)_m}$

The heat drop in moving blades is equal to increase in relative vel. of steam passing through the blad

$(\Delta h)_m = \dfrac{C_{r_0}^2 - C_{r_1}^2}{2}$

$C_{bl}(C_{w_1} + C_{w_0}) = (\Delta h)_T \longrightarrow (\Delta h)_f + (\Delta h)_m$

$C_{r_0} = C_{f_0} \operatorname{cosec} \phi$

$C_{r_1} = C_{f_1} \operatorname{cosec} \theta$

$C_{w_1} + C_{w_0} = C_{f_1} \cot\theta + C_{f_0} \cot\phi$

Assumption

Flow velocity remains same

i.e. $C_{f_0} = C_{f_1} = C_f$

$$\therefore R_d = \frac{\frac{C_{r_0}^2 - C_{r_1}^2}{2}}{(\Delta h)_T}$$

$$= \frac{C_f^2 (\operatorname{cosec}^2\phi - \operatorname{cosec}^2\theta)}{2 \times C_{bl} \times C_f (\cot\theta + \cot\phi)}$$

$$= \frac{C_f [(1 + \cot^2\phi) - (1 + \cot^2\theta)]}{2 \times C_{bl} \times (\cot\theta + \cot\phi)}$$

$$= \frac{C_f (\cot^2\phi - \cot^2\theta)}{2 \times C_{bl} \times (\cot\theta + \cot\phi)}$$

$$= \frac{C_f (\cot\phi + \cot\theta)(\cot\phi - \cot\theta)}{2 \times C_{bl} (\cot\phi + \cot\theta)}$$

$$\Rightarrow R_d = \frac{C_f}{2 C_{bl}} (\cot\phi - \cot\theta)$$

If $R_d = \frac{1}{2}$

$$\therefore \frac{1}{2} = \frac{C_f}{2 C_{bl}} (\cot\phi - \cot\theta)$$

$$\Rightarrow C_{bl} = C_f (\cot\phi - \cot\theta) \longrightarrow (1)$$

$C_{bl} = CE - DE$

$= C_{f_1} \cot\alpha - C_{f_1} \cot\theta$

$\Rightarrow C_{bl} = C_f (\cot\alpha - \cot\theta)$ $(\because C_{f_1} = C_{f_0} = C_f)$ → (2)

<u>Considering C_{f_0}</u>

$C_{bl} = DF - CF$

$= C_{f_0} \cot\phi - C_{f_0} \cot\beta$

$\Rightarrow C_{bl} = C_f (\cot\phi - \cot\beta)$ $(\because C_{f_1} = C_{f_0} = C_f)$ → (3)

Equating (1) & (2), $\phi = \alpha$

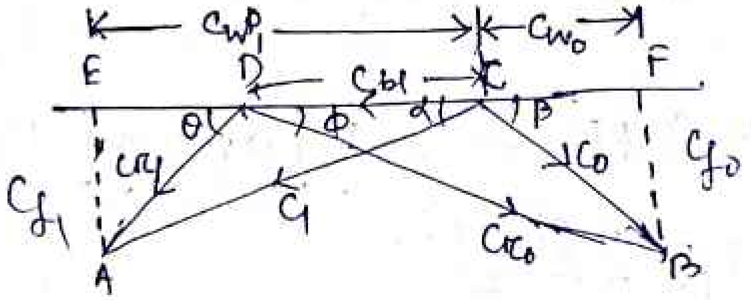
from (1) & (3), $\theta = \beta$

Any rea^n turbine having flow vel. same, & degree of rea^n ½ & $\phi = \alpha$, $\theta = \beta$. Then this is known as <u>parson's turbine.</u>

ΔAEC & ΔBDF are similar

$\therefore \boxed{C_{r_0} = C_1}$

Condition for Max^m blade efficiency



Assumptions

(i) The degree of reaction $= R_d = 0.5$

(ii) $\phi = \alpha \rightarrow C_1 = C_{ro}$

(iii) Assuming symmetrical blades (fixed & or moving)

$$W = C_{bl} \times (C_{w1} + C_{wo})$$

$$EF = C_{w1} + C_{wo}$$
$$= EC + CF$$
$$= C_1 \cos\alpha + (DF - DC)$$
$$= C_1 \cos\alpha + (C_{ro}\cos\phi - C_{bl})$$
$$= 2C_1 \cos\alpha - C_{bl} \qquad (\because C_1 = C_{ro})$$

$$W = C_{bl}(2C_1 \cos\alpha - C_{bl})$$
$$= 2C_{bl} C_1 \cos\alpha - C_{bl}^2$$
$$= C_1^2 \left(2\frac{C_{bl}}{C_1}\cos\alpha - \frac{C_{bl}^2}{C_1^2}\right)$$
$$= C_1^2 (2\rho\cos\alpha - \rho^2) \qquad \left(\because \frac{C_{bl}}{C_1} = \rho\right)$$

Energy supplied to the fixed blade is

$$K.E. = \frac{C_1^2}{2}$$

Energy supplied to the moving blades

$$K.E = \frac{C_{r0}^2 - C_{r1}^2}{2}$$

∴ Total energy supplied to the blades

$$(E)_T = \frac{C_1^2}{2} + \frac{C_{r0}^2 - C_{r1}^2}{2}$$

$$= \frac{C_1^2}{2} + \frac{C_1^2}{2} - \frac{C_{r1}^2}{2} \quad (\because C_{r0} = C_1)$$

$$= \frac{2C_1^2}{2} - \frac{C_{r1}^2}{2}$$

$$= C_1^2 - \frac{C_{r1}^2}{2}$$

$$C_{r1}^2 = C_1^2 + C_{b1}^2 - 2C_1 C_{b1} \cos\alpha$$

$$\therefore (E)_T = C_1^2 - \frac{C_1^2 + C_{b1}^2 - 2C_1 C_{b1} \cos\alpha}{2}$$

$$= \frac{C_1^2}{2} - \frac{C_{b1}^2}{2} + C_1 C_b \cos\alpha$$

$$= \frac{C_1^2}{2}\left[1 - \rho^2 + 2\rho\cos\alpha\right]$$

Blade efficiency (η_B) $= \dfrac{\text{Work Done}}{E_T}$

$$= \frac{C_1^2(2\rho\cos\alpha - \rho^2)}{\frac{C_1^2}{2}(1 - \rho^2 + 2\rho\cos\alpha)}$$

$$= \frac{2(2\rho\cos\alpha - \rho^2)}{1-\rho^2+2\rho\cos\alpha}$$

$$= \frac{2(2\rho\cos\alpha - \rho^2)}{1+2\rho\cos\alpha - \rho^2}$$

$$= \frac{2(1+2\rho\cos\alpha - \rho^2 - 1)}{1+2\rho\cos\alpha-\rho^2}$$

$$= \frac{2(1+2\rho\cos\alpha-\rho^2) - 2}{1+2\rho\cos\alpha-\rho^2}$$

$$= 2 - \frac{2}{1+2\rho\cos\alpha-\rho^2}$$

$$= 2 - \frac{2}{T} \quad (\text{where } T = 1+2\rho\cos\alpha-\rho^2)$$

$\eta_B = \text{Max}^m$

$\Leftrightarrow T = \text{Max}^m$

$\Leftrightarrow \frac{dT}{d\rho} = 0$

$\Rightarrow \frac{d}{d\rho}(1+2\rho\cos\alpha-\rho^2) = 0$

$\Rightarrow 2\cos\alpha - 2\rho = 0$

$\Rightarrow 2\rho = 2\cos\alpha$

$\Rightarrow \boxed{\rho = \cos\alpha}$

$$\therefore \eta_B = 2 - \frac{2}{T}$$

$$= 2 - \frac{2}{1 + 2\rho\cos\alpha - \rho^2}$$

$$= 2 - \frac{2}{1 + 2\times\cos^2\alpha - \cos^2\alpha}$$

$$= 2 - \frac{2}{1 + \cos^2\alpha}$$

$$= \frac{2 + 2\cos^2\alpha - 2}{1 + \cos^2\alpha}$$

$$\boxed{\eta_B = \frac{2\cos^2\alpha}{1 + \cos^2\alpha}}$$

23/11/03

Q:- In a stage of impulse reaction turbine operating with 50% degree of reaⁿ. The blades are identical the outlet angle of the moving blade is 17° & the absolute discharge velocity of the steam is 50 m/s in the dirⁿ at 110° to the motion of the blade. Draw the vel. diagram & calculate the work done per kg of mass flow per sec.

Solⁿ:- Given Datas;

$$R_d = 0.5$$

$$\phi = 17°$$

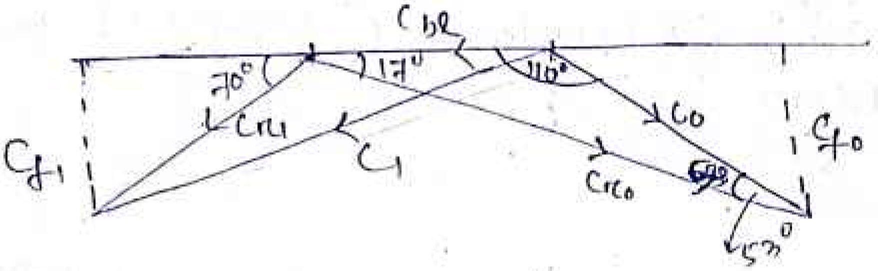
$$\Rightarrow \alpha = 17° \quad (\because \phi = \alpha)$$

$C_o = 50\ m/s = C_{r1}$

$\beta = 180° - 110° = 70°$.

$\Rightarrow \theta = 70°$

Cru ~ cy



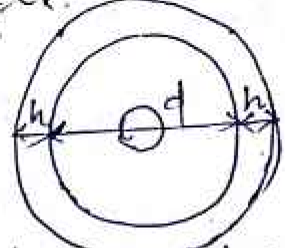
(1) Work done / kg = $C_w \times C_{bl}$

Calculation of blade height of a Reaⁿ. turbine

$D_m = (d+h)$, Area flow $= \pi D_m h$

$\dot{m}_s = \dfrac{\pi (d+h) \times h \times V_{f1}}{v_s}$

$= \dfrac{\text{Area of flow} \times \text{Vel. of flow}}{\text{sp. vol. of steam}}$



Q:- In a reaⁿ. turbine the blade tips are inclined at 35° & 20° in dirⁿ. of motion, the drum diamt. is 1m & blades are 100mm high. At this stage the steam has a pr. of 1.7 bar & dryness 0.935. If the speed of the turbine is 250rpm. find the mass of the steam flow & power developed.

$\theta = 35°$, $\phi = 20°$, $C_{bl} = \dfrac{\pi D N}{60}$

Q:- At a stage of Reaⁿ turbine the rotor diamt. is 1.4m & speed ratio 0.7. If the blade outlet angle is 20° (ϕ) & rotor speed 3000 rpm. Find the blade inlet angle & diagram efficiency. Also find the %age increase in diagram efficiency & rotor speed of the turbine is designed to run at the best theoretical speed.

max^m blade eff.

24/IX/08

Solⁿ:- Given Datas:

rotor diamt. = D = 1.4m.

Speed Ratio = $\rho = \frac{C_{bl}}{C_1} = 0.7$

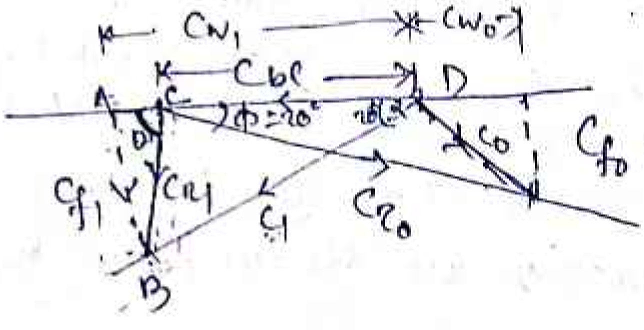
Blade outlet angle = $\phi = 20°$.

rotor speed = N = 3000 rpm.

$C_{bl} = \frac{\pi D N}{60} = \frac{3.14 \times 1.4 \times 3000}{60} = 219.91\ m/s = 220\ m/s$

$\therefore C_1 = \frac{C_{bl}}{0.7} = 314\ m/s$

Assuming $\phi = \alpha$ & $\theta = \beta$



$\Rightarrow \theta = ?$

Diagram efficiency = $\eta_{bl} = 2 - \frac{2}{T}$

$= 2 - \frac{2}{1 + 2\rho\cos\alpha - \rho^2}$

$\Rightarrow \eta_{bl} = 2 - \frac{2}{1 + 0.760 \times 2 \times \cos 20^\circ - (0.7)^2}$

$= 0.90$

Now Max^m blade efficiency is

$\eta_B = \frac{2\cos^2\alpha}{1 + \cos^2\alpha}$

$= \frac{2\cos^2 20^\circ}{1 + \cos^2 20^\circ}$

$= 0.94$

$\therefore$ %age increase in diagram efficiency is

$= \frac{0.94 - 0.90}{0.90} \times 100$

$= 4.425$

$C_{w_1} = C_1 \cos\alpha$

$= 314 \cos 20 = 295$ m/s

$AC = C_{w_1} - C_{bl} = 295 - 220$

$= 75$ m/s

In ΔABD,

~~tan~~ $\tan\alpha = \frac{AB}{AD} = \frac{C_{w}}{C_{f_1}}$ ~~=~~ $\frac{AB}{AD} = \frac{C_f}{C_{w_1}}$

$\Rightarrow$ ~~C_{f_1}~~ $\Rightarrow C_{f_1} = 107.37$

$\tan\theta = \frac{75}{107.37} = 0.69$

$\Rightarrow$

$$\tan \theta = \frac{C_{f1}}{C_{w1} - C_{bl}} = \frac{107.37}{75} = 1.43$$

$\Rightarrow \theta = 55^\circ$

Q:- The following data relates to a stage of reaction turbine.

Mean diameter of rotor = 1.5m.

Speed ratio = 0.72

Blade outlet angle = 20°

rotor speed = 3000 rpm.

· Determine (i) blade efficiency

(ii) %age increase in diagram efficiency & rotor speed of the rotor is designed to run at the best theoretical speed.

Sol^n:- Given Data:

$D = 1.5\,m.$, $\phi = 20^\circ$

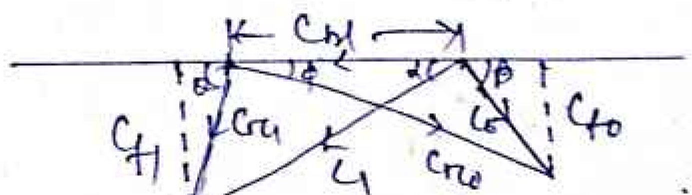
$\rho = 0.7 \Rightarrow \frac{C_{bl}}{C_1} = 0.72 \Rightarrow C_1 = \frac{236}{0.72} = 327.7\,m/s$

$C_{bl} = \frac{\pi D N}{60} = \frac{3.14 \times 1.5 \times 3000}{60}$

$C_{r0} = C_1$

$= 236\ m/s$

$\phi = \alpha$
$\theta = \beta$
$C_{r0} = C_1$



Blade eff. = $\eta_{bl} = 2 - \frac{2}{?}$

$= 2 - \frac{2}{1+2f\cos\alpha \cdot f^2}$

$= 2 - \frac{2}{1+2\times 0.92\times\cos 20^\circ}$

$= 0.909$

≈ 0.91

Now Max^m blade eff.

$(\eta_{bl})_{max} = \frac{2\cos^2\alpha}{1+\cos^2\alpha}$

$= 0.94$

$\therefore$ %age increase in blade eff. is

$= \frac{0.94 - 0.91}{0.91} \times 100$

$= 3.19$

$f = \cos\alpha$

$\Rightarrow \frac{C_{bl}}{C_1} = \cos\alpha$

$\Rightarrow C_{bl} = C_1\cos\alpha = 327.72\times\cos 20^\circ$

$= 307.9$

$\Rightarrow \frac{\pi D N}{60} = 307.9$

$\Rightarrow N_2 = ?$

% increase in speed $= \frac{N_2 - N_1}{N_1} \times 100$

Various Losses In Turbine

Internal Losses in Turbine

(i) Nozzle Loss :- When the steam travels through the nozzle (in case of a simple impulse turbine) due to the friction, the enthalpy drop decreases & due to the drop the vel. of the steam at the nozzle exit is reduced

(ii) Blade Loss :- When the steam guides over the blades (fixed or moving) due to friction bet^n the blade surface & the relative vel. decreases.

Reheat factor.

The factor which decreases the enthalpy drop during the motion of steams over the blades is called as the Reheat factor. Due to this factor the temp. when the steam will exit from the blade is increase. This is due to blade friction.

(iii) Wheel Loss :- When the steam flows over a particular stage or row of blade mounted over a rotor disc. There is a friction bet^n disc surface & steam. Due to this friction the work o/p of the turbine reduced.

(iv) Mechanical Loss (or, Rotor Loss)

As the rotor mounted over the shaft by bearing arrangement, there will be some friction loss due to which the useful work value will decrease.

(v) Steam Loss :- Due to the leakage of steam through the turbine casing some energy is lost with the steam which reduces available energy for the conversion to work.

(vi) Radiation Loss :- If a large temp. difference exists bet? the turbine casing & atmosphere. Some heat is lost ~~with~~ to the surrounding.

(vii) Velocity Loss :- If the steam will exit from the turbine (if it ~~a~~ is a single-stage turbine) then after the expansion still the vel. of the steam is very high. Some energy is lost to the high velocity at the exit. To minimize this loss various compounding arrangement is attached or multistage expansion is preferred

Governing Of Turbines

01/11/0.98

~~Nozzle~~

(I) Nozzle governing

(II) Throttle governing

(iii) Bypass governing

Nozzle governing

The steam flow rate is divided into a summation of mass flow rate by allowing the steam to pass through a set of nozzle & each nozzle is attached with a separate valve. Depending on the load on the turbine the mass flow rates through the nozzles are varied. If the turbine has to operate at lower loads, then without the governing system the turbine speed will be higher than the normal operating speed. By regulating the valves attached to the nozzles the mass flow rate of steam is decreased. Which will decrease the speed of the turbine with the normal speed.

→ If the load on the turbine is increased, then the speed of the turbine is decreased. By regulating the valves the steam flow rate is increased which will increase the speed of the turbine to the normal speed.

Throttle Governing

In this type of governing system the mass flow rate of the steam is controlled by a throttle valve whose operation is controlled by a combination of a servomotor & centrifugal governor

The Arrangement of Throttle Governing

The total throttle governing system consists of a gear pump, an oil sump, two regulating wings or pistons, two discharge pipe, a pivot, a connecting rod from the fulcrum of the centrifugal governor to the piston rod of the servometer, a centrifugal governor.

Operation

When the turbine is operating at the normal speed the position of the flyballs of the centrifugal governor maintains their normal posⁿ & the spear is also positioned at its normal posⁿ. The movement of centrifugal governor is controlled by a gear & lever mechanism attached to the turbine shaft.

- When the pivot is in betⁿ the link of piston rod of the servo motor & the fulcrum of the governor.
- The piston rod of the servo motor is attached with two wings; ~~so that~~ & the gear pump is used to supply high pr. oil to the servo motor cylinder.

Design of servomotor cylinder

It is attached with two discharge openings & three suction or, over flow openings along with two wings of the piston. The discharge pipe of the servo motors are attached with a common cylinder having a piston which is attached to the spear.

Operation

When the load on the turbine is high it will cause in the decrease of speed. As the governor speed is controlled by the turbine speed, then the centrifugal force on the fly ball will decrease. It will cause the decrease in amplitude of the rotation.

21/X/08

Condenser

Types

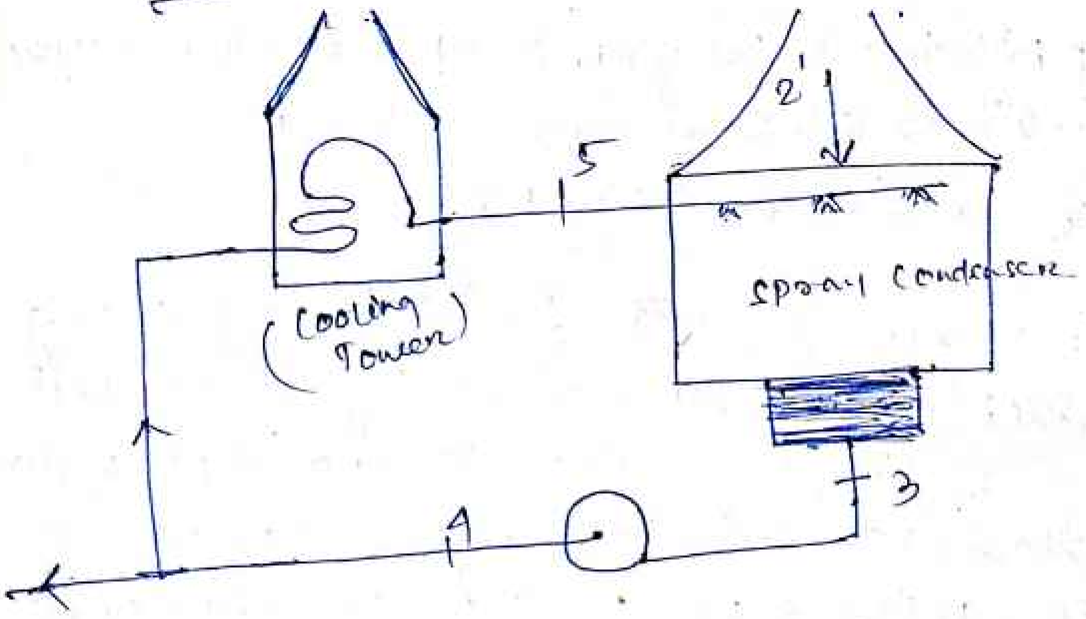
Broadly classified into 2 types.

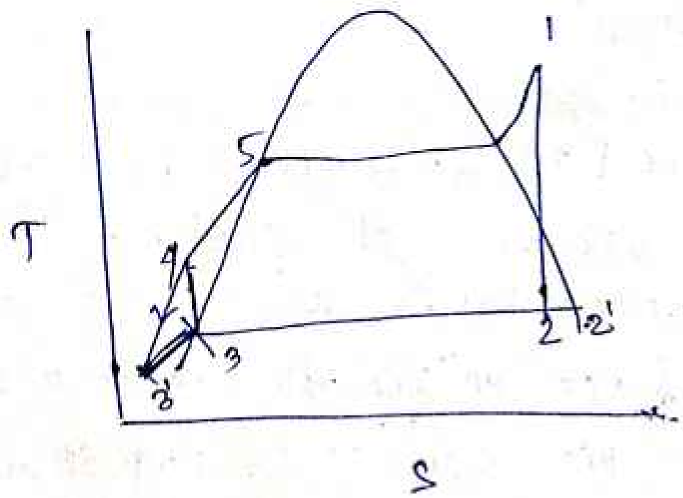
(I) Direct contact type

(II) Surface condenser

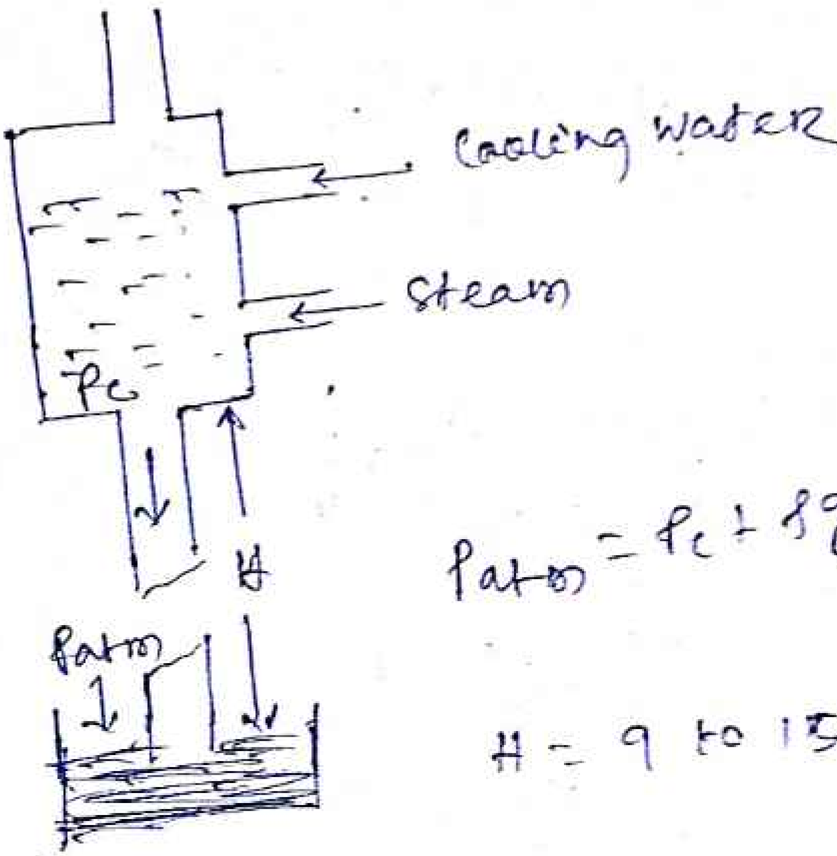
Direct Contact Type

(1) Spray condenser





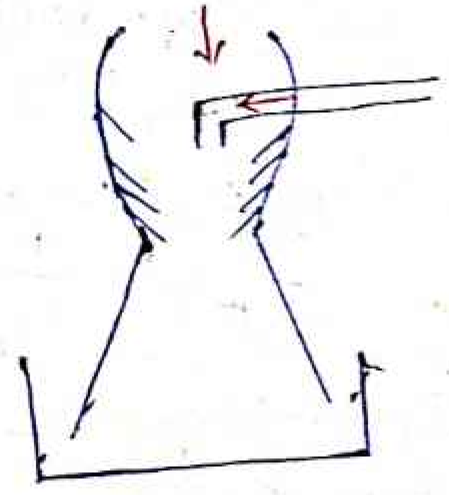
(ii) Barometric Direct Contact Condenser



$$P_{atm} = P_c + \rho g H$$

$H = 9$ to 15 m.

(iii) Jet Condenser



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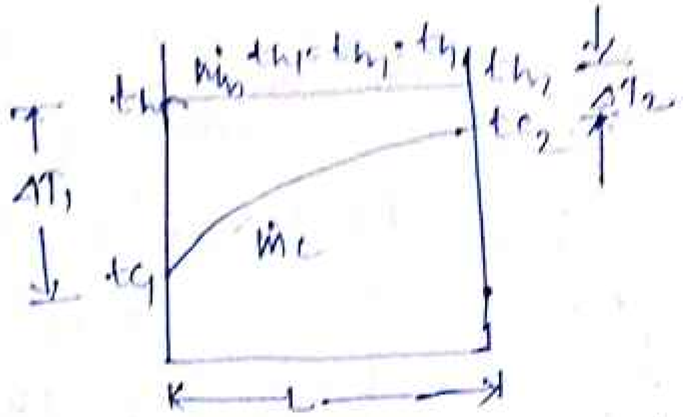
Surface Condenser

→ Shell & tube heat exchanger



$A_o U_o = A_i U_i = \frac{1}{R_e}$

$$LMTD = \theta_m = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

$$Q = U_o A_i \theta_m$$
$$= U_o A_o \theta_m$$

h_o

hot fluid

h_i ↑ cooling water flows inside the shell

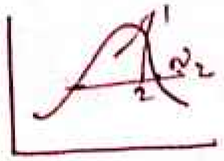
hot fluid

$$T_i \text{ ---}\frac{1}{h_i A_i}\text{---}\frac{L_i}{KA}\text{---}\frac{1}{h_o A_o}\text{--- } T_o$$

$$\frac{1}{R_e} = \frac{1}{\frac{1}{h_i A_i} + \frac{L_i}{KA} + \frac{1}{h_o A_o}}$$

Neglecting thermal resistance of conduction

$$\Rightarrow \frac{1}{R_e} = \frac{1}{\frac{1}{h_i A_i} + \frac{1}{h_o A_o}}$$



$$\dot{m}_a (h_g - h_f) \cdot t_h = \dot{m}_c c_{p_c} (\Delta t)_c$$

$p_a \times (\dot{m}_s v_2)$
$= \dot{m}_a R_a T_{sh}$

Q: A surface condenser receives 250 tonnes of hour of steam at 40°c with 12% moisture. Cooling water enters at 32°c & leaves at 38°c. The condenser pr. is 0.078 bar. The vel. of circulating water is 1.8 m/s. The tube outer diam. is 25.4 mm & its thickness is 1.25 mm. The overall HT co-efficient is 2600 $\frac{W}{m^2K}$. Then find mass flow rate of cooling water.

(ii) Mass of air leaking to the condenser

(iii) Length of the condenser.

Sol^n:

$\dot{m}_c = \frac{250 \times 1000}{3600}$

At 40°c

$x = 12\% = 0.88$

$v_g = ?, v_f = ?$

$t_{c_1} = 32°c$

$v_2 = v_f + x v_{fg}$

$t_{c_2} = 38°c$

$P_{sh} = 0.078$ bar

$V = 1.8$ m/s

$d_o = 25.4$ mm $= 0.0254$ m

$d_i = d_o - 2t$, $U = 2600 \frac{W}{m^2K}$

$\dot{m}_h (h_g - h_f) t_h = \dot{m}_c C_{p_c} (\Delta T)_c$

$C_{p_c} \to 4.2$

$h_m = h_f|_{40°c}$

$P_{sh} = P_a + P$

Condenser Efficiency

It is defined as the ratio of actual rise in cooling water temp. to the max. possible temp. rise.

Vacuum Efficiency

It is defined as the ratio of actual vacuum in the condenser to the difference of barometric pr. & ~~sat.~~ absolute pr. of ~~wet~~ steam at the condenser temp..

Q:- Exhaust steam having quality of 0.9 enters a surface condenser at an absolute pr. of 0.13 bar & comes out as water at 45°C. The ~~stub~~ circulating water enters at 30°C & leaves at 40°C. Distinguish the quantity of circulating water & condenser efficiency.

Soln.:-

From steam table

At $p = 0.13$ bar

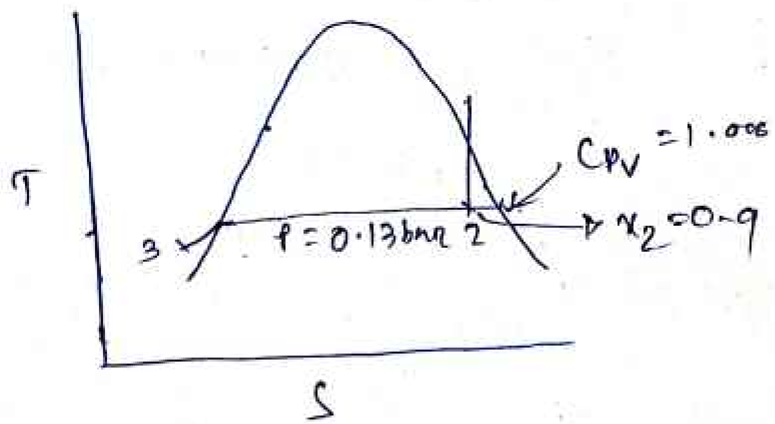
$t_{sat} = 50°C$

$t_{c_1} = 30°C$

$t_{c_2} = 40°C$

$\dot{m}_c = ?$

$\frac{\dot{m}_c}{\dot{m}_s} = ?$



Heat rejected by steam = Heat gained by the cooling water

⇒ Per ~~for~~ 1 kg of steam heat rejected ~~enthalpy drop~~ $= h_2 - h_3$

$h_2 = h_f + x_2 h_{fg} \big|_{0.13\text{ bar}}$

$h_3 = C_p (t - t_{ref.})$ ← 4.18 for water. $t_{ref} = 0$ (always for water)

$= C_p \times 45$

$= 4.18 \times 45$

Net amt. of heat rejected $= \dot{m}_s (h_2 - h_3)$

$\therefore \dot{m}_s (h_2 - h_3) = \dot{m}_c C_p (t_{c_2} - t_{c_1})$

$\Rightarrow \frac{\dot{m}_c}{\dot{m}_s} = 51.77$ kg of water / kg of steam

$\eta_{condenser} = \frac{t_{c_2} - t_{c_1}}{t_{sat} - t_{c_1}}$ ← max^m possible temp. rise

$= 47.5$

Q:- for a surface condenser the following data are given

Mean condenser temp. = 35°c.

Temp. of the hot well = 30°c
i.e. (After exit from condenser)

The vacuum inside the condenser = 69 cm of Hg

Barometer reading = 76 cm

Condensate collected at the rate of 16 kg/min.
i.e. steam flow rate

Cooling water enters at 20°c & leaves at 32.5°c.

Cooling water flow rate is 3700 kg/hr.

Calculate ~~the~~

(i) The mass of air present per 1 unit of condenser

(ii) The quality of steam at the condenser inlet.

(iii) Vacuum efficiency.

(iv) Condenser efficiency.

Solⁿ.:-

$P_c = 76\,cm - 69\,cm$

condenser pr. $= 7\,cm$ of Hg

$= (0.07\,(h) \times 13,600 \times 9.8)$ Pa

$= P_a + P_w$ ← Partial Pr. of air & " " water.

$= P_a + (P_{sat})_{35°C}$

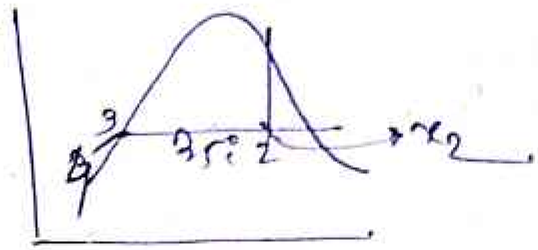
Q

Assuming the vol. of water $= V = 1\,m^3$

$P_a \times 1 = m_a \times R_a \times (273 + 35)$

$\Rightarrow m_a = ?$

$\dot{m}_s[(h_2 - h_3) + (h_3 - h_4)]$



$h_3 = h_f|_{35°C}$

$h_2 = h_f + x_2 h_{fg}$

$= \dot{m}_s((h_f + x_2 h_{fg} - h_f) + C_p(35-30)]$ $(h_g - h_f)|_{35°C}$

$h_3 - h_4 = C_p(t_{cond.} - t_{subcool})$

$= C_p(35-30)$

h_f

$\Rightarrow \dot{m}_s(x_2 h_{fg} + C_p \times 5) = \dot{m}_c \times C_p(32.5 - 20)$

$\Rightarrow x_2 = ?$

$$\eta_{vacuum} = \frac{69}{76 - (P_{sat})|_{35°C}}$$

Spec at 35°C → $P_{sat} = 0.023\,bar$

$= 0.023 \times 10^5$ Pa

$= \rho g h$

$\Rightarrow h = ?$

$$\eta_{condenser} = \frac{t_{c_2} - t_{c_1}}{35^\circ - t_{c_1}}$$

27/10/…

Cooling Towers

(I) Wet Cooling Tower

(II) Dry Cooling Tower

Wet Cooling Tower

$$\text{Approach}(A) = t_{c_2} - WBT$$

t_{c_2} → exit temp. of cooling tower of cooling water

$$\text{Range} = (R) = t_{c_1} - t_{c_2}$$

Cooling eff.

$$\eta_{cooling} = \frac{\text{Actual cooling}}{\text{Max}^m\text{. possible cooling}}$$

$$\eta_{cooling} = \frac{t_{c_1} - t_{c_2}}{t_{c_1} - WBT}$$

(I) Heating & Humidification

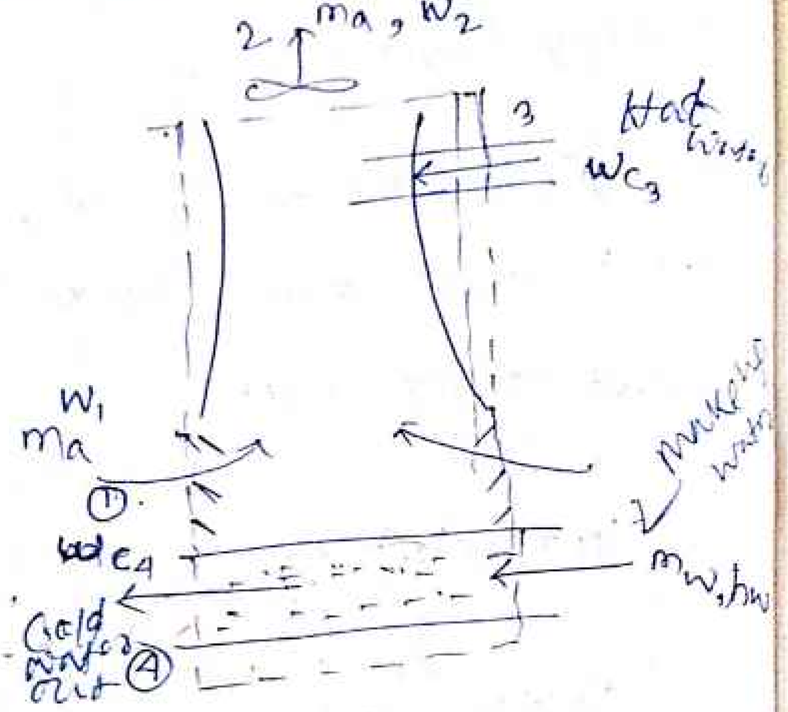
(II) According to the principle

(I) Mechanical Cooling Tower < forced / Induced

(II) Convective cooling Tower or, Natural cooling tower.

Design Analysis of Cooling Tower

29/x/08



$m_w = m_a(W_2 - W_1)$

$m_a \rightarrow$ Mass of dry air

Applying Energy balance eqn.

$$m_a h_1 + m_w h_w + \omega_{C_3} \times h_3 = m_a h_2 + \omega_{C_4} h_4$$

$\omega_{C_3}, \omega_{C_4}$ are mass flow rate of cooling water

$$\Rightarrow \omega_{C_3} h_3 - \omega_{C_4} h_4 = m_a(h_2 - h_1) - m_w h_w$$

$$= m_a(h_2 - h_1) - m_a(W_2 - W_1) h_w$$

Let $\omega_{C_3} = \omega_{C_4} = \omega_c$

$$\Rightarrow \omega_c(h_3 - h_4) = m_a(h_2 - h_1) - m_a(W_2 - W_1) h_w$$

$$\Rightarrow h_3 - h_4 = \frac{m_a}{\omega_c}\left[(h_2 - h_1) - (W_2 - W_1) h_w\right]$$

$$\Rightarrow C_{pw}(t_3 - t_4) = \frac{m_a}{\omega_c}\left[(h_2 - h_1) - (W_2 - W_1)h_w\right]$$

$$\Rightarrow \boxed{t_3 - t_4 = \frac{m_a}{\omega_c \times C_{pw}}\left[(h_2 - h_1) - (W_2 - W_1)h_w\right]}$$

$t_3 - t_4$ = Range

Approach (A) = $t_4 - t_{WBT}$

Q. Water at 30°c flows into a cooling tower at the rate of 1.15 kg per kg of air. Air enters the tower at the ~~rate of~~ DBT of 20°c & relative humidity of 60% & leaves at a DBT of 28°c & 90% relative humidity. Makeup water is supplied at 20°c. Determine

(a) The temp. of water leaving the tower.

(b) The fraction of water evaporated.

(c) Approach & Range of the cooling tower.

(d) The cooling efficiency of the cooling tower?

Sol.:- Given Datas;

$t_3 = 30°c$,

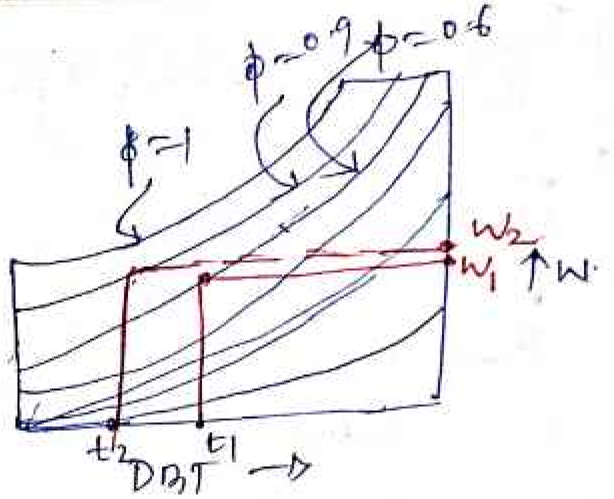
Assuming $m_a = 1\,kg$

$\therefore \omega_{c_3} = \omega_{c_4} = \omega_c = 1.15\,kg/kg$ of air.

$t_1 = 20°c$, $\phi_1 = 0.6$

$t_2 = 28°c$, $\phi_2 = 0.9$

$t_w = 20°c$



$$\phi = \frac{P_v}{P_s}$$

$P_s \leftarrow (P_{sat})_{t_1}$

~~$W_1 = 0.622 \frac{P_w}{P_t - P_w}$~~

$$W_1 = 0.622 \frac{P_v}{P_t - P_v}$$

$$m_w = m_a (W_2 - W_1) = 1 (W_2 - W_1)$$

$$t_3 - t_4 = \frac{m_a}{\omega_c \times C_{pw}} \left[(h_2 - h_1) - (W_2 - W_1) h_w \right]$$

$m_a \to 1$; $\omega_c \to 1.15$; $C_{pw} \to 4.18$; $h_w \to C_{pw} \times t_w$

$(h_2 - h_1) \to C_{pm}(t_2 - t_1)$, $C_{pm} \to 1.022$

At state 1: t_{WBT} = ? (From psychrometric chart)

$t_3 - t_4 \to$ Actual cooling

$t_3 - t_{WBT} \to$ Max^m. possible cooling

$$\eta_c = \frac{t_3 - t_4}{t_3 - t_{WBT}}$$

Mod...

Nuclear Power Plant

01/XI/08

Classification of Nuclear Power Plant

(I) Heterogenous
(II) Homogenous

Heterogenous Nuclear Power Plant

In the nuclear plant where nuclear fuel & moderator or coolant are not in direct contact. But the coolant flows over it.

Homogenous Nuclear Power Plant

where fuel & moderator or coolant are mixed

Depending upon nuclear flux

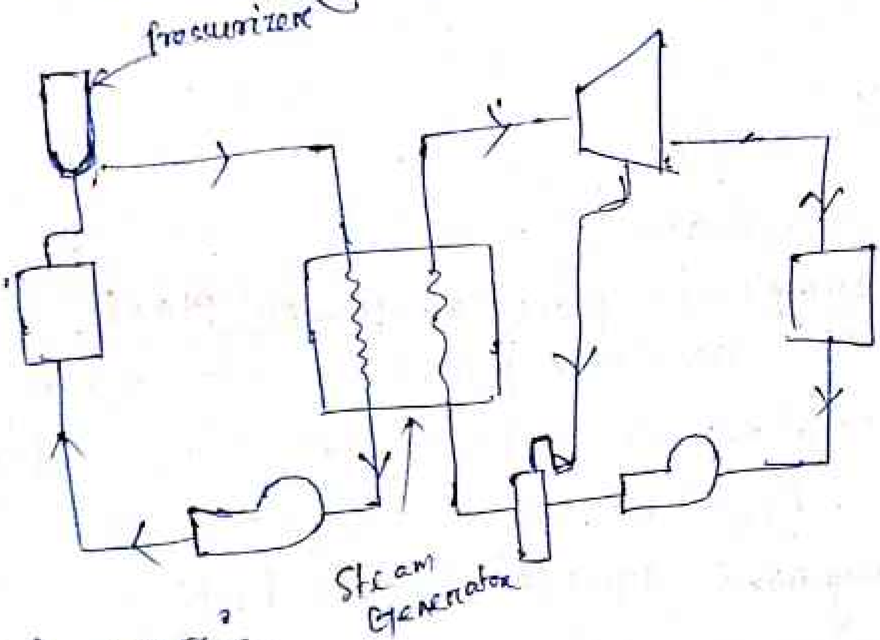
(I) Thermal Nuclear Power Plant
(II) Fast → Coolant used known as liquid metal
→ Moderator is not used.

Type of Nuclear flux	Type of coolant	Type of Moderator	Type of fuel
Thermal	Light water (H_2O),	H_2O, D_2O	Uranium
Fast	Heavy water (D_2O),	Graphite.	→ Enriched Uranium
	Carbon dioxide,	→ No moderator	Normal uranium (unstable)
	Liquid metals		→ Thorium.
			Plutonium.

03/11/08

Light temp. coolant & moderator.

(1) Pressurized ~~Reactor~~ Water Reactor (PWR)



300°C → 350°C
↳ P_s (85 - 90 bar)

A typical PWR, it contains 200 fuel assembly (Bunch of rods of fuel)

& each assembly contains 264 fuel rods &
24 guide tubes for control rods.
Grid space is given to separate the array of bunch

Fission	Fusion
(1) When heavy unstable nucleon is bombarded with neutrons, the nucleus splits into fragments of equal mass & energy is released.	(1) Some light elements fuse together with the release of energy.

Nuclear Reactors

A nuclear reactor is an apparatus in which nuclear fission is produced in the form of a controlled self-sustaining chain reaction.

① Fast Reactors

Sol^n:- Given Datas;

No. of stages = 20

Steam supply pr. = 15 bar, 300° C

Exhaust pr. = 0.1 bar.

Stage eff. of turbine, η_{stage} = 80 %

Reheat factor = 1.06

Total Power developed = 10665 kW.

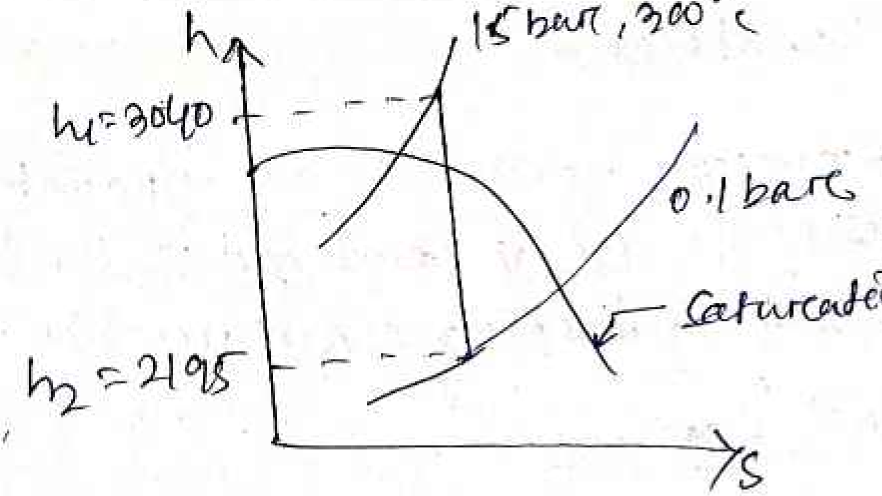
Steam flow rate, m_s

Steam pr. at a certain stage = 1 bar abs., $x = 1$

Blade exit angle = 25°

Blade speed ratio = $f = \frac{C_{bl}}{C_1} = 0.75$

Height of the blade = $h = \frac{1}{12} D$ (mean dia. of rotor)



$(\Delta h)_{isentropic} = h_1 - h_2 = 3040 - 2195 = 845 \ \frac{kJ}{kg}$.

$\eta_{overall} = \eta_{stage} \times$ Reheat factor.

$= 0.8 \times 1.06 = 0.848$

W.D. = Actual enthalpy drop

$= (\Delta h)_{isentropic} \times \eta_{overall}$

$= 845 \times 0.848$

$= 716.56 \ \frac{kJ}{kg}$

W.D. per stage per kg $= \frac{716.56}{20} = 35.83$ kJ

Total power = No. of stages $\times \dot{m}_s \times$ W.D/kg stage.

$\Rightarrow 10665 = 20 \times \dot{m}_s \times 35.83$

$\Rightarrow \dot{m}_s = 14.88$ kg/s.

(ii) Mean Diamt. of Rotor, D

W.D per kg per stage $= C_{bl} \times C_w$

$= C_{bl}(2C_1 \cos\alpha - C_{bl})$

$= C_{bl}(2 C_1 \cos 28 - C_{bl})$

Also $\frac{C_{bl}}{C_1} = 0.75$

$\Rightarrow C_1 = \frac{C_{bl}}{0.75} = 1.33 C_{bl}$

i.e. W.D per kg per stage $= C_{bl}(2 \times 1.33 C_{bl} \cos 25 - C_{bl})$

$= 1.41 C_{bl}^2$

$\therefore 1.41 C_{bl}^2 = 35.83 \times 10^3$

$\Rightarrow C_{bl} = 159.41$ m/s

$\Rightarrow C_1 = 212$ m/s

$C_{f_1} = C_1 \sin \alpha = ?$

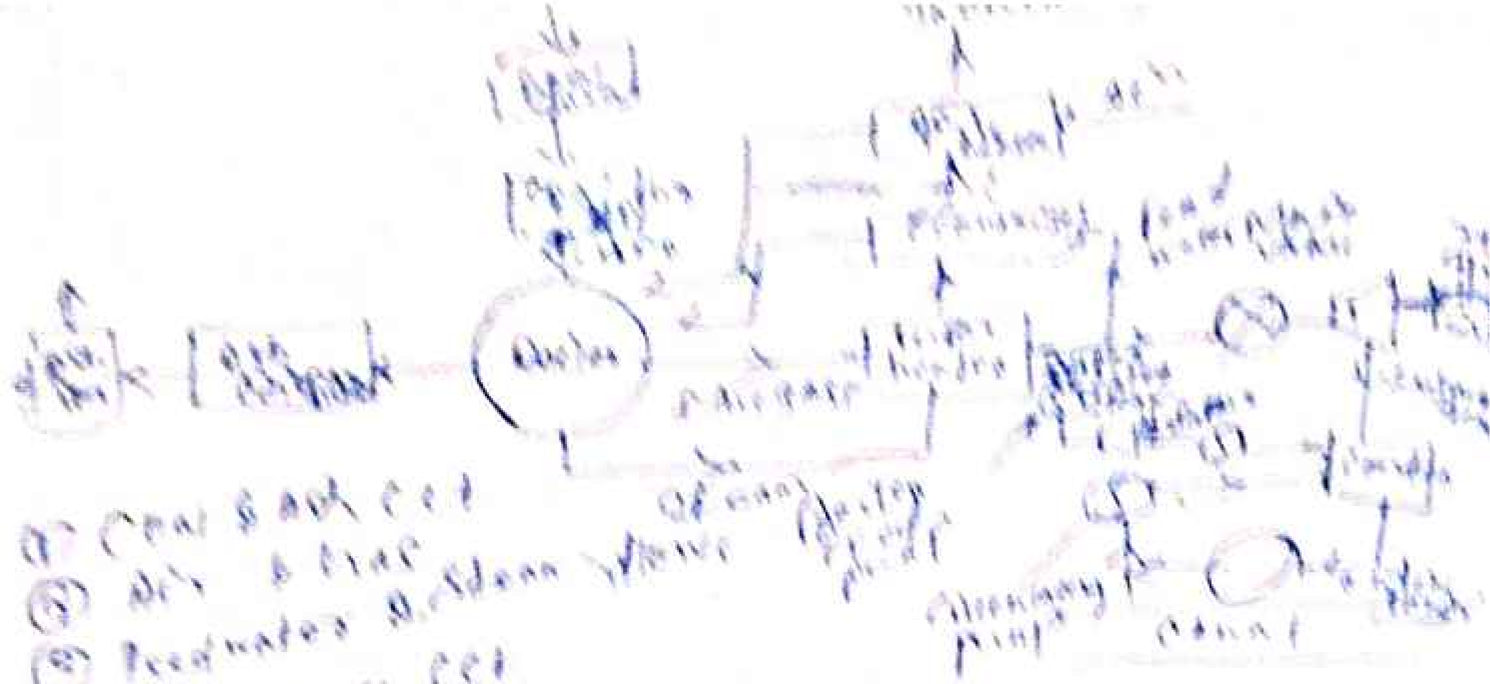
$v_g \Big|_{1 bar} = ?$

$\dot{m}_s = \frac{\pi (d+h) \times h \times C_{f_1}}{v_g}$

$\Rightarrow D = ?$

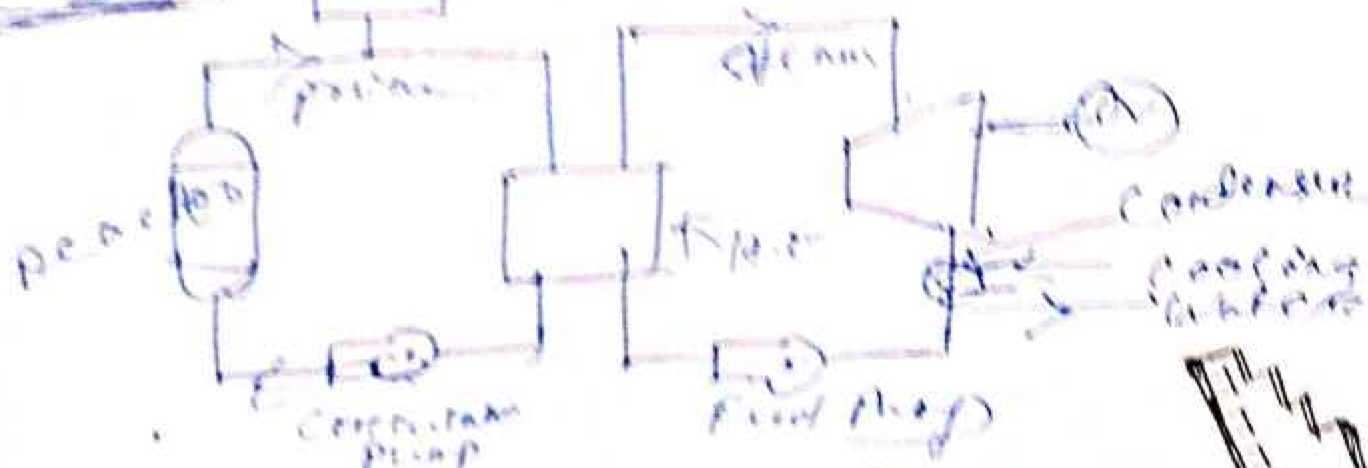
$h = \frac{D}{12} = ?$

$C_{bl} = \frac{\pi D N}{60} = ?$



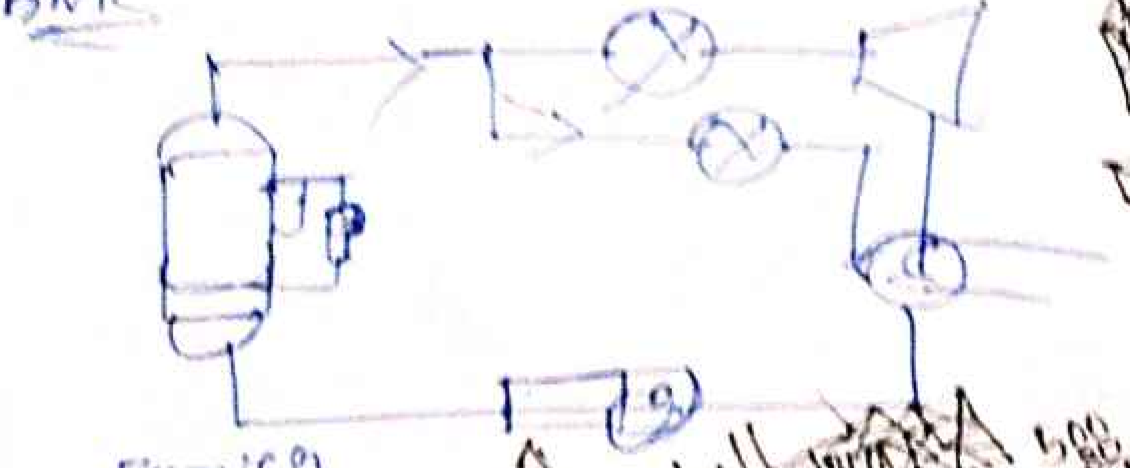
1. Coal & ash cct
2. Air & flue
3. Feedwater & steam flow
4. Cooling " cct

PWR



Water acts both coolant & moderator

BWR



Energy And Power

capacity to do work — Energy

- Energy appears in many forms but has one thing in common: energy is possessed of the ability to produce a dynamic, vital effect. Energy is associated with physical substance, but is not a substance itself.
- Energy exists in various forms e.g. mechanical, thermal, electrical etc. One form of energy can be converted into other by the use of suitable arrangements. Out of all these forms of energy, electrical energy is preferred due to the following advantages.
  1. can be easily transported from one place to another.
  2. Losses in transport are minm.
  3. Can be easily subdivided.
  4. Economical in use.
  5. Easily converted into other forms of energy.
  6. Easily controlled & regulated to suit requirements.

Ability to produce or create work

Power: Rate of doing work.

- Any physical unit of energy when divided by a unit of time automatically becomes a unit of power.
- The term "Power" is generally used in connection with the mechanical & electrical forms of energy.
- Power is primarily associated with mechanical work & electrical energy.
- Therefore, power can be defined as the rate of flow of energy & can state that a power plant is a unit built for production & delivery of a flow of mechanical & electrical energy.
- In common usage, a m/c or assemblage of equipment that produces & delivers a flow of mechanical or electrical energy is a power plant.
- Hence an I.C. Engine is a power plant, a water wheel is a power plant, etc.
- By the term plant is that assemblage of equipment, permanently located on some chosen site which receives raw energy in the form of substance capable of being operated on in such a way as to produce electrical energy for delivery from the power plant.

	9:00-10:05	10:05-11:10	11:10-11:15	11:15-12:10	12:10-01:05	01:05-02:00	02:00-02:55	02:55-03:50
MON	------	AMOS (F-13)			DMC (F-13)			
TUE	AMOS (F-13)				PPE (F-12)			
WED	PPE (F-12)				DMC (F-13)			
THU		DMC (F-13)			PPE (F-12)	MD PROJECT II		
FRI		PPE (F-12)			AMOS (F-13)	NC & SM LAB		
SAT		AMOS (F-13)			DMC (F-13)			

Sources of Energy

The various sources of energy are :

1. Fuels
   - Solids – Coal, coke anthracite etc.
   - Liquids – Petroleum & its derivatives
   - Gases – Natural gas blast furnace gas etc.
2. Energy stored in water
3. Nuclear Energy.
4. Wind Power
5. Solar Energy
6. Tidal power.
7. Geothermal Energy.
8. Thermoelectric power.

			Capacity
Narora Atomic power Station	Narora	U.P.	440MW
Rajasthan "	Rawatbhata	Rajasthan	1180MW
Tarapur Atomic "	Tarapur	Maharastra	1400MW
Kakrapar "	Kakrapur	Gujarat	440MW
Kaiga Nuclear PP.	Kaiga	Karnataka	660 MW
Kudankulam "	Kudankulam	T.N.	

→ 1kg of uranium is equivalent to energy obtained by 4500 tonnes of high grade coal.

Problems Related to Damped ~~Vibrⁿ~~ free vibrⁿ of Single Degree of freedom Systems

Q:- The mass of a spring-mass-dashpot is displaced by a distance of 0.05m from the equilibrium position and released. Find the eqⁿ of motion for the system for the cases, when (i) $\xi = 1.5$ (ii) $\xi = 1$ (iii) $\xi = 0.5$

Solⁿ:- Case-I

When $\xi = 1.5$, System is overdamped.

The eqⁿ. of motion for overdamped system is

$$x = C_1 e^{\lambda_1 t} + C_2 e^{\lambda_2 t} \longrightarrow (I)$$

where two roots λ_1 & λ_2 are given by

$$\lambda_1 = \left(-\xi + \sqrt{\xi^2 - 1}\right)\omega_n$$

$$\& \quad \lambda_2 = \left(-\xi - \sqrt{\xi^2 - 1}\right)\omega_n$$

for the given case, for $\xi = 1.5$, $\lambda_1 = -0.38\,\omega_n$
$\lambda_2 = -2.62\,\omega_n$

Substituting in eqⁿ. (I)

$$x = C_1 e^{-0.38\omega_n t} + C_2 e^{-2.62\omega_n t} \longrightarrow (II)$$

The value of const. C_1 & C_2 can be determined from initial conditions.

$x = 0.05$ at $t = 0$

$\dot{x} = 0$ at $t = 0$.

Differentiating eqⁿ. (II), we get

$$0.05 = C_1 + C_2 \longrightarrow (III)$$

$$\& \quad 0 = -0.38 C_1 \omega_n - 2.62 C_2 \omega_n$$

$$\text{or,} \quad 0 = 0.38 C_1 + 2.62 C_2 \longrightarrow (IV)$$

Solving (III) & (IV), we get

$$C_1 = 0.058, \; C_2 = -0.008$$

The solⁿ. becomes

$$x = 0.058 e^{-0.38\omega_n t} - 0.008 e^{-2.62\omega_n t}$$

Case 2 : When $\xi_p = 1$, System is critically damped.

The eqn of motion for such system is

$$x = (C_1 + C_2 t) e^{-\omega_n t}$$

Differentiating w.r.t t, we have

$$\dot{x} = -\omega_n (C_1 + C_2 t) e^{-\omega_n t} + C_2 e^{-\omega_n t}$$

Applying initial conditions,

$$0.05 = C_1$$

& $$0 = -\omega_n C_1 + C_2$$

or $$C_2 = \omega_n C_1 = 0.05\omega_n$$

Thus, the solⁿ becomes,

$$x = (0.05 + 0.05\omega_n t) e^{-\omega_n t}$$

$$x = 0.05(1 + \omega_n t) e^{-\omega_n t}$$

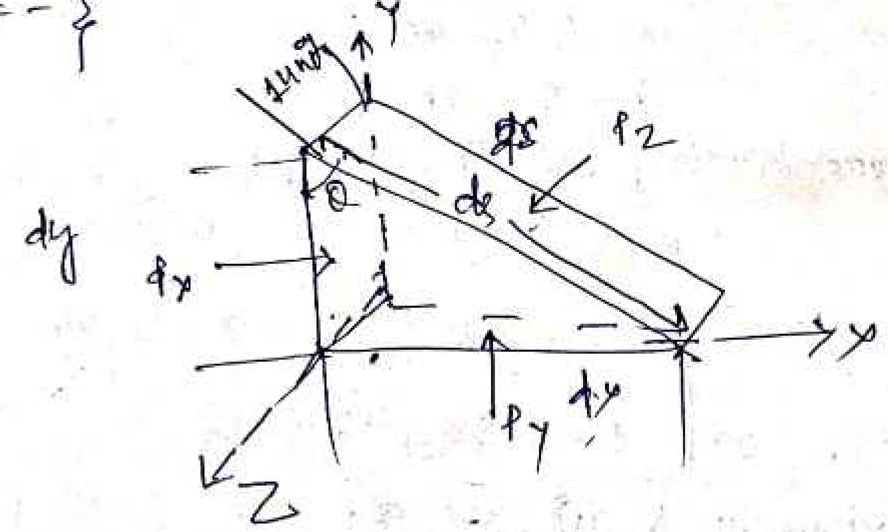
Case 3 : When $\tau = 0.5$, the system is underdamped.

The eqn. of motion for underdamped system is

$$x = A e^{-\xi \omega_n t} \sin(\sqrt{1 - \xi^2}\, \omega_n t + \phi_1)$$

Differentiating w.r.t. time, we have.

$\dot{x} = -\xi$



Pressure And its Measurement

Fluid pr. at a point

Consider a small area dA in a large mass of fluid. ~~is stationary~~ If the fluid is stationary, then the force exerted by the surrounding on the area will always be ⊥r. the surface.

Let dF is the force acting on the area dA in the normal condⁿ.

Then the ratio of $\frac{dF}{dA}$ is known as the intensity of pr. or simply pr. This ratio is represented by P.

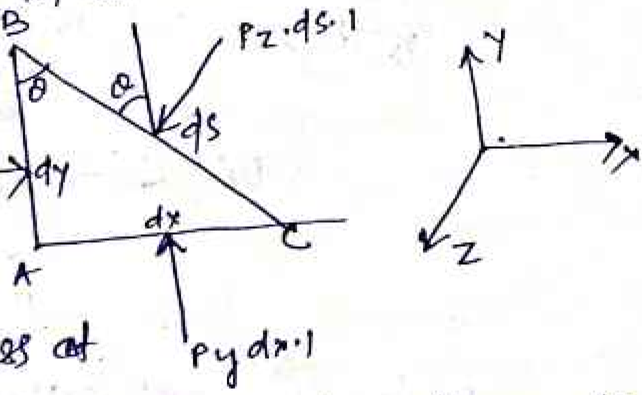
Mathematically, the pr. at a point in a fluid at rest is

$$\boxed{P = \frac{dF}{dA}}$$

∴ If the force(F) is uniformly distributed over the area(A), the pr. at any pt. $P = \frac{F}{A}$

Pascal's law

It states that the pr. or intensity of pr. at a point in a static fluid is equal in all dirⁿs.



The fluid element is $P_x \cdot dy \cdot 1$ of very small dimensions i.e. dx, dy & ds

Consider an arbitrary fluid element in a fluid mass at rest.

Let the width of the element ⊥r. to the plane of paper is unity. P_x, P_y & P_z are the pr. or intensity of pr. acting on the face AB, AC & BC respectively.

Let ∠ABC = θ

Then the forces acting on the element are

1. Pr. force normal to the surface and
2. Weight of the element in the vertical dirⁿ.

force on the face AB $= P_x \times$ Area of face AB
$= P_x \, dy \, 1$

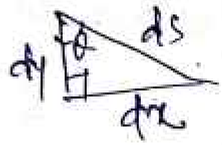
Similarly force on the face AC $= P_y \times dx \times 1$
" BC $= P_z \times ds \times 1$

Weight of element = (Mass of element) $\times g$
$= (\text{vol} \times \rho) \times g$
$= \left(\frac{AB \times AC}{2} \times 1\right) \rho g$

$W = mg$
$= V \times \rho \times g$
$= V \times \frac{m}{V} \times g$

ρ = Density of fluid

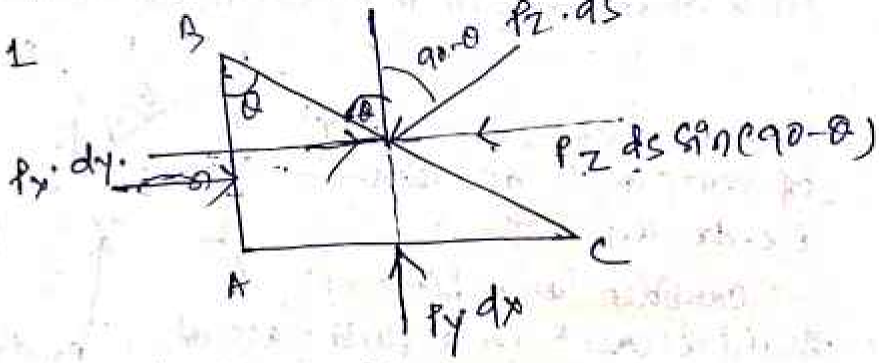
= Mass of element $\times g$
$= \text{vol} \times \rho \times g$
$= \frac{1}{2}$ (Area of $\Delta \times$ depth) $\times \rho g$
$= \frac{1}{2} dx \cdot dy \cdot 1 \times \rho g$
$= \frac{1}{2} dx \, dy \, \rho g$



$\sin\theta = \frac{dx}{ds}$

$\cos\theta = \frac{dy}{ds}$

Resolving the forces in x-dirⁿ.



$P_x \, dy \cdot 1 - P_z \, ds \sin(90-\theta) \cdot 1 = 0$

$\Rightarrow P_x \, dy - P_z \, ds \cos\theta = 0$

But $ds \cos\theta = dy = AB$

$\therefore P_x \, dy - P_z \, dy = 0$

$\Rightarrow P_x = P_z \longrightarrow (1)$

Similarly Resolving forces in y-dir^n, we get

$$P_y\, dx \times 1 - P_z \times ds \times 1 \times \cos(90° - \theta) - \frac{dx\, dy}{2} \times 1 \times \rho g = 0$$

$$\Rightarrow P_y dx - P_z ds \sin\theta - \frac{dx\,dy}{2} \times \rho g = 0$$

But $ds \sin\theta = dx$ & ~~the~~ Also the element is very small & hence weight is negligible.

$$P_y\, dx - P_z\, dx = 0$$

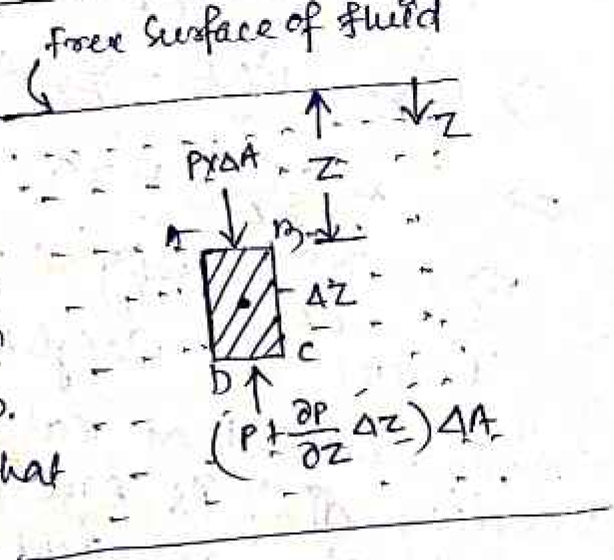
$$\Rightarrow P_y = P_z \rightarrow (2)$$

from (1) & (2), we get $P_x = P_y = P_z$.

The above eq^n shows that the pr. ~~p~~ at any pt. in x, y & z dir^n is equal.

Pr. variation in a fluid at rest

The pr. at any point in a fluid at rest is obtained by Hydrostatic Law. Which states that "the rate of increase of pr. in a vertical downward dir^n must be equal to the sp. weight of the fluid at that point."



Consider a small fluid element

Let ΔA = C/S area of element

ΔZ = Height of fluid element

p = Pr. on The face AB

Z = ~~Pr.~~ Distance of fluid element from free surface of Liq.

The forces acting on the fluid element are

1. Pr. force on AB = $p \times \Delta A$ & acting $\perp^r$ to the face AB in the Downward dir^n.

2. Pr. force on CD = $\left(p + \frac{\partial p}{\partial z} \Delta z\right) \times \Delta A$ acting

$\partial z \rightarrow \partial p$

$1 \rightarrow \frac{\partial p}{\partial z}$

$\Delta z \rightarrow \frac{\partial p}{\partial z} \Delta z$

$\perp^r$ to face CD, vertically upward dir^n.

3. Weight of fluid element = Density × g × vol.

Wt = $\rho \times g \times (\Delta A \times \Delta z)$

$w = mg$
$= \rho \times v \times g$

4. Pr. force on surfaces BC & AD are equal & opposite. For Equilibrium of fluid element, we have.

$$p \Delta A - \left(p + \frac{\partial p}{\partial z} \Delta z\right) \Delta A + \rho g (\Delta A \times \Delta z) = 0$$

$$\Rightarrow p \Delta A - p \Delta A - \frac{\partial p}{\partial z} \Delta z \Delta A + \rho g \Delta z \Delta A = 0$$

$$\Rightarrow + \frac{\partial p}{\partial z} = + \rho g$$

$$\Rightarrow \frac{\partial p}{\partial z} = w \quad (\because \rho g = w) \longrightarrow (1)$$

Where w = Weight Density of fluid.

Eq^n (1) states that rate of increase of pr. in a vertical dir^n is equal to weight density of the fluid at that point. This is Hydrostatic Law

By Integrating the above eq^n, we get

$$\int dp = \int \rho g \, dz$$

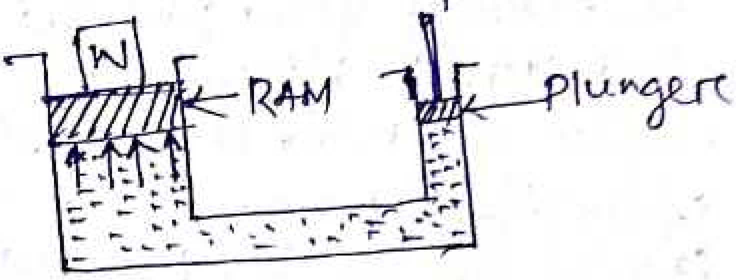
$$\Rightarrow \boxed{p = \rho g z}$$

Where p is the pr. above atmospheric p. & z is the height of the pt. from free surfaces.

$$\Rightarrow z = \frac{p}{\rho g}$$

Here z = Pressure Head.

Q:- A Hydraulic press has a ram of 30cm diameter & a plunger of 4.5cm diameter. Find the weight lifted by the hydraulic press when the force applied at the plunger is 500N.



Sol:- Dia. of ram, $D = 30cm = 0.3m$

Dia of plunger, $d = 4.5cm = 0.045m$

force on plunger, $F = 500N$

Find weight lifted $= W$

Area of ram, $A = \frac{\pi}{4} D^2 = \frac{\pi}{4}(0.3)^2 = 0.07068\ m^2$

Area of plunger, $a = \frac{\pi}{4} d^2 = \frac{\pi}{4}(0.045)^2 = 0.00159\ m^2$

Pressure intensity due to plunger

$= \frac{\text{force on plunger}}{\text{Area of plunger}} = \frac{F}{a} = \frac{500}{0.00159}\ N/m^2$

Due to Pascal's law, the intensity of pr. will be equally transmitted in all dirn. : Hence the pr. intensity at the ram, $= \frac{500}{0.00159} = 314465.4\ N/m^2$

But pr. intensity at ram $= \frac{\text{weight}}{\text{Area of ram}}$

$= \frac{W}{A} = \frac{W}{0.07068}\ N/m^2$

$\frac{W}{0.07068} = 314465.4 \Rightarrow W = 22222N = 22.222KN$

Kinematics of flow or Fluid Kinematics

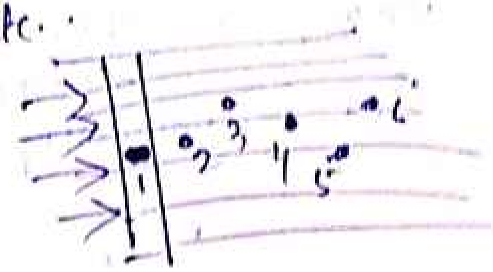
The branch of fluid mechanics which deals with the motion of fluid particles without considering the forces is known as Kinematics.
→ So fluid kinematics deals with the description of motion of the fluid particle w.r.t. space & time. The fluid motion can be studied by two ways.

(i) Lagrangian Method
(ii) Eulerian Method.

Lagrangian Method :- In this method any individual fluid particle is selected & it is followed during its motion & its velocity, accⁿ., density, etc., are described.

Eulerian Method :- In this method any point in the flow field is selected & the observations are made at that point to determine vel., accⁿ., density, etc.

→ This method is commonly used in fluid mechanics



Types of Fluid Flow

(i) Steady & (ii) Unsteady flows

→ Steady flow is defined as that type of flow in which the fluid characteristics like vel., pr., density, etc., at a point do not change with time.

Mathematically, $\left(\frac{\partial v}{\partial t}\right)_{x_0, y_0, z_0} = 0$

$$\left(\frac{\partial p}{\partial t}\right)_{x_0, y_0, z_0} = 0$$

$$\left(\frac{\partial \rho}{\partial t}\right)_{x_0, y_0, z_0} = 0$$

Where (x_0, y_0, z_0) is a fixed point in fluid flow field.

Unsteady flow is that type of flow, in which the vel., pr. or density at a point changes with w.r.t. time. Mathematically, $\left(\frac{\partial v}{\partial t}\right)_{x_0, y_0, z_0} \neq 0$

$$\left(\frac{\partial p}{\partial t}\right)_{x_0, y_0, z_0} \neq 0$$

Uniform & Non-Uniform flows

Uniform flow is defined as that type of flow in which the vel. at any given time does not change w.r.t. space (i.e. the length of dir^n of the flow).

Mathematically, $\left(\frac{\partial v}{\partial s}\right)_{t = const} = 0$

where ∂v = Change of vel.

∂s = Length of flow in the dir^n s.

Non-Uniform flow is that type of flow in which the velocity at any given time changes w.r.t space.

Mathematically, $\left(\frac{\partial v}{\partial s}\right)_{t = const} \neq 0$

✓ Laminar And Turbulent flows

Rate of flow or Discharge (Q)

It is defined as the quantity of a fluid flowing per second through a section of a pipe or a channel.

→ For an incompressible fluid (or Liquid) the rate of flow or discharge is expressed as the vol. of fluid flowing across the section per second.

→ for compressible fluids, the rate of flow is usually expressed as the weight of fluid flowing across the section.

(i) For Liquids the units of Q are m^3/s or litre/s

(ii) For gases the units of Q is kgf/s or Newton/s.

Consider a liquid flowing through a pipe in which.

A = C/S area of pipe

V = Average vel. of fluid across the section

∴ Discharge, Q = A × V

Continuity Eq^n.

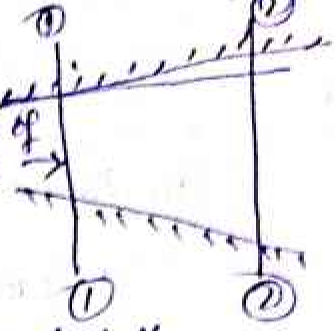
The eq^n. based on the principle of conservation of mass is called continuity eq^n.

→ Thus for a fluid flowing through the pipe at all C/S, the quantity of fluid per second is const.

Consider two cross-sections of a pipe.

Let v_1 = Avg. vel. of c/s 1-1
ρ = Density at section 1-1
A_1 = Area of pipe at section 1-1

& v_2, ρ_2, A_2 are corresponding values at section ②-②.



Then Rate of flow at section 1-1 = $\rho_1 A_1 v_1$
Rate of " " 2-2 = $\rho_2 A_2 V_2$.

According to law of Conservation of mass
Rate of flow at section ①-① = Rate of flow at section 2-2

Q = AV

$$\rho_1 A_1 v_1 = \rho_2 A_2 v_2$$

This eq^n. is applicable to the compressible as well as incompressible fluids & is called continuity Eq^n.

→ If the fluid is incompressible, then $\rho_1 = \rho_2$ & continuity eq^n. reduces to

$$A_1 v_1 = A_2 v_2$$

Mass of flow rate = $\frac{\text{Mass}}{\text{time}}$, $\left(\frac{kg}{s}\right)$

$= \left(\text{Density} \times \frac{\text{Vol.}}{\text{time}}\right)$

$\therefore \frac{dm}{dt} = \dot{m} = \rho \times Q$ [Q = Discharge Rate (m^3/s)]

A × vel. = $\frac{\text{Area} \times \text{length}}{\text{time}}$

$\Rightarrow \boxed{\dot{m} = \rho Q = \rho A V}$

5

20/02
→ Satyapriya Samal
→ Chandan Kr. Gupta
→ Jeevan Hickap
→ Sangram Beher

1500 } 3000
1500 } 2000
5000
5000
10,000

3000
3000
6000

40 Chap coupling

+ Design of Sliding Contact bearings
+ Journal bearings
+ foot step bearings.
+ Types and selection of ball and roller bearings. Dynamic & Static load ratings, Bearing life, Problem illustration.

Lubricants

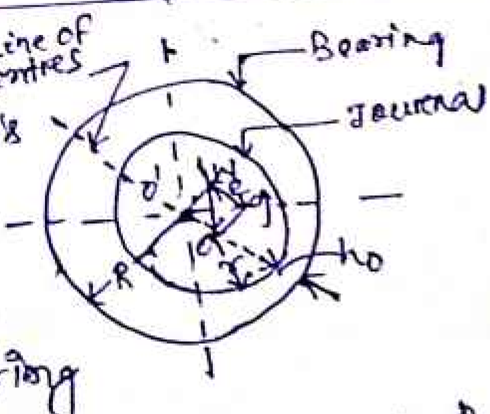
+ The lubricants are used in bearings to reduce friction bet^n the rubbing surfaces & to carry away the heat generated by friction. It also protects the bearing against corrosion.

All lubricants are classified into the following three groups -

1. Liquid — mineral oils & synthetic oils.
2. Semi-liquid — grease
3. Solid — Graphite

Terms used in Hydrodynamic Journal Bearing

A Hydrodynamic journal bearing in which O is the centre of the journal & O' is the centre of the bearings.



Let D = Dia. of the bearing
d = dia. of the journal
& l = Length of the bearing

1. Diametral Clearance :- It is the difference bet^n the dia. of the bearing & the journal.

$$\therefore \quad c = D - d$$

2. Radial clearance :- It is the difference bet^n the radii of the bearing & the journal.

$$c_1 = R - r = \frac{D-d}{2} = \frac{c}{2}$$

3. Diametral clearance ratio :- It is the ratio of the diametral clearance to the dia. of the journal.

$$\frac{c}{d} = \frac{D-d}{d}$$

4. Eccentricity :- It is the radial distance (in which) and for centre (O) of the bearing & the displaced centre of of the bearing under load. It is denoted by e.

5. Min. oil film thickness :- It is the min. distance of b/w the bearing & journal, under complete lubrication cond^n. It is denoted by h_o & occurs at the line of centres. Its value may be assumed as $c/4$.

6. Attitude or eccentricity Ratio :- It is the ratio of the eccentricity to the radial clearance

$$\epsilon = \frac{e}{c_1} = \frac{c_1 - h_o}{c_1} = 1 - \frac{h_o}{c_1} = 1 - \frac{2h_o}{c}$$
$$\left(\because c_1 = \frac{c}{2}\right)$$

7. Short & Long bearing

If the ratio of the length to the diameter of the journal (i.e. l/d) is less than 1, then the bearing is said to be short bearing.

If $\frac{l}{d} > 1 \rightarrow$ Long bearing.

→ When the length of the journal (l) is equal to the diam. of journal (d), then the bearing is called Square bearing

Bearing Characteristic Numbers & Bearing Modulus for journal Bearings

→ In Design of bearing the co-eff. of friction is of great importance, because it affords a means for determining the loss of power due to bearing friction.

→ It has been shown by experiments that the co-eff. of friction for a full lubricated journal bearing is a function of three variables i.e.

(i) $\frac{ZN}{p}$ (ii) $\frac{d}{c}$ & (iii) $\frac{l}{d}$

∴ The co-eff. of friction may be expressed as

$$\mu = \phi\left(\frac{ZN}{p}, \frac{d}{c}, \frac{l}{d}\right)$$

Where μ = Co-eff. of friction.

ϕ = A functional relationship

Z = Absolute viscosity of the lubricant in kg/m-s

N = Speed of the journal in r.p.m.

P = Bearing pr. on the projected bearing area in N/mm²

$= \frac{\text{Load on the journal}}{l \times d}$

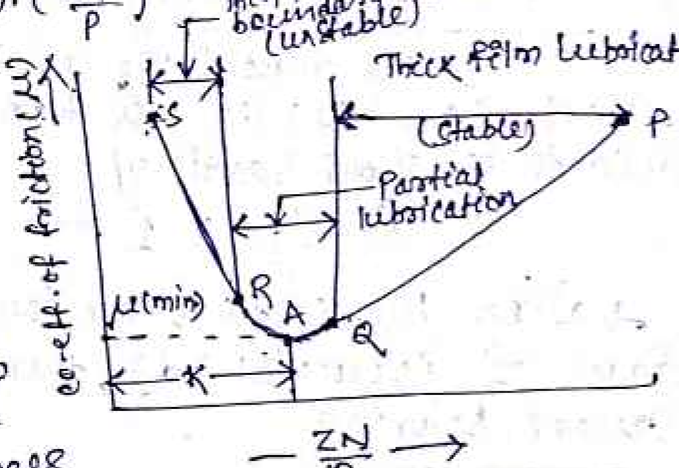
d = Diamt. of the journal

l = Length of the bearing.

C = Diametral clearance.

The factor $\frac{ZN}{P}$ is termed as bearing characteristic No. & is a dimensionless No.

→ The variation of co-eff. of friction with the operating values of bearing characteristic no. ($\frac{ZN}{P}$)



→ The curve PQ represents the region of thick film lubrication.

→ Betⁿ. Q & R, the viscosity (Z) or the speed (N) are so low or the pr. (P) is so great that their combⁿ. $\frac{ZN}{P}$ will reduce the film thickness so that partial metal to metal contact will result.

→ The thin film or boundary lubrication or imperfect lubrication exists betⁿ. R & S on the curve.

→ This is the region where the viscosity of lubricant ceases to be a measure of friction characteristics but the oiliness of the lubricant is effective in preventing complete metal to metal contact.

→ The part PQ represents the stable operating condⁿ., since from any point of stability, a decrease in viscosity (Z) will reduce $\frac{ZN}{P}$.

→ This will result in a decrease in co-eff. of friction (μ) followed by a lowering of bearing temp. that will raise the viscosity (Z)

→ The minⁿ. amount of friction occurs at A & at this point the value of $\frac{ZN}{P}$ is known as bearing modulus which is denoted by K.

→ The bearing should not be operated at this value of bearing modulus, because a slight decrease in speed or slight increase in pr. will break the oil film & make the journal to operate with metal to metal contact.

→ This will result in high friction, wear & heating. In order to prevent such cond?, the bearing should be designed for a value of $\frac{ZN}{p}$ at least three times the min? value of bearing modulus (K).

→ If the bearing is subjected to large fluctuations of load & heavy impacts, the value of $\frac{ZN}{p} = 15K$

→ When the value of $\frac{ZN}{p}$ is greater than K, then the bearing will operate with thick film lubrication or under hydrodynamic cond?.

→ When the value of $\frac{ZN}{p}$ is less than K, then the oil film will rupture & there is a metal to metal contact.

Co-eff. of friction for Journal Bearing.

To determine the co-eff. of friction for well lubricated full journal bearings, the following empirical relation may be used

Co-eff. of friction

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + K \quad \text{— (When Z is in kg/m-s & p is in N/mm}^2\text{)}$$

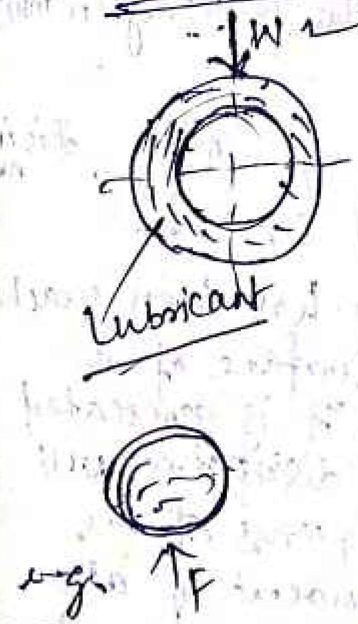
Where Z, N, p, d & c have usual meaning.

K = factor to correct for end leakage.

It depends upon the ratio of the length to the dia. of the bearings (i.e. l/d)

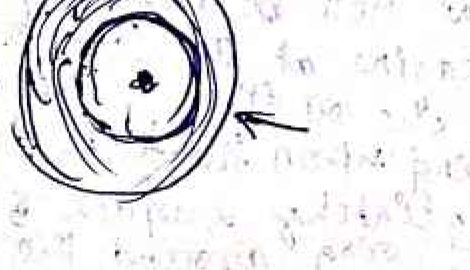
= 0.002 for l/d ratios of 0.75 to 2.8.

Hydrodynamic



Hydrostatic

Load supporting fluid film, separating two surface is created by external sources like pump, supplying fluid under pr.



Critical pr. of the journal Bearing

The pr. at which the oil film breaks down so that metal to metal contact begins, is known as critical pr. or the min^m operating pr. of the bearing. It may be obtained by the following empirical rel^n, i.e.

Critical pr. or min^m operating pr.,

$$p = \frac{ZN}{4.75\times10^6}\left(\frac{d}{c}\right)^2\left(\frac{L}{d+L}\right)\ N/mm^2$$

(When Z is in $\frac{kg}{m\text{-}s}$)

Sommerfeld No.

Sommerfeld No. is also a dimensionless parameter used extensively in the design of journal bearings.

Mathematically,

$$\text{Sommerfeld no.} = \frac{ZN}{p}\left(\frac{d}{c}\right)^2$$

For Design purpose, its value is taken as

$$\frac{ZN}{p}\left(\frac{d}{c}\right)^2 = 14.3\times10^6 \quad (Z \text{ is in } kg/m\text{-}s,\ p \text{ is in } N/mm^2)$$

Heat Generated in a Journal Bearing.

The heat generated in a bearing is due to the fluid friction & friction of the parts having relative motion. Mathematically, heat generated in a bearing

$Q_g = \mu.W.V$ N-m/s or J/s or watts

μ = co-eff. of friction.

W = Load on the bearing in N.

= Pr. on the bearing in N/mm² × Projected area of the bearing in mm²

= p × (l × d)

V = Rubbing velocity in m/s = $\frac{\pi dN}{60}$, d is in metre

N = Speed of the journal in rpm.

After the thermal equilibrium has been reached, heat will be dissipated at the outer surface of the bearing at the same rate at which it is generated in the oil film. The amount of heat dissipated will depend upon the temp. difference, size & mass of the radiating surface & on the amount of air flowing around the bearing.

Heat Dissipated by the bearing.

$$Q_d = CA\,(t_b - t_a)\ \text{J/s or W}$$

Where C = Heat dissipation co-eff. in $W/m^2/°C$

A = Projected area of the bearing in m^2 $= l \times d$

t_b = Temp. of the bearing surface in °C

t_a = " of the surrounding air in °C.

→ It has been shown by experiments that the temp. of the bearing (t_b) is approx. mid way betⁿ the temp. of the oil film (t_o) & the temp. of the outside air (t_a). In other words

$$t_b - t_a = \frac{1}{2}(t_o - t_a)$$

→ The temp. of the oil film is often called as the operating temp. of the bearing.

→ The mass of the oil to remove the heat generated at the bearing may be obtained by equating the heat generated to the heat taken away by the oil.

We know that the heat taken away by the oil.

$$Q_t = m \cdot S \cdot t \text{ J/s or watts}$$

m = Mass of the oil in kg/s

S = Sp. heat of the oil. Its value may be taken as 1840 to 2100 J/kg/°C

t = Difference betn. outlet & inlet temp. of the oil in °C.

Design Procedure for Journal Bearing

1. Determine the bearing length by choosing a ratio of l/d from table
2. Check the bearing pr., $P = \frac{W}{l.d}$ from table for probable satisfactory value.
3. Assume a lubricant from table & its operating temp (t_0). This temp. should be betⁿ. 26.5°C & 60° with 82°C as a maxᵐ. for high temp. installations such as steam turbines.
4. Determine the operating value of $\frac{ZN}{P}$ for the assumed bearing temp. & check this value with corresponding values in Table, to determine the possibility of maintaining fluid film operⁿ.
5. Assume a clearance ratio c/d from table.
6. Determine the co-eff. of friction (μ) by using the relation as discussed.
7. Determine the heat generated by using the formula
8. Determine the heat dissipated by " "
9. Determine the thermal equilibrium to see that the heat dissipated becomes at least equal to the heat generated. In case the heat generated is more than the heat dissipated then either the bearing is redesigned or it is artificially cooled by water.

Q:- Design a journal bearing for a centrifugal pump from the following data:

Load on the journal = 20,000 N; Speed of the journal = 900 rpm: Type of oil is SAE 10, for which the absolute viscosity at 55°c = 0.017 kg/m-s, Ambient temp. of oil = 15.5°c, Maxᵐ. bearing pr. for the pump = 1.5 N/mm² Calculate also mass of the lubricating oil reqd. for artificial cooling, if rise of temp. of oil be limited to 10°c. Heat dissipation co-eff. = 1232 W/m²/°c

Soln :- Given $W = 20,000\ N$ $t_0 = 55°C$

t_0 = oil film temp. $N = 900\ rpm$ $Z = 0.017\ kg/m\ s$

$t_a = 15.5°C$, $P = 1.5\ N/mm^2$

$l = ?$ $t = 10°C$, $C = 1232\ W/m^2\ °C$.

1) First of all, let us find the length of the journal (l). Assume the dia. of the journal (d) as 100mm. from table, we find the ratio of l/d for centrifugal pumps varies from 1 to 2. $1 \le \frac{l}{d} \le 2$

Let us take $\frac{l}{d} = 1.6$

$$l = 1.6\ d = 1.6 \times 100 = 160\ mm.$$

2.) Bearing pr., $P = \frac{W}{ld} = \frac{20,000}{160 \times 100} = 1.25$

Since the given bearing pr. for the pump is $1.5\ N/mm^2$, therefore the above value of P is safe & hence the dimensions of l & d are safe.

3) $\frac{ZN}{P} = \frac{0.017 \times 900}{1.25} = 12.24$

from table, we find that the operating value of

$$\frac{ZN}{P} = 28$$

∴ Safe.

→ The min. value of the bearing modulus at which the oil film will break is given by

$$3K = \frac{ZN}{P}$$

∴ Bearing modulus at the min. point of friction,

$$K = \frac{1}{3}\left(\frac{ZN}{P}\right) = \frac{1}{3} \times 28 = 9.33.$$

Since the calculated value of bearing characteristic no. $\frac{ZN}{P} = 12.24$ is more than 9.33, therefore the bearing will operate under hydrodynamic condition.

4) From table, we find that for centrifugal pumps, the clearance ratio $c/d = 0.0013$

5) We know that co-eff. of friction

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{P}\right)\left(\frac{d}{c}\right) + k$$

$$= \frac{33}{10^8} \times 12.24 \times \frac{1}{0.0013} + 0.002$$

$$= 0.0031 + 0.002 = 0.0051$$

6) Heat generated, $Q_g = \mu W V$

$$= \mu W \left(\frac{\pi d \cdot N}{60}\right) W \quad \left(\because v = \frac{\pi d N}{60}\right)$$

$$= 0.0051 \times 20,000 \left(\frac{\pi \times 0.1 \times 900}{60}\right)$$

$$= 480.7 \, W$$

7) Heat dissipated

$$Q_d = C A (t_b - t_a) \quad (\because A = l \times d)$$

$$= C\, l\, d\, (t_b - t_a) \, W$$

We know that

$$(t_b - t_a) = \frac{1}{2}(t_o - t_a)$$

$$= \frac{1}{2}(55^\circ - 15.5^\circ)$$

$$= 19.75^\circ C$$

$$\therefore Q_d = 1232 \times 0.16 \times 0.1 \times 19.75$$

$$= 389.3 \, W \quad (l \,\&\, d \text{ are taken in metres})$$

» We see that the heat generated is greater than the heat dissipated which indicates that the bearing is warming up. Therefore, either the bearing should be redesigned by taking $t_o = 63^\circ C$ or the bearing should be cooled artificially.

We know that the amount of artificial cooling reqd. = Heat generated − Heat dissipated

$$= Q_g - Q_d$$

$$= 480.7 - 389.3 = 91.4 \, W$$

Mass of lubricating oil reqd. for artificial cooling

Let m = Mass of the lubricating oil reqd. for artificial cooling in kg/s.

We know that the heat taken away by the oil

$$Q_t = m \cdot s \cdot t$$

$$= m \times 1900 \times 10 = 19000\, m \, W$$

$$(\because \text{Sp. heat of oil } (S) = 1840 \text{ to } 2100 \, J/kg/^\circ C)$$

Equating this amount of artificial cooling reqd

$1900m = 91.4$

$\Rightarrow m = \frac{91.4}{19000} = 0.0048$ kg/s $= 0.288$ kg/min

Thrust Bearings

A thrust bearing is used to guide or support the shaft which is subjected to a load along the axis of the shaft.

- Such type of bearings are used in turbines & propeller shafts. The thrust bearings are of following types

① foot step or pivot bearing

② Collar bearings

In a foot step or pivot bearing, the loaded shaft is vertical & the end of the shaft rests within the bearing.

In case of collar bearing, the shaft continues through the bearing. The shaft may be vertical or horizontal with single collar or many collars.

In designing, it is assumed that the pr. is uniformly distributed throughout the bearing surface.

Let W = Load transmitted over the bearing surface.

R = Radius of the bearing surface (or shaft)

A = c/s area of the bearing surface.

p = Bearing pr. per unit area of the bearing surface betn rubbing surfaces

μ = Co-eff. of friction

N = Speed of the shaft in rpm.

When the pr. is uniformly distributed over the bearing area, then $p = \frac{W}{A} = \frac{W}{\pi R^2}$

& total frictional torque,

$T = \frac{2}{3} \mu W R$

∴ Power lost for friction

$$P = \frac{2\pi N T}{60} \text{ watts}$$

→ When the counter boring of the shaft is considered, then the bearing pr.

$$p = \frac{W}{\pi(R^2 - r^2)}$$, where r = Radius of counter bore

& the total frictional torque,

$$T = \frac{2}{3} \mu W \left(\frac{R^3 - r^3}{R^2 - r^2} \right)$$

If $W \geq 20 kN$ ⇒ $d \geq 100 mm$ $\frac{L}{d} = 2$, $L = 2d$

Q1:- A foot step bearing supports a shaft of 150 mm dia which is counter bored at the end with a hole dia. of 50mm. If the bearing pr. is limited to $0.8 N/mm^2$ & the speed is 100 rpm, find:

1. The load to be supported.
2. The power lost in friction.
3. The heat generated at the bearing.

Assume coeff. of friction = 0.015

Sol:- Given, D = 150mm, or, R = 75mm

d = 50mm, or r = 25 mm, $p = 0.8 N/mm^2$

N = 100 rpm, $\mu = 0.015$

Load to be supported

Let W = load to be supported

Assuming that the pr. is uniformly distributed over the bearing surface,

∴ Bearing pr. (P), $p = \frac{W}{\pi(R^2 - r^2)}$

$$\Rightarrow 0.8 = \frac{W}{\pi[(75)^2 - (25)^2]} = \frac{W}{15710}$$

$$\Rightarrow W = 0.8 \times 15710 = 12568 N$$

2. Power lost in friction.

Total frictional torque

$$T = \frac{2}{3}\mu W\left(\frac{R^3 - r^3}{R^2 - r^2}\right)$$

$$= \frac{2}{3} \times 0.015 \times 12568 \left[\frac{75^3 - 25^3}{75^2 - 25^2}\right] \text{ N-m}$$

$$= 125.68 \times 81.25$$

$$= 10212 \text{ N-mm} = 10.212 \text{ N-m}$$

∴ Power lost in friction

$$P = \frac{2\pi N T}{60} = \frac{2\pi \times 100 \times 10.212}{60} = 107\,W$$

$$= 0.107 \text{ kW}$$

3. Heat generated at the bearing.

We know that heat generated at the bearing

= Power lost in friction

= 0.107 kW or kJ/s

= 0.107 × 60 = 6.42 kJ/min.

Rolling Contact Bearings

→ In rolling contact bearings, the contact betⁿ the bearing surfaces is rolling instead of sliding as in sliding contact bearings.

→ The ordinary sliding contact bearing starts from rest with practically metal-to-metal contact & has a high co-eff. of friction.

→ In rolling contact bearing it has a low starting friction.

→ Due to the low friction offered by rolling contact bearings, these are called antifriction bearings.

Advantages & Disadv. of Rolling contact bearings over Sliding contact bearings

Adv. 1. Low starting & running friction except at very high speeds.

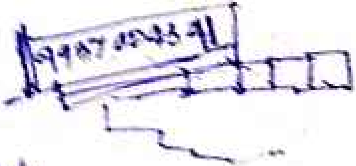
2. Ability to withstand momentary shock loads.

3. Accuracy of shaft alignment.

4. Low cost of maintenance, as no lubrication is reqd while in service.

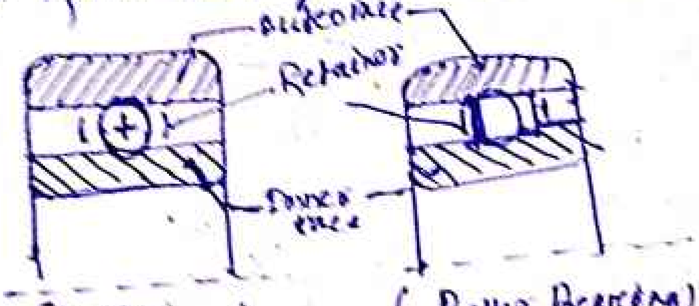
Demerits

1. More noisy at very high speeds.
2. Low resistance to shock loading.
3. Design of bearing housing is complicated.
4. More initial cost.



Types of Rolling Contact Bearings

1. Ball Bearings → are used for light loads
2. Roller Bearings → " " heavier loads.



(Ball bearing) (Roller Bearing)

→ The Ball & Roller bearing consists of an inner race which is mounted on the shaft or journal & the outer race which is carried by the housing or casing.

→ In betⁿ the inner & outer race, there are balls or rollers

→ A no. of balls or rollers are used & these are held at proper distances by retainers so that they do not touch each other.

→ The retainers are thin strips & is usually in two parts which are assembled after the balls have been properly spaced.

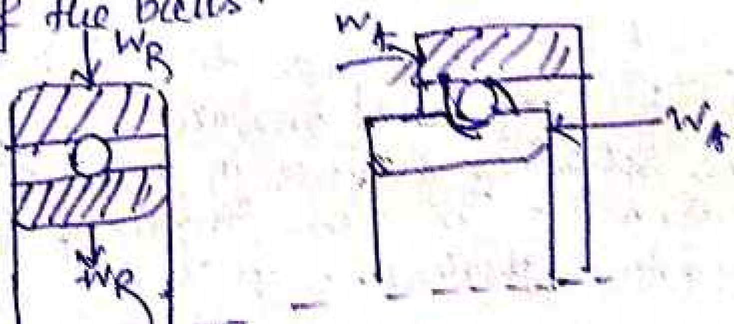
The rolling contact bearings, depending upon the load to be carried, are classified as

① Radial bearings

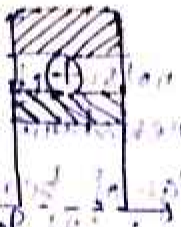
②. Thrust bearings

→ When a ball bearing supports only a radial load (W_R) the plane of rotation of the ball is normal to the centre line of the bearing.

→ The action of thrust load (W_A) is to shift the plane of rotation of the balls.

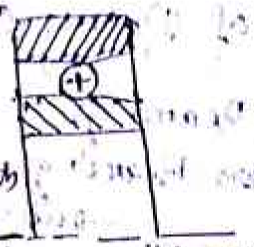


Types of Radial Ball Bearings



1. Single row deep groove bearing.

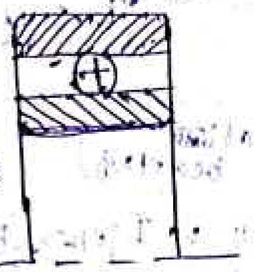
→ The load carrying capacity of a ball bearing is related to the size & no. of the balls.



2. Filling notch bearing

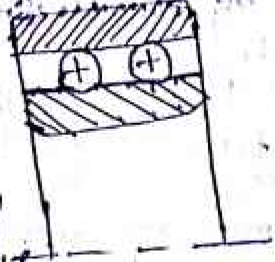
These bearings have notches in the inner & outer races which permit more balls to be inserted than in a deep groove ball bearings.

→ It has a ~~greater~~ larger bearing load capacity.



3. Angular contact bearing

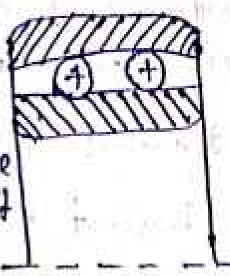
These bearings have one side of the outer race cut away to permit the insertion of more balls than in a deep groove bearing but without having a notch cut into both races.



4. Double row bearing

The double row bearing is appreciably narrower than two single row bearing.

→ These bearing made with radial or angular contact betn. the balls & races.



5. Self-Aligning bearing

These bearing permit shaft deflection within 2-3 degrees.

→ Normal bearing are too small to accomodate any appreciable misalignment of the shaft relative to the housing.

Self-Aligning bearings are of two types

(a) Externally Self-Aligning bearing

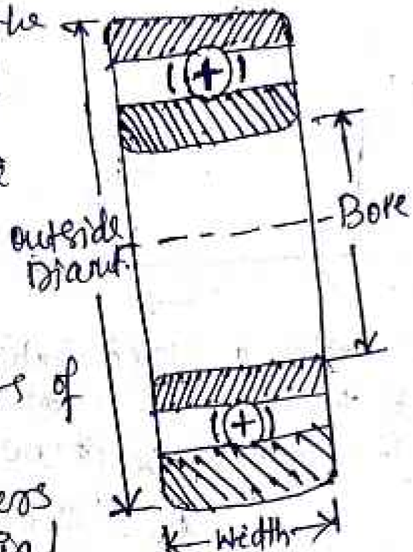
(b) Internally Self-Aligning bearing.

(a) E.S.A.B – The outside dia. of the outer race is ground to a spherical surface which fits in a mating spherical surface in a housing.

(b) I.S.A.B – The inner surface of the outer race is ground to a spherical surface.

Standard Dimensions & Designations of Ball Bearings

- Dimensions are a fun. of the bearing bore & the series of bearing.
- There is no standard for the size & no. of steel balls.
- The bearings are designated by a no.
- In general, the no. consists of 3 digits.
- Additional digits or letters are used to indicate special features e.g. deep groove, filling notch etc.



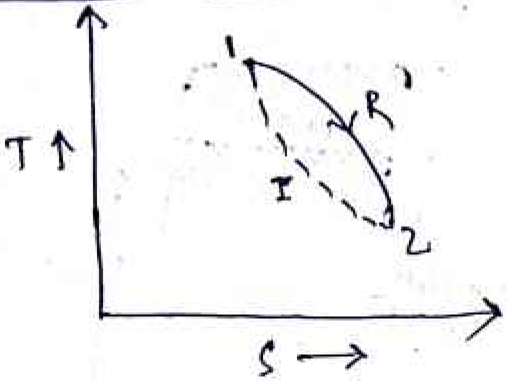
- The last 3 digits give the series & the bore of the bearing
- The last two digits from 04 onwards, when multiplied by 5, give the bore diam. in mm.
- The 3rd from the last digit designates the series of the bearing.
- The most common ball bearings are available in four series as follows

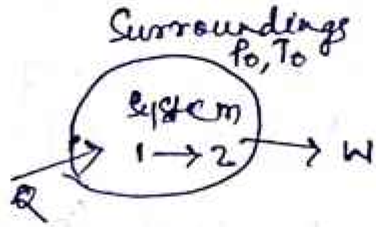
1. Extra Light (100), 2. Light (200)
3. Medium (300), 4. Heavy (400)

Notes

- If a bearing is designated by the no. 305, it means that the bearing is of medium series whose bore is 05 × 5, i.e. 25 mm
- The extra Light & Light series are used where the loads are moderate & shaft sizes are comparatively large & also where available space is limited.
- The medium series has a capacity 30 to 40 percent over the light series.
- The heavy series has 20 to 30 % capacity over the medium series. This series is not used extensively in industrial Appln.

Max^{m} Work in a Reversible Process





Let us consider a closed stationary system undergoing a reversible process R from state 1 to state 2 interacting with the surroundings at P_0, T_0. Then by the 1st law

$$Q_R = dU + W$$

$$\Rightarrow Q_R = (U_2 - U_1) + W_R \rightarrow (1)$$

If the process were irreversible, as represented by dotted line I connecting the same equilibrium end states.

$$Q_I = (U_2 - U_1) + W_I \rightarrow (2)$$

from eq^{n}. (1) & (2).

$$Q_R - Q_I = (U_2 - U_1) + W_R - U_2 + U_1 - W_I$$

$$\Rightarrow Q_R - Q_I = W_R - W_I \rightarrow (A)$$

Now, $\Delta S_{sys.} = S_2 - S_1$

& $\Delta S_{surr.} = -\frac{Q}{T_0}$

By the 2nd law $\Delta S_{univ} \geq 0$

for a Reversible process

$\Delta S_{sys} + \Delta S_{surr.} \rightarrow \Delta S_{univ.} = S_2 - S_1 - \frac{Q_R}{T_0} = 0$

$$\Rightarrow Q_R = T_0 (S_2 - S_1) \rightarrow (3)$$

for an irreversible process,

$$\Delta S_{univ} > 0$$

$$\therefore S_2 - S_1 - \frac{Q_I}{T_0} > 0$$

$$\therefore Q_I < T_0 (S_2 - S_1) \rightarrow (4)$$

from eq^{n}. (3) & (4),

$$Q_R > Q_I$$

from eq^{n}. (A) & (4),

done by ... irreversible process bet the same end states.

$$\theta_1 = t_{h_1} - t_{c_2} \qquad t_{h_1} > t_{h_2}$$

$$\theta_2 = t_{h_2} - t_{c_1} \qquad t_{c_2} > t_{c_1}$$

$$đQ = U\,dA\,(t_h - t_c)$$

✓ $dt_h = \qquad dt_c = t_{c_2} - t_{c_1}$

$$S_{in} - S_{out} + S_{gen} = (\Delta S)_{system}$$

$$S_2 - S_1 = \Sigma \frac{Q}{T} + S_{gen}$$

$$\Rightarrow S_1 - S_2 + \Sigma \frac{Q}{T} + S_{gen} = \frac{m_1 s_1}{s} \cdot \frac{KJ}{Kg\,K} = \frac{KJ}{s \cdot K}$$

Now $đW_{max} = đW + đW_c$.

$$= đQ + dm_1\left(h_1 + \frac{V_1^2}{2} + gZ_1\right) - dm_2\left(h_2 + \frac{V_2^2}{2} + gZ_2\right) - d\left(U + mgZ + \frac{mV^2}{2}\right)_{c.s} + đQ\left(\frac{T_0}{T} - 1\right)$$

$$\Rightarrow đW_{max} = đQ + dm_1\left(h_1 + \frac{V_1^2}{2} + gZ_1\right) - dm_2\left(h_2 + \frac{V_2^2}{2} + gZ_2\right) - d\left(U + \frac{mV^2}{2} + mgZ\right) + \frac{đQ}{T}(T_0) - đQ$$

$$= dm_1\left(h_1 + \frac{V_1^2}{2} + gZ_1\right) - dm_2\left(h_2 + \frac{V_2^2}{2} + gZ_2\right) - d\left(U + \frac{mV^2}{2} + mgZ\right) + T_0\{dS - dm_1 s_1 + dm_2 s_2\}$$

$$\Rightarrow đW_{max} = dm_1\left(h_1 + \frac{V_1^2}{2} + gZ_1 - T_0 s_1\right) - dm_2\left(h_2 + \frac{V_2^2}{2} + gZ_2 - T_0 s_2\right) - d\left(U - T_0 S + \frac{mV^2}{2} + mgZ\right)_{c.s}.$$

The Reversible work Done in a steady flow process

(i) $dm_1 = dm_2 = dm$ (say)

& $d(U - T_0S + \frac{mv^2}{2} + mgZ) = 0$

$\therefore dW_{max} = dm[(h_1 + \frac{V_1^2}{2} + gZ_1 - T_0s_1) - (h_2 + \frac{V_2^2}{2} + gZ_2 - T_0s_2)]$

$\therefore$ Total work o/p

$W_{max} = m[(h_1 + \frac{V_1^2}{2} + gZ_1 - T_0s_1) - (h_2 + \frac{V_2^2}{2} + gZ_2 - T_0s_2)]$

$= (H_1 + \frac{mV_1^2}{2} + mgZ_1 - T_0S_1) - (H_2 + \frac{mV_2^2}{2} + mgZ_2 - T_0S_2)$

$mh_1 = H$
$ms_1 = S_1$

Let $H - T_0S = B$

Where B is called Keenan fun.

$\therefore W_{max} = (B_1 + \frac{mV_1^2}{2} + mgZ_1) - (B_2 + \frac{mV_2^2}{2} + mgZ_2)$

$B + \frac{mV^2}{2} + mgZ$ = Availability fun. for a steady flow process.

$= \Psi$

$W_{max} = \Psi_1 - \Psi_2$

Reversible work for a closed system or Non-flow system

$dW_{rev}|_{closed\ sys.} = -d(E - T_0S)_\sigma$

for a closed system, $dm_1 = dm_2 = 0$

$\therefore$ The eqn. becomes,

$dW_{max} = -d(U - T_0S + \frac{mV^2}{2} + mgZ)_\sigma$

$= -d(E - T_0S)_\sigma$

Where $E = U + \frac{mV^2}{2} + mgZ$

for a change of state of the system from the initial state 1 to the final state 2,

$W_{max} = E_1 - E_2 - T_0(S_1 - S_2)$ → (1)

i.e. $-[(E_2 - E_1) - T_0(S_2 - S_1)]$

Neglecting K.E. & P.E., eqn. (1) becomes

$W_{max} = (U_1 - T_0S_1) - (U_2 - T_0S_2)$

$= (U_1 - U_2) - T_0(S_1 - S_2)$

Useful work   Pdv → Displacement W.D.

All of the work W of the system would not be available for delivery, since a certain portion of it would be spent in pushing out the atmosphere.

i.e. W_c = Work o/p by the system − W.D. on the surround.

Unsteady

$W_c = W_{act} - \int_1^2 P_0 dV$

$= W_{actual} - P_0 (V_2 - V_1)$

Max^m useful work by a closed system

V.C.

$W_{max}|_{useful\ work} = W_{max\ for\ a\ rev.} - P_0 \int_1^2 dV$ ← vol. doesn't changes

$= \int p\,dV$

$W_{1-2} = $ since $dV = 0$, $W_{1-2} = 0$

open system $W_{1-2} = (P_2 - P_1) V$

$W_{1-2} = \int V dP$

$W_{mu} = (U_1 - U_2) - T_0 (S_1 - S_2) - P_0 (V_2 - V_1)$

$= (U_1 - T_0 S_1 + P_0 V_1) - (U_2 - T_0 S_2 + P_0 V_2)$

$= \phi_1 - \phi_2$

Where $U - T_0 S + P_0 V = \phi$ = Availability func. w.r.t. a closed system

Max^m. useful work for an open system

Steady

Max^m. work o/p. for a control vol. having a steady flow process

$W_{mu} = W_{max} - \int_1^2 P_0 dV = W_{rev.}$ or W_{max}

$= \psi_1 - \psi_2$

$dW_{mu} = dm_1 (h_1 + \frac{V_1^2}{2} + gZ_1 - T_0 S_1) - dm_2 (h_2 + \frac{V_2^2}{2} + gZ_2 - T_0 S_2) - d(E - T_0 S) - \phi(P_0 dV)$

$= dm_1 (h_1 + \frac{V_1^2}{2} + gZ_1 - T_0 S_1) - dm_2 (h_2 + \frac{V_2^2}{2} + gZ_2 - T_0 S_2) - d(E - T_0 S) - d(P_0 V)$

$= dm_1 (h_1 + \frac{V_1^2}{2} + gZ_1 - T_0 S_1) - dm_2 (h_2 + \frac{V_2^2}{2} + gZ_2 - T_0 S_2) - d(E - T_0 S + P_0 V)$

Availability :- It is defined as the ·

- → Available Energy or Exergy
- → Unavailable Energy or Anergy

In a steady flow system, the vol. of the system does not change. Hence the max^m. useful work would remain same. i.e. no work done on the atmosphere.

$$(dW_u)_{max} = dW_{max}$$

for unsteady-flow open system or a closed system the vol. of the system changes ∴

$$\therefore (dW_u)_{max} = dW_{max} - P_0 dV$$