PIE Tech

POLLACHI INSTITUTE OF ENGINEERING AND TECHNOLOGY

(Approved by AICTE and Affiliated to Anna University)

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Department of Mechanical Engineering

Regulation 2021

III Year – V Semester

ME3591 - Design of Machine Elements

ME3591	DESIGN OF MACHINE ELEMENTS	L	T	P	C
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COURSE OBJECTIVES

1. To learn the various steps involved in the Design Process.
2. To Learn designing shafts and couplings for various applications.
3. To Learn the design of temporary and permanent Joints.
4. To Learn designing helical, leaf springs, flywheels, connecting rods and crank shafts for various applications.
5. To Learn designing and select sliding and rolling contact bearings, seals and gaskets.
(Use of PSG Design Data book is permitted)

UNIT – I FUNDAMENTAL CONCEPTS IN DESIGN 12

Introduction to the design process - factors influencing machine design, selection of materials based on mechanical properties - Preferred numbers- Direct, Bending and torsional loading- Modes of failure - Factor of safety – Combined loads – Principal stresses – Eccentric loading – curved beams – crane hook and 'C' frame- theories of failure – Design based on strength and stiffness – stress concentration – Fluctuating stresses – Endurance limit –Design for finite and infinite life under variable loading - Exposure to standards.

UNIT – II DESIGN OF SHAFTS AND COUPLINGS 12

Shafts and Axles - Design of solid and hollow shafts based on strength, rigidity and critical speed – Keys and splines – Rigid and flexible couplings.

UNIT – III DESIGN OF TEMPORARY AND PERMANENT JOINTS 12

Threaded fasteners - Bolted joints including eccentric loading, Knuckle joints, Cotter joints – Welded joints- Butt, Fillet and parallel transverse fillet welds – welded joints subjected to bending, torsional and eccentric loads, riveted joints for structures - theory of bonded joints.

UNIT – IV DESIGN OF ENERGY STORING ELEMENTS AND ENGINE COMPONENTS 12

Types of springs, design of helical and concentric springs–surge in springs, Design of laminated springs - rubber springs - Flywheels considering stresses in rims and arms for engines and punching machines-- Solid and Rimmed flywheels- connecting rods and crank shafts

UNIT – V DESIGN OF BEARINGS AND MISCELLANEOUS ELEMENTS 12

Sliding contact and rolling contact bearings - Hydrodynamic journal bearings, Sommerfeld Number, Raimondi & Boyd graphs, -- Selection of Rolling Contact bearings –Design of Seals and Gaskets.

TOTAL: 60 PERIODS

OUTCOMES: At the end of the course the students would be able to

1. Explain the design machine members subjected to static and variable loads.
2. Apply the concepts design to shafts, key and couplings.
3. Apply the concepts of design to bolted, Knuckle, Cotter, riveted and welded joints.
4. Apply the concept of design helical, leaf springs, flywheels, connecting rods and crank shafts.
5. Apply the concepts of design and select sliding and rolling contact bearings, seals and gaskets.

TEXT BOOKS:

1. Bhandari V B, "Design of Machine Elements", 4th Edition , Tata McGraw-Hill Book Co, 2016
2. Joseph Shigley, Richard G. Budynas and J. Keith Nisbett "Mechanical Engineering Design", 10th Edition, Tata McGraw-Hill , 2015.

REFERENCES:

1. Ansel C Ugural, "Mechanical Design – An Integral Approach", 1st Edition, Tata McGraw-Hill Book Co, 2004.
2. Merhyle Franklin Spotts, Terry E. Shoup, and Lee EmreyHornberger, "Design of Machine Elements" 8th Edition, Printice Hall, 2004.
3. Robert C. Juvinall and Kurt M. Marshek, "Fundamentals of Machine component Design",6th Edition, Wiley, 2017.
4. Sundararajamoorthy T. V. and Shanmugam .N, "Machine Design", Anuradha Publications, Chennai, 2003.
5. Design of Machine Elements | SI Edition | Eighth Edition | By Pearson by M. F. Spotts, Terry E. Shoup, et al. | 25 March 2019

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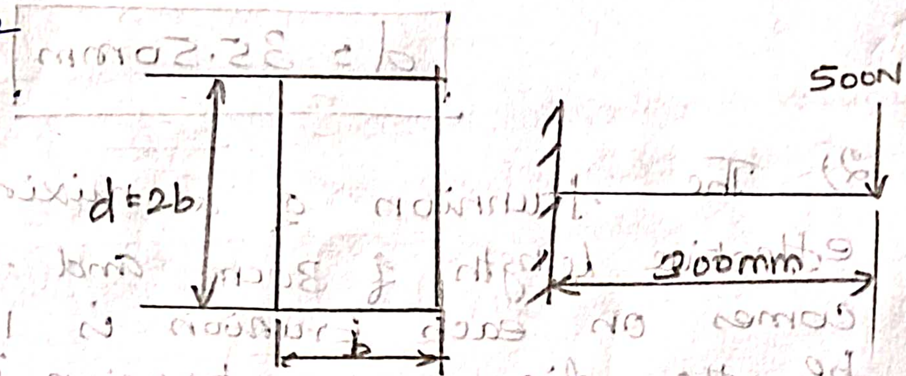
	1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**1**	**2**	**3**
1	2	2	3					1	1			2	3	2	2
2	2	2	3					1	1			2	3	2	2
3	2	2	3					1	1			2	3	2	2
4	2	2	3					1	1			2	3	2	2
5	2	2	3					1	1			2	3	2	2
Low (1) ; Medium (2) ; High (3)															

UNIT-I Fundamental concept in design:

1. An electric motor weighing 500N is mounted on a short cantilever beam of uniform rectangular cross section. The weight of motor act at a distance of 300mm from the support. The depth of the section is twice the width. Det the cross section of the beam. The allowable stress in the beam is $40 N/mm^2$



data:

$P = 500N$

$l = 300 mm$

$d = 2b$

$\sigma_b = 40 N/mm^2$

Soln:

1) $I = \frac{bd^3}{12}$

2) $Z = \frac{bd^2}{6}$

Bending stress $\sigma_b = \frac{M_b}{Z}$

$$\sigma_b = \frac{\text{Load}(P) \times \text{Length}(l)}{Z}$$

$$= \frac{P \times l}{Z} \rightarrow (1)$$

$$Z = \frac{bd^2}{6} = \frac{b(2b)^2}{6} = \frac{4b^3}{6} \rightarrow (2)$$

Eqn (2) in (1)

$$\sigma_b = \frac{P \times l}{\frac{4b^3}{6}}$$

$$40 = \frac{500 \times 300 \times 6}{4 \times b^3}$$

$$b^3 = 5622$$

$$\boxed{b = 17.78\,mm}$$

$$d = 2b = 2 \times 17.78$$

$$\boxed{d = 35.50\,mm}$$

2) The trunnion of a mixing machine have an effective length of 30cm and the weight which comes on each trunion is 12500N what should be the dia of the trunnion if fiber stress is not to exceed $35\,N/mm^2$?

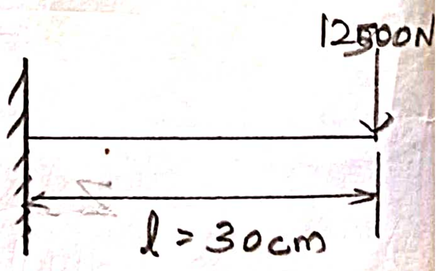
data:

1) length (l) = 300mm
2) $P = 12500\,N$
3) $\sigma_b = 35\,N/mm^2$
4) circular and Cantilever

To find

Diameter (d)

Soln:



(1) Modulus of section] $Z = \frac{\pi d^3}{32}$

2) Bending stress] $\sigma_b = \frac{M_b}{Z}$

$$= \frac{P \times l}{Z}$$

$$\sigma_b = \frac{P \times l}{\frac{\pi d^3}{32}}$$

$$35 = \frac{12500 \times 300 \times 32}{\pi d^3}$$

$$d^3 = \frac{12500 \times 300 \times 32}{35 \times \pi}$$

$$\boxed{d = 47.79 \text{ mm}}$$

3) A Cantilever of span 500mm carries a vertical downward load of 6 kN at free end. Assume the factor of safety of 3. yield value of 350 mpa and find the economical section for the cantilever among.

(a) circular cross section of diameter 'd'

(b) Rectangular section of depth 'h' and width 't' with $\frac{b}{t} = 2$

(C) I section of depth 7t, and flange width 5t where t is thickness. specify the dimension and cross sectional area.

data:

length $(l) = 500$mm

$P = 6\text{kN} = 6 \times 10^3 N$

$\sigma_y = 350 \text{mpa} = 350 \times \frac{10^6}{10^6} = N/mm^2$

$$\boxed{= 350 \text{ N/mm}^2}$$

F.O.S = 3

depth $(h) = 2t$

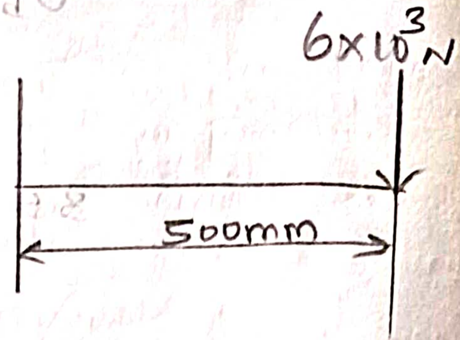
To find:

a) circular cross section dia (d)

b) Rectangle section (h, t) (depth, width)

Soln :

(1) Circular cross section



Factor of Safety $FOS = \dfrac{\text{Yield Stress}}{\text{Bending stress}}$

$$3 = \frac{350}{\sigma_b}$$

$$\boxed{\sigma_b = 116.67\ N/mm^2}$$

(1) Modulus of section $Z = \dfrac{\pi d^3}{32}$

2) Bending stress $\sigma_b = \dfrac{M_b}{Z}$

$$116.67 = \frac{p\times l}{Z}$$

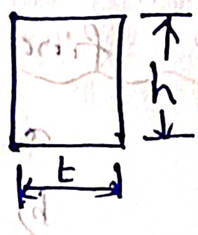
$$116.67 = \frac{6\times10^3 \times 500}{\dfrac{\pi \times d^3}{32}}$$

$$d^3 = \frac{6\times10^3 \times 500 \times 32}{116.67 \times \pi}$$

$$\boxed{d = 63.98\ mm}$$

(ii) Rectangular cross section depth (h) width (t)

Modulus of section



$$Z = \frac{bd^2}{6} = \frac{th^2}{6}$$

$$Z = \frac{t(2t)^2}{6}$$

$$Z = \frac{4t^3}{6} \longrightarrow (1)$$

(ii) Bending stress $\sigma_b = \frac{M_b}{Z}$

$$116.67 = \frac{P \times l}{Z}$$

$$116.67 = \frac{6 \times 10^3 \times 500}{\frac{4t^3}{6}}$$

$$116.67 = \frac{6 \times 10^3 \times 500 \times 6}{4t^3}$$

$$t^3 = \frac{6 \times 10^3 \times 500 \times 6}{4 \times 116.67}$$

$$\boxed{t = 33.79 \text{ mm}}$$

$h = 2t = 2 \times 33.79$

$$\boxed{h = 67.57 \text{ mm}}$$

Stress variable loading

1. Static Stress:

The stress which does not change in magnitude or in direction.

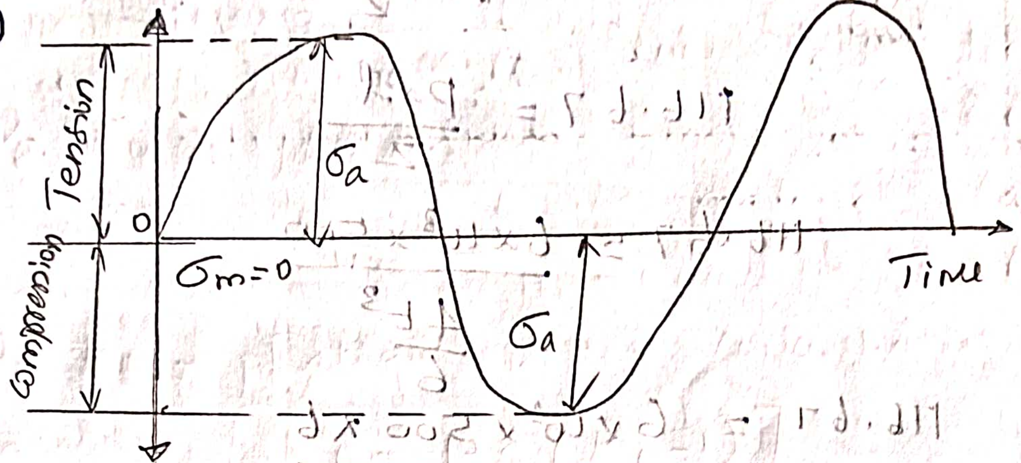
Varying Stresses:

Varying stress refer to the stress in which magnitude or direction or both are changing.

The following are the types of variable stress.

(i) completely Reversed (or) cyclic stresses:

Stresses which change from one value of tension to the same value of compression is known as completely reversed as cyclic stresses (σ_y)

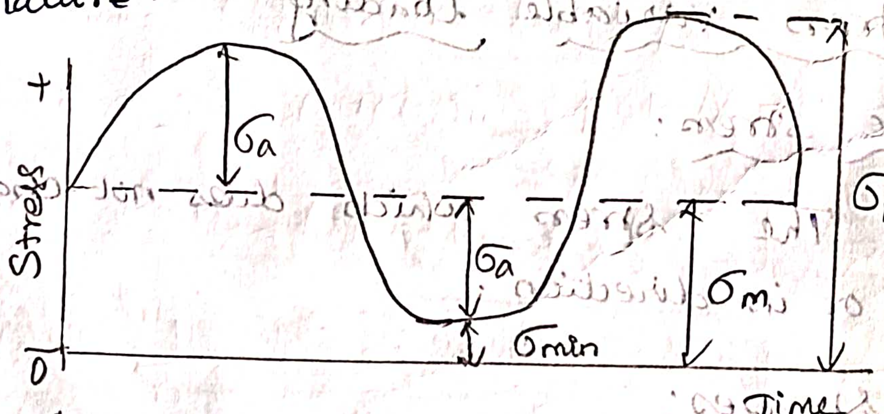


(ii) Fluctuating stresses:

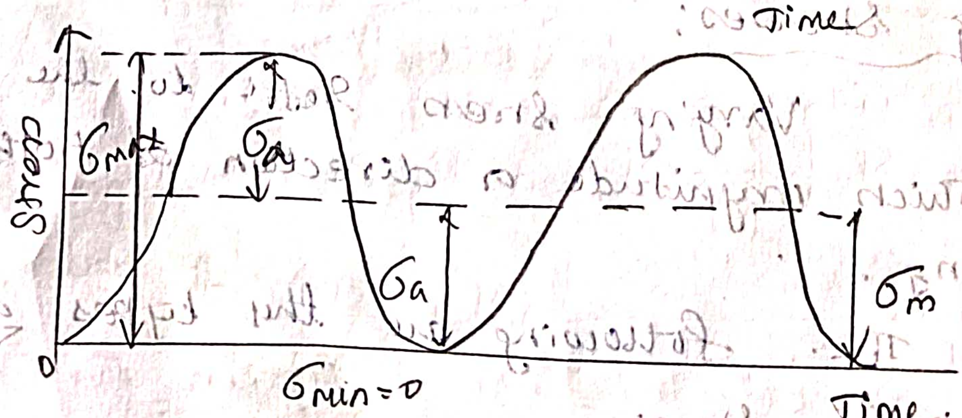
Stresses which vary from a min value to a max of same nature (compressive or tensile) are called fluctuating stresses.

(iii) Repeated stress:

Repeated stress refer to a stress which varies from zero to a max value of same nature.



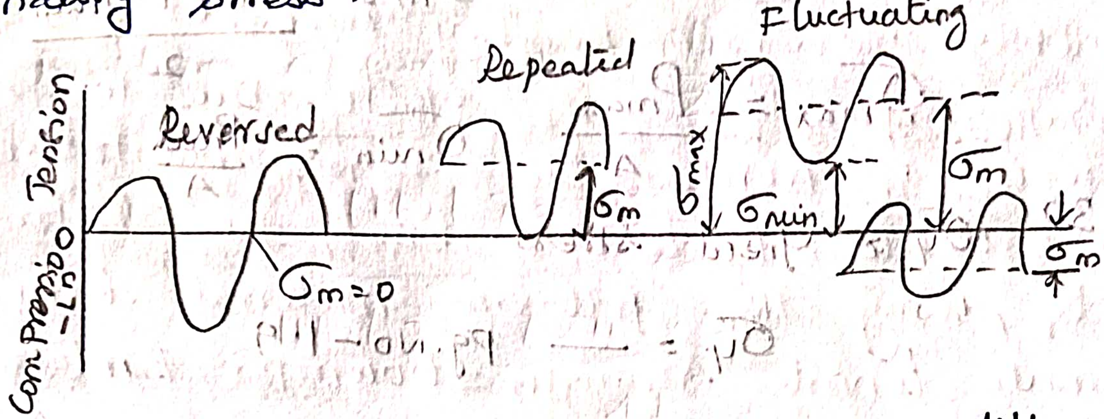
Fluctuating Stress



Repeated stress

(iv) Alternating stress:

Stress varying from a min value to a max value of the opposite nature is known as alternating stress.



Term used for the variable stress condition

1. Mean or average stress $\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$

2. Variable (or) amplitude stress $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$ → Pg. No 7.6

3. Stress ratio $R = \frac{\sigma_{max}}{\sigma_{min}}$

If it is a completely reversed stress

$$\sigma_{min} = -\sigma_{max}$$

$$R = \frac{\sigma_{max}}{\sigma_{min}} = \frac{\sigma_{max}}{-\sigma_{max}} = -1$$

1) Soderberg eqn:

P. NO: 7.6

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + K_f \frac{\sigma_a}{\sigma_{-1}}$$ → ductile Material

(or)

Fatigue stress concentration factor K_f and factor K_L, K_{sf}, K_{s2} and

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + K_f \frac{\sigma_a}{(\sigma_{-1}) \cdot K_L \cdot K_{sf} \cdot K_{S_2} \cdot K_R \cdot K}$$ → 1.158

(or) $\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_{-1}}$ → 1.158

where

1) n = factor of safety

2) σ_m = Mean stress $\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$

$$\sigma_{max} = \frac{P_{max}}{A} \qquad \sigma_{min} = \frac{P_{min}}{A}$$

3) σ_y = Yield stress

$$\sigma_y = \text{—} \quad \text{pg. NO} - 1.9$$

4) K_f = Fatigue stress concentration factor

$$\left[K_f = 1 + q(K_t - 1)\right]$$

where 'q'. value is not given in question

Assume $\boxed{q = 1}$

Assume $K_t = 0.8$ [when question is not given]

5) $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$ [pg. NO - 7.6]

6) σ_{-1} = Endurance limit

$\sigma_{-1} = 0.46\,\sigma_u$ [P. NO - 1.43] [If not gn in question]

Goodman eqn: (brittle material)

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma_{-1}} \qquad \text{PSG} - 7.4$$

Considering fatigue stress concentration factor K_f, and factor K_L, K_{SF}, K_{SZ} and K

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_u} + K_f \frac{\sigma_a}{K_L \cdot K_{SF} \cdot K_{SZ} \cdot K_R \cdot K}$$

or

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_{-1M}} \quad \left[\text{Assume } K_f = 1\right]$$

1) n = factor of safety

2) $\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2}$

$$\sigma_{max} = \frac{P_{max}}{A}, \quad \sigma_{min} = \frac{P_{min}}{A}$$

3) $\sigma_a = \dfrac{\sigma_{max} - \sigma_{min}}{2}$

$$\sigma_{max} = \frac{P_{max}}{A} \quad \sigma_{min} = \frac{P_{min}}{A}$$

4) σ_u = ultimate stress

$\sigma_u = \underline{\quad\quad}$ P.g - 1.9

5) σ_{-1} = endurance limit

$\sigma_{-1} = \longrightarrow$ pg. 1.43

6) $K_f = 1 + q(K_t - 1)$ Assume $q = 1$

$K_t = 0.8$

or

$K_f = D/d$ (or) R/r

7) K_L = load factor (DB 1.43)

K_{sf} - Surface finish factor

K_{sz} - size factor (DB 1.43) $\tan\theta = \dfrac{\sigma_a}{\sigma_m}\sqrt{2}$

K_R = Reliability factor

Gerber Eqn:

$$\sigma_a = \sigma_{-1}\left[1 - \left(\frac{\sigma_m}{\sigma_u}\right)^2\right]$$

Combined steady and variable stresses: - P.NO - 7.6

$$\sigma_{eq} = \frac{\sigma_y}{n} = \sigma_m + K_f \frac{\sigma_a \sigma_y}{\sigma_{-1}} \quad \tau_{eq} = \frac{\tau_y}{n} = \tau_m + K_f \frac{\tau_a \tau_y}{\tau_{-1}}$$

shear stress

Pb: 1 (1.185)

1) A 45mm dia steel shaft is subjected to a static axial load of 125KN (tensile). A completely reversed bending moment of 358 N-m is superimposed on the shaft. Due to the presence of notches, the stress concentration factor was estimated to be 1.5. Find the factor of safety, using the following material data. ultimate tensile strength = 600 mpa, Endurance limit = 250 mpa.

data:

1) $d = 45mm$
2) $P = 125\,kN = 125\times10^3 N$
3) Reverse bending $(M_b\,max) = 358$ N-m
$= 358\times10^3$ N-mm
$(M_b\,min) = -358\times10^3$ N-mm
4) Stress factor $(K_t) = 1.5$
5) $\sigma_u = 600\,mpa = 600 N/mm^2$
6) $\sigma_{-1} = 250\,mpa = 250 N/mm^2$

Soln: find : Fos (or) N

From using Goodman eqn

$$\frac{1}{n} = K_f\left(\frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma_{-1}}\right) \rightarrow (1)$$

$K_f = 1 + q(K_t - 1)$ Assume $q=1$

$= 1+1[1.5-1]$

$\boxed{K_f = 1.5}$

2) $\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2} \longrightarrow (2)$

$\sigma_{max} = \dfrac{M_{b\,max}}{Z}$

$Z = \dfrac{\pi d^3}{32}$ ⟶ (Circular Shaft)

$= \dfrac{\pi \times (45)^3}{32}$

$\boxed{Z = 8.94 \times 10^3}$

$\sigma_{max} = \dfrac{M_{b\,max}}{Z} = \dfrac{358 \times 10^3}{8.94 \times 10^3} = 40.04\ N/mm^2$

$\sigma_{min} = \dfrac{M_{b\,min}}{Z} = \dfrac{-358 \times 10^3}{8.94 \times 10^3} = -40.04\ N/mm^2$

$\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2} = \dfrac{40.04 - 40.04}{2}$

$\boxed{\sigma_m = 0}$

3) $\sigma_a = \dfrac{\sigma_{max} - \sigma_{min}}{2} = \dfrac{40.04 + 40.04}{2}$

$\sigma_a = 40.04\ N/mm^2 \longrightarrow (3)$

Eqn (2) and (3) in eqn 1

$\dfrac{1}{n} = k_f \left(\dfrac{\sigma_m}{\sigma_u} + \dfrac{\sigma_a}{\sigma_{-1}} \right]$

$= 1.5 \left[0 + \dfrac{40.04}{250} \right]$

$\dfrac{1}{n} = 1.5 (0.160]$ $\qquad 1 = 0.24n$

$\boxed{n = 4.16}$ $\qquad 1 - 0.24$

n_2

2) A simply supported beam has concentrated load at the centre which fluctuates a value from P to 4P. The span of the beam is 500mm and its cross section is circular with a diameter of 60mm. Beam material is cold drawn 0.2% carbon steel. Calculate the max permissible value of P for a factor of safety of 1.3. Beam surface is ground. ultimate stress of 700mpa yield stress is 500mpa, $\sigma_{-1} = 330$ mP. calculate the value of P. Take size of factor is 0.85 and surface finish factor 0.9

1.188

data:

1. load $P_{max} = 4P$

 $P_{min} = P$

2. length $(l) = 500mm$

3. Dia $(d) = 60mm$

4. ultimate stress $(\sigma_u) = 700mpa = 700 N/mm^2$

5. Yield stress $(\sigma_y) = 500mpa = 500 N/mm^2$

6. Endurance limit $(\sigma_{-1}) = 300mpa = 330 N/mm^2$

7. F.O.S (or) $n = 1.3$

8. $K_{sz} = 0.85$ 9. $K_{sf} = 0.9$

To find

P

Soln: [Sodeberg eqn]

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + K_f \left[\frac{\sigma_a}{\sigma_{-1} [K_{sf} \times K_{sz})} \right] \rightarrow (1)$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\sigma_{max} = \frac{P_{max}}{A} = \frac{4P}{\frac{\pi}{4} d^2} = \frac{4P}{\frac{\pi}{4} (60)^2}$$

$$\boxed{\sigma_{max} = (1.415 \times 10^{-3})\, P \ N/mm^2}$$

$$\sigma_{min} = \frac{P_{min}}{A} = \frac{P}{\frac{\pi}{4}(60)^2} = \cancel{2.8227}\ 3.536 \times 10^{-4} P \ N/mm^2$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$= \frac{[(1.415 \times 10^{-3}) + (3.538 \times 10^{-4})] P}{2}$$

$$\boxed{\sigma_m = (8.754 \times 10^{-4})\, P \ N/mm^2} \rightarrow (3)$$

2) $K_f = 1 + q(K_t - 1)$

$= 1 + 1(0.8 - 1)$

$\boxed{K_f = 0.8}$

3) $$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$= \frac{[(1.415 \times 10^{-3}) - (3.538 \times 10^{-4})] P}{2}$$

$$\boxed{\sigma_a = (5.3 \times 10^{-4})\, P \ N/mm^2} \rightarrow (4)$$

Eqn (4) (3) (2) in (1)

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + K_f \left[\frac{\sigma_a}{\sigma_{-1}[K_{s1} \times K_{s2}]}\right]$$

$$\frac{1}{1.3} = \frac{(8.754 \times 10^{-4}) P}{500} + 0.8 \left[\frac{(5.3 \times 10^{-4}) P}{330\,(0.85 \times 0.9)}\right]$$

$$\frac{1}{1.3} = [(1.75 \times 10^{-6}) P + (1.847 \times 10^{-6}) P]$$

$$\boxed{P = 214.067 \times 10^3 N}$$

∴ In Given load at Centre.

$$\boxed{P = \frac{214.067 \times 10^3}{2} = 107.033 \times 10^3 N}$$

Goodsman eqn:

$$\frac{1}{n} = k_d \left[\frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma_{-1}(k_{s1} \times k_{s2})} \right]$$

$$\frac{1}{1.3} = 0.8 \left[\frac{(8.754 \times 10^{-4}) P}{700} + \frac{(5.3 \times 10^{-4}) P}{(300 \times 0.85 \times 0.9)} \right]$$

$$0.77 = 0.8 \left[\left(\frac{8.754 \times 10^{-4}}{700} + \frac{5.3 \times 10^{-4}}{300 \times 0.85 \times 0.9} \right) P \right]$$

$$0.77 = (2.85 \times 10^{-6}) P$$

$$\boxed{P = 270.369 \times 10^3 N}$$

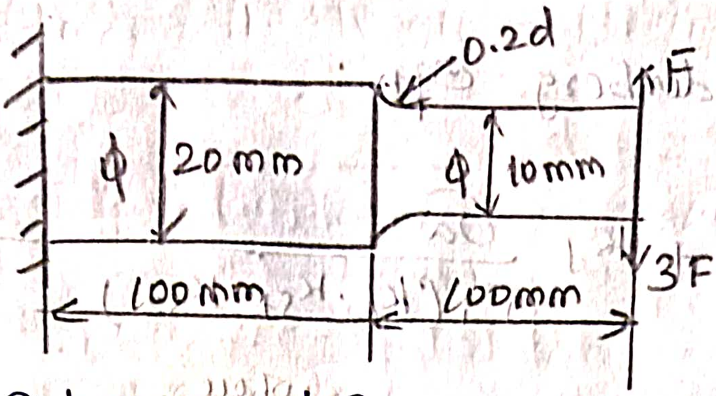
In Given load at centre.

$$P = \frac{270.369 \times 10^3}{2}$$

$$\boxed{P = 135.185 \times 10^3 N}$$

1.144

3. A Cantilever beam fig made of C 40 Steel of circular cross section is subjected to a load that varies from F (compressive) to 3F (tensile). Det the value of F that this beam can withstand. Assume FOS = 2.1, stress concentration factor = 1.42, notch sensitivity factor = 0.3, surface finish factor = 0.85 and size factor = 0.86



data:

D.b Pg.No - 1.9

1\. C40 Steel ultimate Stress $(\sigma_u) = 58\ kgf/mm^2$

$= 58 \times 10\ N/mm^2$

$= 580\ N/mm^2$

Yield Stress $(\sigma_y) = 33\ kgf/mm^2$

$= 33 \times 10 = 330\ N/mm^2$

2\. load max $P = 3F$

min $P = -F$ (In up direction)

3\. factor of safety (FOS) or $n = 2.1$

$K_t = 1.42$

$q = 0.3$

$K_{sf} = 0.85$

$K_{sz} = 0.86$

$l_{max} = 200mm$

$l_{min} = 100mm$

$d_1 = 20\ mm$ $d_2 = 10mm$

$\frac{d_1}{d_2} = 0.2d$

$\frac{20}{10} = 0.2d$

$\boxed{d = 10mm}$

To find

Value of F

Soln:

(i) using Soderberg eqn:

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + K_f \frac{\sigma_a}{\sigma_{-1}(K_L \cdot K_{sf} \cdot K_{sz})} \rightarrow (1)$$

where Assume
K_L = load factor = 1 (not given)
K_{sf} = Surface finish factor
K_{sz} = size factor.

(1) σ_m = Mean Stress

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\sigma_{max} = \frac{M_{b.max}}{Z}$$

$$= \frac{P_{max} \alpha\, l_{max}}{\frac{\pi d^3}{32}}$$

$$= \frac{3F \times 100}{\frac{\pi (10)^3}{32}}$$

Z = circular

$$Z = \frac{\pi d^3}{32}$$

$$\boxed{Z = 98.174\ mm^3}$$

$$\sigma_{max} = \frac{300F}{98.174}$$

$$\boxed{\sigma_{max} = 3.06F\ N/mm^2} \rightarrow (2)$$

$$\sigma_{min} = M_{b\,min}$$

$$= \frac{P_{min} \alpha\, l_{min}}{\frac{\pi d^3}{32}}$$

$$= \frac{-F \times 100}{98.174}$$

$$\boxed{\sigma_{min} = -1.019F\ N/mm^2} \rightarrow (3)$$

Eqn (4) and (3) in (2)

Mean stress $\sigma_m = \frac{3.06F - 1.019F}{2}$

$$\boxed{\sigma_m = 1.02F\ N/mm^2} \rightarrow (5)$$

2) $K_f = 1 + q(K_t - 1)$

$= 1 + 0.3(1.42 - 1)$

$$\boxed{K_f = 1.126} \rightarrow (6)$$

(3) Amplitude Stress

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$= \frac{3.06F + 1.019F}{2}$$

$$\boxed{\sigma_a = 2.0395F} \rightarrow (7)$$

4) Endurance limit (Steel) — [D.B - P.NO - 1.48]

$\sigma_{-1} = 0.5\,\sigma_u$

$= 0.5 \times 580$

$$\boxed{\sigma_{-1} = 290\ N/mm^2} \rightarrow (8)$$

using Soderberg eqn:

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + K_f \frac{\sigma_a}{\sigma_{-1}(K_{sf} \times K_L \times K_{sz})}$$

Assume $K_L = 1$

$$\frac{1}{2.1} = \frac{1.02F}{330} + 1.126\left[\frac{2.0395F}{290(0.85 \times 0.86)}\right]$$

$$\frac{1}{2.1} = 0.014F$$

$$\boxed{F = 34.19\ N}$$

using Goodman eqn

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_u} + K_f \left[\frac{\sigma_a}{\sigma_{-1}(K_L \cdot K_{SF} \cdot K_{SZ})} \right]$$

$$\frac{1}{2.1} = \frac{1.02F}{580} + 1.126 \left[\frac{2.0395F}{290(0.85 \times 0.86)} \right]$$

$$\boxed{F = 37.82\ N}$$

The maximum load the beam can withstand is the minimum of F obtained from Goodman and Soderberg eqn:

$$\therefore \boxed{F = 34.19\ N}$$

Combined Stress (σ, τ)

$$\frac{1}{n} = \sqrt{\left(\frac{\sigma_{eqn}}{\sigma_y}\right)^2 + \left(\frac{\tau_{eqn}}{\tau_y}\right)^2}$$ D.B pg.NO 7.6

where n (or) FOS = Factor of Safety

σ_{equ} = equivalent static stress

$$\sigma_{equ} = \frac{\sigma_y}{n} = \sigma_m + K_f \frac{\sigma_a \cdot \sigma_y}{\sigma_{-1}(K_{sf} \times K_{sz} \times K_L)}$$

σ_y = yield stress - D.B pg-1.9

σ_m = Mean stress $= \frac{\sigma_{max} - \sigma_{min}}{2}$

σ_{max} (or) $\sigma_{min} = \frac{P}{a}$ (or) $\frac{M_b}{Z}$

σ_a = amplitude stress

$\sigma_a = \sigma_{max} - \sigma_{min}$

K_f = Fatigue stress concentration factor

$$K_f = 1 + q(K_t - 1)$$

Assume $q = 1$ (not gn is question)

$K_t = \frac{D}{d}$ (or) Assume $K_t = 0.8$

σ_{-1} = Endurance limit [D.B Pg - 1.43]

Torque con shear stress

τ_{equ} = Equivalent shear stress

$$\tau_{equ} = \frac{\tau_y}{n} = \frac{\tau_m + K_f \tau_a \cdot \tau_y}{\tau_{-1} (K_{SF} \times K_{SL} \times K_L)} \rightarrow 7.6$$

τ_y = Yield stress (D.B Pg - 19)

τ_m = Mean shear stress

$$= \frac{\tau_{max} + \tau_{min}}{2}$$

$$\tau_{max} = \frac{16 M_{t(max)}}{\pi d^3} \quad \tau_{min} = \frac{16 M_{t(min)}}{\pi d^3}$$

$$\tau_a = \frac{\tau_{max} - \tau_{min}}{2}$$

$$\tau_{-1} = \sigma_u \times 0.22 \rightarrow 1.43$$

$$\boxed{\tau_y = \frac{\sigma_y}{2}}$$

4) A hot rolled steel shaft of 40mm dia is subjected to a torsional moment that varies from 330 Nm to −100 Nm and an applied bending moment which rises from 400 Nm to −220 Nm. The material of the shaft has an ultimate strength of 550 MN/m² and yield strength of 410 MN/m². Find the approximate factor of safety using Soderberg eqn allowing endurance limit to be half the ultimate strength and size factor and surface factor to be 0.85 and 0.62 respectively.

data:

Diameter of shaft $d = 40mm$

Max torsional moment $(M_t)_{max} = 330\,Nm = 330 \times 10^3\,N\text{-}mm$

min " " $(M_t)_{min} = -100\,N\text{-}m = -100 \times 10^3\,N\text{-}mm$

Max bending moment $(M_b)_{max} = 400\,N\text{-}m = 400 \times 10^3\,N\text{-}mm$

Min bending moment $(M_b)_{min} = -220\,N\text{-}m = 200 \times 10^3\,N\text{-}mm$

ultimate strength $\sigma_u = 550\,MN/m^2 = 550\,N/mm^2$

Yield strength $\sigma_y = 410\,MN/m^2 = 410\,N/mm^2$

Endurance limit $\sigma_{-1} = \frac{1}{2}\sigma_u$

$\sigma_{-1} = \frac{1}{2} \times 550 = 275\,N/mm^2$

Size factor $K_{sz} = 0.85$

Surface finish factor $K_{sf} = 0.62$

To find

Find factor (n)

Soln:

Combined stress

$$\frac{1}{n} = \sqrt{\left(\frac{\sigma_{equ}}{\sigma_y}\right)^2 + \left(\frac{\tau_{equ}}{\tau_y}\right)^2} \rightarrow (1)$$

→ D.B Pg 7.6

(1) σ_{equ} = Equivalent stress

$$\sigma_{equ} = \frac{\sigma_y}{n} = \sigma_m + K_f\left(\frac{\sigma_a \cdot \sigma_y}{\sigma_{-1}(K_{SF} \cdot K_{SZ})}\right) \rightarrow (2)$$

7.6

$$\sigma_{max} = \frac{M_b\, max}{Z} \qquad Z = \frac{\pi d^3}{32}$$

$$= \frac{440 \times 10^3}{6283.18} \qquad = \frac{\pi \times (40)^3}{32}$$

$$\boxed{\sigma_{max} = 70.02\ N/mm^2} \qquad \boxed{Z = 6283.18\ mm^3}$$

$$\sigma_{min} = \frac{M_b\, min}{Z} = \frac{-220 \times 10^3}{6283.18} = -35.01\ N/mm^2$$

$$\sigma_M = \frac{70.02 - 35.01}{2} = 17.5\ N/mm^2 \rightarrow (3)$$

2) $K_f = 1 + q(K_f - 1)$ Assume $q = 1$

$= 1 + 1(0.8 - 1)$ $K_t = 0.8$

$$\boxed{K_f = 0.8} \rightarrow (4)$$

3) $$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{70.02 - (-35.01)}{2}$$

$$\boxed{\sigma_a = 52.5\ N/mm^2} \rightarrow 5$$

4) Endurance limit

$$\sigma_{-1} = \frac{1}{2}\sigma_u = \frac{1}{2} \times 550 = 275\ N/mm^2 \rightarrow (6)$$

Eqn (6), (5), (4) (8) in (2)

$$\sigma_{equ} = 17.5 + 0.8 \left[\frac{52.5 \times 410}{275 (0.62 \times 0.85)} \right]$$

$$\sigma_{equ} = 136.32 \ N/mm^2 \rightarrow (7)$$

(2) τ_{equ} = Equivalent shear stress → Pg. No - 7.8

$$= \frac{\tau_y}{n} = \tau_m + K_f \cdot \frac{\tau_a - \tau_y}{\tau_{-1} (K_{sf} \times K_{sc})} \rightarrow (8)$$

(i) $\tau_m = \dfrac{\tau_{max} + \tau_{min}}{2}$

$$\tau_{max} = \frac{16 (M_t)_{max}}{\pi d^3}$$

$$= \frac{16 \times 330 \times 10^3}{\pi \times (40)^3} = 26.26 \ N/mm^2$$

$$\tau_{min} = \frac{16 (M_t)_{min}}{\pi d^3} = \frac{16 \times (-100 \times 10^3)}{\pi \times (40)^3}$$

$$\tau_{min} = -7.95 \ N/mm^2$$

$$\tau_m = \frac{26.26 - 7.95}{2} = 9.15 \ N/mm^2 \rightarrow (9)$$

b) we know that $K_f = 0.8$

c) $\tau_a = \dfrac{\tau_{max} - \tau_{min}}{2} = \dfrac{26.26 - (-7.95)}{2} = 17.10$

d) $\tau_y = \dfrac{\sigma_u}{2} = \dfrac{410}{2} = 205 \ N/mm^2 \rightarrow (11)$

e) Endurance limit for shear stress

$\tau_{-1} = 0.22 \sigma_u$ Pg. No - 1.42

$$\tau_{-1} = 0.22 \times 550 = 121 \ N/mm^2 \rightarrow (12)$$

Eqn (12) (11) (10) (9) (4) in (8)

$$\tau_{equ} = 9.15 + 0.8\left[\frac{17.10 \times 205}{121 \times (0.62 \times 0.85)}\right]$$

$$\tau_{equ} = 53.23\ N/mm^2 \rightarrow 13$$

Eqn (7), (13), 11, σ_y in Eqn (1)

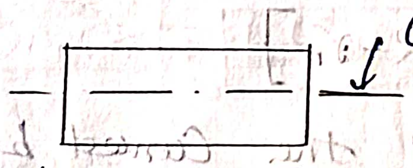
$$\frac{1}{n} = \sqrt{\left(\frac{\sigma_{equ}}{\sigma_y}\right)^2 + \left(\frac{\tau_{eqn}}{\tau_y}\right)^2}$$

$$\frac{1}{n} = \sqrt{\left(\frac{136.32}{410}\right)^2 + \left(\frac{53.23}{205}\right)^2}$$

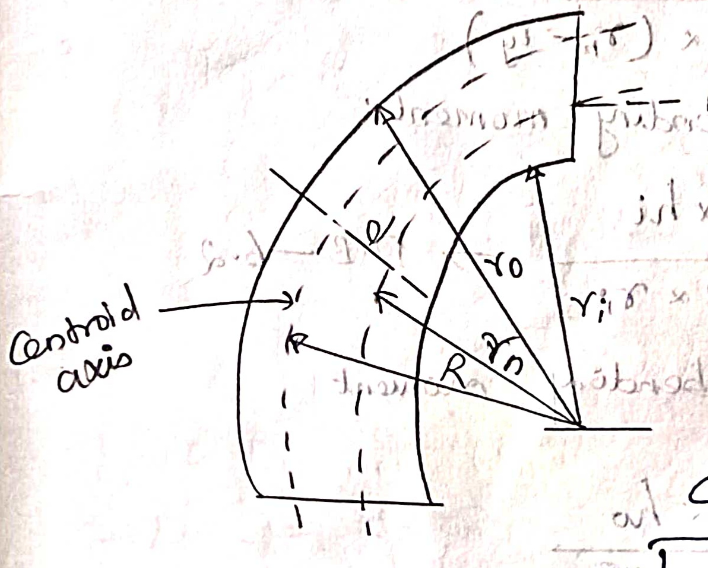
$$\frac{1}{n} = 0.422$$

$$\boxed{n = 2.37}$$

Curved Beams



(1) Axis of curved beams: [D.B - 6.2]



r_i = inner radius

r_o = outer radius

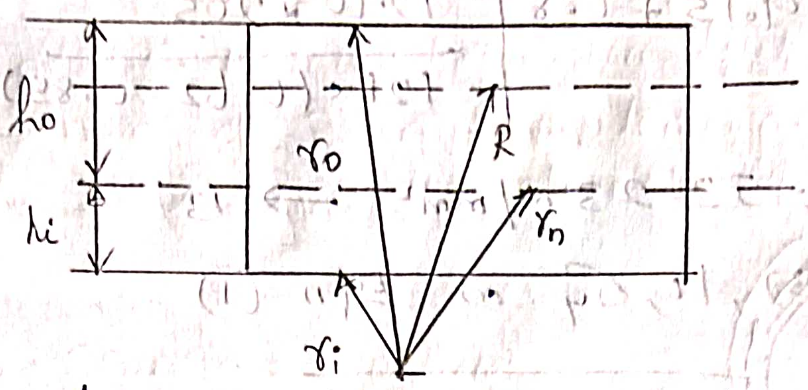
r_n = Neutral radius

R = Centroid axis

e = Distance bet the Centroid axis and Neutral axis

$$\boxed{e = (R - r_n)}$$

Cross section of beam



ho = difference bet outer radius and Neutral axis

$$h_o = r_o - r_n$$

hi = difference bet Neutral and Inner radius

$$h_i = r_n - r_i$$

Formula:

$$r_o = R + d/2$$

$$r_i = R - d/2$$

$$h_i = r_n - r_i$$

$$h_o = r_o - r_n$$

$$e = [R - r_n]$$

Due to bending the Curved beam P.g - 6.2

$$\sigma_b = \frac{M_b \times y}{a \times e \times (r_n - r_y)}$$

Inner radius of bending moment:

$$\sigma_{bi} = \frac{M_b \times h_i}{a \times e \times r_i}$$ → D.B - 6.2

Outer radius of bending moment

$$\sigma_{bo} = \frac{M_b \times h_o}{a \times e \times r_o}$$

Bending Stress for outer fibre.

$$\sigma_b = \frac{M_b \cdot h_o}{a \times e \times r_o} \rightarrow 6.2$$

Bending stress for curved beam

$$\sigma_b = \frac{M_b \times y}{a \cdot e \cdot (r_n - y)} \rightarrow 6.2$$

Bending stress for Inner fibre

$$\sigma_b = \frac{M_b \times h_i}{a \times e \times r_i} \rightarrow 6.2$$

max Bending stress

$$\sigma_{max} = \sigma_b + \sigma_d$$

where $\sigma_b = \frac{M_b \times h_o}{a \times e \times r_o}$ (or) $\frac{M_b \times h_i}{a \times e \times r}$

1) M_b = load x distance (or) length

2) $h_i = r_n - r_i$ $\quad r_n$ = neutral axis
$\quad r_i$ = Inner radius

$$r_n = \frac{(\sqrt{r_o} + \sqrt{r_i})^2}{4}$$ [DB pg - 6.3 (Depend on shape)]

$r_o = R + d/2$

$r_i = R - d/2$

3) Area of section

$a = \pi/4 \, d^2$ (Circular)

$a = \frac{1}{2}(b_i + b_o) \times h$ (Trapezium)

$a = l \times h$ (rectangle)

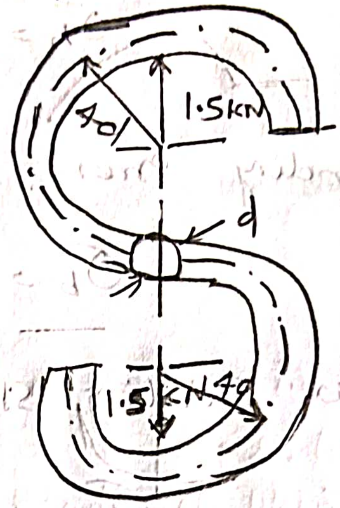
$a = (l \times l)(h \times h)$ [for T section]

4) $e = R - r_n$ [R - DB 6.3 depends on shape]

$\sigma_d = P/A$ [A = depend on shape]

Pb1

1) A link of S-shape made of a round steel bar is shown in fig. Material for the link is steel with a yield stress of 380 mpa in tension. For a factor of safety of 4, find the diameter of the steel bar.



data:

1) $\sigma_y = 380\,mpa = 380\,N/mm^2$
2) F.O.S (or) $n = 4$
3) load $P = 15\,kN = 15\times10^3\,N$
4) Centroid axis $(R) = 4d$

To find Dia (d)

Solution:

$$\sigma_{max} = \sigma_b \times \sigma_d \longrightarrow (1)$$

Bending stress

$$\sigma_b = \frac{M_b \times h_i}{a\times e\times r_i} \quad [D.B: 6.2] \rightarrow (2)$$

M_b = load × [length (or) distance (or) centroid axis]

$= (15\times10^3)\times 4d$

$$\boxed{M_b = (60\times10^3)d} \text{ N-mm}$$

2) $h_i = r_n - r_i$

$$r_n = \frac{(\sqrt{r_o}+\sqrt{r_i})^2}{4} \rightarrow D.B\ 6.3$$

$r_o = R + d/2$

$= 4d + d/2$

$= \frac{8d+d}{2} = \frac{9d}{2}$

$$\boxed{r_o = 4.5d}$$

$r_i = R - d/2$

$= 4d - d/2$

$= \frac{8d-d}{2}$

$$\boxed{r_i = 3.5d}$$

$$r_n = \frac{\left(\sqrt{4.5d} + \sqrt{3.5d}\right)^2}{4} = 3.98 \times d$$

$$h_i = (3.98 \times d) - (3.5d)$$

$$\boxed{h_i = 0.48d}$$

3) Area (circular)

$$a = \pi/4 \, d^2$$

4) $e = R - r_n$

$$= 4d - 3.98d$$

$$\boxed{e = 0.02d}$$

$$\sigma_b = \frac{M_b \times h_i}{a \times e \times r}$$

$$= \frac{(60 \times 10^3) d \times (0.43) d}{\pi/4 \, d^2 \times 0.02d \times 3.5d}$$

$$= \frac{523847.127}{d^2}$$

$$\boxed{\sigma_b = \frac{523.847 \times 10^3}{d^2}} \text{ N/mm}^2$$

Direct Stress:

$$\sigma_d = P/A = \frac{15 \times 10^3}{\frac{\pi}{4} d^2}$$

$$\boxed{\sigma_d = \frac{19.098 \times 10^3}{d^2}} \text{ N/mm}^2$$

$$\sigma_{max} = \frac{\sigma_y}{FOS} = \frac{380}{4} = 95 \text{ N/mm}^2$$

$$\sigma_{max} = \sigma_b + \sigma_d$$

$$95 = \frac{523.84 \times 10^3}{d^2} + \frac{19.098 \times 10^3}{d^2}$$

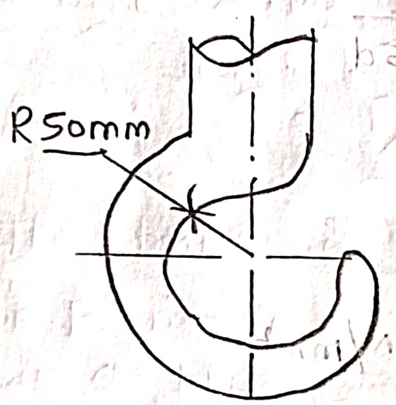
$$95 = \frac{1.0 \times 10^{10}}{d^4}$$

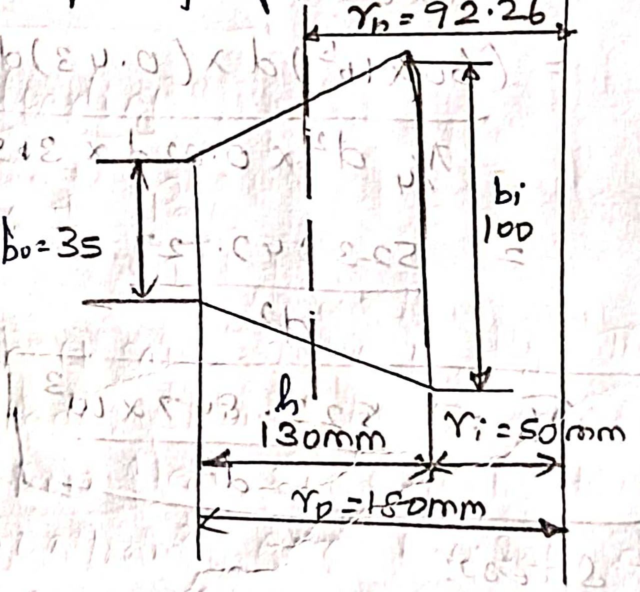
$$d^4 = \frac{1.0 \times 10^{10}}{95}$$

C-clamp

$$\boxed{d = 101.30 mm}$$

Pb2 A crane hook has a section shown in fig which for the purpose of analysis is considered trapezoidal as shown in fig. It is made of plain carbon steel with an yield strength of 380 mpa in tension. Determine the load capacity of the hook for a factor of safety of 3.





data :

1) $\sigma_y = 380 mpa$
2) F.o.s (or) $n = 3$
3) $b_o = 35mm$
4) $b_i = 100mm$
5) $r_i = 50mm$
6) $r_o = 180mm$
7) $r_n = 92.26mm$
8) $h = 130mm$

To find load P

Solution:

$$\sigma_{max} = \sigma_b + \sigma_d \rightarrow (1)$$

$$\sigma_b = \frac{M_b \times h_i}{a \times e \times r_i} \rightarrow (2)$$

(1) $M_b = P \times R$

$$R = r_i + \frac{h(b_i + 2b_o)}{3(b_i + b_o)} \rightarrow \text{P.g } 6.3$$

$$= 50 + \frac{130(100 + 2 \times 35)}{3(100 + 35)}$$

$$\boxed{R = 104.6}$$

$$\boxed{M_b = 104.6\,P \text{ N-mm}}$$

2) $h_i = r_n - r_i$

$$= 92.26 - 50$$

$$\boxed{h_i = 42.26}$$

3) Area $a = \frac{1}{2}(b_i + b_o) \times h$

$$= \frac{1}{2}(100 + 35) \times 130$$

$$\boxed{a = 8775 \text{ mm}^2}$$

4) $e = R - r_n$

$$= 104.6 - 92.26$$

$$\boxed{e = 12.34 \text{ mm}}$$

$$\sigma_b = \frac{104.6 \times 42.26}{8775 \times 12.34 \times 50}$$

$$\boxed{\sigma_b = 8.16 \times 10^{-4} \text{ N/mm}^2}$$

$$\sigma_d = \frac{P}{A} = \frac{P}{8775}$$

$$\sigma_{max} = \sigma_b \propto \frac{P}{8225}$$

$$= 8.16 \times 10^{-4} \propto \frac{P}{8225}$$

$$\boxed{\sigma_{max} = 9.29 \times 10^{-8} (P) \ N/mm^2}$$

$$\sigma_{max} = \frac{\sigma_y}{FOS}$$

$$= \frac{380}{3}$$

$$\boxed{\sigma_{max} = 126.67 \ N/mm^2}$$

$$126.67 = 9.29 \times 10^{-8} P$$

$$\boxed{P = 13.63 \times 10^{8} \ N}$$

Eccentric Axial loading

$$\sigma = P/A \pm \frac{Pey}{I}$$

1

Pb1 The frame of hacksaw shown in fig. The initial tension P in the blade should be 300N. The frame is made of plain carbon steel 30C8 with tensile yield strength of 400N/mm² and F.O.S is 2.5. The cross section of frame is rectangular with a ratio of depth of width as 3 as shown in fig. Det the dimension of the cross section.

data:

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

$P = 300N$

Plain carbon steel 30C8

F.O.S = 2.5

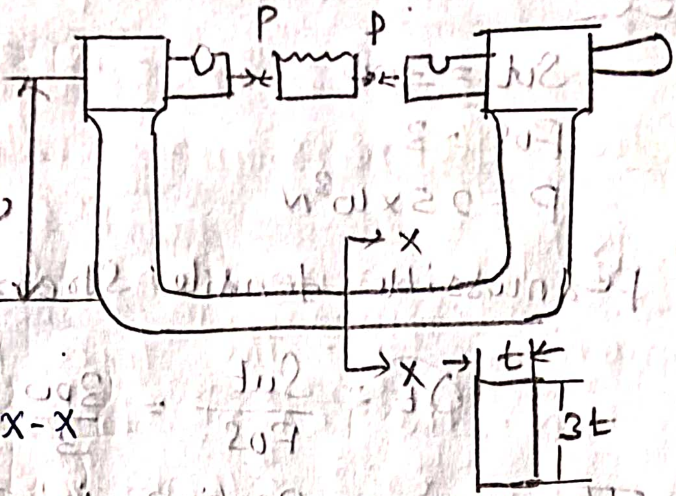
$d = 3t$

Soln:

Permissible Stress

$\sigma_t = \frac{S_{yt}}{FOS} = \frac{400}{2.5}$

$\boxed{\sigma_t = 160\ N/mm^2}$



The stresses at section X-X

1) compressive stress
2) bending stress

Tensile stress maximum at the lower fibre

$$\sigma_c = \frac{P}{A} = \frac{300}{t \times 3t} = \frac{100}{t^2}\ N/mm^2$$

$$\sigma_b = \frac{M_b y}{I} = \frac{(300 \times 200) \times 1.5t}{\frac{1}{12}(t)\cdot(3t^3)} = \frac{40,000}{t^3}\ N/mm^2$$

Super imposing the two stresses and equating into permissible stresses.

$$\sigma_b - \sigma_c = \sigma_t$$

$$\frac{40000}{t^3} - \frac{100}{t^2} = 160$$

$$\frac{40000 - 100t}{t^3} = 160$$

$$40000 - 100t = 160t^3$$

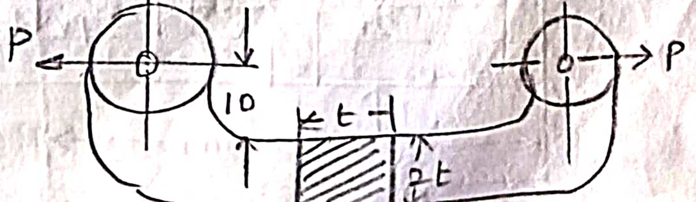
$$160t^3 + 100t - 40000 = 0$$

$$160t^3 + 100t - 40000 = 0$$

solving the eqn;

$\boxed{t = 6.3\ mm}$

2) An offset link subjected to a force of 25 kN shown in fig. It is made of grey cast iron FG 300 and the factor of safety 3. Det the dimension of the cross-section of the link.



Soln:

$S_{ut} = 300$

$FOS = 3$

$P = 25 \times 10^3 N$

permissible tensile stress (σ_t)

$$\sigma_t = \frac{S_{ut}}{FOS} = \frac{300}{3} = 100 \, N/mm^2 \rightarrow (1)$$

The cross section is subjected to direct tensile stress and bending stress. Max stress at the top fibre.

$$\sigma_t = \frac{P}{A} + \frac{M_b y}{I}$$

$$= \frac{25 \times 10^3}{t \times 2t} + \frac{25 \times 10^3 [10+t] \cdot t}{\frac{1}{12} t \cdot (2t)^3} \rightarrow (2)$$

equation (1) & (2)

$$\frac{25 \times 10^3}{t \times 2t} + \frac{25 \times 10^3 (10+t) \cdot t}{\frac{1}{12} t (2t)^3} = 100$$

$$\frac{12500}{t^2} + \frac{37500 (10+t)}{t^3} = 100$$

$$\frac{12500t + 375000 + 37500t}{t^3}$$

$500t + 3750 = t^3$

$t^3 - 500t - 3750$

$$t^3 - 500t - 3750 = 0$$

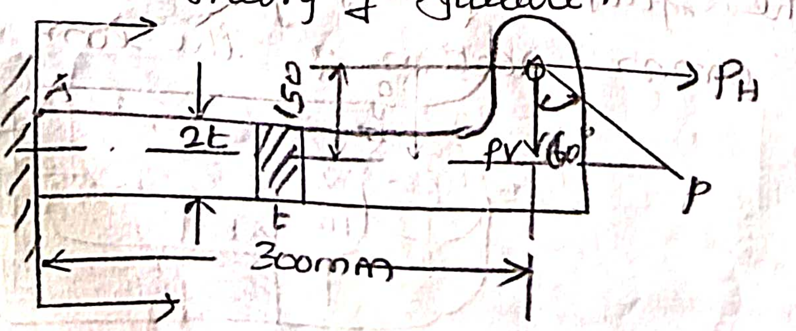
$12500t + 375000 + 37500$

$\frac{50000t + 375000}{100}$

$$\boxed{t = 25.5 \, mm}$$

Theory of failure

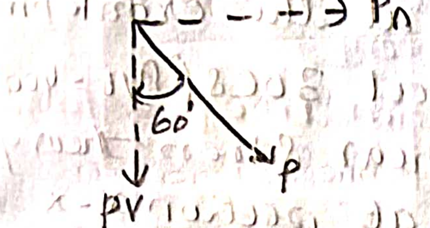
Pb 3 A wall bracket with rectangular cross-section is shown in fig. The depth of the cross section is twice of the width. The force p acting on the bracket at 60 to the vertical is 5KN. The material of the bracket is grey cast iron FG 200 and FOS is 3.5. Det the dimension of the cross-section of the bracket. Assume Max principal stress theory of failure.



Soln:

Maximum stress is maximum at point section A & x section

The force P is resolved horizontal and vertical components.



$P_H = P\sin 60^\circ = 5000\sin 60^\circ = 4330.13\ N$

$P_V = P\cos 60^\circ = 5000\cos 60^\circ = 2500\ N$

Point A subjected to combined and direct tensile stresses

$\frac{2099280 + 2165.07t}{t^3} = 57.14$

Bending moment at section $\alpha-\alpha$

$$M_b = (P_H \times 150) + (P_V \times 300)$$

$2099280 + 2165.07\,t = 57.14\,t^3$

$$= (4330.13 \times 150) + (2500 \times 300)$$

$$M_b = 1399.52 \times 10^3\ N\text{-mm}$$

$57.14t^3 - 2165.07t - 2099280 = 0$

$$\sigma_b = \frac{M_b y}{I} = \frac{1399.52 \times 10^3 \times t}{\frac{1}{12} \times t \times (2t)^3} = \frac{2098.28 \times 10^3}{t^3}$$

The direct tensile stress due to component

$$\sigma_t = \frac{P_h}{A} = \frac{4330.13}{2t^2} = \frac{2165.07}{t^2}\ N/mm^2$$

Vertical component P_v induces shear stress at section $\alpha - \alpha$

$$\sigma_{max} = \sigma_b + \sigma_t$$

$$= \frac{2099.28 \times 10^3}{t^3} + \frac{2165.07}{t^2} \rightarrow (1)$$

The permissible tensile stress

$$\sigma_{max} = \frac{S_{ut}}{FOS} = \frac{200}{3.5} = 57.14\ N/mm^2 \rightarrow (2)$$

Eqn (1) & (2)

$$\frac{2099.28 \times 10^3}{t^3} + \frac{2165.07}{t^2} = 57.14$$

$$t^3 - 37.89t - 36739.24 = 0$$

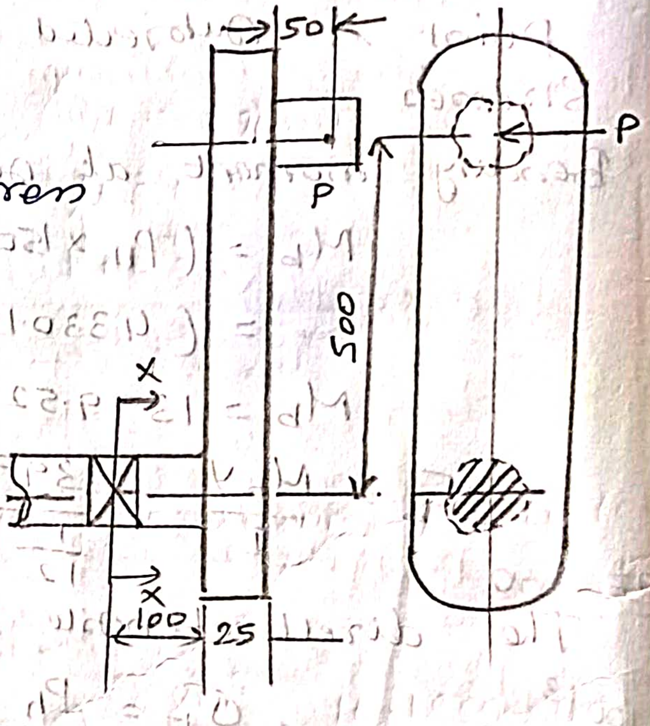
Solving the cubic eqn

$t = 33.65 \simeq 35\,mm$

The dimensions of the $\boxed{C/S = 35 \times 70\,mm}$

Principal Stresses: for various loading combinations

Pb 4 The dimensions of an overhang crank are given in fig. The force P acting at the crank pin is 1kN. The crank is made of steel 30C8 ($S_{yt} = 400$) and F.O.S is 2. using maximum shear stress theory of failure, det the diameter 'd' at section x-x



Soln:

According max shear stress theory

$S_{sy} = 0.5\, S_{yt}$

$= 0.5 \times 400$

$S_{sy} = 200\ N/mm^2$

Permissible shear stress

$\tau_{max} = \frac{S_{sy}}{FOS} = \frac{200}{2}$

$\tau_{max} = 100\ N/mm^2$

The section of the crank pin at section x-x is subjected to combined bending & torsional moments

M_b = load x distance from P to section x x (horizontal)

$= P \times L$

$= 1000 \times (50 + 25 + 100)$

$M_b = 175 \times 10^3\ N\text{-}mm$

M_t = load x vertical distance

$= 1000 \times 500$

$M_t = 500 \times 10^3\ N\text{-}mm$

$$\sigma_x = \sigma_b = \frac{M_b y}{I}$$ bending eqn

$$\frac{M_b}{I} = \frac{\sigma_b}{y} = \frac{E}{R}$$

$$\sigma_b = \frac{M_b y}{I}$$

$$= \frac{175 \times 10^3 \times (d/2)}{(\pi d^4/64)}$$

$$\sigma_x = \left[\frac{1782.54 \times 10^3}{d^3}\right] N/mm^2$$

$$\sigma_y = 0$$

$$\tau = \frac{M_t r}{J}$$ From Torsion eqn

$$\frac{M_t}{J} = \frac{C\theta}{l} = \frac{\tau}{R}$$

$$\boxed{\tau = \frac{M_t \cdot r}{J}}$$

$$= \frac{500 \times 10^3 \times (d/2)}{(\pi d^4/32)}$$

$$\tau = \left(\frac{2546.48 \times 10^3}{d^3}\right) N/mm^2$$

$\therefore$ maximum shear stress is

$$\tau_{max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau)^2}$$

$$= \sqrt{\left(\frac{1782.54 \times 10^3}{2 d^3}\right) + \left(\frac{2546.48 \times 10^3}{d^3}\right)^2}$$

$$\tau_{max} = \frac{2697.95 \times 10^3}{d^3}$$

$$100 = \frac{2697.95 \times 10^3}{d^3}$$

$$\boxed{d = 29.99 \simeq 30 mm}$$

Theory of failure

Pb 1 A bolt is subjected to an axial pull of 8 kN and a transverse shear force of 3 kN. Det the diameter of the bolt required based on

(i) maximum principal stress theory

2) Maximum shear stress theory

Take elastic limit in simple tension is equal to 270 mpa and Poisson's ratio = 0.3

Take fos = 3

data

$P_a = 8\,kN = 8\times10^3\,N$ $\qquad S_{yt} = 270\,mpa$

$P_s = 3\,kN = 3\times10^3\,N$ $\qquad \mu = 0.3$

$\qquad\qquad FOS = 3$

$5092.95 + \sqrt{5092.95 + 1909.}$

find

$\sigma_{max}, \sigma_{min}$

soln:

1) Diameter of the bolt using maximum principal stress theory.

$$\sigma_x = \frac{P_a}{A} = \frac{P_a}{\pi/4\, d^2} \Rightarrow \frac{8\times10^3}{\pi/4\, d^2} = \frac{10.1859\times10^3}{d^2}$$

$\sigma_y = 0$

$$\tau_{xy} = \frac{P_s}{\pi/4\, d^2} = \frac{3\times10^3}{\pi/4\, d^2} = \frac{3.8197\times10^3}{d^2}$$

Principal stresses can be calculated as

$$\sigma_{max} = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

$$= \frac{10.1859\times10^3}{2} + \sqrt{\left(\frac{10.1859\times10^3}{2d^2}\right)^2 + \left(\frac{3.8197\times10^3}{2}\right)^2}$$

$$\sigma_{max} = \frac{11459.13}{d^2}\ N/mm^2$$

$$\sigma_{min} = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

$$= \frac{10.1859\times10^3}{2d^2} - \sqrt{\left(\frac{10.1859\times10^3}{2d^2}\right)^2 + \left(\frac{3.8197\times10^3}{d^2}\right)^2}$$

$$\boxed{\sigma_{min} = \frac{-1273.23}{d^2}}$$

Working Stress

$$FOS = \frac{S_{yt}}{\sigma_w} \Rightarrow \sigma_w = \frac{S_{yt}}{FOS}$$

$$\sigma_w = \frac{270}{3} = 90\,MPa$$

∴ Diameter of bolt using max principal Stress theory.

$$\sigma_w = \sigma_{max}$$

$$90 = \frac{11459.13}{d^2}$$

$$\boxed{d = 11.28\,mm}$$

2) Diameter of bolt using max. shear Stress theory

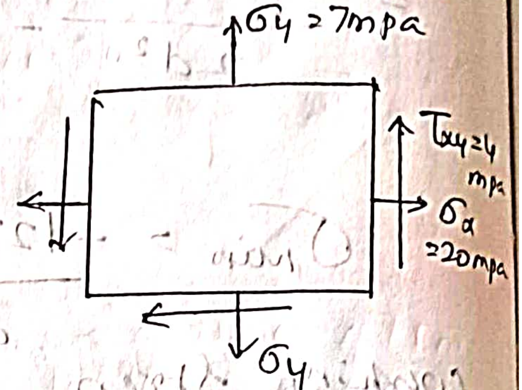
$$\sigma_w = \frac{S_{yt}}{FOS} = \left|\sigma_{max} - \sigma_{min}\right|$$

$$90 = \left|\frac{11459.13}{d^2} - \left(-\frac{1273.23}{d^2}\right)\right|$$

$$\boxed{d = 11.89\,mm}$$

Pb: 11) The stress state in a machine member is given as follows, $\sigma_x = 20$ mpa $\sigma_y = 7$ mpa $\tau_{xy} = 4$ mpa find the principal normal and shear stresses. locate the angle of σ_1 and σ_2 from x-axis.

data:



$\sigma_x = 20 mpa = 20\ N/mm^2$

(1) $\sigma_y = 7 mpa = 7\ N/mm^2$

$\tau_{xy} = 4\ mpa = 4\ N/mm^2$

Soln:

Max principal stress (pg. 7.2)

$$\sigma_1 = \frac{1}{2}\left[(\sigma_x + \sigma_y) + \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$$= \frac{1}{2}\left[(20+7) + \sqrt{(20-7)^2 + 4\times4^2}\right]$$

$$= 21.132\ N/mm^2$$

Min principal stress

$$\sigma_2 = \frac{1}{2}\left[(\sigma_x + \sigma_y) - \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$$= \frac{1}{2}\left[(20+7) - \sqrt{(20-7)^2 + 4\times4^2}\right]$$

$$= 5.87\ N/mm^2$$

Max shear stress

$$\text{Max shear stress } \tau_{max} = \frac{\sigma_1 - \sigma_2}{2}$$

$$= \frac{21.132 - 5.87}{2} = 7.631\ N/mm^2$$

stress may be represented as

$$\tan 2\theta_1 = \frac{2\tau_{xy}}{\sigma_x - \sigma_y} = \frac{2\times4}{20-7} = \frac{8}{13} = 0.615$$

$$2\theta_1 = \tan^{-1}(0.615) = 31.59$$

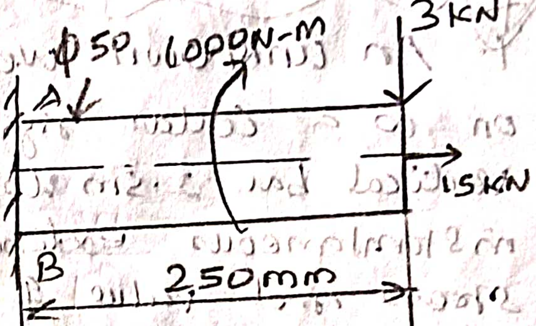
$$\boxed{\theta_1 = 15.8^\circ}$$

θ_2 will be perpendicular to θ_1, angle of σ_2 with x axis

$$\theta_2 = 90 + \theta_1 = 90 + 15.8$$

$$\boxed{\theta_2 = 105.8^\circ}$$ → Principal Stresses for various load combination

Pb: A Shaft as shown in fig 1.41 is subjected to a bending load of 3KN, pure torque of 1000 N-m and an axial pulling force of 15KN. Calculate the stresses A and B.



Soln:

min principal stress of A (Pg. 7.2) $\sigma_y = 0$

$$\sigma_{A\min} = \frac{1}{2}\left[(\sigma_x + \sigma_y) - \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$$= \frac{1}{2}\left(68.76 - \sqrt{68.76^2 + 4 \times 40.74^2}\right)$$

$$= -18.93\ N/mm^2$$

Max shear stress of A (Pg. 7.2)

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

$$= \frac{1}{2}\sqrt{68.76^2 + 4 \times 40.74^2}$$

$$= 53.31\ N/mm^2$$

Min principal stress at B (Same formula)

$$\sigma_{B\min} = \frac{1}{2}\left[(\sigma_x + \sigma_y) - \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$$= \frac{1}{2}\left[(-53.48) - \sqrt{(-53.48)^2 + 4 \times 40.74^2}\right]$$

$$\boxed{\sigma_{B\min} = -75.47\ N/mm^2}$$

Max Shear Stress at B (P8 : P.2)

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

$$= \frac{1}{2}\sqrt{(-53.48)^2 + 4 \times 40.74^2}$$

$$= 48.73\ N/mm^2$$

Impact and shock loading:

1. An unknown weight falls from a distance of 15mm on to a collar rigidly attached to the lower end of a vertical bar 2.5m long and 500mm² cross section. The max instantaneous extension is 2mm. Find the corresponding stress and value of weight falling $E = 2\times10^5\ N/mm^2$

data: $h = 15mm$, $l = 2.5m = 2500mm$ $A = 500mm^2$

$\delta l = 2mm$; $E = 2\times10^5\ N/mm^2$

Soln:

Strain (e) $= \delta l / l = 2/2500 = 0.0008$

Strain (e) = Stress (σ) / Young's modulus (E)

$\sigma = E \times e = 2\times10^5 \times 0.0008 = 160\ N/mm^2$

Equating strain energy and the loss in potential energy.

$$\frac{1}{2} P.(\delta l) = W(h + \delta l)$$

P = Static load = Stress x Area

$= 160 \times 500$

$= 80000N$

$$\therefore \frac{1}{2} \times 80000 \times 2 = W(15+2)$$

$$\boxed{W = 4705.88\ N}$$

Theory of failure

Pb (2) A bolt is subjected to a tensile load of 25 kN and a shear load of 10 kN. Det the dia of the bolt according to 7.2 Theory of failure.

a) max principal stress theory
b) max principal strain theory
c) max shear stress theory

Assume FOS = 2.5, yield point stress in simple tension = 300 N/mm², poisson's ratio = 0.25

data

$P = 25\,kN = 25 \times 10^3\,N$ FOS = 2.5 $\nu = 0.25$

$F = 10\,kN = 10 \times 10^3\,N$ $\sigma_y = 300\,N/mm^2$

Soln: Tensile load

$$\sigma_s = \frac{P}{A} = \frac{25 \times 10^3}{\pi/4\, d^2} = \frac{31.83 \times 10^3}{d^2}$$

Stress due to shear load

$$\tau = F/A = \frac{10 \times 10^3}{\pi/4\, d^2} = \frac{12.732 \times 10^3}{d^2}$$

1. Acc and max principal stress — Pg 7.2

$$\sigma_1 = \frac{1}{2}\left[(\sigma_x + \sigma_y) + \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$\sigma_y = 0$

$$\tau_{xy} = \tau = \frac{12.732 \times 10^3}{d^2}$$

$$\sigma_x = \frac{31.83 \times 10^3}{d^2}$$

Sub in eqn

$$\sigma_1 = \frac{1}{2}\left[\frac{31.83 \times 10^3}{d^2} + \sqrt{\left(\frac{31.83 \times 10^3}{d^2}\right)^2 + 4\left(\frac{12.732 \times 10^3}{d^2}\right)^2}\right]$$

$$= \frac{36.28 \times 10^3}{d^2}\ N/mm^2$$

Min principal stress

$$\sigma_2 = \frac{1}{2}\left[(\sigma_x+\sigma_y) - \sqrt{(\sigma_x-\sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$$= \frac{1}{2}\left[\frac{31.83 \times 10^3}{d^2} - \frac{40.73 \times 10^3}{d^2}\right] = \frac{-4.45 \times 10^3}{d^2} \; N/mm^2$$

a) max principal stress theory

$$\sigma_2 \text{ (or) } \sigma_3 \text{ (or) } (\sigma_1) = \frac{\sigma_y}{FOS} \qquad \sigma_1 = \frac{\sigma_y}{FOS} =$$

$$\frac{36.28 \times 10^3}{d^2} = \frac{300}{2.5}$$

$$d = 17.4 \simeq 18\,mm$$

b) max principal strain theory

$$\sigma_1 - \nu(\sigma_2 - \sigma_3) = \frac{\sigma_y}{FOS} \qquad \sigma_3 = 0$$

$$\frac{36.28 \times 10^3}{d^2} = 0.25 \times \frac{+4.45 \times 10^3}{d^2} = \frac{300}{2.5}$$

$$d = 17.65 \simeq 18\,mm$$

c) max shear stress theory

$$\sigma_1 - \sigma_2 = \frac{\sigma_y}{FOS}$$

$$\frac{36.28 \times 10^3}{d^2} - \frac{-4.45 \times 10^3}{d^2} = \frac{300}{2.5}$$

$$\frac{40730}{d^2} = \frac{300}{2.5}$$

$$\boxed{d = 18.42 \simeq 19\,mm}$$

(1) A rectangular plate 60mm x 10mm with a hole 12mm dia is as shown below, to a tensile load of 12 kN find max stress induced

7.00

data:

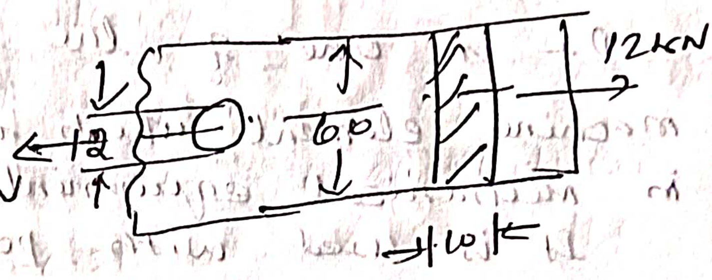
$P = 12\,kN = 12\times10^3\,N$

Plate width $= W = 60mm$

thk $h = 10mm$

Dia of hole $= a = 12mm$

$a/w = 12/60 = 0.2$



find Max stress induced.

Soln: Nominal stress $\sigma_0 = \dfrac{P}{(w-a)h}$

$= \dfrac{12\times10^3}{(60-12)\times10} = 25\,N/mm^2$

$\dfrac{\text{Maximum stress}}{\text{Nominal stress}}$ $\dfrac{\sigma_{max}}{\sigma_0} = K_t \longrightarrow 7.10$

$\sigma_{max} = \sigma_0 \cdot K_t$

for $a/w = 0.2 \longrightarrow 7.10$

$\boxed{K_t = 2.5}$

$\sigma_{max} = \sigma_0 \times 2.5$

$= 25\times 2.5$

$\boxed{\sigma_{max} = 62.5\,N/mm^2}$

UNIT- II

Shaft and Couplings:

Shaft is a rotating element which transmit power from one member to another member.

It is one of the most common and basic machine element which are used in a variety of ways in mechanical equipments.

It is used with power transmission elements such as gear, pulley, fly wheel, cranks etc.

Types of Shaft:

1) Line shaft
2) Spindle
3. Stub shaft
4. Counter shaft

a) Transmission shaft
b) machine shaft.

~~1) Design the rectangular key for a shaft of 50mm diameter. The shearing~~

Design of shaft based on strength against static loading

(1) Simple torsional moment
2) Simple bending moment
3) Combined torsional and bending moment
4) Combined axial load, torsional moment and bending moment.

a) Shaft subjected to simple torsional moments:

Shear strength: $\tau = \dfrac{16 M_t}{\pi d^3}$ → For solid shaft.

$= \dfrac{16 M_t d_o}{\pi (d_o^4 - d_i^4)}$ – For hollow shaft.

M_t : Torsional moment or torque

d = Dia of the solid shaft.

d_o = outside dia of the hollow shaft.

d_i = Inside dia of the hollow shaft.

$[\tau]$ = Allowable shear strength $(0.5 \text{ to } 0.577)\sigma_y$

2) shaft subjected to simple bending moment:

$$\sigma_b = \frac{32 M_b}{\pi d^3} \longrightarrow \text{For solid shaft}$$

$$= \frac{32 M_b d_o}{\pi(d_o^4 - d_i^4)} \longrightarrow \text{for hollow shaft.}$$

M_b = bending moment.

iii) Shaft subjected to combined torsional moment and bending moment:

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2}$$

$$= \frac{1}{2}\sqrt{\left(\frac{32 M_b}{\pi d^3}\right)^2 + 4\left(\frac{16 M_t}{\pi d^3}\right)^2}$$

$$= \frac{16}{\pi d^3}\sqrt{M_b^2 + M_t^2} \longrightarrow \text{Solid shaft.}$$

$$= \frac{16}{\pi d_o^3\left[1 - \left(\frac{d_i}{d_o}\right)^4\right]}\sqrt{M_b^2 + M_t^2} \longrightarrow \text{hollow shaft.}$$

$$\frac{\pi}{16} \times \tau_{max} \times d^3 = \sqrt{M_b^2 + M_t^2} \longrightarrow \text{Solid shaft relation}$$

$\sqrt{M_b^2 + M_t^2} \longrightarrow$ Equivalent twisting moment M_{te}

$$\boxed{M_{te} = \sqrt{M_b^2 + M_t^2}}$$

iv) Shaft subjected to fluctuating load

$$\tau_{max} = \frac{16}{\pi d^3} \sqrt{(K_b M_b)^2 + (K_t M_t)^2} \rightarrow \text{Solid shaft.}$$

$$= \frac{16}{\pi d_o^3 \left[1 - \left(\frac{d_i}{d_o}\right)^4\right]} \sqrt{(K_b M_b)^2 + (K_t M_t)^2} - \text{Hollow shaft.}$$

K_b and $K_t \rightarrow$ 7.21

V) Shaft subjected to combined axial load bending moment and torsional moment:

$$\sigma = \frac{4 P \alpha}{\pi d^2} \rightarrow \text{Solid shaft.}$$

$$\sigma = \frac{4 P \alpha}{\pi d_o^2 \left(1 - \left(\frac{d_i}{d_o}\right)^2\right)} \rightarrow \text{hollow shaft.}$$

α = column action factor pg: 7.21

= 1 for tensile load.

$$= \frac{1}{1 - 0.0044\left(\frac{l}{r}\right)} \quad \ldots \text{ for } \frac{l}{r} < 115$$

$$= \frac{\sigma_y}{\pi^2 n E}\left(\frac{l}{r}\right)^2 \quad \ldots \text{ for } \frac{l}{r} > 115$$

l - length of shaft.

r = radius of gyration.

σ_y = yield strength

n = const for the type of column condition

= 1 for both end hinged

= 2.25 for fixed end

= 1.6 for both end pinned, guided and partly restrained.

bi-axial stress:

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_t + \sigma_b)^2 + 4\tau^2}$$

Substituting

$$\tau_{max} = \frac{16}{\pi d_o^3\left[1-\left(\frac{d_i}{d_o}\right)^4\right]}\sqrt{\left[K_b M_b + \frac{P\alpha d_o}{8}\left[1+\frac{d_i^2}{d_o^2}\right]\right]^2 + (K_t M_t)^2}$$

Design of Shaft for rigidity:

$$\theta = \frac{M_t l}{JG}$$

M_t = Torque on the shaft

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

G = modulus of rigidity

l = length of shaft.

Angle of twist

$$\theta = \frac{584 M_t l}{G d^4} \rightarrow \text{For solid shaft.}$$

$$= \frac{584 M_t l}{G(d_o^4 - d_i^4)} \rightarrow \text{hollow shaft.}$$

$$\boxed{\theta = \frac{2 M_t \times l}{G d}}$$

Design for lateral rigidity:

(i) Reducing the span length

2) Increasing the number of support

3) Selecting the cross section in which the area moment of inertia large in case of tabular or hollow shaft.

Critical or whirling speed of shaft:

$$\omega_c = \sqrt{k/m} = \sqrt{g/\delta}$$

k - Stiffness of the shaft
m = mass of the pulley
δ = Static deflection of the shaft.

Design of shaft for variable loading:

$$\sigma_{max} = \frac{32 M_{b\,max}}{\pi d^3}$$

$$\sigma_{min} = \frac{32 M_{b\,min}}{\pi d^3}$$

$$M_b = P \times e$$

Mean stress $\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2}$

Amplitude stress $\sigma_a = \dfrac{\sigma_{max} - \sigma_{min}}{2}$

Max Shear Stress $\tau_{max} = \dfrac{16 M_{t\,max}}{\pi d^3}$

$$\tau_{min} = \frac{16 M_{t\,min}}{\pi d^3}$$

$$\tau_m = \frac{\tau_{max} + \tau_{min}}{2}$$

$$\tau_a = \frac{\tau_{max} - \tau_{min}}{2}$$

$$\sigma_{eq} = \sigma_m + K_t \times \frac{\sigma_a \times \sigma_y}{\sigma_{-1}}$$

$$\tau_{eq} = \tau_m + K_t \times \frac{\tau_a \times \tau_y}{\tau_{-1}}$$

$$(\tau_{max})_{eq} = \frac{1}{2}\sqrt{(\sigma_{eq})^2 + 4(\tau_{eq})^2}$$

p.1 A electric generator rotates at 200 rpm and it recieves 300 kW from the driving engine. The armature of the generator is 60cm long and located between bearing 120cm centre to centre. Owing to the combined weight of armature and the magnetic pull the shaft is subjected to 9000kg acting at right angle of the shaft. The ultimate stress for the shaft is 4480 kg/cm^2 and shear stress is 3920 kg/cm^2. Find the dia of shaft for a factory of shaft of 6.

data:

$N = 200$ rpm

$P = 300\,kW = 300 \times 10^3\,W$

Armature length $D = 60\,cm = 600\,mm$

Distance $l = 120\,cm = 1200\,mm$

Wt of armature $= 9000\,kg = 9000 \times 9.81 = 88290\,N$

ultimate stress $\sigma_u = 4480\,kg/cm^2$

$= \frac{4480 \times 9.81}{100} = 439.5\,N/mm^2$

$\tau = 3920\,kg/cm^2 = \frac{3920 \times 9.81}{100} = 384.5\,N/mm^2$

F.O.S $= 6$

Soln:

Twisting moment $M_t = \frac{P \times 60}{2\pi N} = \frac{300 \times 10^3 \times 60}{2\pi \times 200}$

$= 14323.95\,N\text{-}m = 14323.95 \times 10^3\,N\text{-}mm$

$\sigma_b = \frac{\sigma_u}{F.O.S} = \frac{439.5}{6} = 73.25\,N/mm^2$

$\tau_{all} = \frac{\tau_u}{F.O.S} = \frac{384.5}{6} = 64.1\,N/mm^2$

$M_b = \frac{W \times l}{4} = \frac{88290 \times 1200}{4} = 26.49 \times 10^6\,N\text{-}mm$

Equivalent twisting moment

$M_{te} = \sqrt{M_b^2 + M_t^2} = \sqrt{(26.49 \times 10^6)^2 + (14232.9 \times 10^3)^2}$

$= 30.11 \times 10^6\,N\text{-}mm$

Equivalent twisting moment

$$M_{te} = \frac{\pi}{16} \times \tau \times d^3$$

$$30.11 \times 10^6 = \frac{\pi}{16} \times 64.1 \times d^3$$

$$d = 133.7 \text{ mm} \simeq 140 \text{ mm}$$

Std dia of shaft = d = 140 mm

Max Normal stress theory ; equivalent bending moment

$$M_{be} = \frac{1}{2}\left[M_b + \sqrt{M_b^2 + M_t^2}\right]$$

$$= \frac{1}{2}\left[26.49 \times 10^6 + \sqrt{(26.49 \times 10^6)^2 + (30.11 \times 10^6)^2}\right]$$

$$= 33.3 \times 10^6 \text{ N-mm}$$

$$M_{be} = \pi/32 \times \sigma_b \times d^3$$

$$33.3 \times 10^6 = \pi/32 \times 73.25 \times d^3$$

$$\boxed{d = 166.68 \text{ mm}}$$

Taking Large of two values of dia d = 166.68 mm

Std dia of shaft d = 120 mm

pb2 : A transmission shaft is supported on two bearings 450 mm apart. Two pulley C and D are located on the shaft at dia of 100 mm and 300 mm, respectively to the right of the left hand bearing. Power is transmitted from pulley C to D. The dia and weight of pulley C are 200 mm and 600 N and those of pulley D are 300 mm and 750 N respectively. Ratio of belt tension is 2 for both pulley. Power to be transmitted by the shaft is 25 kW at 300 rpm. The drive C is vertically downward which from D the drive is upward at an angle of 45° to the horizontal. The shaft is made of C45 steel. $K_b = 1.5$ $K_t = 1.2$ design the shaft.

dia of pulley C ; $D_C = 200$ mm

$D = D_D = 300$ mm

belt wt of pulley C, $W_C = 600$ N

$D = W_D = 750$ N

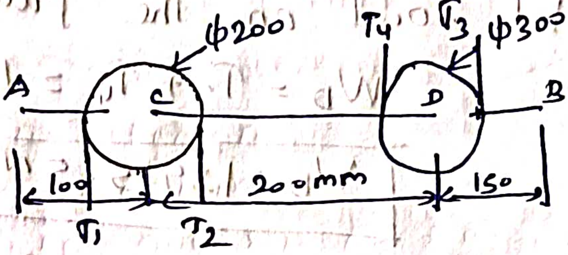
Ratio of belt tension $= \frac{T_1}{T_2} = \frac{T_3}{T_4} = 2$

$P = 25$ kW $= 25 \times 10^6$ W

$N = 300$ rpm

$K_b = 1.5 \quad K_t = 1.2$ Shaft mat: C45 steel

Soln:



$$M_t = \frac{P \times 60}{2\pi N} = \frac{25 \times 10^6 \times 60}{250}$$

$M_t = 795774.72$ N-mm

find total load acting pulley C and D

Force acting on pulley C

T_1 and T_2 be the tension on tight and slack side of the belt

$$M_t = (T_1 - T_2) R_C = T_1 \left(1 - \frac{T_2}{T_1}\right) R_C$$

$$79577.4 = T_1 \left(1 - \frac{1}{2}\right) \times 100$$

$$T_1 = 15915.49 \text{ N} \qquad \frac{T_1}{T_2} = 2$$

loading acts on pulley C

$$W_C = T_1 + T_2 + W_C \text{ (self)}$$

$$= 15915.49 + 7957.5 + 600$$

$$= 24472.99 \text{ N}$$

This load acts vertically downward.

Force acting on pulley D

T_3 and T_4 be the tension on the tight side and slack side of the belt

$$M_t = (T_3 - T_4) R_D = T_3 \left(1 - \frac{T_4}{T_3}\right) R_D$$

$$\frac{T_3}{T_4} = 2$$

$$795774.72 = T_3 \left(1 - \frac{1}{2}\right) 150$$

$$T_3 = 10610.33 N$$

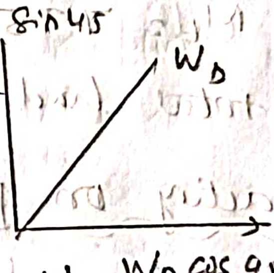
$$T_4 = 5305.17 N$$

Total load on the pulley D

$$W_D = T_3 + T_4 = 10610.33 + 5305.17$$

$$\boxed{W_D = 15915.5 N}$$

This load act at 45° to the horizontal.



Vertical load at D = belt wt of pulley + vertical component of belt tension

$$= 750 + (15915.5 \times \sin 45°)$$

$$W_{DV} = \cancel{10503.96}\ 12002.9 N \text{ (acting upward)}$$

Horizontal component $W_{DH} = W_D \cos 45°$

$$= 15915.5 \times \cos 45 = 11253.9 N$$

(i) considering vertical load only.

R_{AV} and R_{BV}

$$R_{BV} \times 400 = (24471 \times 100) + 12002.9 \times 300 \quad \cancel{+ 10503.96 \times 300 = 2447.99 \times 100}$$

$$\boxed{R_{BV} = \cancel{-1564.2 N}\ 13439 N}$$ acting downward

$$R_{AV} + R_{BV} = 24472.97 + \cancel{10503.96}\ 12002.9 = \cancel{13969.03 N}\ 36473.9 N$$

$$R_{AV} = \cancel{13969.03}\ 36473.9 - R_{BV}$$

$$= 13969.03 - (\cancel{-1564.02}\ 13439) = \cancel{15533.23 N}\ 23034.9 N$$

let us Calculate the bending moment due to vertical loading at various point

BM at $A = 0$

at $C = R_{AV} \times 100 = \overset{23034.9}{\cancel{15533.23}} \times 100 = \overset{2.3\times10^6\ N\text{-}mm}{\cancel{1553323}}\ N\text{-}mm$

$D = R_{BV} \times 150 = \overset{13439}{\cancel{15642}} \times 150 = \overset{2.01\times10^6\ N\text{-}mm}{\cancel{234630}}\ N\text{-}mm$

BM at $B = 0$

ii) considering horizontal force only

R_{AH} and R_{BH} are the reaction at A and B

Taking moment about A

$R_{BH} \times 450 = 11253.96 \times 300$

$R_{BH} = 7502.64\ N$

$R_{AH} + R_{BH} = 11253.96$

$R_{AH} = 11253.96 - 7502.64 = 3751.32\ N$

let us Calculate the bending moment due to horizontal force at various point

BM at $A = 0$

$C = R_{AH} \times 100 = 3751.32 \times 100 = 375132\ N\text{-}mm$

$D = R_{BH} \times 150 = 7502.64 \times 150 = 1125396\ N\text{-}mm$

$B = 0$

Resultant $BM = \sqrt{M_V^2 + M_H^2}$

$C = \sqrt{\overset{(2.3\times10^6)^2}{\cancel{1553323}} + 375132^2} = \overset{2.33\times10^6\ N\text{-}mm}{\cancel{1597998.84}}$

$D = \sqrt{(\underset{2.01\times10^6}{\cancel{234630}})^2 + 1125396^2} = \overset{2.30\times10^6\ N\text{-}mm}{\cancel{1149594.45}}$

$M_b = \overset{2.33\times10^6\ N\text{-}mm}{\cancel{1597998.84\ N\text{-}m}}$

$M_{te} = \sqrt{(K_b M_b)^2 + (K_t M_t)^2}$

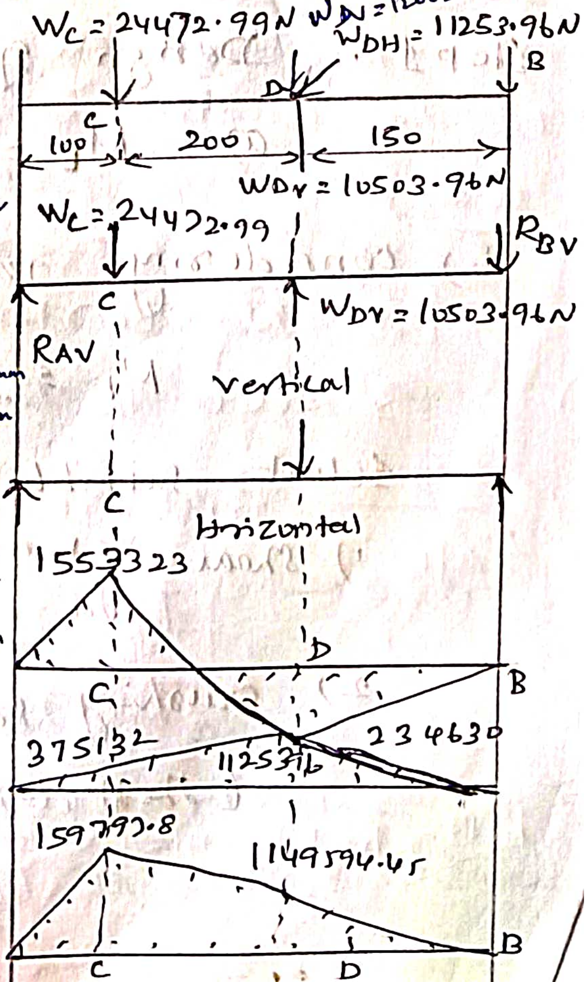
$= \sqrt{(1.5 \times \underset{2.33\times10^6}{\cancel{1597998.84}})^2 + (1.2 \times 79574.72)^2} = 3\times10^6\ N\text{-}mm$

$M_{te} = \frac{\pi}{16} \times \tau \times d^3$

$\overset{3\times10^6}{\cancel{2560183.62}} = \frac{\pi}{16} \times 47.5 \times d^3$

$d = 65.16mm$

$\boxed{d = 70mm}$



Coupling:

Coupling It is used to connect the two shaft is known as coupling.

Types of Coupling

1) Rigid coupling
a) Flange coupling
b) Box (or) Muff (or) sleeve
c) clamp

2) flexible coupling
a) Bushed pin type
b) universal
c) oldham coupling.

Flange coupling:

It is the one type of rigid coupling It is consist of two CI flange key to the shaft end and mould together.

(i) unproducted type flange coupling
(ii) producted type flange coupling
(iii) Marine type flange coupling.

Procedure for flange coupling:

Step 1: Design of shaft

a) $M_t = \frac{P \times 60}{2\pi N}$ — N-mm

b) considering torque transmitted by the shaft.

$$M_t = \frac{\pi}{16} \times \tau_s \times d^3 \quad \text{— 7.135}$$

Select suitable Material

1) Shear stress (key, bolt, shaft) for

$\tau_k = \tau_b = \tau_s = 50\ N/mm^2$

2) Crushing stress for (shaft, key, bolt) $\sigma_{ck} = \sigma_{cb} = \sigma_c = 90\ N/mm^2$

Step 2 Dimension of Flange Coupling

$D = 2d$ $\quad D_1 = 3 \times d$

$L = 1.5d$ $\quad D_2 = 4 \times d$ $\quad$ P.g — 7.134

$t_f = 0.5 \times d$ $\quad t_p = d/4 = 0.25d$

Dimension of Sleeve:

$n = 3$ up to 40mm

$= 4$ up to 100mm

$= 6$ up to 180mm

$D = 2d + 13$ mm

$L = 3.5d$

Step 3 Design of hub.

$$M_t = \frac{\pi}{16} \times \tau_h \times \left(\frac{D^4 - d^4}{D} \right) \rightarrow \text{pg} - 7.135$$

τ_h = Allowable shear strength of flange material.

length of hub is 1.5d

Step 4: Design of key: | Based on dia [DB — 5.16]

width of key (b) :

Thickness of key (h) :

Step 5: Check for shearing for key

$$M_t = l \times b \times \tau_{sk} \times d/2 \rightarrow DB - 7.135$$

$\tau_{sk} = ?$

Case (i)	Case (ii)
Question value $> \tau_{sk}$	Question $< \tau_{sk}$
Design is safe	Design is not safe.

Step 6 check for crushing stress for key.

$$M_t = l \times b/2 \times \sigma_{ck} \times d/2 \rightarrow 7.135$$

Question $> \sigma_{ck}$	Question $< \sigma_{ck}$
Design is safe	Design is not safe
	Change the value of 'h' value in step 4

Step 7: Design of flange.

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

$$\tau_h = ?\ N/mm^2$$

Step 8: Dia of bolt $\longrightarrow$ DB – 7.135

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

Assume $n = 4$ (value not gn)

Step 9: Check for crushing bolt:

$\rightarrow$ 7.135

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

Case (i)

Questsm $> \sigma_{cb}$

Design is safe

Case (ii)

Quessm $< \sigma_{cb}$

Design is not safe.

Change the value of 'n'

Pb1: Det the dimension of flange coupling which connect the motor and a pump shaft. The power to be transmitted a 2 kW at a shaft speed of 960 rpm. Select the suitable material for the parts of the coupling and list the dimension.

data:

1. $P = 2kW = 2\times10^3 W$
2. $N = 960$ rpm
3. Select the suitable material.

C45 / 1.9

CI Grade 30 / 1.5

Assume:

(i) Allowable shear stress for key, bolt, hub shaft

$$\tau_k = \tau_b = \tau_h = \tau_s = 50\ N/mm^2$$

$$\sigma_k = \sigma_b = \sigma_h = \sigma_s = 50\ N/mm^2$$

(2) Allowable crushing shear stress for key bolt, hub, shaft.

$$\tau_{ck} = \tau_{cb} = \tau_{ch} = \tau_{cs} = 90\ N/mm^2$$

$$\sigma_{ck} = \sigma_{cb} = \sigma_{ch} = \sigma_{cs} = 90\ N/mm^2$$

To find

Dimension of flange coupling

Soln:

Step 1 Design of Shaft

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

$$= \frac{(2\times10^3)\times60}{2\pi\times960}$$

$M_t = 19.894$ N-m

$$\boxed{M_t = 19.894\times10^3 \text{ N-mm}}$$

$M_t = \frac{\pi}{16}\times\tau_s\times d^3$ — DB (7.135)

$1984 = \frac{\pi}{16}\times50\times d^3$

$$\boxed{d = 12.65\text{ mm} \simeq 13\text{ mm}}$$ T.25

Step 2 Dimension of flange coupling → 7.134

1. $D = 2d = 2\times13 = 26$ mm
2. $L = 1.5d = 1.5\times13 = 19.5$ mm
3. $D_1 = 3\times d = 3\times13 = 39$ mm
4. $D_2 = 4\times d = 4\times13 = 52$ mm
5. $t_f = 0.5\times d = 0.5\times13 = 6.5$ mm
6. $t_p = d/4 = \frac{13}{4} = 3.2$ mm

Step 3 [Design of hub] — 7.135

$$\boxed{M_t = \frac{\pi}{16}\times\tau_{sh}\times\left(\frac{D^4-d^4}{D}\right)}$$

$$19894 = \frac{\pi}{16}\times\tau_{sh}\times\left(\frac{(26)^4-(13)^4}{D}\right)$$

$$\boxed{\tau_{sh} = 6.15\text{ N/mm}^2}$$

Step 4 : Design of key → 5.16

width of key $(b) = 5mm$
height of key $(h) = 5mm$

Step 5 Check for Shearing key — DB 7.135

$$M_t = L \times b \times \tau_{sk} \times d/2$$

$$19894 = 19.5 \times 5 \times \tau_{sk} \times 13/2$$

$$\boxed{\tau_{sk} = 31.4\ N/mm^2}$$

Case (i)

$$\tau_k > \tau_{sk}$$

$50 > 31.4$ Design is safe.

Step 6 : Check for crushing for key) — 7.135

$$M_t = l \times h/2 \times \sigma_{ck} \times d/2$$

$$19894 = 19.5 \times 5/2 \times \sigma_{ck} \times 13/2$$

$$\boxed{\sigma_{ck} = 62.78\ N/mm^2}$$

$$(\sigma_{ck}) > \sigma_{csk}$$

$90 > 62.78$ → Design is safe.

Step 7 : Design of flange → 7.135

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

$$19894 = \frac{\pi \times 26^2}{2} \times \tau_n \times 6.5$$

$$\boxed{\tau_n = 2.8\ N/mm^2}$$

Step 8 Dia of bolt → 7.135

Assume $n = 4$

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

$$19894 = \pi/4 \times d_b^2 \times 50 \times 4 \times 39/2$$

$d_b = 2.55mm$

Step 4 Check for crushing bolt → 17.135

$M_t = n \times d_b \times t_f \times \sigma_{cb} \times P_1/2$

$19894 = 4 \times 2.55 \times 6.5 \times \sigma_{cb} \times 39/2$

$\boxed{\sigma_{cb} = 15.38\ N/mm^2}$

$\sigma_{cb} > Q_{cb}$: $90 > 15.38$

Design is safe.

Pb 2:

Design a rigid flange coupling to transmit a power of 10kw at 960 rpm bet two coaxial shaft. The shaft is made of alloy steel, flange out of castiron and bolt out of steel. Four bolts are used to couple the flange. The shafts are keyed to the flange hub. The permissible stresses are:

1. Shear stress on shaft = 100mpa
2. Bearing or crushing stress on shaft } = 250mpa
3. Shear stress on key = 100mpa
4. Bearing stress on key = 250 mpa
5. Shearing stress on CI = 200mpa
6. Shear stress on bolts = 100mpa.

data: $P = 10\,KW = 10 \times 10^3\,W$

$N = 960$ rpm

Find: Design of protected type flange coupling.

Soln:

(1) Design of shaft

$M_t = \frac{P \times 60}{2\pi N} = \frac{10 \times 10^3 \times 60}{2\pi \times 960} = 99.4718\ N\text{-}m$

$\boxed{M_t = 99471.8\ N\text{-}mm}$

Allowable Shear Strength = 50 N/mm²
Allowable crushing strength = 90 N/mm²

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

$$99471.8 = \frac{\pi}{16} \times 50 \times d^3$$

$$d = 21.63\,mm \simeq 25\,mm$$

(ii) Dimension of the flange coupling

$D = 2d = 2 \times 25 = 50$

$L = 1.5d = 1.5 \times 25 = 37.5$

$D_1 = 3d = 3 \times 25 = 75\,mm$

$t_f = 0.5d = 0.5 \times 25 = 12.5\,mm$

$D_2 = 4d = 4 \times 25 = 100\,mm$

$t_p = 0.25d = 0.25 \times 25 = 6.25\,mm$

(iii) Design of hub

$$M_t = \frac{\pi}{16} \times \tau_{h} \left(\frac{D^4 - d^4}{D}\right)$$

$$99471.8 = \frac{\pi}{16} \times \tau_{h} \left(\frac{50^4 - 25^4}{50}\right)$$

$$\tau_{sh} = 4.3255\,N/mm^2$$

(iv) Design of key: → 5.16 corresponding $d = 25$

$l = L = 37.5\,mm$

$b = 9\,mm$

$h = 7.5\,mm$

a) Check for shearing

$$M_t = l \times b \times \tau_k \times d/2$$

$$99471.8 = 37.5 \times 9 \times \tau_k \times \frac{25}{2}$$

$$\boxed{\tau_k = 23.57\,N/mm^2}$$

$\tau_k > \tau_{sk}$

Design is safe.

b) Check for Crushing

$$M_t = l \times h/2 \times \sigma_{ck} \times d/2$$

$$99471.8 = 37.5 \times \frac{7.5}{2} \times \sigma_{ck} \times \frac{25}{2}$$

$\sigma_{ck} = 57 \, N/mm^2$ Design is safe.

(v) Design of flange.

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

$$99471.8 = \frac{\pi \times 50^2}{2} \times \tau_h \times 12.5$$

$$\boxed{\tau_h = 2.026 \, N/mm^2}$$

vi) Design of bolts

$$M_t = \frac{\pi}{4} \times d_b^2 \times \tau_h \times n \times \frac{D_1}{2}$$ $n = 3$ for up to [150mm]

$$99471.8 = \pi/4 \times d_b^2 \times 50 \times 3 \times 75/2$$

$$\boxed{d_b = 4.74 mm}$$

Check for crushing

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

$$99471.8 = 3 \times 6 \times 12.5 \times \sigma_{cb} \times \frac{75}{2}$$

$$\boxed{\sigma_{cb} = 11.789 \, N/mm^2}$$ Design is safe.

Bush Coupling:

Step 1 = Design of Shaft

$$M_t = \frac{P \times 60}{2\pi N} = \times 10^3 \, N.mm$$

To find diameter (d)

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

$d = $ — mm

Step 2 : Dimension of flexible coupling

D - 7.108

A - Dia of shaft

B - Outer dia of the flange.

C - Dia of hub

D - pitch circle dia of bolt on flange.

E = length of hub

F - Dia of bolt

G - length of the bush in flange.

H - protective of the bush in flange.

n = no of bolts.

d_b - Dia of bush.

t = clearance.

Step 3 Check for bolt.

Bearing load (W)

$$\boxed{M_t = W \times n \times D/2}$$

$W = \_\_ N$

Direct stress

$$\tau = \frac{W}{\pi/4 \, (F)^2}$$

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

Max bending moment.

$$M_b = W\left(\frac{G}{2} + t\right)$$

$$M_b = \_\_ N\text{-mm}$$

Bending stress

$$\sigma_b = \frac{M_b}{\pi/4 \, (F)^3} = \_\_ N/mm^2$$

Max principal stress

$$\sigma_{max} = \frac{\sigma_b}{2} + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

Max shear stress

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

Cases $\sigma_{max} > \tau_{max}$ Design is safe

$\sigma_{max} < \tau_{max}$ [change F value] Design is not safe 7.108

Step 4 Design of key.

Width of key (w) - - - mm

Thickness of key (h) - — mm

length of key $l_K = E/2$ — mm

Step 5 Check for shearing key : 7.135

$$M_t = l_K \times b \times \tau_K (d/2)$$

$$\tau_K = — \ N/mm^2$$

Case (i) $\tau_K > \tau_{K\,Ans}$ (given)

Design is safe

Case (ii) $(\tau_K)_{Assum} < (\tau_K)_{Ans}$

Design is not safe

Change b value in step: 4 - (5, 6)

Step 6 Check for crushing key)

$$M_t = h/2 \times \sigma_{ck} \times d/2 \times l_K$$

$$\sigma_{ck} — \ N/mm^2$$

Case (i)

$\sigma_{ck} > \sigma_{ck}$ Safe

$\sigma_{ck} < \sigma_{ck}$ Not safe

Change h value in step 4 } 5.16

Step 7: Design of bush

Length of bush $L = G + L + 2/3 \times F$

$$L = — mm$$

Step 8 Bearing (or) crushing stress for bush)

$$\sigma_{bc} = \frac{W}{d_b \times L_{bu}}$$

$$\sigma_{bc} — N/mm^2$$

$\sigma_{bc} > (\sigma_{bc})$ Safe

$\sigma_{bc} < \sigma_{bc}$ - Not safe.

Step 9 Check for hub:

Assume $\tau_h = 15$ (or) $14\ N/mm^2$ → 7.135

$$M_t = \frac{\pi}{16} \tau_h \times \left(\frac{C^4 - A^4}{C}\right)$$

$\tau_h > \tau_h$

$\tau_h < \tau_h$

Step 10 Check for flange

$$M_t = \frac{\pi C^2}{2} \times \tau_h \times G$$

$$\tau_h = - N/mm^2$$

$\tau_h < \tau_h$

Design is not safe

Change the G value in 7.180

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

data $P = 5kW = 5\times10^3$ N

$N = 750$ rpm

Service factor $= 1.2$

Shear stress for shaft, bolt and key $\tau = 50\,N/mm^2$

Shear stress CI $= \tau_c = 15\,N/mm^2$

Bearing stress for bush $\sigma_{bc} = 2.5\,N/mm^2$

crushing stress for key $\sigma_{kc} = 100\,N/mm^2$

soln.

1. Design of shaft.

$$M_{t\,mean} = \frac{P\times60}{2\pi N} = 63.66\,N\text{-}m \text{ (or)}$$

$63661.97\,N\text{-}mm$

service factor is 1.2

$$M_{t\,max} = 1.2\times63661.97 = 76394.37\,N\text{-}mm$$

$$M_{t\,max} = \frac{\pi}{16}\times\tau_s\times d^3$$

$$76394.37 = \frac{\pi}{16}\times50\times d^3$$

$$\boxed{d = 19.8\,mm}$$

2. Dimension of coupling DB — 7.108

$A = 20$ mm

$F = 10$ mm

$n = 3$

$C = 30$ mm

$G = 20$ mm

$t_1 = 2$ mm

$B = 100$

$d_b = 22$ mm

$D = 63$ mm

$H = 12$

3) check for bolt

$$M_{tmax} = W \times n \times D/2$$

$$76394.37 = W \times 3 \times \frac{63}{2}$$

$$W = 808.4 N$$

Direct stress

$$\tau = \frac{W}{\pi/4 (f)^2} = \frac{808.4}{\pi/4 \times (10)^2} = 10.3 N/mm^2$$

Max bending moment M_b

$$= W \left(\frac{G}{2} + t \right)$$

$$= 808.4 \times \left[\frac{20}{2} + 2 \right] = 9700.8 N-mm$$

$$\sigma_b = \frac{M_b}{\frac{\pi}{4} (f)^3} = \frac{9700.8}{\pi/4 (10)^3} = 12.3 N/mm^2$$

Max principal stress $= \frac{\sigma_b}{2} + \frac{1}{2} \sqrt{(\sigma_b)^2 + 4\tau^2}$

$$= \frac{12.3}{2} + \frac{1}{2} \sqrt{(12.3)^2 + 4(10.3)^2} = 18.14 N/mm^2$$

Max shear stress $= \frac{1}{2} \sqrt{(\sigma_b)^2 + 4\tau^2}$

$$= \frac{1}{2} \sqrt{(12.3)^2 + 4(10.3)^2} = 12 N/mm^2$$

Design of key — width and thk $h = 6mm$

$\cancel{b=}$

$$M_{tmax} = l \times \frac{h}{2} \times B \times \frac{A}{2}$$

$$76394.37 = l \times 6/2 \times 100 \times \frac{20}{2}$$

$$l = 25.46 \simeq 26mm$$

Check for shearing

$$M_{tmax} = l \times b \times \tau_K \times d/2$$

$$76394.37 = 26 \times 6 \times \tau_K \times \frac{20}{2}$$

$$\tau_K = 48.7 N/mm^2$$

v) Design of bush:

$$L = G + t - \frac{2}{3} \times F = 20 + 2 - 2/3 \times 10$$

$$L = 15.33 \text{ (or) } 16mm$$

$$\sigma_{bc} = \frac{W}{d_s \times L} = \frac{808.4}{22 \times 16} = 2.3 < 2.5\ N/mm^2$$

Design is Safe;

(vi) Design of hub :

$$M_{t\,max} = \frac{\pi}{16} \times \tau_h \times \left(\frac{C^4 - A^4}{C}\right)$$

$$76394.37 = \frac{\pi}{16} \times \tau_h \left(\frac{30^4 - 20^4}{30}\right)$$

$$\boxed{\tau_h = 17.95\ N/mm^2}$$

More than allowable value is $15\ N/mm^2$

Design is not Safe.

$$C = 2A = 2 \times 20 = 40\ mm$$

$$\tau_h = 6.49\ N/mm^2$$

(vii) Design for Flange :

$$M_{t\,max} = \frac{\pi C^2}{2} \times \tau_h \times h$$

$$76394.37 = \frac{\pi (40)^2}{2} \times \tau_h \times 20$$

$$\tau_h = 1.52\ N/mm^2.$$

Design is safe.

X. **Keys:**

Q1 A 45 mm diameter shaft is made of steel with a yield strength of 400 MPa. A parallel key of size 14 mm wide and 9 mm thick made of steel with yield strength of 340 MPa is to be used. Find the required length of key, if the shaft is loaded to transmit the maximum permissible torque. Use max shear stress theory and assume F.O.S = 2

Data:

$d = 45mm$

$\sigma_{yt} = 400\ N/mm^2$ (shaft)

$W = 14mm$ | Gib and head

$t = 9mm$ | key pg 5.19)

$\sigma_{yt} = 340\ N/mm^2$ (key)

$F.O.S = 2$

find (1) length

Soln:

According to max shear stress theory (shaft)

$$\tau_{max} = \frac{\sigma_{yt}}{2 \times FOS} = \frac{400}{2\times2} = 100\ N/mm^2$$

max shear stress for the key

$$\tau_k = \frac{\sigma_{yt}}{2\times F.OS} = \frac{340}{2\times2} = 85\ N/mm^2$$

Max torque transmitted by the shaft and key

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

$$= \frac{\pi}{16} \times 100 \times 45^3$$

$$\boxed{T = 1.8\times10^6\ N\text{-}mm}$$

Consider failure of key due to shearing

$$T = l w \tau_k \times \frac{d}{2}$$

$$1.8\times10^6 = l \times 14 \times 85 \times \frac{45}{2}$$

$$l = \frac{1.8\times10^6}{26775}$$

$$\boxed{l = 67.2\ mm}$$

Considering crushing

$$T = l w \sigma_{cr} \times d/2$$ max shear stress

$$\left(l \times \frac{t}{2} \times \frac{\sigma_c}{2} \times d/2\right)$$

$$1.8\times10^6 = l \times \frac{9}{2} \times \frac{340}{2} \times \frac{45}{2}$$

$$l = \frac{1.8\times10^6}{17213}$$

$$\boxed{l = 104.6\ mm}$$

Finally taking larger value of length

$l = 104.6 \simeq 105\ mm$

2) It is required to design a square key for fixing a gear on a shaft of 30mm dia. The shaft is transmitting 20kW power at 600rpm to the gear. The key is made of steel 50C4 and the factor of safety is 4. For key material, the yield strength in compression can be assumed to be equal to the yield strength in tension. Det the dimension of the key.

data:

Mtl 50C4 [pg.no - 1.12)

Take $S_{yt} = 460\ N/mm^2$

$P = 20\ kW = 20 \times 10^3 W$

$N = 600\ rpm$

$F.O.S = 4$

$\boxed{S_{yt} = S_{yc}}$

find 1) dimension of key.

Soln:

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

$$20 \times 10^3 = \frac{2\pi\, 600 \times T}{60}$$

$$\boxed{T = 318.309 \times 10^3\ N\text{-}mm}$$

Allowable tensile and shear stress for key material.

$$\sigma_t = \frac{S_{yt}}{FOS} = \frac{460}{4} = 115\ N/mm^2$$

$$\tau = \frac{0.5 S_{yt}}{F.OS} = 57.5\ N/mm^2$$

Square key

$$W = h = d/4 = \frac{30}{4} = 7.5mm$$
$$= 8mm$$

$$l = 1.5d = 1.5 \times 30 = 45mm$$

a) consider shear stress

$$T_{max} = w\, l \times \frac{d}{2} \times \tau$$

$$318.309 \times 10^3 = 8 \times l \times \frac{30}{2} \times 57.5$$

$$\boxed{l = 46.13 \simeq 47\ mm}$$

b) consider crushing stress

$$T_{max} = \frac{h}{2} \times l \times \frac{d}{2} \times \sigma$$

$$318.309 \times 10^3 = \frac{8}{2} \times l \times \frac{30}{2} \times 115$$

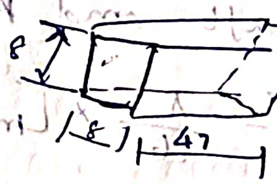
$$\boxed{l = 47\ mm}$$

Larger value is suitable

$l = 47\ mm.$

Dimension of key

$8 \times 8 \times 47\ mm$



26: Q. Design a rectangular key for the following application A shaft 65mm dia transmit power at max shear stress of 67mpa. The shear stress in the key should not exceed 75% of the stress developed in the shaft. The key should be at least 2.5 times strong in crushing compared to shear failure of the key.

<u>data:</u>

$d = 65mm$

$\tau_s = 67 mpa$

$\tau_K = 0.75 \times \tau_s = 50.75 mpa$

$(\sigma_c)_{key} = 2.5 \times \tau_K = 50.25 = 125.65 mpa$

<u>find</u> <u>Rectangular key</u>

pg.NO = 5.16 dimension of parallel keys

$d = 65mm$ $W = 18mm$ $h = 11mm$

$$T = \frac{\pi}{16} d^3 \tau_s = \frac{\pi}{16} \times 65^3 \times 67$$

$$T = 361.28 \times 10^6 \ N\text{-}mm$$

considering shear failure of key

$$T = w \times l \times \frac{d}{2} \times \tau_{key}$$

$$3261.128 \times 10^6 = 18 \times l \times \frac{65}{2} \times 50.25$$

$$l = 122.9 \simeq 123 mm$$

considering crushing failure of key

$$T = l \times \frac{h}{2} \times \frac{d}{2} \times \sigma_c)_{key}$$

$$361.128 \times 10^6 = l \times \frac{H}{2} \times \frac{65}{2} \times 125.65$$

$$l = 160.88 \simeq 161 mm$$

Selecting value is 161mm (comparing two values)

Muff coupling [Sleeve (or) Box]

1) Step 1 (Design of Shaft)

$$M_t = \frac{P \times 60}{2\pi N} \qquad M_t = \_\_ \text{ N-m}$$
$$\_\_ \times 10^3 \text{ N-mm}$$

2) To find diameter

$$M_t = \frac{\pi}{16} \times \tau_s \times d^3 \quad — 7.135$$

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

3) Check for shear stress for muff coupling — 7.133

$$M_t = \frac{\pi}{16} \times \tau_m \times \left(\frac{D^4 - d^4}{D} \right)$$

$$D = 2d + 13 = \_\_ \text{ mm}$$

$$L = 3.5 \times d = \_\_ \text{ mm}$$

$$\tau_m = \_\_ \text{ N/mm}^2$$

Case (i) τ_m (Que) > τ_m (Ans) → Safe

Case (ii) τ_m (Que) < τ_m (Ans) → Not safe.

4) Design of key → 5.16

$b = \_\_$ mm $\quad h = \_\_$ mm

length of key $(l_k) = L/2$

5) Check for shear stress for key

$$M_t = l \times b \times \tau_k \times d/2 \quad — 7.135$$

$$\tau_k = \_\_ \text{ N/mm}^2$$

Case (i) τ_k (Que) > τ_k (Ans) → Safe

(ii) τ_k (Que) < τ_k (Ans) — Not safe (Change the check 'b' 5.16)

6) Check for crushing stress for key

$$M_t = l \times h/2 \times \sigma_{cr} \times d/2 \quad \rightarrow 7.135$$

Case ii) (σ_{ck}) Ques $\geq$ (σ_{ck}) Ans $\rightarrow$ Safe

(σ_{ck}) Que $<$ (σ_{ck}) Ans $\rightarrow$ not safe.

Change the value of 'h' in [pg. no: 5.166]

Step 6 Design of clamping bolt (d_b)

[only for split muff coupling not for muff coupling]

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

Allowable tensile stress for bolts.

Assume $\mu = 0.3$ $n = 4$

$\sigma_t = 70\,N/mm^2$ [$d_b = \_\_ \, mm$]

Pb 1 Design of muff coupling (or) sleeve coupling for the shaft to transmit 35 kW at 350 rpm. The safe shear stress for the shaft is 50 N/mm² and its is 15 N/mm² for the CI muff. The allowable shear and crushing stress for key material are 42 N/mm² and 120 N/mm² respectively.

data:

$P = 35\,kW = 35 \times 10^3\,W$

$N = 350\,rpm$

$\tau_s = 50\,N/mm^2$

$\tau_m = 15\,N/mm^2$

Shear stress for key $(\tau_k) = 42\,N/mm^2$

Crushing stress for key $(\sigma_{ck}) = 120\,N/mm^2$

To find

Design of Muff Coupling.

Soln: Step 1 Design of Shaft

$$M_t = \frac{P \times 60}{2\pi N} = \frac{35 \times 10^3 \times 60}{2\pi \times 350}$$

$M_t = 954.92$ N-m

$M_t = 954.92 \times 10^3$ N-mm

To find Dia.

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

$$954.92 \times 10^3 = \frac{\pi}{16} \times 50 \times d^3$$

$$\boxed{d = 45.98 \text{ mm}}$$

Step 2 (Check for shear stress Muff) - 7.133

$$M_t = \frac{\pi}{16} \times \tau_m \left(\frac{D^4 - d^4}{D} \right)$$

To find D - 7.133

$D = 2d + 13 = 2 \times 46 + 3 = 105$ mm

$$954.92 \times 10^3 = \frac{\pi}{16} \times \tau_m \left(\frac{105^4 - 46^4}{105} \right)$$

$\tau_m = 4.36$ N/mm²

Case (i)

$\tau_m > \tau_m \rightarrow$ Design is safe.

Step 3 Design of key $\rightarrow$ 5.16 (Based on diameter)

$D = 46$ mm

$b = 14$ mm $h = 4$ mm

$L = 3.5 \times d \rightarrow$ 7.133

$= 3.5 \times 46$

$$\boxed{L = 80.5 \text{ mm}}$$

Length of key $(l_k) = \frac{L}{2} = \frac{161}{2} = 80.5\,mm$

Step 4 Check for shear stress for key.

$M_t = l_k \times b \times \tau_k \times d/2 \longrightarrow 7.135$

$984.92 \times 10^3 = 80.5 \times 14 \times \tau_k \times \frac{46}{2}$

$\boxed{\tau_k = 36.83\ N/mm^2}$

Conclusion $\tau_k\,(\text{Allow}) > \tau_k\,(\text{Ans})$

$42 > 36.83 \rightarrow$ Design is safe.

Step 5 Check the crushing stress for key

$M_t = l_k \times \frac{h}{2} \times \sigma_{ck} \times d/2$

$984.92 \times 10^3 = 80.5 \times 9/2 \times \sigma_{ck} \times \frac{46}{2}$

$\boxed{\sigma_{ck} = 114.6\ N/mm^2}$

Conclusion

$\sigma_{ck}\,(\text{Allow}) > \sigma_{ck}\,(\text{Ans})$

$120 > 114 \rightarrow$ Design is safe.

Pb 2 A rigid coupling (or) split coupling type is used to connect two shaft transmitting 15 kW at 200 rpm. The shaft, key, bolt are made up C45 steel and the coupling is CI. Design the coupling.

data:

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

$N = 200\,rpm$

Assume: C45 steel.

$\tau_s = 65\ N/mm^2$

$\tau_m = 14\ N/mm^2$

$\tau_k = 80\ N/mm^2$

$\sigma_{ck} = 180\ N/mm^2$

To find: Design of rigid coupling

Sten: Design of Shaft

$$M_t = \frac{P \times 60}{2\pi N} = \frac{15 \times 10^3 \times 60}{2\pi \times 200}$$

$$= 716.19 \text{ N-m}$$

$$\boxed{M_t = 716.19 \times 10^3 \text{ N-mm}}$$

To find d

$$M_t = \frac{\pi}{16} \times \tau_s \times d^3 \longrightarrow 7.135$$

$$716.19 \times 10^3 = \frac{\pi}{16} \times 65 \times d^3$$

$$d = 38.28 \text{ mm} \simeq 38 \text{ mm}$$

Step 2 [Check for shear stress] for muff $\rightarrow$ 7.133

$$M_t = \frac{\pi}{16} \times \tau_m \left(\frac{D^4 - d^4}{D} \right)$$

$$D = 2d + 13 = (2 \times 38) + 13 = 89 \text{ mm}$$

$$L = 3.5 \times d = 3.5 \times 38 = 133 \text{ mm}$$

$$716.19 \times 10^3 = \frac{\pi}{16} \times \tau_m \left(\frac{89^4 - 38^4}{89} \right)$$

$$\tau_m = 5.35 \text{ N/mm}^2$$

Case i)

$$\tau_m (\text{all}) > \tau_m (\text{Ans})$$

$$14 > 5.35 \longrightarrow \text{Design is safe}$$

Step 3 Design of key $\longrightarrow$ 5.16 (based on dia)

$d = 38$

$b = 12$ mm

$h = 8$ mm

$$\text{Length of key } (l_k) = \frac{L}{2} = \frac{133}{2} = 66.5 \text{ mm}$$

Step 4 Check for shear stress for key (7.138)

$M_t = l_k \times b \times \tau_k \times d/2$

$716.19 \times 10^3 = 66.5 \times 12 \times \tau_k \times \frac{38}{2}$

$\tau_k = 47.23 \, N/mm^2$

case (i) $\tau_k(all) > (\tau_k)$ Ans

$80 > 47.23 \longrightarrow$ Design is safe.

Step 5 [Check for Crushing Stress for key]

→ 7.135

$M_t = l_k \times \frac{h}{2} \times \sigma_{ck} \times d/2$

$716.19 \times 10^3 = 66.5 \times 8/2 \times \sigma_{ck} \times \frac{38}{2}$

$\boxed{\sigma_{ck} = 141.705 \, N/mm^2}$

case (i) $\sigma_{ck}(all) > \sigma_{ck}(Ans)$

$180 > 141.70 \rightarrow$ Design is safe

Step 6: Design of bolt (d_b)

Only for split muff coupling not for muff coupling)

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

$716.19 \times 10^3 = \frac{\pi^2}{16} \times 0.3 \times (d_b)^2 \times 70 \times 4 \times 38$

Assume $\mu = 0.3$

$\sigma_t = 70 N/mm^2$

$n = 4$

$\boxed{d_b = 19.07 \, mm}$

UNIT-3

Types Temporary and permanent joint

Temporary joint

1) Nut and bolts
2) knuckle joints
3) Cotter joint

Cotter joint

* Cotter joint are temporary connection used to connect a rod with another rod are some other machine element.
* A Cotter joint transmit axial tension (or) compression joint
* A cotter joint are made up of mild steel.
* It is taper an inside various from 1/40 to $\frac{1}{24}$

Types of cotter joint

1) Sleeve and cotter joint
2) Socket and spigot joint
3) Jib and cotter joint

Design of Socket and Spigot (or) Cotter joint

Step 1 : failure of rod in tension

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

σ_t = tensile stress (gn in que)

d = —— mm.

Step 2 : failure of rod end at cotter hole.

$$P = \left[\frac{\pi}{4} d_1^2 - d_1 t\right] \times \sigma_t$$

Assume $t = 0.4 \times d$

$t =$ ___ mm

$d_1 = +$ ___ mm $d_1 = -$ ___ mm

Take only positive value.

Step 3 [Tension failure of socket end at cotter hole]

$$P = \left(\frac{\pi}{4} d_2^2 - \frac{\pi}{4} d_1^2 - (d_2 - d_1) \times t\right) \times \sigma_t$$

$d_2 = +$ ___ mm $d_2 = -$ mm

Take positive value

Step 4 (Check crushing failure at the end)

$$P = [d_2 - d_1] \cdot t \times \sigma_c$$

$\sigma_c =$ ___ N/mm^2

Case i) σ_c given $> \sigma_c$ Ans
Design is safe

Case ii) σ_c given $< \sigma_c$ Ans
Design is not safe
(Change in d_2 value)

Step 5 (Shear failure of cotter)

$$P = 2bt\tau$$

$b =$ ___ mm

Step 6 (Crushing failure of cotter joint)

$$P = \frac{\pi}{4}(d_3^2 - d_1^2) \times \sigma_c$$

$d_3 =$ ___ mm

Step 6 (Other parameter length (l_1, l_2, l_3, l) breadth (b_2, b_1)

length $l = 4d$

$l_1 = l_2 = 0.75\, d$

$l_3 = 0.45 \times d$

breadth

$$\tan\alpha = \frac{b_2 - b_1}{l} \qquad \left(\text{Assume } \alpha = \frac{1}{24} \text{ (or) } \frac{1}{40}\right)$$

$$b_2 - b_1 = \_\_\_ \text{ mm}$$

$$b_2 = b + \frac{b_2 - b_1}{2} = \_\_\_ \text{ mm}$$

$$b_1 = \_\_\_ \text{ mm}$$

Pb 2 Design cotter joint to support a completely axial load of 25kN. Use steel for all components. Allowable stress for steel are in tension 50 N/mm², in compression 60 N/mm² and shear is 35 N/mm²

data:

(1) axial load (P) = 25KN = 25×10^3 N

2) Allowable stress in tension (σ_t) = 50 N/mm²

3) Allowable stress in compression (σ_c) = 60 N/mm²

4) Allowable shear stress (τ) = 35 N/mm²

To find (Cotter joint) (Socket and spigot joint)

Soln:

Step 1 (failure of rod in tension)

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

$$25 \times 10^3 = \frac{\pi}{4} \times d^2 \times 50$$

$$\boxed{d = 25.23 \text{ mm (or) } d \approx 25 \text{ mm}}$$

Step 2 failure of rod end at centre joint

$$P = \left[\frac{\pi}{4} d_1^2 - d_1 t\right) \times \sigma_t$$

Assume $t = 0.4d = 0.4 \times 25.23$

$$\boxed{t = 10 \text{ mm}}$$

$$25\times10^3 = \left(\frac{\pi}{4}d_1^2 - d_1\times10\right)\times50$$

$$25\times10^3 = 39.27d_1^2 - 500d_1$$

$$39.27d_1^2 - 500d_1 - 25\times10^3 = 0$$

$$\boxed{\begin{aligned} d_1 &= 32.38\,mm \\ d_2 &= -19.65\,mm \end{aligned}}$$

$$\boxed{d_{1,2} = \frac{-b\pm\sqrt{b^2-4ac}}{2a} \quad \text{using this form}}$$

Take only positive value

Step 3 Tension failure of socket end at cotter hole:

$$P = \left(\frac{\pi}{4}d_2^2 - \frac{\pi}{4}d_1^2 - (d_2-d_1)\times t\right]\times\sigma_t$$

$$25\times10^3 = \left(\frac{\pi}{4}d_2^2 - \frac{\pi}{4}(32.38)^2 - (d_2 - 32.38)\times10)\right]\times50$$

$$= \left(\frac{\pi}{4}\times d_2^2 - 823.46 - 10d_2 + 32.38\right]\times50$$

$$25\times10^3 = \left[\frac{\pi}{4}d_2^2 - 10d_2 - 499.60\right]\times50$$

$$= \left[39.27d_2^2 - 500d_2 - 24983\right]$$

$$= 39.27d_2^2 - 500d_2 - 24983 - 25\times10^3 = 0$$

$$39.27d_2^2 - 500d_2 - 49983 = 0$$

$$39.27d_2^2 - 500d_2$$

$$\boxed{\begin{aligned} d_2 &= 42.61\,mm \\ d_2 &= -29.87\,mm \end{aligned}}$$

Only consider positive value

$$\boxed{d_2 = 42.61\,mm}$$

Step 4 Check Crushing failure of cotter joint

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

$$25\times10^3 = (42.61 - 32.38)\times10\times\sigma_c$$

$$\boxed{\sigma_c = 244.37\,N/mm^2}$$

case (ii)

σ_c given $<$ σ_c Que

$40 < 244.37$ (Design is not safe)

Change the d_2 value:

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

$25 \times 10^3 = (d_2 - 32.38) \times 10 \times 60$

$\boxed{d_2 = 74\,mm}$

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

$25 \times 10^3 = (74 - 32.38) \times 10 \times \sigma_c$

$\boxed{\sigma_c = 59.9\,N/mm^2}$

σ_c given $>$ σ_c Ans

$60 > 59.9 \rightarrow$ Design is safe.

Step 5 shear failure cotter

$P = 2 b t \tau$

$25 \times 10^3 = 2 \times b \times 10 \times 35$

$b = 35.71\,mm$

$\boxed{b = 36\,mm}$

Step 6 - Crushing failure of Cotter joint

$P = \frac{\pi}{4} (d_3^2 - d_1^2) \times \sigma_c$

$25 \times 10^3 = \frac{\pi}{4} (d_3^2 - 32.38^2) \times 60$

$\boxed{d_3 = 39.73\,mm \simeq 40\,mm}$

Step 7 other parameter length (l, l_1, l_2, l_3) breadth (b_2, b_1)

length $= l = 4d$

$l = 4 \times 25 = 100 mm$

$l_1 = l_2 = 0.75 \times d = 0.75 \times 25 = 18.75 mm$

$l_3 = 0.45 \times d = 0.45 \times 25 = 11.25 mm$

breadth

$\tan\alpha = \frac{b_2 - b_1}{l}$ Assume $\alpha = \frac{1}{24}$ or $\frac{1}{40}$

$b_2 - b_1 = \tan\alpha \times l$

$= \tan\left(\frac{1}{24}\right) \times 100$

$\boxed{b_2 - b_1 = 0.073 mm}$

$b_2 = b_1 + \frac{b_2 - b_1}{2} = 36 + \frac{0.073}{2} = 36.03 mm$

$b_2 - b_1 = 0.073$

$b_1 = b_2 - 0.073 = 36.03 - 0.073 = 35.96 mm$

Design Sleeve and cotter joint

Sleeve and cotter joint is similar to the socket and spigot joint except the separate sleeve is fitted over each other & the loader & Same procedure for socket, spigot joint except add the value (in step 7: Instead of $L = 4 \times d$ change is $L = 8 \times d$.

Knuckle Joint

Step 1 find diameter

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

$d = \_\_ mm$

Step 2 Dimension of knuckle joint DB – 7.139

Dia of pin $d_1 = d$

Outer diameter of eye $d_2 = 2d$

Dia of pin head $d_3 = 1.5d$

Thickness of eye $t = 1.25d$

Thickness of fork : $t_1 = 0.75d$

Thickness of pin head $t_2 = 0.5d$

Step 3 [Check for shear stress for knuckle joint]

a) Check for failure of knuckle pin by double shear]

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

$$\tau_K = \_\_\_ N/mm^2$$

Case (i) $(\tau_K)_{que} > \tau_K$ (design safe)

Case (ii) $(\tau_K)_{que} < \tau_K$ (Design is not safe]

Change 'd_1' value [D.B 5.83]

b) Check for rod end by double shear.

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

$$\tau_K = \_\_\_ N/mm^2$$

Case (i) $(\tau_K)_{que} > \tau_K \rightarrow$ Design safe]

Case (ii) $(\tau_K)_{que} < \tau_K \rightarrow$ Design is not safe]

Change 't' value.

c) Force end in double shear.

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

$$\tau_K = \_\_\_ N/mm^2$$

Case (i) $(\tau_K)_{que} > \tau_K \rightarrow$ Design safe]

$(\tau_K)_{que} < \tau_K \rightarrow$ Design not safe

Change 't_1' value]

Step 4 Check for tensile stress (σ_t)

a) Check for rod end in tension

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

$$\sigma_t = \_\_\_ N/mm^2$$

Case (i) $(\sigma_t)_{given} > (\sigma_t)_{Ans} \rightarrow$ Design safe

$(\sigma_t)_{gn} < (\sigma_t)_{Ans} \rightarrow$ Design not safe

(Change 't' value)

(b) Check for fork end intension

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

$$\sigma_t = \text{—} \ N/mm^2$$

Case (i) $\sigma_t > \sigma_b \rightarrow$ Design safe

$\sigma_t < \sigma_b \rightarrow$ Design not safe (change 't_1' value)

Step 5 : Check for compressive stress

a) check for rod end intension

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

$$\sigma_c = \text{—} \ N/mm^2$$

Case (i) $(\sigma_c)_{gn} > (\sigma_c)_{Ans} \rightarrow$ Design safe

$(\sigma_c)_{gn} < (\sigma_c)_{Ans}$ — Design not safe change 't' value.

b) for end in crushing :

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

$$\sigma_c = \text{—} \ N/mm^2$$

Case (i) $(\sigma_c)_{que} > \sigma_c$ (Design safe)

$(\sigma_c)_{que} < \sigma_c$ (Design not safe)

Pb2 A knuckle joint is to transmit a force of 140 kN allowable stress in tension, shear and compression are $75 N/mm^2$, $65 N/mm^2$ and $140 N/mm^2$. Design the joint

data:

$$\text{load } (P) = 140 \text{ kN} = 140 \times 10^3 N$$

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

$$\tau_s = 65 N/mm^2$$

$$\sigma_c = 140 N/mm^2$$

find Design of knuckle joint

Soln: Step 1 Find diameter

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

$$140 \times 10^3 = \frac{\pi}{4} \times d^2 \times 75$$

$$\boxed{d = 48.75 mm \simeq 49 mm}$$

Step 2 Dimension of knuckle jonit

$d_1 = d = 49 mm$

$d_2 = 2d = 98 mm$

$d_3 = 1.5 d = 1.5 \times 49 = 75.5 mm$

$t = 1.25 d = 1.25 \times 49 = 61.2 mm$

$t_1 = 0.75 d = 0.75 \times 49 = 36.75 mm$

$t_2 = 0.5 d = 0.5 \times 49 = 24.5 mm$

Step 3 Check for shear stress for knuckle jonit

a) Check for failure of knuckle pin by double shear

$$P = 2 \times \pi/4 \, d_1^2 \times \tau_k$$

$$140 \times 10^3 = 2 \times \pi/4 \, (49)^2 \times \tau_k$$

$$\boxed{\tau_k = 37.12 \, N/mm^2}$$

Case (i) $(\tau_k)_{gn} > (\tau_k)_{Ans}$

$65 > 37.12 \rightarrow$ Design is safe

b) Check for rod end by double shear

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

$$140 \times 10^3 = (98 - 49) \times 61.2 \times \tau_k$$

$$\boxed{\tau_k = 46.68 \, N/mm^2}$$

Case (i) $(\tau_k)_{gn} > \tau_k (Ans)$ $65 > 46.68$

Design is safe

c) force end double shear

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

$$140 \times 10^3 = (98-49) \times 36.75 \times 2 \times \tau_K$$

$$\boxed{\tau_K = 38.87 \ N/mm^2}$$

case (i) $\tau_K (gm) > \tau_K (ans)$

$65 > 38.87 \rightarrow$ Design is safe.

Step 4 Check for tensile stress (σ_t)

a) Check for rod end in tension

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

$$140 \times 10^3 = \sigma_t \times (98-49) \times 61.25$$

$$\boxed{\sigma_t = 46.65 \ N/mm^2}$$

case (i) $(\sigma_t)_{gn} > (\sigma_t)_{ans}$

$75 > 46.65 \rightarrow$ Design is safe.

b) Check for fork end in tension

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

$$140 \times 10^3 = (98-49) \times 36.75 \times 2 \times \sigma_t$$

$$\boxed{\sigma_t = 38.87 \ N/mm^2}$$

case (i) $(\sigma_t)_{gn} > (\sigma_t)_{ans}$

$75 > 38.87 \rightarrow$ Design is safe.

Step 5 [Check for compressive stress]

a) Check for rod end in tension

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

$$140 \times 10^3 = 49 \times 61.25 \times \sigma_c$$

$$\boxed{\sigma_c = 46.65 \ N/mm^2}$$

Case(i) $(\sigma_c)_{fn} > (\sigma_c)_{An}$

$140 > 46.65 \rightarrow$ Design is safe.

b) For end in crushing:

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

$$140 \times 10^3 = 49 \times 36.75 \times 2 \times \sigma_c$$

$$\boxed{\sigma_c = 38.87 \ N/mm^2}$$

Case(i) $(\sigma_c)_{fn} > (\sigma_c)_{An}$

$140 > 38.87 \rightarrow$ Design is safe.

Riveting.

* Riveting is a joining of same material (or) different material it is also called as the permanent joint
* It is generally made up with mild steel (or) Iron, Some times it is made up of copper and aluminium whether corrosion resistance and light weight are required.

Types of riveting joint

* Lab joint
  a) Single riveted lab joint
  (b) Double " " " (zigzag)
  c) Double (chain riveted)
  d) Triple riveted lab joint
* Butt joint

Design procedure for riveting longitudinal and circumferential joint

Step 1 (To find thickness)

$$t = \frac{P_s \times D}{2 \times \eta_1 \times \sigma_{t_1}}$$ D.B → 7.126

where η_1 = — (7.126)

P_s = working pr

t = — mm

Step 2 to find dia

$d = 6\sqrt{t}$ → DB – 7.126 [$t > 8$ mm]

d = — mm

Step 3 [pitch of rivet 'p']

To find [F_s (strength) (F_t (tearing)] → DB 7.124

Tearing) $F_t = (p-d)\, t \times \sigma_t$

where p = pitch of rivet

F_t = — N ⟹ Eqn 1

Strength)

$F_s = n \times \frac{\pi d^2}{4} \times \tau \times i$ → DB 7.124

F_s = — N → Eqn 2

Eq(1) = Eqn (2) (i = based on no of rivets)

$F_t = F_s$ Assum = 1.875 (butt joint)

we find the value P n = 1 (Lap joint)

p = — mm

$P_{max} = C \times t + 41$ — (DB 7.126)

where $C =$ ___ [DB 7.126]

single (or) double

$P_{max} =$ ___ mm

Now compare the P_{max} and 'p' and choose the less value as 'p'

$$\boxed{P = \text{\_\_ mm}}$$

Step 4 : Thickness of cover plate (t_1)

$t_1 =$ ___ mm [DB 7.127 (single or double]

Step 5 : Margin of rivet (m)

$m =$ ___ mm [DB 7.127

Step 6 Distance bet the two rows of rivet

($P_b =$ ___) (DB 7.126 (chain rivet or zig zag)

$$\boxed{P_b = \text{\_\_ mm}}$$

Step 7 Efficiency of rivet joint (η)

Solve F_t value in step : 3

$F_t =$ ___ N

$$F_c = l \times d \times t \times \frac{\sigma_c}{n}$$

$F_c =$ ___ N

$F_s =$ ___ N (from step : 3)

$$\eta = \frac{\text{least of } F_t \text{ (or) } F_c \text{ (or) } F_s}{p \times t \times \sigma_t}$$

$$\boxed{\eta = \text{\_\_ \%}}$$

circumferential Joint

Step 1 (Diameter) $\rightarrow$ DB 7.127

$d = \_\_\_ mm$

Step 2 (Total no of rivet)

$$i = \left(\frac{D}{d}\right)^2 \frac{P_s}{L} \rightarrow \text{DB } 7.127$$

where $D = 1.6 \times d \rightarrow$ DB 7.124

$i = \_\_\_$

Step 3 [pitch of rivet, P]

$$\eta_c = \frac{P-d}{P} \rightarrow \text{DB } 7.127$$

$$\eta_c = \eta/2$$

$P = \_\_\_ mm$

Step 4 — (No of rows of rivet, r_n)

$$r_n = \frac{i \times P}{\pi(D+t)} \rightarrow \text{DB } 7.127$$

$r_n = \_\_\_$

.143
b: A single riveted lap joint is to be made of 10mm plates. Find the diameter of rivets, pitch and efficiency of the joint. Take $F_{shear} = 64\ N/mm^2$, $F_{tensile} = 80\ N/mm^2$. Also design the joint when its strength to withstand shear of rivets is equal to its strength to withstand tearing of the plate across the line of rivet holes.

data

1) Single Riveted lap joint

2) $t = 10\,mm$

3) Shear Stress (t_{shear}) = $\tau = 64\,N/mm^2$

4) Tensile stress ($t_{tensile}$) = $\sigma_t = 80\,N/mm^2$

5) $F_s = F_t$

To find

Dia and also design of rivet joint

Soln: Step 1 given $t = 10mm$

Step 2 to find dia

$$d = 6\sqrt{t} \rightarrow DB\ 7.126$$

$$= 6\sqrt{10}$$

$$\boxed{d = 18.97mm \simeq 19mm}$$

Step 3 (pitch of the rivet) P

Tearing → DB 7.124

$$F_t = (p-d)\,t \times \sigma_t$$

$$= (p-19) \times 10 \times 80$$

$$F_t = (p-19) \times 800 \rightarrow (1)$$

Strength $F_s = n \times \frac{\pi d^2}{4} \times \tau \times i \rightarrow DB\ 7.124$

$$F_s = 1 \times \frac{\pi}{4} (19)^2 \times 64 \times 1$$

[Assume n=1 lap joint, i=1 single rivets]

$$\boxed{F_s = 18152.81\,N}$$

Eqn (1) = Eqn (2)

$$F_t = F_s$$

$$(p-19) \times 80 \times 10 = 18152.81$$

$$\boxed{P = 41.69 \simeq 42\,mm}$$

$P_{max} = (C \times t) + c \rightarrow$ DB 7.126

$= (1.31 \times 16) + 41$ $\quad$ $C = 1.31$ (Lap Joint)

$\boxed{P_{max} = 54.1\,mm}$

Compare P_{max} and p choose least value

$\boxed{P = 42\,mm}$

Step 4 $\quad$ Thickness of cover plate — DB 7.127

$t_1 = 0.75t$ (single riveted)

$t_1 = 0.75 \times 10$

$\boxed{t_1 = 7.5\,mm}$

Step 5 $\quad$ Margin of rivet

$M = 1.5d \rightarrow$ DB 7.127

$= 1.5 \times 19$

$\boxed{M = 28.5}$

Step 6 $\quad$ Distance bet the two rows of rivet

$P_b = 3d \rightarrow$ DB 7.126 (not mention Chain (or) Zig Zag rivet)

$= 3 \times 19$

$\boxed{P_b = 57\,mm}$

Step 7 $\quad$ Efficiency of rivet joints (η)

Take F_t value in step 3

$F_t = (P - 19) \times 800$

$\boxed{P = 42\,mm}$ — In step: 3

$F_t = (42 - 19) \times 800$

$\boxed{F_t = 18400\,N}$

$$F_c = l\, d \times t \times \frac{\sigma_c}{FOS \text{ (or) } n}$$ (σ_c is not gn)

F_c = It is not satisfied,

F_c is neglect

$$\boxed{F_s = 18152.8\,N} \rightarrow \text{in step 3}$$

$$\eta = \frac{\text{least of } F_t \text{ (or) } F_s}{p \times t \times \sigma_t}$$

$$= \frac{18152.8}{42 \times 10 \times 80}$$

$$\boxed{\eta = 54.02\,\%}$$

Welded Joints

→ It is process of joining metallic parts of heating to a suitable temp, with or without the application of pressure.

Distinct application:

(i) It is substitute for riveted joints

(ii) It is alternative method for casting or forging.

Welding processes:

- Welding
  - Heat (fusion)
    - Thermit
    - Gas
    - Electric arc.
  - Heat & pressure
    - Forge
    - Electric resistance

Pb: A gas tank consist of a cylindrical shell of 2.5 m inner dia. It is enclosed by hemi spherical shell by means of butt welded joint. The thickness of the cylindrical shell as well as the hemispherical cover is 12 mm. Det the allowable internal pressure to which the tank may be subjected, if the permissible tensile stress in the weld is 85 N/mm². Assume efficiency of the welded joint

data:

$d_{inner} = 2.5 m = 2500 mm$

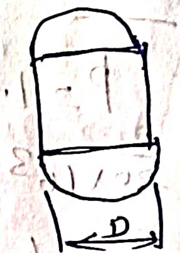
$t = 12 mm$

$\sigma_t = 85 N/mm^2$

$\eta = 0.85$

Find

Allowable internal pressure



Soln:

length of weld = circumference of the cylindrical shell

$l = \pi D = (\pi \times 2500) = 7853.93 mm$

Strength eqn

$$P = \sigma_t \times t \times l \times \eta = 85 \times 12 \times 7853.93 \times 0.85$$

$$P = 6809.4 \times 10^3 N$$

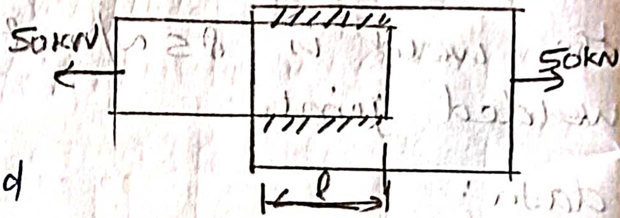
∴ pressure inside the cylinder.

$$\sigma \text{ (or) } p = \frac{P}{\frac{\pi}{4} d^2} = \frac{6809.4 \times 10^3}{\frac{\pi}{4}(2500)^2}$$

$$\boxed{P = 1.39 N/mm^2}$$

2) A Steel plate 100 mm and 10mm thick is welded to another steel plate by means of double parallel fillet welds. The plates are subjected to a static tensile force of 50 kN. Det the required length of the weld, if the permissible shear stress in weld is 94 N/mm²

data:



Width = 100mm

$t = 10mm$

$P = 50kN = 50\times10^3 N$

find

length L=??

Soln:

$$\boxed{P = 1.414\, h\, l\, \tau}$$

$$50\times10^3 = 1.414\times10\times l\times94$$

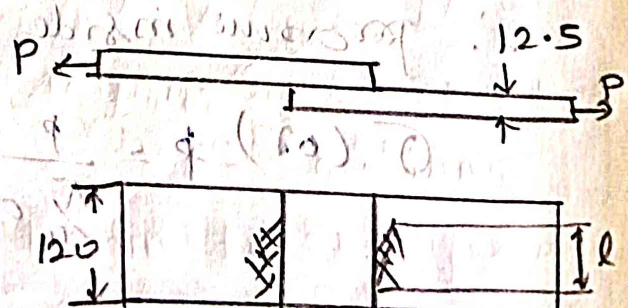
$$l = 37.62mm$$

Adding 15mm length for starting and stopping of the weld

$$l = 37.65 + 15$$

$$\boxed{l = 52.62 \simeq 55mm}$$

3) Two steel plates 120mm wide and 12.5mm thick are joined together by means of double transverse fillet weld. The maximum tensile stress for the plate and the welding material should not exceed 110 N/mm². Find the required length of the length if the strength of weld is equal to the strength of the plates.



data

Wide = 120 mm

$t = 12.5mm$

$\tau = 110\, N/mm^2$

find: length.

Soln:

Plates are subjected to tensile force.

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

$P = (wt)\sigma_t$

$= 120 \times 12.5 \times 110$

$P = 165000\,N$

$P = 1.414 \cdot h l \sigma_t$

$165000 = 1.414 \times 12.5 \times l \times 110$

$l = 84.87\,mm$

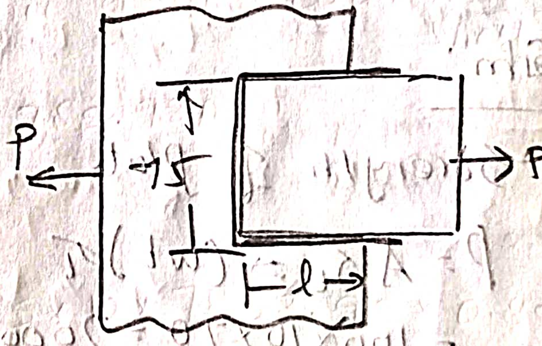
Adding 15mm starting and stopping

$l = 84.87 + 15$

$\boxed{l = 99.87\,mm \simeq 100\,mm}$

4) A plate 75mm wide and 10mm thick is joined with another steel plate by means of single transverse and double fillet welds. The joint is subjected to a maximum tensile force of 55kN. The permissible tensile and shear stresses in the weld material are 70 and 50 N/mm². Det the required length of each parallel fillet weld!

Soln:



The strength of transverse fillet weld is

$P_1 = 0.707\, h l \sigma_t$

$= 0.707 \times 10 \times 75 \times 70$

$P_1 = 37117.5\,N$

Strength of double fillet weld

$P_2 = 1.414\, h l \tau$

$= 1.414 \times 10 \times l \times 50$

$\boxed{P_2 = 707\,l\,N}$

total strength of weld = 55kN

$P_1 + P_2 = 55 \times 10^3$

$37117.5 + 707l = 55000$

$\boxed{l = 25.29\,mm}$

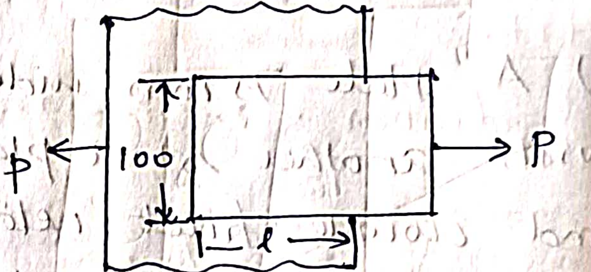
Adding 15mm for starting & stopping of weld

$l = 25.29 + 15 = 40.29\,mm$

$\simeq 45\,mm$

Pb 5 A Steel plate 100mm wide and 10mm thick is joined with another steel plate by means of a single transverse and double parallel fillet welds shown in fig. The strength of the welded joint should be equal to the strength of the plates to be joined. The permissible tensile and shear stresses for the weld material and the plate are 70 and 50 N/mm² resp. find the length of each parallel fillet weld. Assume the tensile force acting on the plate as static

data:



$W = 100mm$

$t = 10 mm$

$\sigma_t = 70 N/mm^2$

$\sigma_t = 50 N/mm^2$

find : Length of weld (L)

Soln:

Strength of plate

$P = A\sigma_t \Rightarrow (wt)\sigma_t$

$= 100 \times 10 \times 70 = 70{,}000N$

Strength of transverse fillet weld P_1

$P_1 = 0.707 h L \sigma_t$

$= 0.707 \times 10 \times 100 \times 70$

$P_1 = 49490N$

Strength of double parallel fillet weld P_2

$P_2 = 1.414 h l \tau$

$= 1.414 \times 10 \times l \times 50$

$= 707 \times l$

Strength of weld joint $\}$ = Strength of plate

$49490 + 707l = 70{,}000$

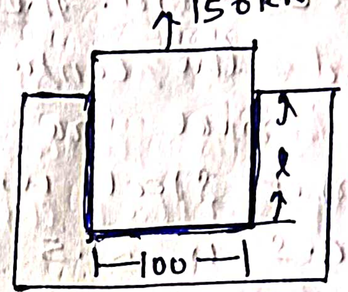
$l = 29.01mm$

15mm for starting and stopping of weld

$l = 29.01 + 15 = 44 mm$

$\simeq 45 mm$

6) Two plates are joined together by means of single transverse and double parallel fillet weld as shown in fig. The size of the fillet weld is 5mm and allowable shear load per mm of weld is 330N. Find the length of each parallel fillet weld.



data:

$t = 5mm$ $\quad P = 150KN = 150\times10^3 N$

$\tau = 330N$

find

Length of parallel weld

$L = \frac{P}{\tau} = \frac{150\times10^3}{330}$

$L = 454.55mm$

Parallel fillet weld

$L = (2l) + 100$

$454.55 = 2l + 100$

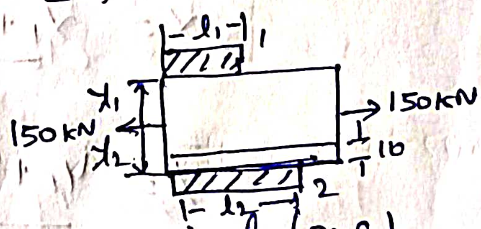
$\boxed{l = 177.27mm}$

Adding 15mm

$l = 177.27 + 15$

$\boxed{l = 195mm}$

7) An ISA 200x100x10 angle is welded to a steel plate by means of fillet welds, as shown fig. The angle is subjected to a static force 150KN and permissible shear stress for the weld is $70N/mm^2$. Det the length of weld of the top and bottom.



data $\quad P = 150\times10^3 N$

$\tau = 70N/mm^2$

find L

Soln:

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

$150\times10^3 = 0.707 \times 10 \times L \times 70$

$\boxed{l = 303.09 mm}$

Moment principle @ CG

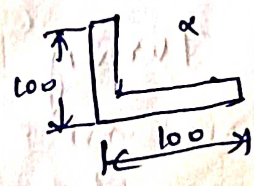
$l_1 y_1 = l_2 y_2$

$l_1(200 - 71.8) = l_2(71.8)$

$l_1 + l_2 = l = 303.09mm$

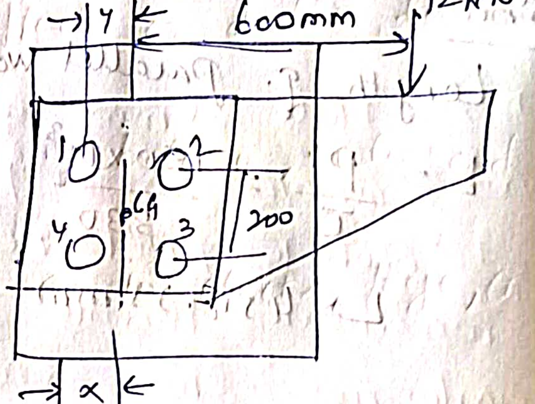
$l_1 = 108.81 mm.$

$l_2 = 194.28 mm$



Pb

The structural connection is subjected to an eccentric force of $P = 12\,kN$ with an eccentricity of 600mm. The centre distance 1 and 2 is 250mm and the centre distance between 2 and is 3 is 200mm. All rivet are identical. The rivet are made from carbon steel with yield strength $400\,N/mm^2$ and f.o.s is 2.5 yield stress in shear is 0.577 times the yield strength. find the minimum area required for the rivet.



data

$P = 12\,kN = 12 \times 10^3\,N$

$e = 600mm$

Centre distance $1 \& 2 = 250mm$

$2 \& 3 = 200mm$

$\sigma_y = 400\,N/mm^2$

$F.O.S = 2.5$

Yield shear $\tau_y = 0.577 \times \sigma_y = 0.577 \times 400 = 230.8\,N/mm^2$

Soln:

Total no of rivet, $n = 4$

$$\bar{x} = \frac{x_1 + x_2 + x_3 + x_4}{n} = \frac{0 + 250 + 250 + 0}{4} = 125mm$$

$$\bar{y} = \frac{y_1 + y_2 + y_3 + y_4}{4} = \frac{200 + 200 + 0 + 0}{4} = 100mm$$

Direct shear load $P_s = \frac{P}{n} = \frac{12000}{4} = 3000N$

Radial distance of rivet $l_1 = \sqrt{125^2 + 100^2} = 160.08\,mm$

$l_1 = l_2 = l_3 = l_4 = 160.08\,mm.$

Turning moment $T = P \times e$

$$P \times e = \frac{F_1}{l_1} \times (l_1^2 + l_2^2 + l_3^2 + l_4^2)$$

$$12000 \times 600 = F_1/l_1 \times 4 l_1^2$$

$$7200000 = \frac{F_1}{160.08} \times 4 \times 160.08^2$$

$$F_1 = 11244.38\,N$$

$$F_2 = F_1 \times \frac{l_1}{l_2} = 11244.38 \times \frac{160.08}{160.08} = 11244.38\,N$$

$$F_1 = F_2 = F_3 = F_4 = 11244.38\,N$$

Angle bet the direct and secondary shear load for these rivets $= \cos\theta_2$ and $\cos\theta_2$

$$\cos\theta_2 = \frac{125}{l_2} = \frac{125}{160.08} = 0.781$$

$$\cos\theta_2 = \cos\theta_3 = 0.781$$

Resultant shear load on rivet 2 & 3

$$R = R_2 = R_3 = \sqrt{P_s^2 + F^2 + 2P_s * F\cos\theta}$$

$$= \sqrt{3000^2 + 11244.38^2 + 2 \times 3000 \times 11244.38 \times 0.78}$$

$$R = 13715.95\,N$$

$$R = A \times \tau/_{FOS}$$

$$A = \frac{R}{\tau/_{FOS}} = \frac{13715.95}{\frac{230.8}{2.5}}$$

$$\boxed{A = 148.57\,mm^2}$$

UNIT - IV

Energy Storing elements and engine components:

Design of spring:

A spring is an elastic member which deflect under the action of load and it regains its orginal shape after the load is removed.

Function of spring.

∝ To provide cushioning effect or reduce the effect of shock or impact loading

∝ To measure force in spring balance, meter and engine indicator.

∝ To store energy such as in clocks, toys, circuit beaker and starter.

Classification of spring

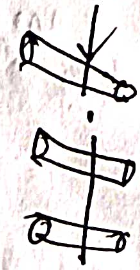
∝ Helical spring :-

Helical spring are made of circular wire coiled in to helical form and the total load is being applied to the axis of the helix.

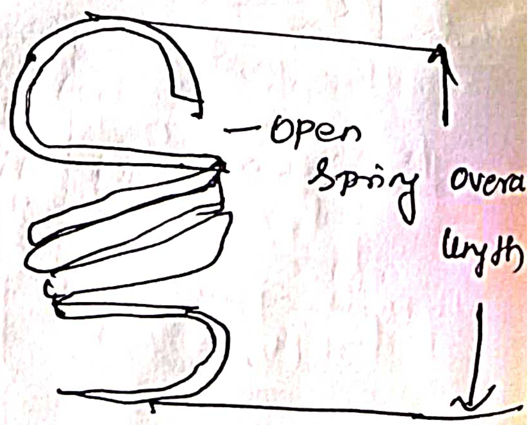
∝ Closed-coiled (or) Tension

∝ open - coiled (or) Compression.

These spring which are sustaining tensile along the axes are called helical tension.



a) compression helial spring



* Spiral spring:

These spring are made of flat strip wounded in the form of sprial and loaded in torsion.

* leaf spring:

A leaf spring consist of flat bar of vary in length, clamped together and supported at both end these acting as a simply supported beam.

* Disc spring:

These are made in the form of a cone disc to carry a high compressive force. Inorder to improve its load carrying capacity.

conical spring:

It is made up of round wire wounded in the shape of the cone. The major stress produced in the spring due to twisting the shear stress may occur.

Pb 1

Helical spring:

1. A gas engine value springs is to have a mean diameter of 37.5mm. The max load which will have to sustain is 450N. with a corresponding deflection of 12.5mm. The spring is to be made of tempered wire. The spring is to be subjected to repeated loading and the fatigue must be considered a low working stress of $300 N/mm^2$. Find the size for wire and number of coils used. Take rigidity modulus as $0.8 \times 10^5 N/mm^2$

data

$D = 37.5\,mm$

Max load on the spring $P = 450\,N$

Deflection of the spring $y = 12.5\,mm$

working stress $\tau = 300\,N/mm^2$

Rigidity modulus $G = 0.85 \times 10^5\,N/mm^2$

find

Size of the wire d

Soln:

Size of wire d: working stress (or) Shear stress

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

$$300 = \frac{1 \times 8 \times 450 \times 37.5}{\pi d^3}$$

$$\boxed{d = 5.2\,mm}$$

2. No of coil used n

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

$$12.5 = \frac{8 \times 450 \times 37.5^3 \times n}{0.85 \times 10^5 \times (5.2)^4}$$

$n = 5$

$\cancel{Y = 12.5\,mm}$ $\boxed{n = 4.15 \simeq 5}$

Assume end of the spring coil

$\therefore$ Total no of coils $n_t = n + 2$

$= 5 + 2$

$$\boxed{n_t = 7}$$

2) A helical spring made of C50 steel has an outside diameter of 80mm and a wire dia of 12mm. The spring has to support a maximum axial load of 1kN. Det the max shear stress and total deflection. If the spring have 10.5 coils with end ground flat. Also det f.o.s. Take $G = 0.89 \times 10^2$ kN/mm²

data

$D_o = 80$ mm

$d = 12$ mm

$P = 1$ kN

$n = 10.5$

$G = 0.89 \times 10^2$ kN/mm²

$= 0.89 \times 10^3$ N/mm²

Find

1) Max shear stress τ

2) Total deflection y

3) F.O.S

Soln:

Mean coil dia (D)

$D = D_o - d$

$= 80 - 12$

$\boxed{D = 68 \text{ mm}}$

Spring index $C = \frac{D}{d} = \frac{68}{12}$

$C = 5.66$

Wahl stress factor K_s

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

$= \frac{4 \times 5.66 - 1}{4 \times 5.66 - 4} + \frac{0.615}{5.66}$

$\boxed{K_s = 1.25}$

Max shear stress τ

$\tau = K_s \frac{8PC}{\pi d^2}$

$= \frac{1.25 \times 8 \times 1000 \times 5.66}{\pi \times 12^2}$

$\tau = 125.11$ N/mm²

ii) Total deflection (y)

$y = \frac{8PC^3 n}{G d}$

$= \frac{8 \times 1000 \times 10.5 \times (5.66)^3}{0.89 \times 10^5 \times 12}$

$\boxed{y = 14.26 \text{ mm}}$

(iii) fos (n_s)

C_{50} steel

$\sigma_y = 340$ N/mm²

(data book)

$n_s = \frac{\text{Yield stress } (\sigma_y)}{\text{Working stress } (\tau)}$

$= \frac{340}{125.11} = 2.72$

$\boxed{n_s = 3}$

3. A helical spring is subjected to a load varying from 400N to 1000N, having the spring index of 6 and the design factor of safety is 1.25. The compression of the spring at the maximum load is 30mm. Design the helical compression spring. Take the yield stress in shear as 110 N/mm². Endurance stress in shear as 350 N/mm² and modulus of rigidity for the spring material as 80 × 10³ N/mm²

Data

$P_{min} = 400N$

$P_{max} = 1000N$

$C = 6$

$n_s = 1.25$

$y = 30mm$

$\tau_y = 110 N/mm^2$

$\tau_{-1} = 350 N/mm^2$

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

find: Design of spring

1) Dia of wire (d)

$P_m = \frac{P_{max} + P_{min}}{2} = \frac{1000-400}{2}$

$P_a = 300N$

Shear stress factor K_{sh}

$K_{sh} = 1 + \frac{0.615}{C} = 1 + \frac{0.615}{6}$

$K_{sh} = 1.1025$

$\tau_m = \frac{8 K_{sh} P_m C}{\pi d^2} = \frac{8 \times 1.1025 \times 700 \times 6}{\pi d^2}$

$\tau_m = 11791.47/d^2$

data book $C = 6$ $K_c = 1.15$

wahl stress factor K_s

$K_s = K_{sh} \times K_c = 1.1025 \times 1.15$

$\boxed{K_s = 1.2678}$

Amplitude shear stress τ_a

$\tau_a = \frac{8 K_s P_a C}{\pi d^2} = \frac{8 \times 1.2678 \times 6}{\pi d^2}$

$\tau_a = 5811.51/d^2$

Repeated loading

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

$\frac{1}{1.25} = \frac{\left(\frac{11791.47}{d^2}\right) - \frac{5811.51}{d^2}}{110} + \frac{\left(\frac{2 \times 5811.51}{d^2}\right)}{350}$

$\frac{\frac{5879.96}{d^2}}{110} + \frac{\frac{11623.02}{d^2}}{350}$

$\frac{1}{1.25} = \frac{87.571}{d^2}$

$d^2 = 109.46 = 10.6mm$

2. Mean coil dia D;

Spring Index $C = D/d$

$D = C \times d = 6 \times 10.6$

$D = 63.6mm$

3. No of active turn n;

$y = \frac{8 P_{max} C^3 n}{G d}$

$30 = \frac{8 \times 1000 \times 6^3 n}{80 \times 10^3 \times 10.6}$

$n = 15$

$n_t = n + 2 = 15 + 2 = 17$

(iv) Solid length of the spring (L_s)

$$L_s = dn + 2d$$
$$= (10.6\times15) + (2\times10.6)$$
$$L_s = 180.2\,mm$$

v) Free length of the spring (L_f)

$$L_f = L_s + y = 180.2 + 30 = 210.2\,mm$$

(vi) Pitch of the coil P

$$P = \frac{L_f - L_s}{n_t} + d = \frac{210.2 - 180.2}{17} + 11$$
$$P = 12.76\,mm$$

(vii) Helix angle of the coil α;

$$\alpha = \tan^{-1}\left(P/\pi D\right) = \tan^{-1}\left(\frac{12.76}{\pi\times63.6}\right) = 3.65^\circ$$

(viii) Spring rate q

$$q = \frac{P_{max}}{y} = \frac{1000}{30} = 33.33\,N/mm^2$$

4) A Safety valve of 60mm dia is to blow off at pressure of $1.2\,N/mm^2$. It is placed on its seat by a close coiled helical spring. The max lift of the valve is 10mm. Design a suitable compression spring of spring index 5 and providing an initial compression of 35mm. The max shear stress in the material of the wire is limited to $500\,N/mm^2$. The modulus of rigidity for the spring matl is $0.80\times10^5\,N/mm^2$. Cal (1) Dia of spring 2) Mean coil dia 3) no of active turns. 4) Pitch of the coil.

data:

$d = 60\,mm$ $\qquad$ $\tau = 500\,N/mm^2$

$p = 1.2\,N/mm^2$ $\qquad$ $G = 80\,N/mm^2$

$\delta_2 = 10\,mm$

$C = 5.$

$\delta_1 = 35\,mm$

To find 1) D 2) D_m 3) N_t 4) Pitch.

Soln:

1) Dia of spring wire

P_1 = Area x max. pressure

$= \frac{\pi}{4} \times 60^2 \times 1.2$

$P_1 = 3394 N$

Max Compression of spring

$\delta_{max} = \delta_1 + \delta_2 = 35 + 10$

$= 45 mm$

$P = \frac{3394}{35} \times 45$

$P = 4364 N$

Wahl's stress factor

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

$= \frac{4\times5-1}{4\times5-4} + \frac{0.615}{5}$

$K = 1.31$

max shear stress (τ)

$\tau = K \frac{8PC}{\pi d^2}$

$= \frac{1.31 \times 8 \times 4364 \times 5}{\pi d^2}$

$500 = \frac{72780}{d^2}$

$d = 12.06 mm \simeq 12.7$

2) Mean of coil dia

$C = D/d$ $D = Cd$

$D = 5 \times 12.7 = 63.5 mm$

3) No of active turn.

$\delta = \frac{8PC^3 n}{Gd}$

$= \frac{8 \times 4364 \times 5^3 \times n}{80 \times 10^3 \times 12.7}$

$\delta = 45/4.3$

$\delta = 10.5 \approx 11$

$n_t = n + 2 = 11 + 2 = 13$

4) Pitch of the coil

Free length of coil (L_F)

$L_F = n_t d + \delta_{max} + 0.15\delta_{max}$

$= 13 \times 12.7 + 45 + 0.15 \times 45$

$L_f = 216.85 mm$

Pitch of coil $= \frac{\text{free length}}{n_t - 1}$

$= \frac{216.85}{13-1}$

Pitch of coil $= 18.1 mm$

Design of leaf spring:

Pbl A Semi - elliptical spring has 10 leaves with two full long leaves extending 650 mm. It is 65 mm wide and is made of 7 mm thick. Design a helical spring with mean coil dia 100 mm. which will have approximately the same value of induced stress and deflection for load.

data:

$G = 0.85\times10^5\ N/mm$

No of leaves $n = 10$

No of full length $n_e = 2$

No of graduated $= n_g = 8$

Leaf spring $2L = 650$ mm

$L = 325$ mm

width of leaves $b = 65$ mm

Thick of leaves $t = 7$ mm

For helical spring

Mean Diameter $D = 100$ mm

Soln

$$\sigma = \frac{6PL}{nbt^2} = \frac{6\times P\times 325}{10\times 65\times 7^2} \Rightarrow 0.0612P$$

$$\sigma = 0.0612P \rightarrow (1)$$

Max Shear stress induced helical spring

$$\sigma = \frac{8PD}{\pi d^3} = \frac{8\times P\times 100}{\pi d^3} \quad K_s = 1$$

$$\sigma = \frac{254.6P}{d^3} \rightarrow (2)$$

The value of stress induced in both spring

$$0.0612P = \frac{254.6P}{d^3}$$

$$d^3 = 4159.55 = 16\text{mm}$$

Deflection of leaf spring

$$y = \frac{12PL^3}{Ebt^3(3n_e+2n_g)}$$

$$= \frac{12\times P\times(325)^3}{2\times10^5\times65\times7^3(3\times2+2\times8)}$$

$$y = 4.2\times10^{-3}P \rightarrow (3)$$

Deflection of helical spring

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

$$= \frac{8\times P\times(100)^3\times n}{0.85\times10^5\times16^4}$$

$$y = 1.436\times10^{-3}P\times n \rightarrow 4$$

Solve (3). & (4)

$$4.2\times10^{-3}P = 1.436\times10^{-3}\times P\times n$$

$$n = \frac{4.2\times10^{-3}\cancel{P}}{1.436\times10^{-3}\cancel{P}}$$

$$\boxed{n = 2.92 \simeq 3}$$

4.2 Design of flywheels.

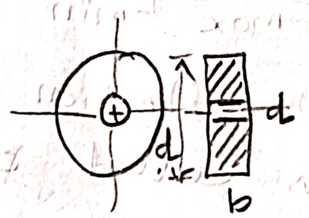
A flywheel is a heavy rotating mass which is placed between power source and driven member to act as a reservoir of energy.

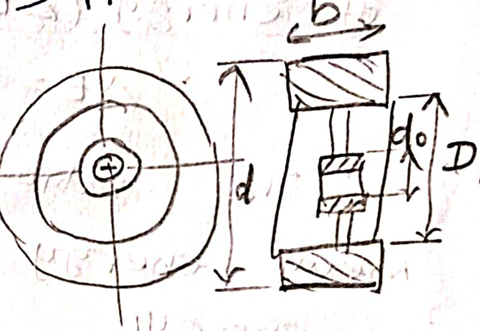
The primary function of flywheel is to act as "energy accumulator"

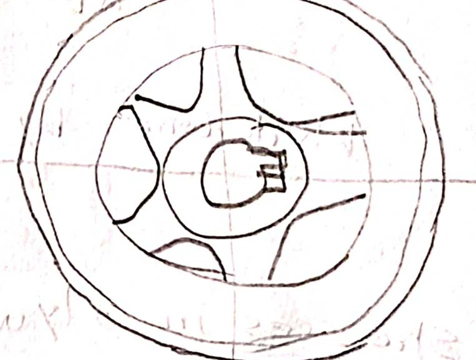
It will absorb energy when the demand is more than the energy being supplied.

Types

1) Disc type
2) web type
3) Arm type







Flywheel Effect and Co-efficient of fluctuation of speed:

Kinetic energy $E = \frac{1}{2} m v^2$

$= \frac{1}{2} m \omega^2 k^2$

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

I – mass moment of inertia

mk^2

m – mass of flywheel

k – Radius of gyration

ω – Angular speed

$$\Delta E = \frac{1}{2} I \left[\omega_{max}^2 - \omega_{min}^2 \right]$$

$$\boxed{\Delta E = I \omega^2 K_s}$$

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

$= m k^2 \omega^2 C_s$

$$\boxed{\Delta E = 2 E C_s}$$

$K_s \rightarrow$ Co efficient of fluctuation speed $= \frac{\omega_{max}^2 - \omega_{min}^2}{\omega}$

$\omega \rightarrow$ Mean angular speed $= \frac{\omega_{max} + \omega_{min}}{2}$

$N = \frac{N_1 + N_2}{2}$, $\omega = \frac{\omega_1 + \omega_2}{\omega}$ $C_s = \frac{N_1 - N_2}{N}$ or $\frac{\omega_1 - \omega_2}{\omega}$

Mass of fly wheel $m = \dfrac{\Delta E}{K^2 \omega^2 K_s}$

$m =$ volume x density

$m = \pi D \times A \times \rho$

A - cross sectional area of rim = $b \times h$

Co-efficient of fluctuation energy (K_E)

The difference between maximum and minimum energy during the cycle is called fluctuation of energy (ΔE).

$$\Delta E = E_{max} - E_{min}$$

The ratio of fluation of energy to the mean energy is called co-efficient of fluctuation of energy.

$$K_E = \frac{E_{max} - E_{min}}{E} = \frac{\Delta E}{E}$$

Work done by cycle $= \dfrac{P \times 60}{n}$ ← no. of working stroke per min

$n = N \to 2$ stroke

$n = N/2 \to 4$ stroke

Stresses in flywheel Rim :

Tensile stress due to centrifugal force

$$\sigma_t = \frac{\gamma v^2}{g} = \rho v^2 \qquad v = \frac{\pi D N}{60}$$

Bending stress $\sigma_b = \dfrac{\pi^2 v^2 D \rho}{n^2 h}$

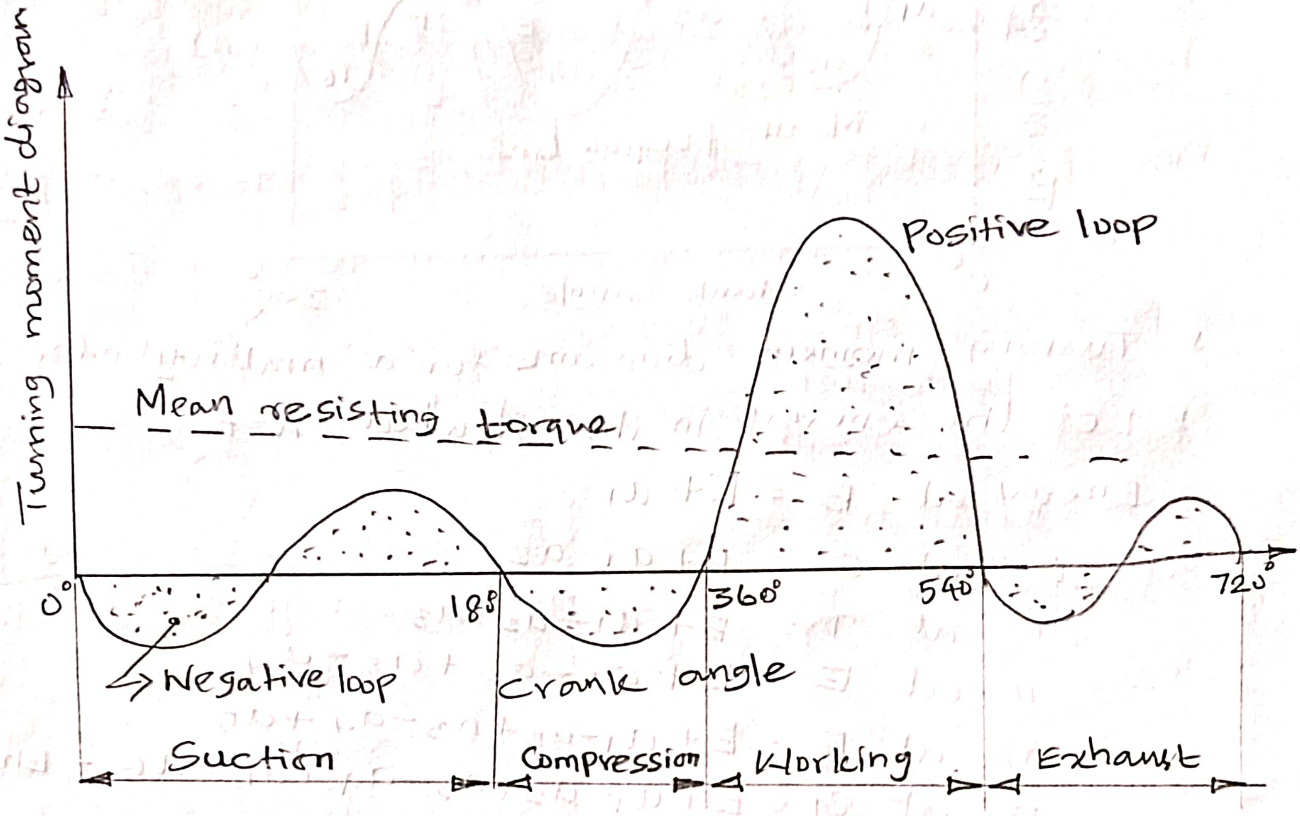
$$\sigma_{total} = \frac{3}{4}\sigma_t + \frac{1}{4}\sigma_b$$

$$= (0.75 \times \rho v^2) + \frac{0.25 \times \pi^2 v^2 D \rho}{n^2 h}$$

$$= \rho v^2 \left[0.75 + \frac{4.935 R}{n^2 h} \right]$$

$\sigma_{total} \leq 40 MN/m^2$.

Turning moment diagram for a four stroke internal combustion engine.



Crank has turned 720° or 4π radians

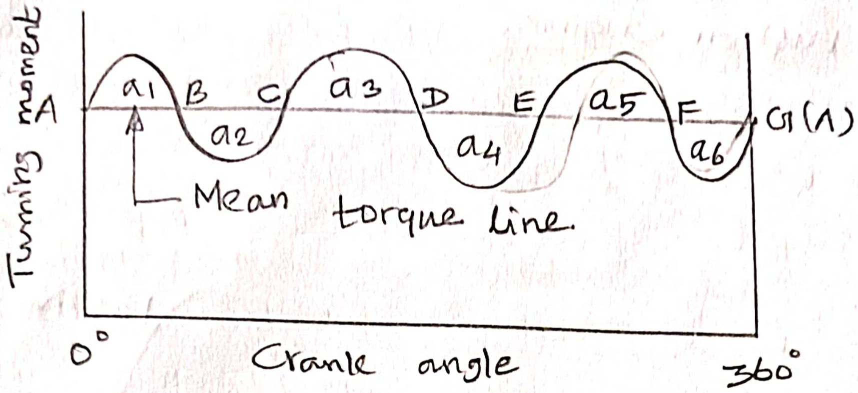
Suction :- Pressure inside the cylinder is less than atmospheric pressure, therefore negative loop is formed.

Compression :- Work is done on the gases, higher negative is formed.

Working :- Fuel burns the gases expand, large positive loop is formed.

Exhaust : Work is done on the gases, negative loop is formed.

Maximum Fluctuation of Energy:



* Turning moment diagram for a multicylinder engine.
* Let the energy in the flywheel $A = E$

Energy at $B = E + a_1$

" at $C = E + a_1 - a_2$

" at $D = E + a_1 - a_2 + a_3$

" at $E = E + a_1 - a_2 + a_3 - a_4$

" at $F = E + a_1 - a_2 + a_3 - a_4 + a_5$

" at $G = E + a_1 - a_2 + a_3 - a_4 + a_5 - a_6$ = Energy at A

Suppose :-

Maximum energies is at B.

Minimum energies at E.

$\therefore$ Maximum energy in the flywheel $= E + a_1$

Minimum energy in flywheel $= E + a_1 - a_2 + a_3 - a_4$

$\therefore$ Maximum fluctuation of energy

ΔE = Maximum Energy − Minimum energy

$= (\cancel{E} + \cancel{a_1}) - (\cancel{E} + \cancel{a_1} - a_2 + a_3 - a_4)$

$= a_2 - a_3 + a_4$

Coefficient of fluctuation of Energy (C_E)

It is defined as the ratio of the maximum flucturtion of energy to the workdone per cycle.

$$C_E = \frac{\text{Maximum fluctuation of energy}}{\text{Work done per cycle.}}$$

1) Workdone / cycle = $T_{mean} \times \theta$

$\theta = 2\pi \rightarrow$ 2 stroke

$\theta = 4\pi \rightarrow$ 4 stroke

$$T_{mean} = \frac{P \times 60}{2\pi N} = \frac{P}{\omega}$$

$\therefore \omega = \frac{2\pi N}{60}$ ← angular speed

(or)

$$\text{Workdone per cycle} = \frac{P \times 60}{n}$$

← no. of working strokes per min.

$n = N \rightarrow$ 2 stroke

$n = \frac{N}{2} \rightarrow$ Foure Stroke

1) The intercepted areas between the output torque curve and the mean resistance line of turning moment diagram for a multicylinder engine, taken in order from one end are as follows

$-35, +410, -285, +325, -335, +260, -365, +285 - 260\,mm^2$.

The diagram has been drawn to a scale of 1mm = 70N-m and 1mm = 4.5°. The engine speed is 900 rpm and the fluctuation in speed is not to exceed 2% of the mean speed.

Find the mass and cross section of the flywheel rim having 650 mm mean diameter. The density of the material of the flywheel may be taken as 7200 kg/m³. The rim is rectangular with the width 2 times the thickness. Neglect effect of arms etc.

Given data

$N = 900$ rpm

$\therefore \omega = \dfrac{2\pi \times 900}{60} = 94.26$ rad/s

$\omega_1 - \omega_2 = 2\% \,\omega$ (or) $\dfrac{\omega_1 - \omega_2}{\omega} = 2\% = 0.02$

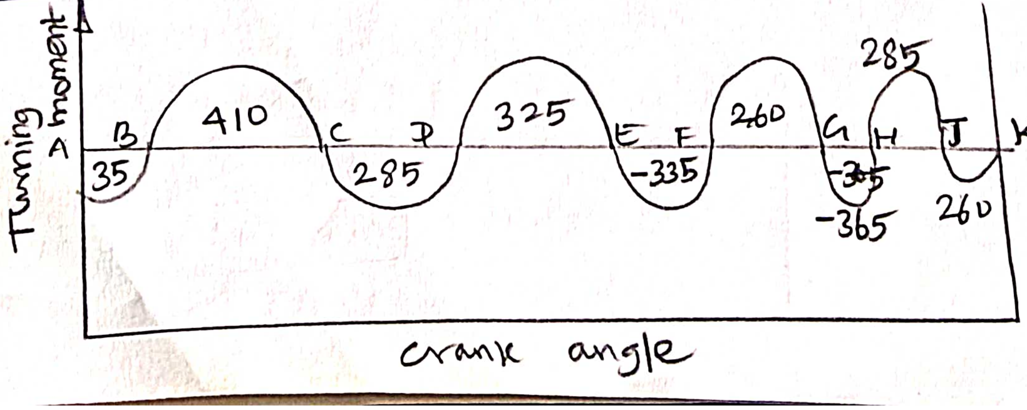
$D = 650\,mm = 0.650\,m$

$R = 325\,mm = 0.325\,m$

$\rho = 7200\,kg/m^3$

Find
1) mass of flywheel rim
2) cross section of flywheel rim.

Solution



1 mm = 70 N-m (Turning moment)

1 mm = $4.5^\circ = \frac{\pi}{40}$ (crank angle)

$\therefore = 70 \times \frac{\pi}{40} = 5.5$ N-m

Total energy at A = E

Energy at B = E − 35

" " C = E − 35 + 410 = E + 375

" D = E + 375 − 285 = E + 90

" E = E + 90 + 325 = E + 415

" F = E + 415 − 335 = E + 80

" G = E + 80 + 260 = E + 340

" H = E + 340 − 365 = E − 25

" K = E − 25 + 285 = E + 260

" L = E + 260 − 260 = E @ A.

$\therefore$ Maximum energy = E + 415

minimum energy = E − 35

$\therefore$ Maximum fluctuation energy (ΔE)

$\Delta E = (E + 415) - (E - 35) = 450\ mm^2$

$= 450 \times 5.5$

$\Delta E = 2475$ N-m

$\Delta E = m R^2 \omega^2 C_s \Rightarrow m (0.325)^2 \times (94.26)^2 \times 0.02$

$m = \frac{2475}{18.77} = 132$ kg.

2) cross section of the flywheel rim

$A = b \times t \Rightarrow 2t \times t = 2t^2$

mass of fly wheel

$132 = A \times 2\pi R \times \rho = 2t^2 \times 2\pi \times 0.325 \times 7200$

$132 = 29409\, t^2$

$t^2 = 0.0044$ or $t = 0.067$ m

$\boxed{t = 67\ mm}$

$b = 2t = 2 \times 67$ $\boxed{b = 134\ mm}$

2) The area of the turning moment diagram for one revolution of a multi cylinder engine with reference to the mean turning moment, below and above the line.

−32, +408, −267, +333, −314, +226, −374, +260 & −244 mm²

The scale for 1 mm = 2.4° and 1 mm = 650 N-m respectively. The mean speed is 300 rpm with percentage speed fluctuation of ± 1.5 %. If the hoop stress in the material of the rim is not to exceed 5.6 MPa, determine the suitable diameter and cross section for the flywheel, assuming the width is equal to 4 times the thickness. The density of the materials may be taken as 7200 kg/m³. Neglect the effect of the boss and arms.

Given data:

$N = 300$ rpm

$\sigma_t = 5.6\,MPa = 5.6 \times 10^6\,N/m^2$

$\rho = 7200\,kg/m^3$

Solution.

Angular velocity $\omega = \frac{2\pi N}{60}$

$$\omega = \frac{2\pi \times 300}{60} = 31.42\,rad/s$$

Velocity of flywheel, V

$$V = \frac{\pi D N}{60} = \frac{\pi \times D \times 300}{60} = 15.71\,D\ m/s$$

Hoop stress $\sigma_t = \rho v^2$

$$5.6 \times 10^6 = 7200 \times (15.71 D)^2$$

$$= 1.8 \times 10^6 D^2$$

$$D^2 = \frac{5.6 \times 10^6}{1.8 \times 10^6} = 3.11$$

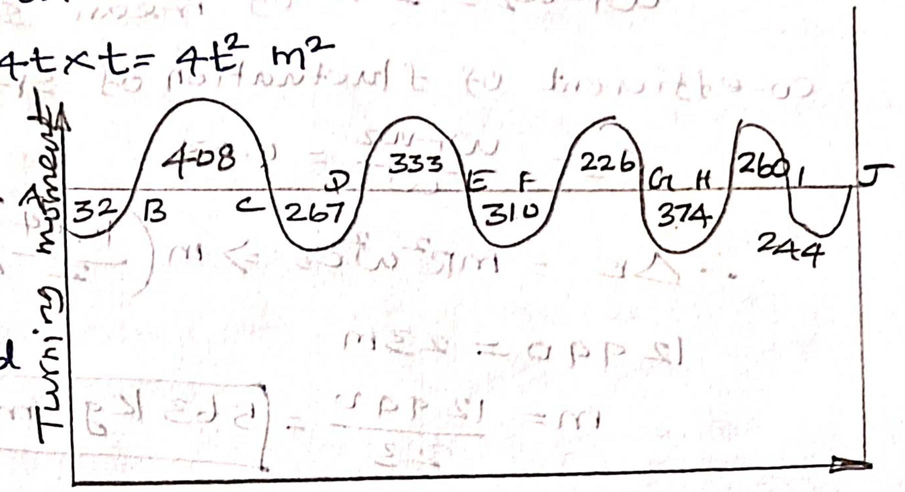
$$\boxed{D = 1.764\,m}$$

crank angle

Cross-section of the flywheel :-

Cross-sectional area of the rim

$A = b \times t = 4t \times t = 4t^2 \ m^2$



Scale

Crank $1mm = 2.4^\circ$

$= 2.4 \times \frac{\pi}{180}$

$1mm = 0.042$ rad

Turning moment $1mm = 650$ N-m

$1mm^2$ on the turning moment diagram

$= 650 \times 0.042$

$= 27.3$ N-m

Let total energy A = E

Energy at B = E − 32

" C = E − 32 + 408 = E + 376

" D = E + 376 − 267 = E + 109

" E = E + 109 + 333 = E + 442

" F = E + 442 − 310 = E + 132

G = E + 132 + 226 = E + 358

H = E + 358 − 374 = E − 16

I = E − 16 + 260 = E + 244

Energy at J = E + 244 − 244 = E = Energy at A

∴ Energy maximum at E & minimum at B

We know that maximum fluctuation of energy

ΔE = Maximum energy − Minimum energy

= (E + 442) − (E − 32)

$= 474 mm^2$ ← $1mm^2 = 27.3$ N-m

∴ = 474 × 27.3

$\boxed{\Delta E = 12940 \text{ N-m}}$

Fluctuation of speed ± 1.5 of mean speed.

$1.5 - (-1.5) = 3$

$\omega_1 - \omega_2 = 3\%$ of mean speed 0.03ω

Co-efficient of fluctuation of speed

$$C_s = \frac{\omega_1 - \omega_2}{\omega} = 0.03$$

$$\therefore \Delta E = mR^2\omega^2 C_s \Rightarrow m\left(\frac{1.764}{2}\right)^2 (31.42)^2 \times 0.03$$

$$12940 = 23m$$

$$m = \frac{12940}{23} = \boxed{563\ kg = m}$$

Mass of the flywheel rim

$$m = A \times \pi D \times \rho$$

$$563 = 4t^2 \times \pi \times 1.764 \times 7200$$

$$t^2 = \frac{563}{159624} = 0.00353$$

$$t = 0.0594m \Rightarrow 59.4mm \Rightarrow 60mm.$$

$$b = 4t = 4 \times 60 = 240\ mm.$$

Result

Mass of rim $(m) = 563\ kg$

thickness $(t) = 60mm$

width $(b) = 240mm$.

3) A Punching machine makes 25 working strokes per min, and is capable of Punching 25 mm diameter holes in 18 mm thick steel plates having an ultimate shear strength of 300 MPa.

The Punching operation takes place during 1/10th of revolution of the crank shaft.

Estimate the Power needed for the driving motor, assuming efficiency of 95%. Determine suitable dimensions for the rim cross section of the flywheel which is to revolve at 9 times the speed of the crank shaft. The permissible coefficient of fluctuation of speed is 0.1.

The flywheel is to be made of cast iron having a working stress (tensile) of 6 MPa and density of 7250 kg/m³. The diameter of the flywheel must not exceed 1.4 m owing to space restrictions. The hub and the spokes may be assumed to provide 5% of the rotational inertia of the wheel.

Check for the centrifugal stress induced in the rim.

Given data:

$n = 25$

$d_1 = 25\,mm$

$t_1 = 18\,mm$

$\tau_u = 300\,MPa = 300\,N/mm^2$

$\eta_m = 95\% = 0.95$

$C_s = 0.1$

$\sigma_t = 6\,MPa = 6\,N/mm^2$

$\rho = 7250\,kg/m^3$

$D = 1.4\,m$ or $R = 0.7\,m$.

Solution

$Area = \pi d_1 \times t_1 = \pi \times 25 \times 18 = 1414\ mm^2$

Maximum shearing force required for Punching

$$F_s = A_s \times \tau_u = 1414 \times 300 = 424200\ N$$

Energy required per stroke

$= $ Avg shear force $\times$ Thickness of Plate

$= \frac{1}{2} F_s \times t_1 \Rightarrow \frac{1}{2} \times 424200 \times 18$

$= 3817.8 \times 10^3$ N-mm

Energy required per min

$= $ Energy / stroke $\times$ No. of working strokes/min

$= 3817.8 \times 10^3 \times 25$

$= 95.45 \times 10^6$ N-mm $\Rightarrow 95450$ N-m

Power needed for the driving motor

$= \dfrac{\text{Energy required per min}}{60 \times \eta_m}$

$= \dfrac{95450}{60 \times 0.95} = 1675$ W.

$P = 1.675$ kW

Dimensions for the rim cross-section.

$A = b \times t = 2t \times t = 2t^2$

Punching operation takes place $1/10^{th}$ revolution of crank shaft

$\therefore\ 9/10$ energy is stored

Maximum Fluctuation of energy

$\Delta E = \frac{9}{10} \times$ Energy / stroke

$= 9/10 \times 3817.8 \times 10^3 = 3436 \times 10^3$ N-mm

$\boxed{\Delta E = 3436\ \text{N-m}}$

5%. → 95%.

Maximum fluctuation of energy

$(\Delta E)_{rim} = 0.95 \times \Delta E = 0.95 \times 3436$

$= 3264$ N-m

Fly wheel revolve at 9 times the speed of Crank shaft and 25 working strokes/min.

$N = 9 \times 25 = 225$ r.p.m

Mean angular speed $\omega = 2\pi \times \frac{225}{60}$

$\omega = 23.56$ rad/s

Maximum fluctuation of energy (ΔE)

$3264 = mR^2\omega^2 C_s \Rightarrow m \times (0.7)^2 (23.56)^2 0.1$

$3264 = 27.2m$

$m \Rightarrow \frac{3264}{27.2} \Rightarrow 120$ kg

Mass of the fly wheel (m)

$120 = A \times \pi D \times \rho \Rightarrow 2t^2 \times \pi \times 1.4 \times 7250$

$120 = 63782 t^2$

$t^2 = \frac{120}{63782} = 0.00188$

$$\boxed{\begin{array}{l} t = 0.044\text{m} = 44\text{mm} \\ b = 2t = 2 \times 44 = 88\text{mm} \end{array}}$$

Check centrifugal stress

$v = \frac{\pi D N}{60} = \frac{\pi \times 1.4 \times 225}{60} = 16.5$ m/s

$\sigma_t = \rho v^2 = 7250(16.5)^2 = 1.97 \times 10^6$ N/m²

$\sigma_t = 1.97$ MPa

$\boxed{1.97 < 6 \text{ MPa}}$

Stress induced is less than permissible value.

4) The Punching Press Pierces 35 holes per minute in a plate using 10 kN-m of energy per hole during each revolution. Each Piercing takes 40% of the time needed to make one revolution. The Punch receives power through a gear reduction unit which in turn is fed by a motor driven belt Pulley 800 mm diameter and turning at 210 rpm. Find the Power of the electric motor if overall efficiency of the transmission unit is 80%. Design a cast iron flywheel to be used with the punching machine for a coefficient of steadiness of 5, if the space consideration limit the maximum diameter 1.3 m

Allowable shear stress in shaft material = 50 MPa

Allowable tensile stress for CI = 4 MPa

Density of cast iron = 7200 kg/m³.

Given data:

No. of holes = 35/min

Energy/hole = 10 kN-m = 10,000 N-m

$d = 800\,mm = 0.8\,m$

$N = 210\,rpm$

$\eta = 80\% = 0.8$

$\frac{1}{C_s} = 5 \therefore C_s = \frac{1}{5} = 0.2$

$D_{max} = 1.3\,m$

$\tau = 50\,MPa = 50\,N/mm^2$

$\sigma_t = 4\,MPa = 4\,N/mm^2$

$\rho = 7200\,kg/m^3$

Solution

1) Power of the electric motor

Energy used = No. of holes × Energy used per hole

$= 35 \times 10{,}000$

$= 350\,000$ N-m/min

$P = \frac{\text{Energy used/min}}{60 \times \eta} = \frac{350\,000}{60 \times 0.8}$

$P = 7297\,W = 7.29\,kW$

2) Design of cast iron fly wheel

overall efficiency of transmission 80%.

$E_T = \frac{10{,}000}{0.8} = 12{,}500$ N-m

velocity of belt $V = \pi d N = \pi \times 0.8 \times 210 = 528$ m/min

$\therefore$ Net Tension $= \frac{P \times 60}{V} = \frac{7292 \times 60}{528} = 828.6\,N$

Time required to punch hole $= \frac{0.4}{35} = 0.0114$ min

Distance moved by belt = velocity of the belt × Time required to punch hole

$= 528 \times 0.0114$

$= 6.03\,m$

Energy supplied to punching hole E_B

E_B = Net tension × Distance travelled by belt

$= 828.6 \times 6.03$

$E_B = 4996$ N-m

$\Delta E = E_T - E_B = 12500 - 4996 = 7504$ N-m.

1) Mass of the flywheel.

$\Delta E = m R^2 \omega^2 C_s$

$7504 = m \times \left(\frac{1.2}{2}\right)^2 \times \left(\frac{2\pi \times 210}{60}\right)^2 \times 0.2$

$m = \frac{7504}{34.85} = 215.3 kg$

$\boxed{m = 215.3 \text{ kg}}$

2) cross sectional dimensions of the flywheel rim

$A = b \times t = 2t \times t = 2t^2$

$m = A \times \pi D \times \rho \Rightarrow 2t^2 \times \pi \times 1.2 \times 7200$

$215.3 = 54.3 \times 10^3 t^2$

$t^2 = 0.00396$

$\boxed{t = 0.063 \approx 65 mm.}$

$\boxed{b = 2t = 2 \times 65 = 130 mm}$

3) Diameter and length of hub

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

dia of shaft $T_{mean} = \frac{P \times 60}{2\pi N} \Rightarrow \frac{7292 \times 60}{2\pi \times 210}$

$\boxed{T_{mean} = 331.5 \text{ N-m}}$

Assume

maximum torque = 2 mean torque

$T_{max} = 2 T_{mean}$

$= 2 \times 331.5 \text{ N-m}$

$= 663 \text{ N-m}$

$\boxed{T_{max} = 663 \times 10^3 \text{ N-mm}}$

Maximum Torque transmitted by the shaft (T_{max})

$$T_{max} = \frac{\pi}{16} \tau d_1^3$$

$$663 \times 10^3 = \frac{\pi}{16} \times 50 \times d_1^3$$

$$d_1^3 = 67.5 \times 10^3 \Rightarrow d_1 = \sqrt[3]{67.5 \times 10^3}$$

$$\therefore \boxed{d_1 = 40.7 \simeq 45 mm}$$

Diameter of Hub

$$d = 2d_1 = 2 \times 45 = \boxed{90 mm}$$

Length of Hub = width of rim(b)

$$\boxed{l = b = 130 mm}$$

4) cross sectional dimensions of the elliptical cast iron arms.

a_1 = major axis.
b_1 - minor axis $= 0.5 a_1$
n = no. of arms = 6

} Assume.

Maximum bending moment, Assume as. cantilever

$$M = \frac{T}{RN} (R - r) = \frac{T}{Dn} (D - d)$$

$$= \frac{663}{1.2 \times 6} (1.2 - 0.09).$$

$$= 102.2 \text{ N-m} = 102\,200 \text{ N-mm}$$

Section modulus of for cross sectional arms.

$$Z = \frac{\pi}{32} b_1 a_1^2 = \frac{\pi}{32} \times 0.5\, a_1 (a_1^2)$$

bending stress $\sigma_t = \frac{M}{Z} \Rightarrow 4 = \frac{M}{Z} = \frac{102\,200}{0.05\, a_1^3}$

$$a_1^3 = \frac{2044 \times 10^3}{4} = \boxed{a_1 = 80 mm}$$

$$b_1 = 0.5 a_1 = 0.5 \times 80 = \boxed{40 mm = b_1}$$

5) Dimensions of key

standard dimensions of rectangular key

$w = 16mm \rightarrow$ width of key

$t = 10mm =$ thickness of key.

Considering shear failure.

$$T_{max} = L w \tau \frac{d_1}{2} \Rightarrow L \times 16 \times 50 \times \frac{45}{2} = 18 \times 10^3 L$$

$$L = \frac{663 \times 10^3}{18 \times 10^3} = 36.8 \simeq 38mm$$

$$\boxed{L = 38mm}$$

∴ Total stress in the which should not be greater than 4MPa

$$V = \frac{\pi D N}{60} = \frac{\pi \times 1.2 \times 210}{60} = 13.2 m/s$$

Total stress in the rim

$$\sigma = \rho v^2 \left[0.75 + \frac{4.935 R}{n^2 . t}\right]$$

$$= 7200 (13.2)^2 \left[0.75 + \frac{4.935 \times 0.6}{6^2 \times 0.065}\right]$$

$$= 1.25 \times 10^6 (0.75 + 1.26)$$

$$= 2.5 \times 10^6 N/m^2$$

$$\sigma = 2.5 \text{ MPa}$$

∴ Since it is less than 4MPa, so design is Safe.

UNIT- V

Bearings:-

Bearings:-

Bearing is a Stationary Machine element Which Supports rotating Shafts (or) axles and its Confines their motion. Naturally, a bearing will be required to offer Minimum frictional resistance to moving parts so as to result in minimum power loss

Classification Of Bearings

a) Based on the type of load acting on Shaft

a) Radial bearing

b) Thurst bearing

Radial bearing:-

The load acts perpendicular to the direction of motion of moving parts.

Thurst bearing:-

The thurst bearing, the pressure act along (or) Parallel to the axis of the Shaft

b) Based On the nature of contact

a) Sliding Contact

b) Rolling Contact bearing

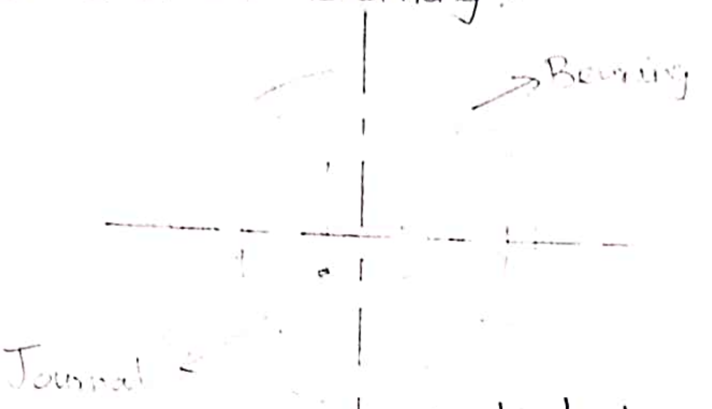
Sliding Contact bearing:-

Depending On the nature of Contact the journal bearing are classified

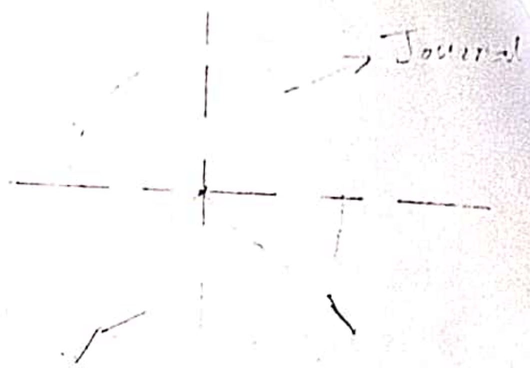
1. Full journal bearing
2. Partial journal bearing
3. Fitted bearing

Rolling

1. Full Journal Bearing:-



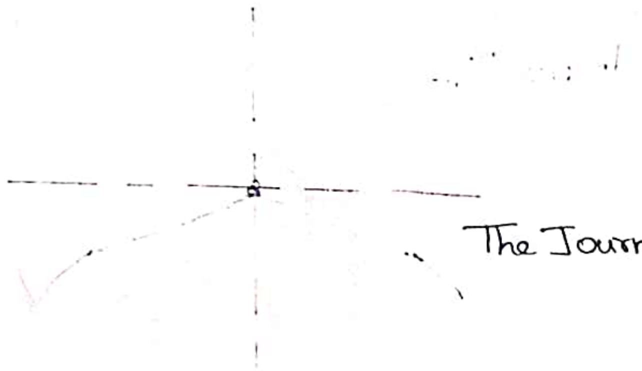
2. Partial bearing:



Contact of
Journal with oil
of 360°

The angle
Varies in
b/w 90° to 180
& 120°

3. Fitted bearing:-



The Journal and bearing are equal

Hydrostatic Bearing

The bearing in which flim of fluid pressure is Obtained by Supplying the lubricant under high pressure Such that the force exerted by Pressure Support the load Shaft at all point is called hydrostatic bearing

Hydrodynamic bearing:-

In hydrodynamic lubrication System, a thin flim of lubrication is Created b/w the Shaft and bearing. The principle of hydrodynamic lubrication in journal bearing

Rolling Contact bearing:-

1. Starting friction is low
2. Lubrication is Simple
3. It requires less axial space & more diameter Space

Sliding Contact bearing:-

1. Starting friction is high
2. Lubrication is Somewhat Complicated
3. It requires more axial Space than diametral space

Sommerfield Number (S)

The Reynold's equation does not have any General Solution. Assuming no side flow, Sommerfield Proposed a Solution and defined a parameter is know as Sommerfield number

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

z' = Absolute Viscosity in N-s/m²

n' = Revolution per Second

P = Bearing Pressure

Operating Pressure:-

The minimum Operating pressure, also know as critical pressure" is the pressure at which the Oil flim breakes down

$$P = \frac{Zn}{4.75\times10^6}\left(\frac{D}{C}\right)^2\left(\frac{L}{D+L}\right) \text{ N/mm}^2$$

$$h_0 = C/4$$

Heat dissipation of bearing:-

$$H_g = \mu W V \quad N\text{-}m/s \text{ (or) } W$$

Where μ = Co-efficient of friction

W = load N = $P \times LD$

L = Length of bearing in m

D = Diameter of bearing in m

P = Pressure on bearing in N/m^2

V = Rubbing Velocity

$$V = \frac{\pi D N}{60} \; m/s$$

Heat dissipated

$$H_d = \frac{(\Delta t + 18)^2 L \times D}{K} \; Kgf/min$$

Δt = temp. rise of bearing

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

t_o = temp. of oil

t_a = ambient temp

Design procedure for Sliding Contact bearing

Step: 1

Calculation of diameter

$$P = \frac{2\pi N M_t}{60} \text{ (or) } M_t = \frac{\pi}{16} \times \tau \times D^3$$

D = ____ mm

Step: 2

Select the Suitable Value of $\frac{L}{D}$ ratio. Determine the length of bearing

[DB Pg. no: 7.31]

Step:3

Calculate the bearing Pressure

$$P = \frac{W}{LD} \quad [\text{DB pg no: 7.31}]$$

Step:4

Select the Clearance and findout clearance ration. [DB pg no: 7.32]

Step:5

Select the Suitable oil and its visocity at Operating temp which preferably with in 60° to 75°c

Step:6

Calculate the bearing characteristic number: $\frac{Zn}{P}$

It should be greater than the min. Value [DB pg 7.31]

Step:7

Determine the Sommerfield number and min. flim thickness [DB pg: 7.40]

$$h_o > D/4$$

Step:8

Calculate the Co-efficient of friction using Petroff equation (or) MacKee's equation

Step:9

Determine the heated generated and heat dissipated (Hd). If the generated heat is more than the dissipated heat.

Solved Problem:- for Sliding Contact bearing:-

Design a journal bearing for Contrifugal pump with the following data, Diameter of the journal = 150mm, Load on bearing = 40KN, Speed of Journal = 900rp

Given data

Diameter of the journal, $D = 150mm$

Load on bearing, $W = 40KN$

Speed $n = 900rpm$

To Find

Design of journal bearing

Solution

Step: 1

Centrifugal pump, $\frac{L}{D}$ = ratio is $\frac{L}{D} = 1 to 2$ [DB pg 7.31]

Let us take $\frac{L}{D} = 1.5$

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

Step: 2

$P = \frac{W}{L \times D} = \frac{4000}{150 \times 225} = 1.185 N/mm^2$ [DB pg 7.31]

This pressure is within the safe limit (0.7 to 1.4 N/m

Step: 3

Selecting of lubricating oil

The Min. Value of $\frac{Zn}{P}$. [DB pg 7.31]

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

$$Z_{min} = \frac{2844.5}{900}$$ [$P = 1.185 N/mm^2 = 11.85 kgf/cm^2$]

$\frac{C}{R} = 0.001$ [radial Clearance]

Radial clearance, $C_1 = R \times 0.001$
$= 50 \times 0.001$
$= 0.05mm$

Diameter clearance $C = 0.05 \times 2$
$= 0.1mm$

$$\frac{D}{C} = \left(\frac{100}{0.1}\right) = 1000$$

Bearing pressure, $P = \frac{W}{LD} = \frac{20000}{150 \times 100}$
$= 1.333\ N/mm^2$

Which is less than allowable Value
[7 to 14 kgf/cm

W.K.T $= \frac{Zn}{P} = 2800$

$$Z = \frac{2800 \times 13.33}{1440}$$

$Z = 25.9$; Say 26

Select SAE 50 (DB 7.41]

From Mckee's equation

Co-efficient of friction, $\mu = \frac{33.25}{10^{10}}\left[\frac{Zn}{P}\right]\left[\frac{D}{C}\right] + K$

$= \frac{33.25}{10^{10}} \times 2800 \times 1000 + 0.002$

$= 0.0113$

Sliding Velocity, $V = \frac{\pi D n}{60} = \frac{\pi \times 0.1 \times 1440}{60}$

$V = 7.54\ m/s$

Heat generated $H_g = \mu \times W \times V$
$= 0.0113 \times 20000 \times 7.54$
$= 1705.5W$

Heat dissipated = $H_d = q\,A\,(T_o - T_a)$

q = energy dissipating Co-efficient

A = Area of radiating Surface ≐ L×D

$H_d = 0.00125 \times 10 \times 15 \times (75-30)$

$= 8.4375 \times 4.186$

$= 35.32\ KJ/min$

$= 588.6\ KJ/s$

Material are rubber (or) Modulated plastic laminate [DB pg 7.30]

Rolling Contact Bearing:-

In rolling Contact bearing, the Contact b/w element is rolling instead of Sliding. The Shaft is Supported on roller or balls. Since the Contact b/w the bearing elements rolling, this type has a Very Small friction and thus, it also Called antifriction bearing

Types Of Rolling Contact Bearing:-

Bearing may be calassified as follow

1. Based on the type of rolling element
   a) Ball bearing
   b) Roller bearing
2. Based on the load to be carried
   a) Radial
   b) Angular Contact
   c) Thrust bearings

Types Of Radial bearing:-

Designed to take up the radial force, they Can also take up axial thurst to Certain extend. Various type of radial and thurst ball bearing.

Deep groove ball bearing:-

It has a deep Continuous raceway all over the Circumference.

Self-Aligning ball bearing:-

These bearing are used Which (or) Where a misalignment b/w axes of Shaft is likely to exist. These are available in two ways

1. Self aligning external
2. Self-aligning Internal

Angular Contact bearing:-

It has a Single row of balls. The Centre Of Contact b/w the ball and races make an angle Called Contact angle. It is mainly used to take up high axial thurst. It has two types

1. One directional, 2. Two directional

Filling notch bearing

These bearing have two notches in inner and outer race. Through this notch, additional ball are inserted Which increase its radial load Capacity. The thurst load Carrying Capacity of this Bearing is reduced

Types of Thrust ball bearing:

Thrust ball bearing are used to carry pure thrust load. There are three type of thrust ball bearing

1. One directional flate race
2. One - directional grooved race
3. Two - directional grooved race.

Bearing life:

The rating of life of a group of identical bearing is define as the number of revolution (or) hours at constant speed that 90% of group of identical will complete

$$\frac{L}{L_{10}} = \left[\frac{\ln\left[\frac{1}{P}\right]}{\ln\left[\frac{1}{P_{10}}\right]} \right]^{1/b}$$

Where L - Required life of bearing in million revolution

L_{10} - Calculated life of selecting bearing, for given load, for 90% survival

P - Probability of survival of 90% or 0.9

P_{10} - Probability of survival

b - is a constant

Load life Relationship:

The relationship b/w the dynamic load carrying capacity (C), the equivalent dynamic load (P) and the bearing life is given by

$$L = \left(\frac{C}{P}\right)^b$$

b = Constant — 3 for ball bearing; 10/3 for Roller bearing

Problem based on roller bearing :-

Problem pg.no. 5.78, Ex 14

Given data

SKF Series 222C

Radial load, $F_r = 4kN = 4\times10^3 N$

Axial load, $F_a = 2kN = 2\times10^3 N$

Life $L = 1000$ hrs

$n = 1000$ rpm

Sol

$$\frac{L_{95}}{L_{90}} = \left[\frac{\ln[1/R_{95}]}{\ln[1/R_{90}]}\right]^{1/b}$$

$\frac{F_a}{F_r} = \frac{2000}{4000} = 0.5$, Assume Serivce factor, $S = 1.2$

Equivalent load, $P = [X F_r + Y F_a] S$

$= [(0.067\times4000) + (4.4\times2000)]1.2$

$\boxed{P = 13776N}$

$\frac{C}{P} = 6.81$ [DB pg 4.7]

$C = 6.81\times13776$

$\boxed{C = 93814.5N}$

$\frac{C}{P} = 7.26$ [DB pg 47]

life $L = 11500$ hrs

Expected life $L = \frac{60 n L_h}{10^6} = \frac{60 \times 1000 \times 11500}{10^6} = 690$ million revolution

$$\frac{L_{95}}{11500} = \left[\frac{\ln\left[\frac{1}{0.95}\right]}{\ln\left[\frac{1}{0.90}\right]}\right]^{1/1.17}$$

$$\frac{L_{95}}{11500} = 0.54 \Rightarrow L_{95} = 0.54 \times 11500$$

$$\boxed{L_{95} = 6215.9 \text{ hrs}}$$

$\frac{C}{P} = 5.94$ [Corresponding 6215.9 hrs and 1000rpm; DB pg 4.7]

$$\frac{C}{P} = 5.94 \Rightarrow \frac{C}{13776} = 5.94$$

$$C = 5.94 \times 13776$$

$$\boxed{C = 81830 N}$$

Verified
[signature] 01.09.22
HOD/Mech