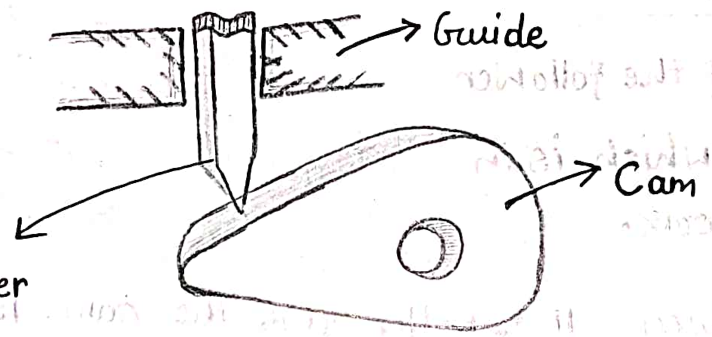
Theory of Machine & Mechanism

Unit - 1 Cams and Followers

The cam is a mechanical member which provides the motion to a follower by direct contact.

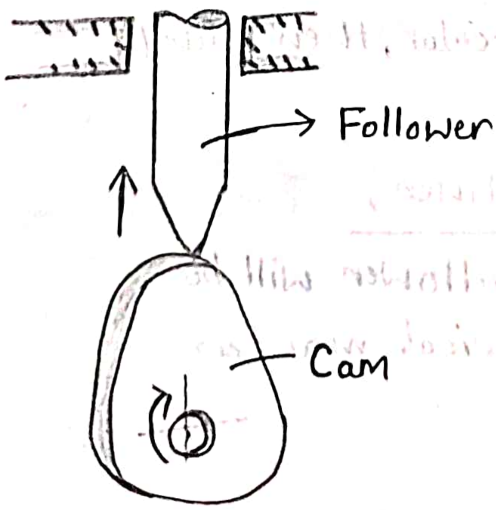
The cam may be rotating or reciprocating where as the follower may be rotating, reciprocating or oscillating.

The driver member is called Cam, the driven member is called Follower and there will be same arrangement of frame supports the cam and guide the follower.



Suppose that, we rotate the cam (Driver) in the clock-wise dirven member is called Follower (Driven) in the upward direction. The means, we can say that, the follower follows the cam.

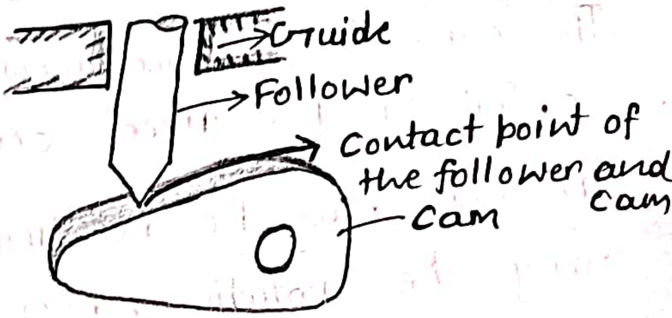
The guide is provided for the definite move only in upward and downward direction not in left and right direction.



Type of follower :

(i) According to the shape of the contact point of the follower;

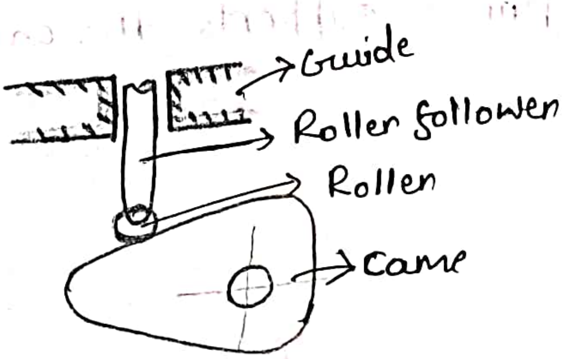
(a) Knife - Edge Follower ;



The end part of the follower which will contact the cam, will be in the shape of knife edge.

The ~~shape~~ sharp knife - edge will wear the surface of the cam. Also there will be side thrust in between the follower and guide.

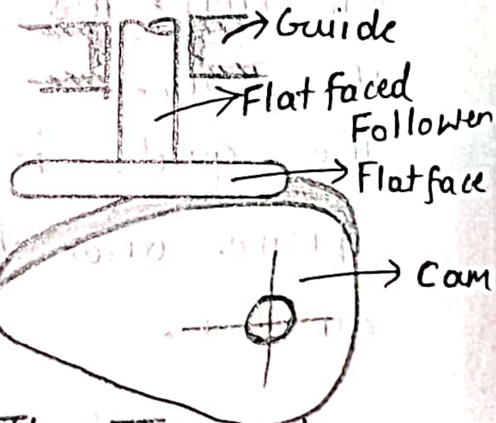
(b) Roller Follower ;



The end point of the follower contains a roller which is in the contact with cam.

Rate of wear between the roller and the cam is reduced. Also the side thrust in between guide and follower can be required. Roller follower is widely used in mechanical system.

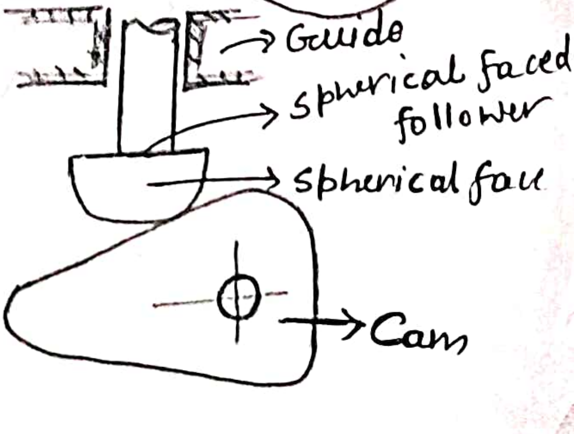
(c) Flat faced or Mushroom Follower;



The contact face of follower will be in the shape of flat manner.

If the flat face is circular, then called Mashroom follower.

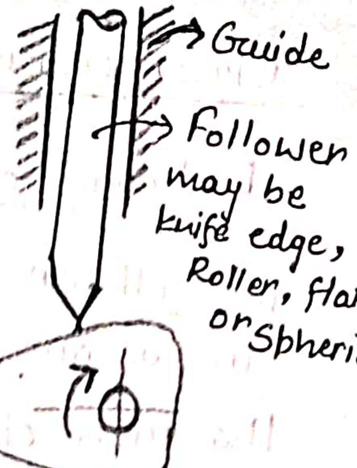
(d) Spherical Face follower;



The contact face of follower will be in the shape of spherical manner

(ii) According to the motion of the follower;

a) Reciprocating or Translating follower;



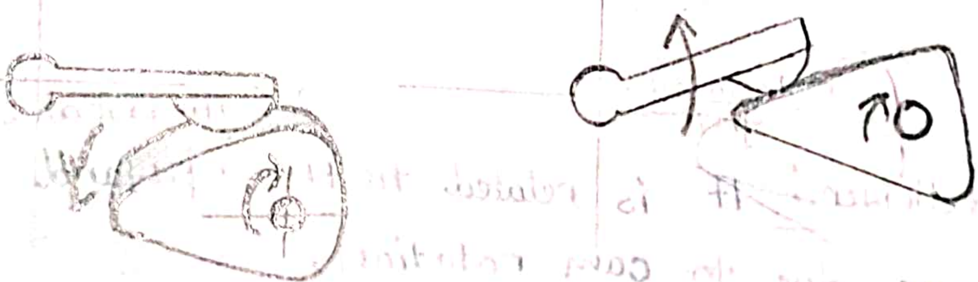
When the follower can only reciprocate or translate in the guide, then called reciprocating follower.

Due to the rotation of Cam (clockwise-direction), the follower can only ~~oscillate or rotate~~ reciprocates within the guide.

b) Oscillating or Rotating follower;

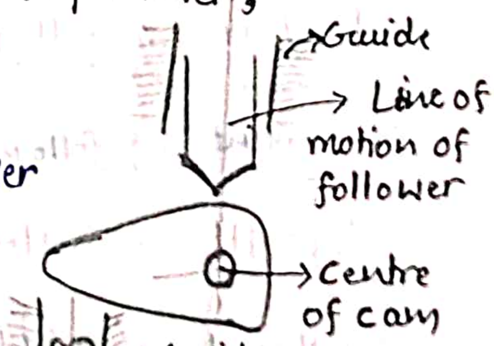
When the follower can only oscillate or rotate, then called oscillating follower.

With the rotation of can (clockwise - direction) the follower can easily oscillate about point 'O'.



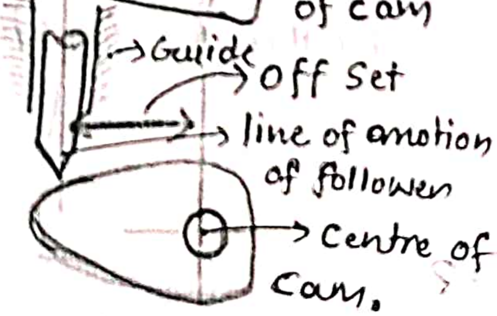
(iii) According to the path of motion of the follower;

a) Radial Follower



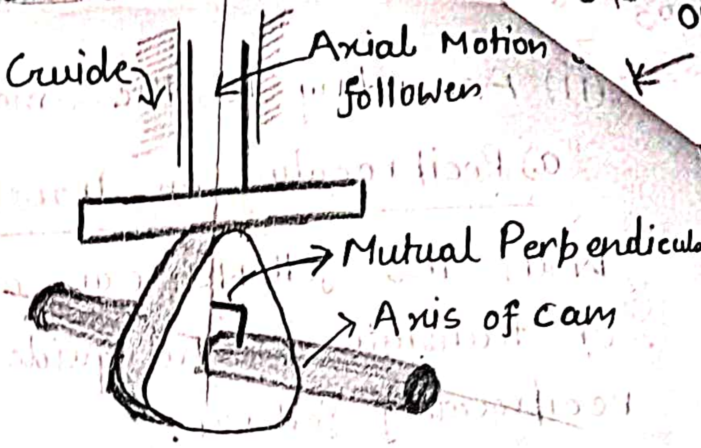
When the line of motion of the follower passes through the centre of cam, then called radial follower.

b) Off-set follower



When the line of motion of the follower is away from the centre of the cam, then called of set follower.

Classification of Cam ;



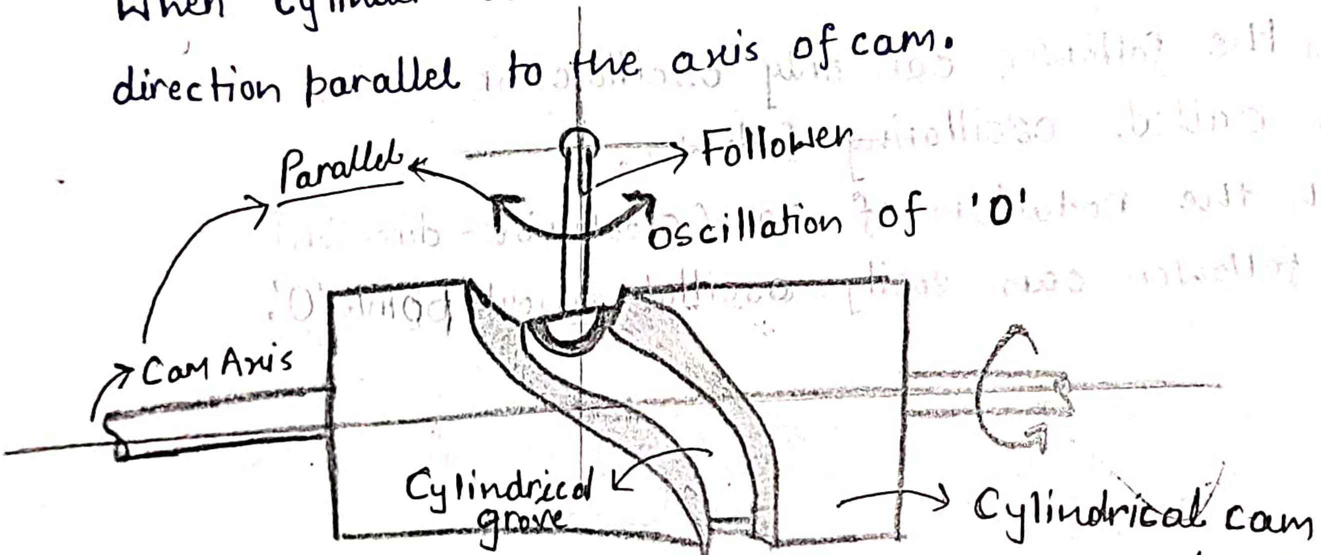
1. Radial or Disc Cam ;

The follower reciprocate or oscilates in a direction perpendicular to the axis of cam.

2. Cylindrical Cam ;

The follower reciprocates or oscilates in a direction parallel to the axis of cam.

When cylinder cam rotates the follower oscillates in a direction parallel to the axis of cam.



Rise of follower : It is related to the upward movement of follower due to cam rotation.

Return of follower : It is related to the downward movement of follower due to cam rotation.

Dwell of follower : In this position, the follower neither move upward and downward during the cam rotation.

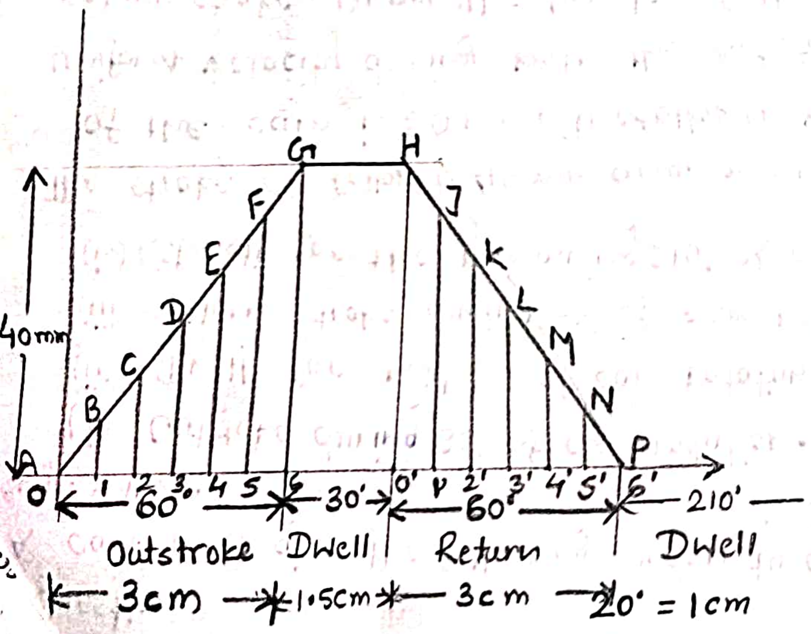
Question

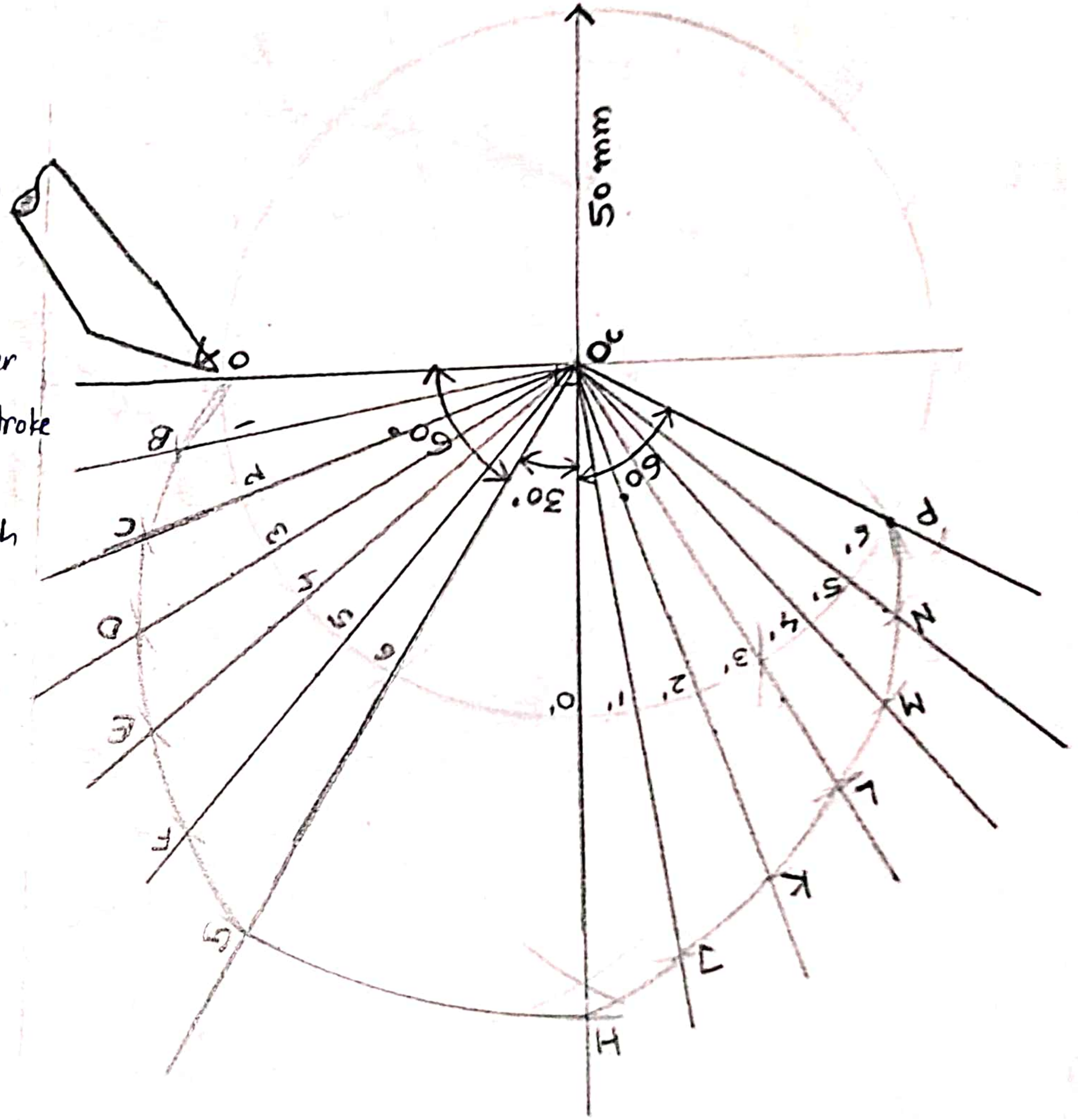
A cam is to give the following motion to a knife edge follower:

(i) Outstroke during 60° of cam rotation.
(ii) Dwell for the next 30° of cam rotation.
(iii) Return stroke during next 60° of cam rotation.
(iv) Dwell for the remaining 210° of cam rotation.

The stroke of the follower is 40 mm and the minimum radius of the cam is 50 mm. The follower moves with uniform velocity during both the outstroke and return stroke. Draw the profile of the cam when the axis of the follower passes through the axis of camshaft.

Displacement Diagram,





A

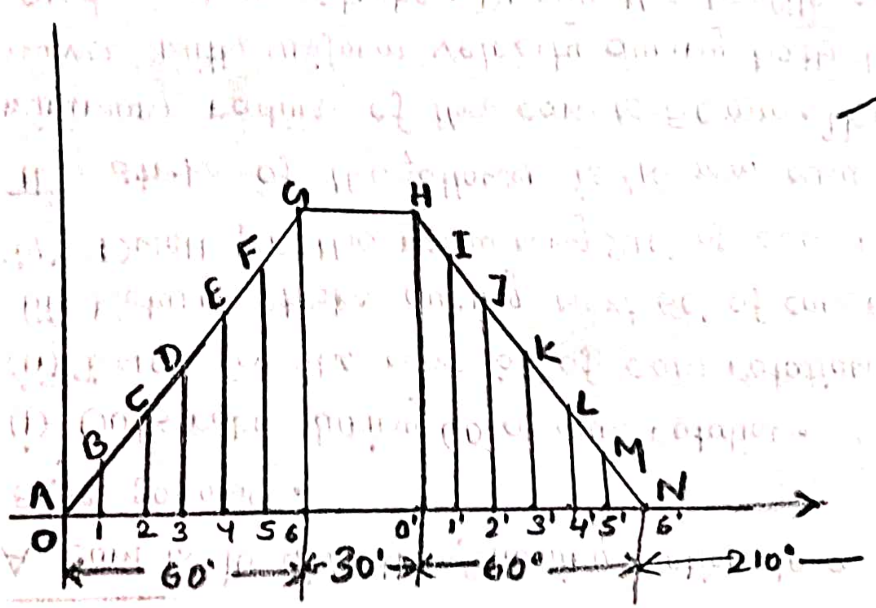
Question.

5) A cam is to give the following motion to a knife edge follower;

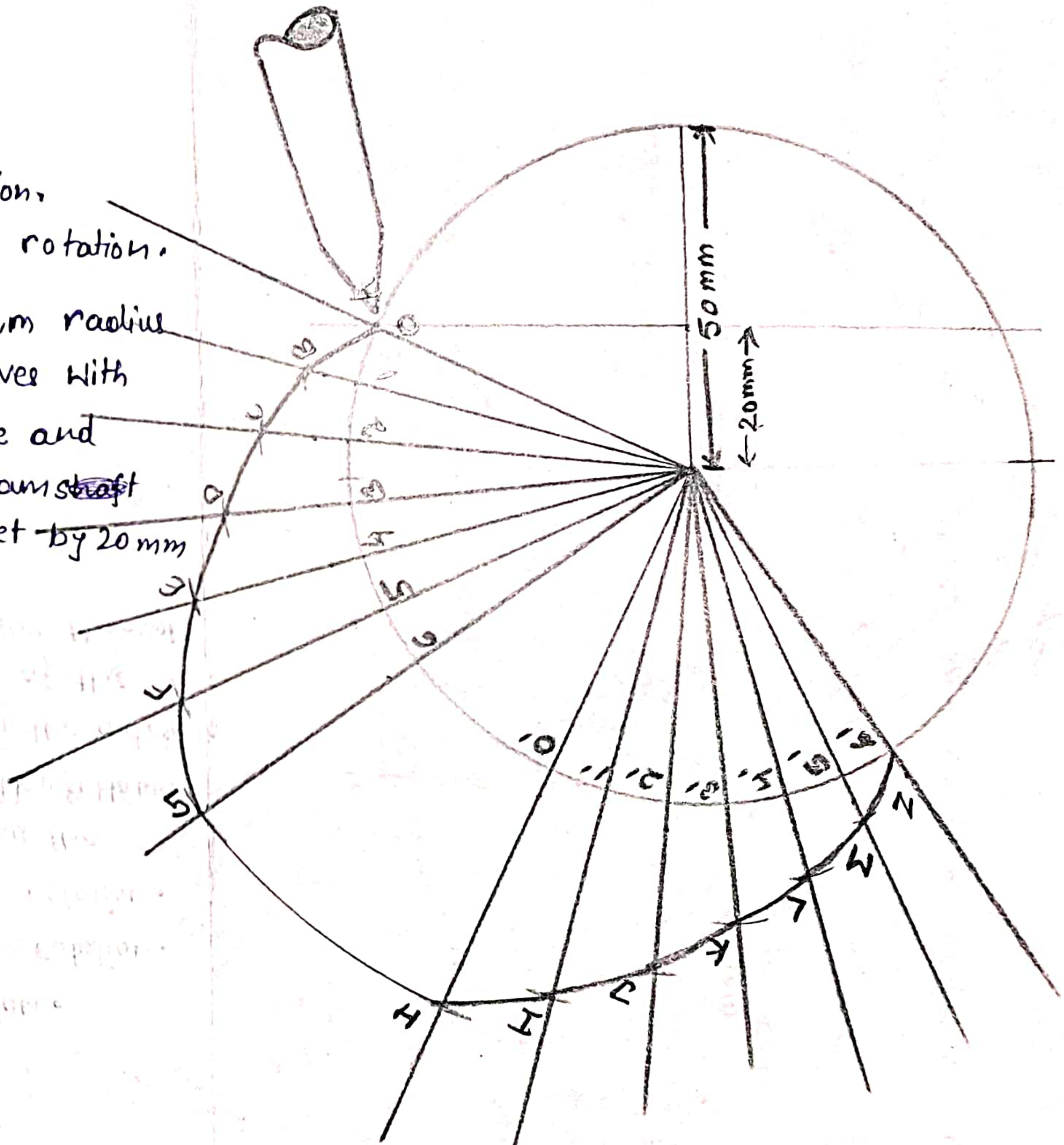
(i) Outstoke during 60° of cam rotation.
(ii) Dwell for Next 30' of cam rotation,
(iii) Return stroke during 60° of cam rotation,
(iv) Dwell for the remaining 210° of cam rotation.

The stroke of follower 40 mm and minimum radius of the cam is 50 mm. The follower moves with uniform velocity during both the outstroke and return stroke. Draw the profile of the camshaft when the axis of the follower is offset by 20 mm from the axis of the camshaft.

Displacement Diagram;



1cm = 20°



Question

A cam, with a minimum radius of 25 mm, is to be designed to give a roller follower.

(i) To rise the follower through 50 mm during 120° rotation of cam.
(ii) To keep the follower at rest through next 30°.
(iii) To lower the follower through next 60°
(iv) To keep the follower at rest through next 150°.

The diameter of roller is 20 mm.

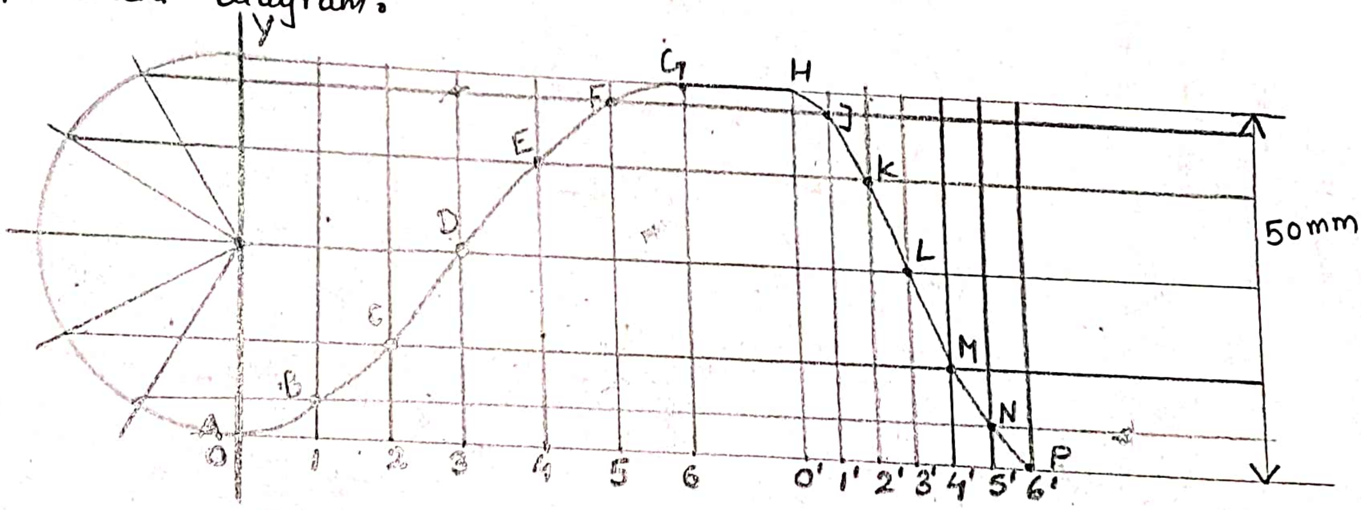
Draw the profile of cam when

a). The line of stroke of follower passes through the axis of camshaft.

b). The line of stroke is offset 15 mm from the axis of cam shaft. The displacement of valve, while being raised and lowered, is to take place with simple harmonic motion.

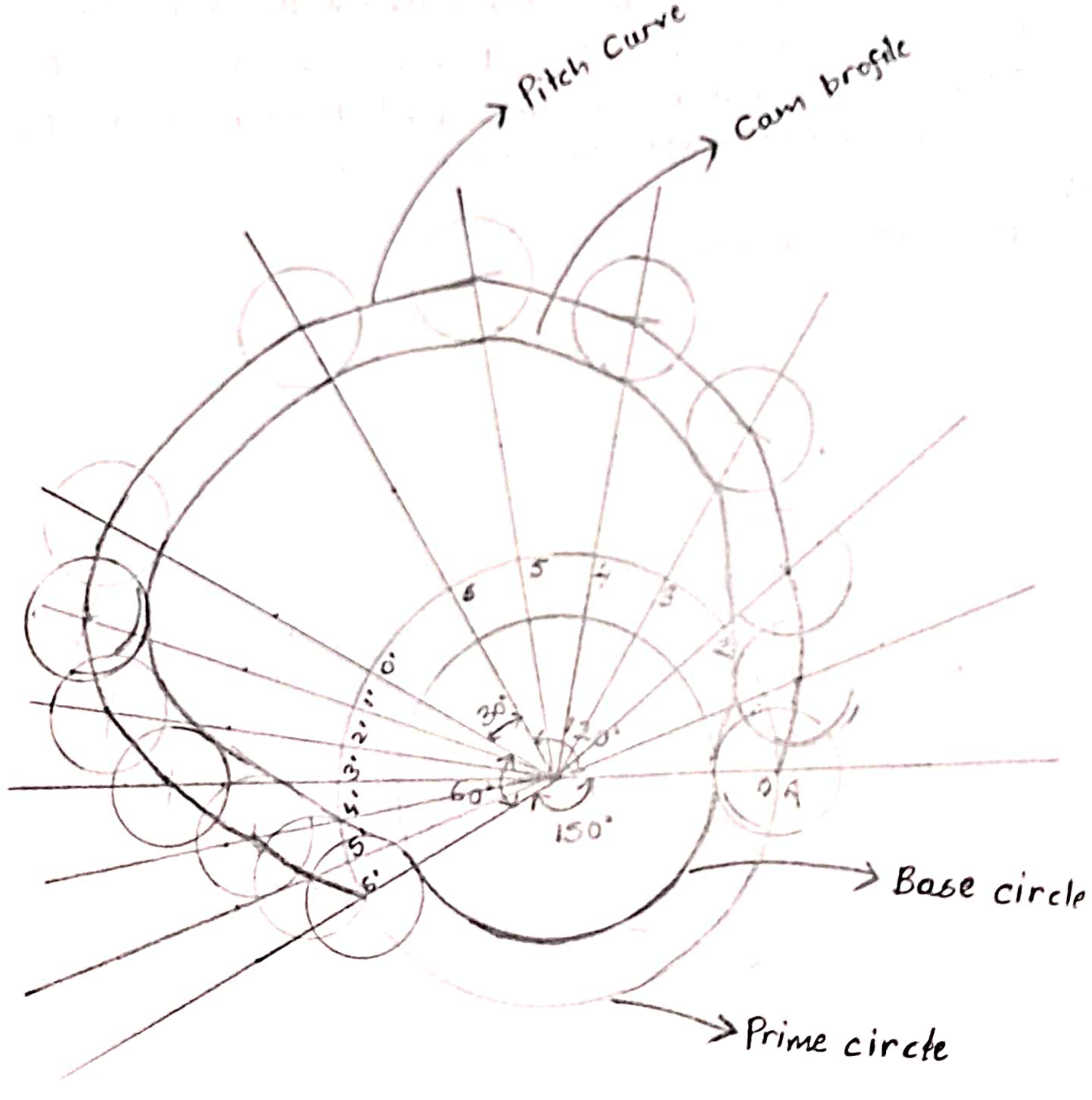
a).

Displacement diagram.

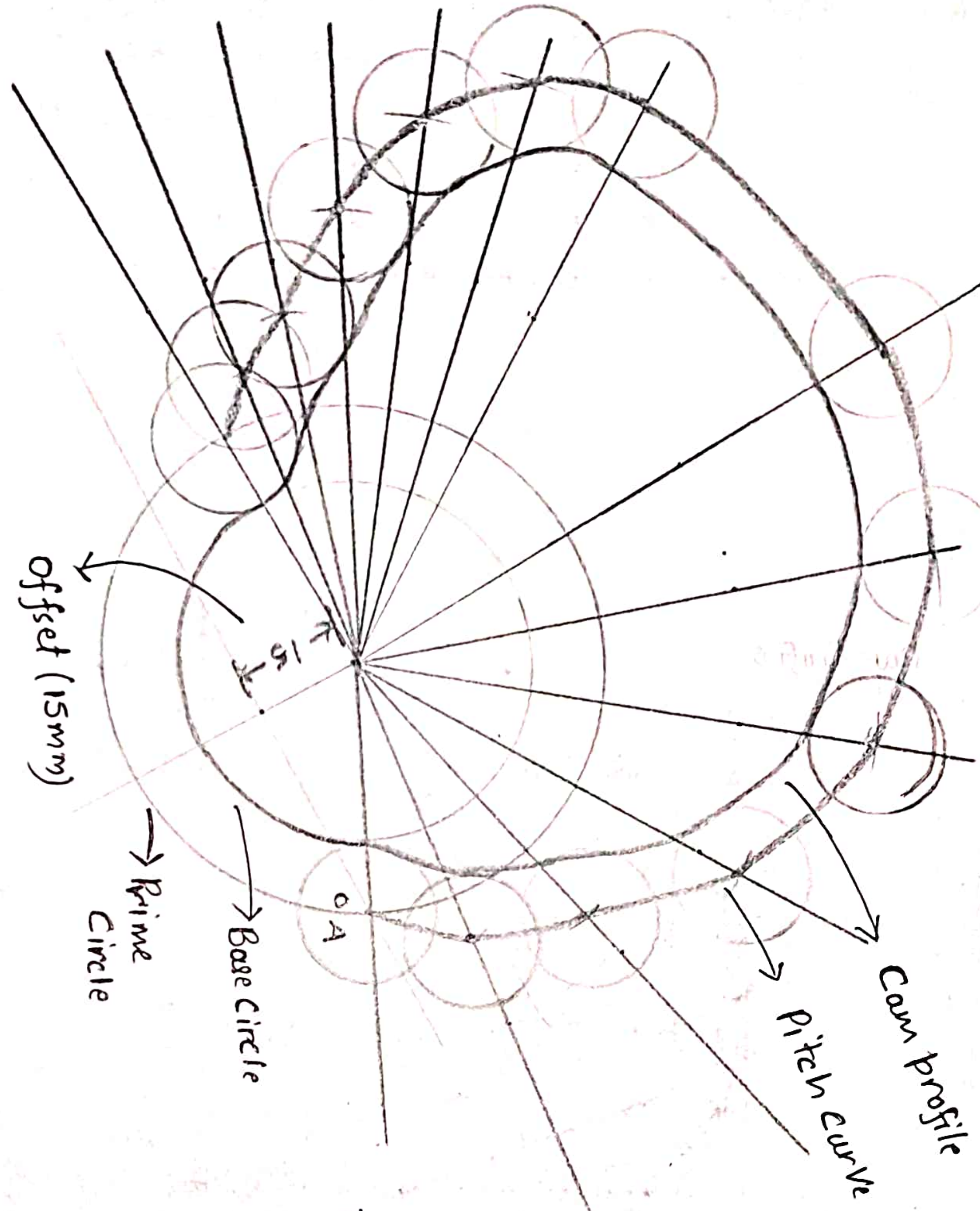


Prime Circle Radius = Base circle Radius + $\frac{\text{Roller Diameter}}{2}$

= 35 mm.



b).



Question

A cam with a miniamum radius of 40 mm, is to be designed to give a knife Edge follower. Stroke of the follower is 37.5 mm.

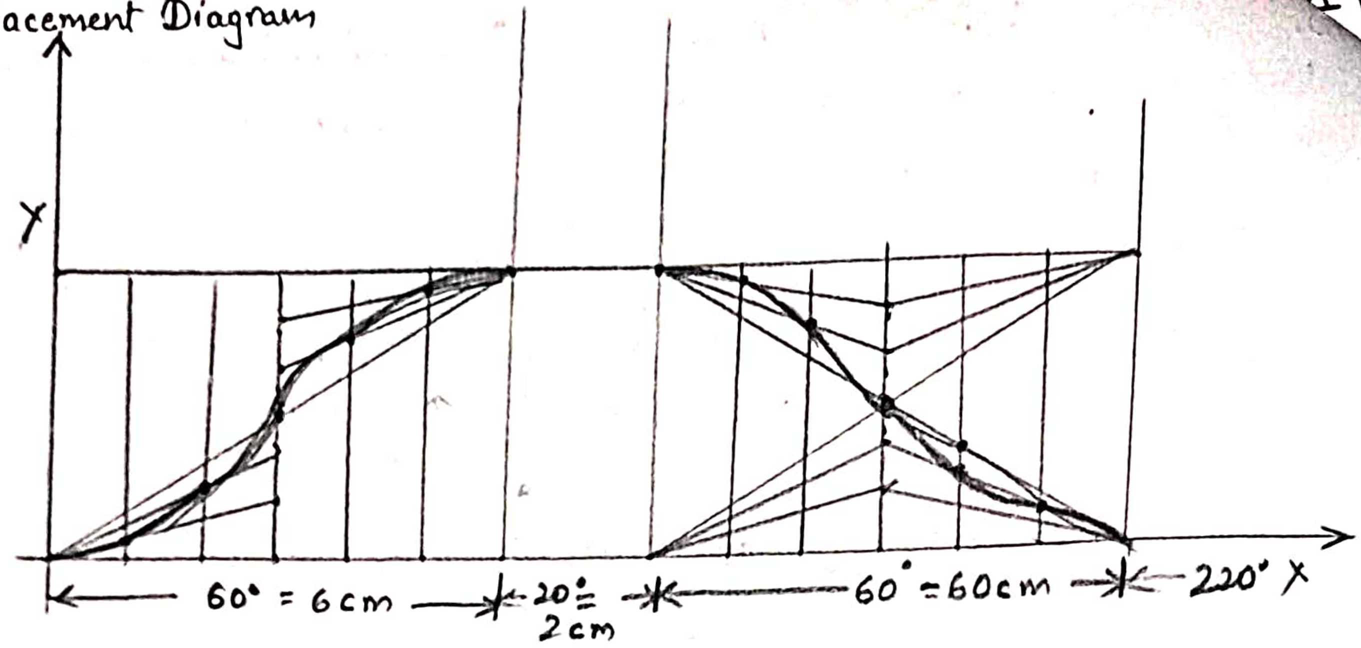
(i) Outstroke during 60° rotation of the ~~crank~~ cam.

(ii) Dwell through next 20°

(iii) Return stroke during next 60°.

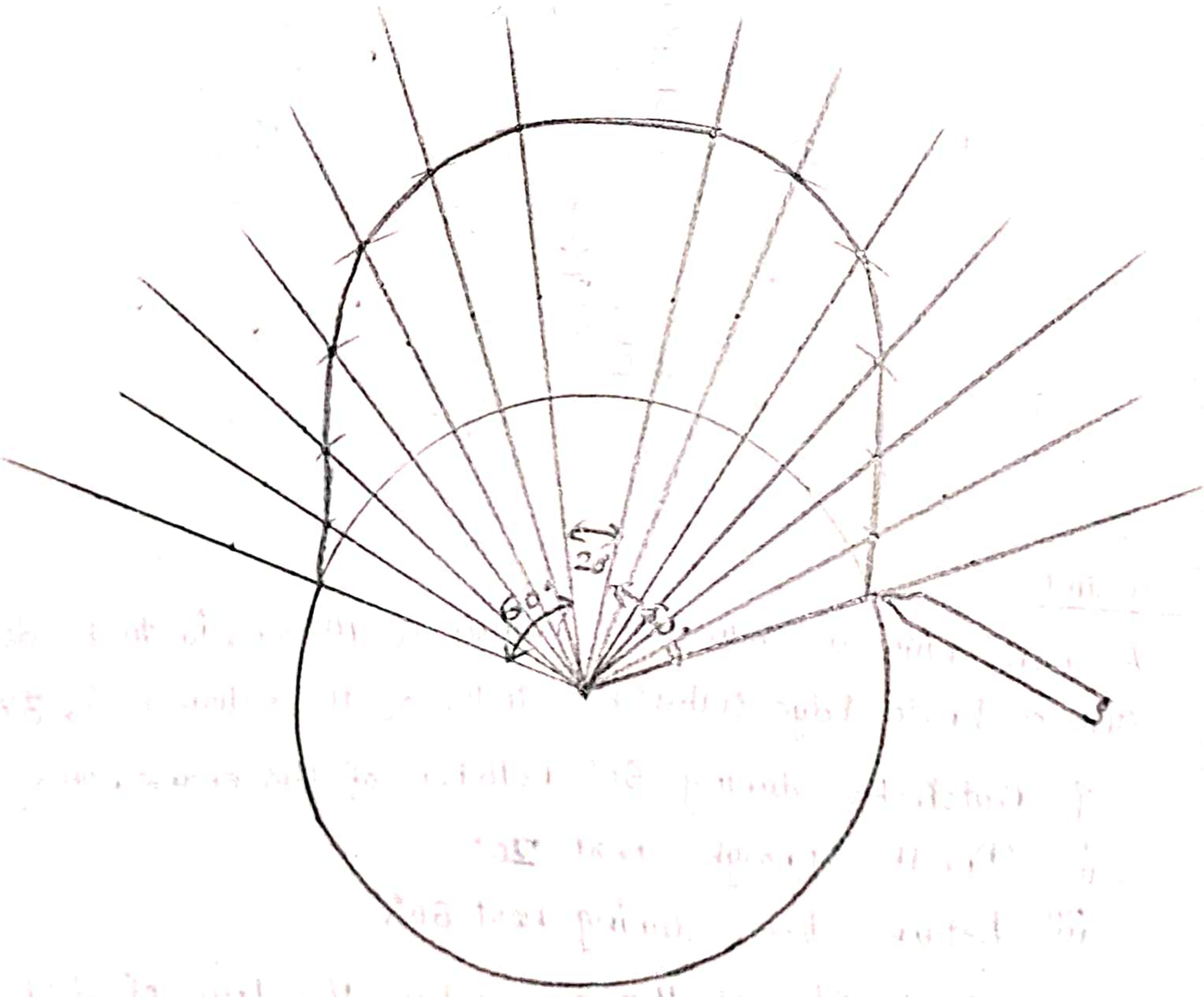
Draw the profile of the cam when the line of stroke of the follower passes through axis of cam shaft.

Displacement Diagram



1cm = 10°

Cam Profile



Analytical Method - Follower moves with S.H.M.

When follower moves with simple Harmonic Motion:-

S - stroke of Follower

θ_o and θ_R - Angular displacement of the cam during outstroke and return stroke in radians.

ω - Angular velocity of cam in radian/second.

1. Max^m. Velocity of the follower on the outstroke

$$\boxed{v_o = \frac{\pi \omega S}{2\theta_o}}$$

2. Max^m. Acceleration of the follower on outstroke

$$\boxed{a_o = \frac{\pi^2 \omega^2 S}{2\theta_o^2}}$$

3. Max^m. acceleration of the follower on Returnstroke

$$\boxed{a_R = \frac{\pi^2 \omega^2 S}{2\theta_R^2}}$$

4. Max^m. Velocity of the follower on Returnstroke

$$\boxed{v_R = \frac{\pi \omega S}{2\theta_R}}$$

Analytical Method - Follower moves with Uniform acceleration and uniform Retardation

Maximum Velocity of follower, During Outstoke

$$\boxed{v_o = \frac{2\omega S}{\theta_o}}$$

Maximum Velocity of follower During Returnstroke

$$\boxed{v_R = \frac{2\omega S}{\theta_R}}$$

Maximum Acceleration of follower During Outstroke,

$$\boxed{a_o = \frac{4 \omega^2 s}{\theta_o^2}}$$

Maximum Acceleration of follower, During Return stroke,

$$\boxed{a_R = \frac{4 \omega^2 s}{\theta_R^2}}$$

Question;

1. A cam is to be designed for a knife edge follower with the following data,
   1. Follower lift = 40 mm during 90° of cam rotation with simple harmonic motion.
   2. Dwell for the next 30°
   3. During the next 60° of cam rotation, the follower returns to its original position with simple harmonic motion.
   4. Dwel during the remaining 180°

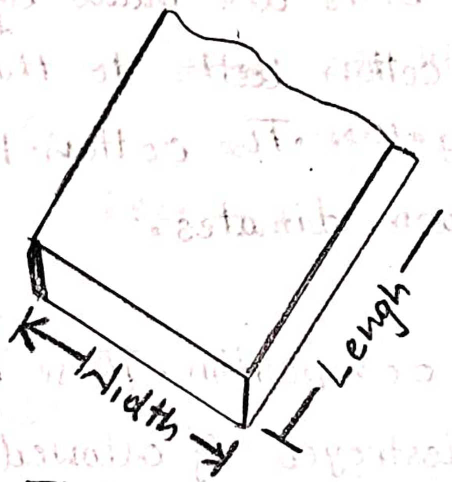
   The radius of the base circle of cam is 40 mm. Determine the maximum velocity and acceleration of the follower during its ascent and descent; if the cam rotates at 240 rpm.

2. A Cam is to be designed for a knife edge follower with the following data.
   1. Follower lift = 40 mm during 100° of cam rotation with uniform acceleration.
   2. Dwell for next 80°,
   3. During the next 90° of cam rotation, the follower returns to its original position with uniform acceleration.
   4. Dwell during the remaining 90°.

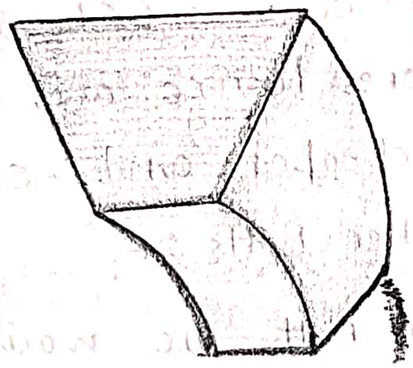
   The radius of the base circle of the cam is 50 mm. Determine the maximum velocity and acceleratiof, if the cam rotates at 900 rpm.

UNIT - 2 Power Transmition

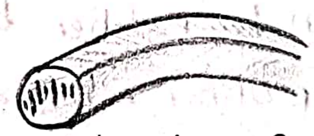
Belt Drive :- Belts are used to transmitting the power of one shaft to another shaft, The belt may be flat, V-type or circular type. The belts are running over the pulleys. The pulleys are mounted on the two shafts.



Flat Belt
(Rectangular cross section)



V - Belt
(Trapezoidal cross-section)



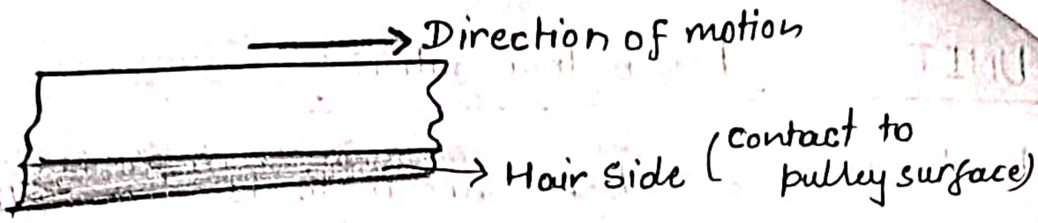
Circular Belt
(circular Crosssection)

Flat Belt over a Pulley.

Material Used For Belts :

The material used for belts and ropes must be strong, flexible and durable. It must have a high co-efficient of friction.

1. Leather Belts :- The hairside of the leather is smoother and harder than the ~~leather~~ flesh side, but the flesh side is stronger. The fiber on the Hair side are perpendicular to the surface, due to which the Hair side keeps a strong contact with the pulley surface. It provide great tensile strength.



The leather may be either Oak - tanned or Mineral salt tanned.

2. Cotton or Febric Belts :- Febric belts are made by flooding canvass (Strong cloth) or cotton ~~belts~~ to three or more layers and stitching together. The cotton belts are cheaper and suitable in warm dimates.

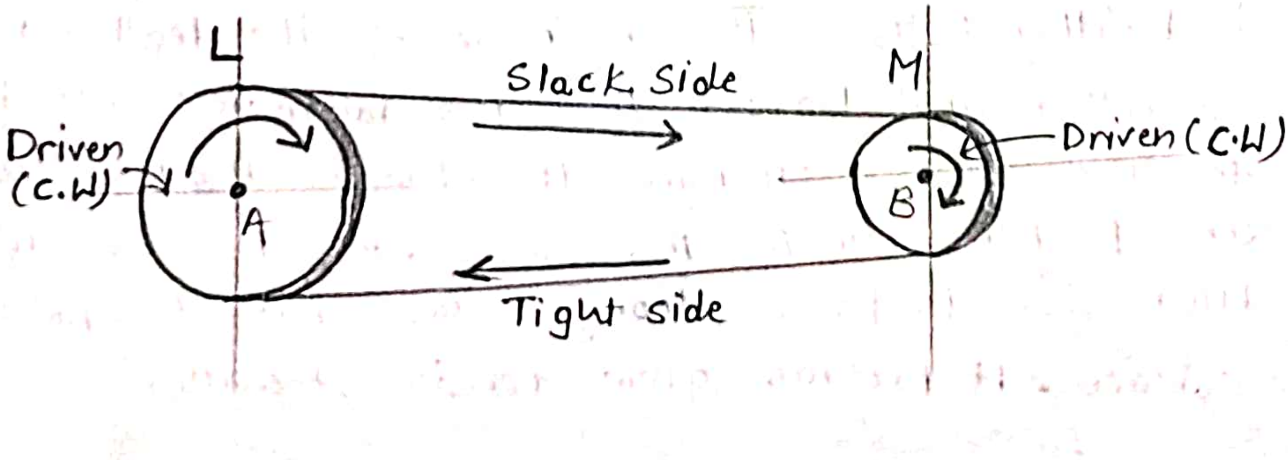
3. Rubber Belts :-

These belts are made by rubber composition. These belts are very flexible but are quickly destroyed if allowed to come into contact with heat, oil or greace.

4. Balata Bets : These belts are similar to rubber belts, but Balata gum is used in place of rubber. These belts are acid proof and water proof. These Balata belts should not be at temperature above 40°c because at this temperature the Balata begins to soften and become sticky.

Types of Flat Belt Drives

1. Open Belt drives :



Open belt provides the same direction of rotation of Driver and Driven pulleys.

The pulley A (Driver) pulls the belt from one side (lower end RQ) and delivers it to the another side (Upper end MN).

The tension in the lower end and will be greater than that of upper ends, ~~are calle~~ hence the lower end and upper called Tight slide and slack side respectively.

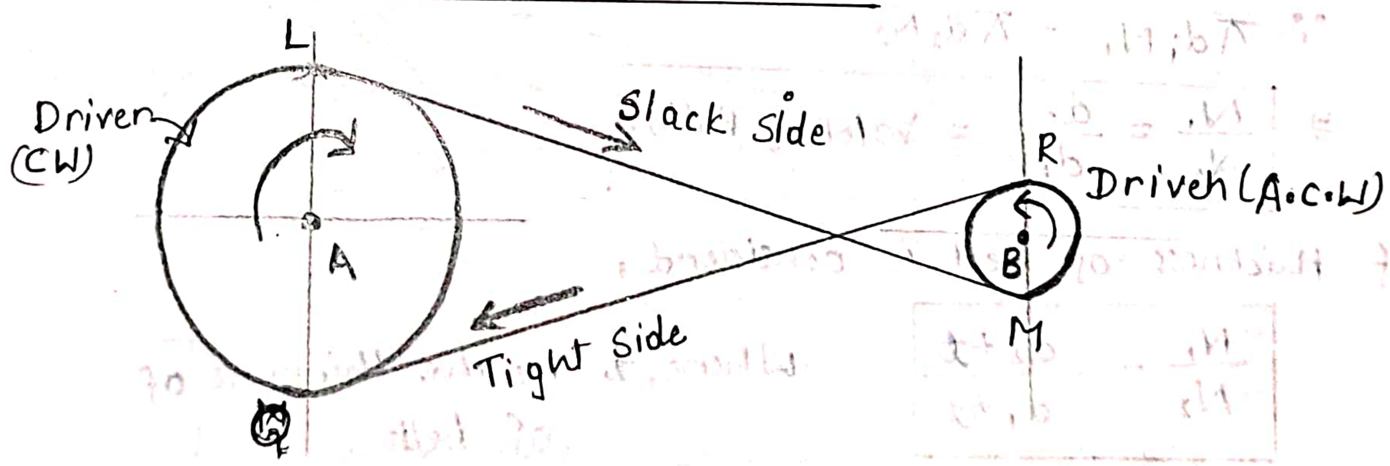
Note; The small pulley (B) will rotate more the Bigger pulley (A)

$$N_B > N_A$$

If the size of both pulleys are same, both pulleys are of same rotation.

$$N_A = N_B$$

2. Crossed or Twist Belt Drive :



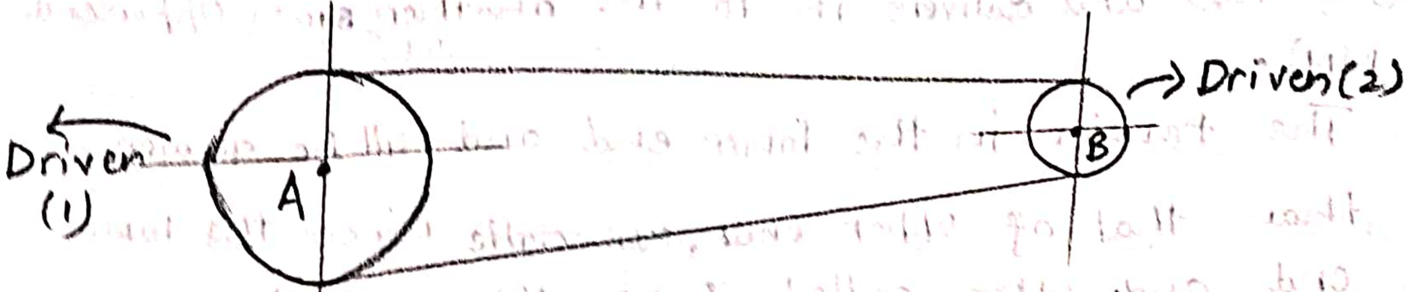
Cross Belt drives provides the opposite direction of rotation of Driver and driven.

QR → Tight Side

LM → Slack side

Velocity ratio of Belt Drive

Velocity ratio of Belt drive is defined as the ratio of rotation of Driver to rotation of Driven or follower.



d_1 = diameter of Driver

d_2 = diameter of Driven

N_1 = Speed of Driven

N_2 = Speed of Driven

Now, length of belt which passes through the driver in one minute $= \pi d_1 N_1$

the length of belt which passes through the driven in one minute $= \pi d_2 N_2$

$\because \pi d_1 N_1 = \pi d_2 N_2$

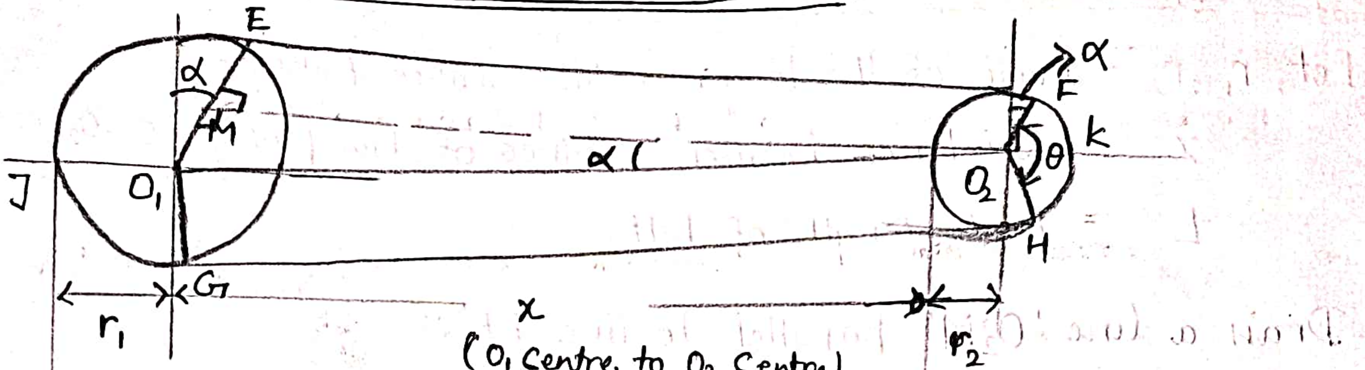
$$\Rightarrow \boxed{\frac{N_1}{N_2} = \frac{d_2}{d_1} = \text{Velocity Ratio.}}$$

If thickness of belt is considerd,

$$\boxed{\frac{N_1}{N_2} = \frac{d_2+t}{d_1+t}}$$

Where, t be the thickness of of belt

Angle of Lap or Angle of contact :-



r_1 = Radius of larger pulley
r_2 = Radius of smaller pully
x = Distance between centres of the two pelleys (O_1 & O_2)

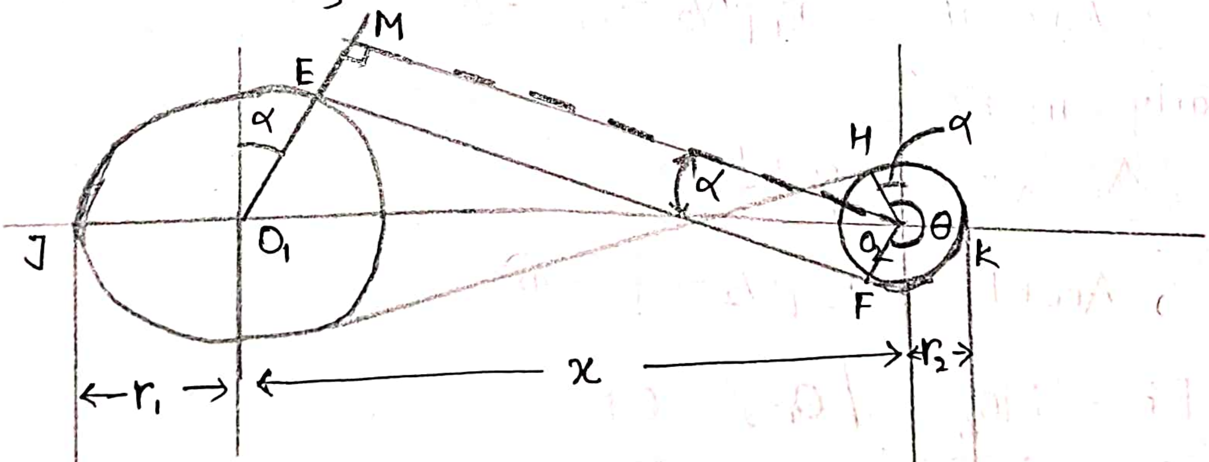
Now, in triangle O_1MO_2

$$\sin\alpha = \frac{O_1M}{O_1O_2} = \frac{O_1E - ME}{O_1O_2}$$

$$\boxed{\sin\alpha = \frac{r_1 - r_2}{x}}$$

$$\theta = (180° - 2\alpha) \cdot \text{ in degree}$$
$$= (180° - 2\alpha) \times \frac{\pi}{180°} \text{ in radian.}$$

In cross belt,



In triangle, O_1MO_2

$$\boxed{\sin\alpha = \frac{O_1M}{O_1O_2} = \frac{O_1E + EM}{O_1O_2} = \frac{r_1 + r_2}{x}}$$

$$\theta = (180° + 2\alpha), \text{ in degree}$$
$$= (180° + 2\alpha)\frac{\pi}{180}, \text{ in Radian}$$

Length of open Belt Drive

Let, r_1, r_2 = Radii of the larger and smaller pulleys.

x = Distance between centres of two pulleys (O_1, O_2)

L = Total length of Belt.

Draw a line 'O_2M' parallel to line 'FE',

Let, $\angle MO_2O_1 = \alpha$ rad.

Now, total Length of Belt,

$$L = \text{Arc } GJE + EF + \text{Arc } FKH + HG$$

$$= (\text{Arc } JE + EF + \text{Arc } FK) \times 2$$

$$\left[\begin{array}{lcl} \because 2\text{ Arc } JE & = & \text{Arc } GJE \\ EF & = & HG \\ \text{and } 2\text{ Arc } FK & = & \text{Arc } FKH \end{array}\right]$$

Now, in JO_1E,

$$\left(\frac{\pi}{2} + \alpha\right) = \frac{\text{Arc } JE}{r_1}$$

$$\Rightarrow \text{Arc } JE = r_1\left[\pi/2 + \alpha\right] \quad -(\text{I})$$

Similarly, in FO_2K,

$$\left(\frac{\pi}{2} - \alpha\right) = \frac{\text{Arc } FK}{r_2}$$

$$\Rightarrow \text{Arc } FK = r_2\left[\pi/2 - \alpha\right] \quad -(\text{II})$$

Now, $EF = MO_2 = \sqrt{O_1O_2^2 - O_1M^2}$

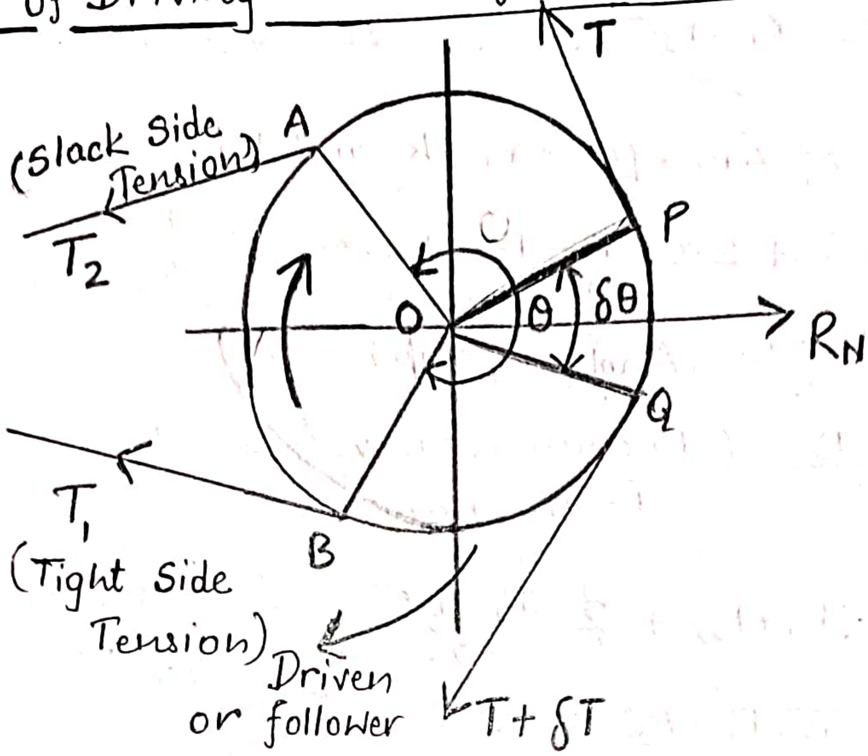
$$EF = \sqrt{x^2 - (O_1E - ME)^2}$$

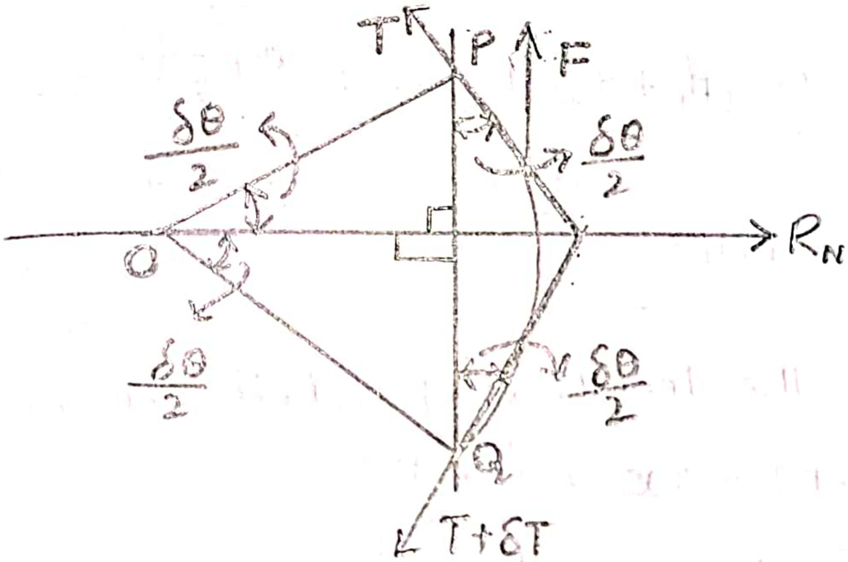
$$EF = \sqrt{x^2 - (r_1 - r_2)^2}$$

$$EF = \sqrt{x^2\left[1 - \frac{(r_1 - r_2)^2}{x^2}\right]} = x\sqrt{1 - \left(\frac{r_1 - r_2}{x}\right)^2}$$

$$EF = x\sqrt{1 - \left(\frac{r_1 - r_2}{x}\right)^2}$$

Ratio of Driving Tensions for Flat Belt drive ;





Let, T_1 = Tension in the belt on the tight side

T_2 = Tension in the belt on the slack side

θ = Angle of contact in radian (Angle made by arc AB)

Now, consider a small portion of the belt 'PQ' subtending an angle '$\delta\theta$' at the centre of the pulley.

The portion 'PQ' of the belt is equilibrium under forces:

1. Tension 'T' in the belt at 'P'.
2. Tension '$T+\delta T$' in the belt at 'Q'.
3. Normal Reaction 'R_N'.
4. Frictional force, "$F = \mu R_N$".

Expanding the expression of EF by Binomial expression;

$$EF = x - \frac{(r_1 - r_2)^2}{2x} -$$

Putting the value of Arc JE, Arc FK and EF

$$L = \pi(r_1 + r_2) + 2\alpha(r_1 - r_2) + 2x - \frac{(r_1 - r_2)^2}{x}$$

$\because$ $\sin\alpha = \frac{r_1 - r_2}{x}$ (Angle of lap)

$\alpha = \frac{r_1 - r_2}{x}$ (Because α is very small is in radian.

then,

$$L = \pi(r_1 + r_2) + \cancel{2}\frac{(r_1 - r_2)^2}{x} + 2x - \frac{(r_1 \cancel{-} r_2)^2}{\cancel{x}}$$

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x}$$

$$\text{or } L = \frac{\pi}{2}(d_1 + d_2) + 2x + \frac{(d_1 - d_2)^2}{4x}$$

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Length of cross Belt;

Proceed same as the, length of open belt drive:-

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x}$$

$$= \frac{\pi}{2}(d_1 + d_2) + 2x + \frac{(d_1 + d_2)^2}{4x}.$$

Now, equating all the Horizontal forces;

$$T\sin\frac{\delta\theta}{2} + (T+\delta T)\sin\frac{\delta\theta}{2} = R_N$$

$$\Rightarrow T\frac{\delta\theta}{2} + T\frac{\delta\theta}{2} + \delta T \cdot \frac{\delta\theta}{2} = R_N$$

$$\Rightarrow T\frac{\delta\theta}{2} + T\frac{\delta\theta}{2} = R_N$$

$$\Rightarrow T\delta\theta = R_N$$

Now, equating all the vertical forces;

$$T\cos\frac{\delta\theta}{2} - (T+\delta T)\cos\frac{\delta\theta}{2} + F = 0$$

$$\Rightarrow T - (T+\delta T) + F = 0$$

$$\Rightarrow T - T - \delta T + F = 0$$

$$\Rightarrow F = \delta T$$

$$\Rightarrow \mu R_N = \delta T$$

$$\Rightarrow R_N = \frac{\delta T}{\mu}.$$

$$\Rightarrow T\delta\theta = \frac{\delta T}{\mu}$$

$$\Rightarrow \frac{\delta T}{T} = \mu\,\delta\theta$$

Now, intrigrating the above expresion,

$$\Rightarrow \int_{T_2}^{T_1}\frac{\delta T}{T} = \int_0^{\theta}\mu\,\delta\theta$$

$$\Rightarrow \log_e\left(\frac{T_1}{T_2}\right) = \mu\theta \quad \text{— (A)}$$

$$\Rightarrow \frac{T_1}{T_2} = e^{\mu\theta} \quad \text{— (B)}$$

$$\Rightarrow 2.3\log(T_1/T_2) = \mu\theta \quad \text{— (C)}$$

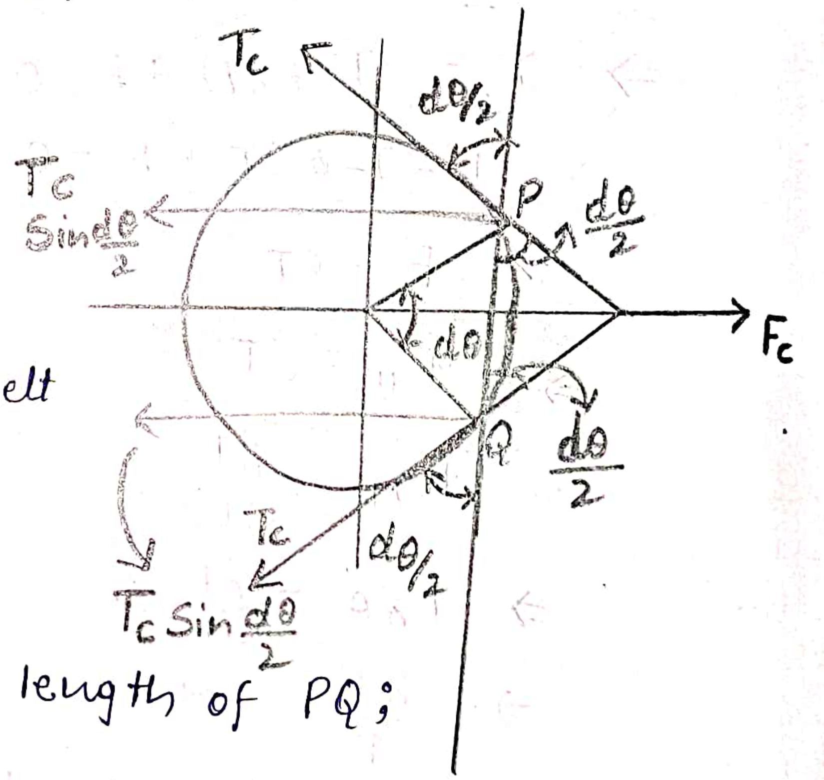
Centrifugal Tension :-

Due to the continuously running of pullys, there is some centrifugal force which increases the tension in tight side as well as slack side.

The tension developed by the centrifugal action is called centrifugal tension.

At lower belt speed (less than 10 m/s), the centrifugal tension can be neglected. But if the belt speed is high, then we should consider the effect of centrifugal tension.

Consider a small portion 'PQ' of the belt, which makes an angle of '$d\theta$' at the centre of pulley.



Let,

m = mass of length per unit length

v = linear velocity of belt

r = radius of pulley

T_c = Centrifugal Tension.

Now, firstly find out the length of PQ;

in section; OPQ,

$$d\theta = \frac{\widehat{PQ}}{r} \Rightarrow \widehat{PQ} = r\,d\theta$$

Mass of unit length of belt = m kg

Mass of length of $\widehat{PQ}$ = $r\,d\theta$, of belt = $m \cdot r\,d\theta$ kg.

$\therefore$ Centrifugal force acting on belt PQ $= (m r\,d\theta)\frac{v^2}{r}$

$$\therefore F_c = (m r\,d\theta)\frac{v^2}{r}.$$

Now, for the equilibrium position;

Resolve all the forces horizontally and equate with the centrifugal force.

$$T_c \sin\frac{d\theta}{2} + T_c \sin\frac{d\theta}{2} = F_c$$

$$\Rightarrow T_c \times \frac{d\theta}{2} + T_c \times \frac{d\theta}{2} = (m\, r\, d\theta)\frac{v^2}{r}$$

$$\Rightarrow T_c \times d\theta = m\, r\, d\theta \frac{v^2}{r}$$

$$\Rightarrow \boxed{T_c = m v^2}$$

1. If centrifugal Tension (T_c) is considered, then tension in tight side will be $(T_1 + T_c)$, and tension in slack side will be $(T_2 + T_c)$.

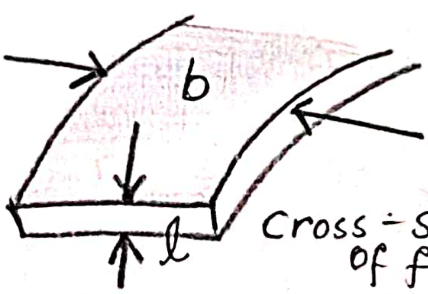
2. Power Transmitted,

$$P = [(T_1 + T_c) - (T_2 + T_c)]\, v$$

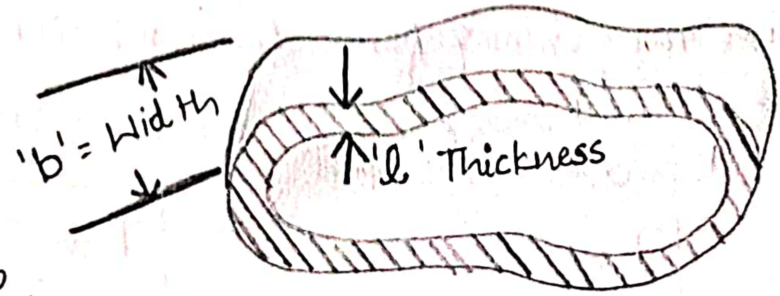
$$\therefore P = [T_1 - T_2]\, v$$

Maximum Tension or Stress on belt :-

In this topic, we will obserb the maximum tension produced in the belt will no fail. In other words, we have to find the safe stress produced in the belt due to tension in the belt. ~~due to the~~



Cross-section of flat belt



Suppose max. induced stress in Belt $= \sigma_m$

$$\sigma_m = \frac{\text{max. Tension}}{\text{Surface area over which the Tension acts}}$$

$$\sigma_m = \frac{T_m}{b \times t}$$ Where, $T_m = T_1 + T_c$.

$$\boxed{T_m = \sigma_m (b \times t)}$$

Condition for Maximum Power Transmission :-

Power transmission,

$$P = (T_1 - T_2)\, v \quad -(1)$$

Also, $\frac{T_1}{T_2} = e^{\mu\theta} \Rightarrow T_2 = T_1 / e^{\mu\theta}$

$$\therefore P = \left(T_1 - \frac{T_1}{e^{\mu\theta}}\right) v$$

$$= T_1 \left(1 - \frac{1}{e^{\mu\theta}}\right) v$$

$$P = T_1 (Z)\, v$$ Where, $Z = 1 - \frac{1}{e^{\mu\theta}}$

Now, we know that,

$$T_m = T_1 + T_c \Rightarrow T_1 = T_m - T_c$$

$$P = (T_m - T_c) Z v = (T_m - m v^2) Z v$$

$$P = Z (v T_m - m v^3)$$

↳ mass of belt per unit length

For the max. power transmission;

$$\frac{dP}{dv} = 0 \Rightarrow Z(T_m - m 3 v^2) = 0$$

$$\boxed{T_m = 3 v^2 m} \Rightarrow \boxed{T_m = 3 T_c} \text{ or} \Rightarrow \boxed{v = \sqrt{\frac{T_m}{3m}}}.$$

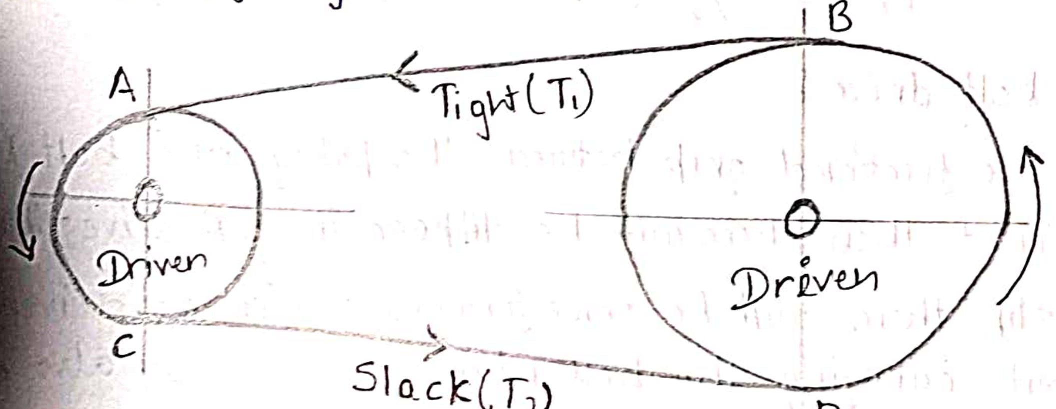
- Transmission will be maximum, if centrifugal Tension (T_c) become one third (1/3) of maximum tension (T_m).

- Transimission of power will be maximum, if velocity of belt (v) becomes $\sqrt{T_m/3m}$.

Initial Tension in the Belt :-

When a belt fitted to a pair of stationary pulley then the induced Tension in belt is called initial Tension (T_0).

Now, When the pulleys are rotated, then there will be induction of Tight side (T_1) and slack side (T_2) Tensions,



When the pulleys are stationary, the tension in portion 'AB' and 'CD' are 'T_0'.

But when the pulleys are rotated, the tension in 'AB' is increased while the tension in 'CD' is decreased. If the material of Belt is perfectly elastic and length of belt remains unchanged then, the increament in Tension in 'AB' will be same as the decrement in Tension in 'CD'.

Let increment or decrement in tension is 'δT'

$\therefore$ $T_1 = T_0 + \delta T$

$T_2 = T_0 - \delta T$

$\therefore$ $\boxed{T_0 = \frac{T_1 + T_2}{2}}$

If centrifugal tension is considered;

$$T_0 = \frac{(T_1 + T_c) + (T_2 + T_c)}{2}$$

$$T_0 = \frac{1}{2}(T_1 + T_2 + 2T_c).$$

Slip in belt drive :

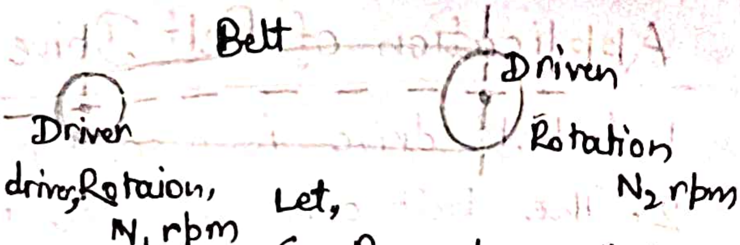
When the frictional grip between the pulley and belt becomes insufficient then there will be slippage in belt drive. Due to the slip there will be some forward motion of driver motion without carrying the belt when it This may also cause some forward motion of the belt without carrying the driven pulley with it.

The phenomena is called slip of the belt and is generally expressed as a percentage.

Actually slip in the belt drive is a phenomenan of the relative motion between belt and pulley. Due to insufficient grip of friction between pulley & belt, there are some relative motion is called slip of the belt.

Now,

Velocity of belt passing over driver,



$$v = \frac{\pi d_1 N_1}{60} - \frac{\pi d_1 N_1}{60}\left[\frac{S_1}{100}\right].$$

$$v = \frac{\pi d_1 N_1}{60}\left[1 - \frac{S_1}{100}\right].$$

Loss of v due to slip

Let,

S_1 = Percentage slip b/w Driver & belt

S_2 = Percentage slip b/w Driven & belt

Now Velocity of belt passing over the driven,

$$\frac{\pi d_2 N_2}{60} = v - v\frac{S_2}{100} = v\left(1 - \frac{S_2}{100}\right)$$

$$\Rightarrow \frac{\pi d_2 N_2}{60} = \frac{\pi d_1 N_1}{60}\left(1 - \frac{S_1}{100}\right)\left(1 - \frac{S_2}{100}\right)$$

$$\Rightarrow \frac{N_2}{N_1} = \frac{d_1}{d_2}\left(1 - \frac{S_2}{100} - \frac{S_1}{100} + \frac{S_1 S_2}{10^4}\right)$$

$$\therefore \boxed{\frac{N_2}{N_1} = \frac{d_1}{d_2}\left(1 - \frac{S_1 + S_2}{100}\right) = \frac{d_1}{d_2}\left(1 - \frac{S}{100}\right)}$$

Where, $S = S_1 + S_2$.

If thickness of belt is consider,

$$\boxed{\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}\left(1 - \frac{S}{100}\right)}$$

Crip in the belt drive,

When the belt passes from slack side to tight side, a certain portion of the belt extends and it contracts again when the belt passes from tight sight to slack side. Due to these changes in length, there is relative motion between the belt and the pulley surfaces. This relative motion is called creep

$$\boxed{\frac{N_2}{N_1} = \frac{d_1}{d_2} \times \frac{E + \sqrt{\sigma_2}}{E + \sqrt{\sigma_1}}}$$

When, σ_1 or σ_2 = Stress in belt on tight & slack side

E = Young's Modulus.

Application of Belt Drive;

1. A belt drive is used for transferring the power.
2. The belt drive is used in mill industry.
3. The belt drive is used in conveyor.

Application of Belt Drive;

1. It can be easily installed and removed.
2. The price of belt is not too high.
3. Simple in construction.
4. Low maintaince cost.
5. No additiond lubrication needed.
6. Power consumption is quite low.

Disadvantages of Belt drive;

1. Loss of power due to slip and creep.
2. Cannot be used for a very short distance.
3. Speed is limited to longer life.
4. No possibility of longer life.
5. Velocity ratio does not remain constant.
6. The operation temperature is just limited withis -35 to 85 degree C. As soon as it exceeds the temperatu then it cause extreme wear and tear.

Properties of Belt Drive;

- The belt should be flexible.
- It should be reliable and durable.
- The material should be capable of withstanding high tensile stress.
- Higher temperature should be resist.
- The weight per unit length should be less.
- Higher coefficient of friction within the belt and pulley.
- It should have an excellent resistance to wear fatigue

Robe Drive

Robe drive are widely used where a large amount of bower is to be transmitted, from one pulley to another.

Frictional grib in case of rope drives is more than the V-drive.

Types of rope drive; There are two types of rope drive. Namely; 1. Fibre Rope
2. Wire Robe.

Fibre Robe The fibre rope operates wellgood, when the pulleys are about 60 m apart.

Fibre robe are made of fibrous material such as hemp (A plant), manila (fibre of philibbine plant) and cottom. Hemp and Manila fibres are rough, hence there fibres are not very flexible and have poor mechanical properties. However, the hemp fibre are less strength as comprare to manila fibres.

Sheave for the fibre ropes;

The fibre ropes are usually circular in cross-section.

Wire Robe The wire robe operates when the pulley are upto 150 m apart. They widly used in cranes, conveyors, elevators, suspention bridge etc.

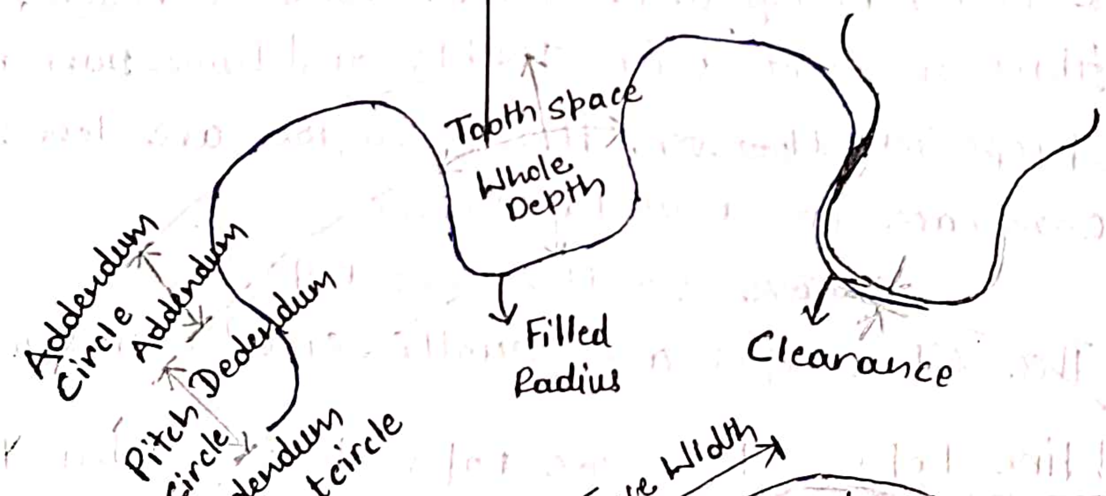
Limitation of steel Robe:

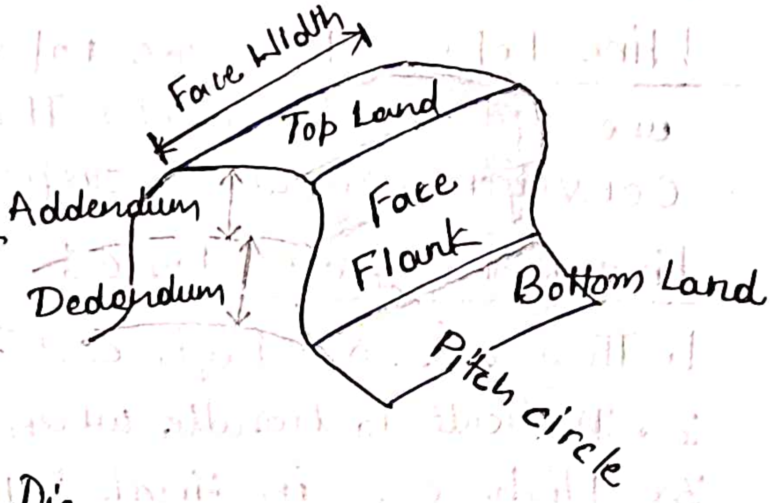
1. They are of high cost
2. Difficult to handle when it is fabricated.
3. High cost in final polishing and finishing.
4. They need highly inspection.
5. They cannot be repair.
6. Steel core wire should never be used at temp. above 400°F or below -40°F.

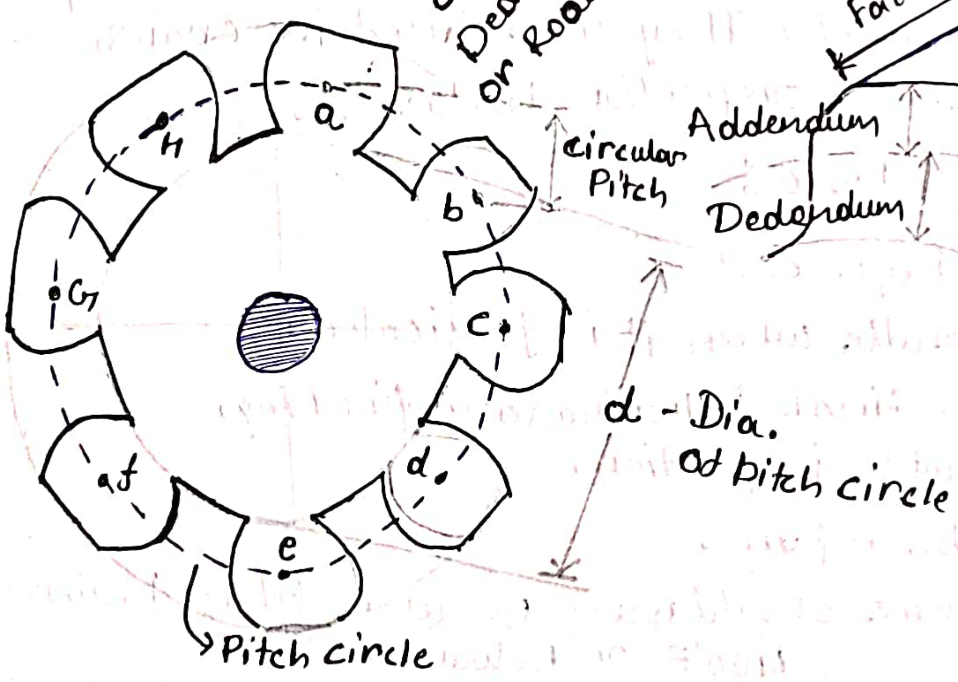
Advantages of Rope drives :

Fibre Drive	Wire Rope
1. They give smooth steady and quiest service.	1. They are more durable.
2. The strong and flexible	2. They are not fail suddenly.
3. They can run in any direction	3. They can withstand shock loads.
4. They are of low cost and economic	4. They are of high efficiency.
5. They are lighter in weight.	

Gear Nomenclature or Gear terminology ;







- **Addendum:** It is a radial distance from top Land to the pitch circle.
- **Dedendum:** It is the radial distance from bottom Land to pitch circle.
- **Whole Depth:** It is the sum of Addendum & Dedendum. It is called whole depth.
- **Addendum Circle:** It is a circle drawn from the top of the teeth.
- **Dedendum Circle:** It is a circle which is drawn through the bottom of the teeth. It is also called root circle.
- **Clearance:** When two gears are moting, then the clearance is define as the radial distance between top of a tooth of a gear and bottom of a tooth of another moting gear.
- **Tooth Thickness:** It is the width of a tooth measured along the pitch circle.
- **Tooth Space:** It is the width of space between the adjacent teeth measured along the pitch circle.
- **Face of the tooth:** It is the surface of tooth above the tooth surface.
- **Flank of tooth:** It is the surface of tooth below the tooth surface.
- **Top Land:** It is the top surface of the tooth.
- **Fillet Radius:** It is the radius that conects the root circle to the profile of the tooth.
- **Pitch Circle:** It is the imaginarry circle on the gear about which it may be supposed to roll without slipping with pitch circle another gear.
- **Circular Pitch:** It is the distance measured along the circumferance of the pitch circle from a point on one tooth to the corresponding point on the adjument tooth. It is denoted by 'pc'. $\boxed{p_c = \pi d / T}$

Diametral Pitch: It is the ratio of the number of teeth of the pitch circle diameter.

$$\boxed{p_d = \frac{T}{d} = \frac{\pi}{p_c}}$$

Module: It is ratio of pitch circle diameter (in mm) to the number of teeth.

$$\boxed{m = \frac{d}{T}}$$

Gear Ratio: It is the ratio of no. of teeth on bigger gear to the number of teeth on the smaller gear.

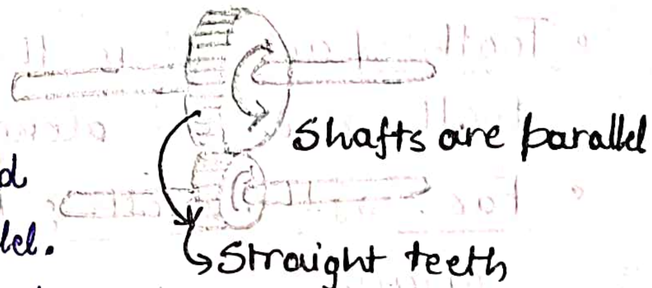
$$\boxed{\text{Gear Ratio} = T/t}$$

Classification of Gears:—

1. According to position of the axes of shaft.

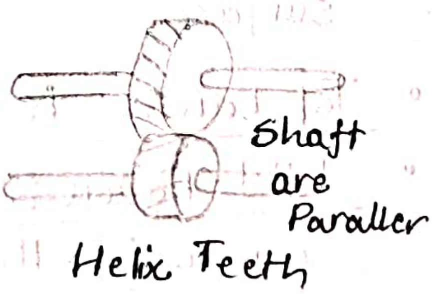
a) Parallel Shaft

(i) Spur Gear:

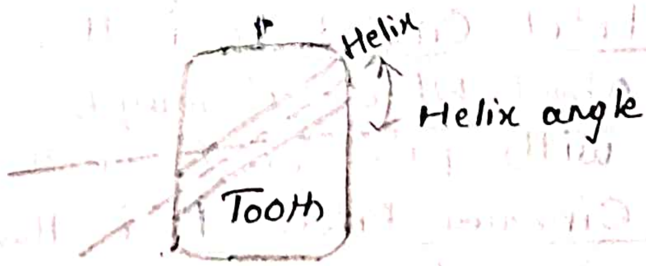


Both the shaft are parallel and both gears surface are co-parallel. The teeth of gear are straight and parallel to axis of gear.

(ii) Helical Gear:



The teeth are inclined or curved and helix in shape. Two mating gears have some helix angle, but have teeth of opposite hands.

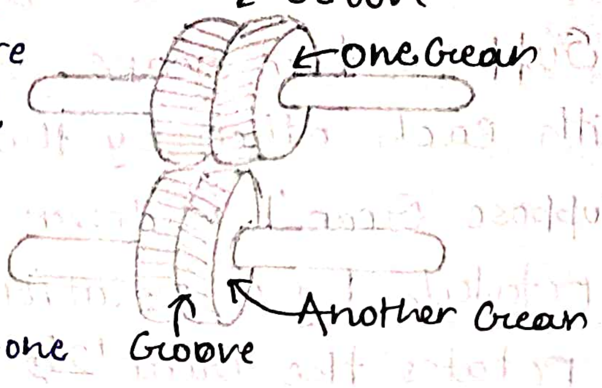


(iii) Double Helical or Herringbone Gears :-

There are two helical gears are combined to construct one duble helical gear.

The teeth of the two rows are seperated by a groove used for tool run out.

If there will be no any grove than called Herringbone Gears.

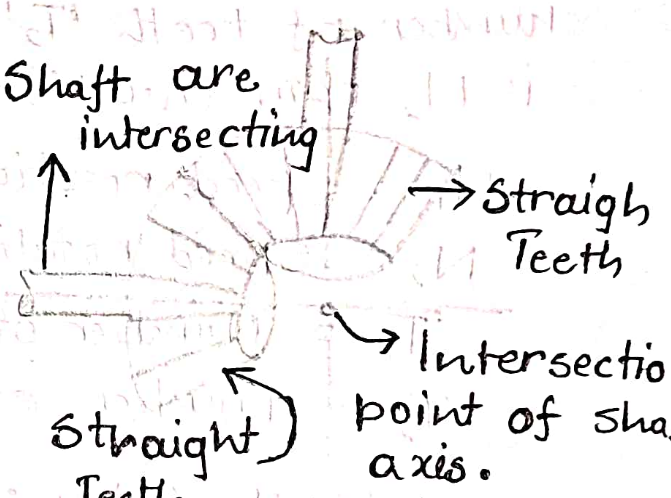


(b) Intersecting Shafts:

(i) Straight Bevel Gears:—

Bevel gear is a part of cone, on which staight teeth are construted.

(ii) Spiral Bevel Gears:—



The interse

2. According to the pheripheral velocity of gears;

a) Low Velocity Gears: If the velocity is less than 2 m/s.

b) Medium Velocity Gears: If the velocity is betweer 3 m/s and 15 m/s.

c) High Velocity Gears: If the velocity is greater than 15 m/s.

3. According to types of gear;

a) External Gear: The teeth are constructed on the external portion of disc.

b) Internal Gear: The teeth are constructed on the internal portion of dics.

c) Rock & Pinion: Rock is a type of gear which have infinite radius.

Velocity Ratio and Train Value :

Velocity ratio is defined as the ratio of rotation of driver to the rotation of driven.

Suppose two gears, Gear 1 and Gear 2 and connected with each other by the means of teeth as shown below. Suppose Gear '1' is driver having number of teeth 'T_1' is rotated by a rotation of N_1 rpm (CW). The Gear '1' rotates the Gear '2', where the Gear '2' consist number of teeth 'T_2' and the rotation of Gear '2' is N_2 rpm (A.C.W.).

N_1 = Speed rotation of gear 1 in rpm
N_2 = Speed rotation of gear 2 in rpm
T_1 = Number of teeth on Gear 1
T_2 = Number of teeth on Gear 2

$$\text{Velocity Ratio} = \frac{N_1}{N_2} = \frac{T_2}{T_1}$$ (Also called speed ratio)

Train value is defined as the ratio of rotation of driven to the rotation of driver.

ie Train value is reciprocal of the Velocity ratio.

$$\therefore \text{Train Value} = \frac{N_2}{N_1} = \frac{T_1}{T_2} = \frac{1}{\text{Velocity Ratio}}.$$

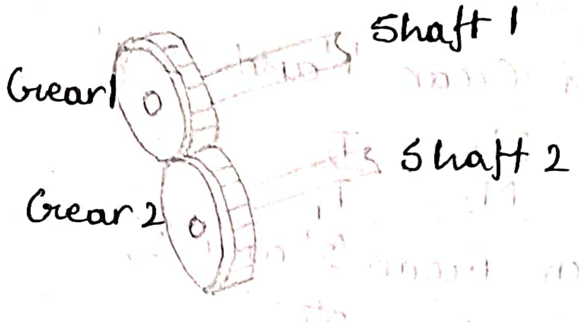
Noted that we will find the velocity ratio of all the gear trains.

Gear Train :

For the power transmission from one shaft to another, two or more than two gears are engaged with each othed (directly or indirectly). Such type of combination is called Gear train.

Types of gear train;

1. Simple gear Train
2. Compound gear Train
3. Reverted gear Train
4. Epicyclic Gear Train

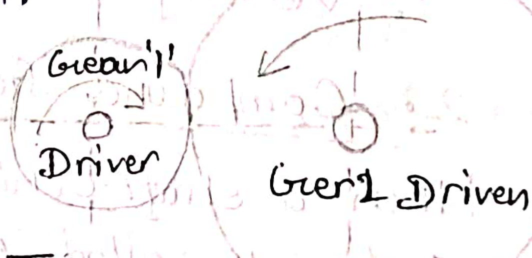


1. Simple Gear Train :

When there is only one gear on each shaft, then the System is called simple gear train.

Suppose there are two gears, '1' and '2' are connected. If driver rotates in clockwise direction then driver will rotate in anti-clockwise direction.

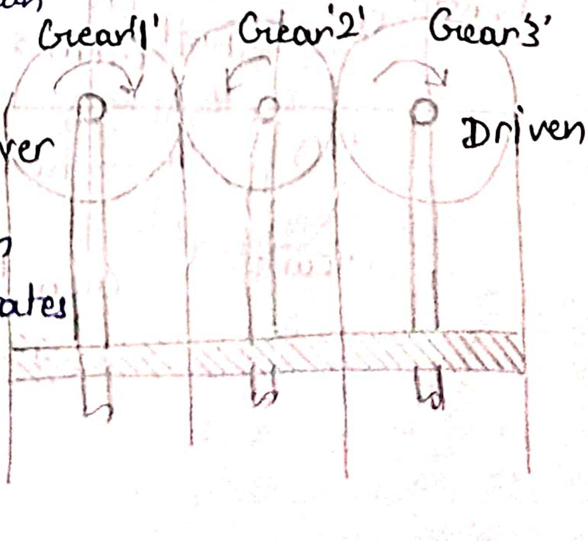
Also, Velocity Ratio $= \frac{N_1}{N_2} = \frac{T_2}{T_1}$



Noted that each shaft consist one gear.

T_1, N_1 & T_2, N_2 are teeth and speed of gear'1' & gear'2'.

Now, let these are three gears fitted in seperated shaft as shown in figure,



Noted that gear'1' rotates in C.W direction, Gear '2' rotates in A.C.W. and finally Gear 3 rotates in C.W. direction.

We can say that;

① If no. of gear are even (2,4,6) then the rotation of of driver and driven are opposite, but it no. of gear are odd (1,3,5) the the rotation of driver & driven are same.

Also,
In Gear '1' and '2',

$$\frac{N_1}{N_2} = \frac{T_2}{T_1}$$

then Gear '2' and '3',

$$\frac{N_2}{N_3} = \frac{T_3}{T_2}$$

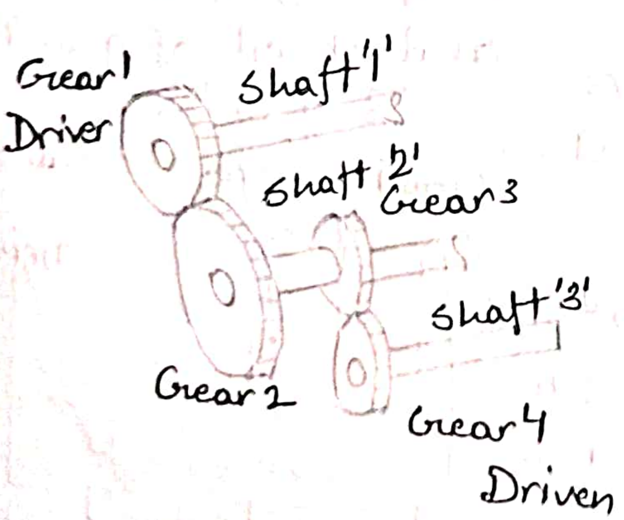
$$\frac{N_1}{\cancel{N_2}} \times \frac{\cancel{N_2}}{N_3} = \frac{\cancel{T_2}}{T_1} \times \frac{T_3}{\cancel{T_2}}$$

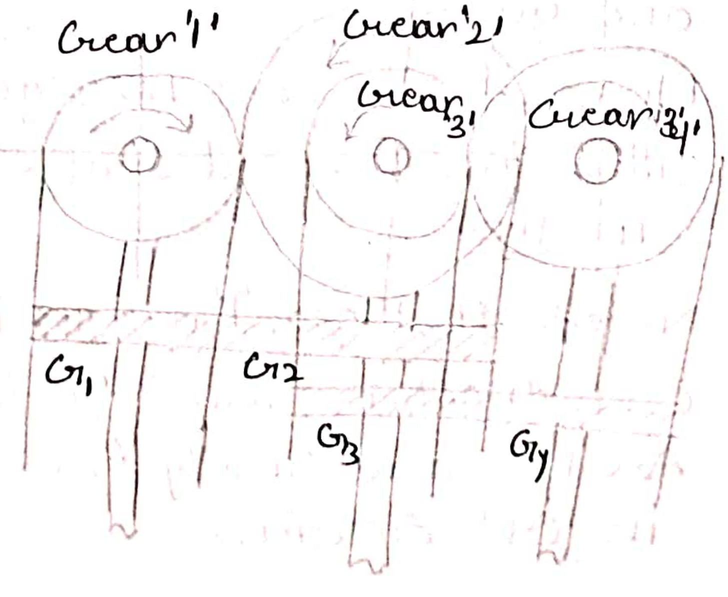
$$\Rightarrow \frac{N_1}{N_3} = \frac{T_3}{T_2}$$

$\because$ Velocity Ratio = $\frac{\text{Driver Speed}}{\text{Driven speed}} = \frac{\text{Driven teeth}}{\text{Driver Speed}}$

2. Compound Gear Train

When a shaft contain more than one gear, then this system is called compound gear train.





Now, for gear '1' and '2'

$$\frac{N_1}{N_2} = \frac{T_2}{T_1}$$

for gear '3' and '4'

$$\frac{N_3}{N_4} = \frac{T_4}{T_3}$$

$$\frac{N_1}{\cancel{N_2}} \times \frac{\cancel{N_3}}{N_4} = \frac{T_4}{T_3} \times \frac{T_2}{T_1} \quad \because \underline{N_2 = N_3}$$

$$\Rightarrow \frac{N_1}{N_4} = \frac{T_4 \times T_2}{T_3 \times T_1}$$

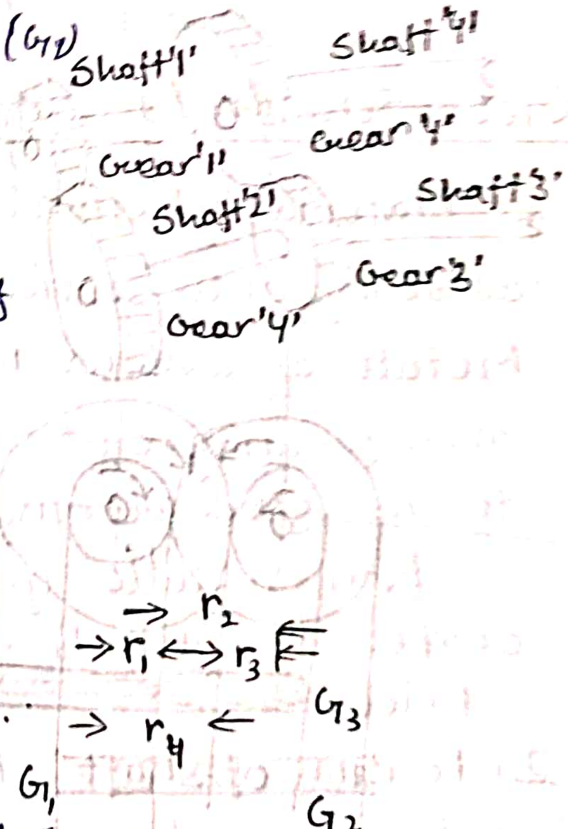
$$\because \text{Velocity ratio} \Rightarrow \frac{N_1}{N_4} = \frac{T_2 \times T_4}{T_1 \times T_3}.$$

3. Reverted Gear Train

When axis of first gear and last gear are co-axial, then the system is called reverted gear train.

Noted that Driver (G_1) and Driven (G_4) Having same rotation.

$$r_1 + r_2 = r_3 + r_4$$



The circular part of module of all the gear is assumed to be same, hence teeth of all the gear is directly perpotional to raddi.

$$T_1 + T_2 = T_3 + T_4.$$

Now for Gear 1 'and' 2'

$$\frac{N_1}{N_2} = \frac{T_2}{T_1}$$

for gear '3' and '4'

$$\frac{N_3}{N_4} = \frac{T_4}{T_3}$$

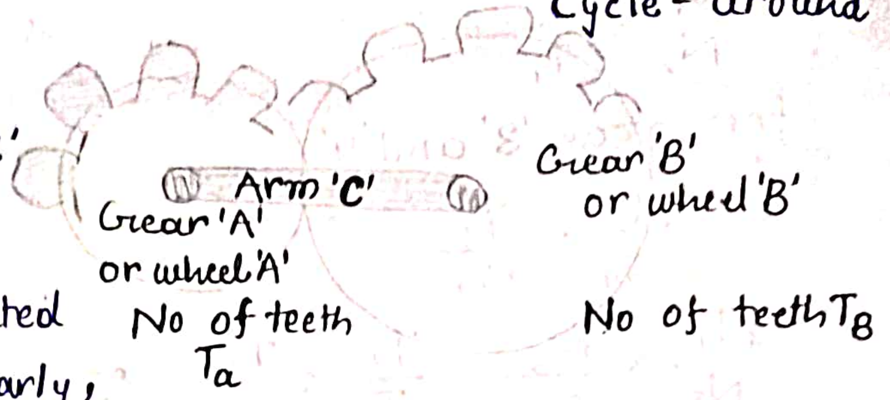
$$\Rightarrow \frac{N_1}{\cancel{N_2}} \times \frac{\cancel{N_3}}{N_4} = \frac{T_2 \times T_4}{T_1 \times T_3}$$

$$\because \text{Velocity ratio} = \frac{N_1}{N_4} = \frac{T_2 \times T_4}{T_1 \times T_3}.$$

4. Epicycle Gear Train / Planetary Gear :- Epi - Upon
Cycle - around

CASE I;

Suppose gear 'B' is fix and gear 'A' and Arm 'C' free to move.



Let the arm 'c' rotated in clockwise, then clearly,
the gear 'A' will rotate upon and around gear B, such motion is called Epicyclic gear train.

CASE II;

Suppose the arc is fixed and gear A ~~as~~ is rotated in CW, then this system is simple gear train.

Due to rotation of gear 'A' and 'B'

$$V.R = \frac{N_A}{N_B} = \frac{T_B}{T_A}$$

Gear Selection for differet application;

1. Material Selection: It depends upon the strength, durability and service condition like wear noice etc. Metals as well as non-metals are used for gear manufacturing. Cast iron is widely used for gear due to its good wearing properties.

   Non-metallc gears such as wood gears, compessed, synthetic gears are used for reducing the noice.

2. Position of shaft: If the motion transfer is required ~~such as wood gears~~ the two parallel shafts then spur gears are used. But if the shaft are non-parallel the bevel gear is used.

3. Mounting of shaft on Gears: Mounting of shaft is related to the assembly of shaft with gear bore with a spacial key.

Method for Lubrication:

1. Grease Lubrication: Grease lubrication can be used for industrial gear system which may be open or closed. The greas must have the correct viscosity properties. Noted that, grease has no any colling effect.

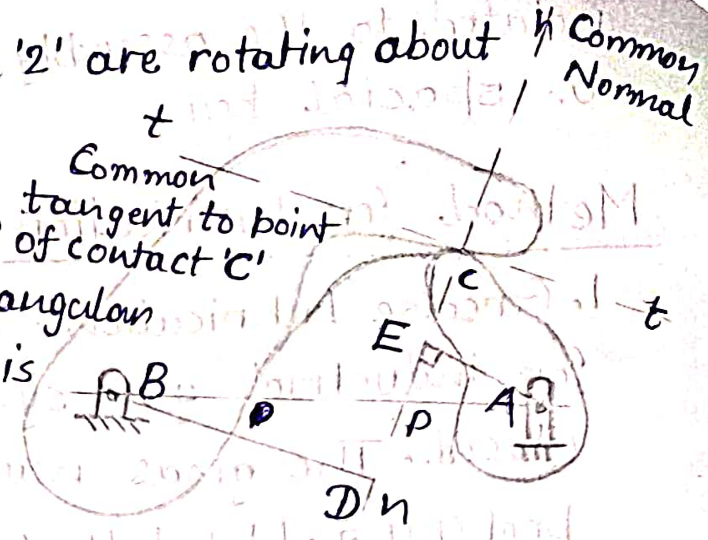
2. Splash Lubrication: The lower gear 1, is immersed in oil or lubricant (both gears can be immersed). When the gear are rotated, the oil is automatically transported to all the moving parts and lubrication points. The level of oil should be always checked and mointered.

3. Spray lubrication: Spray lubrication methods are are used in high speed industrial gears. In order to achieve the high speed industrial gear. In order to achieve the high speed spray of oil lubricant, a nozzles are also used for proper spraying.

4. Forced - Circulation Lubrication: In this method, the oil lubricant is provided at the contact point of teeth of gears, by the mans of force. Inorder to get the forces circulation, an oil pump is used.

Law of gearing or condition for constant velocity ratio;

Let the two curved bodies '1' and '2' are rotating about the centres 'A' and 'B'. The two bodies are in contact at point 'C'. The body '1' is rotating C.W with angular velocity 'ω_1' and the body '2' is rotating A.C.W. with angular velocity 'ω_2'.



t-t is common tangent at point of contact 'C'.
n-n is Common normal at point of contact 'C'.

Draw perpendicular lines 'BD' and 'AE' on the common normal.

P' is a fixed point (Pitch point) obtained from the intersection of 'BA' and common normar.

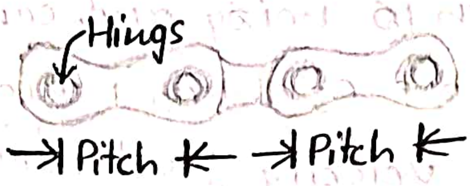
By the law of Gearing; $\boxed{\dfrac{\omega_1}{\omega_2} = \dfrac{BP}{AP}}$

ie. The common normal at point of contact devides the line joining the centre of rotation in the inverse ratio of the angular velocity. In other words, the common normal must passes through a fixed point 'P' (Pitch point) on the line joining of centre of rotation in such a way that there will be constant velocity ratio.

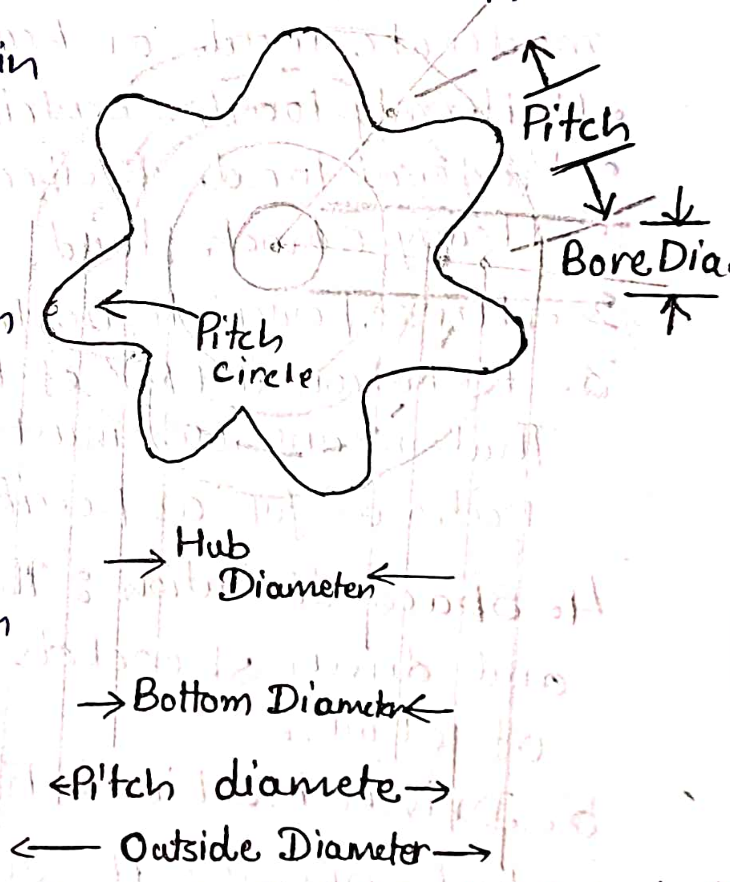
Chain Drive :

Before understanding chain drive, Let us first understand the chain. A chain can be defined as a sepies of likks connected by pin Joint.

Pitch of chain is the distance between hinge centre of an endless chain of a link and corresponding hinge centre of the adjustment link.



A chain drive consist of an endless chain wrapped arround two sporckets. Chain drive is used for transmitting mechanical power from one place to another. There will be no any slippage in between the chain and sprocket hence velocity ratio is maintained.



Pitch circle diameter is defined as the diameter of a circle on which the hinge centre of the chain lie.

Advantages of chain Drive;

- There will be no any slipage, hence perfect vel. ratio obtained.
- Temperature and enwoirnmental conditions do not effect their working.
- They have high transmissing efficiency upto 98%.
- They can operate in wet condition.
- They do not show fire hazard.
- It can with stand abrasive condition.
- Multiple shaft can be driven from a single chain.

Disadvantages of chain drive;

- They require frequent lubrication.
- They have less load capacity compared with gears.
- The operation is noisy and can cause vibration.
- Production cost of chain is high.
- They have velocity fluctuation, when stretched.

Selection of chain;

1. Load consideration: The chain is selected with respect to the load factors such as uniformly load, moderate load or heavy shock load.
   - Uniformly load centrifugal pump.
   - Moderat load Reciprocating pump, wood working machine.
   - Heavy shock load Earth moves, cranes.
2. Horsepower to be transmitted.
3. Rotation (RPM) of the driving and driven sprockets. That means, we must cosider the perfection of velocity ratio as far as possible.
4. Space limitation: The space between the driving and driven sporckets defines the working and length of chain.
5. Driven Machine: It depends upon the machine which operates the driving shocket.

Selection of sprockets;

1. Pinion sprocket: The pinion sprocket must have odd number of teeth which provides uniformly distributed of chain. Generally smaller sprocked is used as driving sprocket.

   Generall 17 teeth are used on smaller sprocket.

2. Larger sprocket : The number of teeth on the larger sprocket is calculated by the number of teeth on pinion Shocket. However, number of teeth is also depend upon the load variation.

Method of Lubrication:

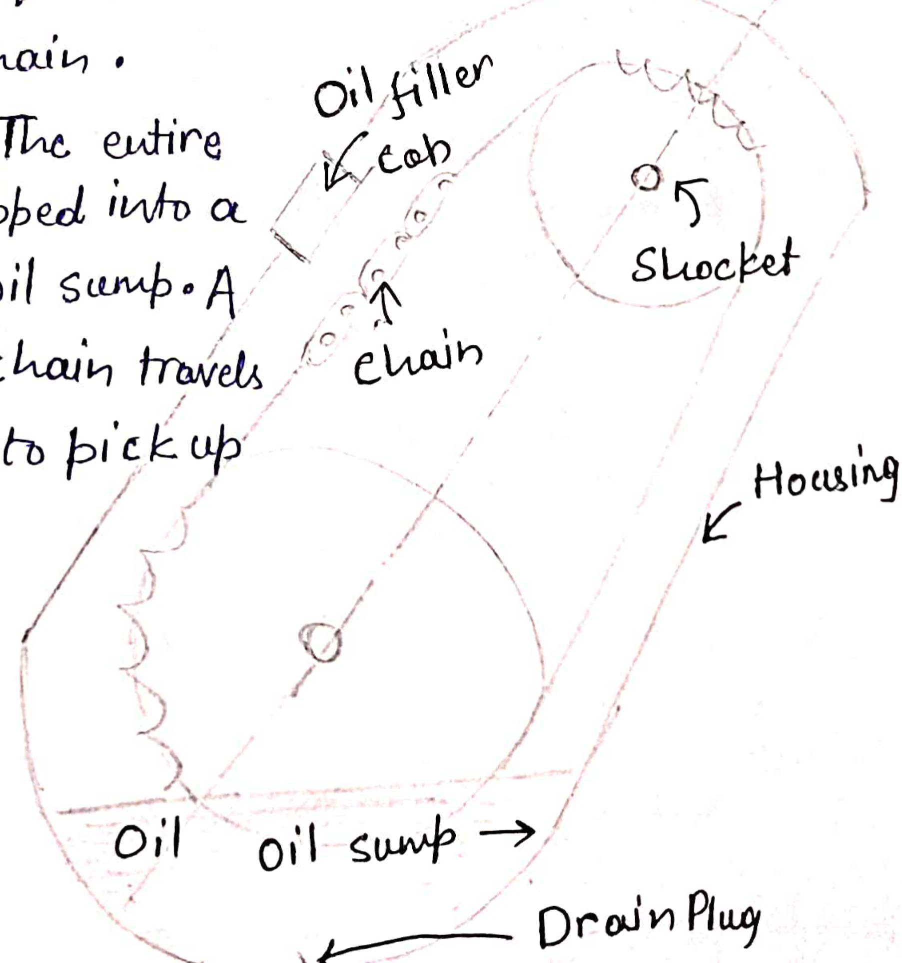
1. Manual Lubrication :- Oil is applied to the chain with a brush or spout can. The oiling frequency should be at least once each day.

It is noted that, we must concentrate on the appearance of Reddish Brown in chain Joints. It indicates that red iron oxide is generated in the chain joint due to insufficient lubrication.

2. Drip Lubrication :- Oil is dripped at a rate from 4-20 drops per minute, depending on speed.

3. Oil steam or Spray lubrication :- The oil is delivered by nozzle technique. Due to nozzle, the oil is well distributed on chain.

4. Oil Bath :- The entire drive system is dipped into a housing with an oil sump. A short section of chain travels through the sump to pick up oil.



UNIT-3

Flywheel

A flywheel used in machines serves as a reservoir, which stores energy during the period when the supply of energy is more than the requirement and release it during the period when the requirement of energy is more than supply.

For example, in internal combustion engine, the energy is developed only during power stroke which is much more than the engine load and no energy is being developed during suction, compression and exhaust stoke.

The excess energy developed during power stroke is absorbed by the flywheel and releases it to the crankshaft during other strokes in which no energy is developed, thus rotating the crankshaft.

When the flywheel absorbs energy, it speed increases and when it releases energy, its speed decreases.

"Flywheel controls the speed variation caused by the fluctuations of the engine turning moment during each circle of operation. It does not controls the speed variation caused by a varying load."

For controlling the speed variations caused by a varying load, Groverner is used.

Co-efficient of fluctuation of speed;

The difference between the maximum and minimum speed of engine during a cycle is called maximum fluctuation of speed. And the ratio of max. fluctuation of speed to the mean speed is called co-efficient of fluctuation of speed.

Let N_1 and N_2 are maximum and minimum speed of engine during a cycle.

N be the mean speed, $N = \frac{N_1 + N_2}{2}$

Max. fluctuation of speed $= N_1 - N_2$

Coefficient of fluctuation of speed $= \frac{N_1 - N_2}{N}$

$$C_s = \frac{N_1 - N_2}{\frac{N_1 + N_2}{2}}$$

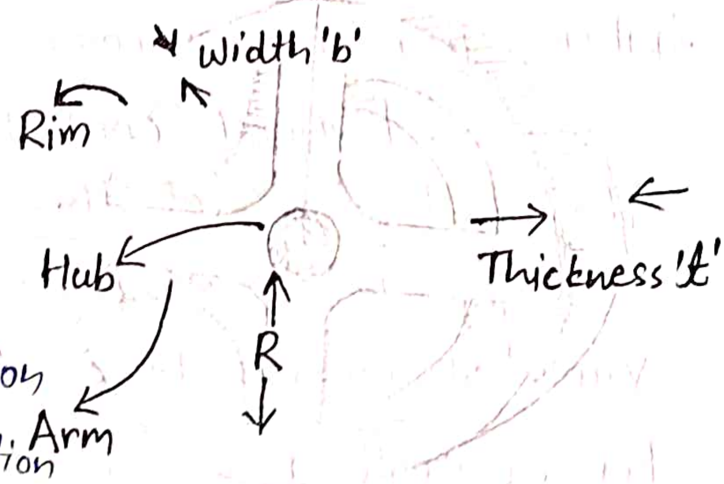
$$\therefore \boxed{C_s \Rightarrow \frac{2(N_1 - N_2)}{N_1 + N_2}}$$

$$\therefore \boxed{C_s \Rightarrow \frac{2(\omega_1 - \omega_2)}{\omega_1 + \omega_2}}$$ (In term of angular speed)

$$\therefore \boxed{C_s = \frac{2(v_1 - v_2)}{v_1 + v_2}}$$ (In term of linear speed)

"The reciprocal of the co-efficient of fluctuation of speed is known as coefficient of steadiness."

Energy stored in fly-wheel:-



I = Mass of MOI of flywheel $= mk^2$

m = mass of flywheel

k = radius of gyration

ω_1 = Max. angular speed of rotation

ω_2 = Min. angular speed of rotation

ω = Mean speed of rotation

$$C_s = \frac{\omega_1 - \omega_2}{\omega} = \frac{N_1 - N_2}{N}$$

Mean K.E. of flywheel $E = \frac{1}{2} I \omega^2 = \frac{1}{2} mk^2 \omega^2$

Max. K.E. on flywheel, $\frac{1}{2} I \omega_1^2$

Min. K.E. on flywheel, $\frac{1}{2} I \omega_2^2$.

As the speed changes from ω_1 to ω_2 the max. fluctuation of energy;

$$\Delta E = \text{max. K.E.} - \text{min. K.E.}$$
$$= \frac{1}{2} I \omega_1^2 - \frac{1}{2} I \omega_2^2$$
$$= \frac{1}{2} I [\omega_1^2 - \omega_2^2]$$
$$= \frac{1}{2} I [(\omega_1 + \omega_2)(\omega_1 - \omega_2)]$$
$$= I \cdot \frac{\omega_1 + \omega_2}{2} \times (\omega_1 - \omega_2)$$
$$= I \, \omega \, (\omega_1 - \omega_2)$$
$$= I \, \omega^2 \, \frac{(\omega_1 - \omega_2)}{\omega} = I \omega^2 C_s$$
$$= m k^2 \omega^2 C_s$$
$$\Delta E \Rightarrow 2 E C_s \quad (E = \tfrac{1}{2} I \omega^2, \; I \omega^2 = 2E).$$

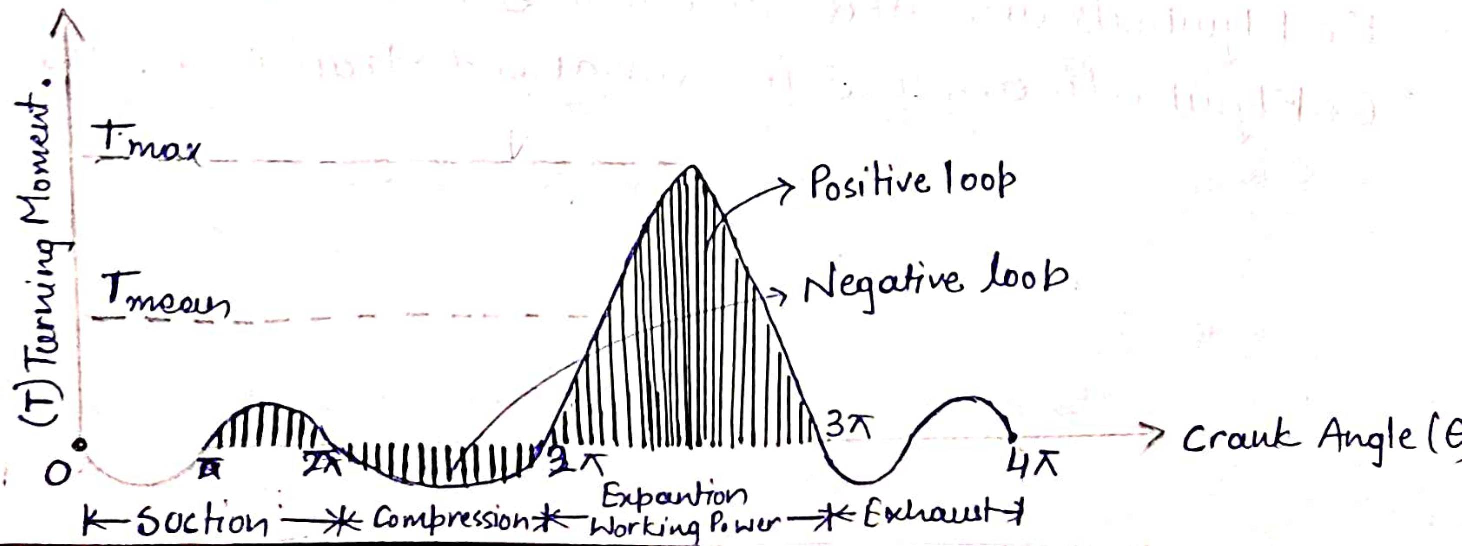
Coefficient of fluctuation of energy;

It is the ratio of the maximum fluctuation of energy to the workdone per cycle.

$$C_E = \frac{\text{Max. fluctuation of energy}}{\text{Workdone per cycle}}$$

$$\therefore \boxed{C_E = \frac{\Delta E}{W/\text{cycle}}}$$

Turning moment diagram for single cylinder 4-stroke IC engine



This diagram is called turning moment diagram for single cylinder 4-stroke IC engine. There are four stroke as follows.

a. Suction stroke:- Crank angle varies from 0°-180° (0 to π rad.).

b. Compression stroke:- Crank angle varies from 180°-360° (π to 2π rad.).

c. Expansion stroke:- Crank angle varies from 360°-540° (2π to 3π rad.).

d. Exhaust stroke:- Crank angle varies from 540°-720° (3π to 4π rad.).

Noted that, "there are two complete revolution of crank."

During the suction stroke, the pressure inside the cydinder is less than the atmospheric pressure hence the turning moment diagram is negative.

During the compression stroke, the piston works on the gases, hence a large negative loop is obtained.

During the expansion stroke, the work is done by the gases on the piston, hence a very large positive loop is obtained.

During exhaust stroke, the work is done by the piston on gases, hence a negative loop is obtaind

Application of Fly wheel:

1. In Wind turbines, flywheels are used.
2. Fly wheel are used in automobile.
3. In electric cars, the flywheels are used to boost speed.
4. Flywheels are used in locomotive propulsion system.
5. Flywheels are used to controls the direction of satellites.
6. Flywheels are used in advanced transit buses.

Note : Suppose N_1 and N_2 are the maximum & minimum speeds respectively.

Clearly, Mean Speed $= \frac{N_1 + N_2}{2} = N$

Max. fluctuation of speed $= N_1 - N_2$

Coefficient of fluctuation of speed, $C_S = \frac{N_1 - N_2}{N}$

Max. K.E $= \frac{1}{2} I \omega_1^2$, Min. K.E. $= \frac{1}{2} I \omega_2^2$

Max. fluctuation of energy, ΔE = Max K.E − Min K.E

$= 2 E C_S$

E = mean KE $= \frac{1}{2} I \omega^2$

$= 2 \times \frac{1}{2} I \omega^2 C_S$

$= I \omega^2 C_S$

for I :- a. If radius of gyration is given,

$$\boxed{I = mk^2}$$

b. If flywheel acts as solid wheel/disc

$$\boxed{I = \frac{mR^2}{2}}$$

c. if real flywheel, $\boxed{I = mR^2}$

Numerical;

Qn. The mass of fly-wheel of an engine is 6.5 ton., the radius of gyration is 1.8 m. If the maximum fluctuation of energy is 56 kN-m and the mean speed of the engine is 120 rpm. Find,

(i) Maximum & Minimum speed

(ii) Coefficient of fluctuation of speed

(iii) Mean kinetic energy of flywheel.

Qn. In a turning moment diagram, the areas above and below the mean torque line taken in order are 4400, 1150, 1300 & 4550 mm² respectively. The scale of turning moment diagram are, Turning Moment, 1 mm = 100 N-m

crank angle, 1mm = 1°

Find the mass of flywheel required to keep the speed 297 and 303 rpm (between) radius of gyration is 0.525 m.

Qn. The flywheel of an engine has a radius of gyration of 1m and mass 2500 kg. The starting torque of the engine is 1500 N-m and may be assume constant. Determine, (i) Angular acceleration of the flywheel.

(i) Kinetic energy of flywheel after 10 sec from start.

[TMM-UNIT-3-PDF-1]

GOVERNORS

Governor is a device used for maintaining the mean speed of an engine. In other words, We can say that it regulates the mean speed of an engine.

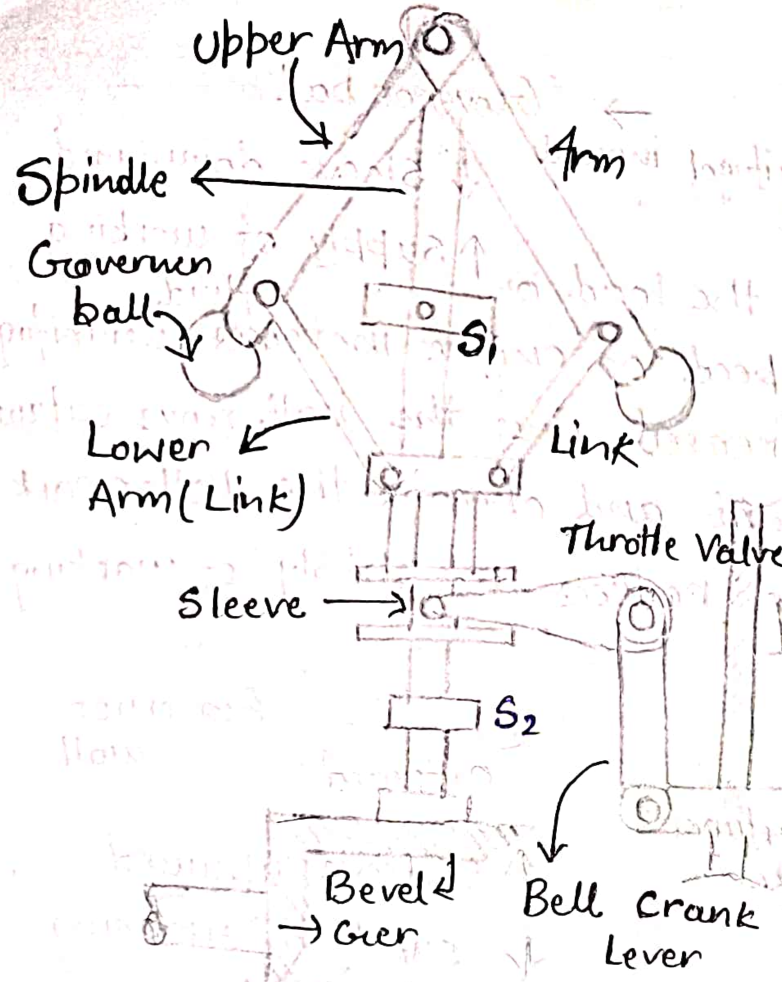
When the load on the engine increases, the speed of engine decreases. The governor will act in such a way that it will increase the supply of working fluid.

Similarly, when the load on the engine decreases the speed of engine increases. The governor is act in such a way that it will decrease the supply of working fluid.

Basic diagram and working of governor;

Governor works on the principle of centrifugal force. It consists of two balls of equal mass called governor balls or fly balls.

Governor balls are attached to the one end of arm, are pivoted to a spindle. The ~~spindle~~ upper ends of arm are pivoted to a spindle. The spindle is driven by the engine through bevel gear. The lower arm of links are connected to a sleeve which is keyed to the spindle.



The sleeve rotate with the spindle but can slide up and down. There are two stoppers S_1 & S_2 on the spindle prevents the upward and downward motion of sleeve. i.e the sleeve can travel in between S_1 and S_2 only.

The sleeve is connected by a Bell-crank lever to a throttle valve. The throttle valve controle the supply of working fluid.

Working Of Governer

Case -I :- When the load on the engine increases, the speed of the engine decreases. Also the speed of spindle decreases. Hence the centrifugal force on the governor balls will also decreases. The governor ball will move inward and hence the sleeve will operate the Bell-Crank lever and will increases the supply of working fluid by increasing the opening of throttle valve. Finally the speed of engine is increased.

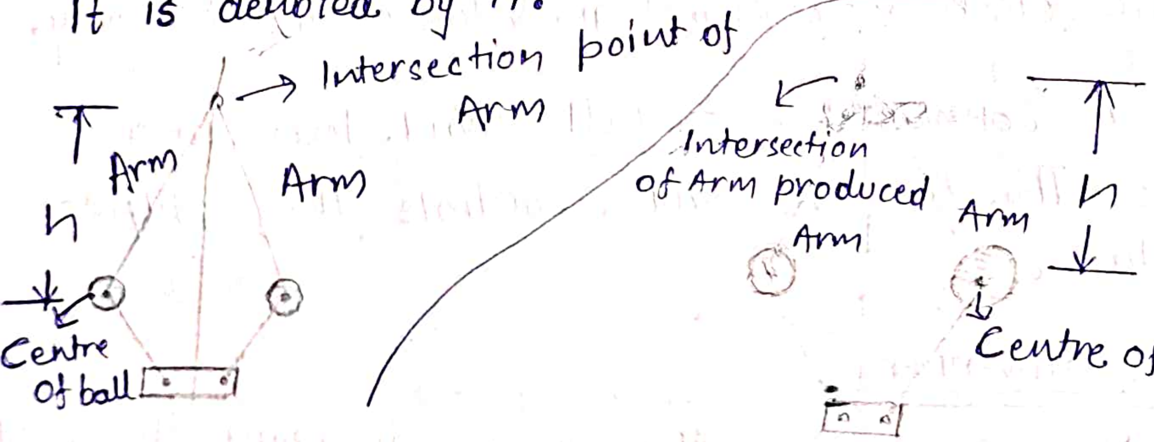
↑ Load ↓ Speed of engine ↓ Speed of spindle ↓ Centrifugal force → inward Grovernor ball
↓ Sleeve downward
↑ Supply of working fluid

Case II :— Similarly, when the load on the engine decreases, the speed of engine increases. Centrifugal force on the balls are increased. thus the ball move outward. The sleeve will rise upward and operates the Bell-crank lever. The Bell crank lever reduces the supply of working fluid by throttle valve.

↓ Load ↑ Speed of engine ↑ Speed of spindle ↑ Centrifugal force ← Outward Grovernor wall
↑ Sleeve upward
↓ supply of working fluid.

Terms used for Grovernor :–

1. Height of Grovernor : The verticle distance between centre of ball and the intersection point of arm and arm produced. It is denoted by 'H'.



2. Sleeve Lift : The vertical distance travelled by the sleeve is called sleeve lift.
3. Equilibrium Speed : The speed at which the balls, arms links are in complete equilibrium and sleep does not tend to move upward and down ward.

Types of Governors

1. Watt Governor
2. Porter Governor
3. Proell Governor
4. Hartnell Governor.

Watt Governor

Watt governor is a simplest form of centrifugal governor.

Let, m = mass of ball

W = Weight of the ball (mg)

T = tension in arm

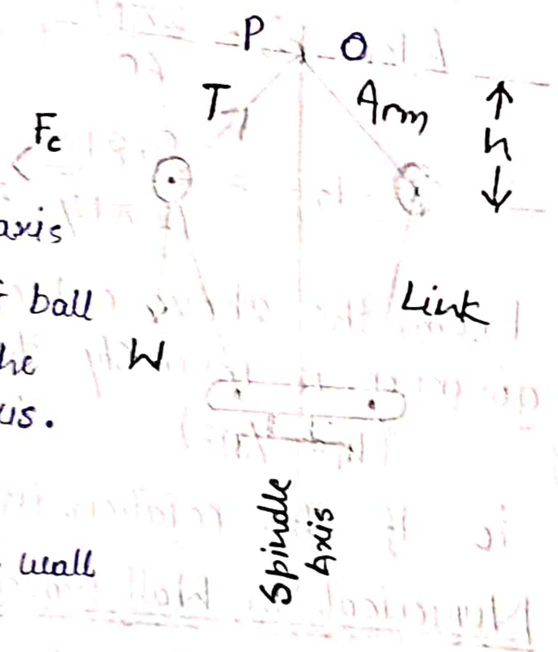
ω = Angular velocity of arm and ball about the spindle axis

r = radius of path of rotation of ball

↳ = Horizontal distance between the centre of ball and spindle axis.

h = Height of governor

F_c = Centrifugal force acting on the wall $= mr\omega^2$



Assume that, weights of arms, Links and sleeve are negligible as compared to the weight of the ball.

Now, there are three forces acting on the ball;

(i) Centrifugal force, $F_c = mr\omega^2$, acting outward.

(ii) Weight of the ball, $W = mg$, acting vertically downward.

(iii) Tension 'T' in upper arm.

Noted that, we will not consider any tension in lower arm (link), because we assume the sleeve as weightless.

Now we will find the expression of 'h'.

Taking the moment of all the force about 'O' and equate to zero.

$T \times 0 - F_c \times h + W \times r = 0$

C.W. $\rightarrow$ -ve

A.C.W $\rightarrow$ +ve

$\Rightarrow F_c \times h = W \times r$

$\Rightarrow m r \omega^2 \times h = mg \times r$

$\Rightarrow h = g/\omega^2$ — ①

Suppose, g is expressed in m/s^2 and 'ω' is in rad/s.

Also, $\omega = \frac{2\pi N}{60}$, where 'N' is rotational speed in rpm.

$\therefore h = \frac{9.81}{(2\pi N/60)^2} = \frac{9.81 \times 3600}{4\pi^2 N^2} = \frac{894.56}{N^2} = \frac{895}{N^2}$ meters

From, the above expression, we can see that the height of governor is inversely the proportional to the square of rotation. ($h \propto 1/N^2$)

ie If the rotation increases, the height will decreases.

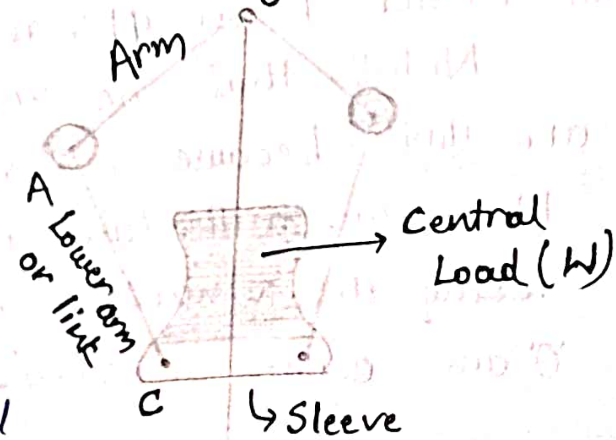
<u>Numerical on Watt Governor</u>

Qn. Calculate the change in hight of watt governor when its speed increases from 201 rpm to rpm.

<u>Portor Governor</u>

Porter governor is just a modification of watt governor.

A central load (W) is placed at the sleeve.



Let, m = Mass of ball

w = Hight of ball = mg

M = Mass of central load

W = Weight of central load

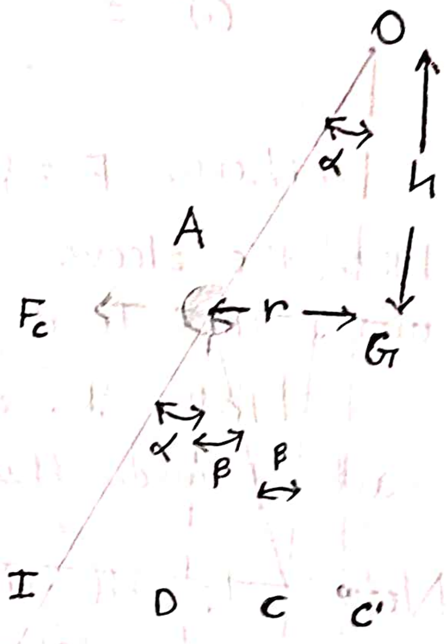
h = height of governor

r = ~~Height of governor~~ radius of rotation of ball

Fc = Centrifugal force acting on ball($mr\omega^2$)

Now, consider the left-hand half portion of governor (because the whole governor is symmetrical about spindle axis) and apply instantaneous centre method, the equilibrium of the lower arm (AC) is cosidered. Instantaneous centre 'I' can be obtained by intersection of 'OA' produced and a perpendicular line on spindle axis, ie intersecting point of OA and CD.

Note that, we consider only half portion of governor, hence at point 'C', the load will be $\frac{W}{2} = \frac{mg}{2}$.



α = angle between upper arm & governor axis ($\angle AOG$)

β = angle between lower arm & governor axis ($\angle ACG$)

Note :- If CC' is not given then W/2 will act at point C'.

Now, taking moment about 'I',

$$F_c \times AD - w \times ID - \frac{W}{2} \times IC = 0$$

$$F_c = (mg)\frac{ID}{AD} + \frac{W}{2}\left(\frac{ID + DC}{AD}\right)$$

$$\Rightarrow mg \tan\alpha + mg\left(\frac{ID}{AD} + \frac{DC}{AD}\right)$$

$$\Rightarrow mg \tan\alpha + mg(\tan\alpha + \tan\beta)$$

$$\Rightarrow mg \tan\alpha + mg \tan\alpha\left(1 + \frac{\tan\beta}{\tan\alpha}\right)$$

$$F_c = \tan\alpha\left[mg + \frac{mg}{2}(1+k)\right]$$

$$mr\omega^2 = \tan\alpha\left[mg + \frac{mg}{2}(1+k)\right]$$

$\tan\alpha = \frac{ID}{AD}$

$IC = ID + DC$

$\tan\beta = \frac{DC}{AD}$.

$k = \frac{\tan\beta}{\tan\alpha}$

Now, from the triangle ; OAG,

$$\tan\alpha = \frac{AG}{OG} = \frac{r}{h}.$$

Putting in eq. (1);

$$mr\omega^2 = \frac{r}{h}\left[mg + \frac{mg}{2}(1+k)\right]$$

$$\Rightarrow \omega^2 = \frac{mg + \frac{mg}{2}(1+k)}{m \times h} \quad — (11)$$

also $\omega = \frac{2\pi N}{60}$, N is in RPM.

Note :- Suppose friction at sleeve is considered.

$$\omega^2 = \frac{mg + \frac{mg \pm F}{2}(1+k)}{m \times h}$$

Where, F = frictional force.

When the sleeve move upward, the friction will act down ward, Thus total load at 'C' = $\frac{mg+F}{2}$.

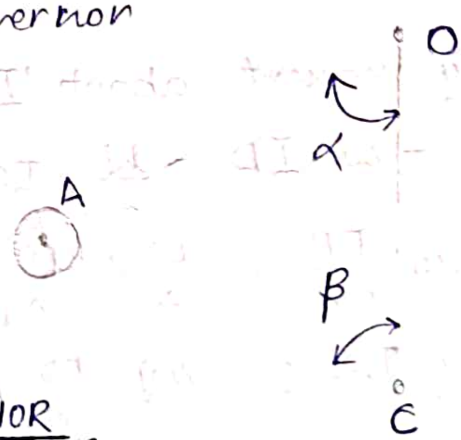
When the sleeve moves downward the friction will act upward, thus the total load at 'C' = $\frac{mg-F}{2}$.

Note :- If all the arm are of equal length and point 'O' and 'C' are on governor axis.

OA = OC,

hence. $\tan\alpha = \tan\beta$

$$\Rightarrow k = \frac{\tan\beta}{\tan\alpha} = 1.$$



NUMERICAL ON PORTER GOVERNOR

Qn. A portor governor has equal arms each 250 mm long and pivoted at the axis of rotation. Each ball has a mass of 3kg and the mass of central load on the sleeve is 14kg. Radius of rotation of the ball is 150 mm when the governor ~~at~~ begins to lift and 200 mm when the governor is at maximum speed. Find the maximum & minimum speed of governor and also range of speed.

(i) Friction at the sleeve is neglect

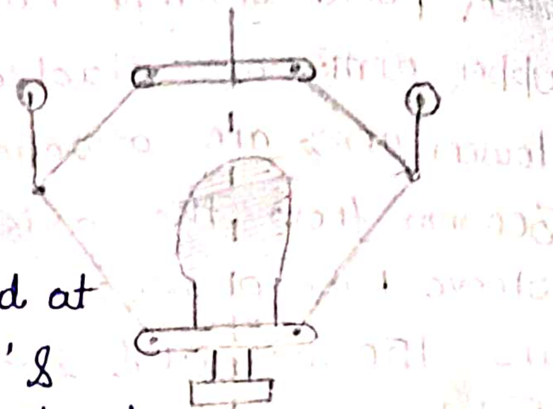
(ii) Friction of 15 N at the sleeve is taken.

Qn. A porter governor has all four arms 250 mm long. The upper arms are attached on the axis of rotation and the lower arms are attached to the sleeve at a distance of 300 mm from the axis. The mass of each ball is 5 kg and sleeve has a mass of 50 kg. The extreme radii of rotation are 150 mm and 200 mm. Determine the range of speed Friction force is neglect. [TMM - PDF - 5]

Qn. A porter governor has equal arms each 250 mm long and pivoted on the axis of rotation of. ball is ~~15~~ 5 kg ~~when the governor beg~~ and the mass of central load on the sleeve is 15 kg. The radius of rotation of the ball is 150 mm when the governor begins to lift & 200 mm when the governor at maximum speed. Find the minimum and minimum speeds and range of speed of governor.

Qn. The arms of a portor governor are each 250 mm long and pivoted on the governor axis. The mass of each ball is 5 kg and the mass of the central sleeve is 30 kg. The radius of rotation of the ball is 150 mm when the sleeve begins to rise and reaches a value of 200 mm for maximum speed. Determine the speed range of governor. If the friction at the sleeve is equivalet of 20 N of load at the sleeve, determine how the speed range is modified.

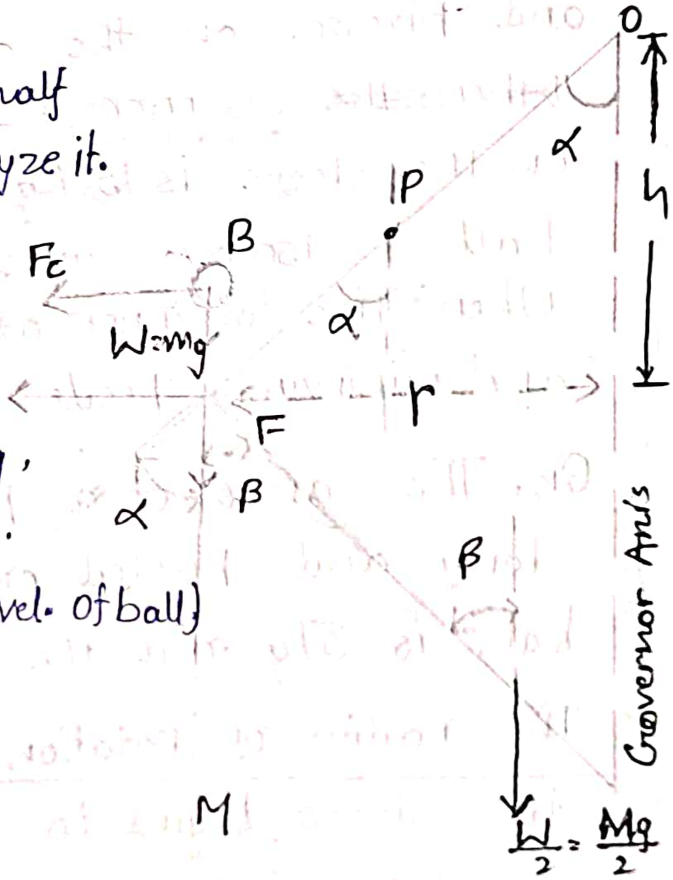
Proell Governor



The proell governor also consist a central load at the sleeve.

The Governor balls are fixed at 'B' and 'C' of link 'BF an 'CG'. 'BF' & 'CG' are the extensions of the 'DF' and 'EG'. The arms 'FP' and 'GQ' are pivoted at point 'P' and 'Q' respectively.

Now, we will consider the one-half of the proell governor and analyze it.



m = mass of ball,

w = Weight of ball = mg

M = mass of central load,

W = weight of central load = Mg,

r = Radius of rotation of ball.

ω = Rotation at ball (angular vel. of ball)

F_c : Centrifugal force ($mr\omega^2$)

h = Height of governor.

Total Moment Along 'I',

$$F_c \times BM = w \times IM + \frac{W}{2}(ID)$$

$$\Rightarrow F_c \times BM = mg \times IM + \frac{Mg}{2}(ID)$$

$$\Rightarrow F_c = mg \times \frac{IM}{BM} + \frac{Mg}{2} \times \frac{IM+MD}{BM}$$

$$\Rightarrow F_c = \frac{1}{BM}\left[mg \times IM + \frac{Mg}{2}(IM+MD)\right]$$

Multiplying and Dividing by FM;

$$\Rightarrow F_c = \frac{FM}{BM}\left[mg \times \frac{IM}{FM} + \frac{Mg}{2}\left(\frac{IM}{FM} + \frac{MD}{FM}\right)\right]$$

$$\Rightarrow F_c = \frac{FM}{BM}\left[mg \times \tan\alpha + \frac{Mg}{2}(\tan\alpha + \tan\beta)\right]$$

$$F_c = \frac{FM}{BM} \times \tan\alpha \left[mg + Mg(1 + \tan\beta/\tan\alpha)\right]$$

$$\Rightarrow F_c = \frac{FM}{BM} \times \tan\alpha \left[mg + Mg(1+q)\right] \quad -① \qquad \left(q = \frac{\tan\beta}{\tan\alpha}\right)$$

Now, $F_c = mr\omega^2$

$\tan\alpha = \frac{r}{h}$.

$$mr\omega^2 = \frac{FM}{BM} \times \frac{r}{h} \left[mg + \frac{Mg}{2}(1+q)\right]$$

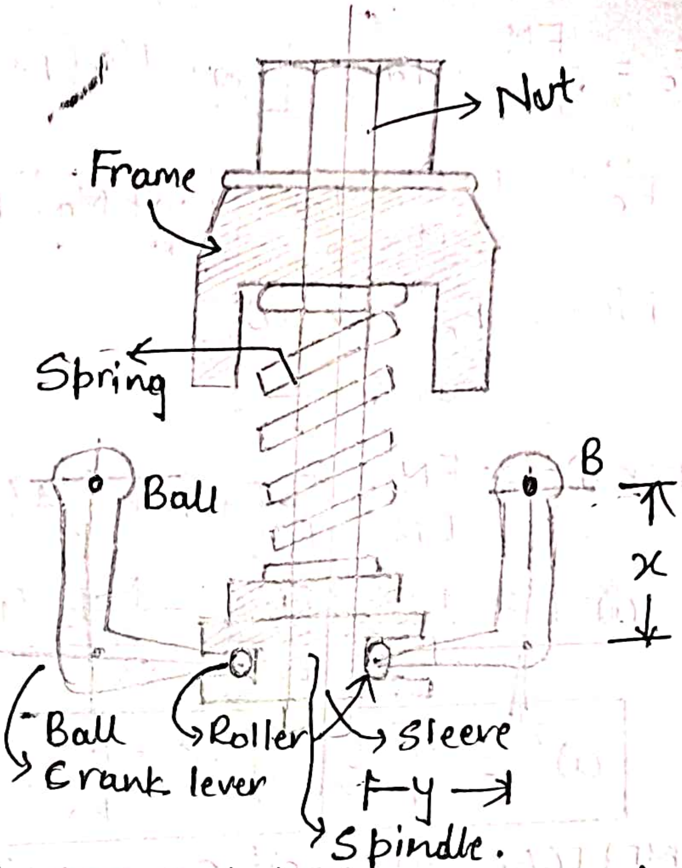
$$\omega^2 = \frac{FM}{BM} \times \left[\frac{mg + Mg/2\,(1+q)}{m}\right]\frac{1}{h}$$

$$\boxed{\omega^2 = \frac{FM}{BM} \times \left[\frac{m + M/2\,(1+q)}{m}\right] \times \frac{g}{h}.}$$

NUMERICAL ON PROELL GOVERNOR

Qn. A proell governor has equal arm of length 300 mm. The upper and lower ~~arm~~ end of the arms are povited on the axis of the governor. The extension arm of the lower links are each 80 mm long and parallel to the axis when the radii of rotation of the balls are 150 mm and 200 mm. The mass of each ball is 10 kg and the mass of centre load is 100 kg determine the range of governor.

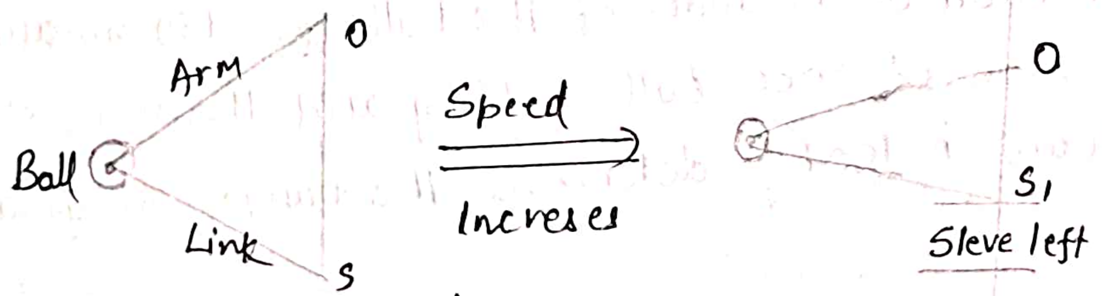
Harthell Governor



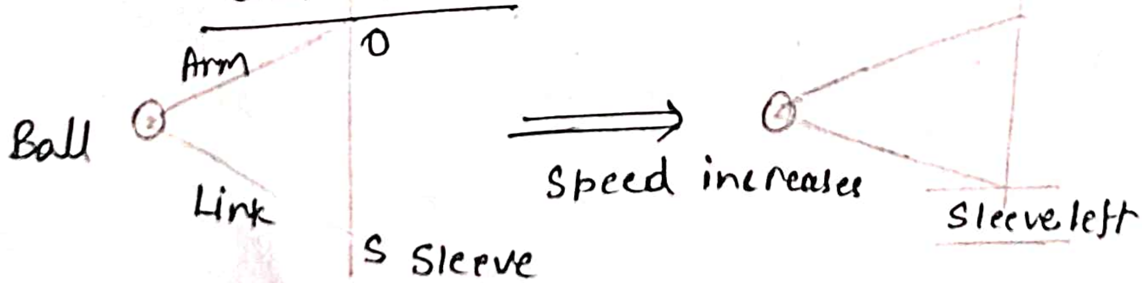
Some Important Definations Related to Governor:

1. Sensitiveness of governor; A Governor is said to be Sensitive, if the displacement of sleeve is higher due to the change in speed.

Suppose two governors, Governor ① & ⑪ as shown below;



Governor '1'



Governor '2'.

Due to changes in speed, there will be sleeve lifts in both governors. From the diagram, we can see that, sleeve lift in Governor '1' is more than the sleeve lift in governor '2'.

i.e. Sleeve lift (Governor 1) > Sleeve lift (Governor 2)

$$\Rightarrow SS_1 > SS_2$$

Thus we will say that, Governor '1' is more sensitive than Governor '2'.

Also, the sensitiveness of governor can be defined as the ratio of the difference between the maximum speed & minimum speed to the mean speed.

$$\text{Sensitivity} = \frac{N_2 - N_1}{N}$$

$$= \frac{2(N_2 - N_1)}{N_1 + N_2}$$

$$= \frac{2(\omega^2 - \omega_1)}{\omega_1 - \omega_2}$$

Note :- Sensitivity is a desirable property of governor.

2. Stability of Governor : A Governor is said to be stable when for every speed with in the working range, there is only one radius of rotation of governor ball at which the governor is in equilibrium.

Note - A Governor is said to be Unstable if the radius of rotation decreases as the speed increases.

3. Isochronism:

A Governor is said to be Isochoronious, when the range of speed is zero. That means, for all the positions of the sleeve, Balls, the governor has the same equilibrium speed.

let, r_1 = Minimum radius of speed

r_2 = Maximum ~~speed~~ radius of speed

N_1, N_2 are the speeds, then for the isochronious condition, $N_1 = N_2$.

4. Hunting—

It is related to the sensitiveness. Suppose a Governor is too sensitive, and load decreases slightly. Clearly the speed increases and the lift of sleeve will be too high (because too sensitive). Thus sleeve immediately reaches at highest position and control-valve will cut off the fuel supply. As the fuel supply is cut, the speed of engine will decreases and sleeve falls sudden. Due to the fall of sleeve, there will be wide opening of supply of fuel (excess than requirement).

This cycle is repeated indefinitely. This phenomenon is culled Hunting of governor.

Application of Governor;

1. It maintains the mean speed of the speed. We can also say, it regulates the mean speed of engine.
2. Govenors are used in railway engines.
3. Govenors are used at that place, where manual handling not posible.

Comparison Between Flywheel and Governor :-

FLYWHEEL	GOVERNOR
1. It controls the fluctuation of speed, due to the variation in turning moment.	1. It regulates the mean speed, due to the variation of load.
2. It regulates the speed in only one cycle.	2. It regulates the speed over a period of time.
3. Flywheel stores energy when supply greater than requirement and releases the energy when requirement is greater than supply.	3. Governor regulates the speed by regulating the quantity of the charge (Working fluid).
4. Flywheel has no control over supply of working fluid.	4. Governor takes care the supply of working fluid.
5. It is not essential element of an engine. It is where there will be undesirable cycle fluctuation.	5. It is essential in every engine because it is related to the fuel supply.
6. It is relatively heavy element	6. It is a light machine part.
7. It is used in punching machines, rolling mills etc.	7. It is used in Engines, Turbine etc.

Unit-4 Brakes Dynamometers, Clutches and Bearings

Function of Brakes and Dynamometer;

Brake: A brake is a device which offers the functional resistance to the moving body and the functional resistance retards the motion of the body and the body comes to rest.

The brake absorbs the kinetic energy of the moving body. This energy is dissipated in the form of heat and it is dissipited in the surrounding air (or water which is circulated through the passage in the break drum).

The function of brake depends upon:-

a. Co-efficient of friction
b. peripheral velocity
c. Projected Area
d. Pressure
e. Energy absorbed.

Dynamometer: A dynamometer is a device used for the measurement of the frictional resistance or frictional torque. This frictional resistance or frictional torque is obtained by applying a break. Hence, dynamometer is also a breake ~~[illegible]~~ in addition it has a device to measure the frictional ~~[illegible]~~ resistance or frictional torque.

Types of brake;

a. Shoe Brake

b. Band brake

c. Internal expanding shoes brake

d. Disc brake.

Types of dynamometer;

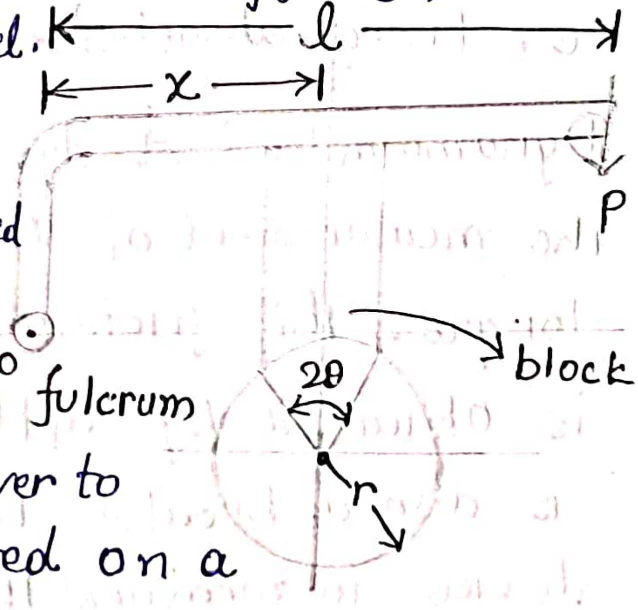
a. Rope brake dynamometer

b. Hydraulic Dynamometer

c. Eddy current Dynamometer.

SHOE BRAKE : Block or shoe brake, it consist of block or shoe which is pressed against the rim of a revolving drum. The block is made of a softer material than the rim of the wheel.

The function between the block and the wheel causes a tangential braking force, which retards the motion of wheel.



The block is pressed against the wheel by a force applied to one end of a lever to which the block is rigidly fixed. The other end of a lever to which the block is pivoted on a fulcrum 'O',

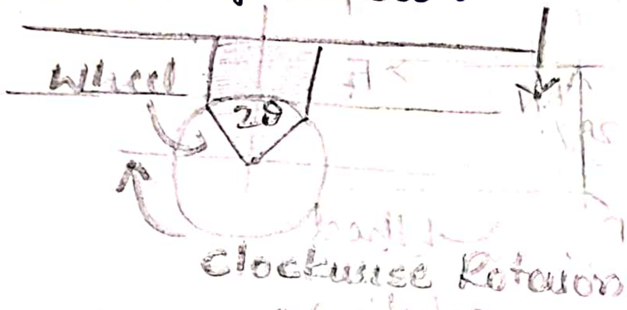
Let 'P' be force applied at the end of lever

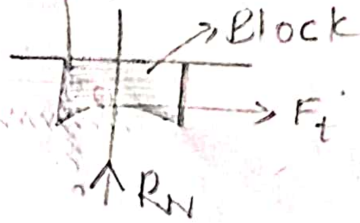
'r' radius of wheel

'2θ' Angle of contact surface of the block.

'μ' be coefficient of friction
'l' length of lever
'x' be distance between fulcrum 'O' and centre line of block.

Suppose, two conditions of clockwise and anti-clockwise rotation of wheel.





R_N = Normal Reaction on the block.
F_t = Tangential braking force or the frictional force.

If the angle of contact is less than $60°$, then we can assume that the normal pressure between the block and the wheel is uniform.

$$\therefore F_t = \mu R_N \quad -(i)$$

and braking torque;

$$\boxed{T_B = F_t \cdot r = \mu R_N r} - (ii)$$

Noted that; C.W. rotation of wheel drum, leftward Friction on
Friction on block, rightward.

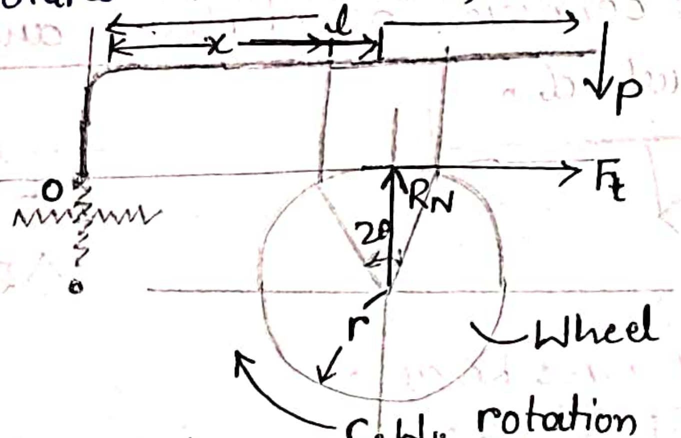
Similarly for anti-clockwise rotation of wheel;

$$F_t = \mu R_N$$

$$\boxed{T_b = \mu R_N r.}$$

Now we will take thee Cases :-

(i) When the line of action of tangetial braking force (F_t) passes through the fulcrum 'O' of the lever and the wheel rotates in clockwise,



Now, taking moments about the fulcrum 'O'

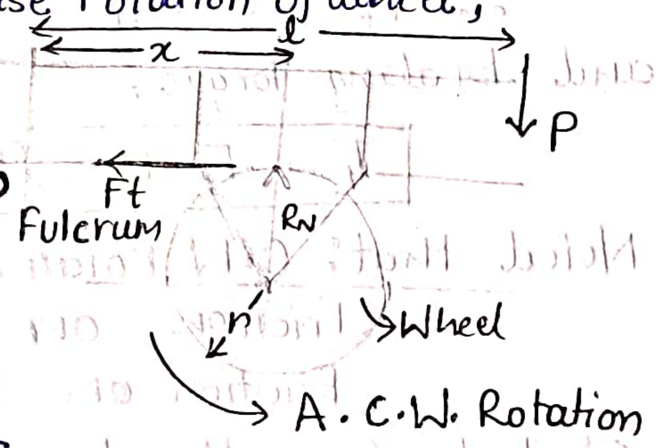
$$R_N \times x + F_t \times 0 = P \times l$$

$$R_N = \frac{Pl}{x}$$

Now breaking torque;

$$T_B = \mu R_N r = \frac{\mu P l \cdot r}{x}$$

Similarly for anticlockwise rotation of wheel;



Taking moment about the fulcrum 'O';

$$R_N = \frac{Pl}{x}$$

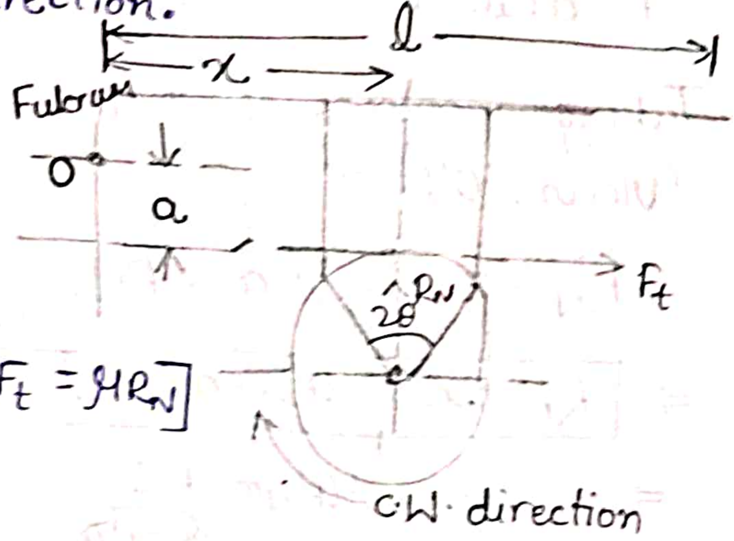
Now, Braking Torque,

$$T_B = F_t \cdot r = \mu R_N r$$

$$\Rightarrow \boxed{T_B = \frac{\mu P \cdot l \cdot r}{x}}$$

(ii) When the line of action of tangential breaing force (F_t) passes through a distance 'a' below the fulcrum '0' and the wheel rotates in CW direction.

Taking moment about the fulcrum '0';



$$R_N \times x + F_t \times a = P \times l$$

$$\boxed{R_N = \frac{Pl}{x+\mu a}} \qquad \left[\because F_t = \mu R_N\right]$$

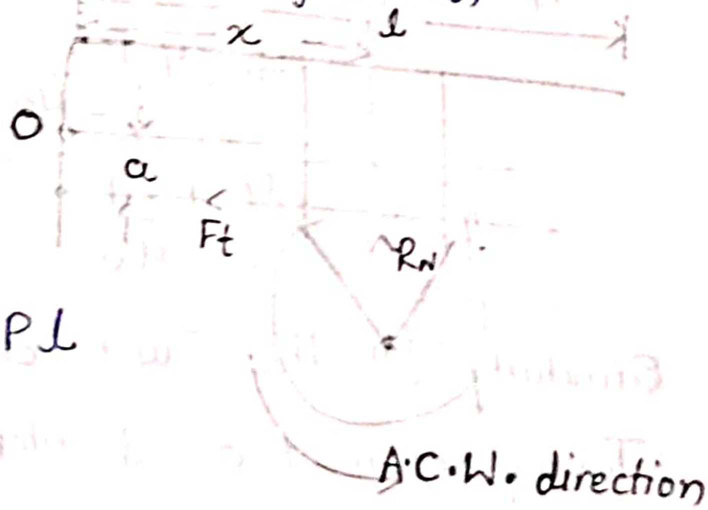
Now Braking torque;

$$T_B = F_t \cdot r = \mu R_N \cdot r$$

$$\boxed{T_B = \frac{\mu P l r}{x + \mu a}}$$

Similarly for the anticlockwise rotation of wheel;

Taking moment about the fulcrum '0';



$$R_N \times x = Pl + F_t \times a$$

$$R_N \times x - \mu R_N \times a = Pl$$

$$\Rightarrow \boxed{R_N = \left(\frac{Pl}{x-\mu a}\right)}$$

Now breaking torque;

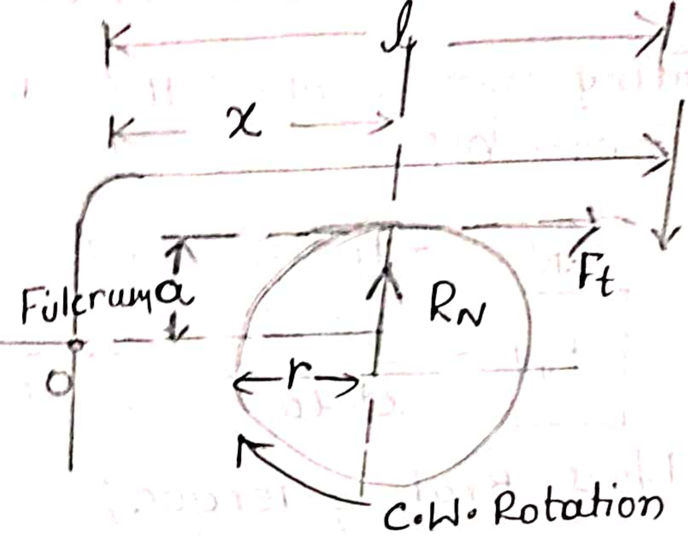
$$T_B = F_t \times r = \mu R_N \times r$$

$$= \mu \times \frac{Pl}{x - \mu a} \times r$$

$$\boxed{T_B = \frac{\mu P l r}{x - \mu a}}$$

(iii) When the line of action of the tangetial braking force (F_t) passes through a distance 'a' above fulcrum 'O' and the wheel rotates in C.W. direction.

Taking moments about the fulcrum 'O';



$$R_N \times x = F_t \times a + P \times l$$

$$\Rightarrow \boxed{R_N \times x = \mu R_N \times a + P \times l}$$

$$\Rightarrow R_N \times x - \mu R_N a = P l$$

$$\Rightarrow \boxed{R_N = \frac{P l}{x - \mu a}}$$

Now, Braking Torque;

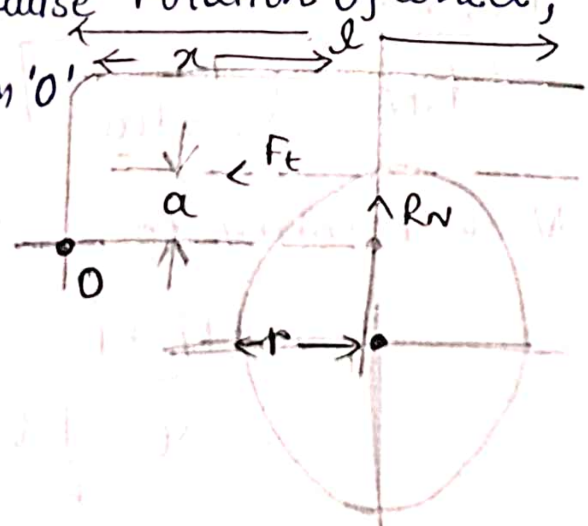
$$T_B = F_t \times r = \mu R_N \times r$$

$$\Rightarrow \mu \frac{P l}{x - \mu a} \times r$$

$$\boxed{T_B = \frac{\mu P l r}{x - \mu a}}$$

Similarly for the anti-clockwise rotation of wheel;

Taking moment about fulcrum 'O';



$$R_N \times x + F_t \times a = P \times l$$

$$\Rightarrow R_N \times x + \mu R_N \times a = P \times l$$

$$\Rightarrow R_N = \frac{P l}{x + \mu a}$$

Now braking power

$$\boxed{T_B = F_t \times r = \mu R_N r} \Rightarrow \boxed{T_B = \frac{\mu P l r}{x + \mu a}}$$

Now, we have three cases of action of brakes for C.W. and A.C.W. rotation of wheel drum.

From the above calculation and cases, we will to observe case (ii) with A.C.W rotation and case (iii) with C.W. rotation we can see that; equations (A) and (B) are same;

$$R_N x = \underbrace{Pl}_{\text{moment of applied force}} + \underbrace{\mu R_N a}_{F_t \times a \text{ moment of frictional force}} \longrightarrow (A) \text{ or } (B).$$

We can say that, Moment of frictional force ($\mu R_N a$) adds to the moment of force (Pl).

ie. frictional force is helps to apply the brake, Such type of brakes are called "Self energizing brakes".

When the frictictional force is greater enough to apply the brake with no external force then the brake is called "Self-locking brakes".

Now rearranging equation (A).

$$P = \frac{R_N (x - \mu a)}{l}$$

If $x \leq \mu a$, then 'P' will be negative or equal to zero. This means no external force is needed to apply the brake and hence the brakes is self-locking

for self-locking ; $x \leq \mu a$

Note : 1, The brake should be self energizing and not the self locking.

2, In order to avoid this self locking $x \geq \mu a$.

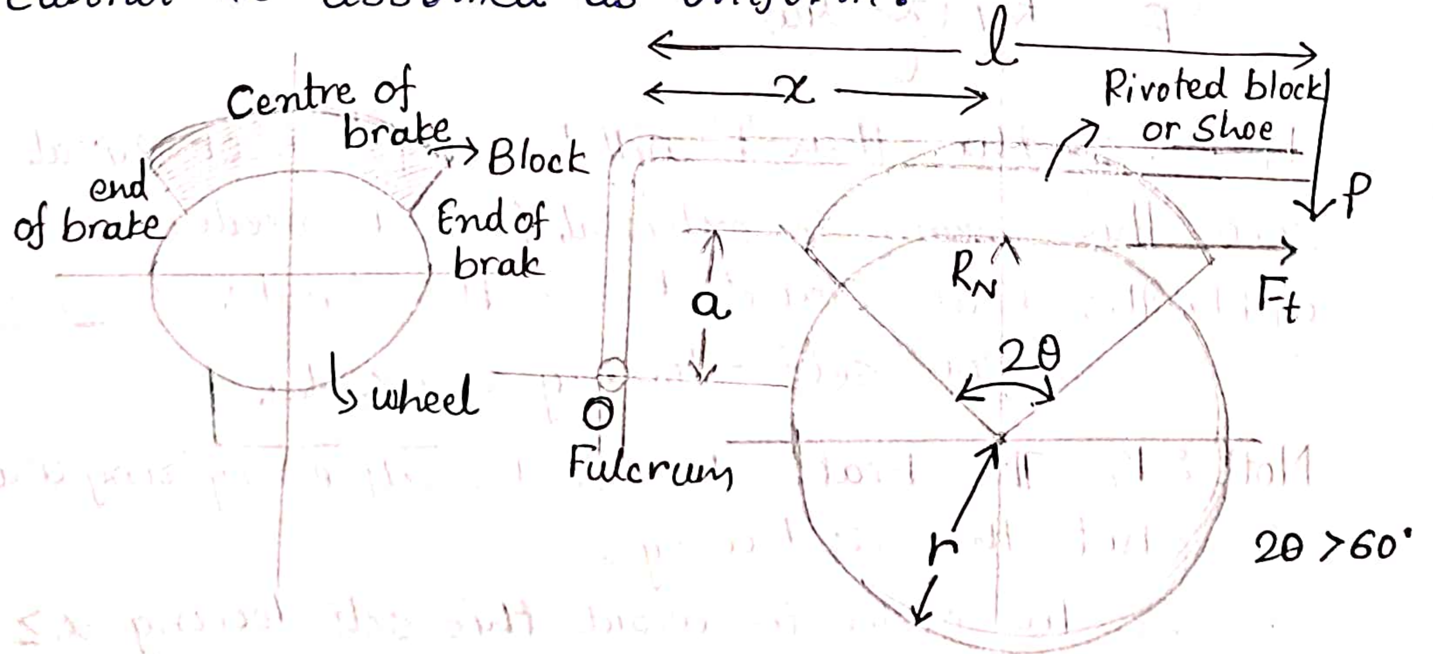
Numerical on Simple Shoe Brake:-

Qn. The brake drum of a single block brake is rotating at 500 rpm in the C.W. direction. The diameter of drum is ~~300N Angle~~ of 400 mm and the force required at the end of lever to apply the brake is 300 N. Angle of contact is 30°, Legth of level is 1 m. The line of action of tangetial braking force is belowe a distance of 25 mm and the distance between the fulcrum centre and central line of block is 300 mm. Determine the breaking torque, if coeficient of friction is 0.3. [TOM-PDF-4.3-Rec].

Pivoted Block or Shoe Brake

The normal pressure between the block and wheel is considering as uniform, when the contact angle is less than 60°.

But if the angle of contact is greater than 60° then the pressure between block and wheel cannot be assumed as uniform.



The pressure between the block and wheel at the ends ~~of~~ is less than the pressure at the centre.

The barking torque is given by:

$$T_B = F_t \times r$$

$$\boxed{T_B = \mu' \cdot R_N \cdot r}$$

When μ' = Equivalant coefficient of friction

$$\boxed{\mu' = \frac{4\mu \sin\theta}{2\theta + \sin 2\theta}}$$

Where μ is actual coefficient of friction.

Numerical Based on Pivoted Block

Qn. Diameter of drum is 250 mm and angle of contact is 90°. If the operating force of 700 N is applied at the end of lever and co-efficient of friction between drum and black is 0.35. Determine the torque transmitted. The line of action of tangetial braking forces passes through a distance of 50 mm above the fulcrum of lever. Length of lever is 450 mm and the distance between the fulcrum and centre line of block is 200 mm. The wheel rotates in clockwise direction.

Double Block Brake

In a double block brake, the braking action is doubled by the use of two blocks and the two blocks may be operated by same force.

P = applied force

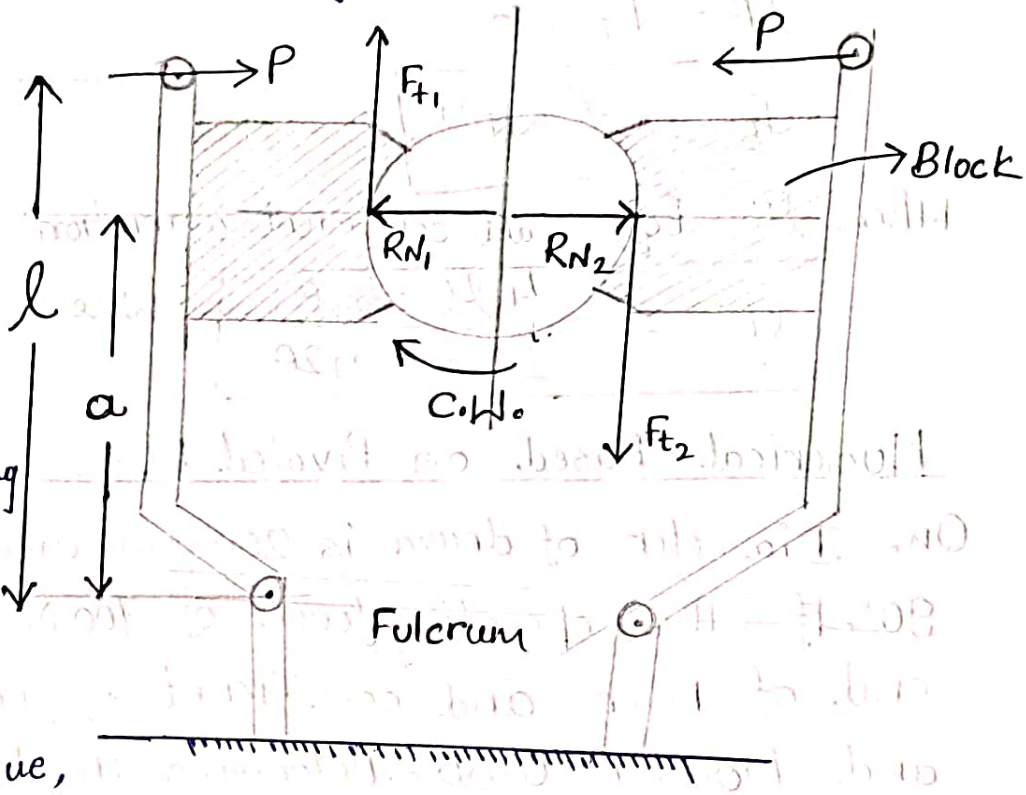
R_{N_1}, R_{N_2} are the normal reactions

$F_{t_1} = \mu R_{N_1}$

$F_{t_2} = \mu R_{N_2}$

are tangential braking forces,

O_1, O_2 are the fulcrums,



The braking torque,

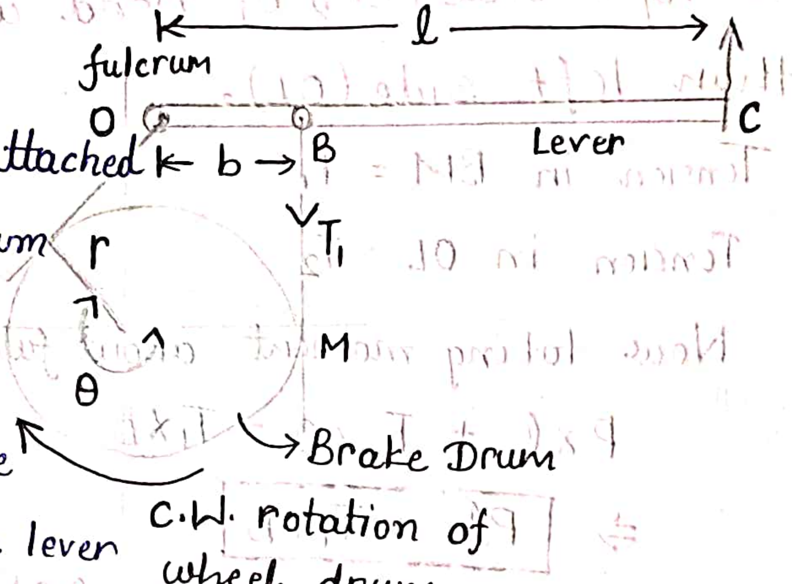
$$T = (\mu R_{N_1} \times r) + (\mu R_{N_2} \times r)$$

$$T = (\mu R_{N_1} + \mu R_{N_2}) r$$

The value of R_{N_1} is obtained by talking moments of forces about O_1. similarly R_{N_2} is obtained by taking moments of all forces about 'O_2'.

Types of brakes;

b. **Band Brake**: A band brake consist of a flexible band of leather, one or more ropes, or a sheet lined with friction material, which embraces a part of the circumferance of the drum.



One end of band is attached to a fixed pin or fulcrum (o) of the lever at a distance 'b' from the fulcrum. When a force 'P' is applied to the lever at 'c', the lever turns about the fulcrum 'O' and it will tight the band on the drum and hence brakes are applied.

Let,

T_1 = tension in the tight side of band

T_2 = Tension in the slack side of band

θ = Angle of lap or angle of embrace

μ = Coefficient of friction between band and drum.

r = Radius of drum

Now, the ratio of tension in band can be given as;

$$\frac{T_1}{T_2} = e^{\mu\theta} \qquad \theta \text{ in radian}$$

Braking force on the wheel drum $= (T_1 - T_2)$

Braking Torque $= T_B = (T_1 - T_2) r$.

Case (i) wheel rorates in C.W. direction.

Noted that due to the C.W. rotation of wheel, the tight side (BM) of bond will be more tight than left side (OL).

Tension in BM = T_1

Tension in OL = T_2

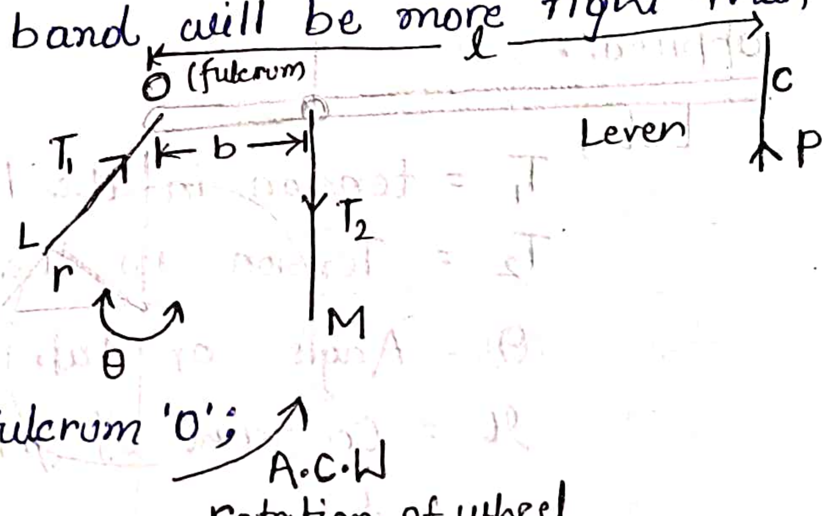
Now taking moment about fulcrum 'O';

$$P \times l + T_2 \times 0 = T_1 \times b$$

$$\Rightarrow \boxed{Pl = T_1 \times b}$$

Case (ii) Wheel rotates in ACW direction.

Noted that, due to the A.C.W. rotation of wheel, the left side (OL) of band will be more tight than right side (BM).



Tension in OL = T_1

Tension in BM = T_2

Taking moment about fulcrum 'O';

$$Pl + T_1 \times 0 = T_2 \times b$$

$$\Rightarrow \boxed{Pl = T_2 b}$$

Numerical on Band Brake

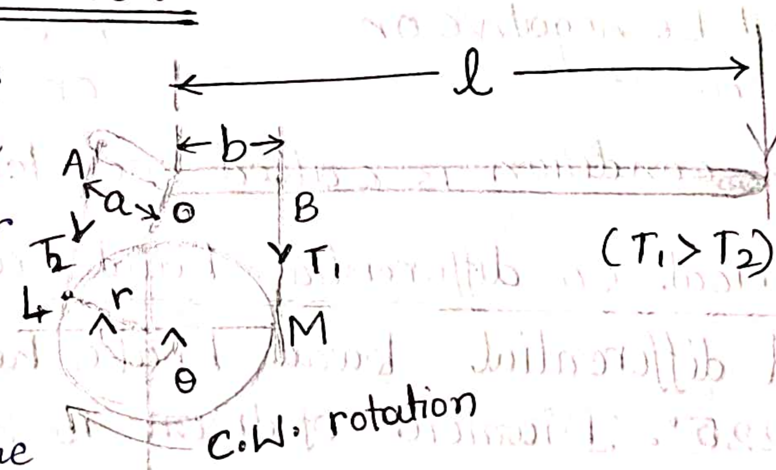
Qn. A band brake act on the 3/4 th of the circumference of a drum of 450 mm diameter. The bond brake provides a breaking torque of 225 N-m. One end of the bond is attached to a fulcrum of lever and the other end is attached at a distance of 100 mm from the fulcrum. Force is applied at a distance of 500 mm from the fulcrum.
$\mu = 0.25$.

[Tom - Rec - 4.6]

Find the applied force for,
(i) A.C.W. direction & (ii) C.W. direction.

Differential Band Brake :

One end of band is connected to 'B', of a lever AOC. The lever AOC is pivoted at fulcrum 'O'.



Noted that for the tightening of band, OA > OB

Taking moment about 'O',

$$Pl + T_1 \times b = T_2 \times a$$

$$\Rightarrow Pl = (T_2 a - T_1 b) \quad -(1)$$

Similarly for the A.C.W. rotation of wheel;

$$\Rightarrow Pl = (T_1 a - T_2 b) \quad -(2)$$

Now from equation (1) and (II)

$$Pl + T_1 b = T_2 a \quad -(1) \quad \text{C.W}$$

$$Pl + T_2 b = T_1 a \quad -(II) \quad \text{A.C.W}$$

We can say that '$T_1 b$' and '$T_2 b$' are helping in applying the brake for the C.W and A.C.W rotation of wheel respectively. Such condition is called Self energizing brake.

Again consider eqn ①, ②

$Pl = (T_2 a - T_1 b)$ — ① C.W

$Pl = (T_1 a - T_2 b)$ — ⑪ A.C.W

Suppose in eq(1)

$T_2 a \leq T_1 b$

P will be negative or zero

Suppose in eq(2)

$T_1 a \leq T_2 b$

P will be negative or zero.

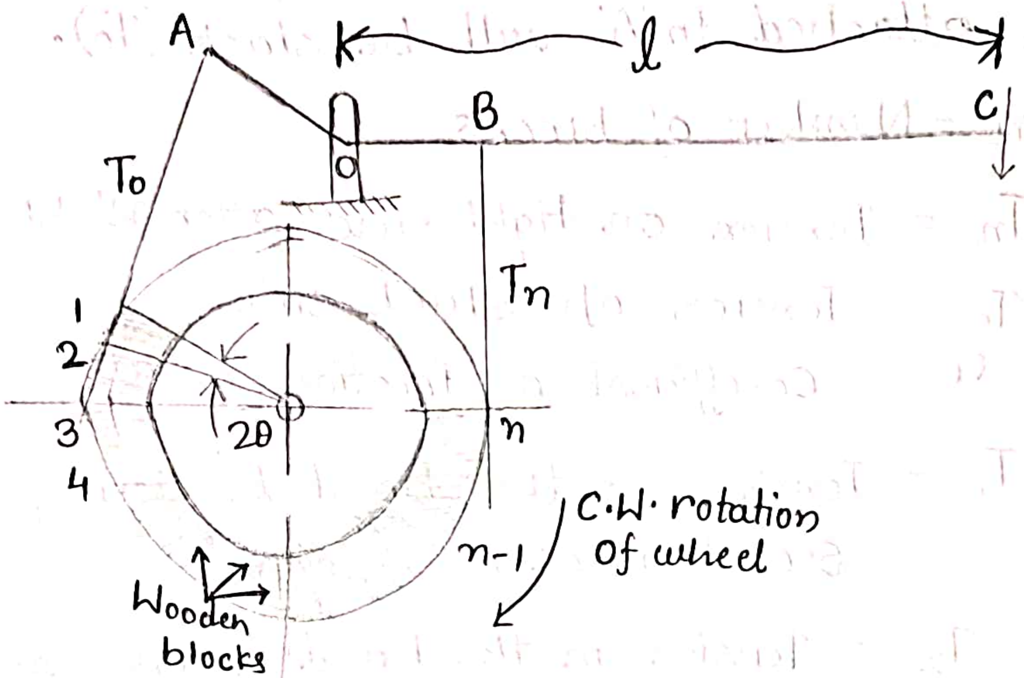
Such condition is called "self locking brake".

Numerical on differential Band brake

Qn. A differential band brake has an angle of contact of 225°. Diameter of drum is 350 mm and co-efficient of friction is given as 0.3, breaking torque is given as 350 N-m. The distance between fulcrum and left end of band is 150 mm. Distance between the fulcrum and right end of band is 35 mm. Length of lever is 500 mm. Find (i) Applied force for C.W. rotation of wheel
(ii) Applied force for A.C.W. rotation of wheel.
(iii) Condition for self locking in C.W. rotation.

[TOM - Rec - 4.6]

Band And Block Brake



Band and block brake is just a modification of Band brake. It consists of a number of wooden blocks fixed inside the flexible steel bond.

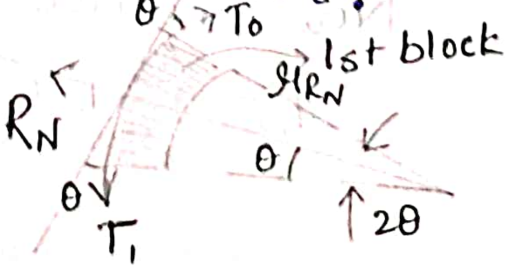
The friction between wooden blocks and wheel drum provides the braking action.

When the brake is applied, the wooden blocks are pressed against the wheel drum.

The wooden block have high co-efficient of friction hence Band and Block brake is more efficient than bond brake.

Also the wooden block can replaced easily after worn out.

Let there are 'n' number of wooden blocks, each subtends an angle '2θ' at the drum centre.



Let the rotation of wheel is in clockwise. Clearly, the band attached to 'B' will be tight (T_n), but the band attached to 'A' will be slack (T_0).

n = Number of blocks

T_n = Tension on tight side after 'n' blocks.

T_0 = Tension on slack side

μ = Co-efficient of friction.

T_1 = Tension in the band between first and Second block.

T_2 = Tension in the band between second & third block.

Considering the first block, which is under equilibrium position under the forces;

(a) Tension 'T_0' on the slack side

(b) T_1 on the tight side

(c) Normal reaction, R_N

(d) Frictional force, μR_N

" The frictional force on the wheel drum will be in the opposite direction of the rotation of drum (ie in ACW) and frictional force on the block will be opposite to the friction on wheel drum. Thus frictional force on block will be in C.W. direction

Now, resolve the forces tangentially;

$$T_1 \cos\theta - T_0 \cos\theta = \mu R_N \quad (T_1 > T_0)$$

$$(T_1 - T_0)\cos\theta = \mu R_N \quad -①$$

Now resolve the forces radially;

$$T_1 \sin\theta + T_0 \sin\theta = R_N$$

$$\Rightarrow (T_1 + T_0)\sin\theta = R_N \quad -②$$

Dividing eqs ① ②;

$$\frac{(T_1 - T_0)\cos\theta}{(T_1 + T_0)\sin\theta} = \frac{\mu R_N}{R_N}$$

$$\Rightarrow \frac{T_1 - T_0}{T_1 + T_0} \times \frac{1}{\tan\theta} = \mu$$

$$\Rightarrow T_1 - T_0 = (T_1 + T_0)\tan\theta\,\mu$$

$$\Rightarrow T_1 - T_0 = T_1 \mu \tan\theta + T_0 \mu \tan\theta$$

$$\Rightarrow T_1 - T_1 \mu \tan\theta = T_0 + T_0 \mu \tan\theta$$

$$\Rightarrow T_1(1 - \mu\tan\theta) = T_0(1 + \mu\tan\theta)$$

$$\Rightarrow \frac{T_1}{T_0} = \frac{1 + \mu\tan\theta}{1 - \mu\tan\theta}$$

Similarly, if we consider 2nd block;

$$\frac{T_2}{T_1} = \frac{1 + \mu\tan\theta}{1 - \mu\tan\theta}$$

Hence for the 'nth' block;

$$\boxed{\frac{T_n}{T_{n-1}} = \frac{1 + \mu\tan\theta}{1 - \mu\tan\theta}}$$

Now, We have to find the ratio of tension $\left(\frac{T_n}{T_0}\right)$;

$$\frac{T_1}{T_0} \times \cancel{\frac{T_2}{T_1}} \times \cancel{\frac{T_3}{T_2}} \times \cdots\cdots \times \cancel{\frac{T_n}{T_{n-1}}} = \left(\frac{1+\mu \operatorname{Tan}\theta}{1-\mu \operatorname{Tan}\theta}\right)^n$$

$$\therefore \boxed{\frac{T_n}{T_0} = \left(\frac{1+\mu \operatorname{Tan}\theta}{1-\mu \operatorname{Tan}\theta}\right)^n}$$

Now, Braking Torque;

$\boxed{T_B = (T_n - T_0)r}$ where 'r' be effective radius of band

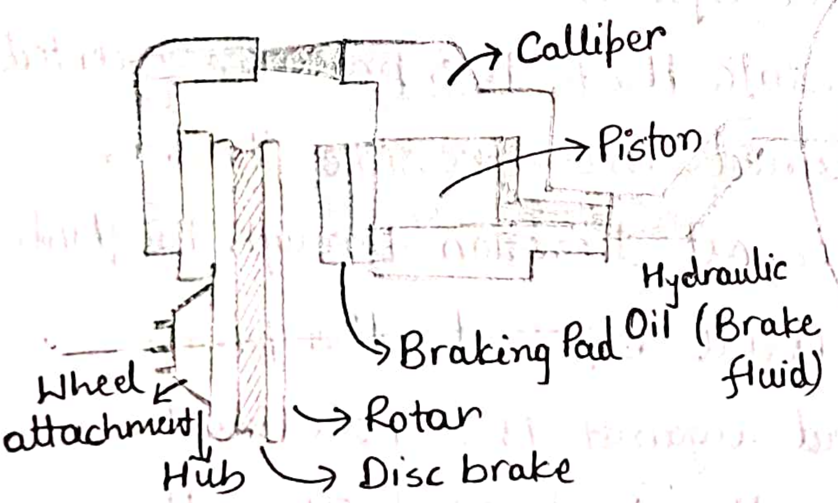
Numerical on band and block brake :

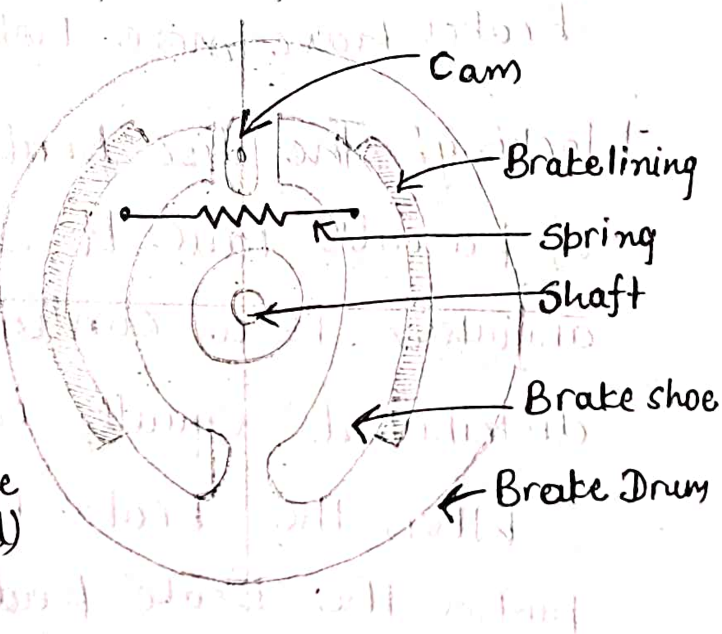
Qn. Band and block brake have 12 blocks, each of which subtends an angle of 18° at the drum centre, is applied to a rotating drum of diameter 800 mm. The blocks are 100 mm thick. The drum and flywheel mounted on the same Shaft have a mass of 1600 kg and have a combine radius of gyration of 500 mm. The two ends of the band are attached to the pins on the opposite sides of the brake fulcrum at a distance of 35 mm and 140 mm from the fulcrum. The coefficient of friction between the blocks and drum is taken as 0.3. A force 150 N is applied at a distance of 800 mm from the fulcrum to the apply the brake. Wheel rotates with C.W dir. Find (i) The maximum breaking torque, (ii) The angular retardation of the braking drum (iii) The time taken by the system to come to rest from the rated speed 240rpm

internal expanding brake

Internal expanding brakes are widely used in automobile. It consist of two brake shoes which are lined with the friction material of high coefficient of friction and good wearing properties and it mounted inside the brake drum. A cam is provided in between these shoes and it connected with the brake pedal. The spring is provided to return to its initial position after releasing the brakes. When brake pedal is pressed the cam rotates which pushes against the brake drum and brakes are applied.

Disc Brake





Wheel Hub: The disc rotor is attached to the wheel hub and rotate with it. The wheel of hub is bolted to the hub.

Brake Pad: It makes contact with the rotar disc and when the friction applies between the pad and rotor, there will be reduction in vehical speed.

Piston : It applies the braking force on the brake p
~~and rotor, there~~ when the brake lever is pressed.

Disc Rotor : It is the rotating part of disc brake. When the brakes are applied, a lot of heat is generated due to the friction between rotor and brake pad.

This heat decreases the braking efficiency; hence the rotor is drilled with vent holes. This hole helps in dissipation of heat.

These holes are provided for increase in Air flow, which helps the heat dissipation quickly. Generally front brakes have more holes than the rear brakes.

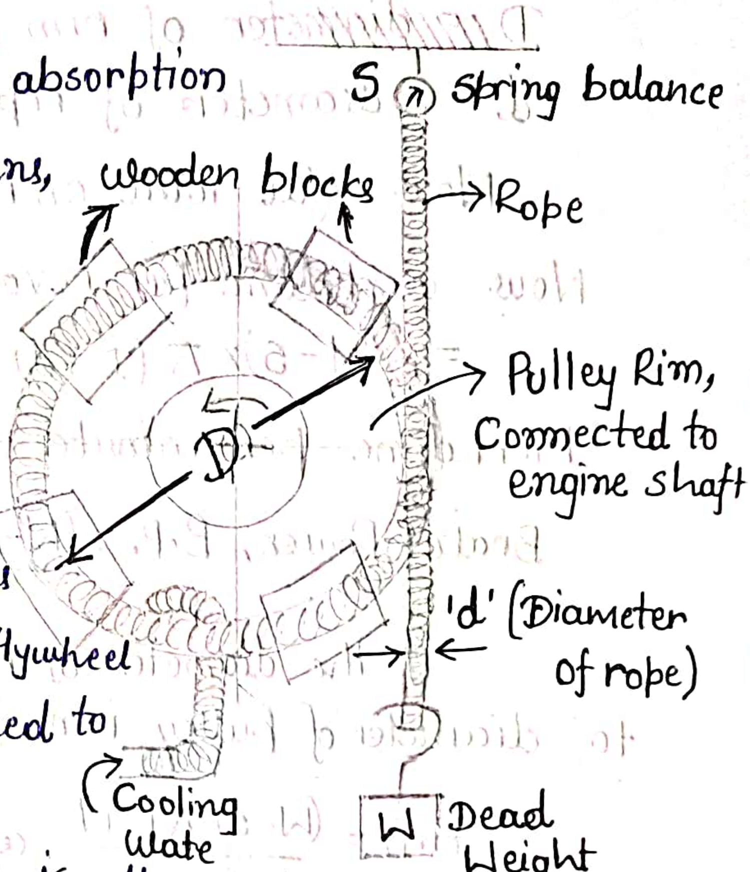
Working; The disc brake system works on the principle of pascal's law. It state that, the pressure exerted anywhere is a contained incompressible fluid is distributed equally in all direction through the fluid.

When the brake pedal is pressed, the piston pushes the brake pad against the rotating disc. Due to friction between the pad and disc, the rotating disc slows down and there will be retardation of vehicle speed.

Dynamometer :- Dynamometer is a device used for the measurement of frictional resistance or frictional torque.

1. Rope Brake Dynamometer :

Rope brake dynamometer is absorption type dynamometer. That means, the whole energy or power produced by the engine is absorbed by the frictional resistance.



One or two or more ropes are wrapped around the flywheel or rim of pulley which is fixed to the shaft of an engine.

The upper end of rope is attached to a spring balance, while the other end of rope is connected to a dead weight. The dead weight is used for keeping the position of rope.

The wooden blocks are used at some intervals of rope, to prevent the slippage between the rim and rope. A cooling arrangement is necessary, if the brake power of the engine is very large.

Measurement → For the measurement of power of an engine, the engine is made to run at a constant speed. Clearly, the torque transimitted by the engine must be equal to the frictional torque due to the ropes.

Let,

N = constant speed of engine shaft (rpm)

W = Dead Weight

S = Spring balancing reading

D = diameter of rim pullley

d = diameter of rope

Now total load on brake = $(W-S)$ Newton

Now workdone per revolution,

$= (W-S) \times \pi (D+d)$ Newton-meter

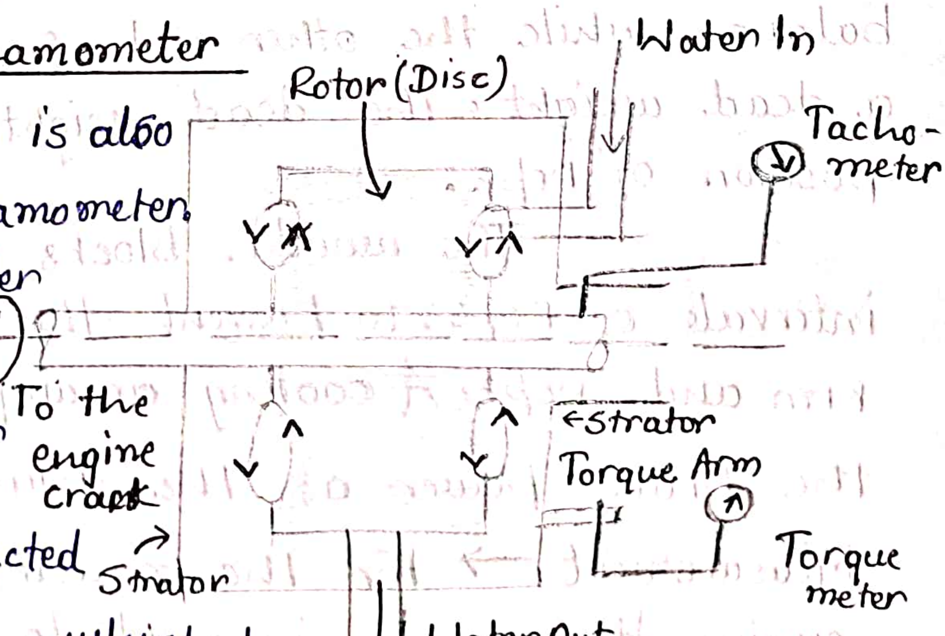
Workdone per minute $= (W-S)\pi(D+d)N$ — RPM

Brake Power, $B.P. = \dfrac{(W-S)\pi(D-d)N}{60}$ Nm/s (watts)

if the diameter of rope (d) is neglected as compared to diameter of pulley rim ie $d <<< D$.

$$B.P. = \frac{(W-S)\pi D N}{60} \text{ watts}$$

2. Hydraulic Dynamometer



Hydraulic Dynamoter is also absorption type dynamometer. It is also called Water brake dynamometer, because it used water or oil.

A shaft is connected to the engine crank, which power is to be mesured. Also the other end of shaft is connected to the rotor (rotating disc). This rotating disc is mounted in casing.

When the shaft is rotating by the engine, the water enter through the water passage (water in). Thus there will be water vortex. This shaft rotates the rotor disc.

Due to the tobulence of the water, it tends to rotate whole the casing (frame).

Noted that, the casing (frame) is fixed. ie there will be some resistance can be measured by torque meter.

Power = Torque x Rotation

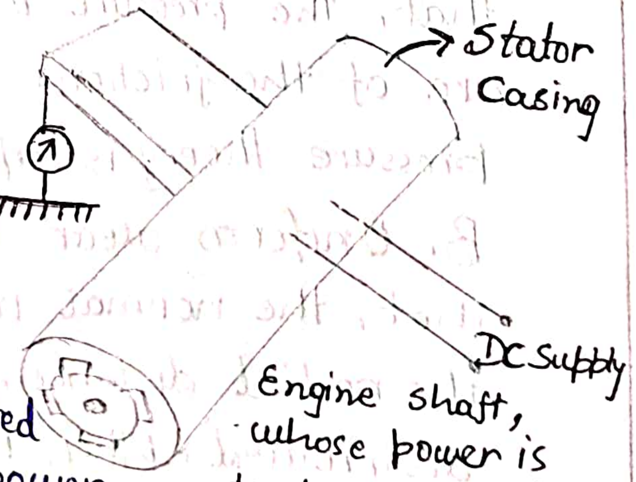
$$\boxed{P = T \times \omega}$$

ω can be measured by tachometer. $\left[\omega = \frac{2\pi N}{60}\right]$

3. Eddy Current Dynamometers;

An eddy current dynamometer works on the principle of Faraday's Law of electromagnetic induction.

It converts mechanical energy into electrical energy.



Stator Casing -

It is a stationary member.

The non-magnetic rotor is inserted in stator casing, which (rotor) power is to be measured. The stator is connected to the coil of D.C source or D.C. supply.

Cooling water arrangments are provided for the disipation of heat.

When the rotor rotates, it causes constant changes in flux density at all points of stator resulting formation of Eddy-current. It opposes the motion of rotor.

An arm is connected to the body of stator. At the end of arm, a pointer is connected, which can measure the torque produced in the rotor. By knowing the speed of rotor, the amount of power can be known.

$$\boxed{P = T\omega}$$

ω → Rotational speed; P → Power; T → Torque

Clutches: For the transmission of rotary motion from one shaft to another, which axes are coincident, clutches or friction clutches are used. Friction clutch helps the transmition of rotary motion frequently when needed and stop as per requirement in automobile.

There are two theories, which are used for used for the derivation and working of clutches,

A. Uniform Pressure Theory: In this theory we consider that, the pressure is uniformly distributed over the entire are of the friction. When the clutch is new, the uniform pressure theory is applied.

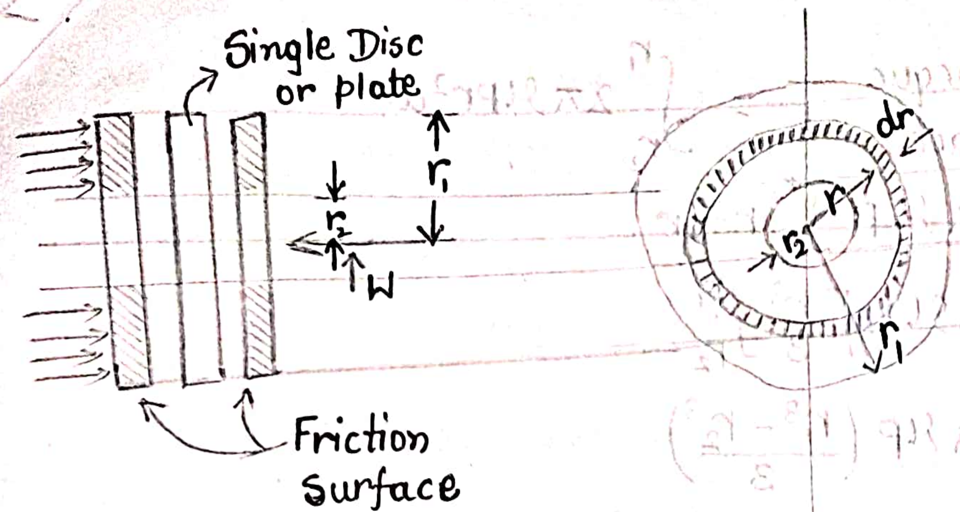
B. Uniform wear Theory: In this theory, we consider that, the normal intensity of pressure varries with the radial distance. ie Pressure is not uniformly distributed. When the cluch is old.

1. Single Disc or Single plate clutches:

Consider two friction surfaces, maintained in contact by an axial load 'W'.

Let, T = Torque transmitted by clutch

P = Intensity of pressure.



r_1 and r_2 = External and internal radii of frictional face surface

μ = Co-efficient of friction

Let us consider an elementary ring of radius 'r' and thickness 'dr'.

Now, area of contact of ring $= (2\pi r)dr$

Normal or axial force on ring, δW = Pressure x ring area

$$\boxed{\delta W = P \times (2\pi r)dr}$$

Frictional force on ring, $F_r = \mu \, \delta W$

$$\boxed{F_r = \mu P (2\pi r)dr}$$

Frictional torque on the ring, $T_r = F_r \times r$

$$T_r = \mu P (2\pi r) \cdot dr \cdot r$$

$$\boxed{T_r = 2\pi \mu P \cdot r^2 dr}$$

Now, we will consider the two cases;

a) Uniform Pressure Theory :-

Clearly, the pressure is uniformly distributed over the frictional surface.

$$\therefore \boxed{P = \frac{W}{\pi(r_1^2 - r_2^2)}}$$

Also we know that, frictional torque on elementary ring

$$\boxed{T_r = 2\pi \mu P \cdot r^2 dr}$$

$\therefore$ Total frictional torque overall the surface, $T = \int_{r_2}^{r_1} 2\pi \mu p r^2 dr$

$$T = 2\pi\mu p \int_{r_2}^{r_1} r^2 dr$$

$$T = 2\pi\mu p \left[\frac{r^3}{3}\right]_{r_2}^{r_1}$$

$$T = 2\pi\mu p \left(\frac{r_1^3 - r_2^3}{3}\right)$$

$$T = 2\pi \mu \frac{W}{\pi(r_1^2 - r_2^2)} \times \left(\frac{r_1^3 - r_2^3}{3}\right)$$

$$\boxed{T = \frac{2}{3}\mu W \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]}$$

or $\boxed{T = \mu W \times R}$, R, mean Radius, $\frac{2}{3}\left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^3}\right)$

(b) Uniform wear Theory →

In this theory, $p \times r$ = Constant,

$$p \times r = c \quad -②$$

In other words, the value of pressure rises with the decrement of radius.

Now, as we know that,

Normal force on the ring, $\delta W = p \times 2\pi r dr$

$\Rightarrow \delta W = \frac{c}{r} \times (2\pi r) dr$ { From ②; $p = c/r$!

$\Rightarrow \delta W = 2\pi c \, dr$

Total force acting on the frictional surface,

$$W = \int_{r_2}^{r_1} 2\pi c \, dr = 2\pi c(r_1 - r_2)$$

$$\therefore \boxed{W = 2\pi c (r_1 - r_2)}$$

$$\boxed{c = \frac{W}{2\pi(r_1 - r_2)}} \quad -③$$

Now also frictional torque acting on the ring,

$$T_r = 2\pi \mu P r^2 dr$$

$$= 2\pi \mu \frac{c}{r} \times r^2 dr$$

$$T_r = 2\pi \mu c \cdot r dr$$

$\therefore$ Total frictional torque, $T = \int_{r_2}^{r_1} 2\pi \mu c \cdot r dr$

$$T = 2\pi \mu c \cdot \left[\frac{r^2}{2}\right]_{r_2}^{r_1}$$

$$\Rightarrow T = 2\pi \mu c \frac{(r_1^2 - r_2^2)}{2}$$

$$\Rightarrow T = \pi \mu c (r_1^2 - r_2^2)$$

$$\Rightarrow T = \mu \pi \frac{W}{2\pi(r_1 - r_2)} \times (r_1 + r_2)(r_1 - r_2)$$

$$\Rightarrow \boxed{T = \frac{1}{2} \mu W (r_1 + r_2)}$$

$$\Rightarrow \boxed{T = \mu W R} \quad R, \text{mean Radius}, \frac{r_1 + r_2}{2}$$

Note;

(i) In general, total frictional torque,

$T = \mu W R$ Where, $R = \frac{2}{3}\left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right)$

For Uniform pressure

$R = \frac{1}{2}(r_1 + r_2)$

For Uniform Wear.

(ii) Since, Pr = Constant for uniform wear

$\therefore$ $P_{max} \times r_2$ = Constant,

ie At inner radius pressure intensity is maximum

$\therefore$ $P_{min} \times r_1$ = Constant,

ie At outer radius, pressure intensity is minimum.

(iii) Generally both sides of clutch disc are effective ie there are two effective side, $n = 2$.

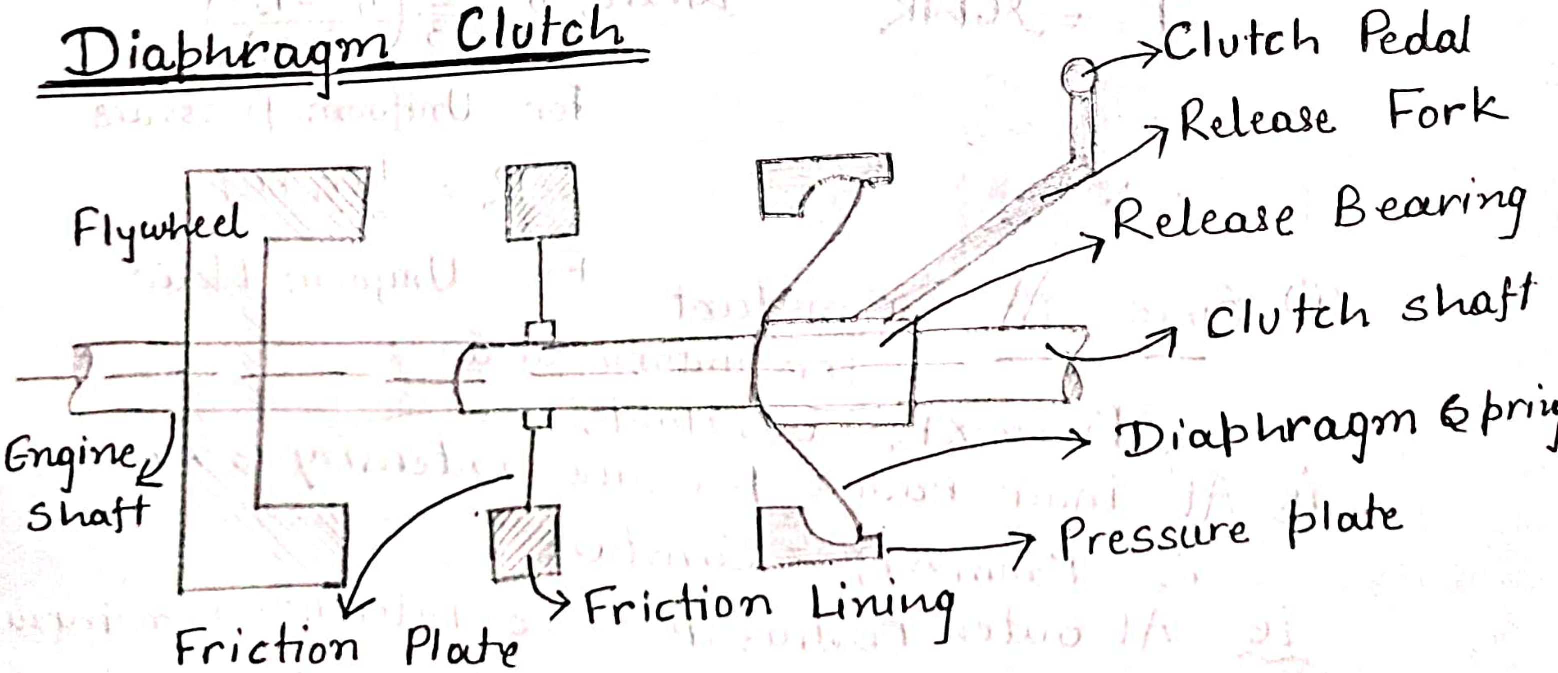
Numerical on single plate clutch :

Qn. A single plate clutch with both sides effective has outer and inner diameters of 300 mm and 200 mm respectively. The maximum intensity of pressure at any point in the contact surface is not to exceed 0.1 N/mm^2. Coefficient of friction is 0.3. Determine the power transmitted by clutch at a speed of 2500 rpm. [Tom - Rec-4.9]

Qn. A single plate clutch is required to transmit 8 kW at 1000 rpm. The axial pressure is limited to 70 kN/m^2. The mean radius of plate 4.5 times the radial width of friction surface. If the both sides of plates are effective and coefficient of friction is 0.25. Find,

(i) inner and outer radius of plate and mean radius.
(ii) Width of friction lining. [Tom - Rec -4.9]

Diaphragm Clutch



Diaphragm Clutch :

Diaphragm clutch consist of diaphragm spring, frictional plate, presssure plate etc.

The diaphragm spring is in the form of conical steel disc, which outer periphery is located in pressure plate.

Construction :-

1. Flywheel → Flywheel is connected to the engine crank-shaft. When the pressure plate press the friction plate against the flywheel, the torque from the flywheel is transmitted to the clutch shaft.

2. Pressure Plate → Pressure plate is used to press the friction plate against the flywheel. This is carried by the diaphragm spring.

3. Friction plate → Friction plate has friction lining on both sides. The friction lining is actually responsible for the torque transmission.

4. Diaphragm sping ⇒ It is circular in shape and used to keep pressure on pressure plate.

5. Release Bearing :→ It is maunted on the clutch shaft.

Working of Diaphragm Clutch;

Disengagement :

a). Driver presses the clutch pedal, release fork will press the release bearing.

b). The release bearing presses the middle portion of the diaphragm, thus the middle portion of the diaphragm will move inward.

c). Due to the inward movement of middle portion of diaphragm, outside part of diaphragm moves backward as well as pressure plate also moves backwards.

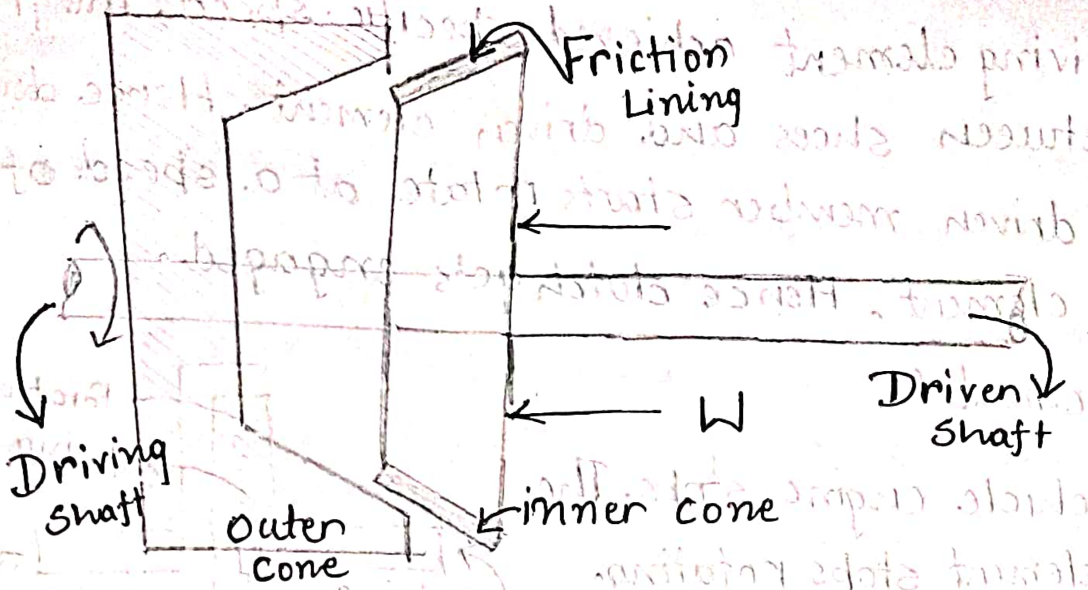
d). Due to the backward movement pressure on friction on friction plate gets removes. Thus there is no friction between plates and flywheel. Hence no power transmission takes place. Hence the clutch gets disengages.

Engagement :-

Similarly, when the clutch pedal is released, the outside portion of diaphragm moves inside and presses the the pressure plate on friction plate. The friction plate gets the contact with the flywheel and torque is transmitted from flywheel to the clutch shaft. Hence clutch engages.

Cone Clutch: Cone clutch is a type of friction clutch. If two rotating surfaces, which are lined with friction material bring in contact in rotating condition, they both start revolving at same speed.

1. In a cone clutch, there are a outer cone which is connected to driving shaft and an inner cone which is connected to driven shaft.
2. The outer cone has friction lining on its inner conical surface and inner cone has friction lining on its conical surface (outer).



Engagement :

In the normal position of clutch, when a vehicle is running, the inner cone is pressed inside the outer cone. Therefore due to friction occurs between them, Power is transmitted. Hence the clutch is is engage position.

Disengagement :

When the driven presses the clutch pedal, the inner cone moves towards the right side against compressing spring. Hence there is no friction between the outer cone and innercone. Therefore no power transmission takes place. Hence the clutch is in disengage position

Centrifugle clutch:

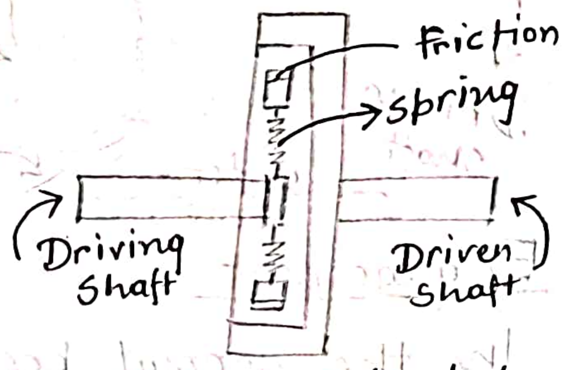
Engagement :

1. When vehicle engine starts, the driving element start rotating.
2. When driving element ~~achieved~~ start rotating, the shoes expande Radially due to centrifugal force.

3. When driving element achieved specific speed, the fric[tion]
starts between shoes and driven element. Hence due to
friction, driven member starts rotate at a speed of
driving element. Hence clutch gets engaged.

Disengagement :

When Vehicle engine stops, The driving element stops rotating. Hence shoes comes inwards the spider. Hence clutch gets disengaged.

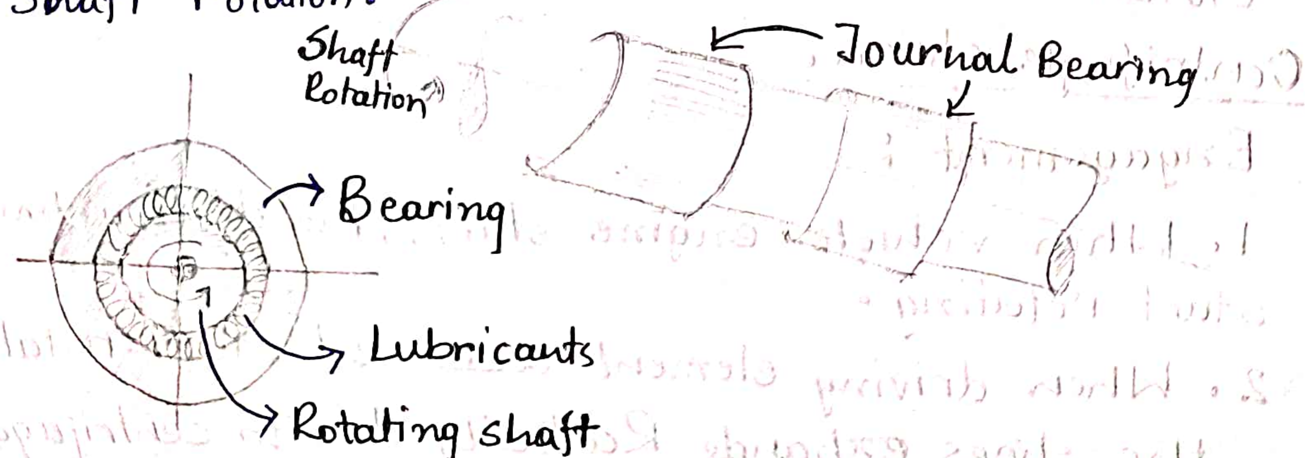


Centrifugal clutch

Bearings:

As we know that, the shaft is used for the transmission of power from one end to another. Obviousl[y] the shaft needs a good support to ensure stability and frictionless rotation. This support for the shaft is known as "Bearing".

Some proper amount of lubricants are interested in between the shaft and bearing for the reduction in friction. The bearing provides the correct position of shaft rotation.



1. Simple Pivot Bearing

The rotating shaft are frequently subjected to axial thrust. This axial thrust are taken by the flat pivot bearing. The axial thrust of propeller shaft of ships the shaft of steam turbine can be taken by flat pivot bearing. The one end of rotating shaft is inserted in the flat pivot bearing.

The one end of rotating shaft is inserted in the flat pivot bearing.

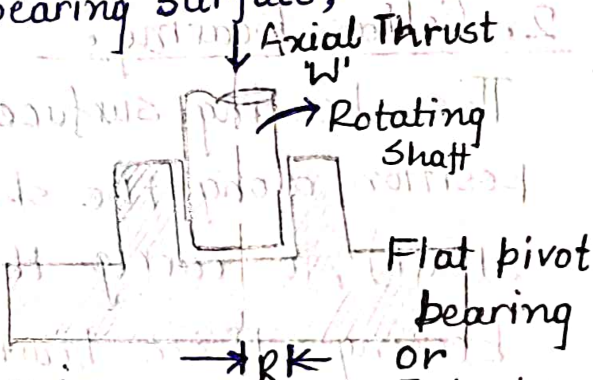
W = load transmitted over the bearing surface,

R = Radius of bearing surface

P = Intensity of pressure

T = Total frictional Torque

μ = Coefficient of friction



Clearly, there will be sliding friction in between the contact surface of rotating shaft & flat bearing.

Case -1: Uniform Pressure Theory →

That means, the pressure the uniformly distributed over the rubbing surface and bearing surface is 'New'.

Pressure Intensity, $P = \dfrac{W}{(\pi R^2)}$ → Total Area of bearing surface.

Frictional Torque, $T = \frac{2}{3}\mu W R$

Power lost due to friction, $P = T\omega = T \times \dfrac{2\pi N}{60}$; N- is in RPM.

Case -2: Uniform Wear Theory →

That means, the pressure the uniformly distributed and bearing is Old. $P \times r$ = Constant.

$W = 2\pi C R$

Total frictional torque, $T = \frac{1}{2}\mu W R$

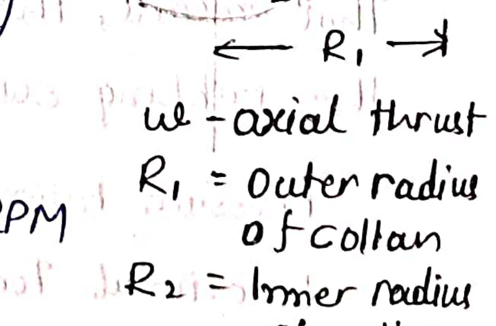
Power lost in friction, $P = T\omega = T \times \frac{2\pi N}{60}$ → N is in RPM

Numerical on simple Pivot Bearing;

Qn. Estimate the power lost in friction for (i) Uniform pressure and (ii) uniform wear.
A vertical shaft of 100 mm diameter rotating at 1500 rpm in a flat foot-step bearing. The shaft carries a load of 15 kN. $\mu = 0.05$. [Tom - Rec - 4.10]

2. Collar Bearing:

The bearing surface which can be connected at any position along the shaft (but not at the end of the shaft), to carry the axial thrust, is called as collar bearing. These bearings are also called Thrust bearing. The surface of collar bearing may be flat or conical.



Case 1; Uniform Pressure Theory

Pressure Intensity, $P = \frac{W}{\pi(R_1^2 - R_2^2)}$

Frictional Torque, $T = \frac{2}{3}\mu W\left(\frac{R_1^3 - R_2^3}{R_1^2 - R_2^2}\right)$

Power lost in friction, $P = T \times \omega = \frac{T 2\pi N}{60}$ if N is in RPM

W - axial thrust
R_1 = Outer radius of collar
R_2 = Inner radius of collar.

Case 2; Uniform Wear Theory

$P \times r$ = constant,

axial thrust, $W = 2\pi C(R_1 - R_2)$

Frictional Torque, $T = \frac{1}{2}\mu W(R_1 + R_2)$

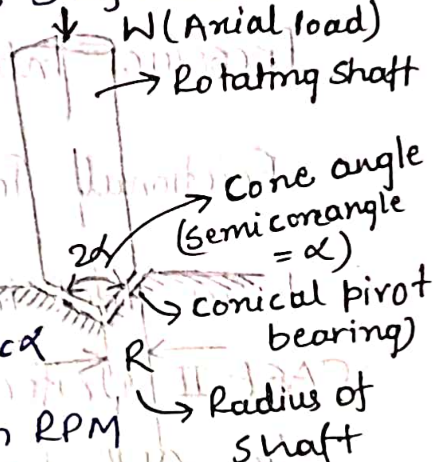
Power lost, $P = T\omega = \frac{T 2\pi N}{60}$ if N is in RPM.

Numerical on collar bearing;

Qn. In a thrust bearing, external and internal diameters are 320 mm and 200 mm. Total axial load is 80 kN and intensity of pressure is 350 kN/m². The shaft rotates at 400 rpm. $\mu = 0.06$. Calculate the power lost in friction and the number of collar required.

3. Conical Pivot Bearing

The bearing surface at the end of a shaft will have a conical surface,



Case 1. Uniform pressure theory;

ie New Bearing,

Pressure Intensity, $P = \frac{W}{\pi R^2}$

Total frictional Torque, $T = \frac{2}{3} \mu W R \cdot \operatorname{Cosec}\alpha$

Power lost, $P = T \times \omega = \frac{T \times 2\pi N}{60}$ N is in RPM

Case 2. Uniform Wear Theory;

ie old bearing,

Axial Load, $W = 2\pi C R$

Total frictional Torque; $T = \frac{1}{2} \mu W R \cdot \operatorname{Cosec}\alpha$

Power Lost, $P = T \times \omega \Rightarrow P = \frac{T 2\pi N}{60}$ N is in RPM.

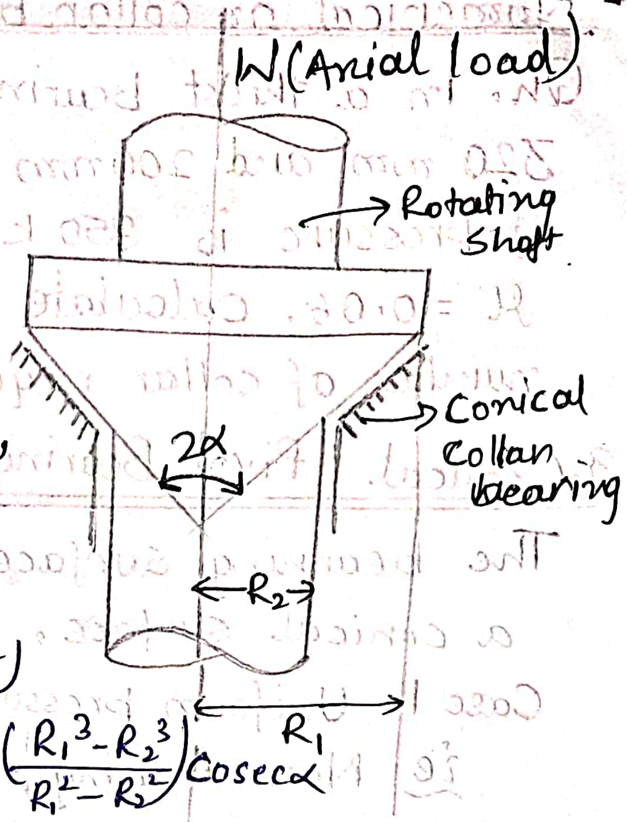
Numerical on conical Pivot bearing;

Qn. A conical pivot with angle of cone as 120°, Support a verticle shaft of diameter 300 mm. It is subjected to a load of 20 kN. $\mu = 0.05$, Speed of shaft is 210 rpm. Calculate the power lost

(i) Uniform Pressure

(ii) Uniform wear.

4. Conical Collar Bearing;



R_1 = External Radius of conical collar bearing

R_2 = Internal Radius of conical collar bearing.

CASE - I Uniform Pressure Theory,

ie New Bearing

Pressure Intensity, $P = \dfrac{W}{\pi(R_1^2 - R_2^2)}$

Frictional Torque, $T = \dfrac{2}{3} \mu W \left(\dfrac{R_1^3 - R_2^3}{R_1^2 - R_2^2}\right) \operatorname{cosec}\alpha$

Power lost in friction, $P = T\omega = T \times \dfrac{2\pi N}{60}$

CASE-II Uniform wear Theory,

ie Old bearing,

Frictional Torque, $T = \dfrac{1}{2} \mu W (R_1 + R_2) \operatorname{cosec}\alpha$

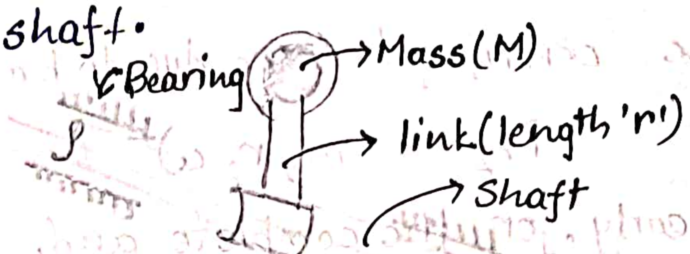
Power Lost in friction, $P = T\omega = T \times \dfrac{2\pi N}{60}$.

Numerical on conical collar bearing

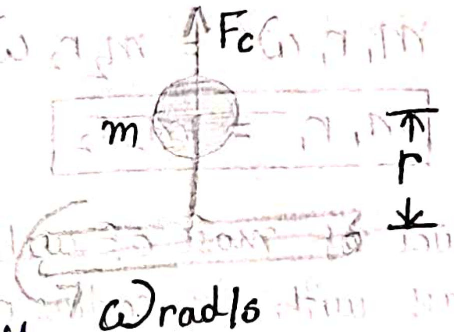
Qn. A conical pivot with angle of cone as 100° supports a load of 18 kN. The external radius is 2.5 times the internal radius. The shaft rotates at 150 rpm. If the intensity of pressure is 300 kN/m² and $\mu = 0.05$ then find the power lost.

UNIT - 5

Balancing; Suppose a body of mass 'm' is mounted on a link, and the link is fixed to a shaft.



Now, suppose the shaft is rotating with some angular velocity (ω rad/s). Clearly as the shaft rotate, the mass (m) will also rotate with the shaft with a radius of rotation of 'r'.

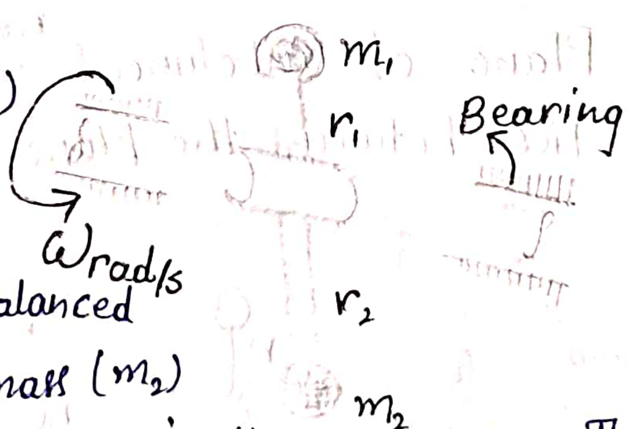


Due to the rotation of shaft, the mass 'm' will subjected to a centrifugal force. This centrifugal force will tend to bend the shaft and hence the mass (m) is called unbalanced mass.
The unbalanced centrifugal force can be expressed as

$$\boxed{F_c = m r \omega^2}$$

Balancing of single rotating mass; We will balance the single unbalance rotating mass. There are two ways as follows;

(A) Balancing of sigle rotating mass by a single mass rotating in the same plane;



Suppose an unbalanced mass (m_1) rotates with angular velocity 'ω' with radius 'r'.

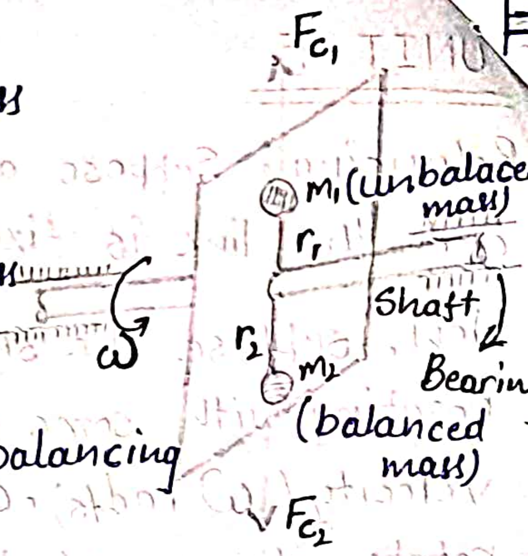
In order to balance the unbalanced mass, we will connect an other mass (m_2) Just opposite to the unbalanced mass in the same plan. This mass (m_2) is called balanced mass at radius 'r_2'.

The centrifugal force due to unbalaced mass

$$F_{c_1} = m_1 r_1 \omega^2$$

The centrifugal force due to balanced mass

$$F_{c_2} = m_2 r_2 \omega^2$$



Clearly, for the complete and proper balancing

$$F_{c_1} = F_{c_2}$$

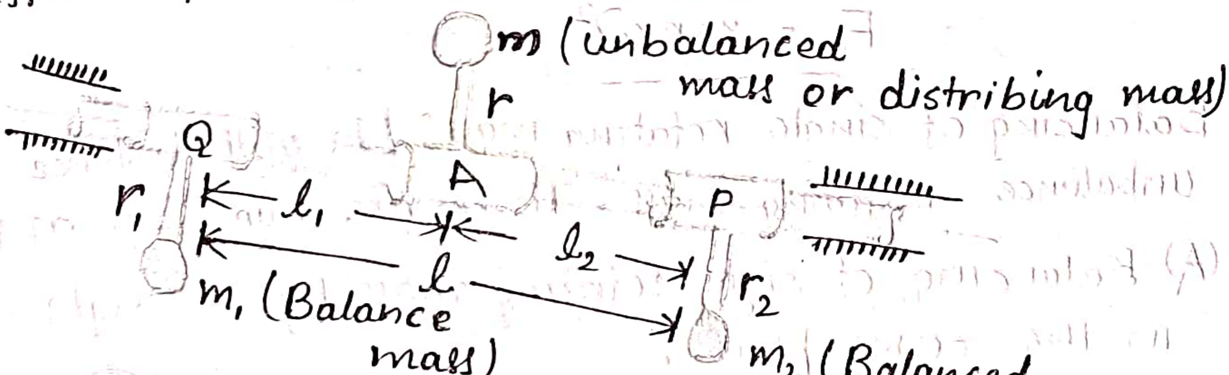
$$\Rightarrow m_1 r_1 \omega^2 = m_2 r_2 \omega^2$$

$$\Rightarrow \boxed{m_1 r_1 = m_2 r_2}$$

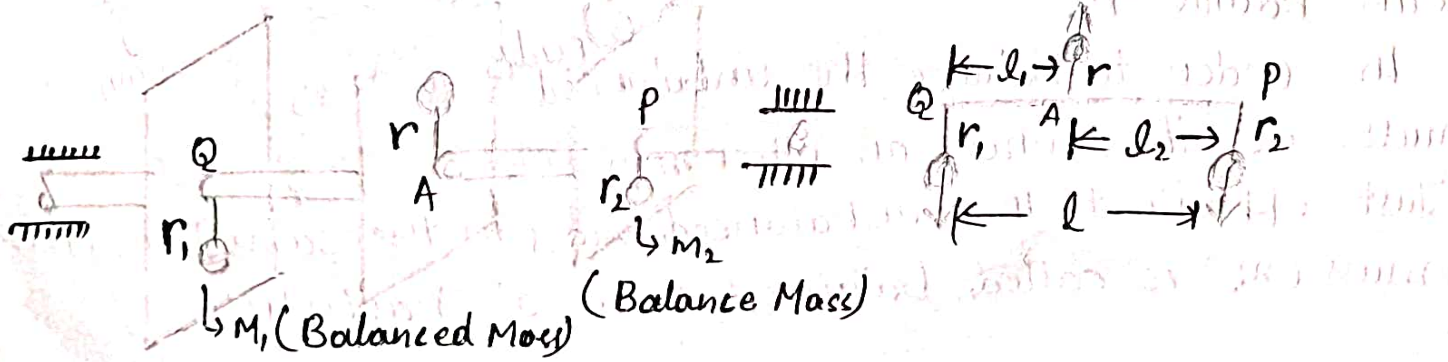
∴ Product of mass of unbalanced mass with its radius of rotation. = Product of mass of balanced mass with its radius of rotation.

(b) Balancing of a single rotating mass by two masses in different planes;

CASE I;



Plane of unbalanced mass (Disturbing mass) lies between the planes of the two balancing masses.



There is one unbalanced mass (m) which have radius of rotation (r), which rotates with angular velocity (ω). The unbalanced mass (m) is balanced by two balanced masses (m_1 & m_2) which are connected at a distance of l_1 and l_2. All the three masses in the three different planes.

Suppose the centrifugal force acted on the unbalanced mass is [$F_c = mr\omega^2$] and centrifugal forces acted on the balanced masses are:

$$F_{c_1} = m_1 r_1 \omega^2 \text{ and } F_{c_2} = m_2 r_2 \omega^2.$$

clearly For complete balancing:

$$F_c = F_{c_1} + F_{c_2}$$

$$mr\omega^2 = m_1 r_1 \omega^2 + m_2 r_2 \omega^2$$

$$\therefore \boxed{mr = m_1 r_1 + m_2 r_2} \text{ — (i)}$$

Now, we should concentrate on the 'moment'.

Take moment of all forces about 'Q',

$$F_{c_1} \times 0 + F_c \times l_1 = F_{c_2} \times l$$

$$\Rightarrow mr\omega^2 l_1 = m_2 r_2 \omega^2 l$$

$$\therefore \boxed{m_2 r_2 = mr \frac{l_1}{l}} \text{ — (ii)}$$

Take moment about 'P';

$$F_{c_1} \times l = F_c \times l_2 + F_{c_2} \times 0$$

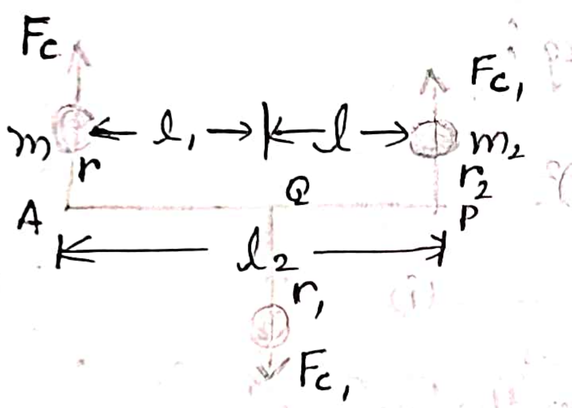
$$\Rightarrow m_1 r_1 \omega^2 l = mr\omega^2 l_2$$

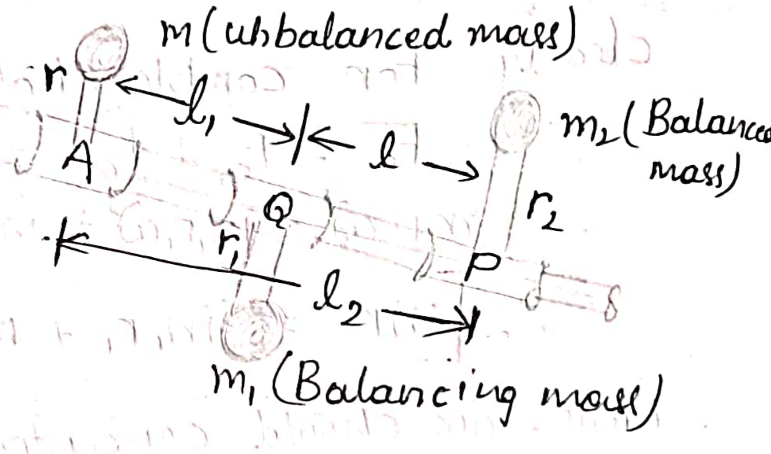
$$\Rightarrow \boxed{m_1 r_1 = mr \frac{l_2}{l}} \text{ — (iii)}$$

From eq (1); we can say that, the total dynamic force on the shaft is equal to zero.
This condition is called "Static Balancing".

From eq (2) and (3); we can say that, the sum of moments of all the forces must be equal to zero.
The condition for zero and total moment zero are called "Dynamic Balancing".

CASE 2.
Plane of unbalanced mass lies on one end of the planes of the two balancing masses.





Now, clearly the dynamic force acting on the shaft must be zero.

$$\therefore F_c + F_{c_2} = F_{c_1}$$
$$\Rightarrow m r \omega^2 + m_2 r_2 \omega^2 = m_1 r_1 \omega^2$$
$$\Rightarrow \boxed{m r + m_2 r_2 = m_1 r_1} \text{ Static Balancing}$$

Now, take moment about 'P',

$$\therefore F_c \times l_2 = F_{c_1} l + F_{c_2} \times 0$$
$$\Rightarrow m r \omega^2 \times l_2 = m_1 r_1 \omega^2 l$$
$$= \boxed{m_1 r_1 = m r \frac{l_2}{l}} \text{ Dynamic Balancing}$$

Now, moment about 'Q'

$$\therefore F_c \times l_1 + F_{c_1} \times 0 = F_{c_2} \times l$$
$$\Rightarrow m r \omega^2 l_1 = m_2 r_2 \omega^2 l$$
$$\Rightarrow \boxed{m_2 r_2 = m r \frac{l_1}{l}} \text{ Dynamic Balancey}$$

Qn. A distubing mass of 600 kg is attached to a shaft. The shaft is rotating with 'ω' rad/s. The C.G. of disturbing mass from the axis of rotation is 270 mm. The disturbing mass is balanced by two masses in two different planes. The C.G. of the balancing masses from the axis of rotation is 450 mm each. The balancing masses is 1.5 m (Distance between two balancing masses) and the distance between the balancing masses and disturbing mass is 300 mm.

Determine;

(i) Find the ~~planes~~ position of other plane.

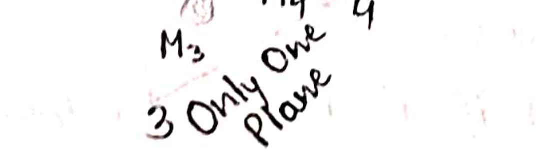
(ii) Magnitude of balancing masses;

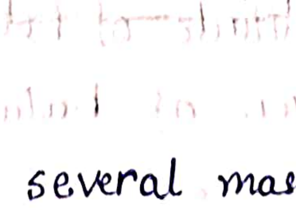
a) planes of balancing masses are on the same side of the plane of the disturbing mass.

b) planes of the balancing masses are on either side of the planes of the disturbing mass. [TMM-UNIT5-PDF 2]

Balancing of Several Masses rotating in same plane;

In this topic, there are several numbers of unbalanced or disturbing masses are rotating in the same plane.





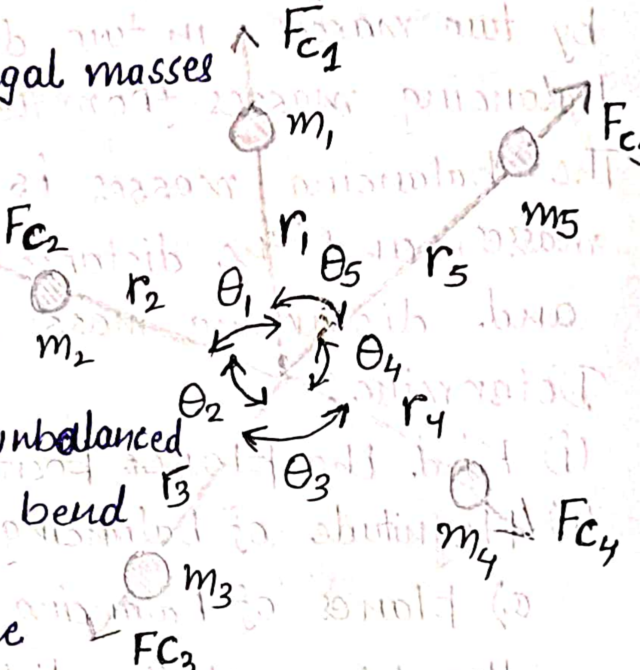
In the above figure, there are several masses $m_1, m_2, m_3 \ldots m_5$ are connected on the rotating shaft and all the masses are on the same plane (Plane 1-2-3-4).

$r_1, r_2 \ldots, r_5$ are the radii of rotation of unbalance masses $m_1, m_2, \ldots, m_5$ respectively.

If the value of masses and radii of rotation are known then centrifugal forces of each mass can be easily calculated. Also the angles between the masses are provided.

$F_{c_1}, F_{c_2} \ldots\ldots F_{c_5}$ are the centrifugal masses $m_1, m_2 \ldots, m_5$ respectively,

$F_{c_1} = m_1 r_1 \omega^2$, $F_{c_2} = m_2 r_2 \omega^2, \ldots\ldots,$
$F_{c_5} = m_5 r_5 \omega^2$.



In this topic all the masses are unbalanced or disturbing which are tend to bend the shaft.

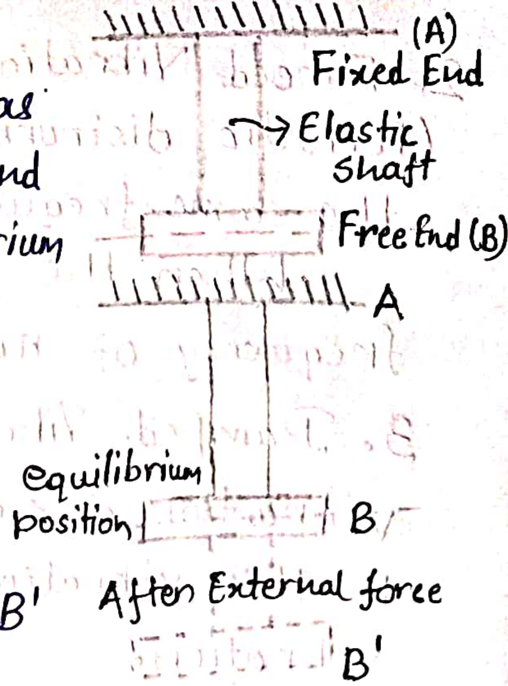
In order to protect the shaft, we will used an another mass (Balanced mass), which is connected on the shaft in the same plane. We will find the magnitude and position of the balance mass.

Qn. A shaft is rotating at a uniform angular speed. Four masses m_1, m_2, m_3 & m_4 of magnitude 300 kg, 450 kg, 360 kg, 390 kg respectively are attached rigidly to the shaft. The masses are rotating in the same plane. The corresponding radius of rotations are 200 mm, 150 mm, 250 mm and 300 mm. The angle made by these masses with horizontal are 0°, 45°, 120° & 255° respectively.

Find (i) Magnitude of balancing mass.

(ii) Position of balancing mass if its radius of rotation is 200 mm. [TMM - UNIT 5 - PDF - 2]

Vibrations ; When the elastic body (Such as spring, shaft, beam) which is fixed at one end is displaced at other end from its equilibrium position by the application of an external force, the elastic body will show vibratory motion after releasing the other end.



Suppose the body (elastic) is displaced by some external applied force. The body end 'B' reaches at 'B''.

Now the elastic body is stretched that means, some work done is completed on the elastic body. The work is stored as strain energy.

Now, when the external applied force is removed and the body released, the body tends to move at equilibrium position. At this stage whole of the strain energy is converted into kinetic energy at the equilibrium position.

At equilibrium position, the body again moves in the opposite direction due to the maximum kinetic energy. Again the whole kinetic energy is converted into stored strain energy, the body again returns to equilibrium position. Due to repeation of storeing strain energy, kinetic energy and movement of elastic body about the equilibrium position, vibrations are induced in the body.

Types of Vibration ;

1. Free Vibration: In thés vibration, no any further external forces are applied, after giving the initial displacement of the body. Such type of virbration is called Free or Natural vibration. Noted that, frictional and other are neglected. The freqancy of free vibration is called Free or natural frequency.

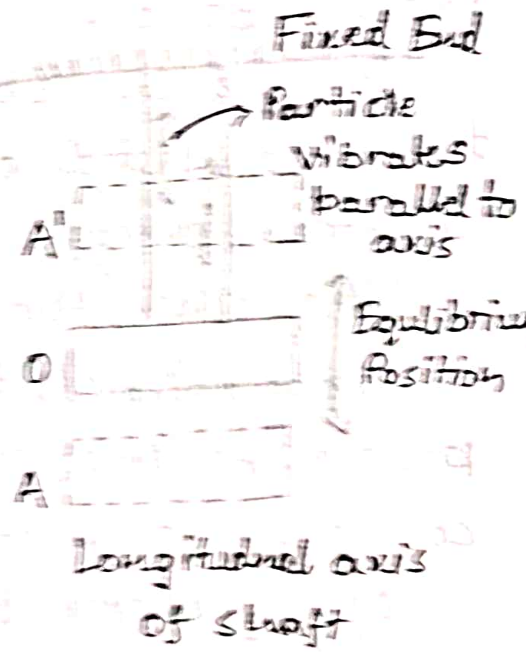
2. Forced Vibration; In this vibration, the external force (Periodic distrurbing force) is applied to the body. It have the same frequency of the applied force frequency.

When the frequency of applied force is same to the frequency of natural vibration, Resonense take place.

3. Damped Vibration; In this vibration, amplitude of vibration decreases over every cycle of vibration motion. In this vibration, a damer is used, which absorbs the vibration.

Types of Vibration;

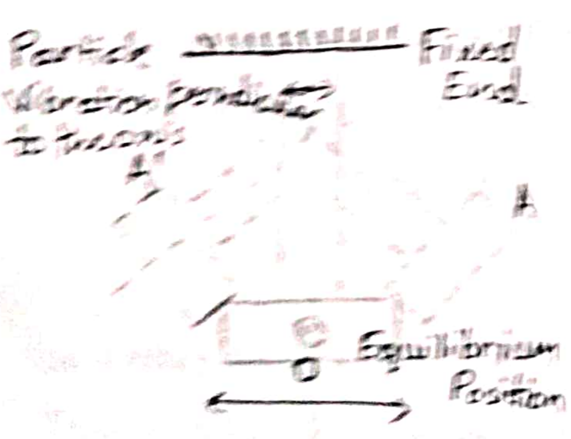
1. Longitudinal Vibrations:



When the particle of shaft vibrates parallel to the longitudinal axis of shaft, then the vibration is called Longitunal vibration.

Noted that, the shaft is subjected to Tension and Compression alternately, resulting Tensile and compressive stresses.

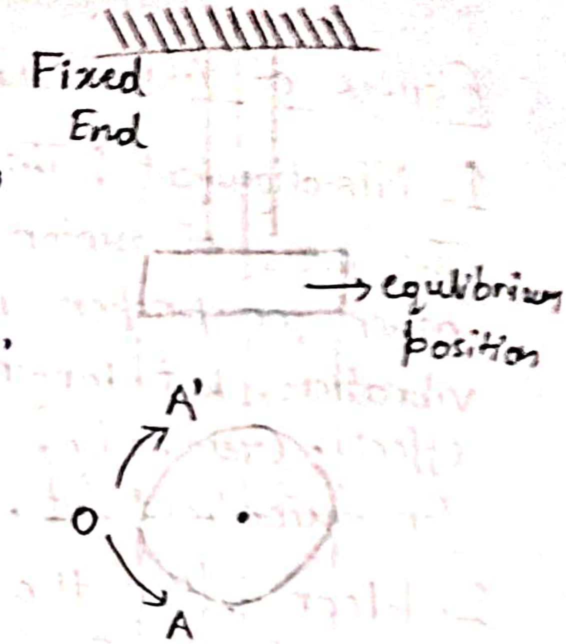
2. ~~Transverse~~ Transverse Vibration



When the particle of shaft vibrates perpendicular to the axis of the shaft, then the vibration is called transverse vibration.

Noted that, the shaft gets straight and Bent alternatly resulting bending stress.

3. Torsional Vibration:



When the particle of shaft vibrates in a circle about the axis of shaft, then the vibration is called torsional vibration. Noted that, the shaft is twisted and unwisted alternatly resulting shear stress.

Terms used in Vibration Motion;

Period of Vibration or Time Period: It is the time after which the motion is repeated itself. The time period is expressed in seconds.

Cycle; It is the motion completed during one time perid.

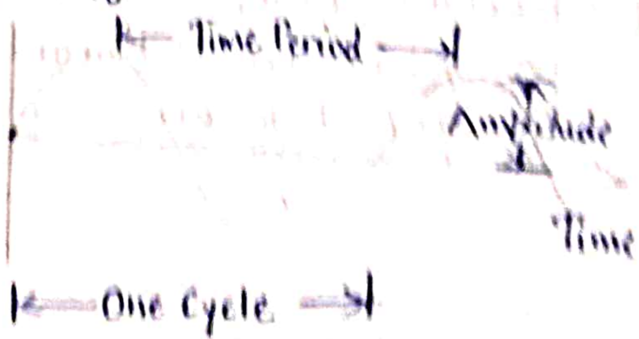
Frequency; It is the number of cycles in one second. In SI unit, the frequency is expressed in "Hertz" (Hz).

$$1\ Hz = 1\ cycle/sec.$$

Amplitude; It is the maximum displacement from the mean position of equilibrium position. It expresed in "milimeters".

Periodic Motion; A motion which repeats itself after the ~~equilibrium position~~ equal intervals of time is called periodic motion.

Degree of freedom; It is the minimum number of factors which are indipendent to each other and defines the complete motion of body.



Causes of Vibration:

1. Mis-alignment : This is very common cause for vibration. The axis of motor and shaft must be parallel to each other for proper running. Small misalignment causes minor vibration, but larger misalignment causes major vibrational effect. Generally, the Thermal expansion is a big factor for misalignment.

2. Wear : After the wear of machine components, the machine can vibrates. Damaged area of roller bearing or gear teeth can cause large vibration.

3. Loose Connections : Loose bearings, loose bolts can causes vibrations.

4. Unbalanced Force in Machine Parts : The rotating or reciprocating unbalanced force in mache component can causes heavy vibrations.

5. External Load : External Load on machine can causes vibrations.

6. Lack of Lubrication; Due to the improper lubrication, the machine part gets friction and wear resulting vibrations.

7. Uses of Spur Gear : ~~Due to the improper lubrication~~ Uses of Simple spur gears generates high vibration as compare to Helical Gears.

8. Improper Foundation of Machine : Improper installation of the machine can cause vibration.

9. Use of poor material : Due to lack of rigidity of machine component there will be ~~various~~ vibrations.

10. Excessive Clearance : Due to the excessive clearance or gap between the mating parts, there will be vibrations.

Harmful Effects On Vibrations;

1. Damaging Machine: The critical vibrations can destroy or damage the machine components.
2. Excessive Stress: Due to the vibrations, there will be stress and strain in machine components.
3. Reduction in Life: Vibration reduces the capacity of machine component.
4. Fatigue Failure: Vibration is responsible for the fatigue failure of machine component, without any warning.
5. Discomfort to humans: The workers who work in an industry get discomfort.
6. Damage of Surface: The surface or foundation of machine can get destroy due to the vibration in machine components.
7. Low Production: Due to the vibration, the production of industry becomes poor, because the product has not of good quality.

Remidies of vibrations;

1. Proper Balancing: We must take proper connection/concentration on the unbalanced rotating or reciprocating masses within the machine.
2. Selection of Gears: We should use the Helical gears in power transmission. The use of helical gear in the place of spur gear can reduce vibration as well as noice.
3. Allignment accuracy: We should check the allignment of driving and driven shaft regularty.
4. High Strength Components: The material used for the components of machine should have high strength. That means, the machine component can withstand with the excessive stresses.

5. Uses of High strength Helical Spring: The heavy machines should have helical spring to absorb the vibrations.

6. Proper Foundation: The base foundation, upon which the heavy machine is to be installed should be rigid.

7. Super vision of connections: All the connection points or joints must be tightly fixed.

8. Monitoring of proper Lubrication; Afte a specific interval of time, the machine components should have proper Lubrication.

UNIT-5 Numerical

Qn. A distibuting mass of 600 kg is attached to a shaft. The shaft is rotating with 'ω' rad/s. The C.G. of disturbing mass from the axis of rotation is 270 mm. The disturbing mass is balanced by masses in two different plans. The C.G. of the balancing mass from the axis of rotation is 450 mm each. The balancing masses is 1.5 m (Distance between the balancing masses and distributing is 300 mm.

Determine,

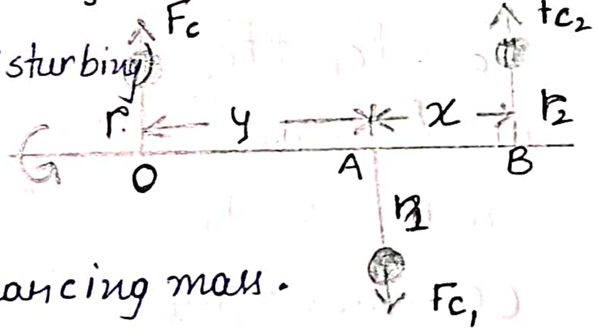
(i) Find the positive of other plane.

(ii) Magnitude of balancing masses;

a). Planes of balancing masses are on the same side of the plane of the distrbing mass.

b). Planes of balancing masses are on either side of the planes of the disturbing mass.

Case-I



m = Mass of unbalanced (disturbing) mass

r = Ratius of rotation of mass.

m_1, m_2 = Masses of Balancing mass.

r_1, r_2 = Radius of rotation of masses m_1, m_2.

from the question,

$$m = 600\ kg,\ r = 270mm = 0.27m$$

$$r_1 = r_2 = 450\ mm = 0.45m$$

$$x = 1.5m,\ y = 300mm = 0.3m$$

Now for static balancing,

$$F_c + F_{c_2} = F_{c_1}$$

$$\Rightarrow mr\omega^2 + m_2 r_2 \omega^2 = m_1 r_1 \omega^2$$

$$\Rightarrow mr + m_2 r_2 = m_1 r_1$$

$$= 600 \times 0.27 + m_2 \times 0.45 = 0.45 \times m_1$$

$\Rightarrow m_1 = m_2 + 360 \quad -①$

Now for dynamic balancing take moment about 'A',

$F_c \times y + F_{c_1} \times 0 = F_{c_2} \times x$

$\Rightarrow m r \omega^2 \times y = m_2 r_2 \omega^2 \times x$

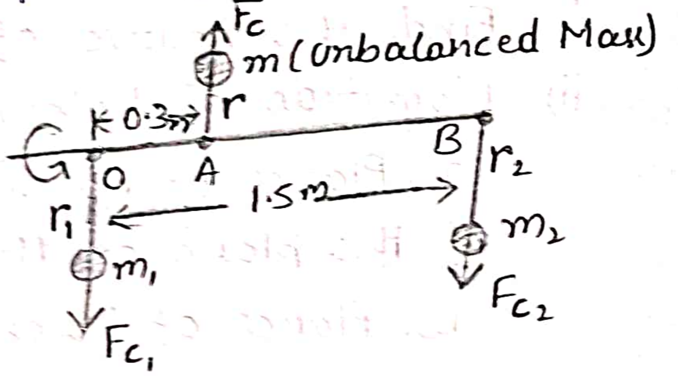
$\Rightarrow 600 \times 0.27 \times 0.3 = m_2 \times 0.45 \times 1.5$

$\Rightarrow m_2 = 72 \text{ kg}$

$\therefore m_1 = 432 \text{ kg}$

Also distance between the plane of 'm' and 'm₂' will be

$y + x = 0.3 + 1.5 = 1.8 m$



Case - II

$m = 600 \text{ kg}, \; r = 0.27 m$

$r_1 = r_2 = 0.45 m$

Now for the static balancing

$F_c = F_{c_1} + F_{c_2}$

$m r \omega^2 = m_1 r_1 \omega^2 + m_2 r_2 \omega^2$

$m r = m_1 r_1 + m_2 r_2$

$600 \times 0.27 = m_1 \cdot 0.45 + m_2 \cdot 0.45$

$m_1 + m_2 = \frac{600 \times 0.27}{0.45} = 360 \quad -②$

Now for dynamic balancing, take moment about O

$F_{c_1} \times 0 + F_{c_2} \times OB = F_c \times OA$

$\Rightarrow m_2 r_2 \omega^2 \times OB = m r \omega^2 OA$

$\Rightarrow m_2 \times 0.45 \times 1.5 = 600 \times 0.27 \times 0.3$

$\Rightarrow m_2 = \frac{600 \times 0.27 \times 0.3}{0.45 \times 1.5} = 72 \text{ kg}$

$\therefore m_1 = 360 - 72 = 288 \text{ kg}$

Also distance between the planes of 'm' & 'm₂'; $1.5 - 0.3 = 1.2 m.$

Qn. A shaft is rotating at a uniform angular speed. four masses m_1, m_2, m_3 and m_4 of magnitude 300 kg, 450 kg, 360 kg, 390 kg respectively are attached rigidly to the shaft. The masses are rotating in the same plane. The corresponding radius of rotations are 200 mm, 150 mm, 250 mm and 300 mm. The angle made by these masses with horizontal are 0°, 45°, 120° and 255° respectively. Find (i) Magnitude of balancing mass.

(ii) Position of balancing mass if its radius of rotation is 200 mm.

Graphical Method

Given, $m_1 = 300$ kg, $m_2 = 450$ kg, $m_3 = 360$ kg, $m_4 = 390$ kg.

$r_1 = 200\ mm = 0.2\ m$

$r_2 = 150\ mm = 0.15\ m$

$r_3 = 250\ mm = 0.25\ m$

$r_4 = 300\ mm = 0.3\ m$

Now, calculate centrifugal forces ; $F_c = mr\omega^2$, it is clear that 'ω^2' is same and common for all the masses (because all the masses rotate on the same shaft), hence there is no need to write 'ω^2' we will concentrating on only (mr).

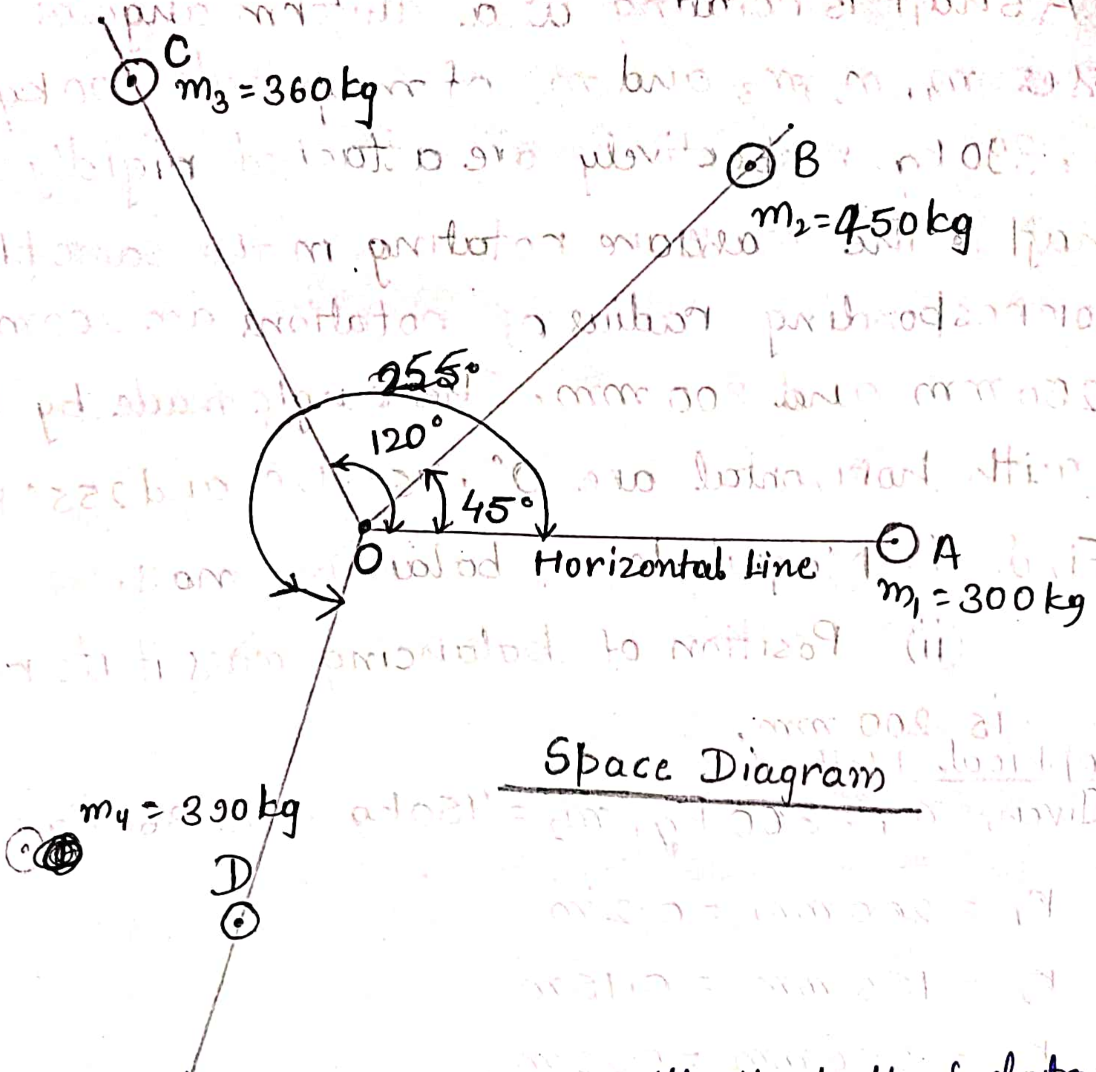
Now, $m_1 r_1 = 300 \times 0.2 = 60$ kg-m

$m_2 r_2 = 450 \times 0.15 = 67.5$ kg-m

$m_3 r_3 = 360 \times 0.25 = 90$ kg-m

$m_4 r_4 = 390 \times 0.3 = 117$ kg-m

Now firstly we draw the space diagram, ie we will represent the position of all the masses.



Space Diagram

Now, we will draw force polygon with the help of datae, $m_1 r_1$, $m_2 r_2$, $m_3 r_3$ and $m_4 r_4$. Take a suitable scale for the drawing of force poygon.

Suppose $60\,kg\text{-}m = 3\,cm$

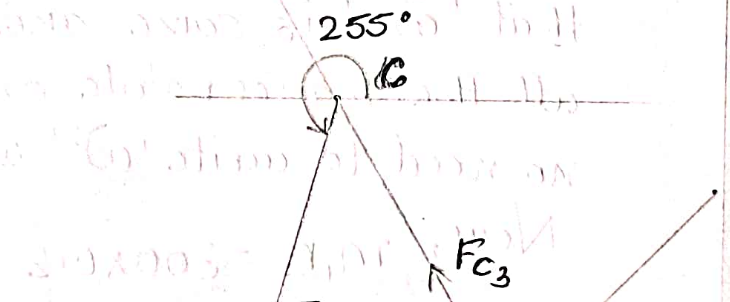
$\therefore 1\,kg\text{-}m = 0.05\,m$

$\therefore 67.5\,kg\text{-}m = 3.375\,m$

$90\,kg\text{-}m = 4.5\,cm$

$117\,kg\text{-}m = 5.85\,cm$

Firstly draw the horizontal magnitude i.e $m_1 r_1$ after this draw other magnitude with refeence $m_1 r_1$ with given angle



$oa = 3\,cm$

$ab = 3.375\,cm$

$bc = 4.5\,cm$

$cd = 5.85\,cm$

Noted that, force polygon is not completed (not closed). hence the system is not balanced (unbalanced). In order to complete the system balance, we will close the polygon. Thus join the last vector 'od'.

The closing side of the polygon 'od' represents the resultant force.

Now, measure 'od'.

By measurement,

$$od = 2{\cdot}74\,cm$$

$\because$ $0.05\,cm = 1$ kg-m

$\therefore$ $2{\cdot}74\,cm = 34.8$ kg-m

Suppose mass of balancing mass is 'm' with radius of rotation 'r'.

From the que, $r = 200\,mm = 0.2\,m$

$\therefore$ $mr = m \times 0.2\,m = 34.8$ kg-m

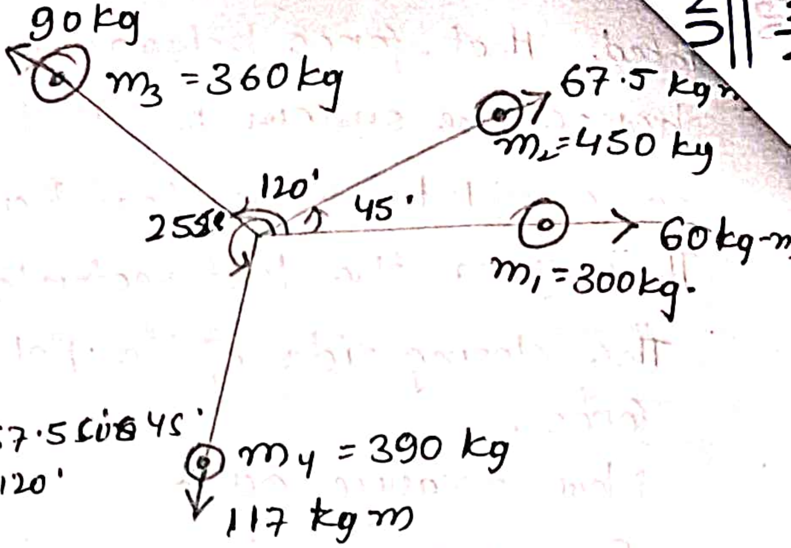
$m = 174$ kg, magnitude of balancing mass,

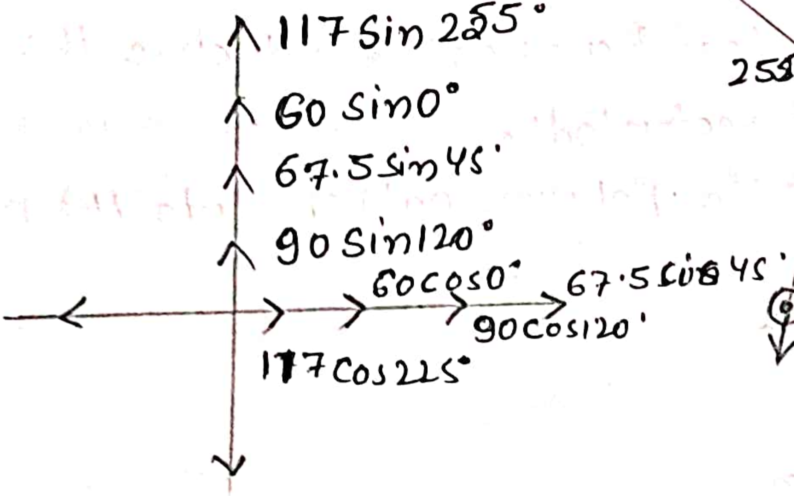
Now, we will find the position of $m = 174$ kg.

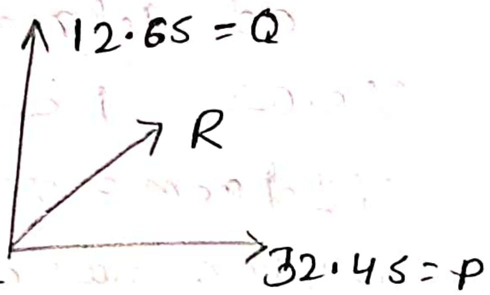
Measure the angle made by 'F_c' with x-axis. By the measurement, '$201°$'.

Thus for complete balancing of system, we will add a mass of $m = 174$ kg at an angle of $201°$ with x-axis.

ANALYTICAL METHOD







'R' Resultant, $= \sqrt{P^2 + Q^2}$

$= \sqrt{32.45^2 + 12.65^2}$

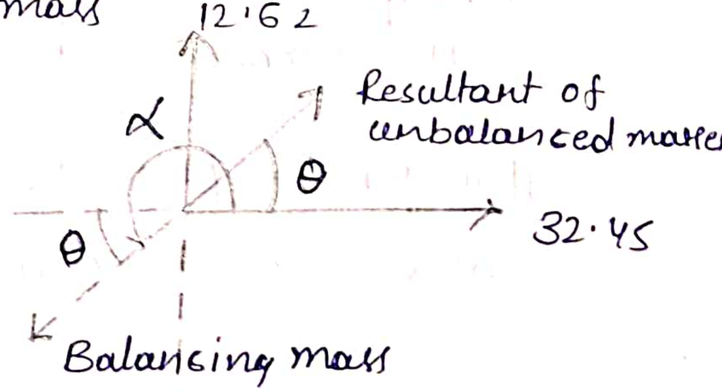
$= 34.8$ (resultant of unbalanced mass)

Now, complete balancing, we will add a balancing mass (m) at a radius of $r = 0.2$ m

$\therefore R = 34.8 = m \times 0.2$

$m = 174$ kg, magnitude of balancing mass

Now, the position of balancing mass will be just opposite to the resultant of unbalance mass



Clearly, $\tan\theta = \frac{12.65}{34.45}$

$\theta = 20.16317°$

Position of balancing mass,

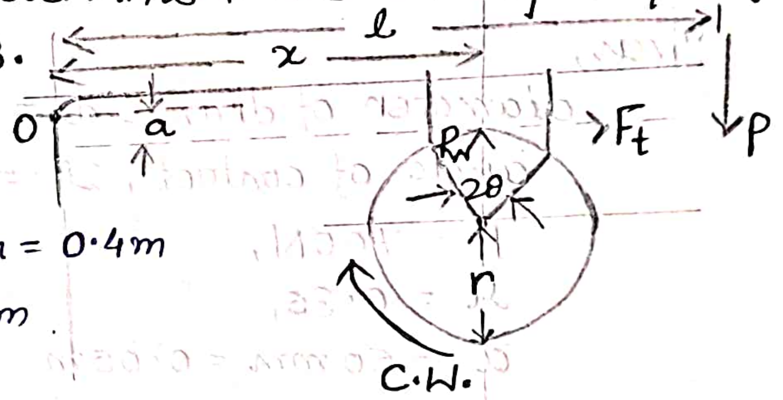
$\alpha = 180° + \theta = 200.16317$

$\simeq 201°$

UNIT-4 Numerical

Numerical on Simple Shoe Brake

Qn. The brake drum of a single block brake is rotating at 500 rpm in C.W. direction. The diameter of drum is 400 mm and the force required at the end of lever to apply the brake is 300N. Angle of contact is 30°, length of lever is 1m. The line of action of tangential braking force is below a distance of 25 mm and the distance between the fulcrum centre and central line of block is 300 mm. Determine the breaking torque if coeficient of friction is 0.3.



given,

$N = 500\,rpm$

diameter of drum, $d = 400\,mm = 0.4\,m$

radius of drum, $r = 0.2\,m$

$2\theta = 30°$,

$l = 1\,m$

$a = 25\,mm = 0.025\,m$

$x = 300\,mm = 0.3\,m$, $\mu = 0.3$.

$\therefore$ Braking Torque, $T = \dfrac{\mu P l r}{x + \mu a}$

$$= \frac{0.3 \times 300 \times 1 \times 0.2}{0.3 + 0.3 \times 0.025}$$

$$= 58.536\ N\,m.$$

Taking moment of all forces about fulcrum 'O';

$$R_N \times x + F_t \times a = P \times l$$

$$\Rightarrow R_N \times x + R_N \mu \times a = P \times l$$

$$\Rightarrow R_N (x + \mu a) = Pl$$

$$\Rightarrow R_N = \frac{Pl}{x + \mu a} = \frac{300 \times 1}{0.3 + 0.3 \times 0.025} = 975.6098\ N$$

$\therefore$ Braking Torque, $T = F_t \times r$

$$= \mu R_N r$$

$$= 0.3 \times 975.6098 \times 0.2 = 58.536\ N$$

Numerical Based on Pivoted Block

Qn. Diameter of drum is 250mm and angle of contact is 90°. If the operating force of 700 N is applied at the end of lever and co-efficient of friction between drum and brake is 0.35. Determine the torque transmitted. The line of action of tangential braking force passes through a distance of 50mm above the fulcrum of lever. Length of lever is 450mm and the distance between the fulcrum and centre line of block is 200mm. The wheel rotate in C.W. direction.

Given,

diameter of drum, $d = 250\ mm = 0.25\ m$

angle of contact, $2\theta = 90°$, $r = 0.125\ m$

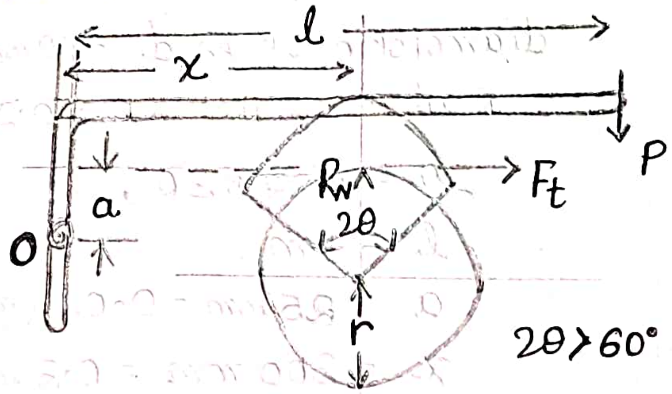
$P = 700N$,

$\mu = 0.35$,

$a = 50\ mm = 0.05\ m$

$l = 0.45\ m$

$x = 200\ mm = 0.2\ m$,



$\because 2\theta = 90°$, $2\theta > 60°$

$\therefore$ Equivalent coefficient of friction,

$$\mu' = \frac{4\mu \sin\theta}{2\theta + \sin 2\theta}$$

$$= \frac{4 \times 0.35 \times \sin 45°}{90° + \sin 90°}$$

$$= \frac{4 \times 0.35 \times 1/\sqrt{2}}{\pi/2 + 1}$$

$$\mu' = 0.3851.$$

Now, taking moments about 'O',

$$P \times l + F_t \times a = R_N \times x$$

$$\Rightarrow P \times l + \mu' R_N a = R_N x$$

$$\Rightarrow R_N = \frac{Pl}{(x - \mu a)} = \frac{700 \times 0.45}{(0.2 - 0.3851 \times 0.05)} = \frac{315}{0.18074} = 1742.78\ N$$

Now Breaking torque,

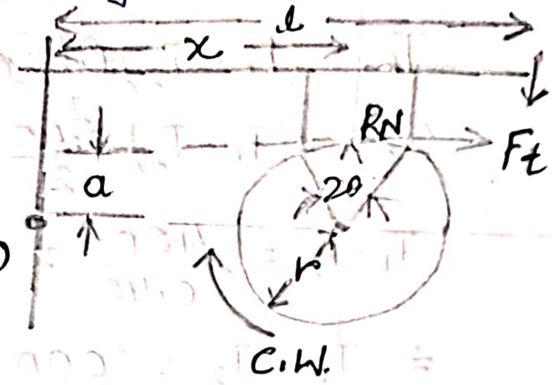
$$T = F_t \times r = \mu' R_N r$$

$$= 0.3851 \times 1742.7868 \times 0.125 = 83.8934\ Nm$$

Numerical on single block brake,

Qn. The diameter of the drum is 180mm and the angle of contact is 60°. If the operating force of 400N is applied at the end of the lever and the coefficient of friction between the lining is 0.3, Determine,

The torque that may be transmitted by the block brake.



given,

Diameter of drum,

$d = 180\ mm = 0.18\ m$

$2\theta = 60°$

$P = 400N$

$\mu = 0.3$

$l = 200 + 250 = 450\ mm = 0.45\ m$

$x = 200\ mm = 0.2\ m$, $a = 60\ mm = 0.06$

∵ Breaking Torque, $T = \dfrac{\mu P l r}{x + \mu a}$

$= \dfrac{0.3 \times 400 \times 0.45 \times 0.9}{0.2 + 0.3 \times 0.06}$

$= \dfrac{48.6}{0.218} = 222.935\ Nm$

Numerical on Band Brake

Qn. A Band brake act on the 3/4th of the circumferance of a drum of 450 mm diameter. The band brake provides a breaking torque of 225 N-m. One end of the band is attached to a fulcrum of lever and the other end is attached at a distance of 100 mm from the fulcrum. Force is applied at a distance of 500mm from the fulcrum. $\mu = 0.25$.

Find applied force for (i) A.C.W. and (ii) C.W

given, from the question, the band brake embraces $\frac{3}{4}$th part of wheel drum.

$\therefore \theta = \frac{3}{4} \times 360° = 270° = 4.7124$ rad.

Now, suppose T_1 = Tight side Tension,

T_2 = Slack side Tension,

$$\Rightarrow \frac{T_1}{T_2} = e^{\mu\theta}$$

$$\Rightarrow T_1 = T_2 e^{0.25\times4.7124}$$

$$T_1 = 3.2482 T_2 \quad -①$$

Now, Breaking Torque

$$T_B = (T_1 - T_2)\frac{\cancel{0.150}}{2}$$

$$225 = (T_1 - T_2)\frac{0.45}{2}$$

$$T_1 - T_2 = \frac{450}{0.45} = 1000$$

$$\Rightarrow T_1 - T_2 = 1000$$

$$\Rightarrow 3.2482 T_2 - T_2 = 1000$$

$$\Rightarrow T_2(2.2482) = 1000$$

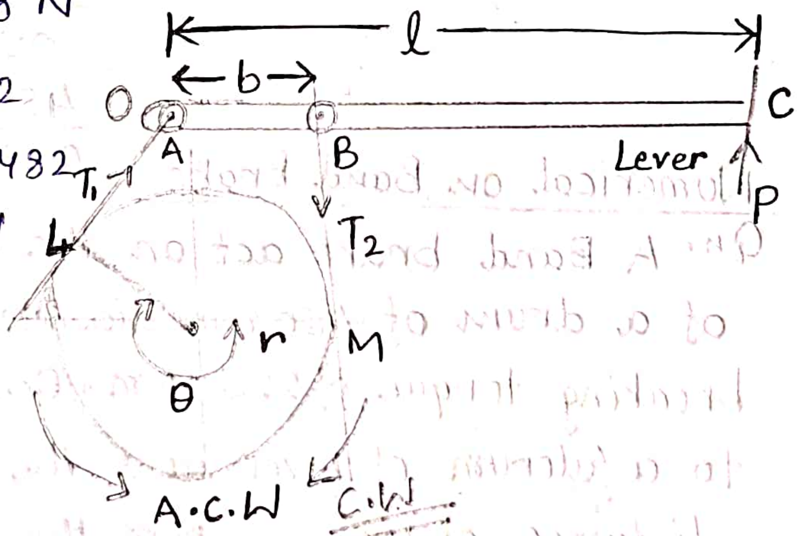
$$\Rightarrow T_2 = \frac{1000}{2.2482} = 444.8$$

$$\therefore T_2 = 444.8 \text{ N}$$

$$\because T_1 = T_2 \times 3.2482$$

$$= 444.8 \times 3.2482$$

$$= 1444.8 \text{ N}$$



① In A.C.W,

$$Pl = T_2 b$$

$$P = \frac{T_2 b}{l}$$

$$= \frac{444.8 \times 0.1}{0.5} = \underline{\underline{88.96\text{N}}}$$

⑪ In C.W.,

$$Pl = T_1 b$$

$$P = \frac{T_1 b}{l} = \frac{1444.8 \times 0.1}{0.5} = \underline{\underline{288.96\text{N}}}$$

Numerical on differential Band Brake;

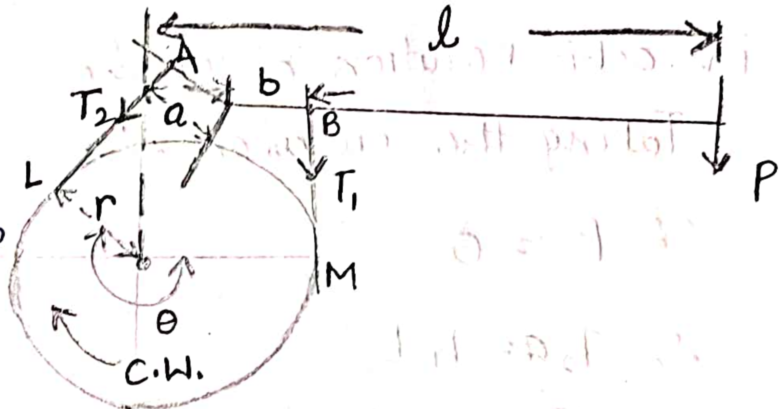
Qn. A differential Band Brake has an angle of contact of 225°. Diameter of drum is 350N-m. Breaking Torque is given as 350 N-m and co-efficient of friction is given as 0.3. The distance between the fulcrum and right end of band is 35mm. Length of lever is 500mm. Find

(i) Applied force for C.W. rotation of wheel.

(ii) Condition for self locking in C.W. rotation.

Given,

$\theta = 225° = 3.93$ rad.



diameter of drum,

$d = 350mm = 0.35m$

Breaking Torque,

$T_B = 350$ N-m

$\mu = 0.3$.

Now, let, T_1 = Tight side Tension

T_2 = Slack side Tension

$$\therefore \frac{T_1}{T_2} = e^{\mu\theta} = e^{0.3 \times 3.93}$$

$$T_1 = 3.2511\, T_2, \quad -(1)$$

$$\because T_B = 350$$

$$\Rightarrow (T_1 - T_2) r = 350$$

$$\Rightarrow (T_1 - T_2) \frac{0.35}{2} = 350$$

$$\Rightarrow T_1 - T_2 = 2000$$

$$\Rightarrow 3.2511 T_2 - T_2 = 2000$$

$$\Rightarrow T_2 = \frac{2000}{3.2511} = 615.176 \text{ N}$$

$$\Rightarrow T_1 = 2615.176 \text{ N}$$

(i) C.W. rotation of wheel,

Taking moment of all the force about fulcrum, 'O';

$T_2 \times a = T_1 \times b + P \times l$ —(2)

$\Rightarrow 615.167 \times 0.15 = 2615.167 \times 0.035 + P \times 0.5$

$\Rightarrow 92.27 = 91.53 + P \times 0.5$

$\Rightarrow P = \frac{0.74}{0.5} = 2.96\,N$

(ii) ~~Apply~~ We have to state condition of self locking in C.W. rotation of wheel,

Taking the equation (2);

$\because P = 0$

$\therefore T_2 a = T_1 b$

$\Rightarrow T_2 OA = T_1 OB$

$\Rightarrow OA = \frac{T_1}{T_2} \times OB$

$= \frac{2615.167}{615.167} \times 0.035$

$= 0.1487\,m$

$= 149\,mm$

Numerical Based on Band and Block Brake

Qn. Band and block brake have 12 blocks, each of which subtends an angle of 18° at the drum centre is applied to a rotating drum of diameter 800 mm. The blocks are 100 mm thick. The drum and wheel mounted on the same shaft have a mass of 1600 kg and have a combined radius of gyration of 500 mm. The two ends of the band are attached to the pins on the opposite sides of the brake fulcrum at a distance of 35 mm and 140 mm from the fulcrum. The coefficient of friction between the block and drum is taken as 0.3. A force 150 N is applied to the break.

Find (i) The maximum breaking torque,

(ii) The angular retardation of break drum,

(iii) The time taken by the system to come to rest from the rated speed of 240 rpm

Wheel rotates with C.W. rotation.

Given, from the question,

n = Number of Blocks = 12

$2\theta = 18°$, $\theta = 9°$

diameter of drum, $d = 800\,mm = 0.8\,m$

thickness of drum, $t = 100\,mm = 0.1\,m$

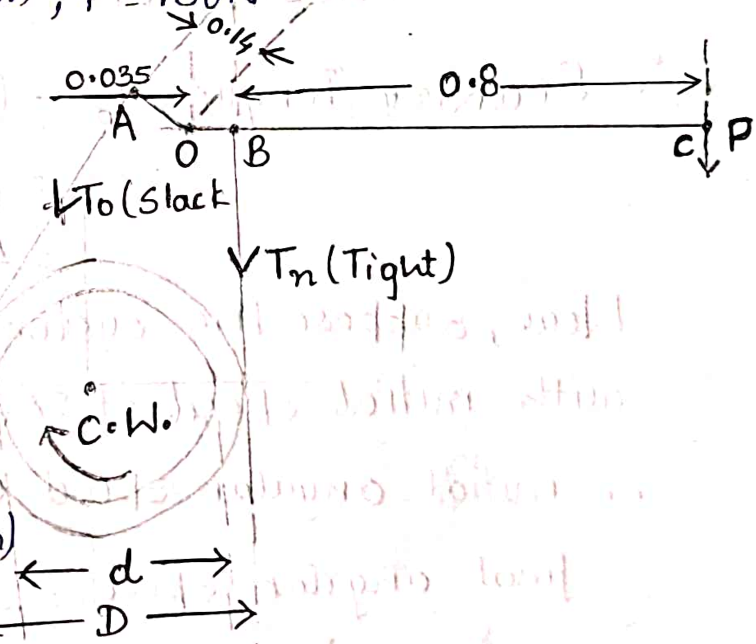
total mass, $m = 1600\,kg$

radius of gyration, $k = 500\,mm = 0.5\,m$

$AO = 140\,mm = 0.14\,m$, $\mu = 0.3$,

$OB = 35\,mm = 0.035\,m$, $P = 150\,N$

$OC = 800\,mm = 0.8\,m$



Diameter of Band;

$D = d + 2t$

$= 0.8 + 2 \times 0.1$

$= 1\,m$

Radius of Band, $r = 0.5\,m$

Now, we know the reaction between the tight tension (T_n) and slack Tension (T_0).

$$\frac{T_n}{T_0} = \left(\frac{1 + \mu \tan\theta}{1 - \mu \tan\theta}\right)^n$$

$$\frac{T_n}{T_0} = \left(\frac{1 + 0.3 \times \tan 9°}{1 - 0.3 \times \tan 9°}\right)^{12}$$

$$\frac{T_n}{T_0} = 3.13$$

$$\Rightarrow T_n = 3.13\, T_0 \quad -(1)$$

Now, taking the moments of all the forces about fulcrum 'O',

$$P \times OC + T_n \times OB = T_o AO$$

$$\Rightarrow 150 \times 0.8 + T_n \times 0.035 = 0.14\, T_o$$

$$\Rightarrow T_o = \frac{120}{0.14} + \frac{0.035}{0.14} T_n$$

$$\Rightarrow T_o = 857.1429 + 0.25\, T_n \quad -②$$

$$\Rightarrow T_o = 857.1429 + 0.25 \times 3.13\, T_o$$

$$\Rightarrow T_o (1 - 0.7825) = 857.14$$

$$\Rightarrow T_o = \frac{857.14}{0.2175} = 3940.87\,N \quad \text{[Slack side]}$$

$$\Rightarrow T_n = 3.13\, T_o$$
$$= 3.13 \times 3940.87 \quad \text{[Tight side]}$$
$$= 12334.9231\,N$$

$\therefore$ Braking Torque, $T_B = (T_n - T_o)\, r$

$$= (12334.9231 - 3940.87)\, 0.5$$
$$= 4197.02\ Nm$$

Now, suppose the system comes in rest in time 't' sec, with initial speed of 240 rpm

$\therefore$ initial angular speed, $\omega_i = \frac{2\pi \times 240}{60} = 8\pi = 25.133\ rad/s$.

final angular speed, $\omega_f = 0$ (because system stops)

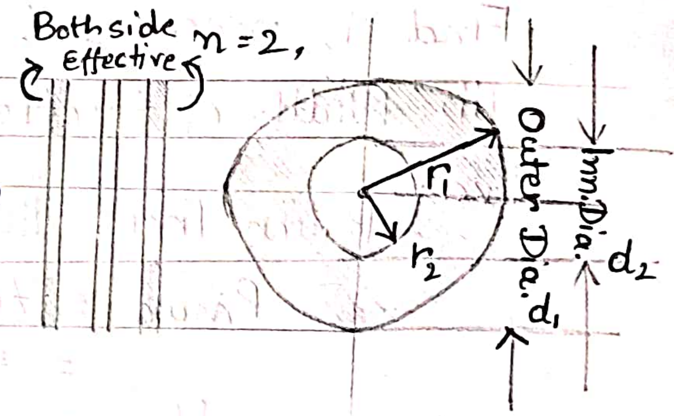
$$\omega_f = \omega_i - \alpha t$$

$$0 = 25.133 - 10.49t$$

$$t = \frac{25.133}{10.49} = 2.396 \text{ Seconds}$$

Numerical on simple plate clutch;

Q.A single plate clutch, with both sides effective has outer and inner diameter of 300 mm and 200mm respectively, The maximum intensity of pressure at any point in contact surface is not to exceed $0.1\,N/mm^2$. Coefficient of friction is 0.3. Determine the power transmitted by clutch at a speed of 2500 rpm.



given,

$$d_1 = 300\ mm = 300\ mm = 0.3\,m$$

$$r_1 = 0.15\,m$$

$$d_2 = 200\ mm = 0.2\,m$$

$$r_2 = 0.1\,m$$

$\mu = 0.3$, $N = 2500$ rpm

$$\therefore \omega = \frac{2\pi N}{60} = \frac{2\pi 2500}{60} = 261.799\ rad/s.$$

Now from the question, maximum pressure intensity is $0.1\,N/mm^2$

$$P_{max} = 0.1 \times 10^6\ N/m^2$$

and at inner ~~diam~~ radius (r_2) there will be max. pressure (P_{max})

Now, applying 'uniform wear Theory',

$$Pr = \text{Constant}$$

$$P_{max}\, r_2 = C$$

$$\Rightarrow C = 0.1 \times 10^6 \times 0.1$$

$$= 10^4\ N/m$$

Now, axial thrust,

$$W = 2\pi C (r_1 - r_2)$$

$$= 2\pi\, 10^4 (0.15 - 0.1)$$

$$\therefore W = 3141.593\ N$$

Now, torque transmitted,

$$T = \frac{1}{2}\mu W (r_1 + r_2) \times n$$

$$= \frac{2}{2}\, 0.3 \times 3141.593 (0.15 - 0.1) = 235.619\ N\text{-}m$$

Now, Power Transmitted,

$$P = T \times \omega$$

$$= 235.6194 \times 261.799$$

$$P = 61684.94\ Nm/s$$

$$= 61.68\ K\,Watt,$$

Qn. A single plate clutch is required to transmit 8 kW at 1000 rpm. The axial pressure is limited to 70 kN/m². The mean radius of plate is 4.5 times the radial width of friction surface. If the both sides of platee are effective and coefficient of friction is 0.25. Find (i) inner and outer radius of plate and mean radius. (ii) Width of friction lining.

given, Power Transmitted, $P = 8\text{ kW} = 8000\text{ W}$

$$P_{max} = 70\text{ kN/m}^2$$
$$= 70000\text{ N/m}^2$$
$$\mu = 0.25$$

r_1 = Outer radius

r_2 = inner radius,

$$\text{Mean Radius} = \frac{r_1 + r_2}{2}$$

$$\therefore \quad 4.5(r_1 - r_2) = \frac{r_1 + r_2}{2}$$

$$(r_1 + r_2) = 9r_1 - 9r_2$$

$$\Rightarrow \quad 8r_1 = 10r_2$$

$$\Rightarrow \quad r_1 = 1.25\,r_2 \quad \text{— (I)}$$

$$\therefore \quad P_{max} \times r_2 = C \quad \text{— (II)}$$

Now, Axial load; $W = 2\pi C(r_1 - r_2)$

$$= 2\pi\, P_{max} \times r_2\,(1.25r_2 - r_2)$$

$$\Rightarrow 2\pi \times 7000 \times r_2^2 \times 0.25$$

$$W = 109955.7429\, r_2^2.$$

Now, Torque transmitted,

$$T = \frac{1}{2} \mu W (r_1 + r_2) n$$

$$= \frac{1}{2} 0.25 \times 109955.7429 \, r_2^2 \times (1.25 r_2 + r_2) \times 2$$

$$\therefore T = 61850.10537 \, r_2^3$$

Now, Power Transmitted,

$$P = T \times \omega$$

$$8000 = 61850.10537 \, r_2^3 \times \frac{2\pi 1000}{60}$$

$$r_2^3 = 0.0012352 \, m$$

$r_2 = 0.1073 \, m = 107.3 \, mm$ = Inner radius,

$$r_1 = 1.25 r_2$$

$$= 1.25 \times 0.1073$$

$= 0.1341 \, m = 134.1 mm$ = Outer radius

$\therefore$ Width of friction lining,

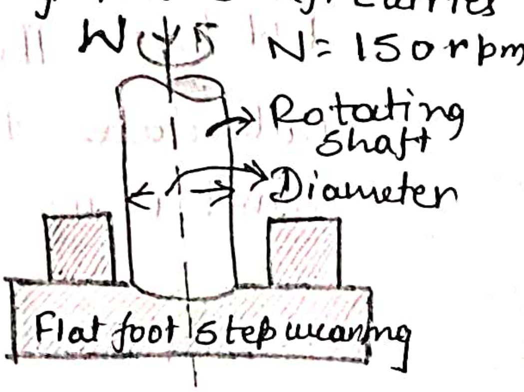
$$r_1 - r_2 = 134.1 - 107.3$$

$$= 26.8 \, mm$$

Numerical on simple pivot Bearing,

Qu. Estimate the power lost in friction for (i) Uniform pressure and (ii) Uniform wear.

A vertical shaft of 100 mm diameter rotating at 1500 rpm is a flat foot - step wearing. The shaft carries a load of 15 KN. $\mu = 0.05$.



Given, $N = 150$ rpm

$D = 100 \, mm = 0.1 \, m$

$r = 0.05 \, m$

$W = 15 \, KN = 15000 N$

$\mu = 0.05$

(i) Uniform Pressure Theory,

Total frictional torque,

$$T = \frac{2}{3} \mu W R$$
$$= \frac{2}{3} \times 0.05 \times 15000 \times 0.05$$
$$= 25 \text{ Nm}.$$

Now, Power lost in friction, $P = T\omega$

$$= T \times \frac{2\pi N}{60}$$
$$= \frac{25 \times 2\pi 150}{60}$$

$$P = 392.699 \text{ Watts}$$

(ii) Uniform Wear Theory,

Total frictional torque,

$$T = \frac{1}{2} \mu W R$$
$$T = 18.75 \text{ N-m}$$

Now, Power lost in friction, $P = T\omega$

$$= \frac{18.75 \times 2\pi 150}{60}$$
$$= \frac{17671.45868}{60} = 294.524 \text{ watt}$$

Numerical on Collar Bearing

Qu. In a thrust bearing external and internal diameter are 320 mm and 200 mm. Total axial load is 80 kN and intensity of pressure is 350 kN/m². The shaft rotates at 400 rpm. $\mu = 0.06$. Calculate the power lost in friction and total number of collar required.

given,

External diameter, $d_1 = 320\ mm = 0.32\ m$

$\therefore$ External radius, $R_1 = 0.16\ m$

Internal diameter, $d_2 = 200\ mm = 0.2\ m$

$\therefore$ internal radius $r_2 = 0.1\ m$

$W = 80000\ N$, $P = 350\,000\ N/m^2$, $N = 400\ rpm$, $\mu = 0.06$,

Assuming uniform pressure theory,

Total frictional torque, $T = \frac{2}{3}\mu W\left(\frac{R_1^3 - R_2^3}{R_1^2 - R_2^2}\right)$

$$= \frac{2}{3} \times 0.06 \times 80000 \times \frac{0.16^3 - 0.1^3}{0.16^2 - 0.1^2}$$

$$\therefore T = 635.12\ N\text{-}m$$

Now, Power Lost, $P = T\omega$

$$= 635.12 \times \frac{2\pi\, 400}{60}$$

$$\therefore P = 26602.03995\ watt$$

Now, Suppose that, there are 'n' number of collars are used. Clearly all the collars (n) will distributed the pressure equally and uniformly.

$\therefore$ Load per collar $\times\, n$ = total load

$$P \times \pi(R_1^2 - R_2^2)n = W$$

$$n = \frac{W}{P\pi(R_1^2 - R_2^2)}$$

$$= \frac{80000}{350\times10^3 \times \pi \times (0.16^2 - 0.1^2)}$$

$$\Rightarrow \frac{80000}{17153.09} = 4.66$$

$\therefore n = 5$ collars are used.

Qn. A conical pivot with angle of cone as 120°, supports a verticle shaft of diameter 300 mm. It is subjected to a lead of 20 kN. Co-efficient of friction is 0.05. Speed of shaft is 210 rpm. Calculate the power lost.
(i) Uniform pressure (ii) Uniform wear.

Given, Cance Angle, $2\alpha = 120°$

$\therefore$ Semi-Conce angle, $\alpha = 60°$

Diameter, $D = 0.3\,m$, $\mu = 0.05$.

$N = 210$ rpm, $W = 20000\,N$

(i) Uniform pressure theory:-

$$T = \frac{2}{3}\mu W R \times \text{Cosec}\,\alpha$$

$$= \frac{2}{3} \times 0.05 \times 20000 \times 0.15 \times \text{cosec}\,60°$$

$$= \underline{115.47\ Nm}$$

Power lost in friction,

$$P = T\omega$$

$$= 115.47 \times \frac{2\pi \cancel{210}^{7}}{\cancel{60}_{2}}$$

$$= 2539.32\ Nm/s$$

$$= \underline{2.53932\ \text{watts}}$$

(ii) Uniform Wear Theory,

$$T = \frac{1}{2}\mu W R \times \text{Cosec}\,\alpha$$

$$= \frac{1}{2} \times 0.05 \times 20000 \times 0.15 \times \text{Cosec}\,60°$$

$$= \underline{86.60\ Nm}$$

Power lost in friction,

$$P = T\omega$$

$$= 86.60 \times \frac{2\pi \cancel{210}^{7}}{\cancel{60}}$$

$$= 1904.48\ Nm/s$$

$$= \underline{1.90448\ \text{Watts}}$$

Numerical on conical collar Bearing

Qn. A conical pivot with angle of cone as 100° supports a load of 18 kN. The external radius is 2.5 times the internal radius. The shaft rotates at 150 rpm. If the intensity of pressure is 300 kN/m² and $\mu = 0.05$, then find the power Lost.

Given,

$2\alpha = 100°$

$\therefore$ Semi cone Angle, $\alpha = 50°$

Load, $W = 18000\,N$

External radius $= R_1$

Internal radius $= R_2$

from question,

$$R_1 = 2.5 R_2$$

$N = 150$ rpm, Pressure, $P = 300{,}000\ N/m^2$, $\mu = 0.05$

Applying uniform pressure theory,

$$P = \frac{W}{\pi(R_1^2 - R_2^2)}$$

$$300{,}000 = \frac{18{,}000}{\pi(\cancel{2}6.25R_2^2 - R_2^2)} = \frac{18000}{1.5\pi R_2^2}$$

$$R_2^2 = \frac{18000}{1.5\pi \times 30000}$$

$$R_2 = 0.060314\,m$$

$$\therefore R_1 = 2.5 \times R_2 = 0.150786\,m$$

Now, frictional torque, $T = \frac{2}{3}\mu W \left(\frac{R_1^3 - R_2^3}{R_1^2 - R_2^2}\right) \text{Cosec}\,\alpha$

$$T = \frac{2}{3} \times 0.05 \times 18000 \times \left(\frac{0.1507^3 - 0.0603^3}{0.1507^2 - 0.0603^2}\right) \text{Cosec}\,50$$

$$T = 131.5995\ Nm$$

Now, Power Lost, $P = T\omega = 131.5995 \times \frac{2\pi \cancel{150}\,5}{\cancel{60}}$

$$= 131.5995 \times 5\pi = 2067.16 \text{ watts}$$

UNIT-3 Numerical

Qn The mass of flywheel of an engine is 6.5 tonne, the radius of gyration is 1.8 m. If the maximum fluction of energy is 56 kN-m and the mean speed of the engine is 120 rpm. Find.

(i) Maximum and minimum speed

(ii) Co-efficient of fluctuation of speed

(iii) Mean kinetic energy of flywheel.

Given, Mass of flywheel, $m = 6.5$ tonne

$= 6.5 \times 10^3$ kg

$\therefore m = 6500$ kg

Radius of gyration, $k = 1.8$m

Maximum fluctuation of enenerg,

$\Delta E = 56$ kN-m

$\therefore \Delta E = 56 \times 10^3$ N-m

Mean speed, $N = 120$rpm

Now, $\because \Delta E = I\omega^2 C_s$ -(1) I = mass moment of inertia of flywheel,

$= mk^2$

$\because I = 6500 \times 1.8^2$

$= 21060$ kg-m²

Now, ω = mean angular velocity

$= \dfrac{2\pi N}{60} = 4\pi$

$\omega = 12.566$ rad/s

$\because 56 \times 10^3 = 21060 \times 12.566^2 \times C_s$

$$C_s = \frac{56 \times 10^3}{21060 \times 12.566^2} = 0.01684.$$

$\because \; C_s = \dfrac{N_1 - N_2}{N}$

where,
N_1 = Max. Speed
N_2 = Min. Speed
N = Mean speed

$$0.01684 = \frac{N_1 - N_2}{120}$$

$$N_1 - N_2 = 2.021 \text{ rpm} \quad -(2)$$

$$\because \; N = \frac{N_1 + N_2}{2}$$

$$120 = \frac{N_1 + N_2}{2}$$

$$\Rightarrow N_1 + N_2 = 240$$

$$\Rightarrow N_1 + N_1 = 240 + 2.021$$

$$\Rightarrow 2N_1 = 242.021$$

$$\Rightarrow N_1 = \frac{242.021}{2} = 121.0105 \text{ rpm} \cong 121 \text{ rpm}$$

$$\therefore N_2 = 118.9895 \text{ rpm} \cong 119 \text{ rpm}$$

Now, Mean K.E of flywheel,

$$E = \frac{1}{2} I \omega^2$$

$$= \frac{1}{2}\, 21060 \times 12.566^2$$

$$= 1662732.869 \text{ J}$$

$$= 1662.733 \text{ kJ}$$

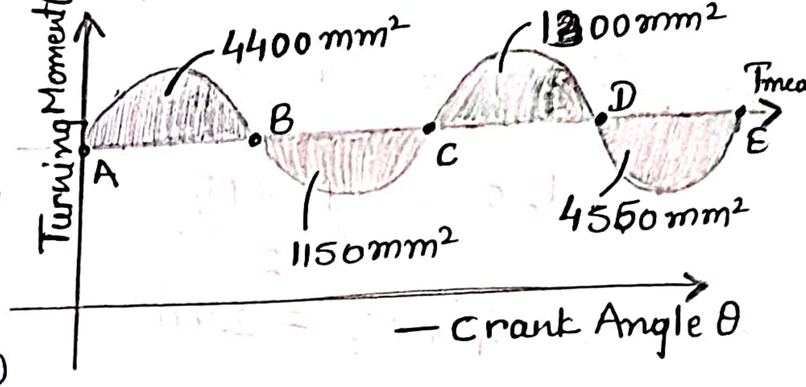
Qn. In a turning moment diagram, the areas above and below the mean torque line taken in order are 4400, 1150, 1300 and 4550 mm^2 respectively. The scale of the turning moment diagram are;

Turning Moment, 1mm = 100 N-m
Crank angle, 1mm = 1°

Find the mass of flywheel required to keep the speed between 297 and 303 rpm. Radius of gyration is 0.525 m

Given, Maximum speed, $N_1 = 303$ rpm
Minimum speed, $N_2 = 297$ rpm,
Radius of gyration, $k = 0.525$ m

Now, we will draw a turning moment diagram, from the given data,



$\because$ 1 mm = 1 Nm

and 1 mm $= 1° = 1 \times \frac{\pi}{180}$ rad.

$\therefore 1\,mm^2 = 100 \times \frac{\pi}{180}$ Nm

$\therefore 1\,mm^2 = \frac{10}{18}\pi$ Nm —①

Now, let total energy at point 'A' = E

Total energy at point 'B' = E + 4400

Total energy at point 'c' = E + 3250 [E+4400 − 1150]

Total energy at point 'D' = E + 3250 + 1300
= E + 4550

Total energy at point 'E' = E + 4550 − 4550 = E.

$\because$ Maximum ~~fluctuation of~~ energy, at D = E + 4550
Minimum ~~fluctuation of~~ energy at C = E + 3250

Now,
max. fluctuation of energy, ΔE = Max Energy − Min Energy
= E + 4550 − E − 3250
= 1300 mm²

$\because \Delta E = 1300 \times \frac{\pi 10}{18}$ N-m
$\Rightarrow 2268.928$ N-m

Also, $\Delta E = I\omega^2 C_s$
$2268.928 = (mk^2)\omega^2 C_s$ —②

Now, Mean Speed, $N = \frac{N_1 + N_2}{2} = \frac{303 + 297}{2} = 300$ rpm

$\omega = \frac{2\pi N}{60} = 10\pi = 31.416$ rad/s

$C_s = \frac{N_1 - N_2}{N} = \frac{6}{300} = 0.02$ $\because$

Putting, in eq 2;

$$2268.928 = m \times 0.525^2 \times (31.416)^2$$

$$\Rightarrow \quad m = 417.033 \text{ kg.}$$

$\therefore$ mass of flywheel $= \underline{417.033 \text{ kg}}$

Qn. The flywheel of an engine has a radius of gyration of 1m, and mass 2500 kg. The starting torque of engine is 1500 N-m and may be assume constant. Determine,
(i) Angular acceleration of the flywheel.
(ii) Kinetic energy of the flywheel after 10 sec from start.

given, $K = 1\text{m}$, $m = 2500$ kg,
$T = 1500$ N-m

(i) Angular acceleration of flywheel,

Let, α = angular acceleration,

$$T = \underbrace{I}_{\text{mass moment of inertia}}\alpha$$

$$T = mk^2\alpha$$

$$1500 = 2500 \times 1^2 \times \alpha$$

$$\alpha = \frac{1500}{2500} = \frac{3}{5} = 0.6 \text{ rad/s}^2$$

(ii) K.E. after 10 sec,

Let, ω_1 = angular velocity at rest
ω_2 = angular velocity after 10 sec,

$$\omega_2 = \omega_1 + \alpha t$$

$$= 0 + 0.6 \times 10$$

$$= 6 \text{ rad/s.}$$

$$\therefore \text{KE after 10 sec} = \frac{1}{2} I\omega^2$$

$$= \frac{1}{2}\, 25000 \times 1^2 \times 6^2$$

$$= 45000 \text{ N-m} = \underline{45 \text{ kN-m}}$$

Numerical on Watt governal

Qn. Calculate the change in hight of wall governor whe its speed increase from 200 rpm to 201 rpm.

given,

initial speed, $N_1 = 200$ rpm

final speed, $N_2 = 201$ rpm

Now, Initial height $= \frac{895}{N_1^2} = \frac{895}{200^2} = 0.22375$ m

$h_1 = 0.022375$ m

$= 22.375$ mm, Final height $= \frac{895}{N_2^2} = \frac{895}{201^2} = 0.22153$ m

Now change in verticle height,

$= h_1 - h_2$

$= 22.375 - 22.153$

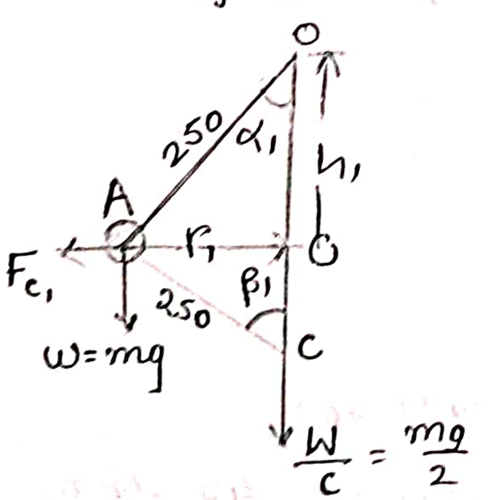
$= 0.222$ mm

Numerical on ~~watt~~ portor governon

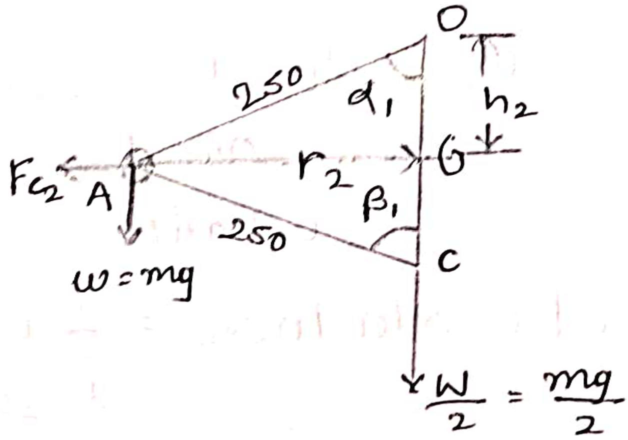
Qn. A porter governor has equal arms each 250 mm long and pivoted at the axis of rotation. Each ball has a mass of 3 kg and the mass of central load on the sleeve is 14 kg. Radius of rotation of the ball is 150 mm when the governer begins to lift and 200 mm. When the governor is at maximum speed. Find the maximum and minimum speed of gover and also range of speed.

(i) Friction at the sleeve is neglect.

(ii) Friction of 15 N at the sleeve is taken.



Minimum Position



Maximum Position

Given,

$AO = AC = 250mm$

$m = 3\ kg$, $M = 14kg$

$r_1 = 150mm$, $r_2 = 200mm$

Now, height of Governor at minimum position,

$OG = h_1 = \sqrt{AO^2 - AG^2} = \sqrt{AO^2 - r_1^2}$

$h_1 = \sqrt{250^2 - 150^2}$

$h_1 = 200\ mm = 0.2m$

Similarly, hight of governor at maximum position,

$h_2 = \sqrt{250^2 - 200^2}$

$h_2 = 150mm = 0.15\ mm$

Now, applying the formula for minimum position,

$$\omega_1^2 = \frac{mg + \frac{Mg}{2}(1+k)}{m \times h} \quad \text{where } k = \frac{\tan\alpha}{\tan\beta} = 1$$

$\because \tan\alpha = \tan\beta$

$$= \frac{2mg + Mg}{mh}$$

$$= \frac{3 \times 9.81 + 14 \times 9.81}{3 \times 0.2} = \frac{17 \times 9.81}{0.6}$$

$$= \cancel{400}\ 277.95$$

$\therefore \omega_1 = \sqrt{277.95}$

$= 16.67$

$\because \frac{N_1 2\pi}{60} = 16.67$

$\Rightarrow N_1 = \frac{16.67 \times 60}{2\pi} = \frac{1000.3099}{2\pi} = 159.204\ rpm$

$\therefore$ Minimum speed $= N_1 = 159.204$ rpm.

Now Similarly apply formula for maximum speed;

$$\omega_2^2 = \frac{mg + \frac{Mg}{2}(1+k)}{m \times h_2}$$

$$= \frac{3 \times 9.81 + 14 \times 9.81}{3 \times 0.15}$$

$$= \frac{17 \times 9.81}{0.45} = \frac{166.77}{0.45} = 370.6$$

$$\omega_2 = \sqrt{370.6}$$

$$= 19.2509 \text{ rad/s}$$

$$\frac{2\pi N_2}{60} = 19.2509$$

$$\Rightarrow N_2 = \frac{19.2509 \times 60}{2\pi} = \frac{1155.0584}{2\pi} = 183.833 \text{ rpm}.$$

$\therefore$ Range of speed $= N_2 - N_1 = 183.833 - 159.204$

$= 24.629$ rpm

(ii) When a friction 'F' of 15 N is considerd,

$$\omega_1^2 = \frac{mg + \frac{Mg \mp f}{2}(1+k)}{m + h_1}$$

$$= \frac{3 \times 9.81 + \frac{14 \times 9.81 - 15}{2}(1+1)}{3 \times 0.2}$$

$$\Rightarrow \frac{29.43 + 137.34 - 15}{0.6}$$

$$\Rightarrow \frac{151.77}{0.6} = 259.6167$$

$$\therefore \omega_1 = \sqrt{259.6167} = 16.11 \text{ rad/s}$$

$$N_1 = \frac{60 \times \omega_1}{2\pi} = \frac{966.6}{6.2831} = 153.876 \text{ rpm}.$$

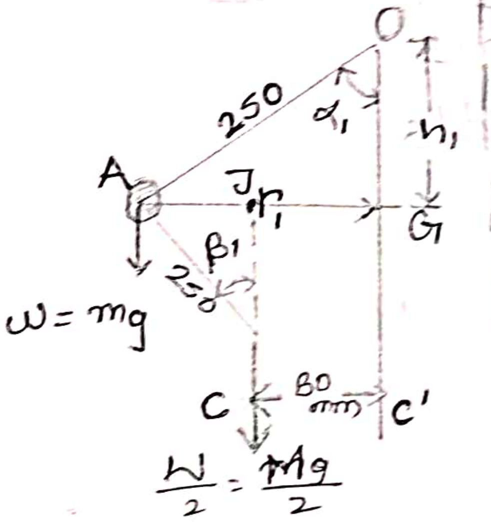
$$\omega_2^2 = \frac{3 + 9.81 + 14 \times 9.81 - 15}{3 \times 0.15} = \frac{151.77}{0.45} = 337.2667$$

$$\omega_2 = \sqrt{337.2667} = 18.36 \text{ rad/s}$$

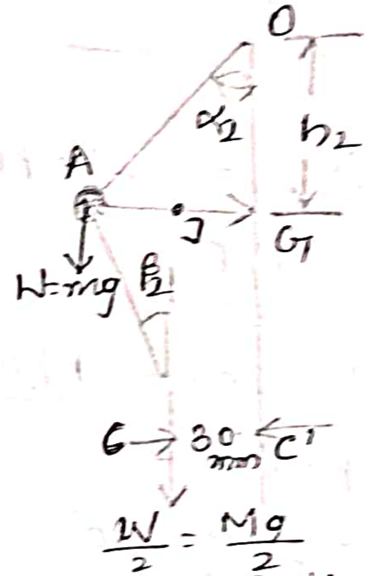
$$N_2 = \frac{60\,\omega_2}{2\pi} = \frac{1101.6}{6.2831} = 175.327 \text{ rpm}$$

$\therefore$ Range of speed $= N_2 - N_1 = 175.327 - 153.876 = 21.451$ rpm

Qn. A porter governor has all four arms 250mm long. The upper arms ~~and~~ are attached on the axis of rotation and the lower arms are attached to sleeve at a distance of 30mm from the axis. The mass of each ball is 5kg and sleeve has a mass of 50kg. The external radii of rotation are 150mm and 200mm. Determine the range of speed.



Minimum Position



Maximum Position

given,

$AO = AC = 250mm = 0.25\,m$

$CC' = 30\,mm = 0.03\,m$

$m = 5\,kg$, $M = 50\,kg$

$r_1 = 0.15\,m$, $r_2 = 0.2\,m$

$\because \tan\alpha \neq \tan\beta$, we have to find their value for both condition

For Minimum position,

$$OG^2 = AO^2 - AG^2$$

$$h_1^2 = AO^2 - r_1^2$$

$$\Rightarrow h_1 = \sqrt{250^2 - 150^2} = 200\,mm = 0.2\,m$$

Also, $AJ = AG - JG$

$AJ = r_1 - CC' = 150 - 30 = 120\,mm = 0.12\,m.$

Now in ΔAOG,

$$\tan\alpha_1 = 0.75 \quad \left[\frac{AG}{OG} = \frac{0.15}{0.2} = 0.75\right]$$

Now, in ΔACJ

$$\tan\beta_1 = \frac{AJ}{CJ} = \frac{0.12}{0.219} = 0.548 \quad (CJ^2 = AC^2 - AJ^2) \quad \therefore CJ = 0.219$$

Now, $$\omega_1^2 = \frac{mg + \frac{Mg}{2}(1+k_1)}{m \times h_1}$$

$$= \frac{5\times 9.81 + \frac{50\times 9.81}{2}(1+0.731)}{5\times 0.2}$$

$$\omega_1 = 21.762 \text{ rad/s}$$

$$N_1 = \frac{\omega_1 \times 60}{2\pi}$$

$N_1 = 207.81$ rpm. (minimum speed)

For maximum position →

$$h_2^2 = 250^2 - 200^2$$

$$h_2 = 0.15$$

$AJ = 200 - 30 = 170$ mm $= 0.17$ m

In ΔAOG; $\tan\alpha_2 = \frac{0.2}{0.15} = 1.333$

ΔACJ; $\tan\beta_2 = \frac{AJ}{CJ} = \frac{0.17}{0.1833} = 0.93$ $\left[CJ^2 = AC^2 - AJ^2 \quad CJ = 0.1833 \right.$

Now, $\omega_2^2 = 5\times 9.81 + \frac{50\times 9.81}{2}(1+0.697)$

$$\omega_2 = 24.926 \text{ rad/s}$$

$N_2 = 238.025$ rpm (maximum speed)

Range of speed; $= N_2 - N_1$

$= 238.025 - 207.81$

$= 30.215$ rpm

Numerical on Proell Governor

Qn. A proell Governor has equal arm of length 300 mm. The upper and lower end of arm are pivoted on the axis of governor. The extension arms of the lower links are each 80 mm long and parrallel to the axis when the radii of rotation of the balls are 150 mm and 200 mm. The mass of each ball is 10 kg and the mass central load is 100 kg. Determine the range of the governor.

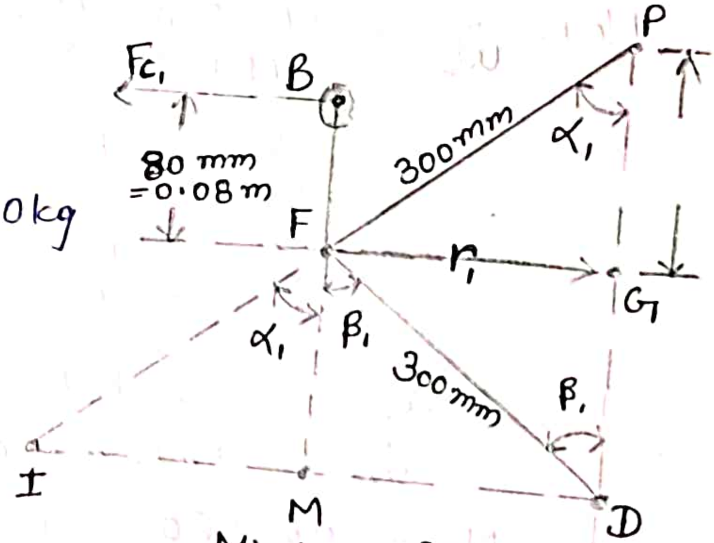
given

Mass of ball, $m = 10\ kg$,

Mass of central load, $M = 100\ kg$

$r_1 = 150\ mm = 0.15\ m$

$r_2 = 200\ mm = 0.2\ m$



Minimum Position

for maximum position,

from ΔPFG;

$PG^2 = PF^2 - FG^2$

$h_1^2 = 300^2 - r_1^2 = 300^2 - 150^2$

$h_1 = 259.808\ mm = 0.25980\ m$

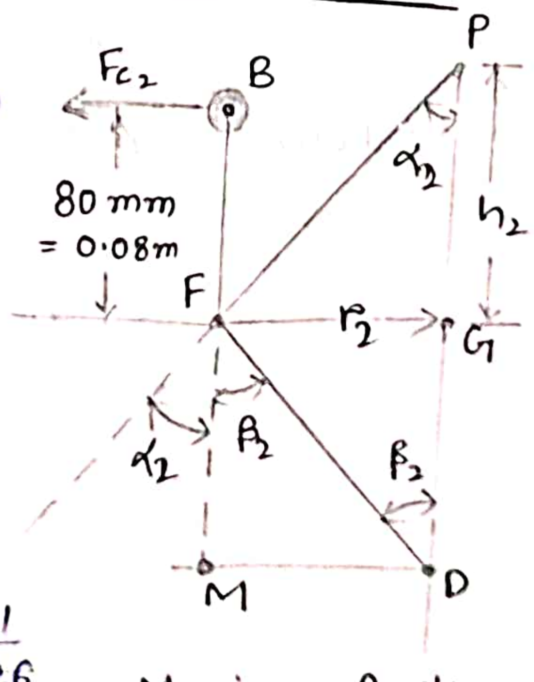
$h_1 \simeq 0.26\ m$,

Similarly, $FM = 0.26$

also, $BM = BF + FM$

$= 0.08 + 0.26$

$= 0.34\ m$



Maximum Position

$$\left[q = \frac{\tan\beta}{\tan\alpha}\right]$$

Now, $\omega_1^2 = \frac{FM}{BM}\left[\frac{m+\frac{M}{2}(1+q)}{m}\right]\frac{g}{h_1}$

$$= \frac{0.26}{0.34}\left[\frac{10+100(2)}{10}\right]\frac{9.81}{0.26}$$

$$= \frac{981 \times 11}{34} = \frac{10791}{34} = 315.617$$

$\omega_1 = \sqrt{315.617} = 17.765\ rad/s$.

$N_1 = \frac{\omega_1 \times 60}{2\pi} = \frac{17.765 \times 60}{6.28318} = \frac{1065.9}{6.28318} = 169.64\ rpm$

(minimum speed).

for Maximum position,

from ΔPFG

$$PG^2 = PF^2 - FG^2$$

$$h_2^2 = 300^2 - 200^2$$

$$h_2 = 223.61\ mm = 0.224\ m.$$

Now, similarly,

$$FM = 0.224$$

also $BM = BF + FM$

$$= 0.08 + 0.224 = 0.304\ m$$

Now, $$\omega_2^2 = \frac{FM}{BM}\left[\frac{m + \frac{M}{2}(1+q_2)}{m}\right]\frac{g}{h_2}$$

$$= \frac{0.224}{0.304}\left[\frac{10 + \frac{100}{2}(1+1)}{10}\right]\frac{9.81}{0.224}$$

$$= \frac{9.81}{0.304} \times 11 = \frac{100.98}{0.304}$$

$$\omega_2^2 = 332.171$$

$$\omega_2 = 18.225\ rad/s$$

$$N_2 = \frac{\omega_2 \times 60}{2\pi} = \frac{18.225 \times 60}{6.28318} = \frac{1093.5}{6.28318} = 174.036\ rpm$$

(maximum speed)

$\therefore$ Range of speed,

$$= N_2 - N_1$$

$$= 174.036 - 169.64$$

$$= 4.396\ rpm$$

THEORY OF MECHANICS & MECHANISMS

UNIT-2

Qn1. Write about belt drive, material used in belt drive, types of belt drive, open belt and cross bet drive and its application.

Qn2. Derive these expression,

- Velocity Ratio of Belt drive
- Angle of contact/lap
- Length of open belt
- length of cross belt
- Driving Tension in belt
- Max^m & Min^m stress in belt
- Power Transmission in belt
- Initial tension in belt.

Qn 3. Write about on rope drive with their types and applicatio of rope drive also for both fibre and wire rope drive.

Qn.4. Write down the gear Nomenclature and terminology. And classify gear on different points.

Qn 5. Explain gear train and its type. Also dertermine the velocity ratio for all types.

Qn.6. Write about chain drive with advantages and dis-advantages and also selection of chain.

UNIT-3

Qn.1. Write about flywheel with its application

Qn.2. Write the governor with its neat sketch and working with is all four types. Also write its application.

Qn. 3. Define, (i) Height of governor, (ii) Sleeve lift, (iii) Equilibrium, (iv) sensitiveness of governor, (v) stability of governor, (vi) Isochronism and (vii) Hunting in governor.

Qn. 4. Write down the difference between flywheel and governor.

All Numericals are important.

UNIT-4

Qn1. Write the function of brakes and dynamoteter and also define both of them.

Qn.2. Write down the types of brake and equation for all four types. Also derive them.

Qn3. Write the types of dynamometer with a neat sketch.

Qn.4. Explain the term clutch with its types also Bearing with their types.

All Numericals are important.

UNIT-5

Qn.1. Explain balancing in deep. Also solve numerical on balancing with both methods, Analytical and graphical representation method.

Qn.2. Write about vibration and its types.

Qn. 3. Explain Causes, effects and remidies of vibration.