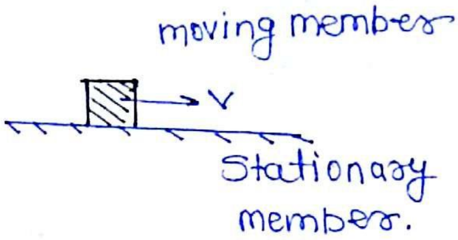
V.V. Imp.
ESE/Gate

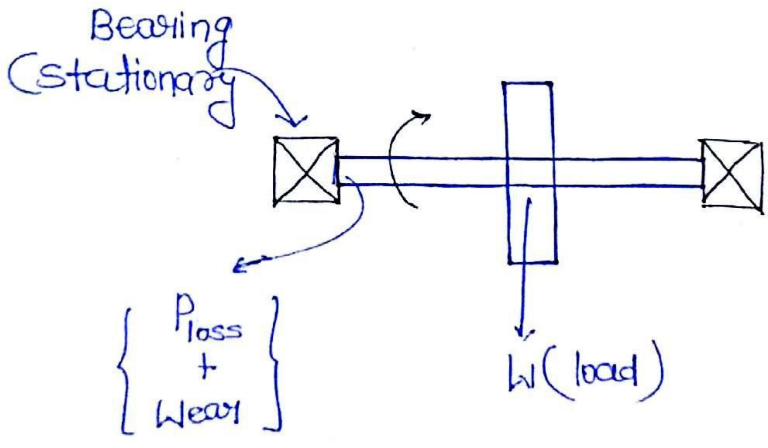
Bearing

Bearing :- when ever relative motion occurs b/w two machine elements. which is stationary, supporting the moving machine element is refered as bearing.



→ Bearing Yes

Bearing According to the shaft :- Bearing is define as m/c element whose function is to support the rotating element shaft, axle,. to guide, Allign and confined it's motion, while preventing the motion in the direction of applied ~~motion~~ load.



There is always power loss accur while overcoming the frictional resistance and also wear occurs hence lubricant require to minimize losses and wear.

Bearing is said to be a good bearing which perform above function with minimum losses and wear.

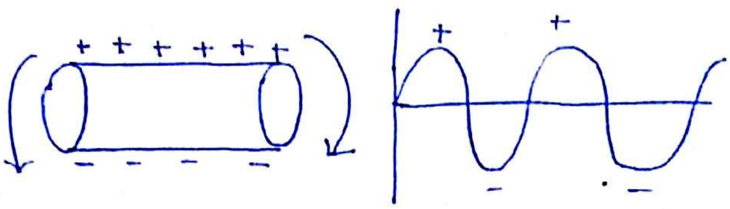
<u>Axle</u>

- only Bending
- No-power transmission
  ⇓
  No- torque
  ⇓
  No- twisting

Axle are Design by "bending" only

<u>Shaft</u>

- Always Bending
- Aim → power transmission
  ⇓
  Always torque
  ⇓
  Always twisting



'Fatigue design'

Real shaft are design by "Fatigue"

Classification of Bearing:-

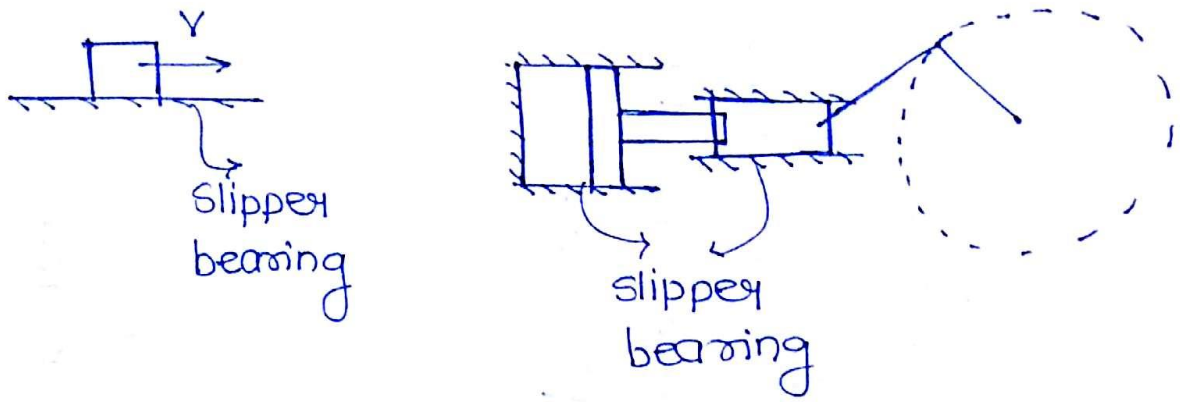
Bearing

$\mu_{rolling} << \mu_{sliding}$

- Sliding contact bearing (sliding b/w two surfaces)
  - Sleeve bearing
    - w.r.t. type of lubrication
      - Thin film lub^n (boundary lub^n)
      - Thick film lub^n
        - Hydrodynamic lub^n
        - Hydrostatic lub^n
      - Solid or zero film lub^n
    - w.r.t. dir^n of load
      - Axial load bearing or thrust bearing
        - Pivot bearing (foot step bearing)
          - Flat pivot
          - Conical pivot
        - Collar bearing
          - Flat collar bearing
          - Conical collar bearing
      - Radial load bearing / Journal bearing
        - Full J.B.
        - Partial J.B.
        - Fitted J.B.
  - Slipper or Guide bearing
    - w.r.t. type of Bushes
      - Solid Bush Bearing
      - Split bush Bearing
- Rolling contact bearing — Anti friction bearing (Rolling b/w two surface)
  - Ball bearing
    - Deep Groove B.B. (Noiseless B.B)
    - Angular Contact B.B.
    - Self aligning B.B
    - Thrust B.B.
  - Roller bearing
    - Cylendrical R.B.
    - Spherical R.B.
    - Taper R.B.
    - Needle R.B

Slipper or Guide Bearing :-

when sliding occurs in a state line dirⁿ bearing refered as slipper bearing.

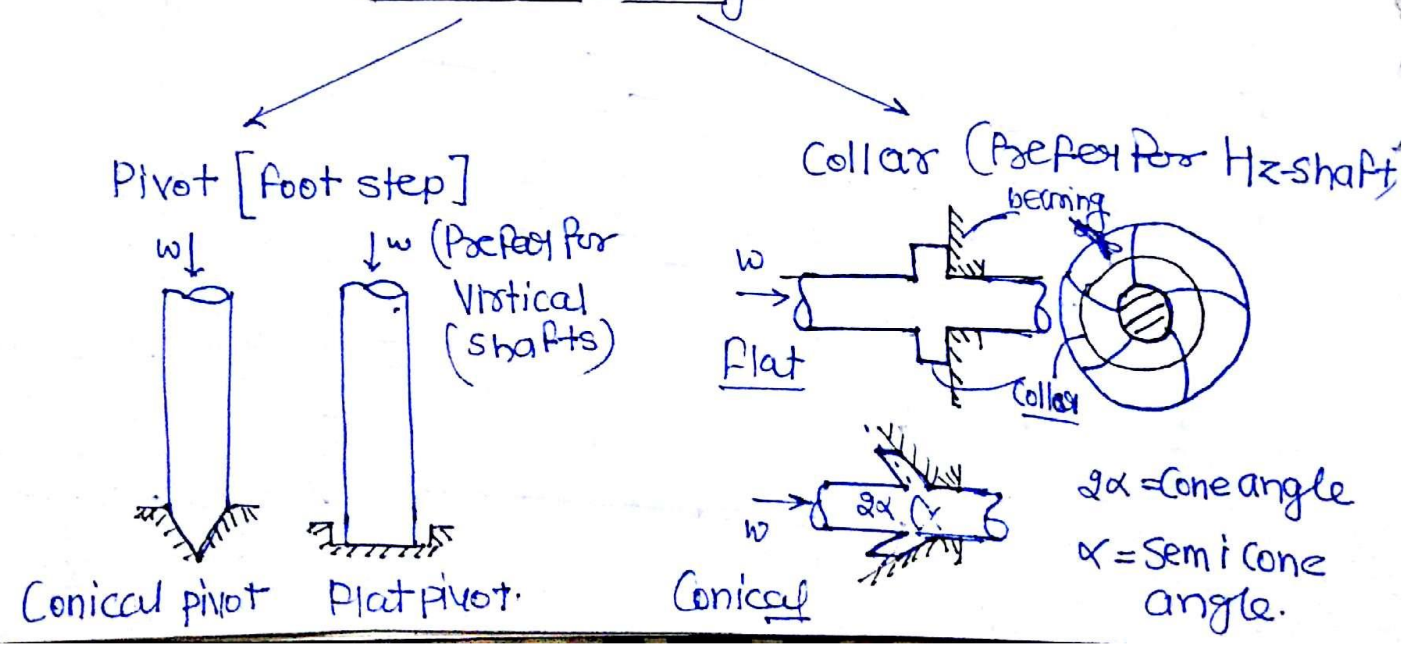


Hence not used for shaft

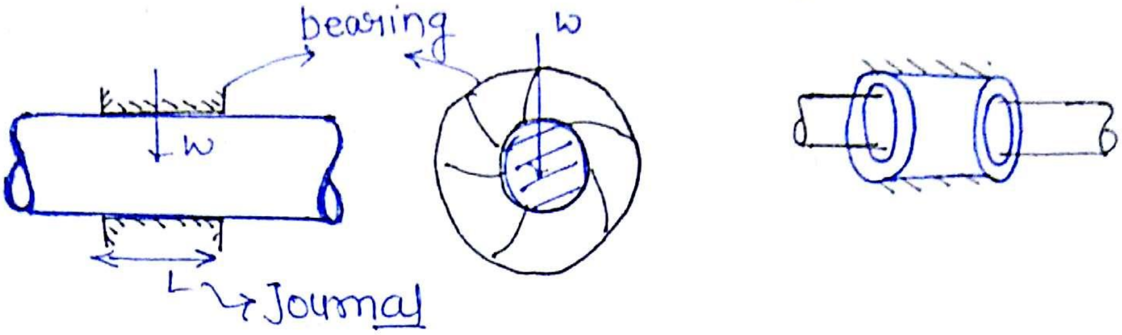
Sleeve Bearing :- When sliding occurs in a circular direction implies that around the pheriferi of cylinder of Circle.

Hence used for shaft.

Axial load bearing
or
Thrust bearing

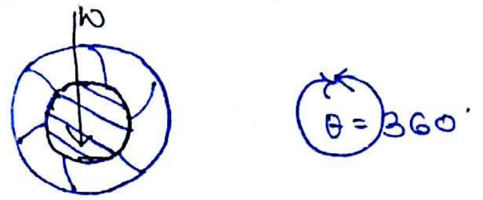


Radial load Bearing / Journal bearing.

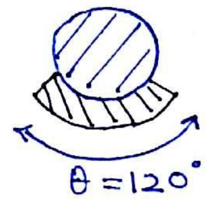


The portion of shaft lies inside the bearing is called journal.

Full J.B.



Partial J.B.

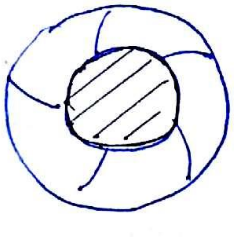


Can only be use when load is acting only in one dirn.

Fitted J.B.

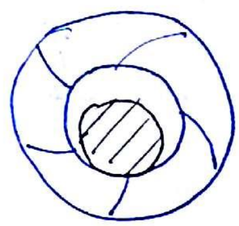
Bearing dia > shaft dia ⇒ with clearance

Bearing dia = shaft dia ⇒ No - Clearance
⇓
Fitted J.B.

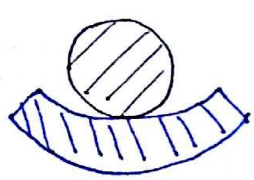


Fitted Full J.B

Filled partial J.B.



with clearance Full. J.B.



with clearance partial J.B.

Solid / zero film lubrication :-

When metal behaves like a lubricant refered as solid lubricant

eg. Graphite, cast iron, teflon.

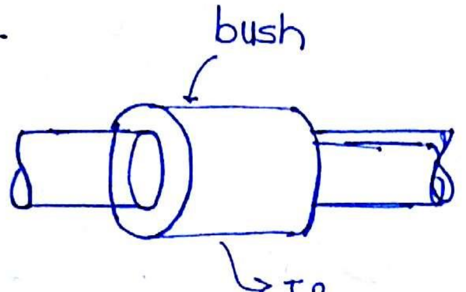
Thin film:- Metal to metal contact will is present at any speed. Lubricant is used to reduced the coefficient of b friction only.

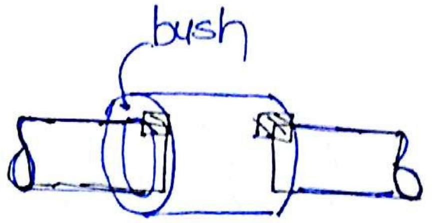
Thick film:- Metal to metal contact will always avoided.

Hydro-dynamic = Metal to metal contact will avoided only at High speed condition

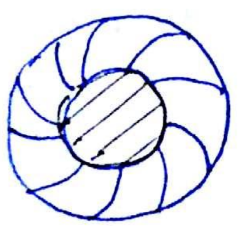
Hydro-static - Metal to metal contact will avoided at stationary condn.

Bush:-

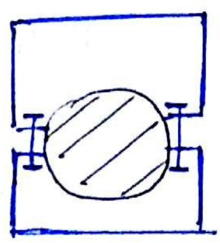




solid - Muff / Sleeve - Coupling.

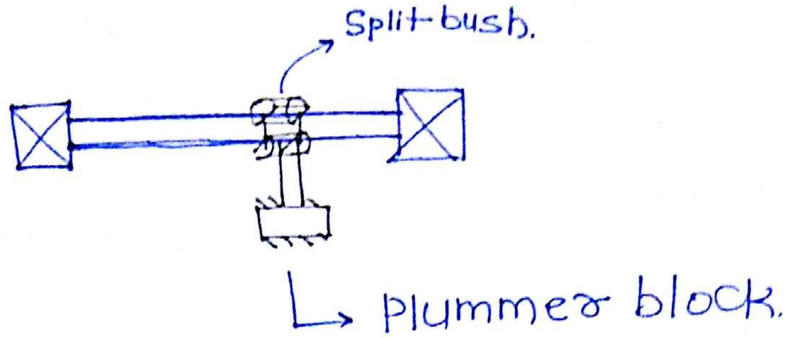


Solid Bush



Split bush

Long shaft



* Plummer block is used to provide intermidiate support for long shaft.

Antifriction Bearing / Rolling Contact bearing.

Ball Bearing
↳ Rolling element
→ Ball → point contact
→ load capacity ↓
→ loss ↓
→ Cost ↓

Roller bearing
↳ Rolling element
→ ~~Roller~~ line Contact
→ load Capacity ↑
→ loss ↑
→ Cost ↑

Ball Bearing. (see photos [illegible]

Deep Groove	Angular Contact	Self Aligning	Thrust
① $\frac{F_r}{F_a} > 1$ ② Constaction simple ③ Cost ↓ ④ most Commanly use ⑤ Noise less BB ⇓ Min Noise in all A.F.B. ⑥ doesn't permit any misallignment b/w shaft & bearung houshing	① $\frac{F_r}{F_a} < 1$ wear more axial load ② more stronger than Deep Groove ③ Constouction difficult ④ preloading of bearing require ⑤ cost ↑ ⑥ Two bearing required to wear load in both direction ⑦ doesn't permit any misallignment b/w shaft & bearing housing	① Permit mis-alignment b/w shaft & bearing houshing. ⇒ Two-row of moving balls are used. ② F_r ✓, F_a ✓	① F_a ✓, F_r ✗ wear only axial load ② Refer for Vertical shaft

Rollor Bearing

Cylendrical R.B.

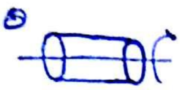
$L/D < 1$

① F_r ✓, F_a ×

only bear radial load

② max. radial load bearing capacity

③ Effective for Horizontal Shafts



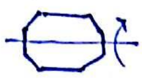
Spherical R.B.

$L/D < 1$

① F_r ✓, F_a ✓

② Permit mis-alignment b/w shaft and bearing housing

→ Two row of moving roller are used.



Tapper R.B.

$L/D < 1$

① $\frac{F_r}{F_a} > 1$

② max. load bearing capacity.

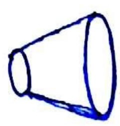
③ prefered Under fatigue and Impact loading.

④ Construction is very diffcult.

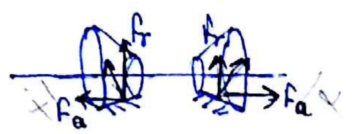
⑤ Pre-loading of bearing require

⑥ Tightning Nut in the races also req.

⑦ max. Cost



⑧ Tapper roller always use in pair



⑨ Doesn't permit any mis-alignment.

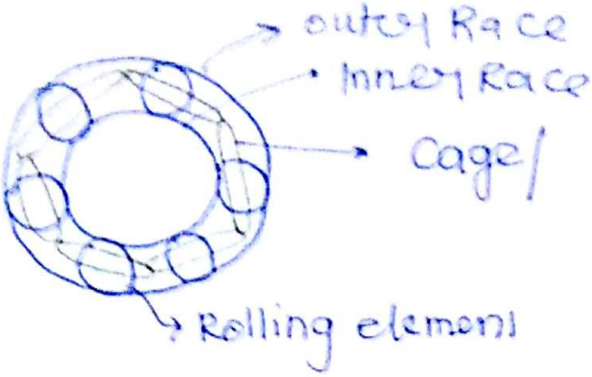
Needle R.B.

$L/D > 1$

① Needle roller are prefered when radial space is a constraint.

② max. load bearing cap. in a given radial space.

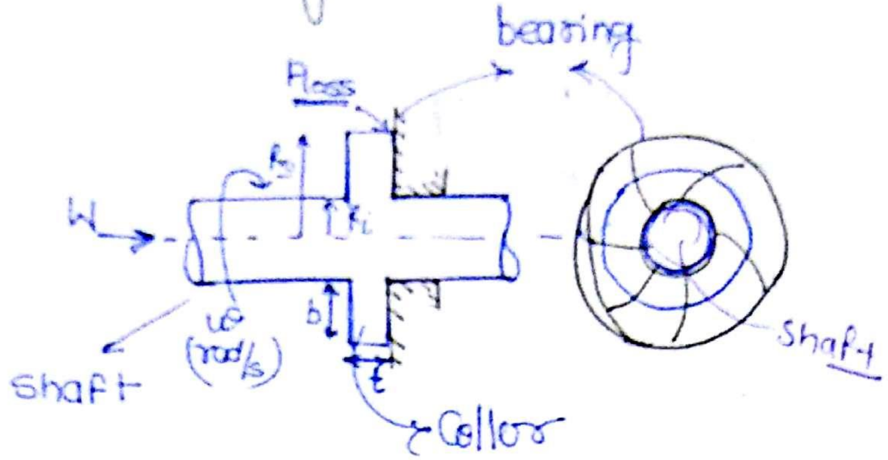
Function of Cage



- Cage is used to avoid the clustering of rolling element at one location.
- Cage is used to maintain constant relative angular position b/w two rolling element.
- Cage is used to avoid metal to metal contact b/w rolling element to minimise losses and wear.

Note:- cage is absent in case of Needle roller bearing.
Because needle rollers are placed all around the pheriferi of shaft.

Flat Collor Bearing :-



- R_o = outer radius of collor
- R_i = Inner radius of collor
- t = thickness of collor
- b = widthof collor

  $b = R_o - R_i$
- W = Axial load.

if $W\uparrow$ → $R_o\uparrow$, $t\uparrow$

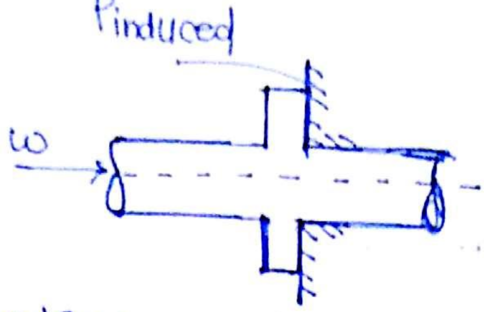
if Radial spaceis constrainst then

→ $n\uparrow$ = no. of collor.

Flat collor bearing

Single collor bearing | multi collor bearing.

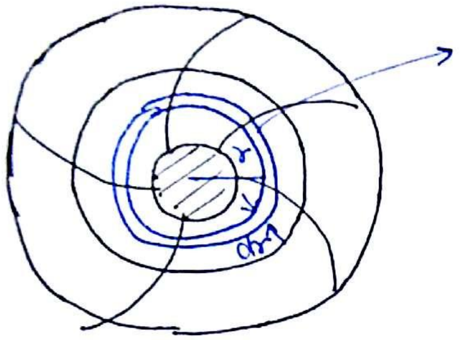
* Single collar bearing:-



Crushing Design:

Design by Uniform pressure theory:-(UPT)

$$P_{induced} = \text{Constant.}$$



$$P_{induced} = P = \text{Constant}$$

$$dW = 2\pi r . dr . P$$

$$\int_0^W dW = \int_{R_i}^{R_o} 2\pi r . dr . P$$

$$W = \pi (R_o^2 - R_i^2) . P$$

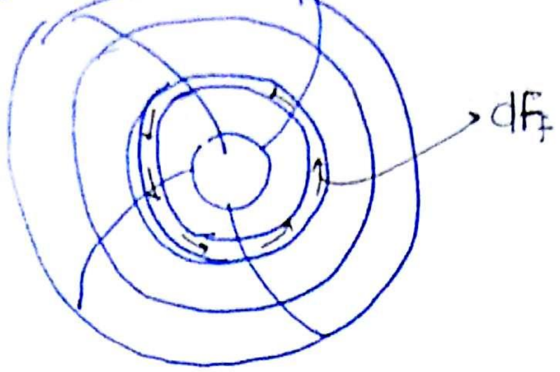
**
$$\boxed{P_{ind.} = \frac{W}{\pi (R_o^2 - R_i^2)}}$$

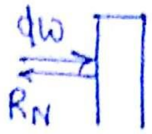
$$P_{ind} \leq P_{per.}$$

$$\frac{W}{\pi (R_o^2 - R_i^2)} < P_{per.}$$

strength of Collar $\quad W_{max} = \pi (R_o^2 - R_i^2) P_{per.}$

Frictional torque :-





$$dF_f = \mu R_N = \mu dW$$

$$dF_f = \mu 2\pi r dr . p \quad , \quad dT_f = dF_r \times r$$

$$\int_0^{T_f} dT_f = \int_{R_i}^{R_o} 2\pi r^2 \mu p dr$$

$$T_F = \frac{2}{3}\mu \pi P(R_o^3 - R_i^3)$$

$$P_{ind} = \frac{W}{\pi(R_o^2 - R_i^2)}$$

Frictional torque:

$$\boxed{T_F = \frac{2}{3}\mu W\left[\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right]}$$

UPT

$$\boxed{P_{loss} = T_F \times \omega}$$

Design by uniform wear theory : (UWT)

$P_{ind} \propto \frac{1}{r}$ Non uniform induced pressure.

$$P_{ind} = \frac{C}{r}$$

$$dW = 2\pi r . dr . P$$

$$\int_0^W dW = 2\pi C \int_{R_i}^{R_o} dr$$

$$W = 2\pi C (R_o - R_i)$$

$$C = \frac{W}{2\pi (R_o - R_i)}$$

$$\boxed{P_{ind} = \frac{W}{2\pi r (R_o - R_i)}}$$

$P_{ind} \uparrow$, $r \downarrow$
$\hookrightarrow R_i$

Safe condⁿ

$(P_{ind})_{max} \leq P_{per.}$ $\quad (P_{ind})_{max} = \frac{W}{2\pi R_i (R_o - R_i)}$

$$\frac{W}{2\pi R_i (R_o - R_i)} \leq P_{per.}$$

$$\boxed{W_{max} = 2\pi R_i (R_o - R_i)}$$ strength of collar by UWT

Frictional torque :-

$$dF_f = \mu R_N = \mu \, dW$$

$$dF_f = \mu \, 2\pi r \, dr . p \quad , \; dT_f = dF_f \times r$$

$$\int_0^{T_f} dT_f = \int_{R_i}^{R_o} \mu 2\pi r^2 dr . p \qquad p = \frac{C}{r}$$

$$T_f = 2\pi \mu C \int_{R_i}^{R_o} r \, dr$$

$$T_f = \pi \mu C \left(R_o^2 - R_i^2 \right)$$

$$T_f = \frac{W \mu \left(R_o^2 - R_i^2 \right)}{2 \left(R_o - R_i \right)}$$

$$\boxed{T_{f_{UWL}} = \frac{\mu W \left(R_o + R_i \right)}{2}}$$

$$\boxed{P_{loss} = T_f \times \omega}$$

UPT

$P_{ind} = C$

$$P_{ind} = \frac{W}{\pi(R_o^2 - R_i^2)}$$

Safe condⁿ

$P_{ind} \le P_{per}$

$$W_{max} = \pi(R_o^2 - R_i^2) P_{per}$$

$$T_{f_{UPT}} = \frac{2}{3}\mu W \left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)$$

$$T_f = \mu W R_{eff}$$

$$R_{eff} = \frac{2}{3}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)$$

$$P_{loss} = T_f \times \omega$$

$$\tau_{ind} = \frac{W}{2\pi R_i t}$$

$$W_{max} = 2\pi R_i t \tau_{per}$$

UWT

$P_{ind} \propto \frac{1}{r}$ $\quad P_{ind} = \frac{C}{r}$

$$P_{ind} = \frac{W}{2\pi r(R_o - R_i)}$$

Safe condⁿ

$(P_{ind})_{max} \le P_{per}$

$$W_{max} = 2\pi R_i (R_o - R_i) P_{per}.$$

$$T_{f_{UWT}} = \mu W \left(\frac{R_o + R_i}{2}\right)$$

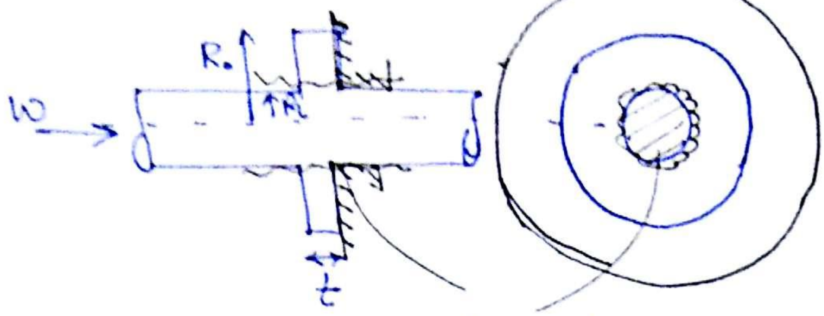
$$R_{eff} = \frac{R_o + R_i}{2}$$

$$P_{loss} = T_f \times \omega$$

$$\tau_{ind} = \frac{W}{2\pi R_i t}$$

$$W_{max} = 2\pi R_i t \tau_{per}.$$

Shear Design of Collar:-



Shearing

sheared Area $= 2\pi R_i t$

$$\tau_{ind} = \frac{W}{2\pi R_i t}$$

Safe Cond^n

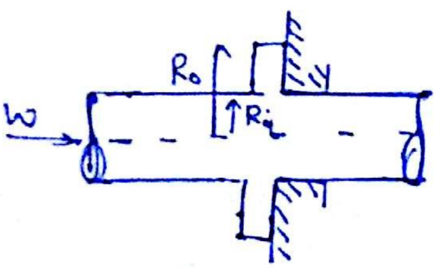
$$\tau_{ind} \le \tau_{per.}$$

$$\frac{W}{2\pi R_i t} \le \tau_{per.}$$

Shear strength of Collar.

$$\boxed{W_{max} = 2\pi R_i t \, \tau_{per}}$$

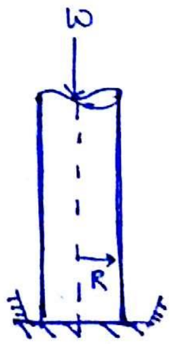
Expression for Friction torque in Case of Flat pivot bearing:-



$\Rightarrow$

$R_i = 0$

$R_o = R$



Flat Collar

Flat Pivot

Flat pivot bearing

$$(T_f)_{UPT} = \frac{2}{3}\mu W R \quad , \quad R_{eff} = \frac{2}{3}R$$

$$(T_f)_{UWT} = \frac{\mu W R}{2} \quad , \quad R_{eff} = \frac{R}{2}$$

$$\frac{(T_f)_{UPT}}{(T_f)_{UWT}} = \frac{4}{3} = 1.33$$

* Hence the frictional torque by UPT is 33% greater than that of by UWT. in case of flat pivot bearing.

* Frictional torque by UPT is always greater that by frictional torque by UWT for all cases.

$$(T_f)_{UPT} > (T_f)_{UWT} \quad \text{Always}$$

Single clutches:-

$(T)_{UPT} > (T)_{UWT}$ For all Cases

Bearing

Aim → W support

⇓

Collors

⇓

T_f

⇓

P_{loss}

Clutch

Aim → Power transmission

⇓

Friction lining

⇓

T_f → power transmit

⇓

axial load by spring

① For the safe design of the bearing it is batter to use uniform pressure theory because power loss occurs in overcoming the frictional Resistance.

only clutches means old clutch if nothing mention

② For the safe design of clutches (old clutches or worn out clutch) it is better to use uniform wear theory because pressure is non-uniformaly distributed over the clutches surface.

③ For the safe design of new clutches it is batter to use uniform pressure theory because pressure is uniformaly distributed over the clutches surface when they are new.

Q.12

$D_0 = 100\text{ mm}$ uniform pressure theory.

$D_i = 40\text{ mm}$

$P = 2\text{ MPa}$, $\mu = 0.4$

$T_f = ?$

$$T_f = \frac{2}{3}\mu W\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)$$

$$P_{ind} = \frac{W}{\pi(R_o^2 - R_i^2)}$$

$$T_f = \frac{2}{3}\mu P \pi (R_o^3 - R_i^3)$$

$$T_f = \frac{2}{3} \times 0.4 \times 2\times10^6 \times \pi \times (50^2 - 20^2) \times 10^{-6}$$

$$T_f = \frac{2}{3} \times 0.4 \times \pi \times (50^2 - 20^2)$$

$$T_f = \underline{\underline{196\text{ Nm}}}$$

Q.36 $P = 5\text{ kW} = T_f \times \omega$

$N = 2000\text{ rpm}$

$\mu = 0.25$

$R_i = 25\text{ mm}$ $R_o = ?$

$P = 1\text{ MPa}$

$$T_f = \frac{P}{\omega} = \frac{5000 \text{ J/s}}{2\times\pi\times 2000/60} = 23.87 \text{ Nm.}$$

$$T_f = \frac{2}{3}\mu \times \pi \times P \times \left(R_o^2 - R_i^2\right)$$

$$27.87 = \frac{2}{3} \times 0.25 \times \pi \times 1\times 10^6 \times \left(R_o^2 - (0.025)^2\right) \times 10$$

$$R_o = 39.4 \text{ mm}$$

Que:- A single flat clucth has $\mu = 0.3$ and the intencity of pressure can't not exceeds 1.5 MPa the outer diameter of friction lining is 200 mm & inner dia is 100 mm. assuming uniform wear theory find out the max. torque that can be transmitted

Sol^n

$$T_{f_{max}}\uparrow = \mu W \uparrow \left(\frac{R_o + R_i}{2}\right)$$

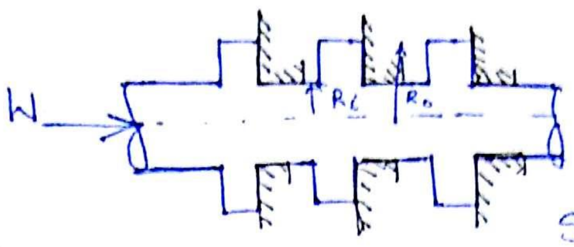
$$W_{max} = 2\pi R_i (R_o - R_i) \times P_{per}.$$

$$T_f = \mu \times 2\pi R_i (R_o - R_i) \times P_{per} \times \left(\frac{R_o + R_i}{2}\right)$$

$$T_f = 0.3 \times 2\pi \times 50 \times (100 - 50) \times 1 \times \left(\frac{100 + 50}{2}\right) \times 10^{-6} \times 10^{+6}$$

$$T_f = 529.875 \text{ Nm}$$

Multi collar Bearing:-



Similar Collar

if $W\uparrow$ → if Radial space is Constraint

$n\uparrow$ = No of Collar.

W_{collar} = load of each Collar

$$\boxed{W_{collar} = \frac{W}{n}}$$

Bearing ⇒ UPT

$$P_{ind} = \frac{W_{collar}}{\pi(R_o^2 - R_i^2)}$$

$$P_{ind} = \frac{W}{n\pi(R_o^2 - R_i^2)}$$

$$P_{ind} \leq P_{per.}$$

$$\frac{W}{n\pi(R_o^2 - R_i^2)} \leq P_{per}$$

$$\boxed{W_{max} = n\pi(R_o^2 - R_i^2)\,P_{per.}}$$

Frictional torque :-

$$T_f = \eta\left[T_{f\,collar}\right]$$

$$T_f = \eta\left[\frac{2}{3}\mu W_{collar}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)\right]$$

$$\boxed{T_f = \frac{2}{3}\mu W\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)}$$

* Hence frictional torque is independant of number of collar.

$$\boxed{\left(T_f\right)_{\substack{\text{single}\\ \text{collar bearing}}} = \left(T_f\right)_{\substack{\text{multi}\\ \text{collar bearing}}}}$$

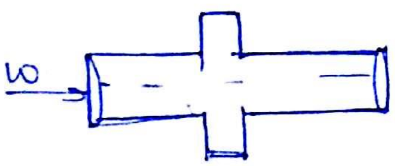
$$\text{Power loss} = T_f \times \omega$$

So

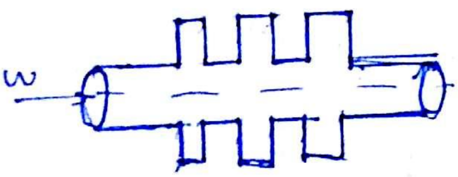
$$\boxed{\left(P_{loss}\right)_{SCB} = \left(P_{loss}\right)_{MCB}}$$

Power losses are same

*



- $W_{collar} = W$
- $P_{ind} = P$
- $T_f = T$
- $P_{loss} = P'$
- $\tau_{collar} = \tau$



- $W_{collar} = \frac{W}{3}$
- $P_{ind} = \frac{P}{3}$
- $T_f = T$
- $P_{loss} = P'$
- $\tau_{collar} = \frac{\tau}{3}$

Que:- Which of the following statement are valid for multicollar thrust bearing

(i) Friction movement is independant of No. of Collar,

(ii) Coeff. of friction is affected by No. of Collar.

(iii) Intensity of Collar is affected by No. of Collar!

Ans I & III

Que:- The thrust of a propeller shaft in a marine engine is taken up by no. of collar in-built ~~wh~~ with the shaft which is 30 cm in dia the total axial thrust is 200 KN and speed is 75 rpm. The coeff. of friction b/w surfaces is 0.05 and uniform intensity of pressure is 0.3 MPa find the external dia. of Collars and No. of the collor require if power loss can not exceeds 16 KW.

Soln $D_i = 30\,cm = 0.3\,m = 0.15\,m$ $(P)_{loss} \leq 16\,kW$

$W = 200\,KN$

$N = 75\,rpm$ $(P_{loss}) \leq T_f \times \omega$

$\mu = 0.05$ $16 \times 10^3 = \frac{2\pi(75)}{60} \times T_f$

$P = 0.3\,MPa$

$T_f = 2037.18\,N\text{-}m$

$$16000 \leq \frac{2}{3} \times k\, w \left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2} \right) \times \omega$$

$$16000 \leq \frac{2}{3} \times \frac{0.05}{100} \times 200 \times 10^3 \times \left\{ \frac{R_o^3 - (0.3)^3}{R_o^2 - (0.3)^2} \right\} \times 2\pi \times \frac{75}{60}$$

$$\frac{16 \times 3 \times 60}{2 \times 5 \times 2 \times 2\pi \times 75} = \frac{R_o^3 - (0.15)^3}{R_o^2 - (0.15)^2}$$

$$R_o = 24.92 \text{ cm}$$

$$D_o = 49.84 \text{ cm.}$$

$$P_{ind} = \frac{w}{n\pi \left(R_o^2 - R_i^2 \right)}$$

$$0.3 \times 10^3 = \frac{200}{n\pi \left((0.2492)^2 - (0.15)^2 \right)}$$

$$n = 5.3 \simeq 6 \quad \underline{\text{Ans}}$$

Que Design a multi collar thrust bearing to take an axial thrust of 16.3 Tonne and the intensity of pressure can't exceed 0.7 MPa the outer dia of Collar are 42 cm & inner dia is 32 cm and the safe shear stress for the collar is 25 MPa.

Sol^n

R_o, R_i, b, t, n ? ?

$W = 16.3$ Ton $\qquad D_o = 42$ cm

$P_{max} = 0.7$ MPa $\qquad D_i = 32$ cm.

$\tau_s = 25$ MPa. $\qquad W = 16.3 \times 10^3$ kg

$W = 16.3 \times 10^3 \times 9.81$ N

$$P_{ind} = \frac{W}{n\pi(R_o^2 - R_i^2)}$$

$$0.7 \times 10^6 \, N/m^2 = \frac{16.3 \times 10^3 \times 9.81}{n \times \pi(0.21^2 - 0.16^2)}$$

$$n = 3.93 \simeq 4 \text{ Collar.}$$

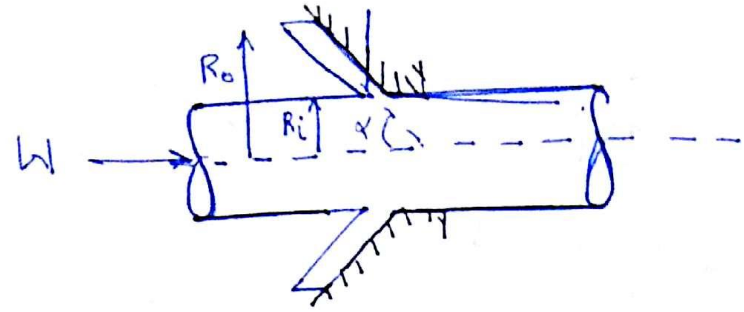
$$\tau = \frac{W_{collar}}{2\pi R_i \times t} \qquad W_{collar} = \frac{W}{4}$$

$$\tau = \frac{16.3 \times 10^3 \times 9.81}{2\pi \times 0.16 \times t \times 4} = 25 \times 10^6$$

$$t = 1.59 \times 10^{-3} \text{ m}$$

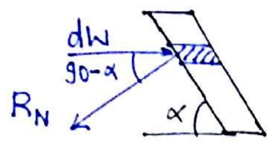
$$t = 1.59 \text{ mm}$$

Conical Collar Bearing :-



α = Semi-Cone Angle

$$b = \frac{R_o - R_i}{\sin\alpha}$$



$$dF_f = \mu R_N = \frac{\mu \, dW}{\sin\alpha}$$

Replace $\Rightarrow$ $\mu \longrightarrow \dfrac{\mu}{\sin\alpha}$

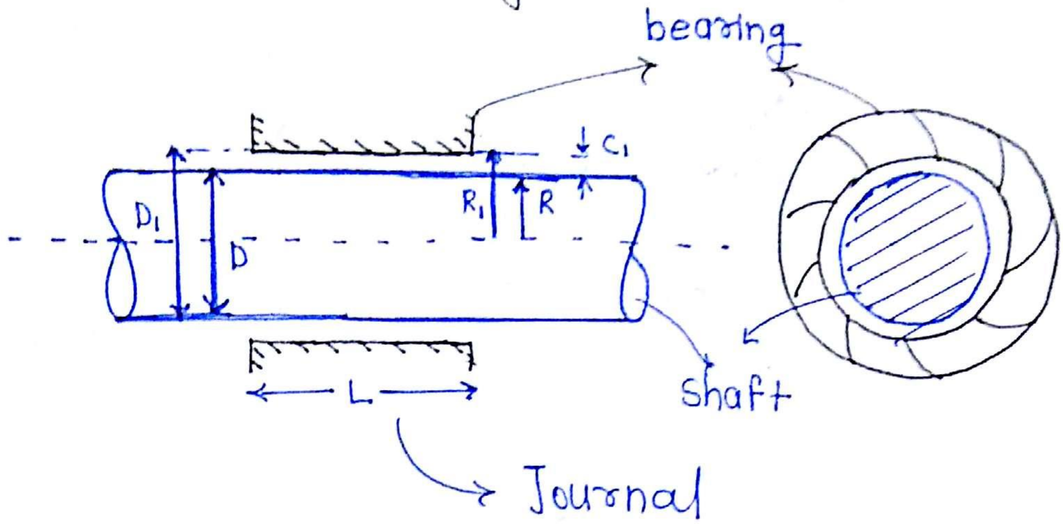
Flat Collar $\qquad$ Conical Collar.

Multi ~~collar~~ Cone Collar bearing. (UPT)

$$P_{ind} = \frac{W}{n\pi(R_o^2 - R_i^2)}$$

$$T_f = \frac{2}{3}\frac{\mu}{\sin\alpha} W \left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)$$

Journal Bearing



- R = Radius of shaft / Journal
- R_1 = Radius of bearing
- $\underline{C_1}$ = Radial clearance
- $C_1 = R_1 - R$
- D = Dia of shaft / Journal
- D_1 = Dia of bearing
- $\underline{C}$ = Diametrical clearance

$$C = D_1 - D = 2(R_1 - R)$$

$$\boxed{C = 2C_1}$$

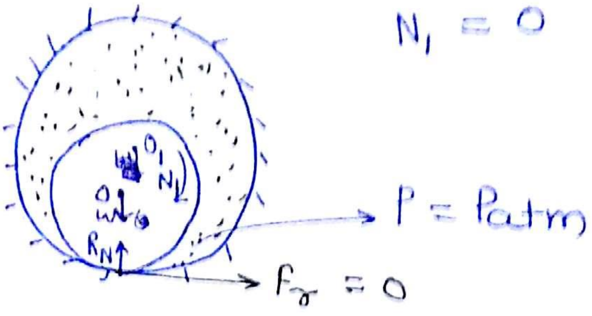
L = length of bearing / journal.

- Journal bearing is a sliding contact Radial load bearing Generally operating with hydrodynamic lubrication.

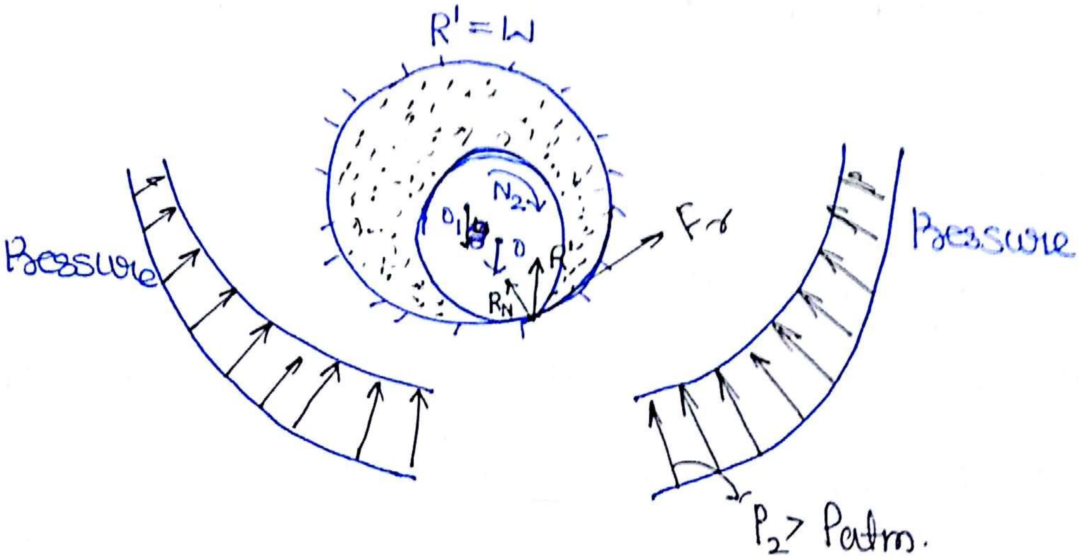
Hydro dynamic lubrication :-

Case - I   stationary condition.

$N_1 = 0$ Clockwise

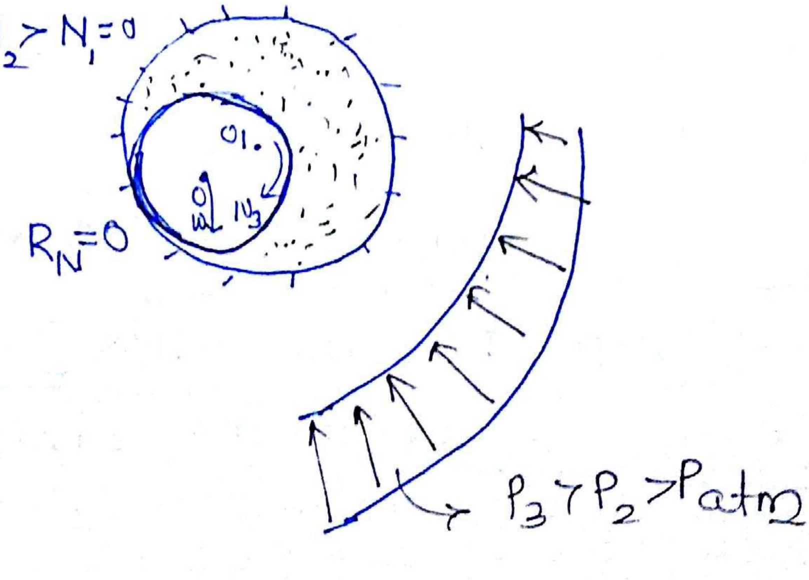


Case - II $N_2 > N_1 = 0$



Case-III $N_3 > N_2 > N_1 = 0$

[Justify]

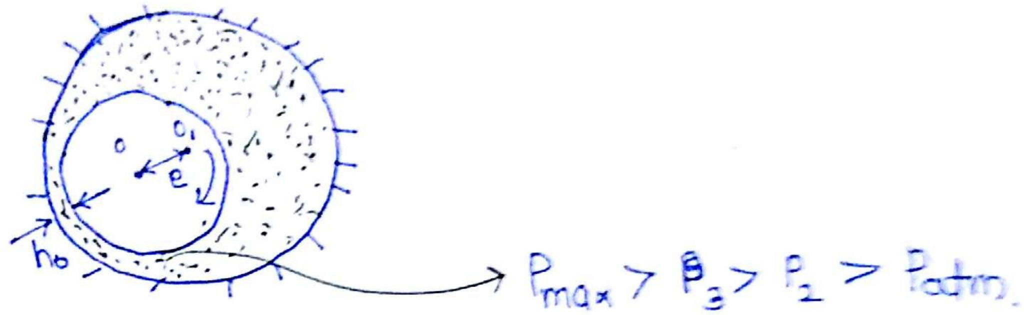


Case IV :- $N_{max} > N_3 > N_2 > N_1 = 0$

Journal bearing at Dynamic Cond^n

or

J.B. at max^n Running Cond^n



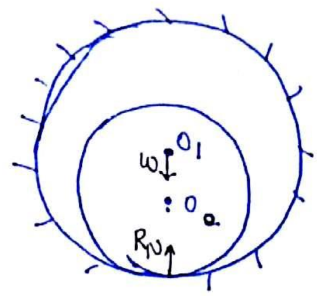
h_o = min film thickness

Conclusion :-

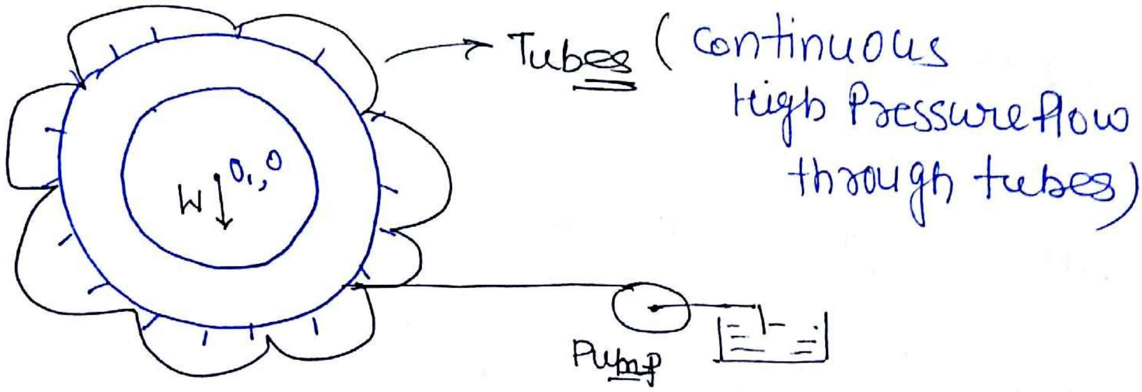
- As the speed of shaft increases pressure of lubricant also increases and this liquid lub. is converted into a sticky film.
- When this lub. enters from wider space to narrow space, pressure of lubricant become maximum this phinomena for lub. is known as waging action/dynamic action and convergent action for lubricant. and this high pressure is responsible to lift the shaft.

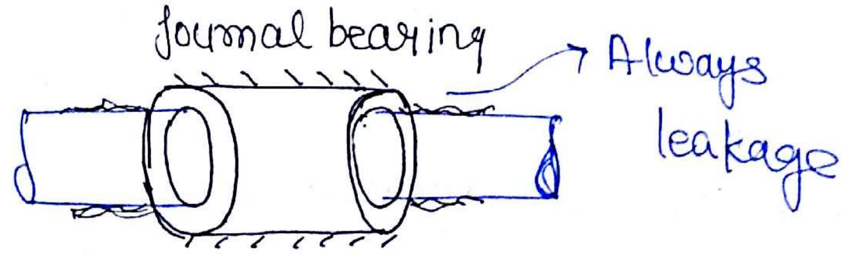
Hydro static lubrication :-

Case - I Stationary condition
(with out lubricant)



Case-II stationary Condn
(with lubricant)

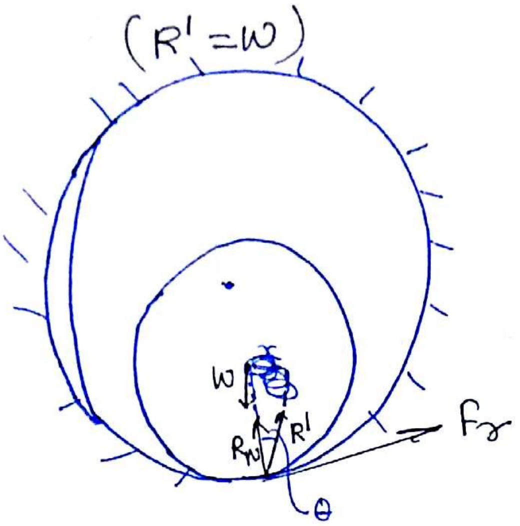


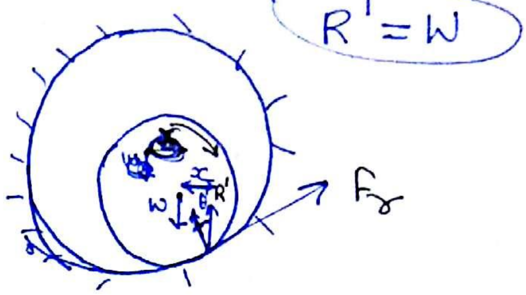


* In case of J.B., always leakage so continuous lubrication require

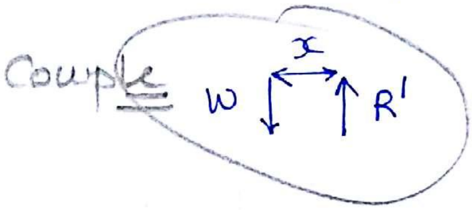
Hydrodynamic	Hydrostatic
① lub is supplied into the bearing at atmosphere condition.	① lub. is supplied at high pressure.
② Pressure of lub. increase due to waging action / dynamic action of lub.	② ↑ due to an external device (pump)
③ Motion of shaft is ecentric wrt bearing houshing	③ Cocentric Motion
④ Metal to Metal Contact is avoided only at high speed condition	④ at stationary condition only
⑤ starting torque is high	⑤ starting torque is less.
⑥ Cost of lubrication less	⑥ Cost of lubrication more
⑦ It is used in application where shaft is subjected to lighter load at stationary Cond^n	⑦ ~~It is used~~ ~~at~~ higher load at stationary Cond^n.
⑧ it is used in application like, IC engine crankshaft, steam & gas turbine, electric motor	⑧ It is used in heavy machinary like Bowl mill in power plant, Vertical turbo generator, large size Concrete.

Frictional torque and power loss in Journal Bearing:—
or Concept of friction circle in J.B.





W & R' forming a Couple in ACW dir^n



$$T_f = W x$$

$$\sin\theta = \frac{x}{R}$$

$$x = R\sin\theta$$

x = Friction Circle Radius

$$T_f = W R \sin\theta$$

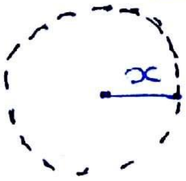
θ = Very less

$$\sin\theta = \theta = \tan\theta$$

θ = angle b/w resultant force & normal rxn
Hence $\mu = \tan\theta$

$$T_f = W R \tan\theta$$

$$\boxed{T_f = \mu W R}\ ,\quad R_{eff} = R$$



Friction circle (take centre at shaft centre &

Conclusion:

1. When shaft is in stationary Condition (absence of Friction). The reaction offerred by bearing is inlined with line of action of load acting from the journal.

2. When shaft is in motion (presence of friction) the resultant R' (resultant of friction & normal reaction) is deviated by a distance x from line of action of the load and this x is known as friction radius

3. Circle drawn from centre of the shaft by taking radius x is known as friction circle.

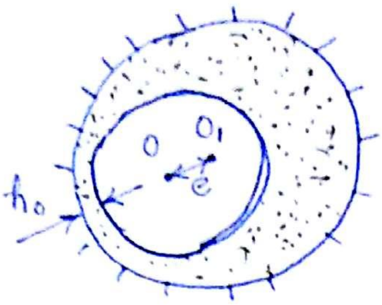
$$x = R \sin \theta$$

$$x = R \mu$$

$$x = f(R, \mu)$$

x depends on R & μ

Terminology used in Journal bearing:-



h_o = min film thickness.

Speed $\uparrow \rightarrow h_o \uparrow \rightarrow e \downarrow$

$W \uparrow \rightarrow h \downarrow \rightarrow e \uparrow$

* Viscosity $\uparrow \rightarrow h \uparrow \rightarrow e \downarrow$
(Z)

temp $\uparrow \rightarrow Z \downarrow \rightarrow h \downarrow \rightarrow e \uparrow$

Eccentricity :- Distance b/w centre of shaft and centre of bearing.

$$e + R + h_o = R_1$$

$$e = R_1 - R - h_o$$

$$e = c_1 - h_o$$

$$e = \frac{c}{2} - h_o$$

$C = D_1 - D_o$

$C_1 = R_1 - R_o$

$2C_1 = C$

Eccentricity Ratio (ε) :- It is define as ratio of eccentricity to the radial clearance.
(Attitude)

$$\epsilon = \frac{e}{c_1} = \frac{c/2 - h_o}{c/2}$$

$$\epsilon = 1 - \frac{2h_o}{c}$$

Bearing clearance :-

C_1 = Radial clearance $(R_1 - R_0)$

If nothing mention take clearance

C = Dimetrial clearance / clearance $(D_1 - D_0)$

Diameter to clearance ratio $= \frac{D}{C}$

Radius to clearance Ratio $= \frac{R}{C_1}$

C/D Ratio :-

if $C\uparrow \rightarrow \frac{C}{D}\uparrow \rightarrow (P_{loss})\downarrow \rightarrow W\downarrow$

$C\downarrow \rightarrow \frac{C}{D}\downarrow \rightarrow (P_{loss})\uparrow \rightarrow W\uparrow$

↳ load capacity

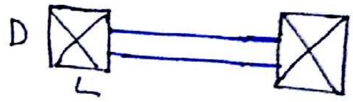
$$\boxed{0.001 \le \frac{C}{D} \le 0.002}$$

* L/D_1 Ratio :→ when find deflection in shaft use (L/D)

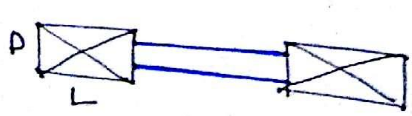
↳ Bearing dia.

Simple support
- short bearing $\frac{L}{D} < 1$

- square bearing $\frac{L}{D} = 1$

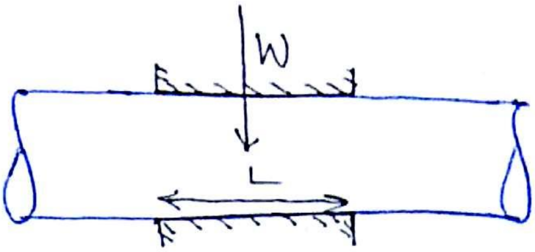


Fixed support
- long bearing $\frac{L}{D} > 1$

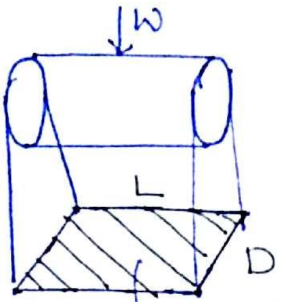


Bearing Pressure:-

{ Compression - Crushing
{ Tearing - tension



$$P_{ind} = \frac{\text{load}}{\text{Projected Area}}$$



$$P_{ind} = \frac{W}{LD}$$

safe condⁿ

$$P_{ind} \le P_{per}.$$

$$\frac{W}{LD} \le P_{per}.$$

$$\boxed{W_{max} = L\,D\,P_{per}.}$$ Strength of J.B.

Power loss:-

$$T_f = \mu W R$$

$$P_{loss} = T_f \times \omega$$

$$P_{loss} = \mu W R \times \omega$$

$$\boxed{P_{loss} = \mu W V}$$

Coefficient of Friction:-

* 'Mcc kee's eqⁿ'

$$\mu = f\left[\left(\frac{Zn}{p}\right), \left(\frac{D}{c}\right), \left(\frac{L}{D_1}\right)\right]$$

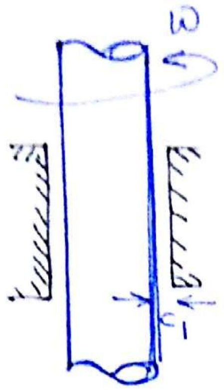
* Petroff's eqⁿ

$$\mu = 2\pi^2\left(\frac{Zn}{p}\right)\left(\frac{D}{c}\right) + K \quad [\text{by Petroff's eq}^n]$$

* $\frac{Zn}{p}$ = Bearing characteristic Number.
* Z = Absolute viscosity of lub. (Pa-s)
* Imp n = speed "In Revolution per sec." (rps)
* p = Bearing pressure $\left(\frac{W}{LD}\right)$
* K = leakage Factor

$$\begin{cases} K = 0.002, & \text{when } 0.75 \le \frac{L}{D_1} \le 2.8 \\ K = 0.003, & \text{when } L/D_1 \ge 2.8 \end{cases}$$

$$\boxed{\mu = 2\pi^2\left(\frac{Zn}{p}\right)\left(\frac{D}{c}\right)}$$ if K not Given



$\tau = \mu \frac{du}{dy}$ (in FM)

$\tau = Z \frac{dV}{dh}$

$V = \pi D n$ (rps)

$\frac{F_s}{A} = Z \frac{V}{C_1}$

$\frac{F_f}{\pi D L} = \frac{Z \pi D n}{C/2}$

Shear force = friction force.

$\Rightarrow \quad F_f = \frac{\pi D L Z \pi D n}{C/2}$

$T_f = F_f \times R$

$$T_f = \frac{\pi D L Z \pi D n . R}{C/2} \quad -(1)$$

$$T_f = \mu W R \quad -(2)$$ (by friction circle)

(1) = (2)

$$\mu W R = \frac{\pi D L Z \pi D n . R}{C/2}$$

$$\boxed{\mu = 2\pi^2 \left(\frac{Zn}{p}\right) \frac{D}{C}}$$

* Sommerfield Number :- Constant for a given bearing always.

Sommerfield number remains constant for a given bearing hence it is used to co-relate the working machines which are operating with same bearing

$$\boxed{S = \left(\frac{Zn}{p}\right)\left(\frac{D}{c}\right)^2}$$

$\boxed{S_1 = S_2}$ for a Given bearing

Imp for Gate:-

$$\boxed{\begin{aligned} T_f &= \mu W R \\ P_{loss} &= \mu W V \\ P_{ind} &= \frac{W}{LD} \\ \mu &= 2\pi^2\left(\frac{Zn}{P}\right)\left(\frac{D}{c}\right) \\ S &= \left(\frac{Zn}{P}\right)\left(\frac{D}{c}\right)^2 \end{aligned}}$$

* Film thickness depends on sommerfield Number

Question:- Natural feed ~~Gr~~ Journal bearing of diameter 50mm and length of 50 mm operates at 20 rps carries a load of 2KN. The lub. used has a viscosity of 20 mPa-s and the radial clearance is 50 micrometer (μm) the sommerfeild Number for bearing is.

Soln

$$S = \left(\frac{Zn}{P}\right)\left(\frac{D}{C}\right)^2$$

for ~~Gay~~ $D_1 \simeq D$ (C is very less)

$Z = 20$ mPa-s $\quad W = 2KN$

$n = 20$ rps $\quad D = 50mm, \; L = 50mm$

$P = \frac{W}{LD}$ $\quad C_1 = 50\,\mu M$

$C_0 = 2C_1$

$$S = \frac{20 \times 10^{-3} \times 20}{\frac{2000}{50 \times 50 \times 10^{-6}}}\left(\frac{50}{50 \times 2 \times 10^{-6}}\right)^2$$

$$S = \frac{2 \times 2 \times 10^{-3} \times 50 \times 50 \times 10^{-6}}{20} \times \frac{1}{4 \times 10^{-12}}$$

$S = 0.125$ Ans

Que:- A Jurnal bearing has a shaft dia is 40 mm & length is 40 mm & shaft rotate at 20 rad/s, Viscosity of lub is 20 mPa-s and radial clearance is 0.02 mm loss of torque due to viscocity is ?

$\omega = 20\ rad/s$

$n = \frac{\omega}{2\pi}$

Soln

$$\mu = 2\pi^2 \left(\frac{Zn}{P}\right)\left(\frac{D}{C}\right)$$

$$\mu = \frac{2\pi^2 \times 20\times10^{-3} \times 20 \times 40\times40\times10^{-6}}{W \times 2\pi} \times \left(\frac{40}{0.04}\right)^2$$

$$T_f = \mu W R$$

$$T_f = \frac{2\pi^2 \times 20\times10^{-3} \times 20\times40\times40\times10^{-6}}{\cancel{W} \times 2\pi} \times \left(\frac{40}{0.04}\right) \times \cancel{W} \times \frac{20}{10^3}$$

$$T_f = \frac{2\pi^2 \times 2\times2\times4\times40\times10^{-6} \times 40\times20\times10^{-3}}{0.04 \times 2\pi}$$

$$T_f = 0.04\ N\text{-}m \quad \text{Ans}$$

Q1103 A lightly loaded full JB has Journal diameter of 50 mm and bush bore of 50.05 mm and the bush length is 20 mm. if rotational speed of journal is 1200 rpm & lub. Viscosity 0.03 Pa-s the power loss in W will be.

Sol^n

$$P = T_f \times \omega$$

$$P = \mu W R \times \omega$$

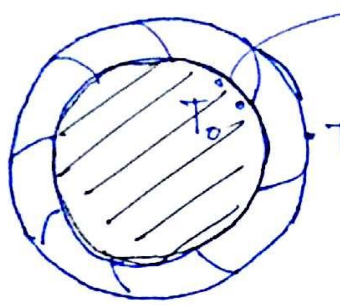
$$P = 2\pi^2 \left(\frac{Zn}{\frac{W}{LD}}\right)\left(\frac{D}{c}\right) \times W \times R \times \omega$$

$$P = 2\pi^2 \times 0.03 \times \frac{1200}{60} \times 0.02 \times 0.05 \times \left(\frac{0.05}{0.05 \times 10^{-3}}\right) \times 0.025 \times \frac{2\pi \times 1200}{60}$$

$$P = 37.2\ W$$

Q. 4.19

Mass Flow rate of Coolant require in Journal Bearing :-



P_{loss} = heat Generated

$HG = P_{loss} = \mu W V$

T_o = Max. temp that lub. can bear.

T_b = outer surface temp. of bearing.

T_a = Ambient temp

Heat Dissipation $= hA\Delta T$

$$HD = h\pi DL(T_b - T_a)$$

$\pi h = C_D \rightarrow$ heat dissipation coefficient

$$HD = C_D DL(T_b - T_a)$$

let us assume $T_b - T_a = \dfrac{T_o - T_a}{2} \Rightarrow T_b = \dfrac{T_o + T_a}{2}$

$$\boxed{HD = C_D DL\left(\frac{T_o - T_a}{2}\right)}$$

Heat dissipation at worst case when (max.) $T_b \simeq T_o$

if

Heat Dissipation > Heat Generated } No Coolant require

Heat Dissipation = Heat Generated

If Heat Dissipation < Heat Generated } Coolant Require

* Monitoring

1. Measuring vibration with the help of accelerometer
2. Measuring internal temp.

Hence $HG - HD = \dot{m}_c C_p (\Delta T)_{Coolant}$

$\dot{m}_c$ = known

Imp G-2017

Question:- General bearing has a shaft dia is 50 mm & length is 50mm operating at 900 rpm. The bearing is lubricated with and oil whose Absolute viscosity & operating temp. are 0.03 Pa-s & 75 °C

Assume $D/c = 1000$, $T_a = 25$ °C, $C_D = 600\ W/m^2K$

(i) Find rate of artificial Cooling require ($C_p = 1.8\ kJ/kgK$)

(ii) Calculate the mass flow rate of Coolant require if temp diff for outlet & Inlet for Coolant is 15 °C.

$$HG = \mu W V$$

$$\mu = 2\pi^2 \left(\frac{Zn}{P}\right)\left(\frac{D}{c}\right) \quad , \quad P = \frac{W}{L.D}$$

$$HG = 2\pi^2 \left(\frac{Zn}{W}\right) LD \left(\frac{D}{c}\right) W V \quad \pi D N/60$$

$$HG = 2\pi^2 \left(\frac{0.03 \times 900}{60}\right)(0.05)^2 (1000)\, \frac{\pi (0.05)(900)}{60}$$

$$HG = 52.32\, W$$

$$HD = C_D\, D L \frac{(T_o - T_a)}{2}$$

$$= 600 \times (0.05)^2 \frac{(75 - 25)}{2} \qquad = 37.5\, W$$

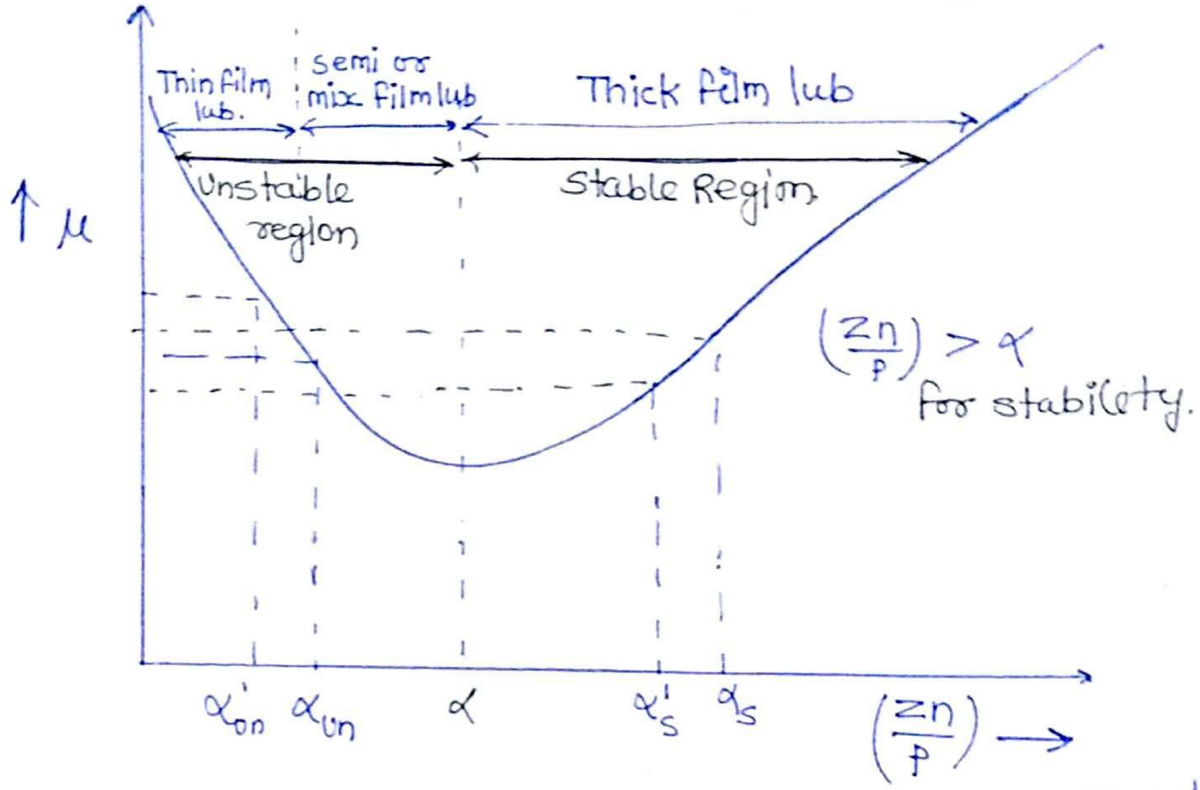
$$AC = HG - HD = 14.82\, W$$

$$\dot{m}\, C_{pc} (\Delta T)_{Coolant} = 14.82$$

$$\dot{m} \times 1.8 \times 15 = 14.82$$

$$\dot{m}_c = 0.54\ gm/se$$

Significance of Bearing characteristic Number and Bearing modulas :- Reynold's Analogy.



same graph with N (speed)

Unstable Region :-

if load $\uparrow \rightarrow P_{loss} \uparrow \rightarrow HG \uparrow, \rightarrow Temp \uparrow \rightarrow Z \downarrow$

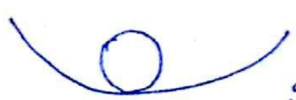
$Z \downarrow \rightarrow \left(\frac{Zn}{p}\right) \downarrow \rightarrow \alpha_{un} \downarrow \rightarrow \mu \uparrow \rightarrow P_{loss} \uparrow -$ temp $\uparrow$

unstable region.

stable region :-

if load $\uparrow \rightarrow P_{loss} \uparrow \rightarrow$ temp $\uparrow \rightarrow Z \downarrow$

$Z \downarrow \rightarrow \left(\frac{Zn}{p}\right) \downarrow \rightarrow \alpha_s \downarrow \rightarrow \underline{\mu \downarrow} \rightarrow \underline{P_{loss} \downarrow} \rightarrow \underline{temp \downarrow}$



stable region

hence

$$\downarrow \mu = 2\pi^2 \left(\frac{Zn}{P\uparrow}\right)\left(\frac{D}{c}\right)$$

is valid only for stable region.

in stable region

$$\downarrow\uparrow P_{loss} = \downarrow \mu W \uparrow V$$

* to achive more stability we increase Viscosity (Z)

Bearing Modulas (α) :-

min value of bearing char. No. for stability is → $\alpha = \left(\frac{Zn}{P}\right)_{min}$ for stability

↓

bearing modulas.

Bearing characteristic Number $= \left(\frac{Zn}{P}\right)$

For Practical case/Design :-

$$\left(3\alpha \le \left(\frac{Zn}{P}\right) \le 5\alpha\right)$$

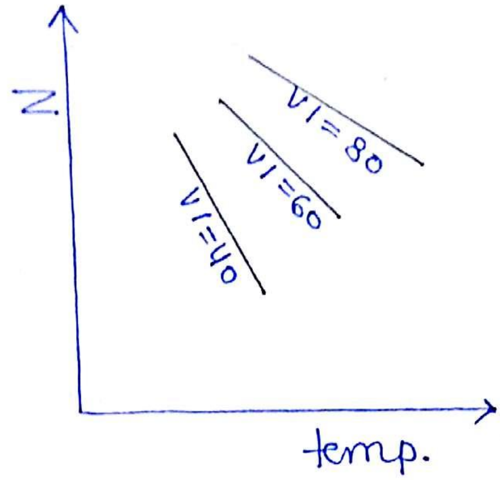
For static load :- $\left(3\alpha \le \frac{Zn}{P} \le 5\alpha\right)$

For High fatigue/impact load

$$\left(13\alpha \le \frac{Zn}{P} \le 15\alpha\right)$$

Viscosity Index :- Rate of change of viscosity w.r.t. temp. is known as viscosity index.

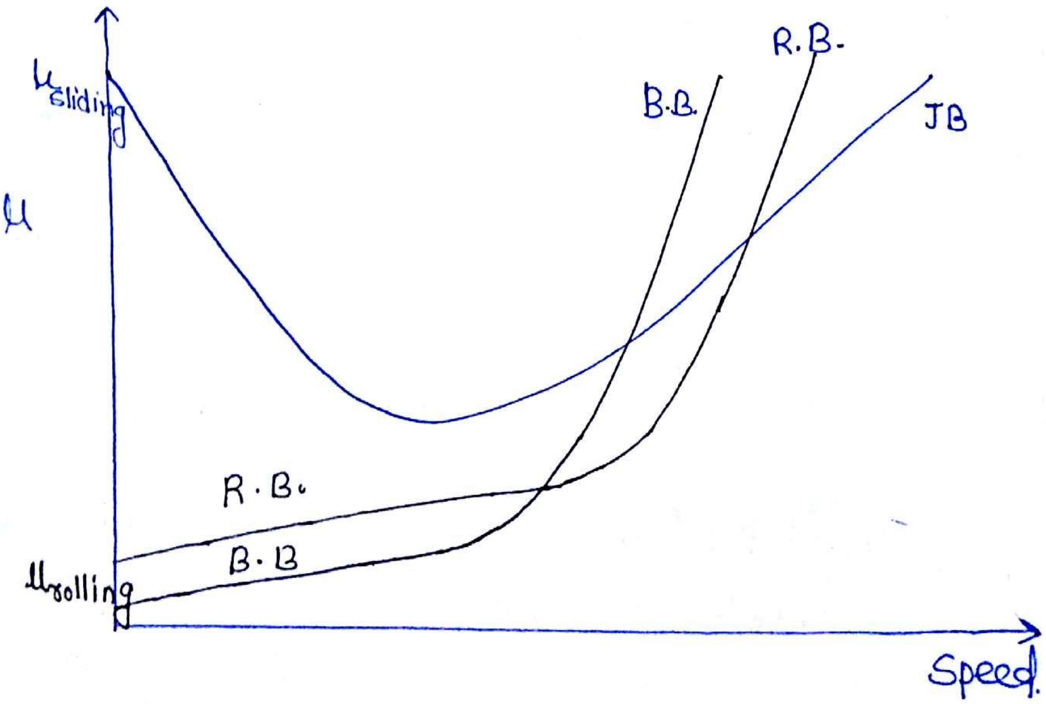
$$Z = A + \frac{B}{T}$$



$0 \leq VI \leq 100$

* Stability doesn't depands on viscosity index (VI)

Anti-Friction Bearing / Rolling Contact bearing :⇒



at high speed :- $\mu_{BB} > \mu_{RB} > \mu_{JB}$

Parameter	J.B.	AFB
1. Speed	Used forhigh speed application	Used for low speed application
2. Load.	only Radial load	both radial & axial load
3. Machine service	machine in Continuous service	interminant service require,(Frequent stoping and starting)
4. Noise	minimum in all bearing	maximum in All bearing.
5. Life	More (shaft will Rot in lub)	less (Metal to metal cont.)
6. starting torque	more	less.
7. Radial space	less space require	More space regire
8. Axial space	more	less
9. Cost	less	more. { Balls Powed metalargy Babbit material
10. Damping (Vib absorb. Capacity. (cap.	more	less.
11. lubrication	liquid lubricant and Continuous lubrication	semi-Solid lubricant (Grease)& periodic lubrication.
12. Type of Failure	Sudden failure without indication	Gives indication before failure by making more noise

	J.B	AFB
13. Application	- IC engine crank shaft - Steam and gas turbine - Concrete mixture - large electric motor	- machine tool spindle - automobile front and rare axes - Gear boxes - Small electric bearing.

Designation of AFB:- by a Company SKF

SKF- $\underset{\text{I}}{\underline{6}}$ $\underset{\text{II}}{\underline{3}}$ $\underset{\text{III}}{\underline{0\ 8}}$

deep groove

I (1-8) - type of bearing ← { 6 = DGBB, 3 = taper RB }

II - type of series (1 to 5)

III 08 X 5 = 40 mm - shaft dia.

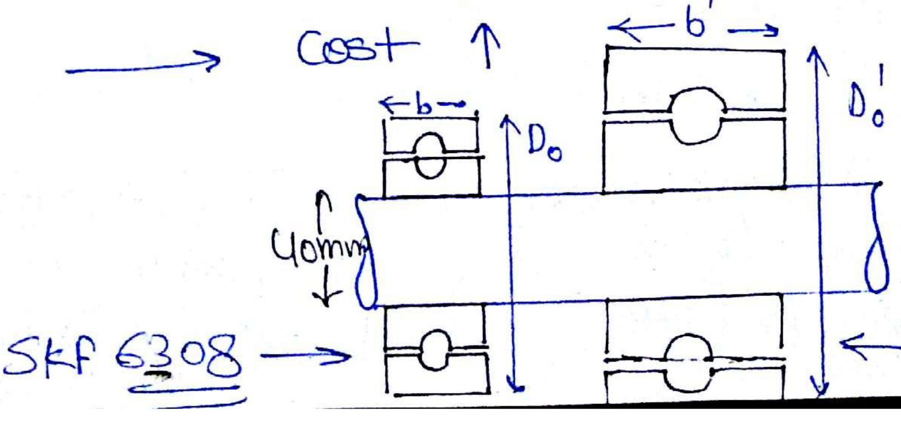
Type of Series:-

1	2	3	4	5
100 Series	200 series	300 Series	400 Series	500 Series
or	or	or	or	or
[Extralight Series]	[light Series]	[medium series]	[Heavy series]	[extra heavy series]

⟶ load capacity ↑

⟶ dim^n ↑

⟶ cost ↑



$D_o' > D_o$

$b' > b$

bearing		shaft diameter
SKF 6300	→	10 mm
SKF 6301	→	12 mm
SKF 6302	→	15 mm
SKF 6303	→	17 mm

by Rule ×5

SKF 6304	20 mm
SKF 6305	25 mm

Indian standard for bearing :-

eg IS 40 BC 03

- 40 → shaft dia
- BC → type of bearing [BC = deep Groove bearing.]
- 03 → type of series

Different terms use while selecting a series for AFB :-

① Equivalent load :- (P_e or P_m)

$$P_e \text{ or } P_m = S[XVF_r + YF_a]$$

S - service factor/shock factor
X - Radial load factor
Y - Axial load factor
V - race rotation factor
F_r - Radial load
F_a - Axial load.

S :- steady load / No. shock $\Rightarrow S = 1$

light shock $\Rightarrow S = 1.5$

moderate shock $\Rightarrow S = 2$

Heavy shock $\Rightarrow S = 3$

Extra heavy shock $\Rightarrow S = 3.5$

V :-

★ Inner race rotation $\Rightarrow V = 1$

Outer race rotation $\Rightarrow V = 1.2$

X, Y :-

Thrust BB → $X = 0, \quad Y = 1$

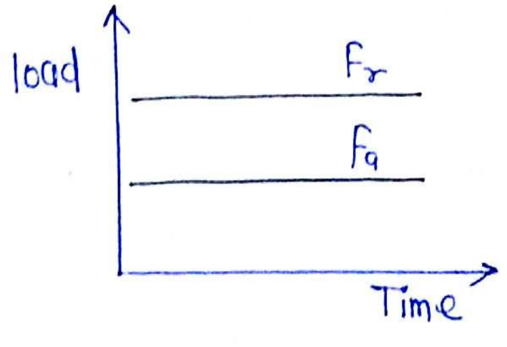
Cy. R.B. → $X = 1, \quad Y = 0$

DG BB → $X > Y$

taper RB → $X > Y$

Angular Contact BB → $Y > X$

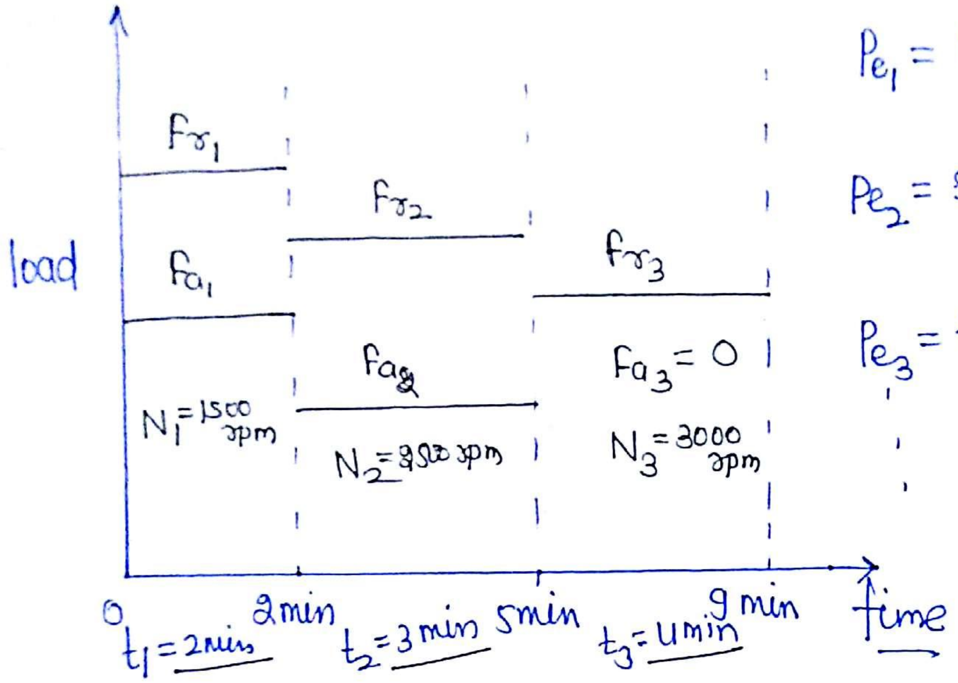
Imp Note :-



Speed = const

The Formula for equivalent load is only valid when load and speed remain constant wrt time

Cubical Mean load :-



$P_{e_1} = S_1 [X_1 V_1 Fr_1 + Y_1 Fa_1]$

$P_{e_2} = S_2 [X_2 V_2 Fr_2 + Y_2 Fa_2]$

$P_{e_3} = S_3 [X_3 Y_3 Fr_3]$

Cubical mean

$$P_e = \sqrt[3]{\frac{n_1 P_{e_1}^3 + n_2 P_{e_2}^3 + n_3 P_{e_3}^3 + n_4 P_{e_4}^3 \ldots}{n_1 + n_2 + n_3 + n_4 \ldots}}$$

n_1, n_2, n_3 .. → Are the no. of revolution that a bearing has undergone in respective region time t_1, t_2, t_3 ..

eg $n_1 = 1500 \times 2 = 3000$ rev.

$n_2 = 2500 \times 3 = 7500$ rev.

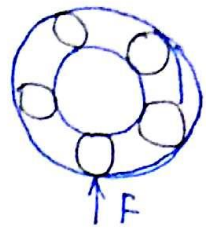
$n_3 = 3000 \times 4 = 12000$ rev.

Imp for Gate

Life of Anti Friction bearing :- life of AFB is define as number of revolution that a bearing has undergone before the evidance of first fatigue failure either in races or in the rolling element.

Fatigue → No yeilding

Fatigue fail → Fracture (crack)



life → No. of Revolution

eg 2000 hr, 600 rpm

life = 2000 × 60 × 600 = 72×10^6 rev.

Nominal life / Rated life / L90 / L10 / life / life with 90% Reliability :

[Always define for Group of identical bearing]

Nominal life or rated life of group of identical bearing is defined as a no. of revolution that 90% of group of identical bearing can serve or exceeds at a given speed without any failure.

Relationship between L and L_{90} :-

$$\frac{L}{L_{90}} = \left[\frac{\ln(1/R)}{\ln(1/R_{90})}\right]^{1/1.17}$$

R = Reliability

eg

$$\frac{L_{60}}{L_{90}} = \left[\frac{\ln(1/0.6)}{\ln(1/0.9)}\right]^{1/1.17}$$

$$L_{60} = 3.85\, L_{90}$$

eg

$$\frac{L_{50}}{L_{90}} = \left[\frac{\ln\left(\frac{1}{0.5}\right)}{\ln\left(\frac{1}{0.9}\right)}\right]^{1/1.7}$$

$$L_{50} = 5\, L_{90}$$

$\Downarrow$

Average life / Half life

Note

* Half life is five times the nominal life.

* $L_{10} = L_{90}$ L_{10} = means 90% realaibility or 10% will fail, 90% safe.

$L_{20} = L_{80}$

$L_{30} = L_{70}$

$L_{40} = L_{60}$

$L_{50} = L_{50}$

eg

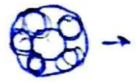
$$\frac{L_{20}}{L_{90}} = \frac{L_{80}}{L_{90}} = \left[\frac{\ln\left(\frac{1}{0.8}\right)}{\ln\left(\frac{1}{0.9}\right)}\right]^{1/1.17} \Rightarrow L_{80}/L_{20} = L_{90}$$

Dynamic load capacity / Dynamic load Rating :- (C)

* It is define as the maximum value of the load that 90% Group of identical bearing can serve 1 million revolution.

* Dynamic load capacity (C) remain constant for a given bearing.

{ Independent of no. of Rolling element



* $P_e > C \Rightarrow L_{90} < 1\ MR$

  $P_e = C \Rightarrow L_{90} = 1\ MR$

  $P_e < C \Rightarrow L_{90} > 1\ MR$

*

Million Revolution

$$\boxed{L_{90} = \left(\frac{C}{P_e}\right)^K} \; MR$$

Million Rev. 10^6

$K = 3$ Ball bearing

$K = \frac{10}{3}$ Roller bearing

$\frac{C}{P_e} \Rightarrow$ loading Ratio

In Gate

$$L_{90} = \left(\frac{C}{P_e}\right)^K$$

$K = 3$ BB

$K = 10/3$ RB.

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Q. 4.8

$C = 22\,KN$

$N = 600\ rpm$

$t = 2000\ hrs.$

$life = 2000 \times 60 \times 600 = 72 \times 10^6\ rev.$

$= 72\ MR$

$L_{90} = \left(\frac{C}{P_e}\right)^K$

$72 = \left(\frac{22}{P_e}\right)^3 \Rightarrow P_e = 5.28\ KN$

$P_e \to F_r \to max$

$P_e \to F_a \to 0$

$F_r = 5.28\ \underline{KN}$

4.12

$t = 8000\ hr$

$k = 3$

$L_{90} = \left(\frac{C}{P_a}\right)^K = \left(\cancel{\frac{C}{P_{e_1}}}\right)^K$

$L \propto \frac{1}{P_e^3}$

$\left(\frac{8000}{L}\right)^{1/3} = \frac{2\cancel{f}}{\cancel{f}}$

$L = 1000\ \underline{hr}$

4.16

$P_{ep} = 30$ KN

$P_{eQ} = 45$ KN

$$\frac{L_P}{L_Q} = \left(\frac{C}{P_{ep}} \times \frac{P_{eQ}}{C}\right)^3 = \left(\frac{45}{30}\right)^3 = \frac{27}{8}$$

Q. 4.18

$C = 16$ KN $\rightarrow$ 1 MR

$$L_{90} = \left(\frac{C}{P_e}\right)^3 MR$$

$$= \left(\frac{16}{2}\right)^3 MR = 512 \times 10^6$$

4.20

$$F L^{1/3} = K$$

$$(2)(540)^{1/3} = C(1)^{1/3}$$

$$C = 16.28 \text{ KN}$$

4.1

$$(3000 \times 60 \times 1000)^{1/3} \times (9800) = (t \times 60 \times 2000)^{1/3} \times (4900)$$

$$t = 12000 \text{ hr}$$

$$L_{90} \propto \frac{1}{P_e^3}$$

$P_e = 4900$

$L_g' = 8 \times 180$ MR

$h \times 60 \times 200 = 8 \times 180 \times 10^6$

$h = 12000$ hr

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Question :- A Deep groove BB has a dynamic load capacity 40500 N. operates on following work cycle

(i) radial load of 15000 N at 500 rpm for 25 % of the time

(ii) radial load of 10,000 N at 700 rpm for 50% of the time

(iii) radial load of 7000 N at 400 rpm for remaining time calculate the expected half life of the bearing in hours.

Soln

$$P_e = S[XVF_r + YF_a] \qquad F_a = 0$$

$$P_e = S\,XV\,F_r$$

$$P_{e_1} = 1 \times 1 \times 1 \times 15000\ N = 15000\ N$$

$$P_{e_2} = 10000\ N$$

$$P_{e_3} = 7000\ N$$

$$P_e = \sqrt[3]{\frac{(0.25 \times 500)(15000)^3 + (0.50 \times 700)(10000)^3 + (0.25) \times 400 \times (7000)^3}{(500 \times 0.25) + (0.50 \times 700) + (0.25 \times 400)}}$$

~~Pe = 22600 N~~ P_e = ~~1119.92 N~~ 11192.38N

$$L_{90} = \left(\frac{C}{P_e}\right)^3$$

$$L_{90} = \left(\frac{40500}{\cancel{111.92}\ 11192.38}\right)^3 = 47.32\ MR$$

$$h \times 60 \times N_{avg} = 47.32 \times 10^6$$

~~h x60 x~~

$$N_{avg} = \frac{\text{total no. of Revolution}}{\text{total time}}$$

$$= \frac{500 \times \frac{1}{4} + 700 \times \frac{1}{2} + 400 \times \frac{1}{4}}{1}$$

$$N_{avg} = 575\ rpm$$

$$h \times 60 \times 575 = 47.32 \times 10^6$$

$$h = 1373.3\ hrs.$$

$$L_{90} = 1373.3\ hrs$$

$$\text{half life} = L_{90} \times 5$$

$$= 1373.3 \times 5$$

$$= 6866.5\ hr$$

Question A DG BB is anticipited to have a life of 400 MR under a load of 10 KN with 80% reliability.

(i) Find out the life L_{30} under a load of 20 KN

(ii) Find out the life with 60% relaiability under a load of 15 KN.

Sol^n

$$L_{80} = 400\,MR \qquad P_e = 10\,KN$$

$$\frac{L_{80}}{L_{90}} = \left[\frac{\ln\left(\frac{1}{0.8}\right)}{\ln\left(\frac{1}{0.90}\right)}\right]^{1/1.17} = \frac{400}{L_{90}}$$

$$L_{90} = \cancel{361.69\,MR}$$
$$210.62\,MR$$

(i)

$$L_{90} = \left(\frac{C}{P_e}\right)^3 \qquad L_{90} \propto \left(\frac{1}{P_e}\right)^3$$

$$\cancel{361.69 = \left(\frac{C}{10}\right)^3} \Rightarrow \cancel{C = 71.23\,KN}$$

(i)

$$\frac{L'_{30}}{L'_{90}} = \left[\frac{\ln\left(\frac{1}{0.70}\right)}{\ln\left(\frac{1}{0.9}\right)}\right]^{1/1.17}$$

$P_e' = 20\,KN$

$$L'_{90} = \frac{210.62}{8}\,MR = 26.32\,MR$$

$$\frac{L'_{30}}{L'_{90}} = \left(\frac{\ln\left(\frac{1}{.7}\right)}{\ln\left(\frac{1}{0.9}\right)}\right)^{1/1.17}$$

$$L'_{30} = 74.6\ \underline{MR} \quad \text{Ans}$$

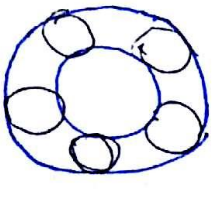
(ii) $P''_e = 15\ KN$

$$L''_{e90} = \frac{210.62}{(1.5)^3} = \underline{\underline{62.4\ MR}}$$

$$\frac{L''_{60}}{L''_{90}} = \left(\frac{\ln\left(\frac{1}{0.6}\right)}{\ln\left(\frac{1}{0.9}\right)}\right)^{1/1.17}$$

$$L''_{60} = 240.5\ \underline{\underline{MR}} \quad \text{Ans}$$

Static load capacity :⇒ Used for theoretical design/static design.
(Basic load capacity)



$\sigma_{ind} = \frac{P}{n.A}$

n = No. of Rolling element

Safe condⁿ

$\sigma_{ind} \leq \sigma_{per}$

$\frac{P}{nA} \leq \sigma_{per}$

$$\boxed{P_{max} = n.A.\sigma_{per}}$$

Basic load cap.

* life is depandent only one the no. of revolutim in actual)