Unit - II

DESIGN OF SHAFTS AND COUPLINGS

Couplings: The elements which join two shafts are known as couplings.

Types of Couplings

① **Rigid Coupling** It is used to connect two shaft which are perfectly aligned. used Low-speed applications where good axial alignment

→ Sleeve (or) muff coupling. (or) Box

→ Clamp (or) Split-muff (or) Compression coupling

→ Flange coupling — Unprotected type / Protected / marine type.

② **Flexible Coupling** It is used to connect two shafts having both lateral and angular misalignment.

→ Bushed pin type — use Reduce the effect of Shock & Impact load.

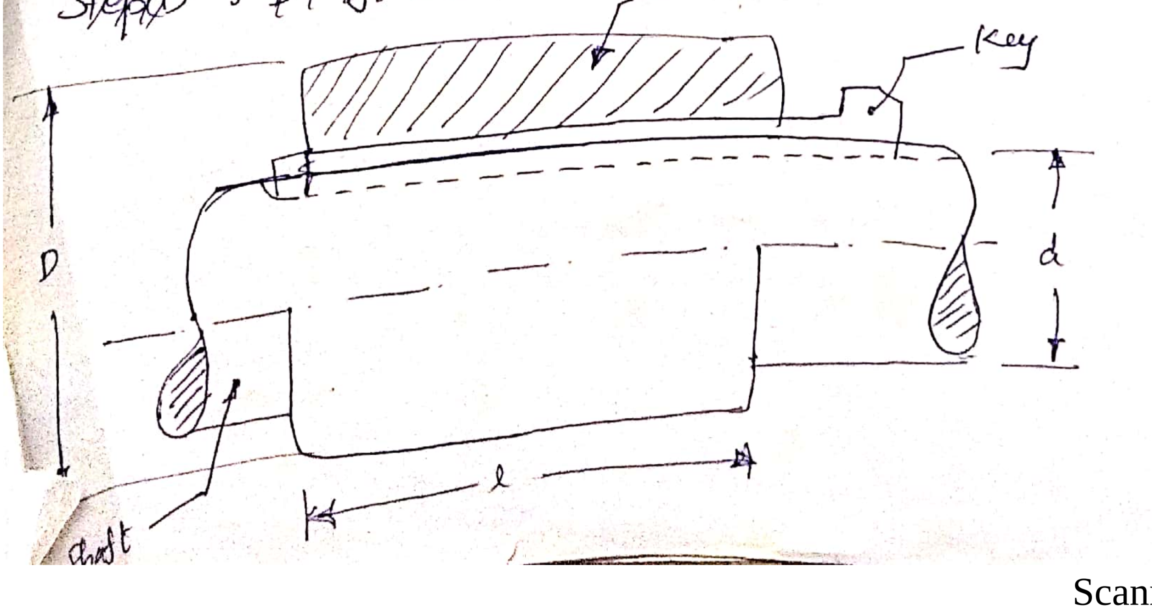
→ Universal coupling

→ Oldham coupling.

① Design of Sleeve (or) muff-coupling: (or) Box

T.133

Step(1) : Design for Sleeve.



1. design for shaft

Step2 :- Design for Sleeve.

$$M_t = \frac{\pi}{16}\tau \left[\frac{D^4 - d^4}{D}\right]$$

(or) T

$T = \frac{P \times 60}{2\pi N}$

$T = \frac{\pi}{16} \times \tau_s \times d^3$

$\tau \rightarrow$ shear strength of the sleeve material $= 14\ N/mm^2$ for cast iron.

Outside diameter of sleeve $D = 2d + 13$ mm

length of sleeve $L = 3.5d$.

Step2: Design for Key

5.16

If we use two key to connect both the shafts (shaft) then the length of each key, $l = \frac{L}{2} = \frac{3.5d}{2}$.

Check for shear Stress

Torque $M_t = F \times r$ $\qquad r = d/2 \quad - \quad A = l \times b$.

$= F \times \frac{d}{2} = \tau A_s \frac{d}{2}$

$T = \tau_s \times l \times w \times \frac{d}{2}$ $\qquad \cancel{M_t = \tau_s l b \frac{d}{2}}$

As - Shearing area (lb)

$\tau \rightarrow$ shear strength of key material 54 to 120 N/mm^2

Check for Crushing

Torque $= \sigma_c A_c \frac{d}{2}$

$T = \sigma_c \times l \times \frac{t}{2} \times d/2$

$= \sigma_c \times l \times \frac{h}{2} \times \frac{d}{2}$

Ac - Crushing area.
$= l \times h/2$

Where σ_c - Crushing strength of key material.

Std value's

$= 150\ N/mm^2$ for steel

$= 80\ N/mm^2$ for C.I

Scanned by CamScanner

Design and make a neat dimensioned sketch of a muff coupling which is used to connect two steel shafts transmitting 40kw at 350 rpm. The material for the shaft and key is plain carbon steel for which allowable shear and crushing stresses may be taken as 40MPa and 80 MPa. respectively. The material for the muff is cast iron for which the allowable sheal stress may be assumed as 15 MPa.

Given data:

$$\text{Power transmitted } (P) = 40\,kw = 40 \times 10^3\,w$$

$$N = 350\,rpm.$$

$$\tau_s = 40\,MPa = 40\,N/mm^2$$

$$\sigma_c = 80\,MPa = 80\,N/mm^2$$

$$\tau_c = 15\,MPa = 15\,N/mm^2$$

To Find

Dimension of sleeve and key.

Solution

1. design for shaft

$$T = \frac{P \times 60}{2\pi N}$$

$$T = 40 \times 10^3 \times 60 / 2 \times \pi \times 350$$

$$T = 1100 \times 10^3\,N.mm.$$

$$T = \frac{\pi}{16} \times \tau_s \times d^3$$

$$1100 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$d^3 = \frac{1100 \times 10^3}{7.86} = 140 \times 10^3$$

$$d = 52 \text{ say } 55\,mm.$$

② Design for Sleeve

$D = 2d + 13$ mm

$= 2 \times 55 + 13 = 123$ say $\boxed{125 \text{ mm.}}$

length of the muff

$L = 3.5\, d$

$L = 3.5 \times 55 = 192.5$

$\boxed{L = 195 \text{ mm}}$

$$T = \frac{\pi}{16} \times \tau \left[\frac{D^4 - d^4}{D} \right]$$

$$1100 \times 10^3 = \frac{\pi}{16} \times \tau \left[\frac{(125)^4 - (55)^4}{125} \right]$$

$$= 370 \times 10^3\, \tau$$

$$\tau = 1100 \times 10^3 / 370 \times 10^3$$

$$\tau = 2.97 \text{ N/mm}^2.$$

3) Design for Key

we find that for a shaft of 55 mm diameter
from Design data book P. 100: 5.19
width of key ~~~~ mm.

psg 5.16

b (or) w = 16 mm

height (h) = 10 mm.

length of the key each shaft

$$l = L/2 = \frac{195}{2} = 97.5 \text{ mm.}$$

Check for Shear.

$M_T = F \times r = F \times d/2$ w - width of the key

$= \tau_s l b \cdot \frac{d}{2}$ (w)

$1100 \times 10^3 = \tau_s \times 97.5 \times 16 \times \frac{55}{2}$

w/d

$$\tau_s = \frac{1100 \times 10^3}{97.5 \times 16 \times \pi/2}$$

$$\tau_s = 22.8 \ N/mm^2$$

Check for Crushing

$$T = \sigma_c \times A_c \times \frac{d}{2}$$

$A_c = l \times h/2$.

$$1100 \times 10^3 = \sigma_c \times l \times h/2 \times d/2.$$

$$= \sigma_c \times 97.5 \times \frac{10}{2} \times \frac{55}{2}$$

$$= \sigma_c \times 97.5 \times 5 \times 22.5$$

$$\sigma_c = \frac{1100 \times 10^3}{97.5 \times 5 \times 22.5}$$

$$\sigma_c = 45.6 \ N/mm^2$$

Design of Key (471)

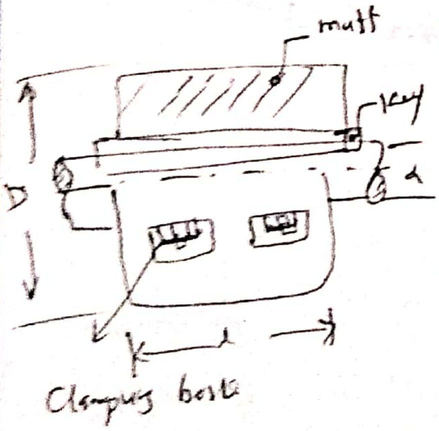
Define Key,
Types of Key
Spline,
Exp: 13.1, 2, 3

Design of Split muff coupling:

(i) Design for Shaft, sleeve & key

(ii) Design for Clamping bolts

$$M_t = \frac{\pi^2}{16} \mu (d_b)^2 \sigma n d.$$



$\mu \rightarrow$ co-efficient of friction, (0.3) assuming.

$n \rightarrow$ number of bolt

$d \rightarrow$ dia of shaft

$d_b \rightarrow$ dia of the bolt

$\sigma \rightarrow$ Allowable tensile strength of the bolt material

h.w/ Ex: 13.5

A rigid type of coupling is used to connect two shaft transmitting 15 kw at 200 r.p.m. The shaft key and bolt also made of C45 Steel and the coupling is of cast iron. Design the coupling.

Same as the previous problem

only add

Design for bolt.

$$M_t = \frac{\pi^2}{16} \times \mu \times (d_b)^2 \times \sigma_t \times n \times d.$$

assume no. bolt $n = 4$,

allowable stress, steel $\sigma_t = 70\,N/mm^2$

$\mu = 0.3.$

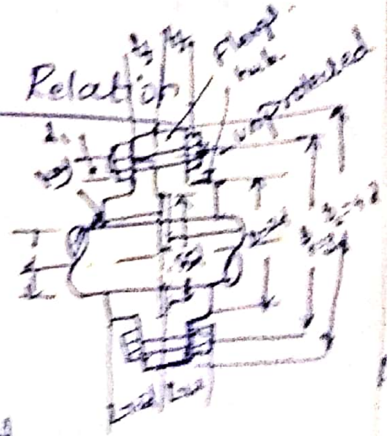
$d = 18.59\,mm$

$d_b = 20\,mm.$

Flange coupling. Protected & unprotected type.

① ~~unprotected type flange coupling~~

Empirical Relation



Step 1 Outer diameter of hub $D = 2d$

length of hub $L = 1.5d$

Pitch circle diameter of bolts $D_1 = 3d$

Outside diameter of flange $D_2 = 4d.$

Thickness of flange $t_f = 0.5d$

Thickness of protective layer, $t_p = 0.25d$

number of bolts $n = 3$ for d upto 40 mm

$= 4$ for d " 100 mm

$= 6$ for d " 180 mm

1. Design of shaft:

Step 1: Empirical Relation

$$P = \frac{2\pi N M_t \text{ (or) } T}{60}$$

$$M_t = \frac{\pi}{16} \times \tau_s \times d^3$$

τ_s – allowable shear strength of shaft

Step 2: Design of hub.

$$M_t = \frac{\pi}{16} \times \tau_h \times \left[\frac{D^4 - d^4}{D}\right]$$

τ_h → allowable shear strength of flange. for cast iron

length of the hub $L = 1.5d$

Step 3: Design for key

Same as previous design

5.19 . shear $T = l \times b \times \tau_k \times d/2$

crush. $T = l \times t/2 \times \sigma_{ck} \times d/2$

Step 4: Design for Flange: based on shear failure (or) $\boxed{t_f = 0.5d}$

The Torque carrying capacity of the flange

$$M_t = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

t_f – flange thickness

Step 5 Design for bolt

Shear str

Load on each bolt –

$$M_t = \frac{\pi}{4} d_b^2 \times \tau_b \times n \times \frac{D_1}{2}$$

$\sigma_{c,k} = 2\tau$

40 20

square key

w.k, t.k

12, 12

PSG. (9)

~~T =~~ $\frac{}{4}$ ~~n~~ d_1

Crushing strength of ~~flange~~ bolt

$$M_t = n \cdot d_b \, t_f \cdot \sigma_{cb} \frac{D_1}{2}$$

Torque type Flange Coupling:

Flexible coupling

The shaft and the flange of a marine engine are to be designed for flange coupling. in which the flange is forged on the end of the shaft. The following are to be con[sidered] in the design.

power of the engine = 3 MW

Speed of the engine = 100 r.p.m.

Permissible shear stress in bolt and shaft = 60 MPa.

No of bolts used n = 8

Pitch circle diameter of bolts = 1.6 x Dia of shaft.

Find 1. diameter of shaft 2. diameter of bolts. 3. thickness of flange 4. dia of flange.

Given:-

$P = 3\,MW = 3 \times 10^6\,W$

$N = 100\,rpm$

$\tau_s = \tau_s = 60\,MPa = 60\,N/mm^2$

$n = 8$

$D_1 = 1.6 \times d.$

To Find

Solution:

1. Dia of Shaft:-

Torque Transmitted by Shaft - $T = \frac{P \times 60}{2\pi N}$

$$T = \frac{3\times10^6\times60}{2\pi N} = 286\times10^3 \text{ N.m} = 286\times10^6 \text{ N.mm}$$

5/9/16

$$\frac{\pi}{16}\times\tau_s\times d^3$$

$$286\times10^6 \quad \pi/16\times60\times d^3 = 11.78d^3$$

$$= \frac{286\times10^6}{11.78} = d, = 289 \text{ Say } 300\text{mm.}$$

$$\boxed{d = 300\text{ mm}}$$

Diameter of bolts!

$$M_t = \frac{\pi}{4}d_b^2\times\tau_b\times n\times\frac{D_1}{2}.$$

$$286\times10^6 = \frac{\pi}{4}\times(d_b)^2\times60\times8\times\frac{1.6\times300}{2}$$

$$= 90490(d_b)^2$$

$$(d_b)^2 = \frac{286\times10^6}{90490} = 3160$$

$$\boxed{d_b = 56.2\text{ mm}}$$

(2) The standard diameter of the bolt is 60mm (M60). The taper on the bolt may be taken from 1 in 20 to 1 in 40.

(3) Thickness of flange!

$t_f = 0.5d = 0.5\times300 = 150$

$$M_t = \frac{\pi d^2}{2}\times\tau_f\times t_f$$

$$286\times10^6 = \frac{\pi\times300^2}{2}\times\tau_f\times150$$

$$\tau_f = 20.2 \text{ MPa / N/mm}^2$$

(4) Dia of flange

$D_2 = 2.2d$

$D_2 = 2.2\times300$

$\boxed{= 660\text{ mm.}}$

Marine type

Thickness of the Flange = d

Taper of bolt = 1 in 2[0] to 1 i[n 40]

Pitch circle diameter of bolts $D_1 = 1.6$

Outside diameter of Flange $D_2 = 2.2$

Shaft dia.	n.
35 to 55	= 4
56 to 150	= 6
151 to 230	= 8
231 to 390	= 10

8/8/16

2

(1)

Design a rigid flange coupling to transmit a torque of 250 N-m between two co-axial shaft. The shaft is made of alloy steel, flange out of cast iron and bolts out of steel. Four bolts are used to couple the flanges. The shafts are keyed to the flange hub. The permissible stresses are given below:

Shear Stress on shaft = 100 MPa.

Bearing (or) Crushing Stress on Shaft = 250 MPa.

Shear stress on keys = 100 MPa.

Bearing stress on keys = 250 MPa.

Shearing Stress on C.I : 200 MPa.

Shear stress on bolts = 100 MPa.

After desigining the various elements, make a neat sketch of the assembly indicating the importance dimensions. The stresses developed in the various numbers may be checked if thumb rules are used for fixing the dimensions.

Given

$T = 250 N\cdot m = 250 \times 10^3 \ N.mm.$

$n = 4$

$\tau_s = 100 \ MPa = 100 \ N/mm^2$

$\tau_{cs} = 250 \ MPa = 250 \ N/mm^2$

$\tau_K = 100 \ MPa = 100 \ N/mm^2$

$\sigma_{cK} = 250 \ MPa = 250 \ N/mm^2$

$\tau_c = 200 \ MPa = 200 \ N/mm^2$

$\tau_b = 100 \ MPa = 100 \ N/mm^2$

To Find

Cast iron Rigid Flange coupling.

Solution:

Step 1 Design for hub

$$T = \frac{\pi}{16} \times \tau_s \times d^3$$

$$250 \times 10^3 = \frac{\pi}{16} \times 100 \times d^3$$

$$d = 23.35 \text{ Say } 25 \text{ mm}$$

Step 2: WKT ... $D = 2d = 2\times 25 = 50$ mm

length of the hub. $L = 1.5d = 1.5\times 25 = 37.5$ mm

Pitch circle diameter of bolts $D_1 = 3d = 3\times 25 = 75$ mm

outside diameter of flange $D_2 = 4d = 4\times 25 = 100$ mm

Thickness of flange $t_f = 0.5d = 0.5\times 25 = 12.5$ mm.

protective circumferential flange $t_p = 0.25d = 6.25$ mm

the allowable ... the number ... permi...

WKT Torque transmitted (T)

$$T = \frac{\pi}{16} \times \tau_s \times \left[\frac{D^4 - d^4}{D}\right]$$

$$250\times 10^3 = \frac{\pi}{16} \times \tau_s \times \left[\frac{50^4 - 25^4}{50}\right]$$

$$\tau_s = 10.86 \text{ MPa}$$

$$\boxed{\tau_s = 10.86 \text{ N/mm}^2}$$

Less than 200 MPa, therefore the design for hub is safe.

Step 3: Design for Key:

WKT dimension $d = 25$ mm

So PSG data book 5.19.

$w = 10$ mm, $t = 8$ mm.

$l = L = 37.5$ mm.

Step 4: Design for flange:

Thickness of the flange (t_f) = 12.5 mm.

WKT

$$T = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

$$250\times 10^3 = \frac{\pi\times (50)^2}{2} \times \tau_c \times 12.5$$

$\tau_c = 5.1$ MPa $\boxed{= 5.1 \text{ N/mm}^2}$

Design is safe.

Step 5: Design for bolt:

WKT $D_1 = 75$ mm

$$T = \frac{\pi}{4} (d_b)^2 \tau_b \times n \times \frac{D_1}{2}$$

$$250\times 10^3 = \frac{\pi}{4} \times (d_b)^2 \times 100 \times 4 \times \frac{50}{2}$$

$\boxed{d_b = 4.6 \text{ mm}}$ design is safe.

Design and draw a cast iron flange coupling for a mild steel shaft transmitting 90kW at 250rpm. The allowable shear stress in the shaft is 40 MPa and the angle of twist is not to exceed 1° in a length of 20 diameters. The allowable shear stress in the coupling bolts is 30 MPa.

Solution Given:

$P = 90 kW = 90 \times 10^3 W$

$N = 250 rpm$

$\tau_s = 40 MPa = 40 N/mm^2$

$\theta = 1° = 1 \times \frac{\pi}{180} = 0.0175 rad$

$l = 20d.$

$\tau_b = 30 MPa = 30 N/mm^2$

To Find

Design of C.I Flange coupling.

Solution:

Step 1: Design of Shaft

$$T = \frac{P \times 60}{2\pi N} = \frac{90 \times 10^3 \times 60}{2 \times \pi \times 250} = 3440 N.m = 3440 \times 10^3 N.mm$$

Considering ~~strength~~ rigidity of the shaft

WKT

$$\frac{T}{J} = \frac{C \times \theta}{l}.$$

$J = \frac{\pi}{32} \times d^4$

Assume take $C = 84 \times 10^3 N/mm^2$

$$\frac{3440 \times 10^3}{\pi/32 \times d^4} = \frac{84 \times 10^3 \times 0.0175}{2d}$$

$$d^3 = 35 \times 10^6 / 73.5$$

$d = 78 \checkmark 80 mm.$

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Two 35 mm Shafts are Connected by a flanged Coupling. The flanges are fitted with 6 bolts on 125 mm bolt Circle. The Shafts transmitted a torque of 800 N.m at 350 rpm. For the safe. Stresses mentioned below, calculate 1. diameter of bolts. 2. thickness of flanges 3. Key dimension 4. hub Length 5. Power transmitted.

Safe shear Stress for shaft material = 63 MPa.

Safe Stress for bolt material = 56 MPa.

Safe Stress for Cast Iron Coupling = 10 MPa.

Safe Stress for key Material = 46 MPa.

Given data!

$d = 35 mm$

$n = 6$

$D_1 = 125 mm.$

$T = 800 N.m = 800 \times 10^3 N.mm.$

$N = 350 rpm.$

$\tau_s = 63 MPa = 63 N/mm^2$

$\tau_b = 56 MPa = 56 N/mm^2$

$\tau_c = 10 MPa = 10 N/mm^2$

$\tau_k = 46 MPa = 46 N/mm^2$

To Find!

Design a flanged Coupling.

Solution!

Step 1. Diameter of bolts.

$$T = \frac{\pi}{4}(d_b)^2 \times \tau_b \times n \times \frac{D_1}{2}$$

$$800 \times 10^3 = \frac{\pi}{4}(d_b)^2 \times 56 \times 6 \times \frac{125}{2}$$

$$(d_b)^2 = 800 \times 10^3 / 16495 = 48.5$$

$$d_b = 6.96 \sim \boxed{8 mm}$$

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Step 2: Thickness of flanges:

$D = 2d$

$= 2 \times 8 = 16$ m

$$T = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

$$800 \times 10^3 = \frac{\pi \times (16)^2}{2} \times 10 \times t_f$$

$$t_f = 10.4 \approx \boxed{12 mm}$$

Step 3: Key dimensions

w.r.t $d = 35$ mm.

width of key $(w) = 12$ mm.

thickness of key $(t) = 8$ mm,

now check the shear stress.

Step 4: Hub length:

$L = 1.5d = 1.5 \times 35$

$$\boxed{L = 52.5 mm}$$

Step 5: power transmitted

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

$$= 2\pi \times 350 \times 800 \times 10^3$$

$$\boxed{P = 29.325 kW}$$

Design and draw a protective type of C.I flange coupling for a steel shaft transmitting 15kW at 200rpm, and having an allowable shear stress of 40 MPa. The working stress in the bolts should not exceed 30 MPa. Assume that the same material is used for shaft and key and that the crushing stress is twice the value of its shear stress. The maximum torque 25% greater than the full load torque. The shear stress for C.I. is 14 MPa.

Given

$P = 15kW = 15 \times 10^3$ W

$N = 200$ r.p.m.

$\tau_s = 40$ MPa $= 40$ N/mm²

$\tau_b = 30$ MPa $= 30$ N/mm²

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$\sigma_{ck} = 2\tau_k.$

$T_{max} = 1.25\, T_{mean}.$

$\tau_c = 14\,MPa = 14\ N/mm^2$

To Find!

Design a Protective Flange Coupling.

Solution:

Step 1 Design for shaft.

$$T_{mean} = P \times 60 / 2\pi N.$$

$$T_{mean} = 716\,N.m = 716 \times 10^3\ N.mm$$

$$T_{max} = 1.25\, T_{mean}$$

$$= 1.25 \times 716 \times 10^3$$

$$= 895 \times 10^3\ N.mm.$$

W.K.T

$$T = \frac{\pi}{16} \times \tau_s \times d^3$$

$$895 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$d^3 = 895 \times 10^3 / 7.86$$

$$d = 48.4 \text{ say } \boxed{50\,mm}$$

Step 2! Empirical Relation:

$D = 2d = 2 \times 50 = 100$ mm.

$L = 1.5d = 1.5 \times 50 = 75$ mm.

$D_1 = 3d = 3 \times 50 = 150$ mm.

$D_2 = 4d = 4 \times 50 = 200$ mm.

$t_f = 0.5d = 0.5 \times 50 = 25$ mm

$t_p = 0.25 \times d = 0.25 \times 50 = 12.5$ mm.

Step 3! Design of hub!

$$T = \frac{\pi}{16} \times \tau_c \times \left[\frac{D^4 - d^4}{D}\right]$$

$$895 \times 10^3 = \pi/16 \times \tau_c \times \left[\frac{100^4 - 50^4}{100}\right]$$

$$\boxed{\tau_c = 4.86\ N/mm^2}$$ Desig is safe.

Step 4: Design for key:

WKT $d = 50\,mm$
$w = 16\,mm$
$t = 16\,mm.$ } from Psg data book Pg. No 5.18

$L = 75\,mm$

WKT crushing check.

$$T = l \times w \times \tau_k \times d/2$$

$$895 \times 10^3 = 75 \times 16 \times \tau_k \times \frac{50}{2}$$

$$\boxed{\tau_k = 29.8\ N/mm^2}$$

Consider key crushing

$$T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$$

$$895 \times 10^3 = 75 \times \frac{16}{2} \times \sigma_{ck} \times \frac{50}{2}$$

$$\boxed{\sigma_{ck} = 59.6\ N/mm^2}$$ design for key safe.

Step 5: Design for flang.

$$t_f = 0.5d = 0.5 \times 50 = 25\,mm.$$

$$T = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

$$895 \times 10^3 = \frac{\pi \times (100)^2}{2} \times \tau_c \times 25$$

$$\tau_c = 2.5\ MPa.$$

Step 6: Design for bolts!

$$T = \frac{\pi}{4} \times (d_b)^2 \times \tau_b \times n \times D_1/2.$$

$$895 \times 10^3 = \frac{\pi}{4} \times (d_b)^2 \times 30 \times 4 \times 150/2$$

$$(d_b)^2 = 895 \times 10^3 / 7070$$

$$d_b^2 = 126.6.$$

$$\boxed{d_b = 11.25\ mm}$$ design is safe.

16/7/17

Design a bushed pin type of flexible coupling to connect a pump shaft to a motor shaft transmitting 32 kw at 960 rpm. The over all torque is 20% more than mean torque. The material properties are as follow.

(i) Allowable shear and crushing stress for shaft and key material is 40 Mpa and 80 MPa.

(ii) The allowable shear stress for cast iron is 15 MPa.

(iii) The allowable bearing pressure for rubber bush is $0.8\ N/mm^2$.

(iv) The material of the pin is same as that of shaft and key.

Draw and neat sketch of the coupling.

Given

$P = 32\ kW = 32 \times 10^3\ W$

$N = 960\ r.p.m$

$T_{max} = 1.2 T_{mean}$.

$\tau_s = \tau_k = 40\ MPa$

$\sigma_{cs} = \sigma_{ck} = 80\ MPa$.

$\tau_c = 15\ MPa$.

$P_b = 0.8\ N/mm^2$

To Find

Design a bushed pin type of flexible coupling.

Solution

Step I Design for Shaft

$$T_{mean} = \frac{P \times 60}{2\pi N} = \frac{32 \times 10^3 \times 60}{2\pi \times 960}$$

$$T_{mean} = 318.3\ N.m = 318.3 \times 10^3\ N.mm$$

$$T_{max} = 1.2 \times 318.3 \times 10^3$$

$$\boxed{T_{max} = 382 \times 10^3\ N.mm}$$

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$$T = \frac{\pi}{16} \times \tau_s \times d^3$$

$$382 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$d = 36.5$ mm says 40 mm.

$$\boxed{d = 40\text{ mm}}$$

Step 2: Design for Pins and rubber bush:

Diameter of Pin $d_1 = \frac{0.5 d}{\sqrt{n_i}}$

Three shaft are $\boxed{n = 6}$ assume.

$$= \frac{0.5 \times 40}{\sqrt{6}} = 8.2\text{ mm}.$$

$\boxed{d_1 = 8.2\text{ mm}}$

bending stress indued due to the compressibility rubber bush the dia of pin d_1 may be taken as $\boxed{d_1 = 24\text{ mm}}$

Step 3: Design for rubber bush:

Over all diameter of rubber bush,

$$d_2 = d_1 + 2 \times 2 + 2 \times 6$$

$$\boxed{d_2 = 40\text{ mm}}$$

$2 \rightarrow$ Brass bush (2 mm thick Two side)

$6 \rightarrow$ Rubber bush.

Diameter of the Pitch circle of the Pin $\boxed{D_1 = 2d + d_2 + 2 \times 6}$

$$= 2 \times 40 + 40 + 2 \times 6$$

Step 3!

$$\boxed{D_1 = 132\text{ mm}}$$

bearing load

$$W = P_b \times d_2 \times l$$

$$= 0.8 \times 40 \times l$$

$$W = 32 l \text{ N}.$$

Max. Torque transmitted by the coupling (T_{max})

$$T = W \times n \times \frac{D_1}{2}$$

$$382 \times 10^3 = 32 l \times 6 \times 132/2$$

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$= 12672\, l.$

$$l = \frac{382 \times 10^3}{12672}$$

$\boxed{l = 30.1 \text{ mm}}$ $\sim \boxed{32 \text{ mm}}$

30.1

$w = 32 \times l = 32 \times \cancel{30} 32$

$= \cancel{963.2}\ \cancel{1024} N.$
$963.2 N$

direct stress due to Pure torque.

$$\tau = \frac{W}{\pi/4 (d_1)^2} = \frac{\cancel{1024}\ \cancel{17310}}{\frac{\pi}{4}(40^2)}$$

1024

$\boxed{\tau = 3.26 \text{ N/mm}^2}$

$M = W\left(\frac{l}{2} + 5\right)$

$= \cancel{19866}$ N.m

$= 21504 N mm$

$\sigma = \frac{M}{Z}$ $Z = \frac{\pi}{32} d_1^3$

27.4 N/mm^2

Max. P.S

14.1 N/mm^2

Step 4! Design for hub

$D = 2d = 2 \times 40 = 80$ mm

$L = 1.5d = 1.5 \times 40 = 60$ mm

$$T = \frac{\pi}{16} \times \tau_c \times \left[\frac{D^4 - d^4}{D}\right]$$

382×10^3

$$\tau_c = \cancel{94.26}\ \frac{\pi}{16} \times \tau_c \times \left[\frac{80^4 - 40^4}{80}\right]$$

$\boxed{\tau_c = 4.05 \text{ MPa}}$

Step 5! Design for key

width of the key $w = 14$ mm

$h = 14$ mm.

Checking ~~Crushin~~ Shear Stress

$$T = l \times w \times \tau_k \times d/2$$

$\boxed{\tau_k = 22.79 \text{ MPa}}$

Checking for crushing stress

$$T = l \times \frac{h}{2} \times \sigma_{cr} \times d/2$$

$$\boxed{\sigma_{cu} = 45.68 \text{ MPa}}$$

Step 6 Design for flange

$$t_f = 0.5d = 0.5 \times 40 = 20 \text{ mm}$$

$$T = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

$$382 \times 10^3 = \frac{\pi \times (80)^2}{2} \times \tau_c \times 20$$

$$\boxed{\tau_c = 1.9 \text{ MPa}}$$

Design is safe.

H.W

Design and sketch a flexible flange coupling (Bush type) to transmitting 5 kW at 750 rpm. with service factor 1.2 for shaft, bolt and key permissible shear stress 50 N/mm² For C.I the shear stress is 15 N/mm² and bearing stress for bush 2.5 N/mm². For key crushing stress 100 N/mm²

28/8/19

Unit - III

Temporary and Permanent Joints

Threaded fasteners - Design of bolted joints including eccentric loading, Knuckle joints, Cotter joints - Design of welded joints, riveted joints for structures theory of bonded joints.

Knuckle Joints Knuckle joint is used to connect two rods whoes axes are either coinciding (or) intersecting and lying in one plane. normally knuckle joints are used to tensile loads,

Application:

→ Cycle chain links

→ Diagonal stays of boilers

→ Valve and eccentric rods

→ Tie rod joints of roof truss.

Design of Knuckle Joints:

Diameter of rod $= d$

" of pin $= d_1$

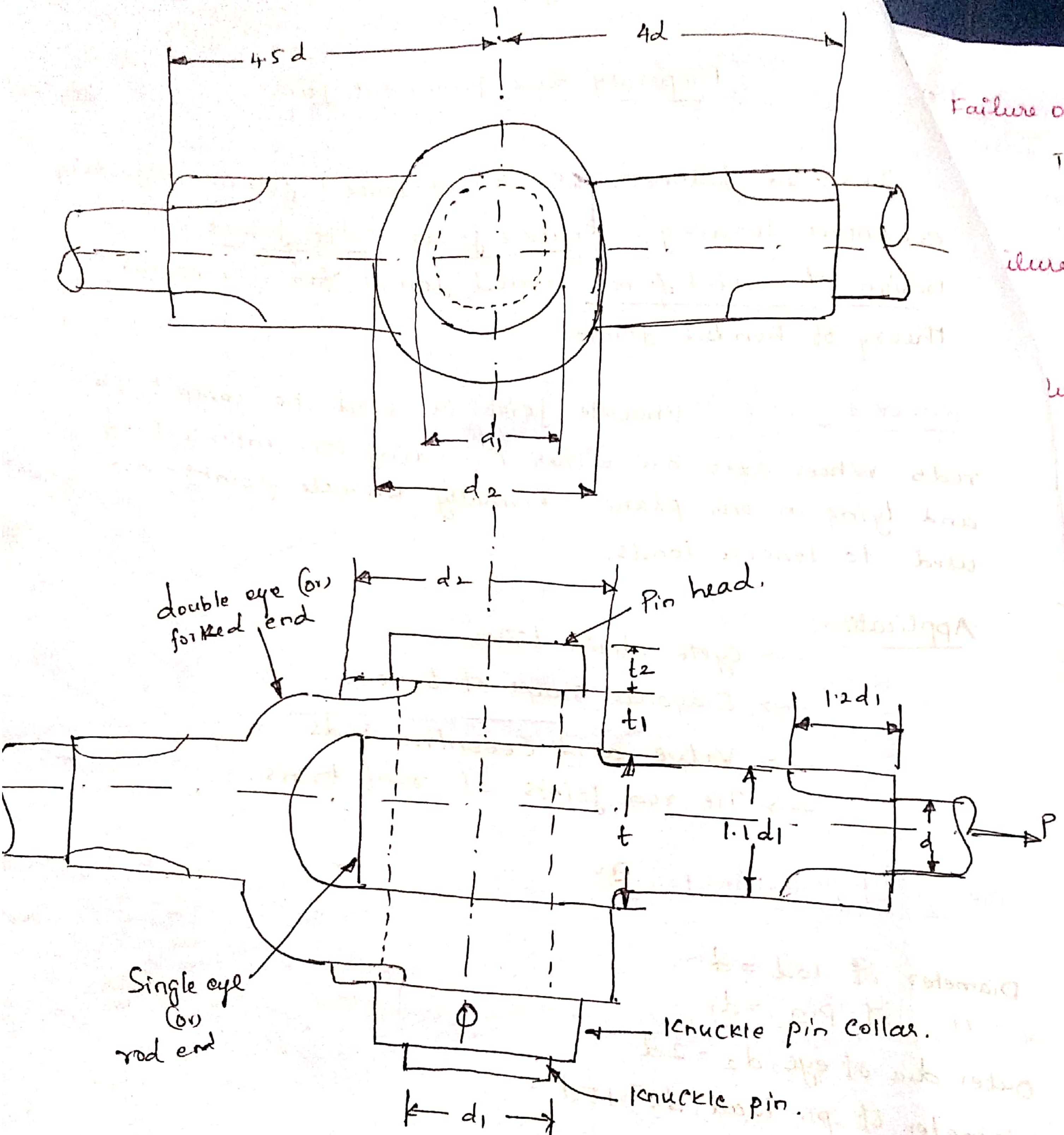
outer dia of eye, $d_2 = 2d$

Diameter of pin head, $d_3 = 1.5d$

Thickness of eye, $t = 1.25d$

Thickness of fork, $t_1 = 0.75d$

Thickness of pin head $t_2 = 0.5d$.



1. Failure of Solid rod in Tension:

Due to the tensile ro load, the rod are subjected to tensile stresses and may fail if this stress exceeds the limit For safe design' we have,

$$P = \frac{\pi}{4} d^2 \times \sigma_t$$

Failure of knuckle pin by double shear:

Two surface of pin that resist shearing.

$$P = 2 \times \frac{\pi}{4} \times d_1^2 \times \tau.$$

3. Failure of the single eye (or) rod end in tension:

$$P = (d_2 - d_1) t \times \sigma_t$$

4. Failure of the single eye (or) rod end in double shear.

$$P = (d_2 - d_1) \times t \times \tau$$

5. Failure of single eye (or) rod end in crushing:

$$P = d_1 \times t \times \sigma_c$$

6. Failure of the forked end in tension:

$$P = (d_2 - d_1) \times 2t_1 \times \sigma_t$$

7. Failure of forked end in double shear.

$$P = (d_2 - d_1) \times 2t_1 \times \tau.$$

8. Failure of the forked end in crushing.

$$P = d_1 \times \cancel{2}t_1 \times 2\sigma_c.$$

$$P = d_1 \times 2t_1 \times \sigma_c$$

9. Failure of knuckle pin by bending.

The loosely fitted knuckle pin experience bending,

$$\sigma_b = \frac{\frac{P}{2}\left[\frac{t_1}{3} + \frac{t}{4}\right]}{\frac{\pi (d_1)^3}{32}}$$

① Design a knuckle joint to transmit 150 kN, design stresses may be taken as 75 mpa in tens., 60 mpa in shear, 150 mpa in compression.

Failu

Given:

$P = 150 kN = 150 \times 10^3 N.$

$\sigma_t = 75 mpa = 75 N/mm^2$

$\tau = 60 mpa = 60 N/mm^2$

$\sigma_c = 150 MPa = 150 N/mm^2$

To Find:

Design a knuckle joint

solution:

1. Failure of the solid rod in tension:

$$P = \frac{\pi}{4} d^2 \times \sigma_t$$

$$150 \times 10^3 = \frac{\pi}{4} d^2 \times 75$$

$$d^2 = \frac{150 \times 10^3}{59} = 2540$$

$$d = 50.4 \sim 52 mm.$$

Diameter of knuckle pin $d_1 = d = 52 mm.$

Outer diameter of eye $d_2 = 2d = 52 \times 2 = 104 mm$

Diameter of knuckle pin head and Collar $d_3 = 1.5d = 1.5 \times 52 = 78 mm$

Thickness of single eye (or) rod end $t = 1.25d = 1.25 \times 52 = 65 mm$

Thickness of fork $t_1 = 0.75d = 0.75 \times 52 = 39 \sim 40 mm$

Pin head $t_2 = 0.5d = 0.5 \times 52 = 26 mm$

2. Failure of the knuckle pin in shear.

$$p = 2\times\frac{\pi}{4}\times(d_1)^2\times\tau.$$

$$150\times10^3 = 2\times\frac{\pi}{4}\times(d_1)^2\times\tau$$

$$= 2\times\pi/4\times(52)^2\times\tau$$

$$\boxed{\tau = 35.3\ N/mm^2}$$

3. Failure of the single eye (or) rod end in Tension

$$p = (d_2-d_1)\times t\times\sigma_t$$

$$150\times10^3 = (104-52)\times65\times\sigma_t$$

$$\boxed{\sigma_t = 44.4\ N/mm^2}$$

4. Failure of single eye (or) rod end in ~~crushing~~ double shear.

$$p = \cancel{d_1\times t\times\sigma_c}.\ (d_2-d_1)\times t\times\tau.$$

$$150\times10^3 = \cancel{52\times65\times\sigma_c}\ (104-52)65\times\tau.$$

$$\boxed{\tau = 44.4\ N/mm^2}$$

5. Failure of single eye (or) rod end in crushing.

$$p = d_1\times t\times\sigma_c$$

$$150\times10^3 = 52\times65\,\sigma_c$$

$$\boxed{\sigma_c = 44.4\ N/mm^2}$$

6. Failure of forked end in tension!

$$p = (d_2-d_1)\,2t_1\times\sigma_t$$

$$150\times10^3 = (104-52)\times2\times40\times\sigma_t$$

$$\boxed{\sigma_t = 36\ N/mm^2}$$

(7) Failure of the forked end in shear

$$P = (d_2 - d_1)\, 2t_1 \times \tau.$$

$$150 \times 10^3 = (104 - 52)\, 2 \times 40 \times \tau.$$

$$\boxed{\tau = 36\ N/mm^2}$$

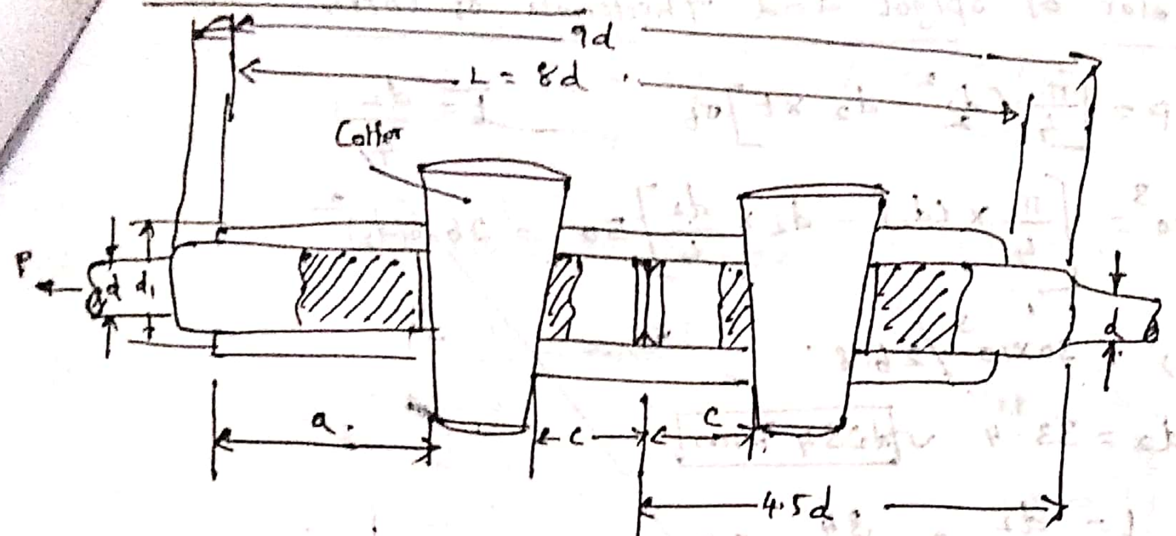
(8) Failure of the forked end in crushing.

$$P = d_1 \times 2t_1 \times \sigma_c$$

$$150 \times 10^3 = 52 \times 2 \times 40 \times \sigma_c$$

$$\boxed{\sigma_c = 36\ N/mm^2}$$

Sleeve and cotter joint!



$d_1 = 2.5\,d$

$d_2 = 1.25\,d$

$L = 8\,d$

$t = \frac{d_2}{4}$

$b = 1.25\,d$

$l = 4\,d$

Design a sleeve and cotter joint to resist a tensile load of 60kN. All parts of the joint are made of the same materiale with the following allowable stress

$\sigma_t = 60$ mpa, $\tau = 70$ MPa, $\sigma_c = 125$ mpa.

Given:

$\sigma_t = 60\ N/mm^2$, $\tau = 70\ N/mm^2$, $\sigma_c = 125\ N/mm^2$

$P = 60 \times 10^3$ N.

To find: Design a sleeve and cotter joint.

Solution:

Step 1! Diameter of the rod!

$$P = \frac{\pi}{4} d^2 \times \sigma_t$$

$$60 \times 10^3 = \frac{\pi}{4} \times d^2 \times 60$$

$$d = 35.7 \sim 36$$

$$\boxed{d = 36\ mm}$$

Step 2! Diameter of enlarged end of rod and thickness of cotter!

$$P = \left[\frac{\pi}{4} (d_2)^2 - d_2 \times t\right] \sigma_t$$

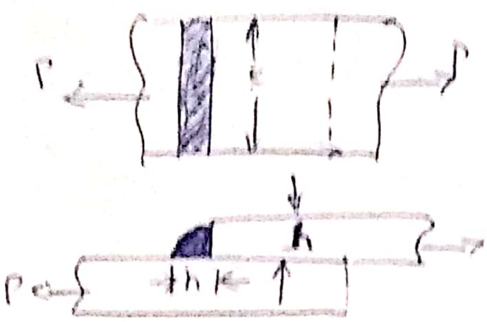
Unit - IV

Elements of welding symbol: Reference line, Arrow, Basic weld symbol, Finish symbol, supplementary symbol, specification,

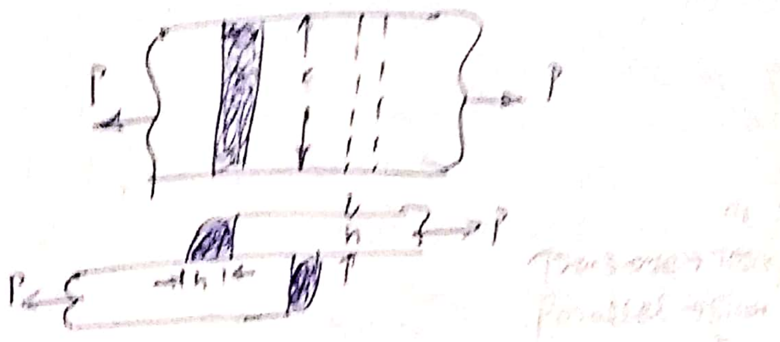
Design of welded joints

Welding can be defined as a process of joining two similar (or) dissimilar metals by heating to a suitable temperature.

① Strength of Transverse fillet welded joint



Single transverse fillet weld



Double transverse weld

Single fillet weld

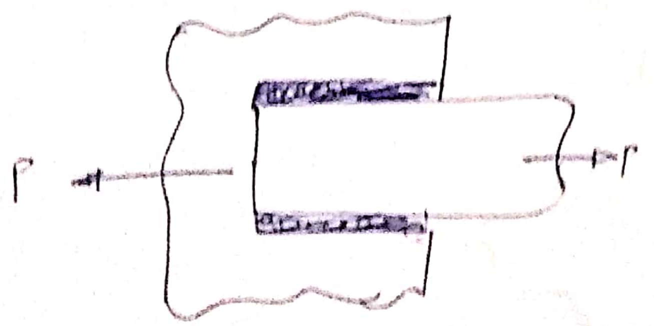
$$\boxed{P = 0.707 h \times l_1 \times \sigma_t}$$

h — size of weld
l → length of weld
σ_t → tensile stress

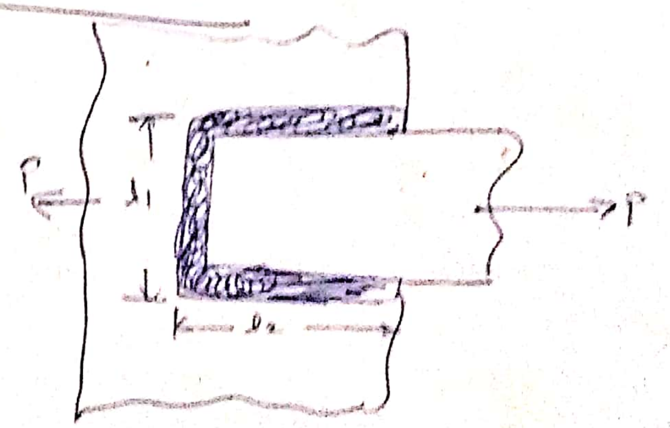
double fillet weld

$$\boxed{P = 2 \times 0.707 h \times l_2 \times \sigma_t}$$

② Strength of Parallel fillet welded joint



Double parallel fillet weld



Combination of transverse and parallel fillet weld

$P = P_1 + P_2$

Single

$P = 0.707 h \times l \times \sigma_t$

Double $P = 2 \times 0.707 h \times l \times \tau = 1.414 h \times l \times \tau$

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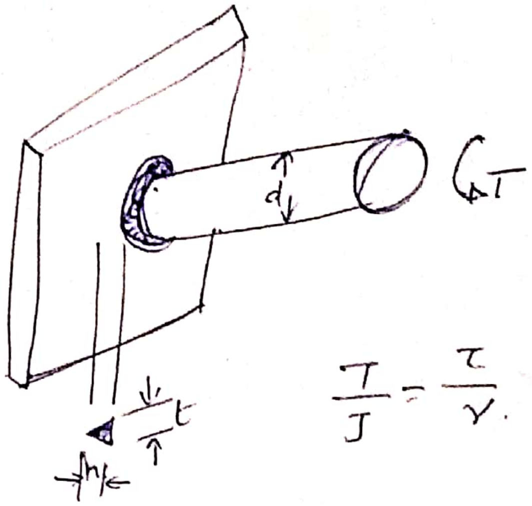
Special cases of fillet welded joints

1. Circular fillet weld subjected to torsion.



d – Dia of rod.

T = Torque action on the rod.

h = Size of weld.

t = Throat thickness.

r = Radius of rod.

$$\frac{T}{J} = \frac{\tau}{r}$$

weld Section

For circular Z (or) J. $= \frac{\pi t d^3}{4}$.

Maximum Shear Stress

$$\tau = \frac{2.83 T}{\pi h d^2}$$ $\boxed{\text{PSG DB 11.3}}$

② Bending Moment.

$$Z = \frac{\pi t d^2}{4}$$

$\boxed{A = 2 t \pi D}$

$t = 0.707 h \pi D.$

$$\sigma_b = \frac{M}{Z} = \frac{M}{\pi t d^2/4}$$

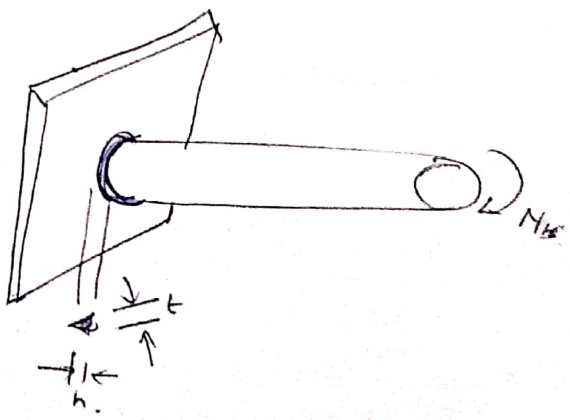
max bending Stress (or) normal stress

$$\sigma = \frac{5.66 M_b}{\pi h d^2}$$ $\boxed{\text{PSG DB 11.3}}$

$P = P_1 + P_2$

$l_1 = -\ \ 112.5$

$l_2 = -\ \ 112.5$



length of the each fillet for fatigue loading

(or) Variable loading

$$\sigma_t = \frac{\text{Stress}}{1.5}$$

$$\tau = \frac{\text{Stress}}{2.7}$$

	Type of Weld	Stress concentration factor
1.	Static load	1
	Reinforced butt weld	1.2
2	Toe of transverse fillet weld (or) normal fillet weld	1.5
	End of Parallel fillet weld (or) longitudinal weld	2.7
	T-butt joint with sharp corners	2.0

$\sigma_t = P/A$.

A = Area of throat $(t) \times l$.

$A = 0.707 h \times l$.

$A = 2 \times 0.707 h \times l$.

A plate 100 mm wide and 12.5 mm thick is to be welded to another plate by means of ~~two~~ double parallel fillet welds. the plates are subjected to a load of 50 kN. Find the length of the weld so that the maximum stress does not exceed 56 N/mm² (Do the calculation under static loading & fatigue loading)

Given

width of the plate (b) = 100 mm

$s = h = 12.5$ mm

$P = 50 \text{ kN} = 50 \times 10^3$ N

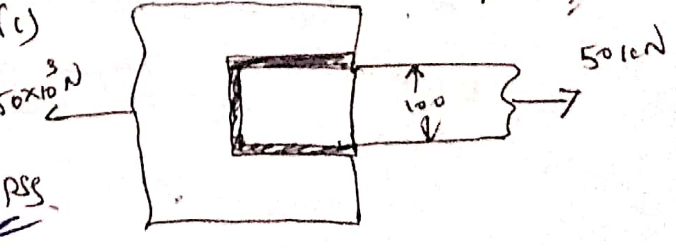
$\tau = 56$ N/mm²

Static: $P = 2 \times 0.707 h \times l \times \tau$, $l = 20.14 + 12.5$

Fatic: $\tau = 56/2.7$, $l = 136 + 12.5$

To Find

length of the weld (l)



Solution

Method 1

$$\sigma_b = \frac{0.707P}{hl} \quad \text{— (11.3) PSG}$$

$\frac{1}{1.414} = 0.707$

$l = 50.5$ mm

$l = 50.5 + 12.5$ $\boxed{l = 63 \text{ mm}}$

$l = 50.5$ mm + weld 12.5 mm

Method 2

Fatigue $\text{Stress}/2.7$

Thickness of the weld 12.5 m

~~Size of weld~~ 12.5 m

$A = 2 \times 0.707 \times h \times l$

$$\tau = \frac{L}{A}$$

$P = 0.707 \times h \times l$.

$50 \times 10^3 = 0.707 \times 12.5 \times l$

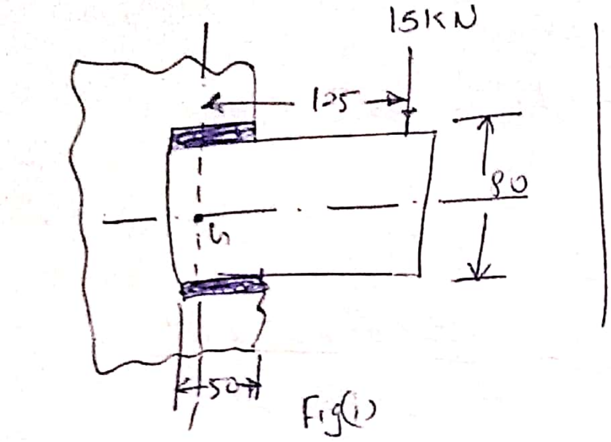
$= 17.675\, l$ mm²

~~$56 = \frac{50 \times 10^3}{17.675\, l}$~~

~~$\boxed{l = 50.5 \text{ mm}}$~~

bracket carrying a load of 15 KN is to be welded as show in fig. Find the size of weld required if the allowable shear stress is not to exceed 80 MPa.

Given



Fig(i)

Given:

$P = 15KN = 15 \times 10^3 N$

$\tau = 80 MPa$ - N/mm^2

$b = 80 mm$.

$l = 50$

$e = 125 mm$

Solution

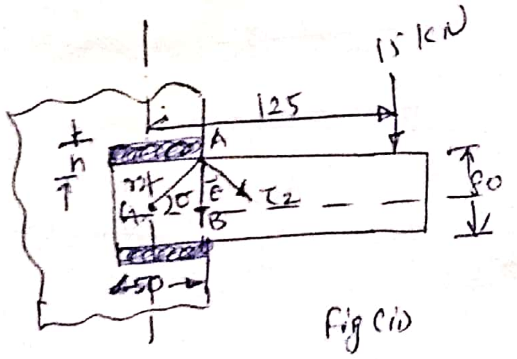
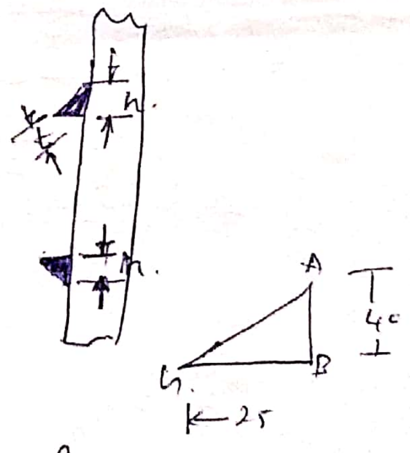


Fig (ii)



WKT

$A = 2 \times t \times l = 2 \times 0.707 h \times l$.

$= 1.414 \times h \times 50 = 70.7 h \ mm^2$

direct or Primary shear stress

$$\tau_1 = \frac{P}{A} = \frac{15 \times 10^3}{70.7h} = \frac{212}{h} N/mm^2$$

We can find fig(ii) $AB = 40 mm$.

$BG = r_1 = 25 mm$.

PSG 11.3

$$J = \frac{t \cdot l(3b^2 + l^2)}{6} = \frac{0.707h \times 50[3(80)^2 + 50^2]}{6}$$

(or)

$$J = \frac{t \times d(3b^2 + d^2)}{6}$$

$$= 127850h \text{ mm}^4.$$

$t = 0.707$

max. radius of the weld

$$r_2 = \sqrt{AB^2 + BG^2} = \sqrt{(40)^2 + (25)^2} = 47 \text{mm}.$$

Shear stress due to turning moment

$$\tau_2 = \frac{P \times e \times r_2}{J} = \frac{15 \times 10^3 \times 125 \times 47}{127850h} = \frac{689.3}{h} \text{ N/mm}^2$$

$$\cos\theta = \frac{r_1}{r_2} = \frac{25}{47} = 0.532$$

WKT Resultant ~~force~~ shear stress

$$\tau = \sqrt{(\tau_1)^2 + (\tau_2)^2 + 2\tau_1 + \tau_2\cos\theta}$$

$$80 = \sqrt{\left[\frac{212}{h}\right]^2 + \left[\frac{689.3}{h}\right]^2 + 2 \times \frac{212}{h} \times \frac{689.3}{h} \times 0.532}$$

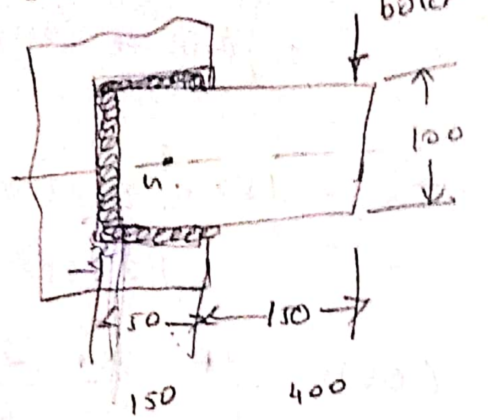
$$= 822/h.$$

$$\boxed{h = 10.3 \text{mm}}$$

H.W

... cantilever to a vertical

Rectangular Steel Plate is welded as a cantilever to a Vertical Column and supports a single concentrated load, P, as shown in Fig. Determine the weld size if shear stress in the same is not to exceed 140 mpa.



Given

$P = 60\ kN$

$b = 100\ mm$

$l = 50\ mm$

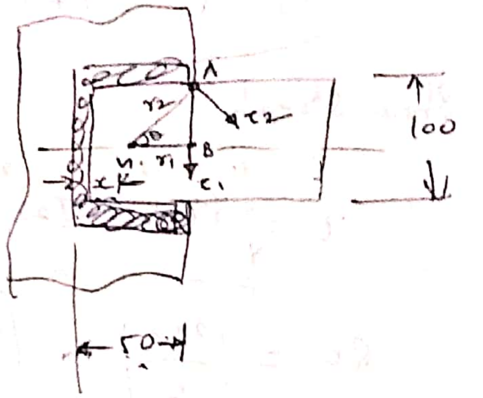
$\tau = 140\ mpa.$

To Find

Size of the weld

Solution.

$$x = \frac{l^2}{2l+b} = \frac{50^2}{2\times50+100} = 12.5\ mm$$



$$J = t\left[\frac{(b+2l)^3}{12} - \frac{l^2(b+l)^2}{b+2l}\right]$$

$$= 275\times10^3\ h\ mm^4.$$

Distance of load the centre of gravity (G)

$$e = 150+50-12.5 = 187.5\ mm.$$

$$r_1 = BG = 50 - x = 50-12.5 = 37.5\ mm.$$

$$AB = 100/2 = 50\ mm,$$

WKT

max radius of the weld

$$r_2 = \sqrt{(AB)^2+(BG)^2} = \sqrt{(50)^2+(37.5)^2} = 62.5\ mm$$

$$\cos\theta = \frac{r_1}{r_2} = \frac{37.5}{62.5} = 0.6$$

Area of weld

$$\tau = P/A \qquad A = 2\times0.707h\times l + 0.707h\times b$$

$$= 0.707\,h\,(2l+b)$$

$$= \frac{60\times10^3}{141.4h} = \frac{424}{h}\ N/mm^2 \qquad = 0.707\,h(2\times50+100) = 141.4\ mm^2$$

$$\tau_2 = \frac{P\times e\times r_2}{J} = \frac{2557}{h}\ N/mm^2$$

$$\tau = \sqrt{(\tau_1)^2+(\tau_2)^2+2\tau_1\times\tau\times\cos\theta}$$

$140 =$

$$\boxed{h = 20.23\ mm.}$$

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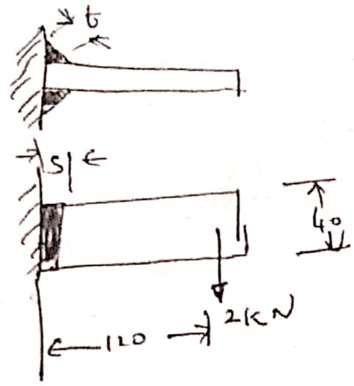
weld joint as show in Fig. is subjected to an eccentric load of 2KN. Find the size of weld, if maximum shear stress in the weld is 25 MPa.



Given

$P = 2\,kN$

$e = 120\,mm$

$d = 40\,mm$

$\tau_{max} = 25\,N/mm^2$ or mpa.

To Find

Size of the weld.

Solution

Area of weld for double transverse weld.

A.

direct stress $(\tau) = P/A$

$A = 2 \times 0.707\, h \times d.$

$= 2 \times 0.707\, h \times 40$

$= 56.56\, h\, mm^2$

$\frac{35.4}{h}$

$\tau = \frac{2 \times 10^3}{56.56 h} = \frac{198.2}{h}\, N/mm^2$

$e = 120\,mm$

$\sigma_b = \frac{4.24\, Pe}{h l^2} = \frac{636.6}{h}\, N/mm^2$

$\sigma_b = \sigma_x.$

$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4(\tau_{xy})^2}$

$\sigma_b = \frac{M}{Z}.$

$M = e \times load.$

$Z = \frac{bl^2}{4.242}$

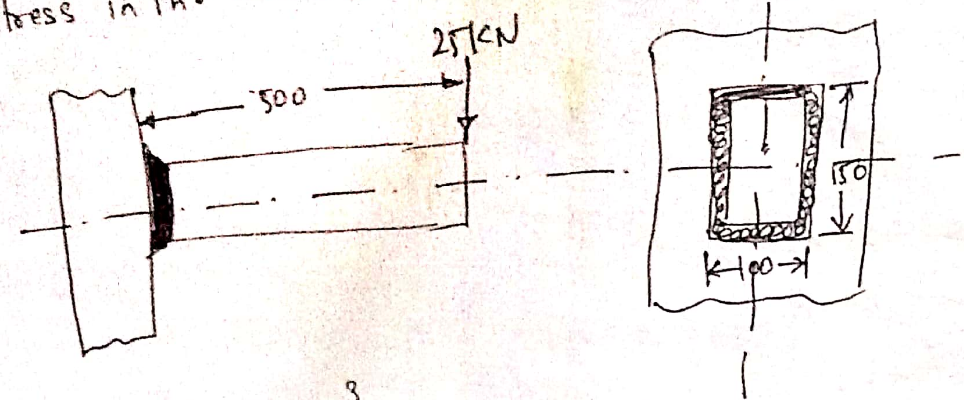
$= \frac{5(40)^2}{4.242} = 377\, h\, mm^2$

$25 = \frac{320}{h}.$

$\boxed{h = 12.8\,mm}$

Q2.

A Rectangular Cross-Section bar is welded to a Support by means of fillet welds as shown in fig. Determine the Size of the welds, if the permissible shear stress in the weld is limited to 75 MPa.



Given:

$P = 25KN = 25\times10^3 N$

$\tau_{max} = 75\ MPa = 75\ N/mm^2$

$l = 100mm$

$b = 150\ mm.$

(distance) $e = 500\ mm.$

Section modulus

$Z = t\left[b.l + \frac{b^2}{3}\right]$

To Find

Size of the weld (h)

Solution

$\tau = \frac{P}{A}$

For weld Rec Area

$A = t(2b+2l)$

$t = 0.707h.$

$= 0.707h(2\times150+2\times100)$

$= 353.5h\ mm^2$

Direct Stress $\tau = \frac{25\times10^3}{353.5h}$

$= \frac{70.72}{h}\ N/mm^2$

bending Moment

$M = P\times e = 25\times10^3\times500 = 12.5\times10^6 N.mm$

$Z = 0.707h\left[150\times100 + \frac{(150)^2}{3}\right] = 15907.5h\ mm^3$

σx (or) $\sigma_b = M/Z = \frac{785.8}{h}\ N/mm^2$

Max. Shear stre

$\tau_{max} = \frac{1}{2}\sqrt{(\sigma x-\sigma y)^2+4(\tau xy)^2} = \frac{1}{2}\sqrt{\left(\frac{785.8}{h}\right)^2+4\left[\frac{70.72}{h}\right]^2}$

$= \frac{399.2}{5}$

$\boxed{h = 5.32\ mm}$

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2

(1)

UNIT-IV

Design of Energy Storing Elements & Engine Components.

7.100 607.105

Design of various types of Springs:

1. Design of compressed helical spring.
2. Design of tension helical spring.
3. Design of concentric (or) composite helical spring.
4. Design of leaf spring.
5. Design of bellivelli spring (or) disc spring.

Spring: Spring is a elastic member; when a external load applied on the spring energy is stored in the form in the form compressed spring, when the unloaded condition energy is released in the form of tension of spring.

Classification of springs

- helical spring
  - Closed coil
  - open coil.
- helical compression spring.
- helical tension spring.
- helical conical spring.
- helical spiral spring.

Spiral spring

Leaf spring.

Disc (or) bellevelli springs.

Spring materials

- High Carbon Steel
- Oil tempered carbon steel
- Stainless steel.
- bar bronze, brass
- Medial carbon alloy steel.

Design of helical Spr

1. Spring stiffness

$q = P/y$

2) Spring index (C) D/d.

d - wire dia

Effect of curvature Shear stress $k_s = 1 + 1/2c$

Wahl's Factor $k = \frac{4c-1}{4c-4} + \frac{0.615}{c}$

7.100.

Design procedure for spring!

$\tau = K_s \frac{8PD}{\pi d^2}$. Find . Desi

1. Selection of spring.
2. calculate coil diameter from shear stress (7.100) sol
3. Find standard wire diameter $c = D/d$.

where $K_s = \frac{4c-1}{4c-4} + \frac{0.615}{c}$

$\forall \; c = D/d$

$y = \frac{8PD^3 n}{G d^4}$.

4. calculate no. of turns from defelection
5. calculate total no. of. turns
6. calculate stiffness -
7. calculate free length and solid length

$L_s = dn + 2d$

$L_f = L_s + y$

8. calculate Pitch $\frac{L_f - L_s}{N}$

9. calculate helix angle

$\alpha = \tan^{-1}\left[\frac{P}{\pi D}\right]$

(1) Design a helical compression spring to carry a load of 1.5KN with a deflection of 40mm Spring index is 5 allowable shear stress is $400\ N/mm^2$, modulus of rigidity $8 \times 10^{10} N/m^2$

Given

$P = 1.5 \times 10^3 N$

$y = 40mm$

$c = D/d = 5$

$\tau = 400\ N/mm^2$

$G = 8 \times 10^{10} N/m^2 = \frac{8 \times 10^{10}}{10^6} N/mm^2 = 8 \times 10^4 N/mm^2$

Find

Design a helical spring.

Solution

1. Selection of spring

helical compression spring.

2. calculate the ~~dia~~ coil diameter from shear stress

$$\tau = K_s \frac{8PD}{\pi \times d^2}$$ | from 7.100

$C = D/d$.

$$K_s = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

$$K_s = \frac{4(5)-1}{4(5)-4} + \frac{0.615}{5}$$

$$K_s = 1.3105$$

$$400 = 1.3105 \times \frac{8 \times 1.5 \times 10^3 \times 5}{\pi \times d^2}$$

$$d = 7.91 \text{ mm}$$

Std dia $\boxed{d = 8 \text{ mm}}$ From 7.105 DB

3. mean ~~coil~~ wire diameter

$$c = D/d \quad 7.100$$

$$5 = D/8$$

$$\boxed{D = 40 \text{ mm}}$$

D = mean dia

d = wire dia.

4. calculate no. of active coils.

From 7.100

$$y = 8PD^3 n / Gd^4$$

$$40 = \frac{8 \times 15 \times 10^3 \times 40^3 \times n}{8 \times 10^4 \times 8^4}$$

$$n = 17.6$$

$$\boxed{n = 18}$$

Total no. of coil.

Assume Square and Ground

N	Ls	Lf
$N = n+2$	$dn + 2d$	$Ls + y$
		$160 + 40$
$= 18+2$	$= 8(18)+2(8)$	$160 + 40$
$= 20$	$= 160mm$	$= 200mm$

calculate Stiffness (q)

$$q = \frac{Gd^4}{8D^3n}$$

$$q = 8\times10^4 \times 8^4 / 8\times40^3\times18$$

$$q = 35.56 \ N/mm^2$$

Pitch (p)

$$P = Lf - Ls/N = 200 - 160/20$$

$$\boxed{P = 2}$$

helix angle

$$\alpha = \tan^{-1}\left[P/\pi D\right]$$

$$\alpha = \tan^{-1}\left[2/\pi\times40\right]$$

$$\boxed{\alpha = 0.9^\circ}$$

(1)

A Compression coil Spring made of an alloy steel having the following specification.
Mean diameter of coil = 50mm ; wire dia = 5mm
Number of active coils = 20. if this spring is Subjected to an axial load of 500 N. Calculate the maximum Shear Stress (Neglect the curvature effect) to which the Spring Material is Subjected.

Given $D = 50mm$, $d = 5mm$ $n = 20$, $P = 500 N$.

To Find Shear stress (τ)

Solution

Maximum Shear Stress $\tau = K_s \frac{8PD}{\pi d^3}$ — Psg. 7.100

$K_s = 1 + \frac{1}{2C}$ For Neglecting the effect of wire Curvature

$C = D/d$

So WKT $C = \frac{50}{5} = 10mm.$

$K_s = 1 + \frac{1}{2 \times 10} = 1.05$

Max. Shear Stress $\tau = 1.05 \times \frac{8 \times 500 \times 50}{\pi \times (5)^3}$

$$\boxed{\tau = 534.7 \ N/mm^2}$$

(2) A helical Spring is made from a wire of 6mm dia and has Outside diameter of 75mm. if the permissible Shear stress is 350 MPa and Modulus of rigidity 84 KN/mm² find the axial load which the spring can carry and the defelection per active turn.

Given! $d = 6mm$, $D_o = 75mm$, $\tau = 350 MPa.$, $G = 84 KN/mm^2$
$G = 84 \times 10^3 N/mm^2$

To Find deflection per active turn (y/n).

Solution $D_o = D + d$
$D = D_o - d = 75 - 6 = 69mm.$
$C = D/d = 69/6 = 11.5$
δ/n = deflection per active turn.

① Neglecting the effect of curvature.

$$\tau = K_s \times \frac{8PD}{\pi d^3}$$

For effect of curvature
Shear stress factor $K_s = 1 + \frac{1}{2C}$

$$K_s = 1 + \frac{1}{2 \times 11.5}$$

$$K_s = 1.043$$

WKT
PSG 7.100 - $\tau = K_s \times \frac{8PD}{\pi d^3}$

$$350 = 1.043 \times \frac{8 \times P \times 69}{\pi \times (6)^3} = 0.848 P$$

$$P = \frac{350}{0.848}$$

$$\boxed{P = 412.7\ N}$$

WKT deflection of the spring (y)

$$y = \frac{8PD^3 n}{Gd^4}$$

$$y = \frac{8 \times 412.7 \times (69)^3 \times n}{84 \times (6)^4 \times 10^3}$$

$$\boxed{\frac{y}{n} = 9.96\,mm}$$

② Considering the effect of curvature.

For Wahl's stress factor

$$K_s = \frac{4C-1}{4C-4} + \frac{0.615}{C} \quad \text{PSG 7.100}$$

$$K_s = \frac{4 \times 11.5 - 1}{4 \times 11.5 - 4} + \frac{0.615}{11.5} = 1.123$$

$$\tau = K_s \times \frac{8PD}{\pi d^3} = 1.123 \times \frac{8 \times P \times 69}{\pi \times (6)^3}$$

$$350 = 0.913 P$$

$$\boxed{P = 383.4\ N}$$

deflection spring

$$y = \frac{8WD^3 n}{Gd^4}$$

$$\frac{y}{n} = \frac{8 \times 383.4 \times (69)^3}{84 \times 10^3 \times (6)^4}$$

$$\boxed{y/n = 9.2\ldots\,mm}$$

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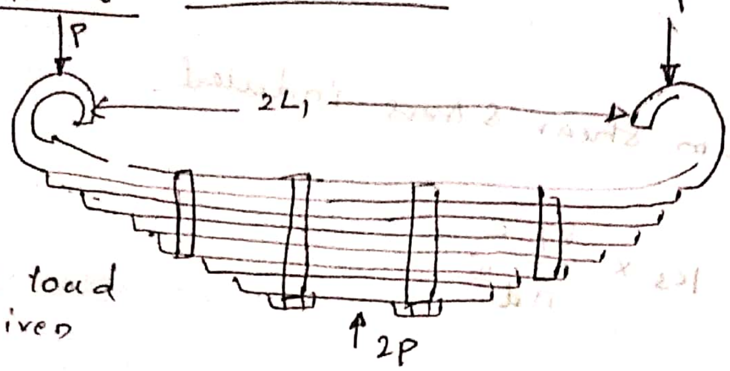
Leaf Spring: Semi-elliptical.



1) calculate load given

2P 7

Overall length (or) Span length $2L_1$

Calculate effective length $2L = 2L_1 - l$.

depth width ratio $= \frac{n \times t}{b}$

$2n_e$ – no. of extra full length Leaves

$2n_g$ – no. of Graduate leaves,

$\boxed{n_g = n - n_f}$

$\boxed{n = n_e + n_g}$

(7.104) Leaf Spring

$\sigma = \frac{6PL}{nbt^2}$, $y = \frac{6PL^3}{Enbt^3}$

initial hat

$c = \frac{2PL^3}{n \cdot E \cdot b \cdot t^3}$

load Exerted on band after the Spring is assembled

$$\boxed{W_b = \frac{2n_e \cdot n_g \cdot P}{n(2n_g + 2n_e)}}$$

Semi ellipti

$$\sigma = \frac{18PL}{b \cdot t^2 (2n_g + 3n_e)}$$

Deflection of Spring

$$Y = \frac{12 \cdot PL^3}{E \times b \times t^3 (2n_g + 3n_e)}$$

Dia of eye

$P = d \times l_1 \times P_b$

Assume $E = 210 \times 10^3 N/mm^2$

bearing Pressure. 8 N/mm²

Length of Leaves

ength of the Smallest leaf $= \frac{\text{Effective length}}{n-1} + \text{ineffective length}$ (6)

master leaf leaves

$$2L_1 + \pi(d+t)2.$$

Radius of curvature.

$$Y(2R-Y) = (L_1)^2$$

(1) Design a leaf spring for the following specific condition., Total load 140 kN, no. of spring supported the load 4, max. nuber of leaves 10, span of the spring 1000 mm, deflection 80 mm, youngs modulus = 200 kN/mm², allowable stress in spring 600 N/mm², ~~band width~~ = 100 mm.

Given

Total load $= \frac{140}{4} = 35\,kN$.

$2L = 1000\,mm$

$\boxed{L = 500\,mm}$

$2L = 2l_1 - l.$

$= 1000 - 100$

$2L = 900\,mm$

$\boxed{L = 450\,mm}$

$2P = 35\,kN$

$P = 17.5\,kN = 17.5\times10^3\,N$

$n = 10$

$y = 80\,mm.$

$2L_1 = 1000\,mm.$

$E = 200\,kN/mm^2 = 200\times10^3\,N/mm^2$

$\sigma = 60\,N/mm^2$

$l = 100\,mm.$

Solution

$$\sigma_b = \frac{6PL}{nbt^2} \quad \text{PSG. 7-104.}$$

$$= \frac{6\times17500\times500}{10\times bt^2} \qquad = \frac{6\times17500\times500}{n\times b\times t^2}$$

$$\boxed{bt^2 = 7.87\times10^3} \text{ — (1)} \qquad nbt^2 = 87.5\times10^3 \text{ — (1)}$$

$= 8500$

$$Y = \frac{6PL^3}{E \cdot n \cdot bt^3} \quad \text{—PSG 7.104.}$$

$$80 = \frac{6 \times 17500 \times (450)^3}{2\times10^5 \times 10 \times bt^3}$$

$$bt^3 = 59.8 \times 10^3 \quad \text{— (2)}$$

$$bt^3 \Rightarrow bt^2 \cdot t = 59.8\times10^3$$

$$7.87\times10^3 \times t = 59.8\times10^3$$

$$t = 7.59$$

$$\boxed{t = 8\,mm}$$

$$bt^2 = 7.87\times10^3$$

$$b = 7.87\times10^3 / 8^2$$

$$b = 116.8$$

$$\boxed{b = 117\,mm}$$

$$t = \frac{6\times17500\times(500)^3}{2\times10^5 \times n\,bt^3}$$

$$nbt^3 = 0.82\times10^6$$

~~82031~~

dividing eq^n 1 & 1

$$\frac{nbt^3}{nbt^2} =$$

$$\cancel{\frac{bt^3}{bt^2} = \frac{82031}{8500}}$$

$$\cancel{\frac{bt^2}{b} =}$$

$$t = \frac{0.82\times10^6}{87.5\times10^3}$$

$$t = 9.37 \sim 10\,mm$$

eqn — 1

$$nbt^2 = 87.5\times10^3$$

$$b = \frac{87.5\times10^3}{n\times t^2}$$

$$\boxed{b = 87.5\,mm}$$

eqn 2

$$nbt^3 = 0.82\times10^6$$

$$t^3 = \frac{0.82\times10^6}{10\times(10)^3} = 82$$

We taking large of the two values

width of the leaves

$$b = 87.5$$

$$\boxed{b = 88\,mm}$$

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Spring under varing load

PSG 7.102

A helical Compression spring made of oil tempered Carbon Steel is Subjected to a load which varies from 400 N to 1000 N. The Spring Index is 6 and the design factor of Safety is 1.25 if the yield Stress in Shear is 770 Mpa. and endurance Stress in shear 350 mpa. Find 1. Size of the Spring wire. 2. diameter of the Spring. 3. Number of turns of the Spring. 4. Free length of the Spring. The compression of the Spring at the maximum load is 30 mm. The modulus of rigidity for the Spring material may be taken as 80 KN/mm².

Given

$P_{max} = 1000\ N$

$P_{min} = 400\ N$

$C = 6$

$n = 1.25$

$\tau_y = 770 MPa$

$\tau_e = 350\ MPa.$

$y = 30$

$G = 80\ KN/mm^2 = 80 \times 10^3\ N/mm^2$

$\boxed{\frac{D}{d} = 6}$

To Find

(i) Size of the Spring wire

(ii) Diameter of the Spring

(iii) Number of turns of the Spring.

(IV) Free length of the Spring

Solution

WKT

Mean load $P_m = \frac{P_{max} + P_{min}}{2} = \frac{1000 + 400}{2} = 700\ N$

Amplitude load $P_a = \frac{P_{max} - P_{min}}{2} = \frac{1000 - 400}{2} = 300\ N.$

WKT

Wahl's Stress factor $K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$

Shear Stress factor $K_s = 1 + \frac{1}{2C}$

Shaft

$$K_s = 1 + \frac{1}{2c} = 1 + \frac{1}{2\times 6} = 1.083$$

Wahl's Stress factor

$$K = \frac{4c-1}{4c-4} + \frac{0.615}{c} = \frac{4\times 6-1}{4\times 6-4} + \frac{0.615}{6}$$

$$\boxed{K = 1.2525}$$

I step! Size of the Spring

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

$$D = 6d$$

WKT mean Shear stress

$$\tau_m = \frac{8 K_{sh} \times P_m D}{\pi d^3}$$

$$= \frac{8 \times 1.083 \times 700 \times 6d}{\pi \times d^3}$$

$$\boxed{\tau_m = \frac{11582}{d^2} \; N/mm^2}$$

Variable Stress

$$\tau_a = \frac{8 K_s \times P_a \times D}{\pi d^3}$$

$$= \frac{8 \times 1.2525 \times 300 \times 6d}{\pi \times d^3}$$

$$\boxed{\tau_a = \frac{5740}{d^2} \; N/mm^2}$$

WKT

$$\frac{1}{n} = \frac{\tau_m - \tau_a}{\tau_y} + \frac{2\tau_a}{\tau_e}$$

$$= \frac{\frac{11582}{d^2} - \frac{5740}{d^2}}{770} + \frac{2 \times \frac{5740}{d^2}}{350}$$

$$d^2 = 505$$

$$\boxed{d = 71 \text{ mm}}$$

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2. Diameter of the Spring -

$\frac{D}{d} = C$

mean diameter $(D) = C \cdot d = 6 \times 7.1 = 42.6$ mm.

Outer diameter of spring $D_o = D + d = 42.6 + 7.1$

$\boxed{D_o = 49.7 \text{ mm}}$

Inner diameter of the spring $D_i = D - d$

$= 42.6 - 7.1$

$\boxed{D_i = 35.5 \text{ mm}}$

3. No. of turns of the Spring -

$$y = \frac{8 P D^3 n}{G d^4}$$

$$30 = \frac{8 \times 1000 \times (42.6)^3 \times n}{80 \times 10^3 \times (7.1)^4}$$

$\boxed{n = 9.87 \sim 10}$

$n' = n + 2 = 10 + 2$

$\boxed{n' = 12}$

4. Free length of the Spring -

$$L_f = n'.d + y + 0.15y$$

$$= 12 \times 7.1 + 30 + 0.15 \times 30$$

$$L_f = 119.7$$

$\boxed{L_f = 120 \text{ mm}}$

UNIT - V BEARINGS

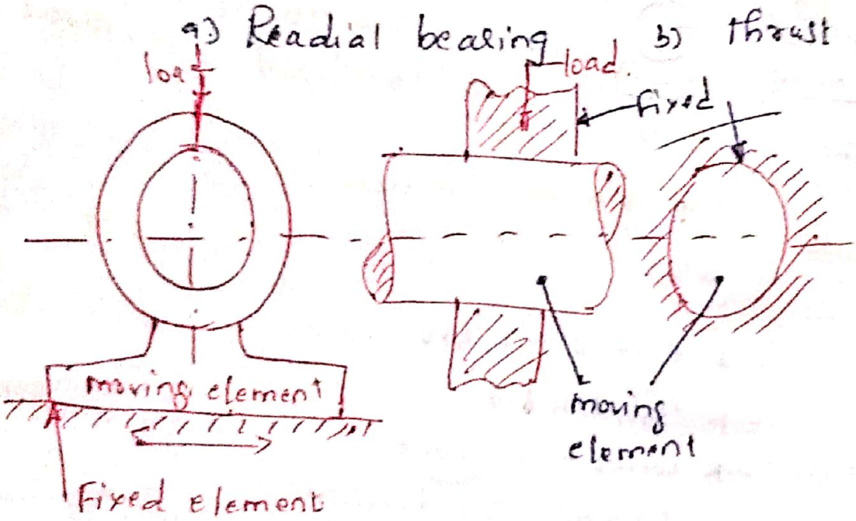
Design of bearings and miscellaneous elements:

bearing:- Bearing is a Mechanical element ~~etc~~ permitting relative motion between two parts, ~~&~~ Such as the shaft and the housing with minimum firiction

classification of bearing!

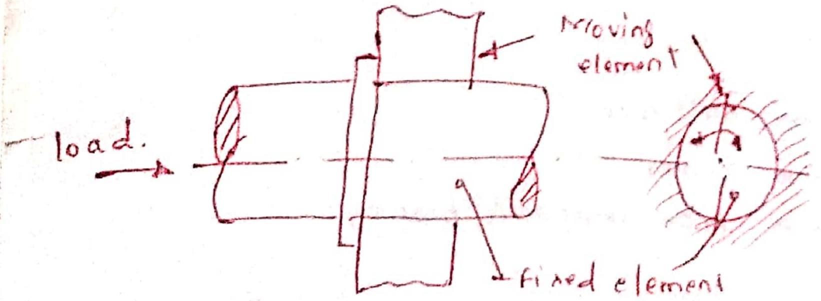
1) Depending upon the direction of load to be supported.

a) Readial bearing b) thrust bearings.



The load acts ⊥ to the direction of motion of the moving element

Radial bearing



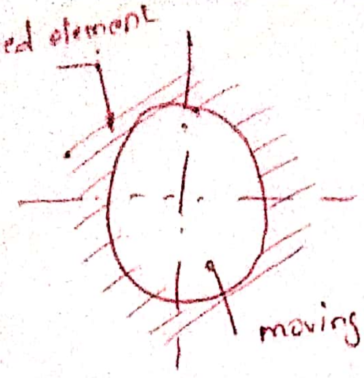
load acts along the axis of rotation.

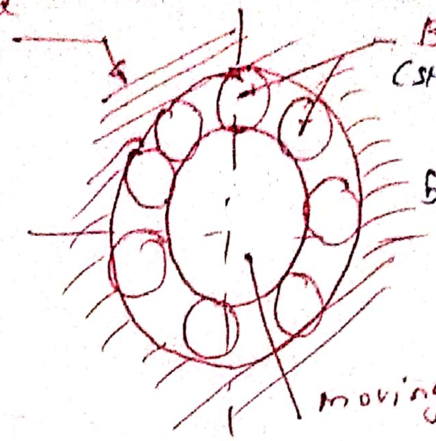
Thrust bearing

② Depending upon the nature of contact!

(a) Sliding contact bearing:-

Sliding takes place along the surfaces of contact b/w the moving element and fixed element ~~&~~ The sliding contact bearing are also known as plain bearings.

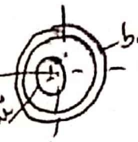




Rolling contact bearing

Types of Sliding contact bearing (Journal bearing).

a) Full journal bearing — angle of contact 360°



b) Partial journal bearing.

⇒ Less than 360° of contact
90° to 180°; 120° is preferred

c) Fitted journal bearing. diameter of Journal are equal.

— There is a no clearance

Properties of Sliding contact bearing:

→ Compressive Strength.
→ Fatigue Strength.
→ Comformability
→ Embeddability
→ Bondability
→ Corrosion resistance.
→ Thermal conductivity
→ Thermal expansion.

material used for Sliding contact bearing:

→ babbit Matal
→ Bronzes
→ cast iron
→ Silver
→ Non-metallic bearing.

Design ~~proce~~ of hydrodynamic journal bearing: (from 7.34 PSG data book all formula)

1. Determine the bearing length by choosing a ratio of l/d

Design for Journal bearing

Step 1: Calculate the diameter of Journal

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

$$T = \frac{\pi}{16} \times \tau \times D^3$$

Assume 150 to 170mm

P – Power given
N – Speed
T – Torque
D – Diameter of the Journal.
τ – Shear Stress

Step 2: Calculate length of the Journal

From PSG Data book 7.31

$\frac{L}{D} = 1$ to 2 we select the ratio based on Machine parameters.

Step 3: Calculate the pressure developed

$$p = \frac{W}{L \times D}$$

W – load
L – length of the Journal
D – dia of Journal
p → Pressure.

Step 4: Select the clearance Ratio

Select the Clearance ratio $\frac{C}{D}$

based on diameter of the journal From PSG 7.33

1 micron $= 10^{-6}$ m.
$= 10^{-3}$ mm.

C → diametral Clearance
D → Journal diameter

Step 5: Select the Lubricating oil

Z/p PSG 7.34

Select the Suitable oil and its viscosity at operating Temperature which is preferably which 60° to 75°

$t_0 = 60°$ to $75°$ (7.35)

Step 6: Select the bearing operating Value

$\frac{Zn}{p}$ PSG 7.31

Z – Absolute viscosity
n – Speed of the Journal, p → 0.7 to 1.4 N/mm^2 Safe limit. 7.31

Step 7: Cal. Select the Coefficient of friction

(McKEES Equation) $\mu = \frac{33.25}{10^{10}}\left[\frac{Zn}{P}\right]\left[\frac{D}{c}\right] + K$ From Psg 7.34

$K \rightarrow$ based on L/D ratio

$\boxed{K - 0.002 \text{ constant}}$

Step 8: Calculate the Heat Generated (H_g)

$$\boxed{H_g = \mu \times W \times V}$$

Rubbing velocity

psg 7.34 $V = \frac{\pi d N}{60}$ m/s

$W \rightarrow$ load in N

Step 9: Cal. Heat dissipated (H_d)

μ – co-efficient of friction

V – Rubbing velocity m/s.

psg 7.35 $H_d = \frac{(\Delta t + 18)^2 LD}{K}$

$\Rightarrow \Delta t = \frac{1}{2}(t_o - t_a)$

K = 437 for heavy
= 775 ligh constouction.

psg 7.34

if not Given 140 t 420

if Given C – Heat dissipation co-efficient [1232 W/m²/°C]

we can used as H_d formula $\boxed{C \times l \times d (t_b - t_a) W}$

Step 10: Cal. Amount of Artificial Cooling Required

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

$$\boxed{H_{AC} = H_g - H_d.} \quad -①$$

Step 11 cal. mass of lubricating oil required for Artificial Cooling

$Q_t = m.S.t$ –②

$m \rightarrow$ mass

$S \rightarrow$ Specific heat oil [1840 to 2100 J/kg/°C]

equating the $H_{AC} = Q$

we get the m

Heat dissipated $(H_d) = C \cdot A \, (t_b - t_a)$

$= C \times l \times d \, (t_b - t_a)$

$|A = l \times d$

Δt, (or) $(t_p - t_a) = \frac{1}{2} (t_o - t_a)$

$= C l d \times \frac{1}{2} (t_o - t_a)$

C → Heat dissipation Co-efficient

t_o → operating temp

t_a → room (or) Absolute Temp

Amount of artificial cooling

$= H_g - H_d$

mass of lubricating oil required for artificial cooling.

$Q_t = m \cdot S \cdot t$

m → mass of the lubricating oil

Specific heat of oil (S) = 1840 to 2100 J/kg/°C.

— * —

① Design a Journal bearing for a centrifugal pump from the following data: Load on the journal = 20000 N; Speed of the Journal = 900 r.p.m. Type of oil is SAE 10, for bearing pressure for the pump = 1.5 N/mm². calculate also mass of lubricating oil required for artificial coolings, if rise of temprature of oil be limited to 10°C. Heat dissipation co-efficient = 1232 W/m²/°C.

② The load on the Journal bearing is 150 KN due to turbine Shaft of 300 mm diameter running at 1800 r.p.m. Determine the following: 1) length of the bearing if the allowable bearing pressure 1.6 N/mm², and 2) Amount of heat to be removed by the lubricant per minute, if the bearing temperature is 60°C and viscosity of the oil at 60°C is 0.02 kg/m-s and bearing clearance is 0.25 mm.

Design Procedure for Journal bearing: ①

1. Determine the bearing length by choosing a ratio of l/d from PSG data book ~~7.29~~ (Pg.No) 7.31
2. Check the bearing pressure $p = W/l.d$ from data book 7.31
3. Assume a lubricant from (7.29) data book. operating temperature (t_o) (or) absolute viscosity. This temperature should be between 26.5°C and 60°C with 8... as a maximum for high temperature installations such as steam turbine.
4. Determine the operating value of $\frac{ZN}{P}$ for the assumed bearing temperature. Z - oil viscosity (or) absolute visc—
5. Assume a clearance ratio c/d from data book Pg.No (7.31)
6. Determine the coefficient of friction (μ) McKess eqn. ~~McKee eq.~~ from data book (7.34 Pg.No.)
7. Determine the heat generated formula from DDB 7.34 = $H_g = \mu W V$ kgf m/min $[V = \pi d N/60]$
8. Determine the heat dissipated DDB 7.34
9. Determine the thermal equilibrium to see that the heat dissipated becomes atleast equal to the heat generated. Incase the heat generated is more than the heat dissipated then either the bearing is redesigned (or) it is artificially cooled by water.

③ A full journal bearing of 50 mm diameter and 100 mm long has a bearing pressure of 1.4 N/mm². The speed of the journal is 900 r.p.m. and the ratio of journal diameter to the diametral clearance is 1000. The bearing is lubricated with oil whose absolute viscosity at the operating temperature of 75°C may be taken as 0.011 kg/m-s. The room temp is 35°C. Find 1. The amount of artificial cooling required 2. The mass of the lubricating oil required, if the difference b/w the outlet and inlet temp of the oil is 10°c. Take specific heat of the oil as 1850 J/kg/°c.

④ A 150 mm diameter shaft supporting a load of 10 kN has a speed of 1500 r.p.m. The shaft runs in a bearing whose length is 1.5 times the shaft diameter. If the diametral clearance of the bearing is 0.15 mm and the absolute viscosity of the oil at the operating temp is 0.011 kg/m-s, find the power wasted in friction.

⑤ A 80 mm long journal bearing supports a load of 2800 N on a 50 mm diameter shaft. The bearing has a radial clearance of 0.05 mm and the viscosity of the oil is 0.021 kg/m-s at the operating temp. If the bearing is capable of dissipating 80 J/s, determine the max safe speed.

⑥ A journal bearing 60 mm is diameter and 90 mm long runs at 450 rpm. The oil used for hydrodynamic lubrication has absoulte viscosity of 0.06 kg/m-s if the diametral clearance is 0.1 mm. find the safe load on the bearing by using Sommerfiled num bearing:

(8) The length and diameter of a Journal bearing are 0.075 m and 0.05 m respectively. The shaft is running at 900 rpm. The bearing temperature is 82°c while the ambient temp is 32°c. The oil used is having a viscosity of ~~0.0128~~ 0.0128 kg/m-s. at the operating temperature of 32°c. Heat dissipation coefficient may be taken as 209.34 W/m²/°c. The bearing is air-cooled with a diametral clearance of 0.05 mm. Compute:

(i) The allowable load on bearing.

(ii) Power loss.

Given:

$l = 0.075\ m = 75\ mm$

$d = 0.05\ m = 50\ mm$

$N = 900\ rpm$

$t_o = 82,\ t_a = 32°$

$Z = 0.0128\ kg/ms$

$C = 209.34\ W/m^2/°c$

$c = 0.05\ mm$

To Find

W

Power loss

Solution

$$P = \frac{W}{l \times d} = \frac{W}{\cancel{0.05}\ 50 \times 75} = \frac{W}{3750}\ N/mm^2$$

$$\mu = \frac{33}{10^6}\left[\frac{ZN}{P}\right]\left[\frac{d}{c}\right] + k = \frac{14.25}{W} + 0.002$$

Heat generated $H_g = \mu W V$ $\quad V = \frac{\pi D N}{60}$

$$= \left[\frac{1425}{W} + 0.002\right] \times W \times 2.356\ W \qquad = \frac{\pi \times 0.05 \times 900}{60}$$

$$= 2.356\ m/s$$

$$H_d = CA\tfrac{1}{2}(t_o - t_a)$$

$$= 209.34 \times 0.05 \times 0.075 \times \tfrac{1}{2}(82 - 32)$$

$$= 39.37\ W$$

$H_g = H_d$

$$\boxed{W = 1233.4\ N}$$

Power loss $= \mu W V$

$\mu = 0.0135$

$$\boxed{\text{Power loss} = 39.22\ W}$$

①

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① Given ②

979

$W = 20000 N$

$N = 900$ r.p.m.

$t_o = 55°C$

$Z = 0.017 kg/m\text{-}s$

$t_a = 15.5°C$

$P = 1.5 N/mm^2$

$t = 10°c$

$C = 1232 W/m^2/°c.$

$K = 0.002$

To Find

Design of Journal bearing.

Solution:

① Find the length of the Journal. (l) Assume the diameter of the Journal (d) as 100 mm. from PSG DDB 7.32 we find the ratio of l/d for centrifugal Pumps 1 to 2. Let us take $l/d = 1.6$

$$l = 1.6d = 1.6 \times 100 = 160 \text{ mm}$$

② bearing pressure.

$$p = \frac{W}{l \cdot d} = \frac{20000}{160 \times 100} = 1.25 \text{ N/mm}^2.$$

[bearing pressure for the pump is 1.5 N/mm², So therefore the above value of p is safe so dimensions of l & d are safe.]

③ $$\frac{Z \cdot N}{p} = \frac{0.017 \times 900}{1.25} = 12.24$$

So we have $\frac{ZN}{p} = 2844.5$ from PSG.DDB 7.31

The minimum value of the bearing ~~Lubrication~~

④ Centrifugal Pump Clearance ratio C/d $= 0.0013$

Assume.

~~1400~~

(5) Coefficient of friction. From PSG DDB 7.34

$$\mu = \frac{33.25}{10^{10}}\left[\frac{Zn}{P}\right]\left[\frac{D}{C}\right]+K.$$

$$= \frac{33.25}{10^{10}}\left[12.24\right] \times \frac{1}{0.0013} + 0.002$$

$$= 0.0051$$

(6) Heat generated

$$H_g = \mu \times W \times V = 0.0051 \times 20000 \times \left(\frac{\pi d N}{60}\right)$$

$$= 0.0051 \times 20000 \times \left(\frac{\pi \times 0.100 \times 900}{60}\right)$$

$$= 480.7 \text{ Watts.} \quad J = \frac{N \cdot m}{s} = W$$

(7) Heat dissipated

$$H_d = \cancel{\frac{(\Delta t + 18)^2 LD}{K}} \text{ — DDB at 7.35}$$

$$= \cancel{\frac{(19.75+18)^2 \times 160 \times 100}{0.437 \times 1000}}$$

$\Delta t = \frac{1}{2}t_o - t_a$

$= \frac{1}{2}(55^\circ - 15.5^\circ)$

$= 19.75$

$C = D - d.$

$\frac{C}{D} = 0.0013$

$$H_d = CA(t_b - t_a)$$

$$= C \cdot l \cdot d\,(t_b - t_a)\, W$$

$$(t_b - t_a) = \frac{1}{2}(t_o - t_a) = \frac{1}{2}(55^\circ - 15.5^\circ) = 19.75^\circ C.$$

$$= 1232 \times 0.16 \times 0.1 \times 19.75$$

$$= 389.3\ W$$

(Redesigned)

Amount of Artificial cooling required. (H_t)

= Heat generated (H_g) − Heat dissipated (H_d)

$$H_t = 480.7 - 389.3 = 91.4\ W$$

mass of lubrication

$H_t = m.S.t$

$= 1900 \times 10\, m$

$m = 91.4/19000 = 0.0048$ kg/s

Specific heat of oil (S) = 1840 to 2100 J/kg°C

Design a journal bearing for a centrifugal Pump with the following data! ③

Diameter of the journal = 150 mm.

Load on bearing = 40 kN

Speed of journal = 900 rpm.

Given!

$D = 150$ mm.

$W = 40$ kN $= 40 \times 10^3$ N

$N = 900$ rpm.

To Find Design a Journal bearing

Solution!

Step 1! Find the length of the journal (l)

For centrifugal pump from PSG 7.31

Let us 1 to 2 $\frac{L}{D}$

take $\frac{L}{D} = 1.5$ (Assume)

lenth of the journal $L = 1.5 \times 150$

$\boxed{L = 225 \text{ mm}}$

Step 2! Bearing Pressure (P)

$$P = \frac{W}{L \times D} = \frac{40000}{150 \times 225}$$

$\boxed{P = 1.18 \text{ N/mm}^2}$ $P = 11.8$ kg/cm²
7 to 14 Design Safe

Step 3! Selection of Lubricating oil

$\frac{ZN}{P}$. From PSG 7.31

$$\left[\frac{ZN}{P}\right]_{min.} = 2844.5$$

$$Z_{min} = \frac{2844.5 \times 11.85}{900}$$

$1.18\ N/mm^2 = 11.8\ kgf/cm^2$

$$Z_{min.} = 37.45 \text{ Centipoise.}$$

1 Poise = 100 CP
1CP = 0.01 Poise.
$1CP = 10^{-3}\ Pa\text{-}s = 10^{-?}\ MPa\text{-}s$

$$\boxed{Z = 40cP.}$$

$Z = 40 \times 10^{-3}\ kg/m\text{-}s.$

From PSG 7.41, at 40cP and 60°C

The Value of SAE 40 oil will be Selected.

Step 4: Calculation of Coefficient of friction:

$$\mu = \frac{33.25}{10^{10}}\left[\frac{ZN}{P}\right]\left[\frac{D}{c}\right] + K. \quad - 7.34$$

$C \rightarrow$ Diametral Clearance, $- 7.32$

based on Shaft diameter, 150 microns

1 micron $= 10^{-6}$ m
1m = 1000mm
150×10^{-6}
10^{3}
150×10^{-3}

$$\frac{D}{c} = \frac{150}{150 \times 10^{-3}} = 1000.$$

150 microns
10^{-6}.

$K = L/D = 1.5$ So let us take

Graph from 7.34.

$K = 0.002$

$$\mu = \frac{33.25}{10^{10}}\left[\frac{40 \times 900}{11.85}\right] \times 1000 + 0.002$$

$$\boxed{\mu = 0.0121}$$

Step 5: Calculate the heat Generated:

7.34

$$H_g = \mu \times W \times V$$

$$V = \frac{\pi DN}{60} = \frac{\pi \times 0.15 \times 900}{60}$$

$$\boxed{V = 7.068\ m/s}$$

$= 0.0121 \times 40000 \times 7.06$

$$\boxed{H_g = 3421.56\ W}$$

Step 6: $H_d = \frac{(K+C)}{K}LD$

$K = 0.484$

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

$= \frac{1}{2}(60° - 28°)$ $H_d = 80.6\ W$

$= 16°$

power total

(2) Design a journal bearing for a centrifugal pump with following data: (5)

Diameter of the journal, D = 150 mm

Load on bearing = 40 KN

Speed of journal = 900 rpm.

Operating temp t_o = 60 °C.

~~Atm~~ Atmospheric t_a = 28 °C.

Given

D = 150 mm

W = 40 kN

N = 900 rpm.

To Find

Design a journal bearing.

Solution.

Step 1. length of the journal L/d

centrifugal pump L/D ratio = 1 to 2 from PSG DDB (7.31)

$L/D = 1.5$

$L = 1.5 \times 150 = 225$ mm.

Step 2: Pressure developed

bearing pressure. $p = \frac{W}{l \cdot d} = \frac{40 \times 10^3}{225 \times 150} = 1.185 \, N/mm^2$

This pressure within the safe limit

(0.7 to 1.4 N/mm^2) from

PSG DB 7.37.

Step 3

Selection of Lubricating oil

$\left[\frac{Zn}{P}\right]_{min} = 2844.5$

$Z_{min} = \frac{2844.5 \times 1.185}{900} = 37.45$ centipoise.

$= 40 \, CP$

$40 \cdot C/D \quad \backslash \quad C/150 \quad c = 1/150$

Step 4 Coefficient of friction.

$D = 150$

$$\mu = \frac{33.25}{10^{10}}\left[\frac{Zn}{P}\right]\left[\frac{D}{c}\right] + K. \qquad \frac{1}{c} = 150$$

$K = 0.002$

$$= \frac{33.25}{10^{10}}\left[\frac{40 \times 500}{1.185}\right]\left[\frac{150}{150 \times 10^{3}}\right] + 0.002$$

$\mu = 0.0103.$

Step 5 Heat generated $H_g = \mu W V$ $\qquad V = \frac{\pi D N}{60}$ m/s.

$= 0.0103 \times 40 \times 10^{3} \times 7.068$

$= 2912.26\,W$

Heat dissipated $H_d = \frac{(\Delta t + 18)^2 LD}{k}$ $\qquad \Delta t = \frac{1}{2}(t_o - t_a)$

$=$

Take cm

$\frac{484}{1000}$

$$= \frac{(60 - 28 + 18)^2 \times 0.15 \times 0.225}{~~0.484~~ \quad 0.484}$$

$H_d = \text{—} \; W.$

$\frac{m^2}{\frac{N.m.S}{S \; N.m}}$ or $\frac{N.m}{Sec}$ $S.$ $\frac{N\,m^2}{m.S}$ $\frac{N.S}{m^2}$

Kurumi ②

Given

$W = 150\,kN \quad = 150 \times 10^{3}\,N$

$d = 300\,mm$

$N = 1800\,r.p.m.$

$p = 1.6\,N/mm^2$

$Z = 0.02\,kg/m\text{-}s$

$c = 0.25\,mm.$

$1cp = 10^{-3}\,\frac{N\text{-}s}{m^2}\,[Pa.s]$

$\boxed{\frac{N.s}{m^2} = kg/m\text{-}s}$

$\frac{kg \times m \times s}{s^2 \times m^2} = kg/m\text{-}s$

~~M = F x a~~

$F = ma$

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

To Find

1) length of the bearing.

2) Amount of heat to be removed by the Lubricant.

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hydrodynamic Lubricated bearing [Assumptions]

→ The Lubricant obeys Newton's Law of viscous flow.

→ The pressure is assumed to be constant throughout the film thickness

→ The Lubricant is assumed to be incompressible.

→ The viscosity is assumed to be constant throughout the film.

→ The flow is one dimensional.
(i.e) The side Leakage is neglected.

Types of lubrication based on the nature of Lubrication the journal bearing.

1. **Thick film type**: The bearing in which the Surfaces are completely separated from each other by the Lubricant is called thick film type.
2. **Thin film type**: The bearing in which although lubricant is present and Surfaces are partially in contact with each other is also termed boundary lubrication

Petroff & Mckee's Equation For friction Co-efficient (μ)

Petroff: Design the bearing depends upon many factors, material of bearing, surface finish, oil viscosity, operating Temperature, speed, dimension radial clearance. from many test found a relation b/w the friction co-efficient and other Parameters for a hydrodynamic

$$\mu = 2\pi^2 \left(\frac{z'n'}{p}\right)\left(\frac{D}{c}\right)$$

Mckee's brothers S.A, T.R) Theoritical Relation for find the μ.

$$\mu = \frac{33.25}{10^{10}} \left(\frac{zn}{p}\right)\left(\frac{D}{c}\right) + k.$$

Step 4 coefficient of friction. $h_0 \cdot c/D$ \ $c/150$ $c = 1/150$

$D = 150$

dimensionless parameter

Sommerfield Number (S)

Sommerfield Number is very important and useful non-dimensional number in hydrodynamic Lubrication Analysis because it contains many of the parameters specified by the designer.

Design procedure

$$S = \frac{Z n'}{P}\left[\frac{D}{c}\right]^2 = 14.3\times10^6$$

PSG pg 7.41

$\boxed{Z - kg/ms}$

- ↳ bearing dimension D & c
- ↳ Viscosity, Z
- ↳ Speed of rotation 'n'
- ↳ bearing Pressure (p)

Raimondi and Boyd Graphs

Raimondi and boyd have transformed the solution of Reynold's equation to chart form. have been prepared to by Raimondi and Boyd Graphs for various design parameters in dimensionless form are plotted with respect to Sommerfield number.

The chart provide accurate solution's for bearings of all proportions.

design Parameters which are given by chart

- h_0/c – minimum oil film thickness
- μ – co-efficient of friction
- Q – flow
- P/P_{max} – Maximum film pressure ratio.

See PSG Pg 7.40

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AP/mech.

Design of rolling contact bearing: It is also called as anti-frictional bearing, mainly difference between sliding contact bearing and antifriction bearing. in sliding contact b- that is contact b/w bearing & shaft to avoid friction. by using Lubrication, Thrust, axial and combination of Thrust, axis Radial loading. its noisy while to Running. PSG: 4.2 4.15

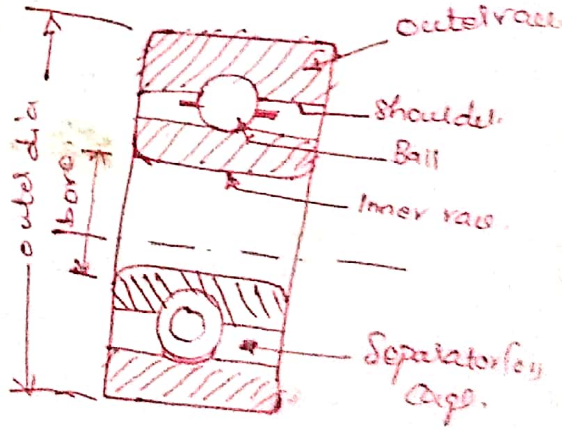
Types of Rolling contact bearing:

1) Based on the type of rolling element
   - → Ball bearing
   - → Roller bearing.

2) Based on the load to be carried:-
   - → Radial
   - → Angular contact
   - → Thrust bearing

Components of rolling contact bearing:

(i) Outer race (ii) Inner race (iii) Rolling (iv) Retaining cage (or) Separator.



Deep groove ball bearing

Types of Radial ball bearing.

1) Deep groove ball bearing →

2) Self-aligning ball bearing.
   - ↳ Self aliging internal
   - ↳ Self aliging external

3) Angular contact ball bearing.
   - → Uni directional
   - → Two "

4) Filling notch bearing

5) Counter bore "

6) Double row bearing

Coefficient of μ ... $\frac{l}{D}$ $\frac{9}{150}$ $c = \frac{1}{...}$

Design of Rolling contact bearing:

~~A80BC03~~ A6416

① Select a suitable deep groove ball bearing for supporting a radial load of 10KN and an axial load of 3KN for a life of 4000hr. at 800 rpm. Select from any series and calculate the expected average life of the selected bearing.

Given data:

Axial load $(F_a) = 3KN = 3\times10^3 N$.

Radial load $(F_r) = 10KN = 10\times10^3 N$.

$L_h = 4000 hr$.

$N = 800 rpm$.

To Find

Design of Deep groove Ball bearing. Life of the bearings.

Solution:

The A80BC03 deep ball grooving.

PSG. Pg. No ~~4.14~~ 4.15 — ~~6~~ ~~6~~ 6416.

$d = 80mm$.

Life of bearing. $\boxed{L = \frac{60 N L_h}{10^6}}$

psg No 4.15 Dynamic load capacity $c = 13000$ kgf

$= 13000\times10$ N. $= 130000$ N

Static load capacity $(C_o) = 12800$ kgf $= 12800\times10$ N.

From data book 4.4

$$F_a/C_o = \frac{3000}{128000} = \cancel{0.023}\ 0.023$$

we have Assume $F_a/C_o = 0.025$ 4.4 (PSG)

$Fa/Fr = \frac{3000}{10\times10^3} = 0.3.$

$\frac{Fa}{Fr} > e$

$0.3 > 0.22$

PSG 4.4

$X = 0.56$

$Y = 2.$

From data book 4.2

Equivalent load

$S \to$ Service factor

Assume 1.5

$P = [XFr + YFa]S$

$= [0.56 \times 10000 + 2 \times 3000]1.5$

$P = 17400 N.$

PSG 100 4.6 hrs and rpm (graph)

WKD hrs = 4000 hr

rpm = 800

$C/P = 5.75$

$\frac{C}{17400} = 5.75$

$\boxed{C = 100050 N.}$ Safe.

$L = \frac{60 \times n \times L_h}{10^6}$

$L = 60 \times 800 \times 400/10^6$

$\boxed{L = 192\ hr.}$

A ball bearing subjected to a Radial load of 5 KN is expected to have a life of 8000 hours at 1450 rpm with a reliability of 99%. cal. the dynamic load capacity of the bearing so that it can be selected from the manufacturer's catalogue based on a reliability of 90%.

Given

$F_r = 5 KN = 5 \times 10^3 N$

$L = 8000$ hrs

$N = 1450$ rpm.

Reliability of 99%.

To Find

Dynamic capacity of bearing (C)

Solution

WKT Probability of Survival.

PSG 4.2

$$\frac{L}{L'_{10}} = \left[\frac{\ln (1/p)}{\ln (1/p_{10})}\right]^{1/b}$$

$L'_{10} = 0.1053$

$b = 1.17$

$$\frac{L}{L'_{10}} = \left[\frac{\ln (1/0.99)}{0.1053}\right]^{1/1.17}$$

$$= \left[\frac{0.0100}{0.1053}\right]^{1/1.17}$$

$$\frac{L}{L'_{10}} = 0.1336$$

$$L'_{10} = \frac{8000}{0.1336} = 59880 \text{ hrs.}$$

From PSGDB 4.6 Corresponding to 59880 hrs and 1450 rpm, The loading $\frac{C}{P} = 9.11$

$C = 9.11 \times 5000 = 45550 N$

The bearing SKF 6018 Series 60

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