LECTURE-1
9/06/2017

MACHINE DESIGN

Enggineering Design :-

Need or Requirement $\xrightarrow{\text{Process}}$ Complete or fullfill

Ⓒ Configuration | Ⓜ Material | Ⓟ Manufacturing process.

Engg. Design is an iterative decision making activity to produce a Plan | drawing to convert resources optimally into a product or device to satisfie the human need.

The ultimate aim of design is to perepare a drawing (i.e to be a selection of appropriate shape, appropriate material, calculation of appropriate dimension and the selection of manufacturing details) in such a way that the resulting m/c component should perform given functionality satisfactorly.

Steps used in desiguing a machine element :-

(1) Specify the function of the machine element.

(2) Define various load acting on the machine component while performing its functionality.

(3) Selection of appropriate shape. (all geometrical properties are known)

(4) Selection of appropriate material. (all mechanical properties are known)

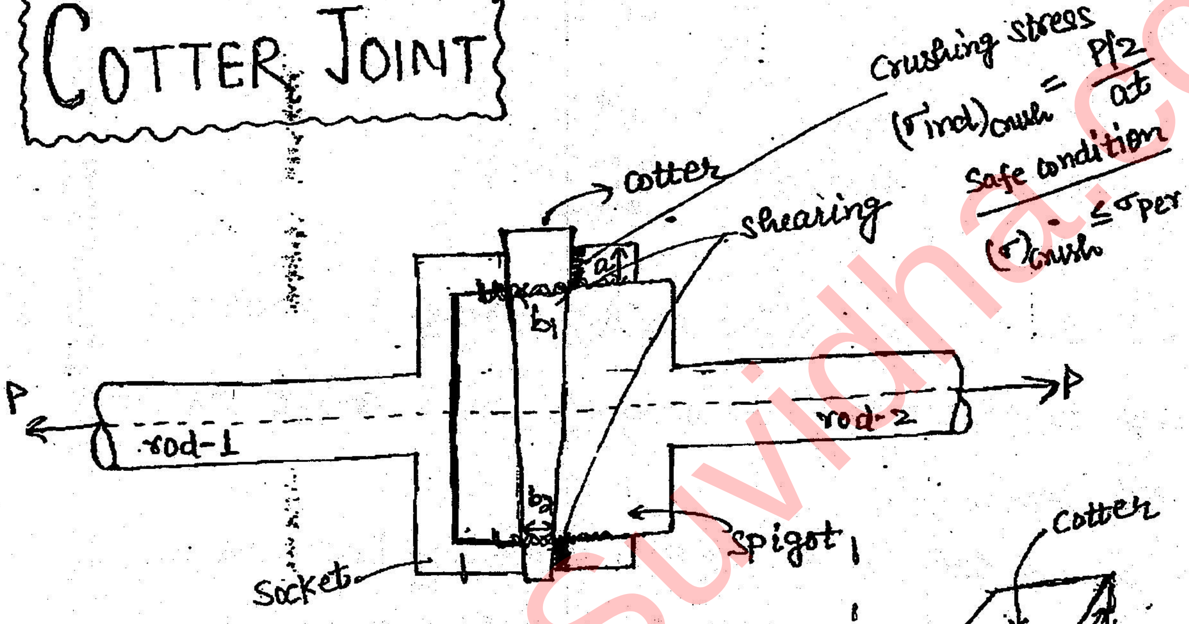
5) Determine mode of failure. [failure stresses are known] or [Theory of failure are known]

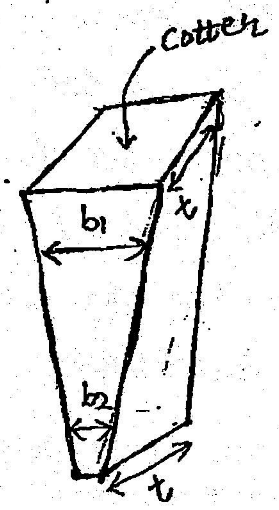
6) Calculation of dimensions by using SOM equations.

7) Define manufacturing process.

8) prepare a drawing or chart.

COTTER JOINT





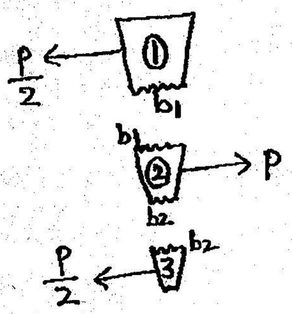
* cotter is a temporary fastner which is inserted b/w ~~two coaxial rod~~ to transmit axial load from one rod to another rod.
* Taper is provided into the width due to easy fastning purpose.

NOTE :- cotter joints are weak in shear.

SHEAR Design of cotter :-

$$\tau = \frac{\text{Shear force}}{\text{Sheared area}}$$

Double shear Failure



$$\tau_1 = \frac{P/2}{b_1 t} = \frac{P}{2b_1 t}$$

$$\tau_2 = \frac{P}{b_1 t + b_2 t}$$

$$\tau_3 = \frac{P/2}{b_2 t} = \frac{P}{2b_2 t} \quad [\tau_{max}]$$

Safe condition

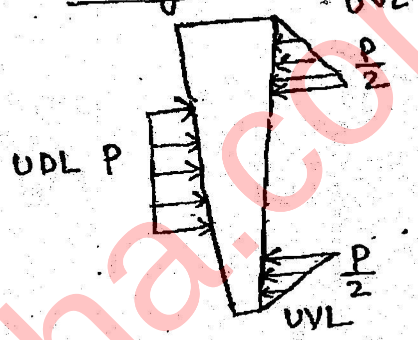
$\tau_{max} \leq \tau_{per}$

$$\boxed{\frac{P}{2b_2 t} \leq \tau_{per}}$$

$P \leq 2b_2 t \; \tau_{per}$

Shear Strength of cotter $\boxed{P_{max} = 2b_2 t \cdot \tau_{per}}$

Bending stress

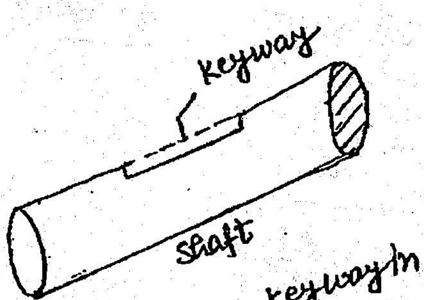


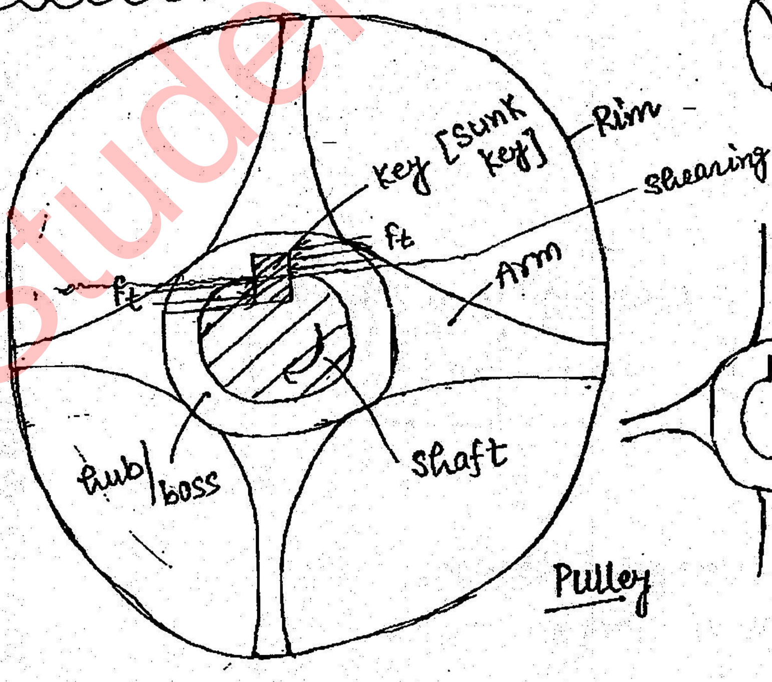
$$(\sigma_b)_{max} = \frac{M_{max} \times Y_{max}}{I_{NA}}$$

$$\boxed{(\sigma_b)_{max} \leq \sigma_{per}}$$

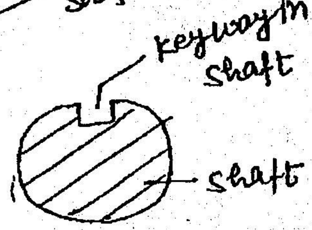
* Cotter is also subjected to crushing and Bending stresses.

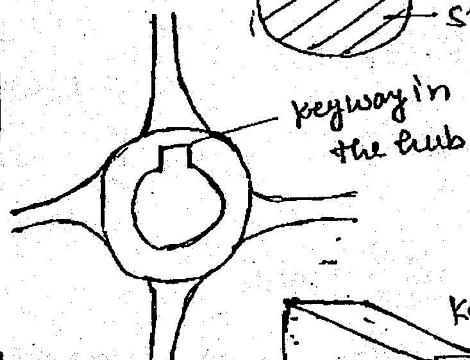
KEYED JOINT

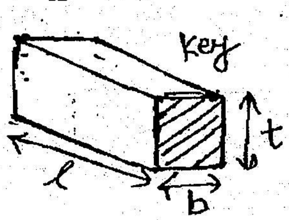




Pulley



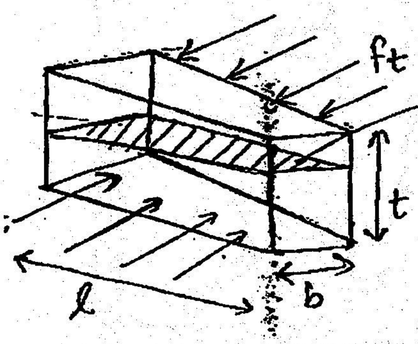




Key is the temporary fastner which is inserted b/w shaft and its assembly (Gear, flywheel, pulley etc) to transmit power by preventing relative motion b/w them.

* Key is the weakest element among shaft, assembly and key.

SHEAR Design of key :-



Single shear failure

$$\boxed{\tau_{key} = \frac{f_t}{bl}}$$

$$P = \frac{2\pi NT}{60}$$

T = known

$$f_t \times \frac{d}{2} = T \qquad d = \text{shaft dia}$$

$$f_t = \frac{2T}{d}$$

$$\boxed{\tau_{key} = \frac{2T}{dbl}}$$

Safe condition

$$\tau_{key} \le \tau_{per}$$

$$\frac{2T}{dbl} \le \tau_{per}$$

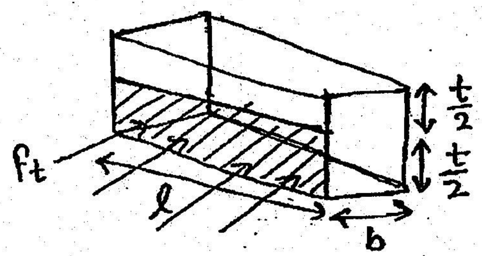
[Generally if b unknown then $\boxed{b = \frac{d}{4}}$]

$$T \le \frac{dbl}{2}\tau_{per}$$

$$\boxed{T_{max} = \frac{dbl}{2}\tau_{per}}$$ Shear strength of key in terms of Torque.

Crushing Design of Key :-

$$(\sigma_{ind})_{crush} = \frac{F_t}{\frac{t}{2} l}$$



and $T = F_t \times \frac{d}{2}$

$$\boxed{(\sigma_{ind})_{crush} = \frac{4T}{dtl}}$$

Safe condition :-

$(\sigma_{ind})_{crush} \le \sigma_{per}$ $\qquad \sigma_{per} = \frac{S_{yt}}{N}$

$\qquad \tau_{per} = \frac{S_{ys}}{N}$

$$\frac{4T}{dtl} \le \sigma_{per}$$

$$T \le \frac{dtl\,\sigma_{per}}{4}$$

$$\boxed{T_{max} = \frac{dtl\,\sigma_{per}}{4}}$$ Crushing strength of key in terms of Torque.

Eg. b, t, l = known

$(T_{max})_{shear} = 70$ KN-mm

$(T_{max})_{crush} = 60$ KN-mm

Strength of key = 60 KN-mm

Actual strength of key = min $[(T_{max})_{shear}, (T_{max})_{crush}]$

eg: b, t = known

$l = ?$

By shearing $l = 10\,mm$

By crushing $l = 12\,mm$

Now $\boxed{l = 12\,mm}$

KEY

Sunk key | Saddle key

Sunk key

- Half part of the key is in the hub and another in the shaft.
- Hence key way is present in both hub and shaft.
- key is resposible to transmit the power.
- power transmission capacity is more.
- Key way present in the shaft hence stress concentration factor are more therefore strength decreases and cost increases.

Saddle key

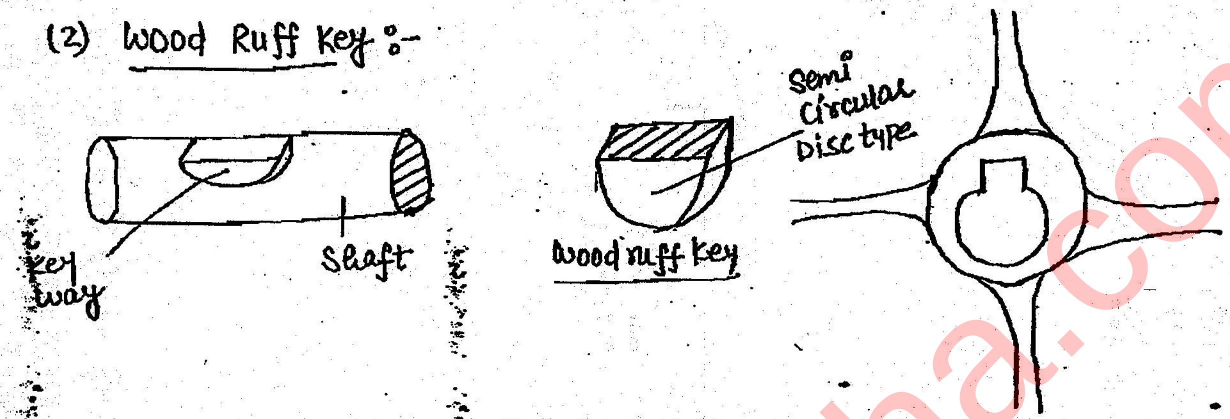
* key is only in the hub.
* There is no key way in the shaft and keyway present only in the hub.
* Friction force is responsible to transmit the power.
* power transmission capacity is less.
* Because there is no key way in shaft hence stress concentration factor are less therefore shaft strength increases and cost decreases.

Types of Sunk Key :-

(i) Feather Key :- Key is fixed either with the shaft or with the hub. It is a type of parallel key.

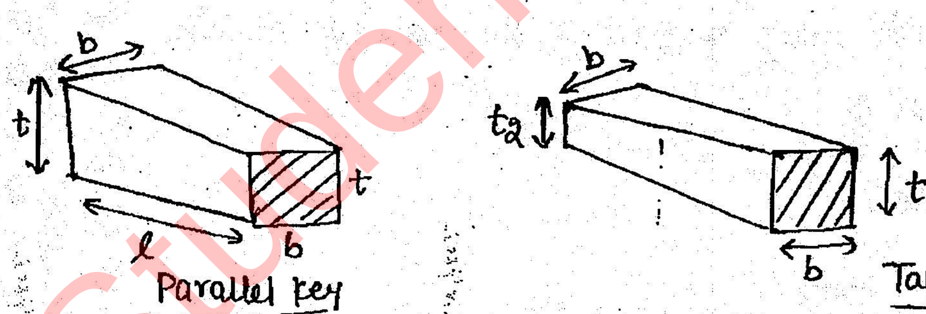
⇒ permit axial relative motion b/w shaft and assembly.

(2) Wood Ruff Key :-



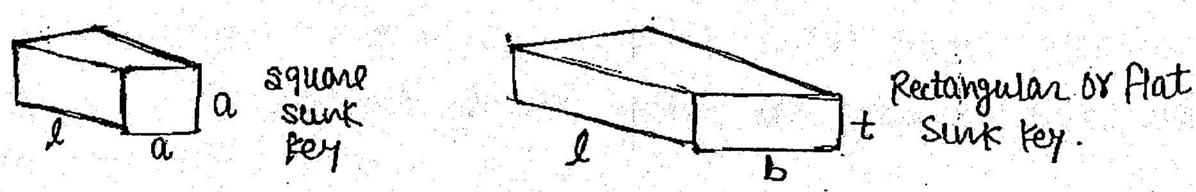
* This key contains semi circular Disc type portion.
* Hence key way is also semi circular Disc type.
* it can align itself hence used in tapered shaft.
* Extra depth of the key in the shaft provide more power transmission capacity.

(3) Parallel and Taper Key :-



Taper is provided in the thickness for easy fastning purpose

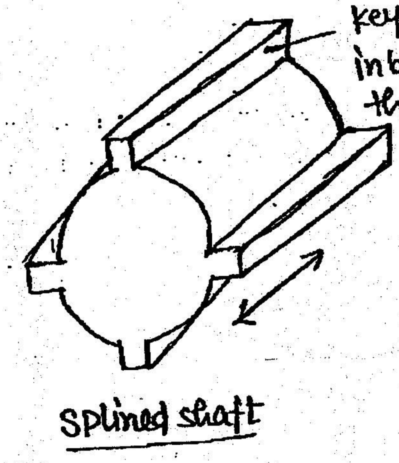
(4) Square Sunk key and Rectangular (Flat) sunk key :-

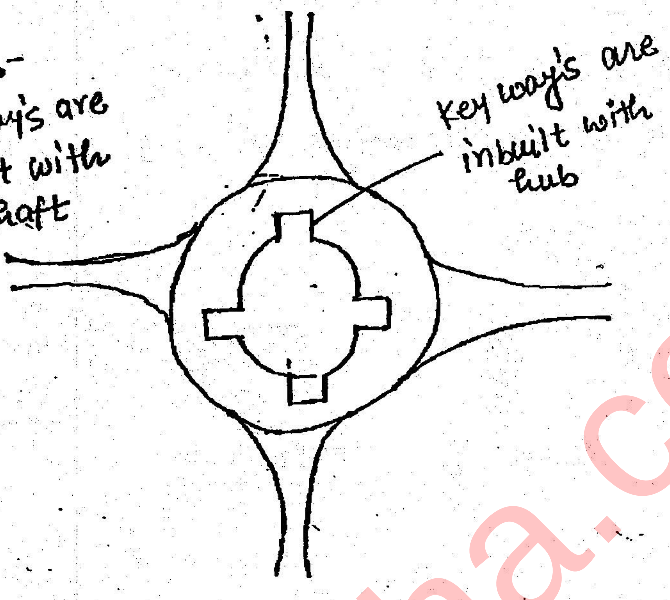


Rectangular sunk key is more stable than square key and used in most industrial application.

→

Splined key or splined shaft :-

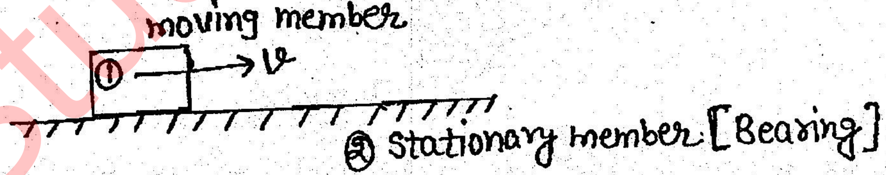




* Permit axial relative motion b/w shaft and assembly
* Used in automobile gearboxes and clutches.

BEARING

Whenever a relative motion occurs b/w two machine elements the machine element which is stationary supporting the moving machine element is reffered as bearing.



Bearing according to the shaft :-

Bearing is defined as a machine element whose function is to support rotating element (shaft, axle) and to guide or confined its motion while preventing the motion into the direction of applied load.

* Due to relative motion b/w shaft and bearing there is always Power loss occur in overcoming the frictional resistance and also wear occurs due to metal to metal contact.

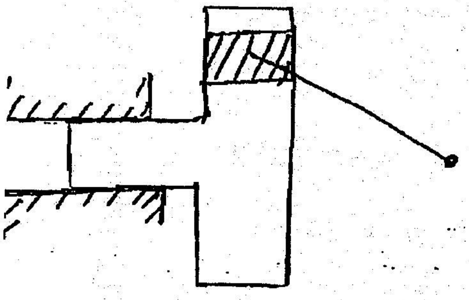
A bearing is said to be a good bearing when perform its functionality with minimum loss and wear and this is provided by using lubricant.

Classification of Bearing :-

- Bearing
  - Sliding Contact Bearing
    - Sleeve Bearing
    - Slipper Bearing or Guide Bearing
    - w.r.t Direction of Load
      - Radial load Bearing OR Journal Bearing
        - Full J.B
        - Partial J.B
        - Fitted J.B
      - Axial load or Thrust Bearing
        - Pivot (Foot step) Bearing
          - Flat Pivot Bearing
          - Conical Pivot Bearing
        - Collar Bearing
          - Flat Collar
          - Conical Collar
    - w.r.t type of Lubrication
      - Thin film lubrication (Boundary lubrication)
      - Thick film lubrication
        - Hydro dynamic lubrication
        - Hydro static lubrication
      - Zero film or Solid lubrication
    - w.r.t type of bushes
      - Solid bushes
      - Split bushes
  - Rolling contact Bearing OR Antifriction Bearing
    - Ball Bearing
      - Deep groove Ball Bearing B.B (Noise less)
      - Angular contact B.B
      - Self Aligning B.B
      - Thrust B.B
    - Roller Bearing
      - Cylindrical Roller Bearing
      - Spherical Roller Bearing
      - Taper R.B
      - Needle R.B

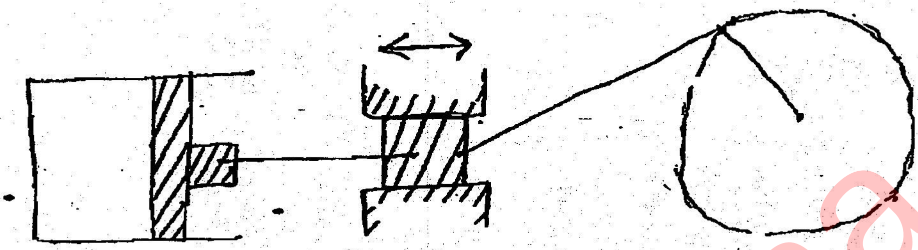
Slipper Bearing or Guide Bearing :-

Sliding occurs in a straight line direction



(1)

(2)



(3)

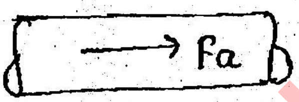
Cross head of EC Engine

Sleeve Bearing :-

Sliding occurs in a circular direction implies that about the periphery of circular or spherical

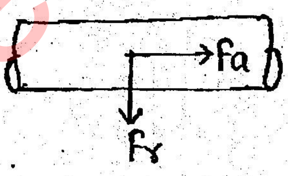
Axial load Bearing or Thrust Bearing

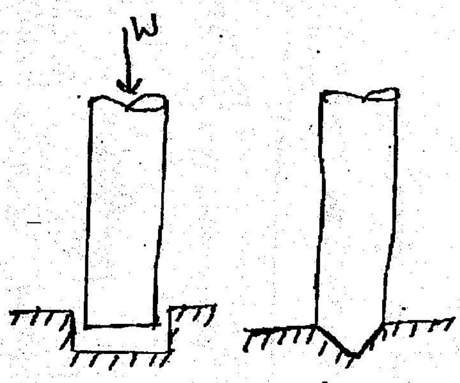
Pivot (foot step)



Only axial load

$F_r = 0$

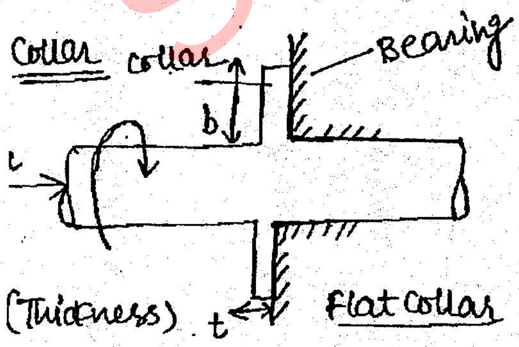




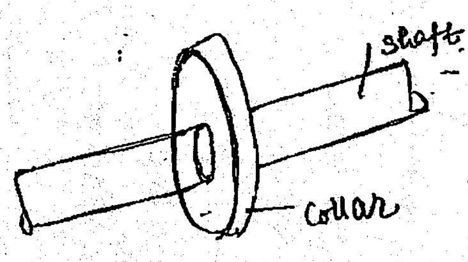
Flat Pivot

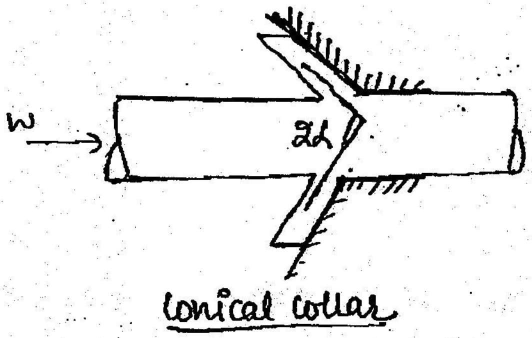
Conical Pivot

Collar



$\Rightarrow$

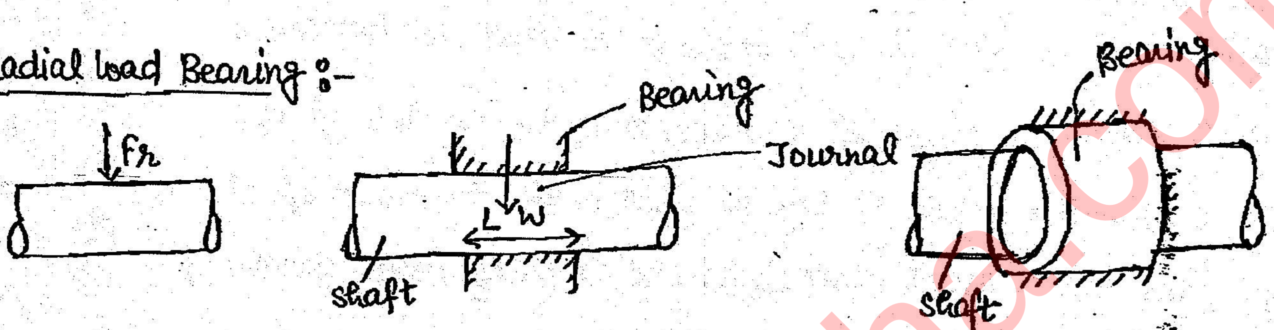


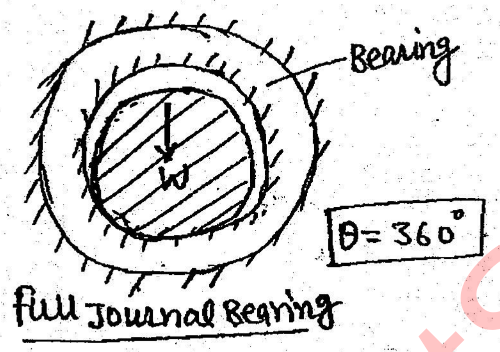


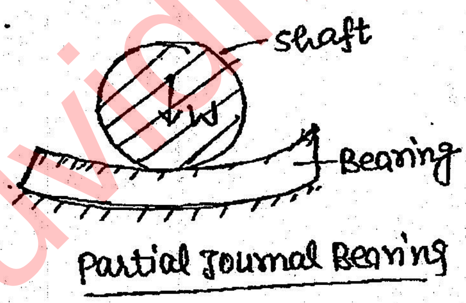
$2\alpha \rightarrow$ Cone angle

$\alpha \rightarrow$ Semi cone angle

Radial load Bearing :-





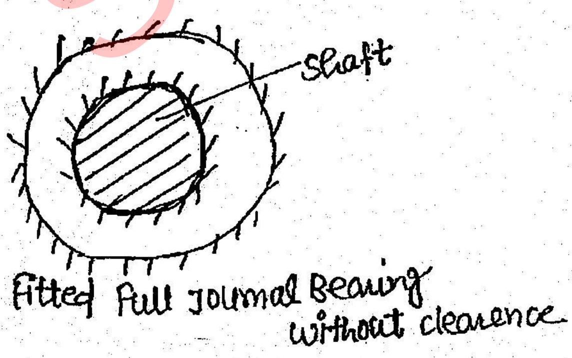


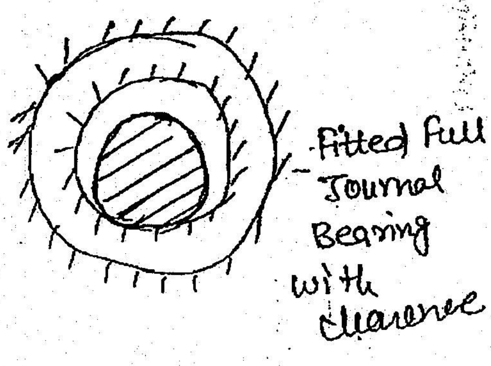
* Partial Journal Bearing is used when load is acting only in one direction.

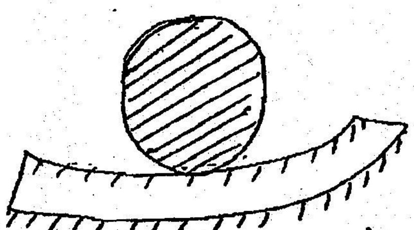
Fitted journal Bearing

Bearing dia > Journal dia with clearence

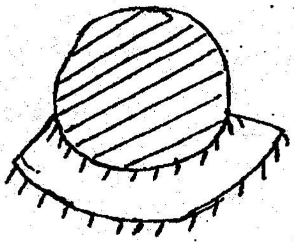
Bearing dia = Journal dia without clearence







Fitted Partial Journal Bearing with clearence



Fitted Partial Journal Bearing without clearence

Q. Square key of a side $\frac{d}{4}$ and length ℓ is used to transmit torque T from the shaft of a diameter d to the hub of a pulley. assume the length of key is equal to the thickness of the pulley. find the average shear stress and crushing stress developed in the key.

Ans:

$$\tau_{key} = \frac{2T}{d\,b\,\ell}$$

$$= \frac{2T}{d \cdot \frac{d}{4} \cdot \ell}$$

$$\boxed{\tau_{key} = \frac{8T}{d^2\ell}}$$

$$\sigma_{ind} = \frac{2T}{d \cdot \frac{t}{2} \ell}$$

$$= \frac{2T}{d \cdot \frac{d}{8} \cdot \ell}$$

$$\boxed{\sigma_{ind} = \frac{16T}{d^2\ell}}$$

LECTURE-2
10/06/2017

Thin film Lubrication :-

metal to metal contact always present.

Thick Film lubrication :-

metal to metal contact is avoided.

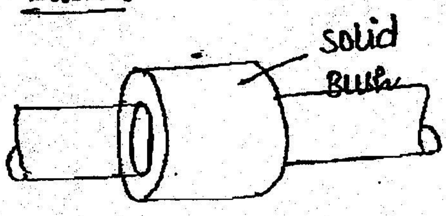
Hydrodynamic Lubrication :-

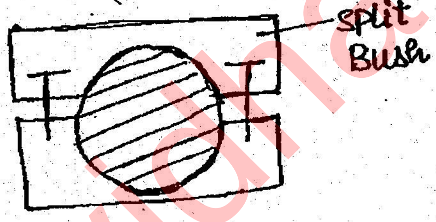
Metal to metal contact avoided only at high speed.

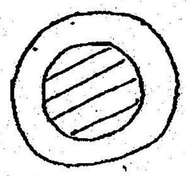
Hydrostatic Lubrication :-

metal to metal contact is avoided at stationary condition.

Bush :-







⇒ Plummer block is used to support long shaft.

Antifriction Bearing :- [$\mu_{rolling} <<< \mu_{sliding}$]

Ball Bearing :- (point contact)

Deep groove Ball Bearing :-
(Noise less bearing)

$$\boxed{\frac{F_r}{F_a} > 1}$$

Angular Contact Ball Bearing :-

$\boxed{\frac{F_r}{F_a} < 1}$

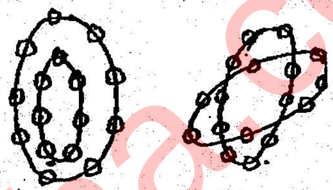
$\boxed{[\text{Load capacity}]_{\text{Angular contact}} > [\text{Load capacity}]_{\text{Deep Groove B.B.}}}$

* Two Bearings required to wear axial load in both direction.

Self Aligning Ball Bearing :-

: permit small misalignment of the shaft with respect to the Bearing housing.

mechanism → 2 raw of moving Balls.



Thrust Ball Bearing :

$\boxed{F_a}$ $\boxed{F_r = 0}$

Roller Bearing :- [Line contact]

Cylindrical Roller Bearing :-

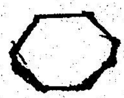
· Max^m load carrying capacity in a given radial space.

$\boxed{F_r}$ $\boxed{F_a = 0}$

$\boxed{\frac{L}{D} < 1}$

Spherical Roller Bearing :-

: · permit small misalignment of the shaft with respect to the Bearing housing.

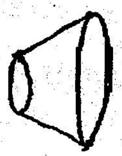


Mechanism → 2 raw of moving rollers are used.

load bearing → $\boxed{F_a}$ and $\boxed{F_r}$

$\boxed{\frac{L}{D} < 1}$

Taper Roller Bearing :-



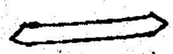
$$\boxed{\frac{F_r}{F_a} > 1}$$

$$\boxed{\frac{L}{D} < 1}$$

Needle Roller Bearing :-

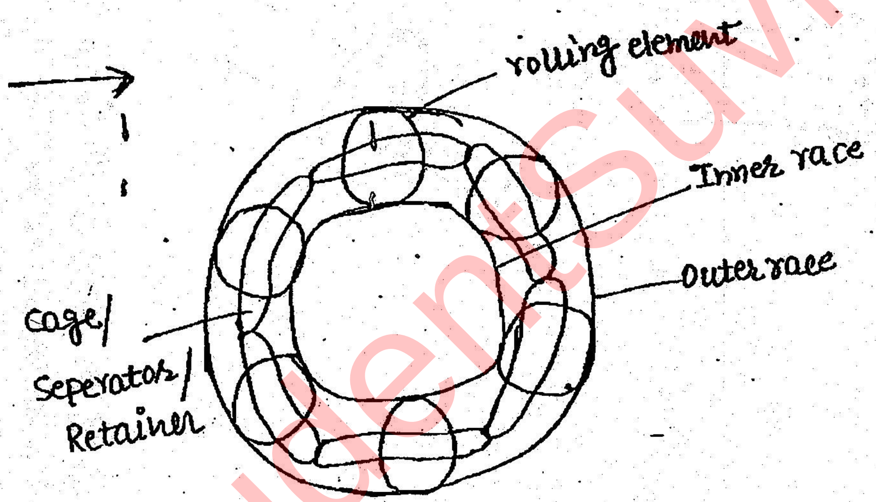
Needle roller are used where radial space are constrained.

$$\boxed{\frac{L}{D} > 1}$$



If $F_r \uparrow\uparrow\uparrow \rightarrow$ Cylindrical roller Bearing

$F_a, F_r \uparrow\uparrow\uparrow \rightarrow$ Taper roller Bearing [Impact or Fatigue load]



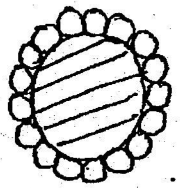
Functions of cage/seperator/Retainer :-

⇒ cage is used to avoid clustering of the rolling elements at one location.

⇒ cage is used to maintain constant relative angular position b/w two adajacent rolling element.

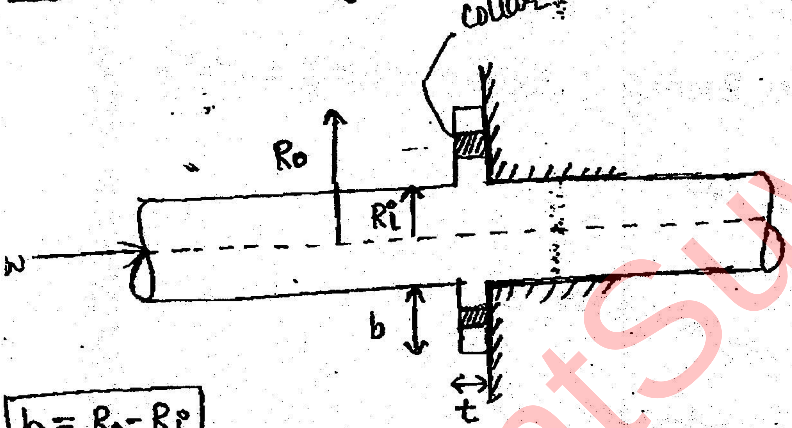
⇒ cage is used to avoid metal to metal contact b/w two rolling element to minimizes losses and wear.

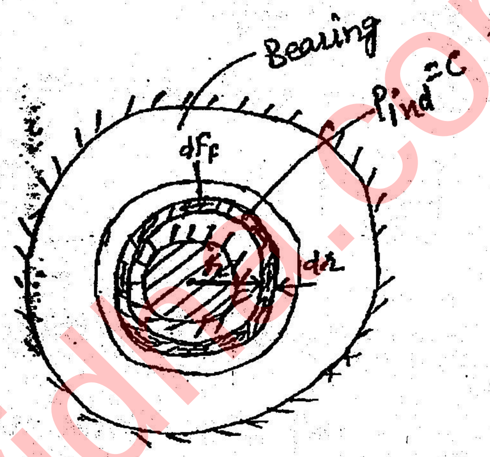
NOTE :- Cage is absent in case of needle roller bearing bcoz needle rollers are placed all around the periphery of the shaft.

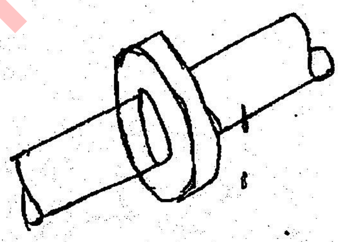


Thrust Bearing
OR
Axial Load Bearing :-

Flat collar Bearing :-







$\boxed{b = R_o - R_i}$

$R_o \rightarrow$ Outer Radius of collar

$R_i \rightarrow$ Inner Radius of collar

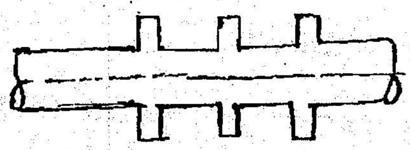
$t \rightarrow$ Thickness of collar

$W \uparrow \rightarrow R_o \uparrow$

$\rightarrow$ If radial space is constraint.

$n \uparrow$ = no. of collar

Flat collar Bearing
- Single collar Bearing
- Multi collar Bearing



Design by uniform pressure theory:-

By U.P.T P_{ind} = constant

$$\boxed{P = C}$$

$dw = 2\pi r \cdot dr\, P$

$\int dw = \int_{R_i}^{R_o} 2\pi r \cdot dr\, P$

$W = 2\pi P \int_{R_i}^{R_o} r\,dr$

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

Pressure induced $\boxed{P_{ind} = \dfrac{W}{\pi (R_o^2 - R_i^2)}}$

Safe Condition

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

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

Strength of collar Bearing (max^m load) $\boxed{W_{max} = \pi (R_o^2 - R_i^2) P_{per}}$

Frictional Torque:-

$df_f = \mu N$

and $N = dw$

$df_f = \mu dw$

$= \mu (2\pi r\, dr) P$

and $dT_f = df_f \times r$

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

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

$$\boxed{T_f = \frac{2}{3} \mu \pi P [R_o^3 - R_i^3]}$$

Here $P = \dfrac{W}{\pi (R_o^2 - R_i^2)}$

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

and

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

$\omega \rightarrow$ angular speed

Design by uniform wear theory (UWT) :-

$P_{ind} \propto \frac{1}{r}$

$$\boxed{P_{ind} = \frac{c}{r}}$$

$dw = 2\pi r \cdot dr \cdot p$

$\int dw = \int_{R_i}^{R_o} 2\pi r dr p$

$w = 2\pi \int_{R_i}^{R_o} r dr \frac{c}{r}$

$W = 2\pi c \int_{R_i}^{R_o} dr$

$= 2\pi c [R_o - R_i]$

$$\boxed{C = P \cdot r}$$

$\therefore W = 2\pi P r [R_o - R_i]$

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

Safe condition:-

$[P_{ind}]_{max} \le P_{per}$

$[P_{ind}]_{max} = \frac{W}{2\pi R_i (R_o - R_i)}$

$\frac{W}{2\pi R_i [R_o - R_i]} \le P_{per}$

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

Frictional Torque :-

$dT_f = r \times dF_f$

$\int dT_f = \int_{R_i}^{R_o} r \cdot \mu (2\pi r) dr \cdot p$

$\int dT_f = 2\pi\mu \int_{R_i}^{R_o} r^2 \cdot \frac{c}{r} dr$

$\int dT_f = 2\pi\mu c \int_{R_i}^{R_o} r\, dr$

$T_f = \pi\mu c (R_o^2 - R_i^2)$

But $c = P \cdot r$

$T_f = \mu\pi P r (R_o^2 - R_i^2)$

But $P = \frac{W}{2\pi r (R_o - R_i)}$

$$\therefore \boxed{T_f = \frac{\mu W (R_o + R_i)}{2}}$$

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

$\boxed{UPT}$ $[P_{ind} = C]$

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

Safe condition

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

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

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

(4) $T_f = \mu W R_{effective}$

$\therefore R_{effec} = \dfrac{2}{3}\left(\dfrac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)$

(5) Power loss $= T_f \times \omega$

(6) $\tau_{ind} = \dfrac{W}{2\pi R_i t}$

(7) $[W_{max}]_{shear} = 2\pi R_i t\, \tau_{per}$

$\boxed{UWT}$ $[P_{ind} \propto \frac{1}{r}]$

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

Safe condition

$[P_{ind}]_{max} \le P_{per}$

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

(3) $T_f = \dfrac{\mu W(R_o + R_i)}{2}$

(4) $R_{effec} = \left(\dfrac{R_o + R_i}{2}\right)$

(5) Power loss $= T_f \times \omega$

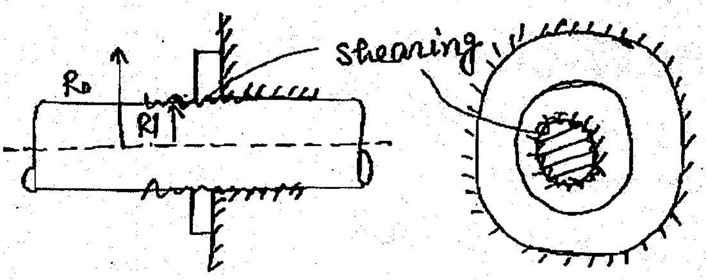
(6) $\tau_{ind} = \dfrac{W}{2\pi R_i t}$

(7) $[W_{max}]_{shear} = 2\pi R_i t\, \tau_{per}$

Shear Design of collar :-

Sheared area $= 2\pi R_i \cdot t$

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



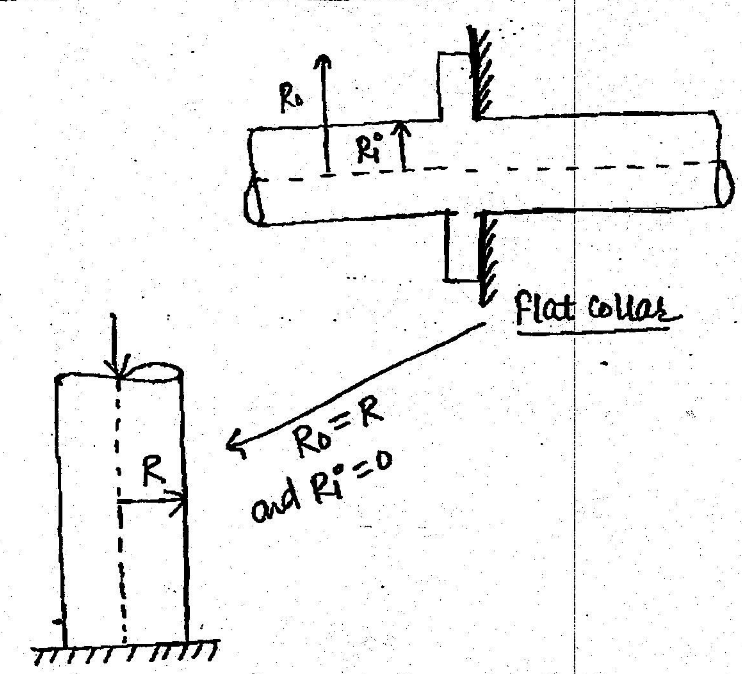
Safe condition :-

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

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

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

Expression for Frictional Torque for Flat pivot Bearing



Flat pivot Bearing

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

$$(T_f)_{UWT} = \frac{\mu W R}{2}$$

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

onclusions:-

1) Hence Frictional Torque by UPT is 33% greater than Frictional Torque by UWT in case of Flat pivot Bearing.

2) There is always Frictional Torque by UPT is greater than Frictional torque by UWT for all cases.

$$\boxed{(T_f)_{UPT} > (T_f)_{UWT}}$$

Bearing	Clutch
⇛ Collar	⇛ Friction lining
⇛ Power loss	⇛ Power transmit

⇛ For safe design of bearing it is better to use UPT because power loss occurs in overcoming the frictional Resistance.

⇛ For safe design of clutches (old or wornout clutches) it is better to use UWT because pressure is non uniformly distributed over the clutch surfaces.

⇛ For the safe design of new clutches it is better to use UPT because pressure is uniformly distributed over the clutch surfaces when they are new.

Ques A clutch has outer and inner diameter 100mm & 40mm respectively. Assuming a uniform pressure of 2MPa and coefficient of friction 0.4. The torque carrying capacity of clutch is —

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

and $$P_{ind} = \frac{W}{\pi[R_o^2 - R_i^2]}$$

$$\therefore \quad T_f = \frac{2}{3}\mu\pi P\left[R_o^3 - R_i^3\right]$$

$$= \frac{2}{3}(0.4)\pi \times 2\times10^6\left[(0.05)^3 - (0.02)^3\right]$$

$$\boxed{T_f = 196 \text{ N-m}}$$

ues A Disc clutch is required to transmit 5 kw at 2000 rpm. The Disc has a friction lining with coefficient of friction 0·25. The bore radius of friction lining is 25 mm. Assume uniform contact pressure of 1 MPa. The value of outside radius for friction lining ___.

$$P = \frac{2\pi N T_f}{60}$$

$$5\times10^3 = \frac{2\pi N T}{60}$$

$$T_f = \frac{60\times5\times10^3}{2\pi\times2000}$$

$$T_f = \underline{23.87\ N\text{-}m}$$

$$\therefore\ T = \frac{2}{3}\mu\pi P\left[R_o^3 - R_i^3\right]$$

$$23.87 = \frac{2}{3}\times0.25\pi\times10^6\left[R_o^3 - (.025)^3\right]$$

$$\therefore\ \boxed{R_o = 39.4\ mm}$$

ues A Disc clutch with a single frictional surface has $\mu = 0.3$. The max^m pressure which can be imposed on friction material is 1·5MPa. The outer ~~radius~~ Dia. of clutch plate is 200 mm and inner diameter is 100mm. Assuming UWT for the clutch plate the max^m Torque that can be transmitted.

$$T_f = \frac{\mu W(R_o+R_i)}{2}$$

$$\therefore\ (T_f)_{max} = \frac{\mu W_{max}(R_o+R_i)}{2}$$

and

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

$$1.5\times10^6 = \frac{W_{max}}{2\pi R_i(R_o-R_i)}$$

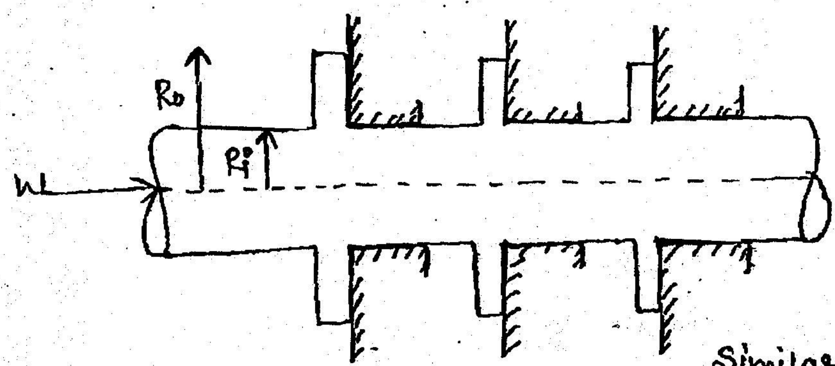
$$1.5\times10^6 = \frac{W_{max}}{2\pi\times\frac{0.1}{2}\left(\frac{.2}{2}-\frac{.1}{2}\right)}$$

$$W_{max} = \underline{23561.94N}$$

$$(T_f)_{max} = \frac{0.3\times23561.94(.1+0.05)}{2}$$

$$\boxed{(T_f)_{max} = 530.1\ N\text{-}m}$$

MULTI COLLAR BEARING :-



Similar collar

$W \uparrow \rightarrow$ if radial space is constraints.

$\therefore\ n \uparrow \rightarrow$ no. of collar

W = Total load

W_{collar} = load on each collar

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

Bearing Design By UPT :-

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

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

Safe condition

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

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

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

Strength of multicollar Bearing.

Frictional Torque :-

$$T_f = (T_f)_{collar} \times n$$

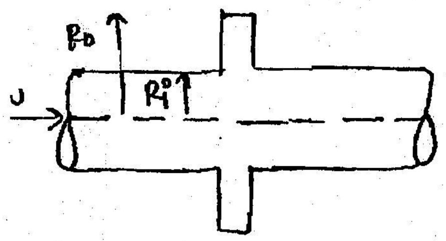
$$= n \times \left[\frac{2}{3}\mu W_{collar}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)\right]$$

$$n \times W_{collar} = W$$

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

Hence frictional Torque is independent of no. of collar.

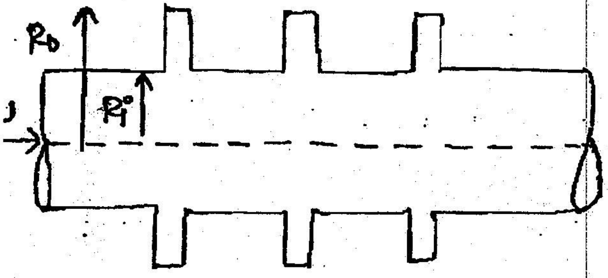
$$\boxed{P_{loss} = T_f \times \omega}$$



$W_{collar} = W$

$P_{ind} = P$

$T_f = T$



$W_{collar} = \frac{W}{3}$

$P_{ind} = \frac{P}{3}$

$T_f = T$

which of the following statements are valid for multicollar Bearing.

1) frictional moment is independent of no. of collars. ✓

2) coefficient of friction is affected by no. of collars. X

3) Intensity of pressure is affected by no. of collars. ✓

Q5 Design of collar Bearing to take a thrust of 16.3 Tonne and the bearing pressure is 0.7 MPa and rotates at 120 rpm. Bearing dimensions are 32 cm inner and 42 cm outer diameter. Coefficient of friction .04 and Safe shear stress for the collar 30 MPa. (assume multi collar Bearing)

Sol:

$P_{ind} = 0.7\ MPa$

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

$R_o = 21 cm$

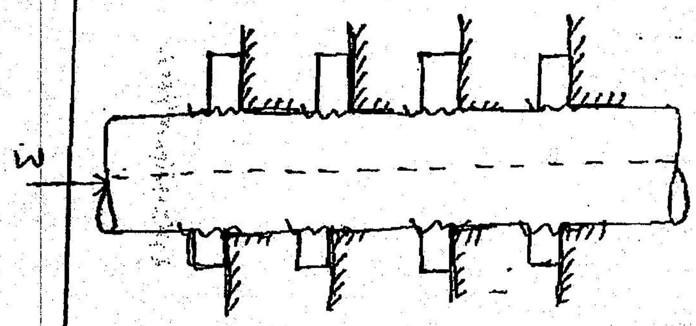
$R_i = 16 cm$

$\tau_{per} = 30 MPa$

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

$$.7 \times 10^6 = \frac{16.3 \times 10^3 \times 9.81}{n\pi(.21^2 - .16^2)}$$

$\boxed{n = 4}$



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

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

Safe condition

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

$$\frac{16.3\times10^3\times9.81}{4\times2\pi\times.16\times t} \le 30\times10^6$$

$$t \ge 13.2$$

$$\boxed{t = 14\,cm}$$

$$b = R_o - R_i$$

$$=$$

Ques The thrust of a propeller shaft in a marine engine is taken up by no. of collars inbuilt with the shaft which is 30 cm in diameter. The total axial thrust is 200KN and the speed of the shaft is 75 rpm. coefficient of friction b/w surface 0.05. and uniform intensity of pressure 0.3MPa. Find the external diameter of the collar and no. of collars required if power loss can't exceed 16 kw.

$$Power = \frac{2\pi N T_f}{60}$$

$$16\times10^3 = \frac{2\pi N T_f}{60}$$

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

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

$$2037.18 = \frac{2}{3}\times.05\times200\times10^3\left[\frac{R_o^3 - .15^3}{R_o^2 - .15^2}\right]$$

$$\boxed{R_o = 24.92\,cm}$$

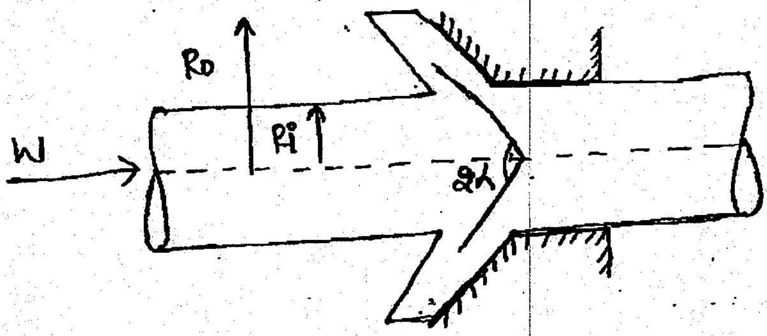
$$P_{ind} = \frac{W}{n\pi(R_o^2 - R_i^2)}$$

$$0.3\times10^6 = \frac{200\times10^3}{n\pi[.2492^2 - .15^2]}$$

$$n = 5.35 \simeq 6$$

$$\boxed{n = 6}$$

Conical collar Bearing :-



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

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

If $n \rightarrow$ no. of collar

For multicollar

$$\boxed{P_{ind} = \frac{1}{n}\frac{W}{\pi(R_o^2 - R_i^2)}}$$

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