UNIT - I

STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS

MACHINE DESIGN

It is the creation of new and vector machines and improving existing one

Classification of design

* Adaptive design
* Development design
* New design

Methods of New design

1. Rational design
2. Empirical design
3. Industrial design
4. Optimum design
5. System design
6. Element design
7. Computer Aided Design (CAD)

Simples stresses in machine members

Note:

$$\text{Factor of safety} = \frac{\text{Ultimate (or) yield stress}}{\text{Working (or) Maximum stress}}$$

Torsional and Bending stresses in machine parts

Torsional stress

Torsion equation is given by,

$$\frac{T}{J} = \frac{G\theta}{l} = \frac{\tau}{r}$$

where

T — Twisting moment in N-mm

J — Polar moment of Inertia in mm^4

$G \rightarrow$ Modulus of rigidity of the shaft in N/mm^2

$\theta \rightarrow$ Angle of twist in radians

$l \rightarrow$ length of shaft

$\tau \rightarrow$ maximum shear stress in N/mm^2

$r \rightarrow$ radius of the shaft in mm

$d \rightarrow$ diameter of the shaft.

For solid shaft.

(i) Torque $T = \frac{\pi}{16} \tau d^3$

(ii) Polar moment of Inertia $J = \frac{\pi}{32} d^3$

For hollow shaft

(i) Torque $T = \frac{\pi}{16} \tau \left(\frac{d_o^4 - d_i^4}{d_o} \right)$

(ii) Polar moment of Inertia $J = \frac{\pi}{32} (d_o^4 - d_i^4)$

Power transmitted by the shaft

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

where

$P \rightarrow$ Power transmitted in watts

$N \rightarrow$ speed in rpm

$T \rightarrow$ Torque acting on the shaft in N-m

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1. A shaft is transmitting 100 kW at 160 rpm. Find the suitable diameter of the shaft, if the maximum torque transmitted exists the mean by 25%. Take maximum shear stress is 70 MPa

Given data

$P = 100\ kW = 100 \times 10^3\ W$

$N = 160\ rpm$

$\tau_{max} = 70\ MPa = 70\ N/mm^2$

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

W.K.T

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

$$100 \times 10^3 = \frac{2\pi \times 160 \times T_{mean}}{60}$$

$$T_{mean} = 5968 \text{ N-m}$$

given that

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

$$= 1.25 \times 5968$$

$$= 7.46 \times 10^3 \text{ N-m}$$

$$\boxed{T_{max} = 7.46 \times 10^6 \text{ N-mm}}$$

For solid shaft

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

$$7.46 \times 10^6 = \frac{\pi}{16} \times 70 \times d^3$$

$$\boxed{d = 82 \text{mm}}$$

Bending stress in straight Beams

Bending equation is,

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R}$$

$M \rightarrow$ Bending moment in N-mm

$I \rightarrow$ moment of Inertia in mm^4

$\sigma \rightarrow$ Bending stress in N/mm^2

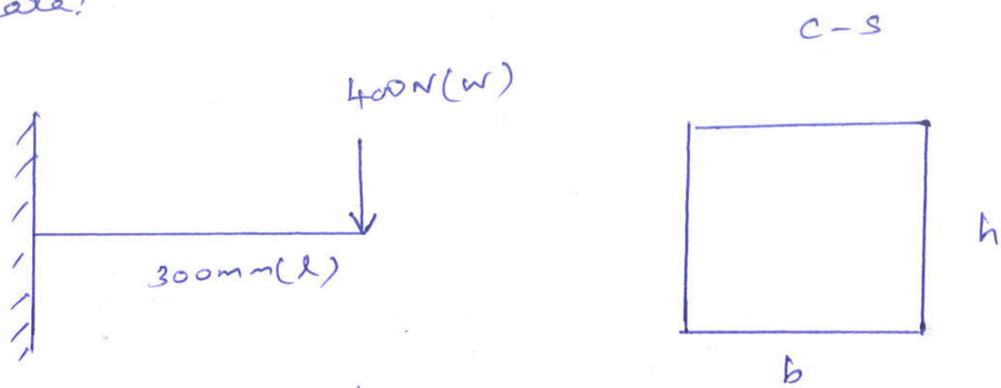
$y \rightarrow$ Distance of fibre from NA in mm

$E \rightarrow$ Youngs modulus in N/mm^2

R → Radius of curvature in mm

(1) A beam of uniform Rectangular section is fixed at one end and carries an electric motor weighs 400 N at a distance of 300 mm from the fixed end. The maximum bending stress in the beam is 40 MPa. Find the width & depth of the beam, if depth is twice that of width.

given data:



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

$h = 2b$

Bending moment: (M)

The given Problem is cantilever with Point load on free end.

From PSGDB P. No 6.4,

$M = Wl = 400 \times 300$

$\boxed{M = 12 \times 10^4 \; N\text{-}mm}$

section Modulus (Z)

The given Section is Rectangular.

∴ From PSGDB P. No 6.1

$Z = \frac{bh^2}{6} = \frac{b \times (2b)^2}{6}$

$\boxed{Z = \frac{2b^3}{3}}$

W.K.T

Bending equation is

$$M = \sigma Z$$

$$12 \times 10^4 = \cancel{\sigma} \; 40 \times \frac{2b^3}{3}$$

$$\boxed{b = 16.51\,mm}$$

given that

$$h = 2b = 2 \times 16.51$$

$$\boxed{h = 33.02\ mm}$$

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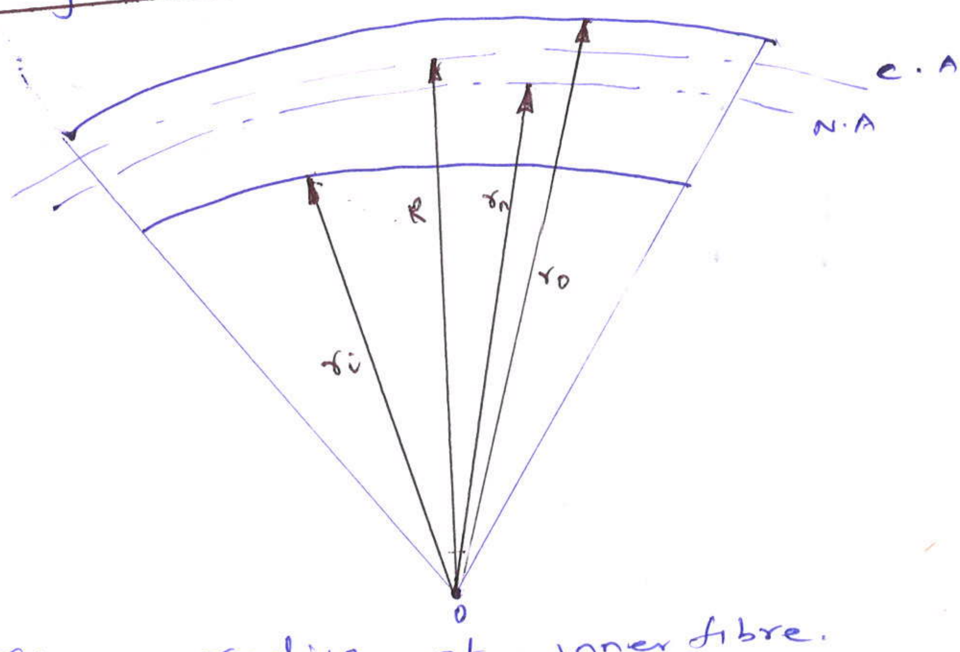
(2) The trunnion of a mixing m/c have an effective length of 30 cm and the weight which comes on each trunnion is 12.5 KN. what should be the diameter of the trunnion, If the stress is not to exceed 35 N/mm²

Ans: Bending moment $M = 3.75 \times 10^6$ N-mm

diameter $d = 103$ mm.

Bending stress in curved beam



r_i → radius at inner fibre.

$r_o \rightarrow$ radius at outer fibre

$r_n \rightarrow$ radius at Neutral AXIS

$R \rightarrow$ radius at centroidal axis

<u>Note:</u>

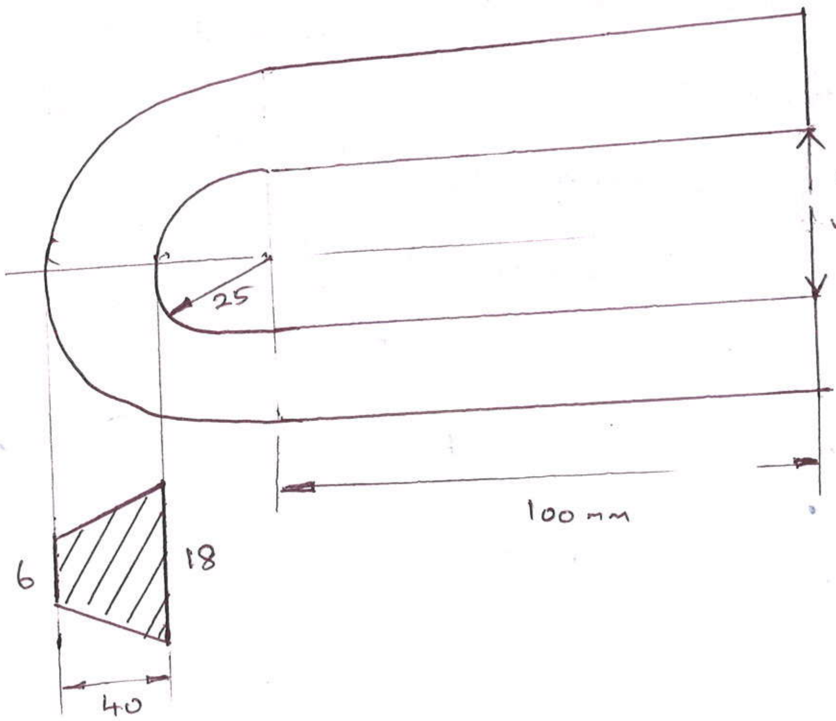
1. Resultant stress $\sigma_R = \sigma_d \pm \sigma_b$

(a) direct stress $\sigma_d = \dfrac{W}{A}$

(b) Bending stress σ_b [P-No: 6.2]

(c) $\pm$ indicates inside & outside fibre respectively.

① The frame of a Punching press is shown in figure. Find the stresses at inner and outer surface of the frame, if load W = 5000N.



<u>given data:</u>

$r_i = 25mm$

$r_o = 25 + 40$

$\boxed{r_o = 65mm}$

$W = 5000 N$

(i) Find r_n

from PSG DB P.No 6.4

For trapezium section

$$r_n = \frac{\frac{1}{2}(b_i + b_o)h}{\left(\frac{b_i r_o - b_o r_i}{h}\right) \ln\left(\frac{r_o}{r_i}\right) - (b_i - b_o)}$$

$$r_n = \frac{\frac{1}{2}(18+6) \times 40}{\left(\frac{18 \times 65 - 6 \times 25}{40}\right) \ln\left(\frac{65}{25}\right) - (18-6)}$$

$$\boxed{r_n = 38.82 \text{ mm}}$$

(ii) R

$$R = r_i + \frac{h(b_i + 2b_o)}{3(b_i + b_o)}$$

$$= 25 + \frac{40(18 + 2 \times 6)}{3(18+6)}$$

$$\boxed{R = 41.67 \text{mm}}$$

(iii) Area of trapezium (A)

$$A = \frac{1}{2} h (b_i + b_o)$$

$$= \frac{1}{2} \times 40 \times (18 + 6)$$

$$\boxed{A = 480 \text{ mm}^2}$$

(iv) eccentricity $e = r_n \sim R$

$$= 41.67 - 38.82$$

$$\boxed{e = 2.85 \text{mm}}$$

(v) direct stress (σ_d)

$$\sigma_d = \frac{W}{A} = \frac{5000}{480}$$

$$\boxed{\sigma_d = 10.42 \text{ N/mm}^2}$$

(vi) Bending moment (M)

$$M = W \times (100 + R)$$

$$= 5000 \times (100 + 41.67)$$

$$\boxed{M = 708.34 \times 10^3 \text{ N-mm}}$$

(vii) Bending stress at inner fibre (σ_{bi})

From PSGDB p.no 6.3

$$\sigma_{bi} = \frac{M\, h_i}{a\, e\, r_i}$$

where $h_i = r_n \sim r_i$

$$= 38.82 - 25$$

$$\boxed{h_i = 13.82 \text{ mm}}$$

$$\sigma_{bi} = \frac{708.34 \times 10^3 \times 13.82}{480 \times 2.85 \times 25}$$

$$\boxed{\sigma_{bi} = 286 \text{ N/mm}^2}$$

(viii) Bending stress at outer fibre (σ_{bo})

From PSGDB p.no 6.2

$$\sigma_{bo} = \frac{M\, h_o}{a\, e\, r_o}$$

where $h_o = r_n \sim r_o = 65 - 38.82$

$$\boxed{h_o = 26.18 \text{ mm}}$$

$$\sigma_{b_o} = \frac{708.34 \times 10^3 \times 26.18}{480 \times 2.85 \times 65}$$

$$\boxed{\sigma_{b_o} = 209 \text{ N/mm}^2}$$

(ix) <u>Resultant stress</u> (σ_R)

$$\sigma_{R_i} = \sigma_d + \sigma_{b_i}$$

$$= 10.42 + 286$$

$$\boxed{\sigma_{R_i} = 296.42 \text{ N/mm}^2}$$

$$\sigma_{R_o} = \sigma_d - \sigma_{b_o}$$

$$= 10.42 - 209.$$

$$\boxed{\sigma_{R_o} = -198.58 \text{ N/mm}^2}$$

② A crane hook carries a load of 20 kN as shown in figure. The section of the hook is rectangular whose horizontal side is 100 mm. Find the stresses at inner and outer fibre.

solution

$r_n = 91.02$ mm

$R = 100$ mm

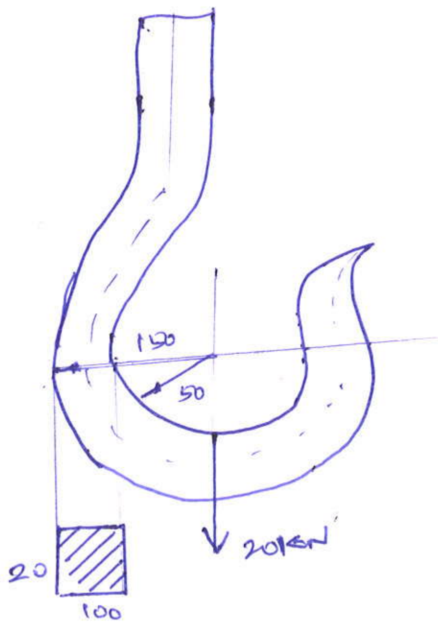
$\sigma_d = 10$ N/mm²

$\sigma_{b_i} = 91$ N/mm²

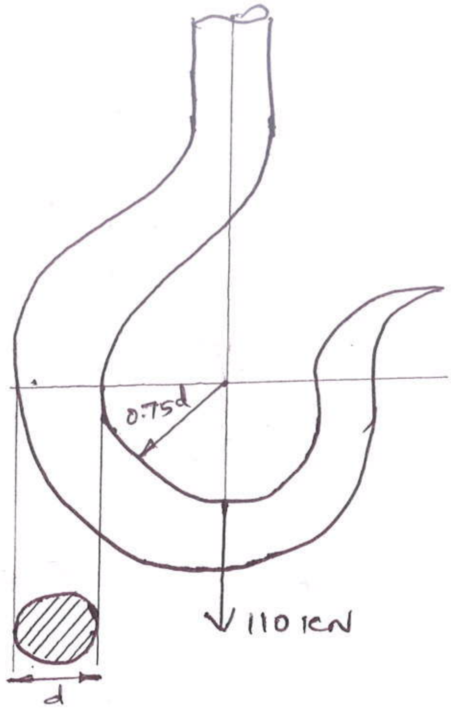
$\sigma_{b_o} = 44$ N/mm²

$\sigma_{R_i} = 101$ N/mm²

$\sigma_{R_o} = -34$ N/mm²



(3) A crane hook has a circular C-S as shown in figure. made of plain carbon steel 55C8 ($\sigma_y = 420\ N/mm^2$). The load acting on the hook is 110 kN. calculate the dimensions of the hook, if the factor of safety as 8.



given data:

$r_i = 0.75d$

$r_o = d + 0.75d$

$\boxed{r_o = 1.75d}$

$W = 110\ kN = 110 \times 10^3\ N$

$F.O.S = 8$

$\sigma_y = 420\ N/mm^2$

W.K.T

$$F.O.S = \frac{\sigma_y}{\sigma_{max}}$$

$$8 = \frac{420}{\sigma_{max}}$$

$$\sigma_{max} = \boxed{\sigma_{R_i} = 52.5\ N/mm^2}$$

(i) r_n

From PSGDB P. No 6.3

For circular section.

$$r_n = \frac{(\sqrt{r_o} + \sqrt{r_i})^2}{4}$$

$$= \frac{[(\sqrt{r_o})^2 + (\sqrt{r_i})^2 + 2\sqrt{r_o r_i}]}{4}$$

$$= \frac{1.75d + 0.75d + 2\sqrt{1.75d \times 0.75d}}{4}$$

$$\boxed{r_n = 1.2d}$$

(ii) R

From PSGDB P.No 6.3

$$R = r_i + \frac{d}{2}$$

$$= 0.75d + 0.5d$$

$$\boxed{R = 1.25d}$$

(iii) Area (A)

$$\boxed{A = \frac{\pi}{4} d^2}$$

(iv) eccentricity (e)

$$e = r_n \sim R$$

$$= 1.25d - 1.2d$$

$$\boxed{e = 0.05d}$$

(v) direct stress (σ_d)

$$\sigma_d = \frac{W}{A} = \frac{110 \times 10^3}{\frac{\pi}{4} d^2}$$

$$\boxed{\sigma_d = \frac{140 \times 10^3}{d^2}}$$

(vi) Bending moment (M)

$$M = W \times R = 110 \times 10^3 \times 1.25d$$

$$\boxed{M = 137.5 \times 10^3 d}$$

(vii) Bending stress at inner fibre (σ_{bi})

From PSGDB P.No 6.2

For inner fibre.

$$\sigma_{b_i} = \frac{M \, h_i}{a \, e \, r_i}$$

where $h_i = r_n \sim r_i$

$= 1.2d - 0.75d$

$$\boxed{h_i = 0.45d}$$

$$\sigma_{b_i} = \frac{137.5 \times 10^3 d \times 0.45d}{\frac{\pi}{4} d^2 \times 0.05d \times 0.75d}$$

$$\boxed{\sigma_{b_i} = \frac{2.105 \times 10^6}{d^2}$$}

(viii) diameter of the hook. (d)

W.K.T

$$\sigma_{R_i} = \sigma_d + \sigma_{b_i}$$

$$52.5 = \frac{140 \times 10^3}{d^2} + \frac{2.105 \times 10^6}{d^2}$$

$$\boxed{d = 207 \text{mm}}$$

Principal stresses for various load combinations

From PSGDB P.No 7.2

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

where $\sigma_1 \rightarrow$ maximum Principal stress

$\sigma_2 \rightarrow$ Minimum Principal stress

Maximum shear stress

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

Types of load

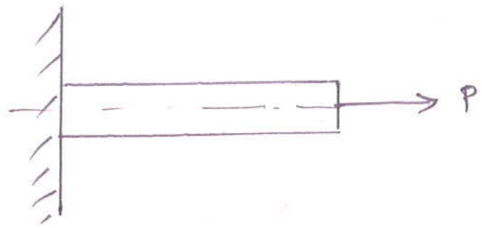
(i) Axial load

(ii) Bending load

(iii) Torsional load

(i) Axial load

* In cantilever beam Axial load only ~~only~~ acting along x direction

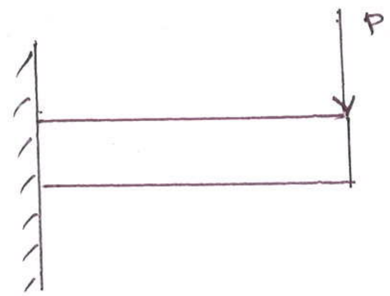


* $\sigma_x = \sigma_{xa} = \frac{P}{A}$

& $\sigma_y , \tau_{xy} = 0$

(ii) Bending load

* Bending load only acting at free end.



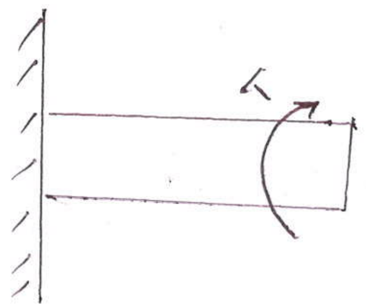
* $\sigma_x = \sigma_{xb} = \frac{M}{Z}$

& $\sigma_y , \tau_{xy} = 0$

* σ_{xb} is +ve → stress at top portion.
  σ_{xb} is −ve → stress at bottom portion.

(iii) Torsional load

* Twisting load only acting on the beam



* $\tau_{xy} = \frac{16\ T}{\pi d^3}$

& $\sigma_x , \sigma_y = 0$

① 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 stresses in the machine member.

Given data:

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

$\sigma_y = 7\ MPa = 7\ N/mm^2$

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

From PSGDB P. NO 7.2

(i) Maximum Principal stress (σ_1)

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

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

$$\boxed{\sigma_1 = 21.13\ N/mm^2}$$

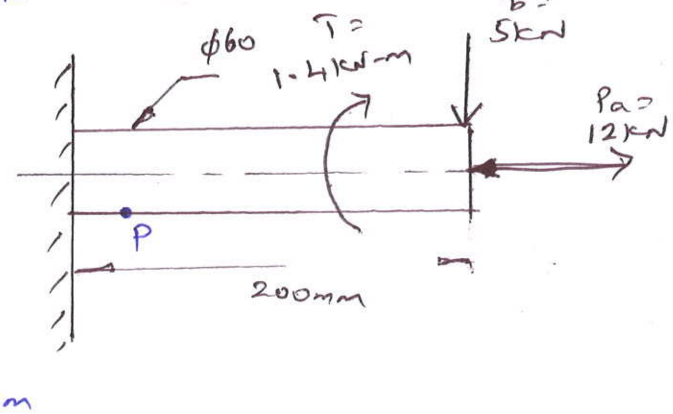
(ii) Minimum Principal stress (σ_2)

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

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

$$\boxed{\sigma_2 = 5.868\ N/mm^2}$$

(2) A. cylindrical bar 60mm diameter and 200mm long is fixed at one end at the free end is loaded as shown in fig. With an axial load of 12 KN, a downward transverse load of 5 KN and a Torque of 1.4 KN-m. calculate the maximum stress



given data:

$P_a = 12\ KN = 12\times 10^3\ N$

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

$T = 1.4\ KN\text{-}m = 1.4\times 10^6\ N\text{-}mm$

$d = 60\ mm$

$l = 200 mm$

Area $A = \frac{\pi}{4} d^2 = \frac{\pi}{4} \times 60^2$

$$\boxed{A = 2827 \ mm^2}$$

(i) Axial load acting on beam.

$$\sigma_{xa} = \frac{P_a}{A}$$

$$= \frac{12 \times 10^3}{2827}$$

$$\boxed{\sigma_{xa} = 4.21 \ N/mm^2}$$

(ii) Bending load acting on beam.

$$\sigma_{xb} = \frac{M}{Z}$$

From PSGDB P.no 6.4

$$M = P_b \times l = 5 \times 10^3 \times 200$$

$$\boxed{M = 1 \times 10^6 \ N\text{-}mm}$$

From PSGDB P. no 6.1

For circular section.

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

$$= \frac{\pi}{32} \times 60^3$$

$$\boxed{Z = 21.20 \times 10^3 \ mm^3}$$

$$\sigma_{xb} = \frac{M}{Z} = \frac{1 \times 10^6}{21.2 \times 10^3}$$

$$\boxed{\sigma_{xb} = 47.16 \ N/mm^2}$$

(iii) Twisting load acting on the beam

W.K.T

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

$$1.4 \times 10^6 = \frac{\pi}{16} \times \tau_{xy} \times 60^3$$

$$\boxed{\tau_{xy} = 33 \text{ N/mm}^2}$$

(i) Maximum Principal stress at point 'P'

Net stress along x direction $\sigma_x = \sigma_{xa} + \sigma_{xb}$

$$\sigma_x = 4.21 + (-47.16)$$

$$\boxed{\sigma_x = -43 \text{ N/mm}^2}$$

[stress is at bottom portion ∴ σ_{xb} (-ve)]

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

$$= \frac{1}{2}\left[(-43+0) + \sqrt{(-43-0)^2 + 4 \times 33^2}\right]$$

$$\boxed{\sigma_1 = 18.28 \text{ N/mm}^2}$$

(ii) Minimum Principal stress at point 'P'

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

$$= \frac{1}{2}\left[(-43+0) - \sqrt{(-43-0)^2 + 4 \times 33^2}\right]$$

$$\boxed{\sigma_2 = -60.45 \text{ N/mm}^2}$$

~~Week 5~~

THEORIES OF FAILURE

From PSGDB P.No 7.3

1. Maximum stress theory (Rankine's theory)

σ_1 (or) σ_2 (or) σ_3 which ever is maximum $= \sigma_{max}$

2. maximum shear theory (Guest's (or) columb's theory)

$$\max\left[(\sigma_1 - \sigma_2) \text{ (or) } (\sigma_2 - \sigma_3) \text{ or } (\sigma_3 - \sigma_1)\right] = \sigma_{max}$$

3. Maximum strain theory (St. Venants theory)

$$\max\left[\sigma_1 - \gamma(\sigma_2 + \sigma_3),\ \sigma_2 - \gamma(\sigma_3 + \sigma_1),\ \sigma_3 + \gamma(\sigma_1 + \sigma_2)\right] = \sigma_{max}$$

4. Maximum strain energy theory

$$\left[\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\gamma(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1)\right] = \sigma_{max}^2$$

5. octahedral (or) Distortion energy theory.

$$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_3\sigma_1 = \sigma_{max}^2$$

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(1) A bolt is subjected to an axial force of 10000 N with a transverse shear force of 5000 N. to be required Find the diameter of the bolt according to (i) maximum principal stress theory (ii) maximum shear stress theory (iii) Maximum principal strain theory (iv) maximum strain energy theory (v) maximum distortion energy theory. Assume permissible tensile stress at elastic limit $= 100\ N/mm^2$ & $\gamma = 0.3$

$P_a = 10000\ N$

$P_s = 5000\ N$

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

$\gamma = 0.3$

solution

$$\sigma_{xa} = \frac{P_a}{A} = \frac{10000}{\frac{\pi}{4} \times d^2}$$

$$\boxed{\sigma_{xa} = \frac{12732}{d^2}}$$

$$\tau_{xy} = \frac{P_s}{A} = \frac{5000}{\frac{\pi}{4} d^2}$$

$$\boxed{\tau_{xy} = \frac{6369}{d^2}}$$

From PSGDB P. no 7.2.

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

$$= \frac{1}{2}\left[\left(\frac{12732}{d^2} + 0\right) + \sqrt{\left(\frac{12732}{d^2} - 0\right)^2 + 4 \times \left(\frac{6369}{d^2}\right)^2}\right]$$

$$\boxed{\sigma_1 = \frac{15372}{d^2}}$$

III^ly

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

$$= \frac{1}{2}\left[\left(\frac{12732}{d^2} + 0\right) - \sqrt{\left(\frac{12732}{d^2} - 0\right)^2 + 4 \times \left(\frac{6369}{d^2}\right)^2}\right]$$

$$\boxed{\sigma_2 = \frac{-2639}{d^2}}$$

(i) Maximum Principal stress theory:

From PSGDB p.no 7.3

$$\text{Max of } [\sigma_1, \sigma_2, \sigma_3] = \sigma_{max}$$

$$\text{Max}\left[\frac{15372}{d^2}, \frac{-2639}{d^2}, 0\right] = 100$$

$$\frac{15372}{d^2} = 100$$

$$\boxed{d = 12.4\,mm}$$

(ii) Maximum shear stress theory

$$\text{Max}\left[(\sigma_1 - \sigma_2), (\sigma_2 - \sigma_3), (\sigma_3 - \sigma_1)\right] = \sigma_{max}$$

$$\sigma_1 - \sigma_2 = \frac{15372}{d^2} - \left(-\frac{2639}{d^2}\right)$$

$$= \frac{18011}{d^2}$$

$$\sigma_2 - \sigma_3 = \frac{-2639}{d^2} - 0$$

$$= -\frac{2639}{d^2}$$

$$\sigma_3 - \sigma_1 = 0 - \left(\frac{15372}{d^2}\right)$$

$$= -\frac{15372}{d^2}$$

$$\text{Max}\left[\frac{18011}{d^2}, -\frac{2639}{d^2}, -\frac{15372}{d^2}\right] = \sigma_{max}$$

$$\frac{18011}{d^2} = 100$$

$$\boxed{d = 13.42\ mm}$$

(iii) Maximum Principal strain theory

$$Max\left[\sigma_1 - \nu(\sigma_2+\sigma_3),\ \sigma_2 - \nu(\sigma_3+\sigma_1),\ \sigma_3 - \nu(\sigma_1+\sigma_2)\right] = \sigma_{max}$$

$$\sigma_1 - \nu(\sigma_2+\sigma_3) = \frac{15372}{d^2} - 0.3\left[-\frac{2639}{d^2} + 0\right]$$

$$= \frac{16163}{d^2}$$

$$\sigma_2 - \nu(\sigma_3+\sigma_1) = \frac{-2639}{d^2} - 0.3\left[0 + \frac{15372}{d^2}\right]$$

$$= \frac{-7251}{d^2}$$

$$\sigma_3 - \nu(\sigma_1+\sigma_2) = 0 - 0.3\left[\frac{15372}{d^2} + \left(\frac{-2639}{d^2}\right)\right]$$

$$= \frac{-3820}{d^2}$$

$$Max\ of\left[\frac{16163}{d^2},\ \frac{-7251}{d^2},\ -\frac{3820}{d^2}\right] = \sigma_{max}$$

$$\frac{16163}{d^2} = 100$$

$$\boxed{d = 12.71\ mm}$$

(IV) Maximum strain energy theory

$$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\nu\left[\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1\right] = \sigma_{max}^2$$

$$\left[\frac{15372}{d^2}\right]^2 + \left[-\frac{2639}{d^2}\right]^2 - 2\times 0.3\left[\left[\frac{15372}{d^2}\times\frac{-2639}{d^2}\right] + \left[\frac{-2639}{d^2}\times 0\right] + \left[0\times\frac{15372}{d^2}\right]\right] = 100^2$$

$$\frac{267.59\times 10^6}{d^4} = 10\times 10^3$$

$$\boxed{d = 12.79\,mm}$$

(V) Maximum distortion energy theory.

$$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_3\sigma_1 = \sigma_{max}^2$$

$$\left(\frac{15372}{d^2}\right)^2 + \left(\frac{-2639}{d^2}\right) + 0 - \left(\frac{15372}{d^2}\right)\left(\frac{-2639}{d^2}\right) - 0 - 0 = 100^2$$

$$\frac{298.818\times 10^6}{d^4} = 10\times 10^3$$

$$\boxed{d = 12.98\,mm}$$

(2) A bolt is subjected to a tensile load of 25 kN and shear load of 10 kN. Determine the dia of the bolt, according to (i) maximum principal stress theory (ii) Maximum principal strain theory (iii) maximum shear stress theory. Assume F.O.S = 2.5, yield stress in tension = 300 N/mm^2; $\nu = 0.25$

Solution

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

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

(i) Maximum Principal stress, $\sigma_1 = \frac{36296}{d^2}$

(ii) Minimum Principal stress $\sigma_2 = \frac{-4466}{d^2}$

Maximum Principal stress theory $d = 17.39$ mm

Maximum Shear stress theory $d = 18.43$ mm

Maximum Principal strain theory $d = 17.66$ mm

3 A cylindrical shaft made of steel yield strength 700 MPa is subjected to static loads consisting of bending moment 10 kN-m and Torsional moment 30 kN-m. Determine the diameter of the shaft using any two theories of failure Assume $\nu = 0.25$, F.O.S = 2, E = 210 GPa

Solution:

$$\sigma_x = \frac{101.86 \times 10^6}{d^3}$$

$$\tau_{xy} = \frac{152.7 \times 10^6}{d^3}$$

$$\sigma_1 = \frac{206.58 \times 10^6}{d^3}$$

$$\sigma_2 = \frac{-104.72 \times 10^6}{d^3}$$

Maximum Principal stress theory $d = 83.88$ mm

Maximum shear stress theory $d = 96.17$ mm.

STRESS CONCENTRATION

* It is defined as the formation of stresses where the rapid change in cross-section or discontinuity in a section.

Stress concentration factor (K_t)

From PSGDB P.No 7.8

$$K_t = \frac{\text{Maximum stress}}{\text{Nominal stress}} = \frac{\sigma_{max}}{\sigma_{nominal}}$$

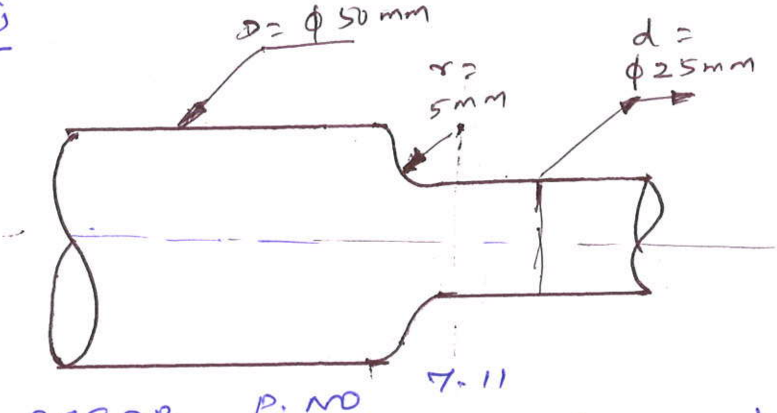
---

(1) A 20 kN tensile load acts on the following members considering the stress concentration. Calculate the maximum stress induced in each member. (i) A stepped shaft of diameters 50mm and 25mm with fillet radius of 5mm (ii) A rectangular plate width 60mm and 10mm thick with a transverse hole of 12mm diameter at the centre.

Solution:

$P = 20\,kN = 20 \times 10^3\,N$

Case (i)



From PSGDB P.NO 7.11

$D = 50\,mm$; $d = 25\,mm$; $r = 5\,mm$

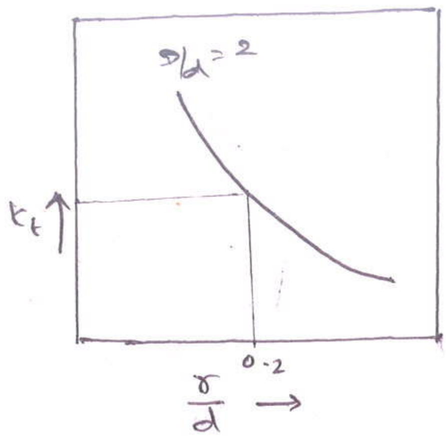
$$\sigma_{nominal} = \frac{P}{A_{min}} = \frac{20 \times 10^3}{\frac{\pi}{4} \times 25^2}$$

$$\boxed{\sigma_{nominal} = 40.74\,N/mm^2}$$

From PSGDB P. no 7.11, From graph

$\frac{r}{d} = \frac{5}{25} = 0.2$; $\frac{D}{d} = \frac{50}{25} = 2$

$\therefore$ $\boxed{K_t = 1.5}$



From PSGDB P. no 7.8

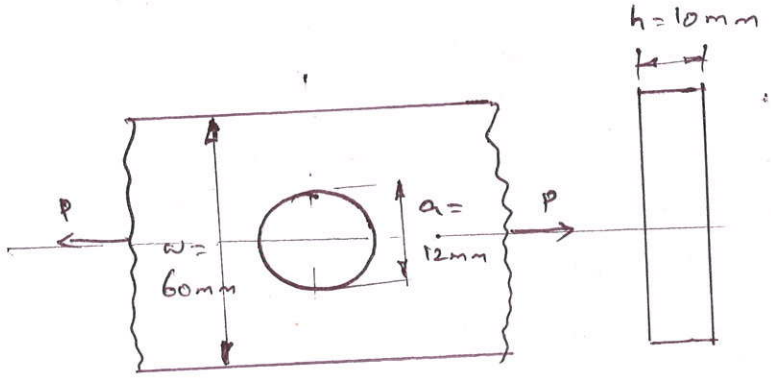
$$K_t = \frac{\sigma_{max}}{\sigma_{nominal}}$$

$$1.5 = \frac{\sigma_{max}}{40.74}$$

$$\boxed{\sigma_{max} = 61.12\ N/mm^2}$$

case (ii)

From PSGDB
P. no 7.10



$\omega = 60$ mm

$a = 12$ mm

$h = 10$ mm

$$\sigma_{nominal} = \frac{P}{(W-a)h}$$

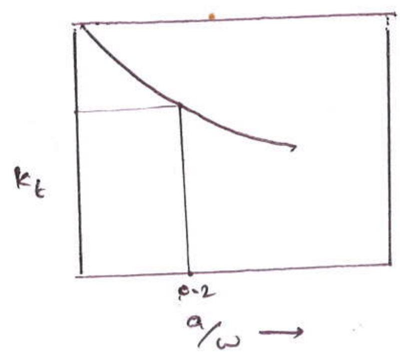
$$= \frac{20 \times 10^3}{(60-12) \times 10}$$

$$\boxed{\sigma_{nom} = 41.67 mm}$$

From PSGDB P. no 7.10 graph

$\frac{a}{\omega} = \frac{12}{60} = 0.2$

$\therefore$ $\boxed{K_t = 2.5}$



W.K.T.

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}}$$

$$2.5 = \frac{\sigma_{max}}{41.67}$$

$$\boxed{\sigma_{max} = 104.18\ N/mm^2}$$

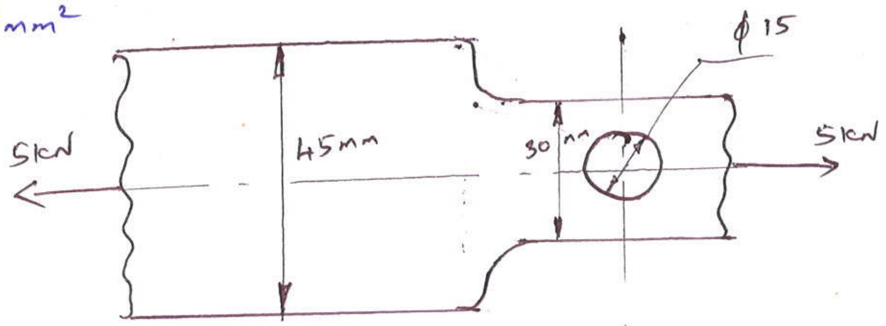
(2) A plate of uniform thickness (t) has two widths of 45mm and 30mm with a fillet radius of 5mm. The smaller width portion has a transverse hole of 15mm diameter. For the plate material, the ultimate tensile strength is 200 N/mm². considering stress concentration effect. Assume F.O.S as 2.5. Find the suitable thickness of the plate, if the max tensile load is 5kN.

<u>given data</u>.

$\sigma_u = 200\ N/mm^2$

F.O.S = 2.5

P = 5 kN
$= 5 \times 10^3\ N$

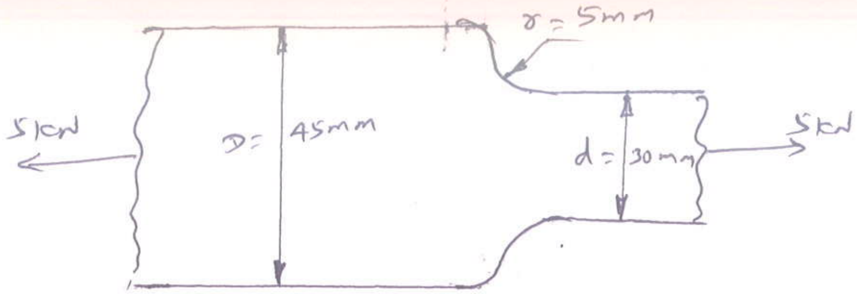


W.K.T $\quad F.O.S = \frac{\sigma_u}{\sigma_{max}}$

$$2.5 = \frac{200}{\sigma_{max}}$$

$$\boxed{\sigma_{max} = 80\ N/mm^2}$$

case (ii) stepped plate without transverse hole.



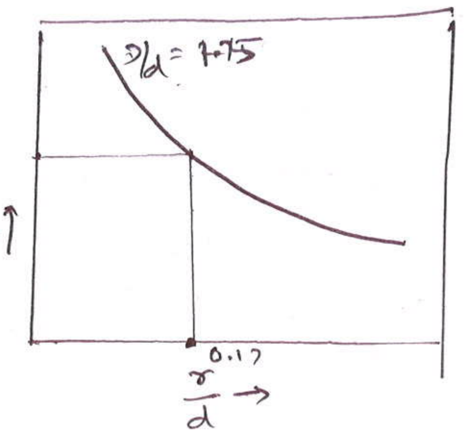
From PSGDB P.no 7.9

$D = 45mm$; $d = 30\ mm$; $r = 5mm$

$\frac{r}{d} = \frac{5}{30} = 0.17$; $\frac{D}{d} = \frac{45}{30} = 1.5$

From PSGDB P.no 7.9, graph

$\frac{r}{d} = 0.17$; $\frac{D}{d} = 1.5$



$\therefore \boxed{K_t = 1.8}$

$$\sigma_{nominal} = \frac{P}{A_{min}} = \frac{P}{d \times t}$$

$$= \frac{5 \times 10^3}{30 \times t}$$

$$\boxed{\sigma_{nom} = \frac{166.67}{t}}$$

W.K.T

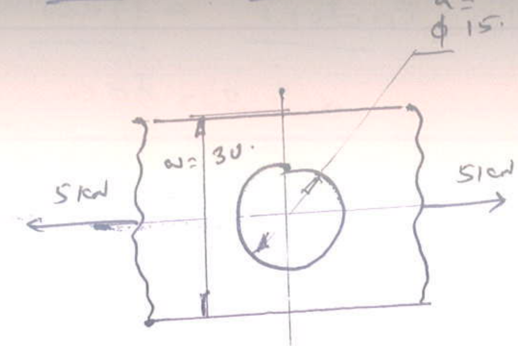
$$K_t = \frac{\sigma_{max}}{\sigma_{nom}}$$

$$1.8 = \frac{80}{\left(\frac{166.67}{t}\right)}$$

$$\boxed{t = 3.75mm}$$

Case (ii) consider smaller width portion with hole

From PSGDB P.no 7.10



$$\sigma_{nom} = \frac{P}{(w-a)t}$$

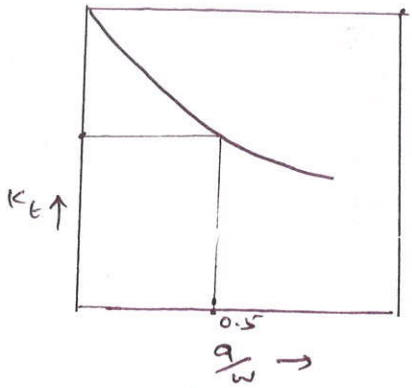
$$= \frac{5 \times 10^3}{(30-15)t}$$

$$\boxed{\sigma_{nom} = \frac{333.33}{t}}$$

From PSGDB P.no 7.10 Graph

$$\frac{a}{w} = \frac{15}{30} = 0.5$$

$$\therefore \boxed{K_t = 2.16}$$



W.K.T $K_t = \frac{\sigma_{max}}{\sigma_{nom}}$

$$2.16 = \frac{80}{\left(\frac{333.33}{t}\right)}$$

$$\boxed{t = 9\,mm}$$

$\therefore$ Suitable thickness of the plate $\boxed{t = 9\,mm}$

DESIGN FOR VARIABLE LOADING

Terms used in variable stress condition

From PSGDB P. no 7.6

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

(ii) Stress amplitude $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$

(iii) Fatigue stress concentration factor $K_f = 1 + q(K_t - 1)$

where $q \rightarrow$ notch sensitivity factor

$K_t \rightarrow$ stress concentration factor

If K_t value is not given in the problem, assume $(K_t = 1)$

(IV) Factors affecting the endurance strength

(a) surface finish factor (K_{SUR})

(b) size factor (K_{SZ})

(c) Load factor (K_L)

(V) Relationship between modified endurance strength to the actual endurance strength

$$\sigma_{-1} = K_{SUR} \times K_{SZ} \times K_L \times (\sigma_{-1})_{actual}$$

(1) A m/c component is subjected to a flexural stress which fluctuates between 300 MN/m² & −150 MN/m². Determine the value ultimate strength of m/c component according to (i) modified goodman relation (ii) soderberg relation. Take yield strength = 0.55 times ultimate strength. Take $\sigma_{-1} = 0.5\,\sigma_u$ & F.O.S = 2

Given data:

$\sigma_{max} = 300\,MN/m^2 = 300\,N/mm^2$

$\sigma_{min} = -150\,MN/m^2 = -150\,N/mm^2$

$n = 2$

$\sigma_y = 0.55\,\sigma_u$

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

From PSG DB P. No 7.6

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{300 + (-150)}{2}$$

$$\boxed{\sigma_m = 75\,N/mm^2}$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{300 - (-150)}{2}$$

$$\boxed{\sigma_a = 225\,N/mm^2}$$

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

K_t is not given $\therefore$ $\boxed{K_t = 1}$

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

$$\boxed{K_f = 1}$$

(i) Modified goodman equation:

From PSGDB P.NO 7.6

$$\frac{1}{n} = K_t \left[\frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma_{-1}} \right]$$

$$\frac{1}{2} = 1 \left[\frac{75}{\sigma_u} + \frac{225}{0.5\,\sigma_u} \right]$$

$$\boxed{\sigma_u = 1050 \ N/mm^2}$$

(ii) soderberg equation:

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

$$\frac{1}{2} = \left[\frac{75}{0.55\,\sigma_u} + 1 \times \frac{225}{0.5\,\sigma_u} \right]$$

$$\boxed{\sigma_u = 1172 \ N/mm^2}$$

(2) A hot rolled steel shaft of 40mm diameter is subjected to a Torsional moment that varies from 330 Nm to −110 N-m and applied Bending moment which rises from 440 N-m to −220 N-m. The material of the shaft has an ultimate strength of 550 MN/m² and yield strength of 410 MN/m². Find the approximate F.O.S using soderberg equation. Allowing $\sigma_{-1} = 0.5\,\sigma_u$, $K_{SZ} = 0.85$, $K_{SUR} = 0.62$.

Given data

$d = 40 mm$

$T_{max} = 330 N-m = 330 \times 10^3 N-mm$

$T_{min} = -110 N-m = -110 \times 10^3 N-mm$

$M_{max} = 440 N-m = 440 \times 10^3 N-mm$

$M_{min} = 220 N-m = 220 \times 10^3 N-mm$

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

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

$\sigma_{-1} = 0.5 \sigma_u \Rightarrow (\sigma_{-1})_{actual} = 0.5 \times 550$

$$\boxed{(\sigma_{-1})_{actual} = 275 N/mm^2}$$

$K_{sz} = 0.85$

$K_{SUR} = 0.62$

Solution:

Torsion

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

$$330 \times 10^3 = \frac{\pi}{16} \times \tau_{max} \times 40^3$$

$$\boxed{\tau_{max} = 26 N/mm^2}$$

$$T_{min} = \frac{\pi}{16} \tau_{min} d^3$$

$$-110 \times 10^3 = \frac{\pi}{16} \times \tau_{min} \times 40^3$$

$$\boxed{\tau_{min} = -9 N/mm^2}$$

From PSG DB P.No 7.6

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

$$\tau_m = \frac{26 + (-9)}{2}$$

$$\boxed{\tau_m = 8.5 \ N/mm^2}$$

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

$$= \frac{26 - (-9)}{2}$$

$$\boxed{\tau_a = 17.5 \ N/mm^2}$$

W.K.T

$$M_{max} = \sigma_{max} \cdot Z$$

[PSGDB 6.1
For circular section
$Z = \frac{\pi}{32} d^3$]

$$440 \times 10^3 = \sigma_{max} \times \frac{\pi}{32} \times 40^3$$

$$\boxed{\sigma_{max} = 70 \ N/mm^2}$$

$$M_{min} = \sigma_{min} \ Z$$

$$-220 \times 10^3 = \sigma_{min} \times \frac{\pi}{32} \times 40^3$$

$$\boxed{\sigma_{min} = -35 \ N/mm^2}$$

From PSGDB P. NO 7.6

mean stress $\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{70 + (-35)}{2}$

$$\boxed{\sigma_m = 17.5 \ N/mm^2}$$

stress amplitude $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{70 - (-35)}{2}$

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

$$(\sigma_m)_{eq} = \sqrt{\sigma_m^2 + 3\tau_m^2} = \sqrt{17.5^2 + 3 \times 8.5^2}$$

$$\boxed{(\sigma_m)_{eq} = 22.9 \ N/mm^2}$$

$$(\sigma_a)_{eq} = \sqrt{\sigma_a^2 + 3\tau_a^2} = \sqrt{(52.5)^2 + 3 \times (17.5)^2}$$

$$\boxed{(\sigma_a)_{eq} = 60.6 \ N/mm^2}$$

modified endurance strength $\sigma_{-1} = K_{sz} \times K_{sur} \ (\sigma_{-1})_{adm}$

$$\sigma_1 = 0.85 \times 0.62 \times 275$$

$$\boxed{\sigma_{-1} = 145 \ N/mm^2}$$

From PSGDB 7.6, soderberg equation

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

$$\frac{1}{n} = \frac{22.9}{410} + 1 \times \frac{60.6}{145}$$

$$\boxed{n = 2.1}$$

2) The pulley is keyed to a shaft midway between two antifriction bearing, the bending moment at the pulley varies from -150 N.m to 450 N.m and Twisting moment in the shaft varies from 50 N.m to 200 N.m. The shaft must be made from cold drawn steel having ultimate strength of 540 N/mm², yield strength of 400 N/mm² in reversed bending. Calculate the required diameter of the shaft in the key slote. Take K_t (stress concentration factor) = 2.3, q (notch sensitivity factor) = 0.9, size factor = 0.88, Factor of safety, n = 1.8 surface finish factor = 0.85

Given Data:

Maximum bending moment, M_{max} = 450 Nm

$\Rightarrow M_{max} = 450 \times 10^3$ N.mm

Minimum bending moment, M_{min} = -150 N.m

$\Rightarrow M_{min} = -150 \times 10^3$ N.mm

Twisting (or) Torsional moment, at maximum } T_{max} = 200 N.m

$\Rightarrow T_{min} = 50 \times 10^{?}$ N·mm

Ultimate tensile stress, $\sigma_U = 540\ N/mm^2$

Yield strength, $\sigma_{yield} = 400\ N/mm^2$

Stress concentration factor, $K_t = 2.3$

Notch sensitivity factor, $q = 0.9$

Size factor, $K_{sz} = 0.88$

Surface finish factor, $K_{sur} = 0.85$

Factor of safety, $(n) = 1.8$

To find :

Diameter of the shaft, $D = ?$

8

Soln :

Bending moment

$$M_{max} = 450 \times 10^3$$

$$\boxed{\text{Bending moment} \quad (\because M = \sigma . z)}$$

$$\sigma_{max} . Z = 450 \times 10^3$$

From PSG Data book, Pg. No. : 6.1,

For circular cross section, $\boxed{Z = \frac{\pi}{32} . d^3}$

$$\sigma_{max} . \left(\frac{\pi}{32}\right) d^3 = 450 \times 10^3$$

$$\sigma_{max} \left(\frac{\pi}{32}\right) (d^3) = 450 \times 10^3$$

$$\sigma_{max} = \frac{(450 \times 10^3)(32)}{(\pi)(d^3)}$$

Maximum bending stress,

$$\Rightarrow \sigma_{max} = \frac{4.6 \times 10^6}{d^3}$$

Similarly,

$$M_{min} = -150 \times 10^3$$

$$\sigma_{min} \cdot Z = -150 \times 10^3$$

$$\sigma_{min} \left(\frac{\pi}{32} \cdot d^3\right) = -150 \times 10^3$$

$$\sigma_{min} = \frac{(-150 \times 10^3)(32)}{(\pi)(d^3)}$$

Minimum bending stress,

$$\Rightarrow \sigma_{min} = \frac{-1.5 \times 10^6}{d^3}$$

From PSG Data book, Pg. No.: 7.6

i) σ Mean stress, $\boxed{\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}}$

$$\sigma_m = \frac{\left(\frac{4.6 \times 10^6}{d^3}\right) + \left(\frac{-1.5 \times 10^6}{d^3}\right)}{2}$$

$$\Rightarrow \sigma_m = \frac{1.55 \times 10^6}{d^3}$$

$$\frac{\left(\quad \right) \left(\quad d^3 \quad \right)}{2}$$

$$\Rightarrow \sigma_a = \frac{3.05 \times 10^6}{d^3}$$

From PSG Data book, Pg. No. : 7.6

Stress concentration factor,

$$\boxed{K_f = 1 + q\,(K_t - 1)}$$

$$K_f = 1 + 0.9\,(2.3 - 1)$$

$$\Rightarrow K_f = 2.17$$

From PSG Data book, Pg. No. : 1.42

For reversed cycle in bending,

endurance strength, $\boxed{\sigma_{-1} = 0.46\,\sigma_U}$

$$\sigma_{-1} = (0.46)\,(540)$$

$$\Rightarrow \underset{\text{(actual)}}{\sigma_{-1}} = 248\ \text{N/mm}^2$$

Modified endurance strength,

$$\boxed{\sigma_{-1} = K_{sz} \cdot K_{SUR} \cdot (\sigma_{-1})_{actual}}$$

$$\sigma_{-1} = (0.88)\,(0.85)\,(248)$$

$\Rightarrow \quad \sigma_{-1} = 186 \ N/mm^2$

From PSG Data book, Pg. No. : 6.7,
equivalent stress,

$$\boxed{\sigma_{eq} = \sigma_m + K_f \cdot \left(\frac{\sigma_a \cdot \sigma_y}{\sigma_{-1}} \right)}$$

$$\sigma_{eq} = \left(\frac{1.55 \times 10^6}{d^3} \right) + (2.17) \left[\frac{\left(\frac{3.05 \times 10^6}{d^3} \right)(400)}{186} \right]$$

$$\sigma_{eq} = \frac{1.55 \times 10^6}{d^3} + \frac{(2.17)(3.05 \times 10^6)(400)}{(d^3)(186)}$$

$$\Rightarrow \quad \sigma_{eq} = \frac{1.55 \times 10^6}{d^3} + \frac{14.23 \times 10^6}{d^3}$$

Equivalent stress,

$$\Rightarrow \quad \sigma_{eq} = \frac{15.78 \times 10^6}{d^3}$$

Twisting (or) Torsional moment.

$$T_{max} = 200 \times 10^3$$

Torsional equation, $\boxed{T = \frac{\pi}{16} \tau d^3}$

$$\frac{\pi}{16} \cdot \tau_{max} \cdot d^3 = 200 \times 10^3$$

a-

$$\tau_{min} = 50 \times 10^3$$

$$\frac{\pi}{16} \tau_{min} d^3 = 50 \times 10^3$$

$$\Rightarrow \tau_{min} = \frac{(50 \times 10^3)(16)}{(\pi)(d^3)}$$

Minimum shear stress,

$$\Rightarrow \tau_{min} = \frac{0.3 \times 10^6}{d^3}$$

From PSG Data book, Pg. No.: 7.6

Mean shear stress, $\boxed{\tau_m = \frac{\tau_{max} + \tau_{min}}{2}}$

$$\tau_m = \frac{\left(\frac{1 \times 10^6}{d^3}\right) + \left(\frac{0.3 \times 10^6}{d^3}\right)}{2}$$

$$\tau_m = \frac{0.65 \times 10^6}{d^3}$$

Similarly,

~~Amplitude~~ shear stress amplitude,

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

$$\tau_a = \left(\frac{1 \times 10^6}{d^3}\right) - \left(\frac{0.3 \times 10^6}{d^3}\right) \Big/ 2$$

.00)

00)

$$\Rightarrow \tau_a = \frac{0.35 \times 10^6}{d^3}$$

From PSG Data book, Pg. No.: 7.6

For Maximum shear theory,

Yield shear stress, $\boxed{\tau_y = \frac{\sigma_y}{2}}$

$$\Rightarrow \tau_y = \frac{400}{2} \Rightarrow \tau_y = 200 \text{ N/mm}^2$$

From PSG Data book, Pg. No. : 1.42

For Reversed torsional moment,

endurance shear strength,

$$\boxed{\tau_{-1} = 0.22\ \sigma_u}$$

$$\tau_{-1} = (0.22)(540)$$

$$\Rightarrow (\tau_{-1})_{actual} = 119 \text{ N/mm}^2$$

Modified endurance shear strength,

$$\boxed{\tau_{-1} = K_{sz} \cdot K_{sur} \cdot (\tau_{-1})_{actual}}$$

$$\tau_{-1} = (0.88)(0.85)(119)$$

$$\Rightarrow \tau_{-1}' = 89 \text{ N/mm}^2$$

From PSG Data book, Pg. No. 7.6

$$\tau_{eq} = \frac{0.65 \times 10^6}{d^3} + (2.17)\left[\frac{\left(\frac{(0.35 \times 10^6)(200)}{d^3}\right)}{89}\right]$$

$$\tau_{eq} = \frac{0.65 \times 10^6}{d^3} + (2.17)\left(\frac{0.35 \times 10^6}{d^3}\right)\frac{(200)}{(89)}$$

$$\Rightarrow \tau_{eq} = \frac{0.65 \times 10^6}{d^3} + \frac{1.71 \times 10^6}{d^3}$$

$$\Rightarrow \tau_{eq} = \frac{2.36 \times 10^6}{d^3}$$

From PSG Data book, Pg. No. : 7.6

combined stress equation

$$\boxed{\frac{1}{n} = \left[\left(\frac{\sigma_{eq}}{\sigma_y}\right)^2 + \left(\frac{\tau_{eq}}{\tau_y}\right)^2\right]^{1/2}}$$

Squaring on both sides on above equation,

$$\left(\frac{1}{n}\right)^2 = \left[\left(\frac{\sigma_{eq}}{\sigma_y}\right)^2 + \left(\frac{\tau_{eq}}{\tau_y}\right)^2\right]^{2/2}$$

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

$$\left(\frac{1}{1.8}\right)^2 = \left[\frac{\left(\frac{15.78 \times 10^6}{d^3}\right)}{400}\right]^2 + \left[\frac{\left(\frac{2.36 \times 10^6}{d^3}\right)}{200}\right]^2$$

$$\frac{1}{3.24} = \frac{1.56 \times 10^9}{d^6} + \frac{0.14 \times 10^9}{d^6}$$

$$\frac{1}{3.24} = \frac{1.7 \times 10^9}{d^6} \Rightarrow d^6 = (1.7 \times 10^9)(3.24)$$

$$\Rightarrow d^6 = 5.508 \times 10^9$$

$$\Rightarrow d = (5.508 \times 10^9)^{1/6}$$

$$\Rightarrow d = 42 \text{ mm}$$

Diameter of the shaft, $d = 42$ mm.

UNIT - II

DESIGN OF SHAFT AND COUPLINGS

SHAFT:-

* It is a rotating m/c element which transmit power from one point to another point

Types of shaft

* line shaft:
  - It is a shaft which transmits Power to several m/c elements.
* spindle:
  - It is a short revolving shaft.
* stub shaft:
  - A shaft integral with an engine, motor
* counter shaft:
  - A shaft which connects motor to line shaft of m/c

DESIGN OF SHAFT BASED ON STRENGTH

I SHAFT subjected to Twisting moment only

note:

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

2. $$T = \frac{\pi}{16} \tau d^3 \rightarrow \text{solid shaft}$$

$$T = \frac{\pi}{16} \tau \left(\frac{d_o^4 - d_i^4}{d_o} \right) \rightarrow \text{hollow shaft.}$$

(1) Find the diameter of a solid steel shaft to transmit 20 kW at 200 rpm. The ultimate shear stress for the steel may be taken as 360 MPa. and a F.O.S = 8 ; If hollow shaft is to be used in place of solid shaft, Find the outside & inside dia

when the ratio of inside to out side is 0.5

Given data:

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

$N = 200\,rpm$

$\tau_u = 360\,MPa = 360\,N/mm^2$

$F.O.S = 8$

$\frac{d_i}{d_o} = 0.5 \Rightarrow \boxed{d_i = 0.5\,d_o}$

WKT

$$F.O.S = \frac{\tau_u}{\tau_{max}}$$

$$8 = \frac{360}{\tau_{max}}$$

$$\boxed{\tau_{max} = 45\,N/mm^2}$$

W.K.T

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

$$20 \times 10^3 = \frac{2\pi \times 200 \times T}{60}$$

$$T = 955\,N\text{-}m$$

$$\boxed{T = 955 \times 10^3\,N\text{-}mm}$$

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

$$955 \times 10^3 = \frac{\pi}{16} \times 45 \times d^3$$

$$\boxed{d = 47.6\,mm}$$

$$\boxed{d \simeq 50\,mm}$$

Diameter of hollow shaft

Instead of solid shaft is replaced by hollow shaft

$$T_{solid} = T_{Hollow}$$

$$T_{solid} = \frac{\pi}{16} \tau_{max} \left(\frac{d_0^4 - d_i^4}{d_0}\right)$$

$$955 \times 10^3 = \frac{\pi}{16} \times 45 \times \left[\frac{d_0^4 - (0.5 d_0)^4}{d_0}\right]$$

$$d_0 = 48.6$$

$$\boxed{d_0 \simeq 50\,mm}$$

So that $d_i = 0.5\, d_0 = 0.5 \times 50$

$$\boxed{d_i = 25\,mm}$$

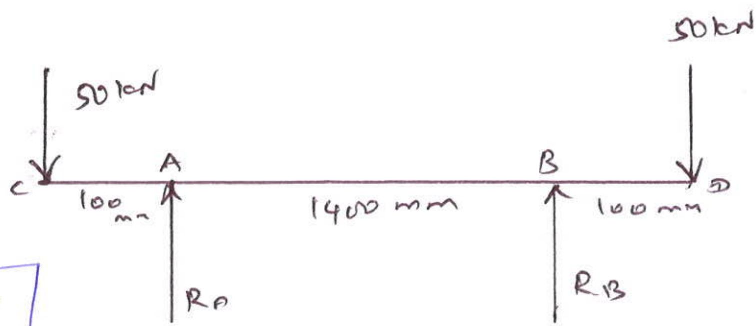
II SHAFT subjected to bending moment only

(1) A pair of wheels of a railway wagon carries a load 50 kN on each axle box acting at a distance of 100 mm out side the wheel base. The gauge of the rails is 1.4 m. Find the diameter of the shaft axle, If the stress is not to exceed 100 MPa.

given data:

$\sigma_b = 100$ MPa

$$\boxed{\sigma_b = 100\ N/mm^2}$$



Max B.M = M = $50 \times 10^3 \times 100$

$= 5 \times 10^6$ N-mm

W.k.T

$$M = \sigma Z$$

$$5 \times 10^6 = 100 \times \frac{\pi}{32} d^3$$

$$d = 79.86 \text{ mm}$$

$$\boxed{d = 80 \text{ mm}}$$

III. Shaft subjected to combined Twisting & bending :

1. Max. Shear stress theory :

$$\tau_{max} = \frac{16}{\pi d^3}\left[\sqrt{M^2 + T^2}\right]$$

T_{eq} = equivalent twisting moment = $\sqrt{M^2 + T^2}$

2. Max. Normal stress theory

$$(\sigma_b)_{max} = \frac{32}{\pi d^3}\left[\frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)\right]$$

equivalent Bending moment $M_{eq} = \frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)$

Note :: Pulley

1. Torque acting on the pulley $T = (T_1 - T_2)R$
2. Ratio of Tensions on pulley $\frac{T_1}{T_2} = e^{\mu\theta}$
3. Ratio b/w Two pulleys $\frac{T_1}{T_2} = \frac{T_3}{T_4}$

Gear

1. Torque acting on the gear $T = W R \cos\alpha$
2. Module of the gear $m = \frac{D}{T}$

3. Force ~~or load~~ acting on the gear

$$F = W\cos\alpha \quad \text{(or)} \quad \frac{2T}{D}$$

where $T \rightarrow$ no of teeth
$D \rightarrow$ dia of gear

Shaft subjected to fluctuating load

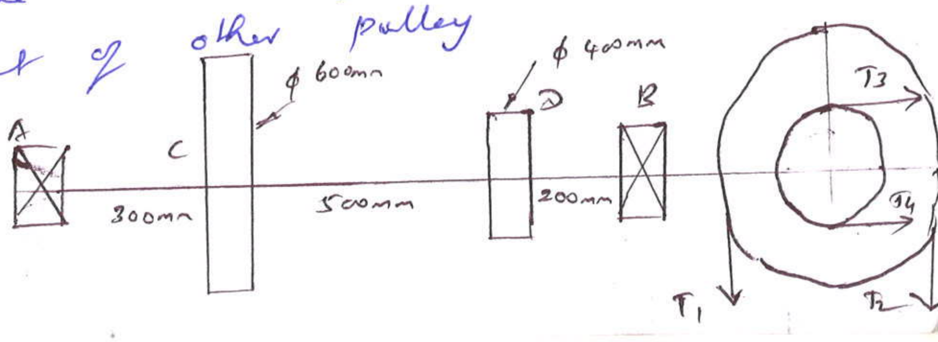
$$T_e = \sqrt{(K_m M)^2 + (K_t T)^2}$$

$$M_e = \frac{1}{2}\left[K_m M + \sqrt{(K_m M)^2 + (K_t T)^2}\right]$$

$K_m \rightarrow$ combined shock & fatigue factor for bending

$K_t \rightarrow$ combined shock & fatigue factor for Torsion.

(1) A shaft is supported by two bearings placed 1m apart. A 600mm diameter pulley is mounted at a distance of 300mm to the right of left hand bearing and this drives a pulley directly below it with a help of belt having max tension of 2.25kN. Another pulley 400mm diameter is placed 200mm to the left of right hand bearing and is driven with the help of electric motor and belt which is placed horizontally to the right. The angle of contact of both pulley is 180° and $\mu = 0.24$. Determine the suitable dia for a solid shaft allowing working stress of 63 MPa in tension & 42 MPa in shear for the material of the shaft. Assume that Torque on one shaft pulley is equal to that of other pulley



given data:

At C

$D_c = 600mm \Rightarrow R_c = 300mm = 0.3m$

At D

$D_D = 400mm \Rightarrow R_D = 200mm = 0.2m$

$\theta = 180^\circ = 180 \times \frac{\pi}{180} = \pi$ radians

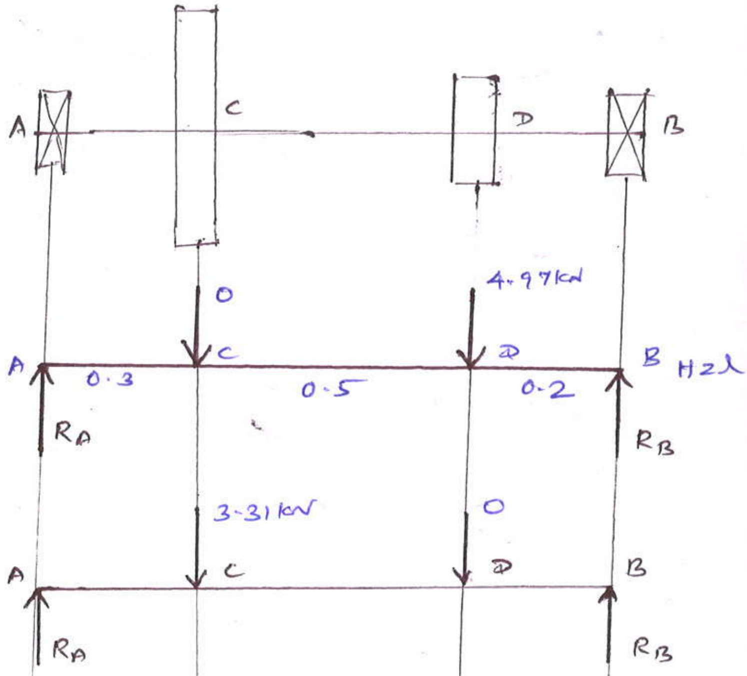
$\mu = 0.24$

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

$\tau = 42\ MPa = 42\ N/mm^2$

$T_1 = 2.25\ kN$

$T_c = T_D$



solution:

From diagram:

(i) Find T_1 & T_2

W.K.T $\frac{T_1}{T_2} = e^{\mu\theta}$

$\frac{2.25}{T_2} = e^{0.24 \times \pi}$

$\boxed{T_2 = 1.06\ kN}$

(ii) Find T_c

$T_c = (T_1 - T_2) R_c$

$= (2.25 - 1.06) \times 0.3$

$\boxed{T_c = 357\ N\text{-}m}$

(iii) Find T_3 & T_4

W.K.T. $\frac{T_1}{T_2} = \frac{T_3}{T_4}$

$\frac{2.25}{1.06} = \frac{T_3}{T_4}$

$$\frac{T_3}{T_4} = 2.12 \Rightarrow T_3 = 2.12\, T_4$$

given that

$$T_C = T_D$$

$$357 = (T_3 - T_4) R_D$$

$$357 = (T_3 - T_4) \times 0.2$$

$$T_3 - T_4 = 1785$$

$[T_3 = 2.12\, T_4]$

$$2.12\, T_4 - T_4 = 1785$$

$$\boxed{T_4 = 1594\, N} = 1.59\, kN$$

$$T_3 = 2.12\, T_4 = 2.12 \times 1594$$

$$\boxed{T_3 = 3379\, N} = 3.38\, kN$$

Horizontal load

Hzl load at C $= 0$

Hzl load at D $= T_3 + T_4 = 3.38 + 1.59$

$= 4.97\, kN$

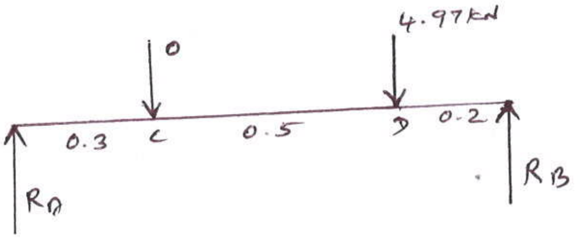
Vertical load

Vzl load at C $= T_1 + T_2 = 2.25 + 1.06$

$= 3.31\, kN$

Vzl load at D $= 0$

Hzl B.M. Diagram



$$R_A + R_B = 4.97$$

$\Sigma M_A = 0$

$$R_B \times 1 - 4.97 \times 0.8 = 0$$

$$\boxed{R_B = 3.98\, kN}$$

$$R_A + R_B = \cancel{3.98}\ 4.97$$

$$R_A + 3.98 = 4.97$$

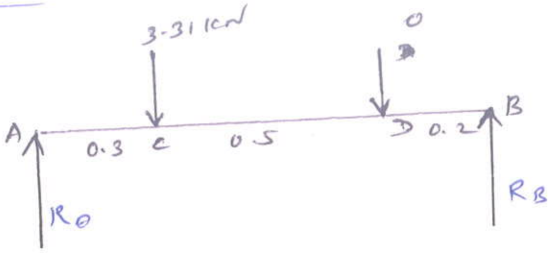
$$\boxed{R_A = 0.99\, kN}$$

$M_B = 0$

$M_D = 3.98 \times 0.2 = 0.796$ kN-m

$M_C = (3.98 \times 0.7) - (4.97 \times 0.5) = 0.301$ kN-m

Vertical B.M diagram



$R_A + R_B = 3.31$

$\Sigma M_A = 0$

$\Rightarrow (R_B \times 1) - (3.31 \times 0.3) = 0$

$\boxed{R_B = 0.993 \text{ kN}}$

$R_A + 0.993 = 3.31$

$\boxed{R_A = 2.317 \text{ kN}}$

$M_B = 0$

$M_D = 0.993 \times 0.2 = 0.1986$ kN-m

$M_C = (0.993 \times 0.7) = 0.6951$ kN-m

$M_A = 0$

Resultant B.M

Resultant B.M at 'C' $= \sqrt{(M_C)_H^2 + (M_C)_V^2}$

$= \sqrt{(0.301)^2 + (0.6951)^2}$

$= 0.758$ kN-m

Resultant B.M. at D $= \sqrt{(M_D)_H^2 + (M_D)_V^2}$

$= \sqrt{(0.796)^2 + (0.1986)^2}$

$= 0.820$ kN-m

$\therefore$ Suitable B.M $M = 0.820$ kN-m

$\boxed{M = 820 \text{ N-m}}$

Suitable Torque $\boxed{T = 357 \text{ N-m}}$

Teq

W.K.T $T_{eq} = \sqrt{M^2 + T^2}$

$T_{eq} = \sqrt{820^2 + 357^2}$

$= 894 \text{ N-M} = 894 \times 10^3 \text{ N-mm}$

W.K.T

$T_{eq} = \frac{\pi}{16} \tau d^3$

$894 \times 10^3 = \frac{\pi}{16} \times 42 \times d^3$

$\boxed{d = 47.68 \text{ mm}}$

$\underline{M_{eq}}$

$M_{eq} = \frac{1}{2} \left[M + \sqrt{M^2 + T^2} \right]$

$= \frac{1}{2} \left[820 + \sqrt{820^2 + 357^2} \right]$

$= 857 \text{ N-m}$

$M_{eq} = 857 \times 10^3 \text{ N-mm}$

W.K.T

$M_{eq} = \sigma Z$

$M_{eq} = \sigma \times \frac{\pi}{32} d^3$

$857 \times 10^3 = 63 \times \frac{\pi}{32} d^3$

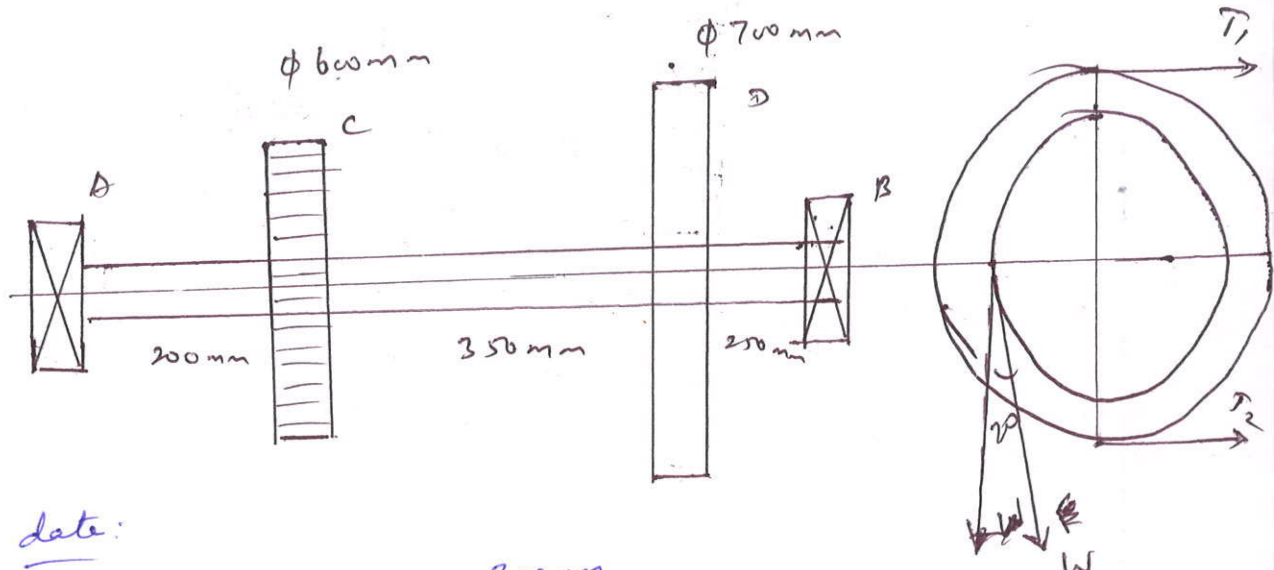
$\boxed{d = 51.75 \text{ mm}}$

$\therefore$ Suitable diameter of the shaft

$d = 51.75$

$\boxed{d \simeq 55 \text{ mm}}$

(2) A shaft is supported on bearings A & B 800 mm between centres. A 20° straight tooth spur gear having 600 mm pitch diameter is located 200 mm to the right of the left hand bearing A. and a 700 mm diameter pulley is mounted 250 mm towards the left of Bearing B. The gear is driven by a pinion with a downward tangential force while the pulley drives a horizontal belt having 180° angle of wrap. The pulley also serves as a flywheel and weighs 2000N. The maximum belt tension is 3000N and tension ratio 3:1. Determine the max B.M and necessary shaft diameter if the allowable shear stress of the material is 40MPa.



given data:

$D_c = 600\text{ mm} \Rightarrow R_c = 300\text{ mm} = 0.3\text{ m}$

$D_D = 700\text{ mm} \Rightarrow R_D = 350\text{ mm} = 0.35\text{ m}$

$\theta = 180° = 180 \times \frac{\pi}{180} = \pi$ radians

$\frac{T_1}{T_2} = \frac{3}{1} \Rightarrow \boxed{T_1 = 3T_2}$

$\tau = 40\text{ MPa} = 40\text{ N/mm}^2$

$T_1 = 3000\text{ N}$

$\alpha = 20°$

$T_2 = \frac{T_1}{3} = \frac{3000}{3} = 1000\text{ N}$

W.K.T.

$$T_D = (T_1 - T_2)\, R_D$$

$$= (3000 - 1000) \times 0.35$$

$$\boxed{T_D = 700 \text{ N-m}}$$

Assume $T_C = T_D$

$$W R_C \cos\alpha = T_D$$

$$W \times 0.3 \times \cos 20 = 700$$

$$\boxed{W = 2483 \text{ N}}$$

At D

H2l load $= T_1 + T_2 = 3000 + 1000$
$= 4000$ N

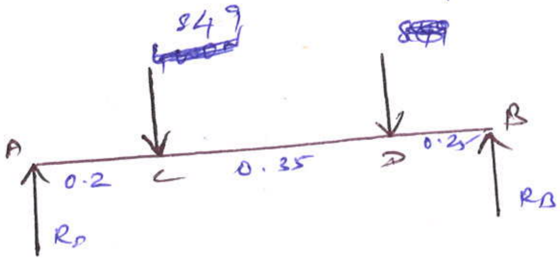
V2l load $= 0$

At C

H2l load $= W \cos 70$
$= 2483 \cos 70$
$= 849$ N

V2l load $= W \sin 70$
$= 2483 \sin 70$
$= 2333$ N

H2l B.M. Diagram



$R_A + R_B = 4000 + 849$
$= 4849$

$\Sigma M_A = 0$

$(R_B \times 0.8) - (849 \times 0.2) - (4000 \times 0.55) = 0$

$$\boxed{R_B = 2962 \text{ N}}$$

$\Rightarrow \quad R_A + 2962 = 4849$

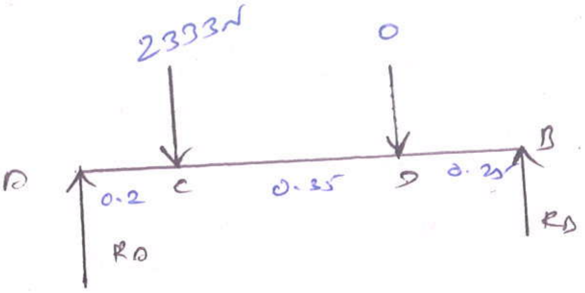
$\boxed{R_A = 1887\ N}$

$M_D = 2962 \times 0.25 = 741\ N\text{-}m$

$M_C = (2962 \times 0.6) - (\cancel{849}\ 4000 \times 0.35)$

$= 377\ N\text{-}m$

vertical B.M.



$R_A + R_B = 2333$

$\Sigma M_A = 0$

$R_B \times 0.8 - 2333 \times 0.2 = 0$

$\boxed{R_B = 583\ N}$

$R_A + R_B = 2333 \Rightarrow R_A + 583 = 2333$

$\boxed{R_A = 1750\ N}$

$M_D = 583 \times 0.25 = 146\ N\text{-}m$

$M_C = (583 \times 0.6) = 350\ N\text{-}m$

Resultant B.M

$(M_C)_{eq} = \sqrt{(M_C)_H^2 + (M_C)_V^2} = \sqrt{377^2 + 350^2}$

$= 514\ N\text{-}m$

$(M_D)_{eq} = \sqrt{(M_D)_H^2 + (M_D)_V^2} = \sqrt{741^2 + 146^2}$

$= 755\ N\text{-}m$

$\therefore$ Max B.M $M = 755\ N\text{-}m$

$\boxed{M = 755 \times 10^3\ N\text{-}mm}$

Max Torque $T = 700\ N\text{-}m = 700 \times 10^3\ N\text{-}mm$

dia of shaft $T = \frac{\pi}{16} \tau d^3$

$700 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$

$d = 44.67$

$\boxed{d = 45\ mm}$

SHAFT COUPLING

* It is a device which is used to make Permanent (or) semi permanent connection of the different length of shaft

Types of coupling

* **Rigid coupling**
  * It is a device which is used to connect two shafts which are perfectly aligned.
    - (i) sleeve (or) Muff coupling
    - (ii) clamp (or) compression (or) split muff coupling
    - (iii) Flange coupling

* **Flexible coupling**
  * It is a device which is used to connect two shafts having both lateral and angular mis alignment.
    - (i) Bushed Pin type coupling
    - (ii) Universal coupling
    - (iii) old ham's coupling

SLEEVE OR MUFF COUPLING

Step 1 Design of shaft

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

From the above relation find 'T'

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

From the above relation find 'd' and converted into standard one.

Step 2: Design of sleeve (or) muff

Outer diameter of sleeve $D = 2d + 13$

length of sleeve $L = 3.5d$

Check τ

$$T = \frac{\pi}{16} \tau \left(\frac{D^4 - d^4}{D} \right)$$

From the above relation find τ and check with the given value

Step 3: Design of key

(i) length of key $l = \frac{L}{2}$

(i) selection of width & thickness

If the relation $\sigma_c = 2\tau$ is satisfied. the key is square

$\therefore$ width & thickness are same.

From PSGDB P.No 5.16 select suitable width (w) & thickness (t) for given diameter (d)

(ii) Check τ

$$T = l \, w \, \tau \, \frac{d}{2}$$

From the above relation Find τ and check with given value

(iii) check σ_c

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

From the above relation Find σ_c and check with the given value.

① Design a muff coupling which is used to connect two steel shaft transmitting 40 kW at 350 rpm. The material for the shaft and key is plain carbon steel for which allowable shear and crushing stresses may be taken as 40 MPa and 80 MPa respectively. The material for the muff is cast iron for which the allowable shear stresses may be assumed as 15 MPa

Given data:

shaft

$P = 40 kW = 40 \times 10^3 W$

$N = 350$ rpm

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

sleeve

$\tau = 15 MPa$
$= 15 N/mm^2$

key

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

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

Step 1 design of shaft

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

$$40 \times 10^3 = \frac{2\pi \times 350 \times T}{60}$$

$$T = 1091 \text{ N-m}$$

$$\boxed{T = 1091 \times 10^3 \text{ N-mm}}$$

W.K.T

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

$$1091 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$d = 51.8 \text{ mm}$$

$$\boxed{d = 55 \text{ mm}}$$

Step 2: Design of sleeve

outer diameter of sleeve $D = 2d + 13$

$= 2 \times 55 + 13$

$$\boxed{D = 123\ mm}$$

length of sleeve $L = 3.5d = 3.5 \times 55$

$$\boxed{L = 192.5\ mm}$$

check. τ

$$T = \frac{\pi}{16}\tau\left(\frac{D^4 - d^4}{D}\right)$$

$$1091 \times 10^3 = \frac{\pi}{16} \times \tau \times \frac{123^4 - 55^4}{123}$$

$$\tau = 3.1\ N/mm^2 < (\tau_{given} = 15\ N/mm^2)$$

Step 3: Design of key

(i) length of key $l = \frac{L}{2} = \frac{192.5}{2}$

$$\boxed{l = 96.25\ mm}$$

(i) Selection of width & thickness

$\sigma_c = 2\tau \Rightarrow 80 = 2 \times 40$

$80 = 80$

$\therefore$ The key is square

From P.no 5.16 ; For $d = 55\ mm$

$$\boxed{w = 16\ mm = t}$$

(ii) Check τ

$$T = l \cdot w \ \tau \ \frac{d}{2}$$

$$1091 \times 10^3 = 96.25 \times 16 \times \tau \times \frac{55}{2}$$

$$\tau = 25.76\ N/mm^2 < (\tau_{given} = 40\ N/mm^2)$$

(iii) check σ_c

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

$$1091 \times 10^3 = 96.25 \times \frac{16}{2} \times \sigma_c \times \frac{55}{2}$$

$$\sigma_c = 51.5\ N/mm^2 < (\sigma_{c\,given} = 80\ N/mm^2)$$

CLOMP OR COMPRESSION OR SPLIT MUFF COUPLING

Step 1: Design of shaft

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

From the above relation Find 'T'

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

From the above relation Find 'd' and convert into standard.

Step 2: Design of sleeve

Outer diameter of sleeve $D = 2d + 13$

length of sleeve $L = 3.5d$

Check of τ

$$T = \frac{\pi}{16} \tau \frac{D^4 - d^4}{D}$$

From the above relation find τ & check with given value

Step 3: Design of key

(i) length of key $l = \frac{L}{2}$

(ii) Selection of width & thickness

From PSGDB P. no 5.16 select suitable width & thickness for the given 'd'

(iii) Check τ

$$T = l\, w\, \tau\, \frac{d}{2}$$

From the above relation Find τ & check with given value

(iv) Check σ_c

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

From the above relation Find σ_c & check with given value

Step 4: Design of bolts :

Torque transmitted $T = \frac{\pi^2}{16} \times \mu \, d_b^2 \times \sigma_t \, n \, d$

where $\mu \rightarrow$ coefficient of friction.

$d_b \rightarrow$ diameter of bolt.

$\sigma_t \rightarrow$ permissible tensile stress

From the above relation find d_b and convert into standard. (P.NO. 5.42)

(1) Design a clamp coupling to transmit 30 kW at 100 rpm. The allowable shear stress for the shaft and key is 40 MPa and number of bolts connecting two halves are six. The permissible tensile stress for the bolts is 70 MPa. The coefficient of friction b/w the muff and shaft surface may be taken as 0.3

Gn data:

Shaft

$P = 30\,kW = 30 \times 10^3\,W$

$N = 100\,rpm$

$\tau = 40\,MPa$

$= 40\,N/mm^2$

Key

$\tau = 40\,MPa$

$= 40\,N/mm^2$

bolt

$n = 6$

$\sigma_t = 70\,MPa$

$= 70\,N/mm^2$

$\mu = 0.3$

Step 1 Design of Shaft

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

$$30 \times 10^3 = \frac{2\pi \times 100 \times T}{60}$$

$$T = 2865\,N\text{-}m$$

$$\boxed{T = 2865 \times 10^3\,N\text{-}mm}$$

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

$$2865 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$d = 71.5 mm$$

$$\boxed{d = 75 mm}$$

Step 2: Design of sleeve

Outer diameter of sleeve $D = 2d + 13$

$$D = 2 \times 75 + 13$$

$$\boxed{D = 163 mm}$$

length of sleeve $L = 3.5d$

$$= 3.5 \times 75$$

$$\boxed{L = 262.5 mm}$$

Step 3: Design of key:

For $d = 75 mm$

P. No 5.16

$w = 22 mm$; $t = 14 mm$

length of key $l = \frac{L}{2} = \frac{262.5}{2} = 131.25 mm$

check τ

$$T = l w \tau \frac{d}{2}$$

$$2865 \times 10^3 = 131.25 \times 22 \times \tau \times \frac{75}{2}$$

$$\tau = 26.46 N/mm^2 < (\tau_{given} 40 N/mm^2)$$

Step 4:

Torque transmitted

$$T = \frac{\pi^2}{16} \times \mu \times d_b^2 \times \sigma_t \; n \; d$$

$$2865 \times 10^3 = \frac{\pi^2}{16} \times 0.3 \times d_b^2 \times 70 \times 6 \times 75$$

$$\boxed{d_b = 22.17\,mm}$$

standard size of bolt.

P. no 5.42 $\boxed{d = 30\,mm}$ ie M 30

FLANGE COUPLING

* It is a coupling having two separate cast iron flanges
  * Unprotected type flange coupling
  * protected type flange coupling

DESIGN OF FLANGE COUPLING

step 1 Design of shaft

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

From the above relation find 'T'

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

From the above relation find 'd' and convert to standard.

Step 2: Design of hub

outer diameter of hub $D = 2d$

length of hub $L = 1.5d$

Check τ

$$T = \frac{\pi}{16} \tau \left(\frac{D^4 - d^4}{D}\right)$$

From the above relation find 'τ' and check with given value.

Step 3: Design of key

length of key $l = L$

(i) selection of width x thickness

From PSGDB P.no 5.16 select suitable width & thickness

(ii) check τ

$$T = l \; w \; \tau \; \frac{d}{2}$$

From the relation Find 'τ' and check with given value

(iii) check σ_c

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

From the above relation find σ_c and check with given value

Step 4: Design of flange:

thickness of flange $t_f = \frac{d}{2}$

check τ

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

From the above relation find τ and check with given value.

Step 5: Design of bolts.

outer dia of flange $D_2 = 4d$

Diameter of bolt circle $D_1 = 3d$

$$T = n \; \frac{\pi}{4} d_1^2 \; \tau \; \frac{D_1}{2}$$

From the above relation find 'd_1' and convert into standard.

Step 6: thickness of Protection:

$$t_p = \frac{d}{4}$$

① Design a protective type C-I flange coupling for a steel shaft transmitting 15 kW at 200 rpm. and having an allowable shear stress of 40 MPa. The working stress in the bolts should not exceed 30 MPa. Assume that the same material is used for shaft and key and that the crushing stress is twice the value of its shear stress. The maximum torque is 25% greater than full load torque. The shear stress for C-I is 14 MPa.

gn data:

shaft:

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

$N = 200$ rpm.

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

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

bolt:

$\tau = 30\ MPa$

$= 30\ N/mm^2$

key:

$\sigma_c = 2\tau$

$\tau = 40\ N/mm^2$

$\sigma_c = 2 \times 40$

$\boxed{\sigma_c = 80\ N/mm^2}$

flange

$\tau = 14\ MPa$

$= 14\ N/mm^2$

hub

$\tau = 14\ MPa$

$\tau = 14\ N/mm^2$

Solution:

step 1 Design of shaft

$$P = \frac{2\pi N T_{mean}}{60}$$

$$15 \times 10^3 = \frac{2\pi \times 200 \times T_{mean}}{60}$$

$$\boxed{T_{mean} = 716\ N\text{-}m}$$

$$T_{max} = 1.25\ T_{mean} = 1.25 \times 716$$

$$T_{max} = 895\ N\text{-}m$$

$$\boxed{T_{max} = 895 \times 10^3\ N\text{-}mm}$$

$$T_{max} = \frac{\pi}{16} \tau d^3$$

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

$$d = 48.5$$

$$\boxed{d = 50\,mm}$$

Step 2: Design of hub

outer diameter of hub $D = 2d$

$= 2 \times 50$

$$\boxed{D = 100\,mm}$$

length of hub $L = 1.5d = 1.5 \times 50$

$$\boxed{L = 75\,mm}$$

check τ

$$T = \frac{\pi}{16} \tau \left(\frac{D^4 - d^4}{D}\right)$$

$$895 \times 10^3 = \frac{\pi}{16} \times \tau \times \left(\frac{100^4 - 50^4}{100}\right)$$

$$\tau = 4.86\ N/mm^2 < \left(\tau_{given} = 14\ N/mm^2\right)$$

Step 3: Design of key

length of key $l = L = 75\,mm$

given that $\sigma_c = 2\tau$

$\therefore$ The key is square.

(i) selection of width & thickness

from PSG DB P.no 5.16

For $d = 50\,mm$

$$\boxed{w = t = 16\,mm}$$

(ii) check τ

$$T = l \ w \ \tau \ \frac{d}{2}$$

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

$$\tau = 29.8 \ N/mm^2 < (\tau_{given} = 40 \ N/mm^2)$$

(iii) check σ_c

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

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

$$\sigma_c = 59.7 \ N/mm^2 < (\sigma_{c \, given} = 80 \ N/mm^2)$$

step 4: Design of flange:

thickness of flange $t_f = \frac{d}{2} = \frac{50}{2}$

$$\boxed{t_f = 25mm}$$

check τ

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

$$895 \times 10^3 = \frac{\pi \times 100^2}{2} \times \tau \times 25$$

$$\tau = 2.28 \ N/mm^2 < (\tau_{given} = 14 \ N/mm^2)$$

step 5: Design of bolts

outer diameter of flange $D_2 = 4d = 4 \times 50$

$$\boxed{D_2 = 200mm}$$

Diameter of bolt circle $D_1 = 3d = 3 \times 50$

$$\boxed{D_1 = 150 \ mm}$$

$$T = n \ \frac{\pi}{4} \ d_1^2 \ \tau \ \frac{D_1}{2}$$

~~$895 \times 10^3 =$~~ For $d = 50mm$ $\boxed{n = 4}$

$$895 \times 10^3 = 4 \times \frac{\pi}{4} \times d_1^2 \times 30 \times \frac{150}{2}$$

$d_1 = 11.25$ mm

standard size of bolt from PSG DB P.no 5.42

$$\boxed{d_1 = 16 mm}$$

step 6: thickness of protection

$$t_p = \frac{d}{4}$$

$$t_p = \frac{50}{4}$$

$$\boxed{t_p = 12.5 mm}$$

FLEXIBLE COUPLING

* It is a coupling which is used to connect shaft subject to one or more kinds misalignment and to reduce the effect of shock & impact loads.

DESIGN OF BUSHED PIN TYPE FLEXIBLE COUPLING

step 1 Design of shaft:

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

From the above relation find 'T'

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

From the above relation find 'd'

step 2: Design of hub:

From PSG DB P.no 7.108

outer diameter of hub D = ~~B~~ 'C'

length of hub L = 'E'

Step 3: Design of key

length of key l = 'E'

(i) selection of width & thickness

Select suitable width & thickness from PSG DB P.No 5.16 for the given diameter

(ii) check for τ

$$T = l \; w \; \tau \; \frac{d}{2}$$

From the above relation find τ & check with the given value

(iii) check for σ_c

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

From the above relation find σ_c & check with given value.

Step 4: Design of flange

From PSG DB 7.1018

thickness of flange t_f = G.

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

From the above relation find τ and check with the given value.

~~Step 5: Design of bush~~

~~length of bush~~ $\cancel{L = G + t = \frac{2}{3}E}$

Step 5:

(i) Bearing load on pin (W)

$$T_{max} = W \times n \times \frac{D}{2}$$

From the above relation find 'W'

(ii) Direct stress due to Torsion in coupling

$$\tau = \frac{W}{\frac{\pi}{4} \times (F)^2}$$ F → Bolt diameter

iii) Maximum bending moment on the Pin (M)

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

G → length of bush
t → clearance

(iv) Bending stress (σ_b)

$$\sigma_b = \frac{M}{\frac{\pi}{4} \times F^3}$$ F → Bolt diameter

(v) Maximum Principal stress

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

step 6: Design of bush

length of bush $L = G + t - \frac{2}{3}F$

TEMPORARY AND PERMANENT JOINTS

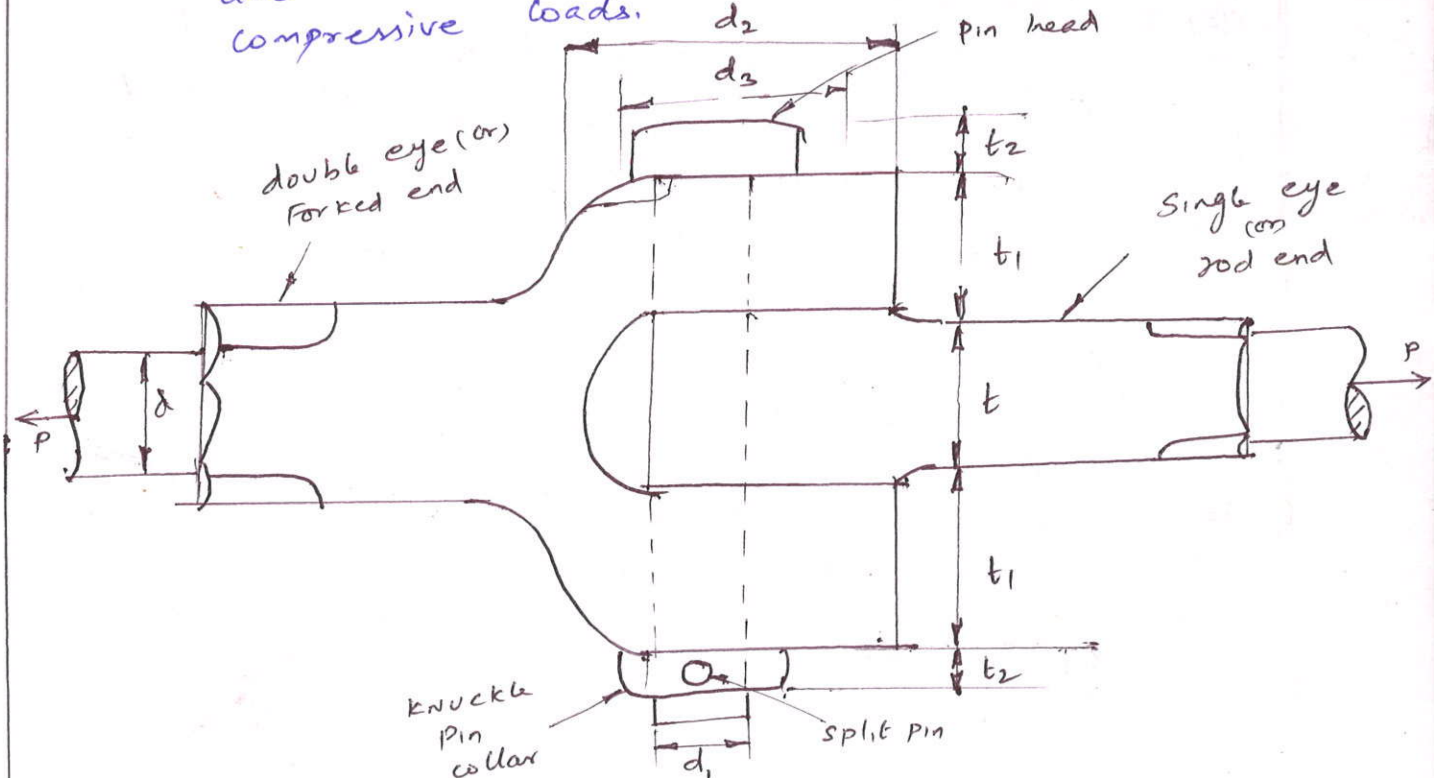
Types of Joints:

* Temporary Joint
  - (a) Knuckle Joint
  - (b) Bolted Joint
  - (c) cotter Joint
* semi Permanent Joint
  - (a) Rivetted Joint
* Permanent Joint
  - (a) Welded Joint

DESIGN OF TEMPORARY JOINTS

KNUCKLE JOINT

* It is used to connect two rods which are under the action of tensile loads & compressive loads.



$d \rightarrow$ diameter of the rod

$d_1 \rightarrow$ diameter of the pin $= d$

$d_2 \rightarrow$ outer diameter of the eye $= 2d$

$d_3 \rightarrow$ diameter of the knuckle pin head & collar $= 1.5d$

$t \rightarrow$ thickness of single eye (or) rod end $= 1.25d$

$t_1 \rightarrow$ thickness of the fork $= 0.75d$

$t_2 \rightarrow$ thickness of the pin head. $= 0.5d$

DESIGN PROCEDURE OF KNUCKLE JOINT

step 1: diameter of the rod (d)

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

From the above relation find 'd'

Step 2: diameter of knuckle pin & thickness of pin head

diameter of knuckle pin $d_1 = d$

thickness of pin head $t_2 = 0.5d$

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

From the above relation find τ & check with given value

step 3: Failure of single eye or rod end in tension

outer diameter of the eye $d_2 = 2d$

thickness of single eye $t = 1.25d$

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

From the above relation find σ_t & check with the given value

step 4: Failure of single eye (or) rod end in shear

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

From the above relation find τ & check with given value

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

$$P = d_1 t \ \sigma_c$$

From the above relation find σ_c & check with given value.

step 6: Failure of forked end in tension

thickness of fork $t_1 = 0.75d$

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

From the above relation Find σ_t & check with given value

step 7: Failure of forked end in shear

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

From the above relation Find τ & check with given value

step 8: Failure of forked end in crushing

$$P = \cancel{d_2}\, d_1\, 2t_1\, \sigma_c$$

From the above relation find σ_c & check with given value.

① Design a knuckle joint to transmit 150 kN. The design stresses may be taken as 75 MPa in tension 60 MPa in shear & 150 MPa in compression

given data

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

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

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

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

step 1 diameter of rod (d)

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

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

$$\boxed{d = 51\ mm}$$

Step 2: diameter of knuckle pin & thickness of pin head

diameter of knuckle pin $d_1 = d = 51\,mm$

thickness of pin head $t_2 = 0.5d = 0.5 \times 51$

$$\boxed{t_2 = 26\,mm}$$

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

$$150 \times 10^3 = 2 \times \frac{\pi}{4} \times 51^2 \times \tau$$

$$\boxed{\tau = 36.71\ N/mm^2} < (\tau_{given} = 60\,N/mm^2)$$

Step 3 Failure of single eye or rod end in tension

outer diameter of the eye $d_2 = 2d$

$= 2 \times 51$

$$\boxed{d_2 = 102\,mm}$$

thickness of single eye $t = 1.25d$

$= 1.25 \times 51$

$$\boxed{t = 63.75\,mm}$$

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

$$150 \times 10^3 = (102-51) \times 63.75 \times \sigma_t$$

$$\sigma_t = 46.14\ N/mm^2 < ((\sigma_t)_{given} = 75\,N/mm^2)$$

Step 4: Failure of single eye (or) rod end in shear

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

$$150 \times 10^3 = (102 - 51) \times 63.75 \times \tau$$

$$\tau = 46.14\ N/mm^2 < (\tau_{given} = 60\,N/mm^2)$$

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

$$P = d_1\, t\, \sigma_c$$

$$150 \times 10^3 = 51 \times 63.75 \times \sigma_c$$

$$\sigma_c = 46.14\ N/mm^2 < (\sigma_{c\,given} = 150\,N/mm^2)$$

Step 6: Failure of forked end in tension 3

Thickness of fork $t_1 = 0.75d = 0.75 \times 51$

$$\boxed{t_1 = 38.25\ mm}$$

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

$$150 \times 10^3 = (102 - 51)\ 2 \times 38.25 \times \sigma_t$$

$$\sigma_t = 38.45\ N/mm^2 < (\sigma_t)_{given} = 75\ N/mm^2$$

Step 7: Failure of forked end in shear

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

$$150 \times 10^3 = (102 - 51) \times 2 \times 38.25 \times \tau$$

$$\tau = 38.45\ N/mm^2 < (\tau_{given} = 60\ N/mm^2)$$

Step 8: Failure of forked end in crushing

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

$$150 \times 10^3 = 51 \times 2 \times 38.25 \times \sigma_c$$

$$\sigma_c = 38.45\ N/mm^2 < ((\sigma_c)_{given} = 150\ N/mm^2)$$

(2) Design a knuckle joint to connect 2 mild steel bars under a tensile load of 25 kN. Assume permissible stresses as follows. $\sigma_t = 65$ MPa, $\tau = 50$ MPa, $\sigma_c = 83$ MPa

(3) A knuckle joint is required to withstand a tensile load of 25 kN. Design the joint if the permissible stresses are $\sigma_t = 56$ MPa, $\tau = 40$ MPa, $\sigma_c = 70$ MPa

Cotter Joint:

* A cotter ~~Joint~~ is a flat wedge shaped piece of Rectangular C-S and its width is tapered

Types of cotter Joint:

* Sleeve and cotter Joint
* Socket and spigot cotter Joint
* Gib and cotter Joint

Design of sleeve and cotter Joint:

Step 1: Diameter of the rod (d)

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

From the above relation Find 'd'

Step 2: Diameter of enlarged end of the rod & thickness of cotter:

Thickness of cotter $t = \frac{d_2}{4}$

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

From the above relation Find 'd_2' & t

$$P = d_2 t \sigma_c$$

From the above relation Find σ_c & check with given value.

Step 3: Outer diameter of sleeve (d_1)

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2) t\right] \sigma_t$$

From the above relation Find (d_1)

Step 4: Width of cotter: (b)

$$P = 2 b t \tau$$

From the above relation Find 'b'

Step 5: Distance of rod from beginning to the cotter hole

$$P = 2\, a\, d_2\, \tau$$

From the above relation find 'a'

Step 6: Distance of rod from its end to cotter hole

$$P = 2(d_1 - d_2)\, c\, \tau$$

From the above relation find 'c'

1. Design a cotter joint to resist a tensile load of 60kN. All parts of the joint are made of the same material with the following allowable stresses: $\sigma_t = 60$ MPa ; $\sigma_c = 125$ MPa ; $\tau = 70$ MPa

Given data:

$P = 60\,kN = 60 \times 10^3\, N$

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

$\tau = 70\, MPa = 70\, N/mm^2$

$\sigma_c = 125\, MPa = 125\, N/mm^2$

Step 1: Diameter of the rod (d)

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

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

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

Step 2: Diameter of enlarged end of the rod & thickness of cotter:

Thickness of cotter $t = \frac{d_2}{4}$

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

$$60 \times 10^3 = \left[\frac{\pi}{4} d_2^2 - d_2 \times \frac{d_2}{4}\right] \times 60$$

$$\boxed{d_2 = 44mm}$$

Thickness of cotter $t = \frac{d_2}{4} = \frac{44}{4}$

$$\boxed{t = 11\ mm}$$

$$P = d_2\, t\, \sigma_c$$

$$60 \times 10^3 = 44 \times 11 \times \sigma_c$$

$$\boxed{\sigma_c = 123.9\ N/mm^2} < (\sigma_{c\,given} = 125\ N/mm^2)$$

Step 3: outer diameter of sleeve

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)\, t\right] \sigma_t$$

$$60 \times 10^3 = \left[\frac{\pi}{4}(d_1^2 - 44^2) - (d_1 - 44) \times 11\right] 60$$

$$d_1 = 58.4$$

$$\boxed{d_1 = 60\ mm}$$

Step 4: width of cotter

$$P = 2\, b\, t\, \tau$$

$$60 \times 10^3 = 2 \times b \times 11 \times 70$$

$$\boxed{b = 38.96\ mm}$$

Step 5: Distance of rod from beginning to the cotter hole

$$P = 2\, a\, d_2\, \tau$$

$$60 \times 10^3 = 2 \times a \times 44 \times 70$$

$$\boxed{a = 9.74\ mm}$$

Step 6: Distance of rod from its end to the cotter hole

$$P = 2(d_1 - d_2)\, c\, \tau$$

$$60 \times 10^3 = 2(60 - 44)\, c \times 70$$

$$\boxed{c = 26.78\ mm}$$

II SOCKET AND SPIGOT COTTER JOINT

4 5

Step 1: Diameter of the rod. (d)

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

From the above relation find 'd'

Step 2: Diameter of spigot (inside dia of socket) & thickness of cotter

thickness of cotter $t = \frac{d_2}{4}$

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

From the above relation find d_2 & t

$$P = d_2 \, t \, \sigma_c$$

From the above relation find σ_c & check with given value.

Step 3: outside diameter of socket (d_1)

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2) t\right] \sigma_t$$

From the above relation find d_1

Step 4: width of cotter (b)

$$P = 2 b t \tau$$

From the above relation find 'b'

Step 5: Diameter of socket collar (d_4)

$$P = (d_4 - d_2) t \, \sigma_c$$

From the above relation find 'd_4'

Step 6: Thickness of collar

$$P = 2(d_4 - d_2)\, c\, \tau$$

From the above relation find 'c'

Step 7: Distance from end of slot to end of rod: (a)

$$P = 2\, a\, d_2\, \tau$$

From the above relation find 'a'

Step 8: Diameter of spigot collar (d_3)

$$P = \frac{\pi}{4}(d_3^2 - d_2^2)\, \sigma_c$$

From the above relation find 'd_3'

Step 9: Thickness of spigot collar: (t_1)

$$P = \pi\, d_2\, t_1\, \tau$$

From the above relation find 't_1'

Step 10: Length of cotter (l)

length of cotter $l = 4d$

1. Design a cotter joint to support a load of 35 kN. The material used is carbon steel for which the following allowable stresses may be used. $\tau = 35\ N/mm^2$; $\sigma_c = 90\ N/mm^2$; $\sigma_t = 50\ N/mm^2$;

2. Design a cotter joint to connect two mild steel rods for a pull of 30 kN. The maximum permissible stresses are 55 Mpa in tension 40 Mpa in shear & 70 Mpa in crushing.

Riveted Joints:

* A rivet is a short cylindrical bore with a head integral to 'd'. The cylindrical portion of the rivet is called shank (or) body & lower portion of shank is known as tail.

Types of Riveted Joints

1) Lap Joint
2) Butt Joint
   * single strap Butt Joint
   * double strap Butt Joint

Note:

* Riveted Joints according to no of rows of rivets
  (a) single Riveted Joints
  (b) double Riveted Joints.
   * chain Riveted Joints.
   * zig-zag Riveted Joints

DESIGN OF BOILER JOINT

* longitudinal Butt Joint
* circumferential lap Joint

longitudinal Butt Joint

Step 1 thickness of boiler shell (t)

From PSGDB P.no 7.126

$$t = \frac{pD}{2 \eta_l \sigma_t}$$

$p \rightarrow$ steam Pressure

$D \rightarrow$ diameter of boiler vessel

$\eta_l \rightarrow$ efficiency of longitudinal Joint [P.no: 7.126]

$\sigma_t \rightarrow$ Permissible tensile stress

Step 2 diameter of rivet (d)

From PSGDB P.no 7.126

$t > 8mm \Rightarrow d = 6\sqrt{t}$

$t < 8mm \Rightarrow$ By shearing & Tearing strength of rivet

i.e: $F_S = F_T$

$i\, n\, \frac{\pi}{4} d^2\, \tau = i\,(p-d)\, t\, \sigma_t$

$i \rightarrow$ no of rivets

$n \rightarrow$ no of strap (or) cover

Step 3: pitch of the rivets (P)

Pitch of the rivet is calculated by equating the Tearing of rivet plate to the shearing of rivet

note * The pitch should not be less than 2d

* $P_{max} = Ct + 41$

* If $P > P_{max}$, then P_{max} is taken as pitch.

Step 4: Distance between the rows of rivets (P_b)

From PSGDB P.no 7.126 select suitable formula According to condition given in Problem.

Step 5 Thickness of strap plate (t_1)

From PSGDB P.no 7.127 select suitable formula according to condition given in Problem.

Step 6 : margin of the plate

From P. No 7.125 select the formula.

1. Design a double riveted butt joint with two cover plates for the longitudinal steam of a boiler shell 1.5m diameter subjected to a steam pressure of 0.95 N/mm². Assume joint efficiency as 75% allowable tensile stress in the plate 90 MPa ; compressive stress 140 MPa and shear stress in the rivet 56 MPa.

given data:

$D = 1.5m = 1500 mm$

$p = 0.95 N/mm^2$

$\eta_\ell = 75\% = 0.75$

$\sigma_t = 90 MPa = 90 N/mm^2$

$\sigma_c = 140 MPa = 140 N/mm^2$

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

Step 1 Thickness of boiler shell

From PSGDB P.No 7.126

$$t = \frac{pD}{2\eta_\ell \sigma_t} = \frac{0.95 \times 1500}{2 \times 0.75 \times 90}$$

$$t = 10.56 mm$$

$$\boxed{t = 11 mm}$$

Step 2. diameter of the rivet (d)

$t > 8 mm \Rightarrow$

$\therefore$ PSGDB P.No 7.126

$$d = 6\sqrt{t} = 6\sqrt{11}$$

[PSGDB P.No 5.29]

$d = 20 mm$

Step 3: Pitch of the rivet (P)

By equating Tearing of the plate to shearing of the plate

i.e. $F_t = F_s$

From P. no 7.124

$F_t = i(p-d) \times t \; \sigma_t$

$= 2(P-20) \times 11 \times 90$

$\boxed{F_t = 1980(P-20)}$

$F_s = i \; n \; \frac{\pi}{4} d^2 \tau$

$= 2 \times 2 \times \frac{\pi}{4} \times 20^2 \times 56$

$\boxed{F_s = 70372 \text{ N}}$

$F_t = F_s$

$1980(P-20) = 70372$

$P-20 = 35.54$

$\boxed{P = 55.54 \text{ mm}}$

From P. no 7.126

$P_{max} = ct + 41$

$= 3.5 \times 11 + 41$

$P_{max} = 79.5 \text{mm}$

$\cancel{P}$ $p < \cancel{p}$ P_{max}

$\therefore$ Pitch $\boxed{P = 60 \text{ mm}}$

Step 4 : Distance b/w rows of rivets (P_b)

Assume chain riveting. P. no 7.126

$$P_b = 2d = 2 \times 20 = 40 \text{ mm}$$

Step 5 : thickness of strap plate : (t_1)

From P. no 7.127

For double cover,

$$t_1 = 0.625\, t$$

$$= 0.625 \times 11$$

$$\boxed{t_1 = 6.875 \text{ mm}}$$

Step 6 : Margin of the plate (m)

P. no 7.125

$$\text{margin } m = 1.5d = 1.5 \times 20$$

$$\boxed{m = 30 \text{ mm}}$$

PERMANENT JOINT

WELDED JOINT

* welded joint are divided into two groups
    * (i) Butt joint
    * (ii) fillet joint

Butt joint

* It is defined as a joint b/w two components lying approximately in the same plane.

Fillet joint

* It is also called a lap joint
* It is defined as the joint b/w two over lapping plates the weld c-s is s lar & joining two sectios is right angles to each other.

Types of fillet joint

* Parallel fillet weld
* Transverse fillet weld.

Types of loading

* Axial load
* Eccentric load.

Axial loading

* only one stresses tensile (or) compresive

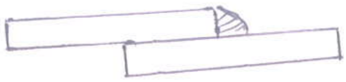
Eccentric loading

* shear stress
* bending stress.

Axial Loading Problems

From PSG DB P. no 11.3

Strength of transverse fillet weld (single)



$$\sigma = \frac{0.707 P}{h l}$$

$P \rightarrow$ tensile load

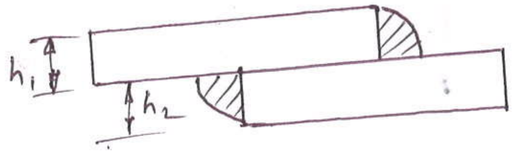
$h \rightarrow$ size of weld

$l \rightarrow$ length of weld

$\sigma \rightarrow$ tensile stress.

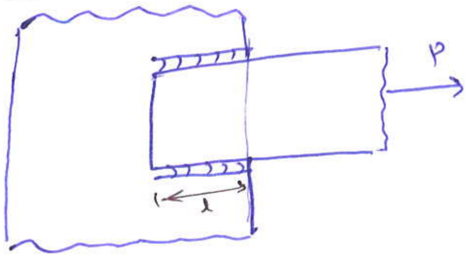
Strength of transverse fillet weld (double)

$$\sigma = \frac{1.414 P}{(h_1 + h_2) l}$$



Strength of || el fillet weld:

Double || el fillet:



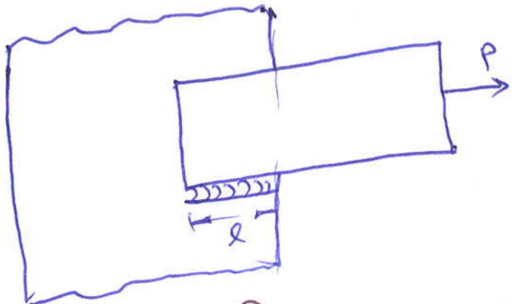
$$\tau = \frac{P}{\text{Area of weld}}$$

$$\tau = \frac{P}{2 l t}$$

$$\tau = \frac{P}{2 \times 0.707 h \times l}$$

$$\boxed{P = 1.414\, h l \tau}$$

Single || el fillet weld



$$\tau = \frac{P}{\text{Area of weld}}$$

$$\tau = \frac{P}{l \times t}.$$

$$= \frac{P}{0.707\, h \times l}$$

$$\boxed{P = 0.707\, h l \tau}$$

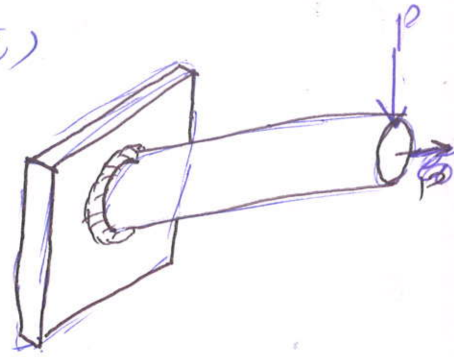
Eccentric loaded welded joints

In eccentric loading there are two types.

(i) welded connections subjected to moment acting in a plane of weld.

(ii) welded connections subjected to moment acting in a plane normal to the plane of weld.

Welded connection subjected to moment acting in a plane of weld



Step 1: direct shear stress (τ)

$$\tau = \frac{\text{Load}}{\text{Area of weld}}$$

Step 2: Bending stress (σ_b)

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

M → Bending moment → $[M = P.e]$

Z → section modulus → P.no 11.5

Step 3: Maximum Normal stress $(\sigma_t)_{max}$

$$(\sigma_t)_{max} = \frac{1}{2}\left[\sigma_b + \sqrt{\sigma_b^2 + 4\tau_x^2}\right]$$

Step 4: Maximum shear stress

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

Welded connections subjected to moment acting in a plane normal to the plane of weld

8

Step 1 Direct (or) Primary shear stress (τ_1)

$$\tau_1 = \frac{P}{\text{Area of weld}}$$

Step 2: Secondary shear stress (τ_2)

$$\tau_2 = \frac{T\,r}{J}$$

where $T \rightarrow$ Twisting moment ($T = P \times e$)

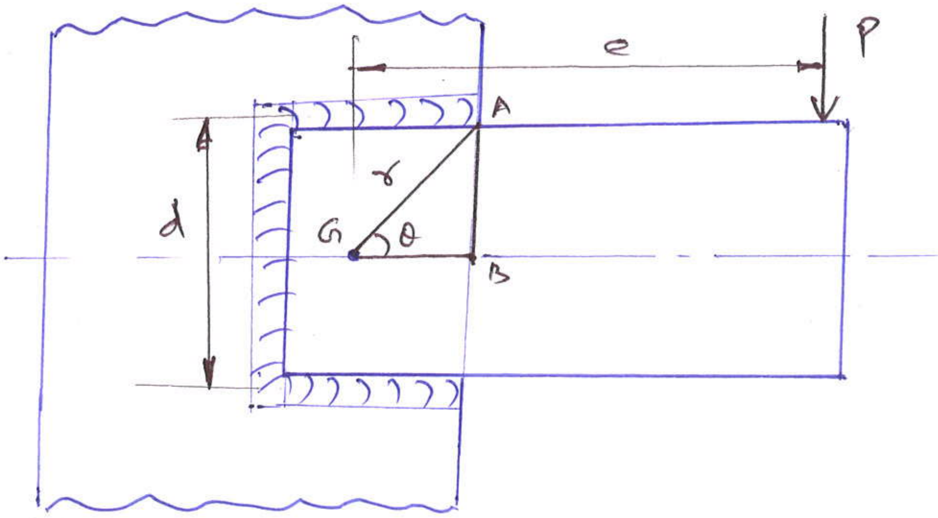
$r \rightarrow \sqrt{GB^2 + AB^2}$

$J \rightarrow$ Polar moment of Inertia (PNO 11.5)

Step 3: Resultant stress

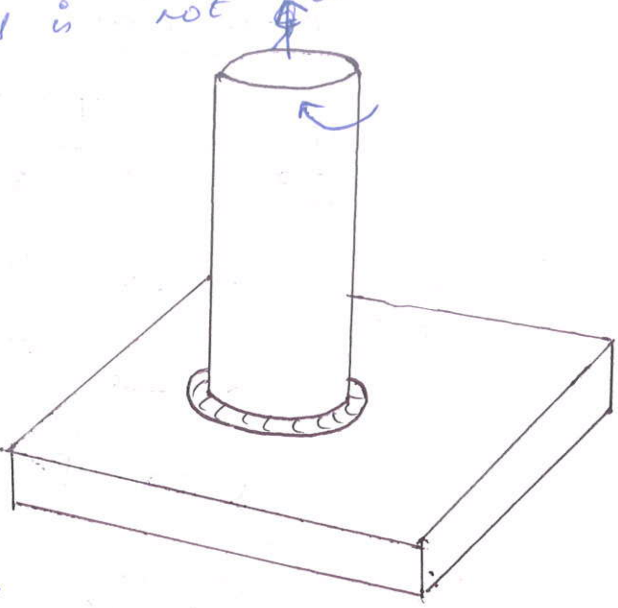
$$\tau_{max} = \sqrt{\tau_1^2 + \tau_2^2 + 2\,\tau_1\,\tau_2 \cos\theta}$$

where $\cos\theta = \frac{GB}{r}$



Axial loading Problems

(1) A 50 mm diameter solid shaft is welded to a flat plate by 10 mm fillet weld as shown in figure. Find the maximum Torque that the welded joint can sustain if the maximum shear stress intensity in the weld material is not to exceed 80 MPa.



gn data:

$d = 50\text{ mm}$

$h = 10\text{ mm}$

$\tau_{max} = 80\text{ MPa}$

$= 80\text{ N/mm}^2$

From PSGDB P.no 11.3

$$\tau_{max} = \frac{2.83\, T}{h d^2 \pi}$$

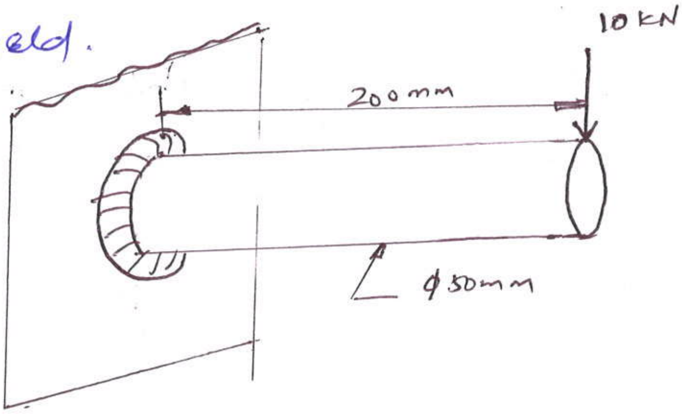
$$80 = \frac{2.83 \times T}{10 \times 50^2 \times \pi}$$

$$T = 2.22 \times 10^6\ \text{N-mm}$$

$$\boxed{T = 2.22 \times 10^3\ \text{N-m}}$$

Eccentric loading Problems

(1) A 50 mm diameter solid shaft is welded to a flat plate as shown in fig. If the size of weld is 15 mm, find the maximum normal and maximum shear stress in the weld.



given data:

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

$e = 200\text{ mm}$

$h = 15\text{ mm}$

$d = 50\text{ mm}$

step 1 direct shear stress (τ)

$$\tau = \frac{P}{\text{Area of weld}}$$

$$= \frac{P}{\pi d \times t} = \frac{P}{\pi d \times 0.707 h}$$

$$= \frac{10 \times 10^3}{\pi \times 50 \times 0.707 \times 15}$$

$$\boxed{\tau = 6 \ N/mm^2}$$

step 2: Bending stress (σ_b)

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

Bending moment $M = Pe = 10 \times 10^3 \times 200$

$$\boxed{M = 2 \times 10^6 \ N\text{-}mm}$$

PSGDB P.NO 11.6

For circular weld

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

$$= \frac{\pi}{4} d^2 \times 0.707 h$$

$$= \frac{\pi}{4} \times 50^2 \times 0.707 \times 15$$

$$\boxed{Z = 20823 \ mm^3}$$

$$\sigma_b = \frac{M}{Z} = \frac{2 \times 10^6}{20823}$$

$$\boxed{\sigma_b = 96 \ N/mm^2}$$

Step 3: Maximum Normal stress $(\sigma_t)_{max}$

$$(\sigma_t)_{max} = \frac{1}{2}\left[\sigma_b + \sqrt{\sigma_b^2 + 4\tau^2}\right]$$

$$= \frac{1}{2}\left[96 + \sqrt{96^2 + 4 \times 6^2}\right]$$

$$\boxed{(\sigma_t)_{max} = 96.4 \text{ N/mm}^2}$$

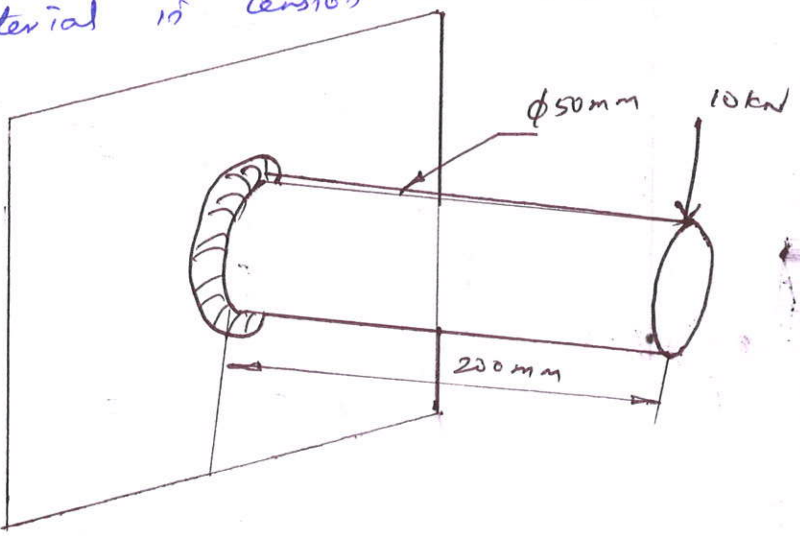
Step 4: Maximum shear stress (τ_{max})

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

$$= \frac{1}{2}\sqrt{96^2 + 4 \times 6^2}$$

$$\boxed{\tau_{max} = 48.37 \text{ N/mm}^2}$$

(2) A 50mm diameter solid shaft is welded at its one edge to the vertical side of rectangular pillar by an all round fillet weld. A load of 10kN is applied at the free end of the shaft which is at a distance of 200mm from the fixed end. Find the size of weld, assuming the permissible stress of weld material in tension is 94 MPa.



given date:

$d = 50$ mm

$e = 200$ mm

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

$(\sigma_t)_{max} = 94$ MPa
$= 94$ N/mm^2

Step 1: direct shear stress (τ)

$$\tau = \frac{P}{\text{Area of weld}}$$

$$= \frac{P}{\pi d t} = \frac{P}{\pi d \times 0.707h}$$

$$= \frac{10 \times 10^3}{\pi \times 50 \times 0.707h}$$

$$\boxed{\tau = \frac{90}{h}}$$

Step 2: bending stress (σ_b)

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

Bending moment $M = Pe$

$$= 10 \times 10^3 \times 200$$

$$\boxed{M = 2 \times 10^6 \text{ N-mm}}$$

From PSGDB P. no 11.6

For circular section

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

$$= \frac{\pi}{4} d^2 \times 0.707h$$

$$= \frac{\pi}{4} \times 50^2 \times 0.707h$$

$$\boxed{Z = 1388h}$$

$$\sigma_b = \frac{M}{Z} = \frac{2 \times 10^6}{1388h}$$

$$\boxed{\sigma_b = \frac{1441}{h}}$$

step 3 Find size of weld (h)

given that

$$(\sigma_t)_{max} = \frac{1}{2}\left[\sigma_b + \sqrt{\sigma_b^2 + 4\tau^2}\right]$$

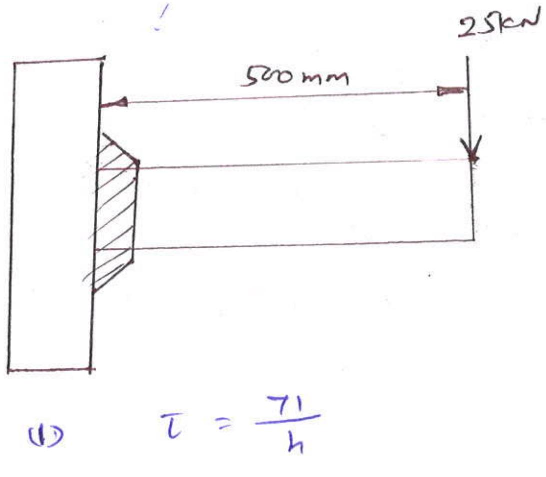
$$94 = \frac{1}{2}\left[\frac{1441}{h} + \sqrt{\left(\frac{1441}{h}\right)^2 + 4\times\left(\frac{90}{h}\right)^2}\right]$$

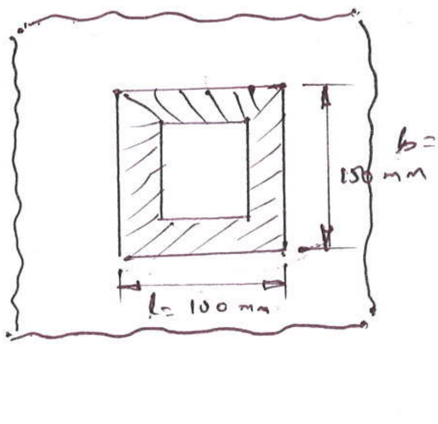
$$188 = \frac{1441}{h} + \frac{1452}{h}$$

$$188 = \frac{2893}{h}$$

$$\boxed{h = 15.4\ mm}$$

(3) A rectangular c-s bar is welded to a support by means of fillet weld as shown in figure. Determine the size of weld, if the permissible shear stress in the weld is limited to 75 MPa.



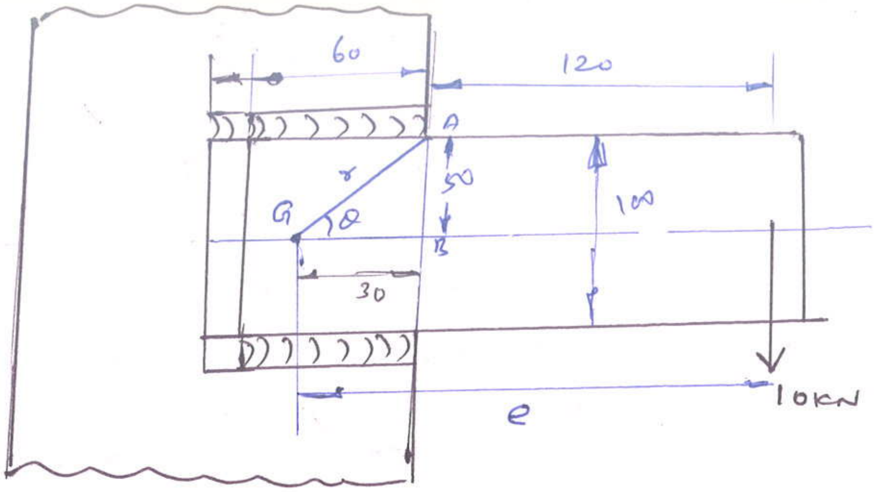


(i) $\tau = \frac{71}{h}$

(ii) $\sigma_b = \frac{786}{h}$

(iii) $h = 5.33\ mm$

(4) A bracket as shown in fig carries a load of 10 kN. Find the size of weld if the allowable shear stress is ~~equal~~ not to exceed 75 N/mm² (13)



given data

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

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

From diagram
$e = 120 + 30$

$\boxed{e = 150\ mm}$

$l = 60mm$

Step 1: direct shear stress (τ_1)

$$\tau_1 = \frac{P}{\text{Area of weld}}$$

$$= \frac{P}{2tl}$$

$$= \frac{P}{2 \times 0.707 h \times l}$$

$$= \frac{10 \times 10^3}{2 \times 0.707 \times h \times 60}$$

$$\boxed{\tau_{1} = \frac{118}{h}}$$

Step 2 secondary shear stress (τ_2)

$$\tau_2 = \frac{T r}{J}$$

$$T = Pe = 10 \times 10^3 \times 150$$

$$\boxed{T = 1.5 \times 10^6\ N\text{-}mm}$$

From diagram $r = \sqrt{GB^2 + AB^2}$

$r = \sqrt{30^2 + 50^2}$

$$\boxed{r = 58.3 \text{ mm}}$$

P. No 11.3

$$J = \left(\frac{b^3 + 3bd^2}{6} \right) t$$

$$= \left(\frac{60^3 + 3 \times 60 \times 100^2}{6} \right) \times 0.707 h$$

$$\boxed{J = 237552 h}$$

$$\tau_2 = \frac{T r}{J} = \frac{1.5 \times 10^6 \times 58.3}{237552 h}$$

$$\boxed{\tau_2 = \frac{368}{h}}$$

step 3: ~~Result~~ Find size of weld (h)

From diagram $\cos\theta = \frac{GB}{r} = \frac{30}{58.3}$

$$\boxed{\cos\theta = 0.515}$$

$$\tau_{max} = \sqrt{\tau_1^2 + \tau_2^2 + 2 \tau_1 \tau_2 \cos\theta}$$

$$\tau_{max}^2 = \tau_1^2 + \tau_2^2 + 2 \tau_1 \tau_2 \cos\theta$$

$$75^2 = \left(\frac{118}{h}\right)^2 + \left(\frac{368}{h}\right)^2 + 2 \times \left(\frac{118}{h}\right)\left(\frac{368}{h}\right) \times 0.515$$

$$\boxed{h = 6 \text{ mm}}$$

⑤ A plate of 200mm width and 600mm long is (14)
welded to a vertical plate by placing it on the vertical plate to form a cantilever with projecting length of 480mm and overlap b/w the plates as 120mm. A vertical fillet weld is done on all three sides. A vertical load 30 kN is applied at the free end of the cantilever plate $\|^{el}$ to its width of 200mm. If the allowable weld stress is 95 MPa, Determine the weld size.

Solution:

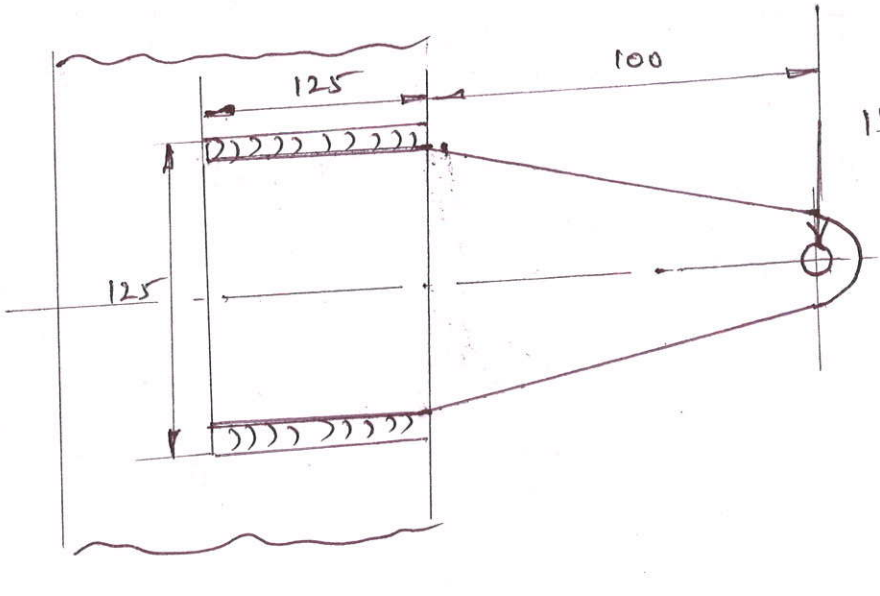
$$\tau_1 = \frac{96.4}{h}$$

$$\tau_2 = \frac{854}{h}$$

$$h = 10\,mm$$

⑥ A bracket is welded to a column as shown in fig. determine the size of weld, If the permissible shear stress in the weld is $80\,N/mm^2$

Solution.



$$\tau_1 = \frac{85}{h}$$

$$\tau_2 = \frac{234}{h}$$

UNIT - IV

SPRING

* It is an elastic member which deflects under the action of load and regain its original shape after the load is removed.

Function of spring

* To reduce the effect of shock
* To measure forces in spring balance.
* To store energy
* To apply force and to control motion

Types of springs :

* Helical spring
  * open coiled Helical spring
  * closed coiled Helical spring
* leaf spring
  * Full elliptical leaf spring
  * Semi elliptical
* spiral spring
* Disc spring
* conical spring

DESIGN PROCEDURE FOR CLOSED COILED HELICAL SPRING:

step 1 wire diameter of the spring

only one axial load

(i) case (i)

From PSGDB P. No 7.100

$$\tau = K_s \frac{8 P C}{\pi d^2}$$

$K_s \rightarrow$ Wahl stress factor [P. No 7.100]

$P \rightarrow$ Maximum load

$\tau \rightarrow$ Maximum shear stress

$C \rightarrow$ spring Index

From the above relation find 'd' and convert into standard value. from ~~13.1~~ P.NO 13.1 then find 'D'

case (ii) Varying load to be given

Varying load means P_{min} to P_{max}

From PSGDB P.NO 7.100

$$\tau = K_s \frac{8 P_{max} C}{\pi d^2}$$

From the above relation Find 'd' and convert into standard value from PSGDB P.no 13.1

then Find 'D'

case (iii) varying load and endurance stress is given

From PSGDB P.NO ~~7.100~~ 7.102

$$\tau_a = \frac{8 K_s P_a C}{\pi d^2}$$

From the above relation Find 'τ_a'

$$\tau_m = 8 \frac{K_{sh} P_m C}{\pi d^2}$$

From the above relation Find 'τ_m'

From the

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

By using the above the relation find 'd' and convert into standard diameter from P.NO 13.1

Step 2: No of turns (n) & Total no of turns (n')

From PSGDB P. No 7.100

$$y_{max} = \frac{8 P_{max} C^3 n}{G d}$$

From the above relation find 'n'

From PSGDB P. No 7.101 & ~~select~~ select suitable formula for Total no of coils (n')

Step 3: Solid length (Ls)

From PSGDB P. No 7.101 select suitable conditions. Formula as per the Ls

Step 4: Free length (Lf)

$$L_f = L_s + y_{max} + 15\% \, y_{max}$$

where $y_{max} \rightarrow$ Maximum deflection

Step 5: Pitch of the coil (P)

$$P = \frac{\cancel{8}\, L_f}{n' - 1}$$

Step 6: To check the spring avoid buckling

From PSGDB P. No 7.101

$$\frac{L_f}{D} < 3$$

$\frac{L_f}{D} > 3 \rightarrow$ spring must be suitably guided.

(1) Design a helical compression spring to carry a load of 1.5 kN with a deflection of 40 mm. Spring index = 5. Allowable shear stress is 400 N/mm² & $G = 8 \times 10^{10}$ N/m².

Given data:

$P = 1.5\ kN = 1.5 \times 10^3\ N$

$y = 40\ mm$

$C = 5 \Rightarrow \frac{D}{d} = 5 \Rightarrow \boxed{D = 5d}$

$\tau_{max} = 400\ N/mm^2$

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

Step 1 wire diameter of the spring

From PSGDB p. no 7.100

$$\tau_{\cancel{max}} = K_s \frac{8 P_{\cancel{max}} C}{\pi d^2}$$

p. no 7.100

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

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

$$\boxed{K_s = 1.3105}$$

$$\tau = K_s \frac{8PC}{\pi d^2}$$

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

$d = 7.91\ mm$

From PSGDB P. no 13.1

$$\boxed{d = 8.23 mm}$$

Gn that $D = 5d = 5 \times 8.23$

Mean dia of coil $\boxed{D = 41.15 mm}$

Step 2: No of turns (n) & Total no of turns (n')

From PSGDB P no 7.100

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

$$40 = \frac{8 \times 1.5 \times 10^3 \times 5^3 \times n}{8 \times 10^4 \times 8.23}$$

$n = 17.56$

$$\boxed{n \simeq 18 \cancel{\text{turns}}}$$

From PSGDB P. no 7.101

For squared and ground end condition

$$n' = n + 2$$

$$= 18 + 2$$

$$\boxed{n' = 20}$$

Step 3: Solid length (L_s)

From PSGDB P. no 7.101

For squared and ground end condition.

$$L_s = nd + 2d$$

$$= (18 \times 8.23) + 2(8.23)$$

$$\boxed{L_s = 164.6 mm}$$

Step 4: Free length (L_f)

$$L_f = L_s + y_{max} + 15\% \text{ of } y_{max}$$

$$= 164.6 + 40 + (40 \times 15\%)$$

$$= 164.6 + 40 + (0.15 \times 40)$$

$$\boxed{L_f = 210.6\,mm}$$

Step 5: Pitch of the coil (P)

$$P = \frac{L_f}{n'-1} = \frac{210.6}{20-1}$$

$$P = 11.08\,mm$$

Step 6: Check to avoid buckling

From PSGDB P. no 7.101

$$\frac{L_f}{D} = \frac{210.6}{41.15}$$

$$\frac{L_f}{D} = 5.12 > 3$$

$\therefore$ The spring must be suitably guided.

---

② Design a closed coil Helical spring for a service load ranging from 2.5kN to 3kN. The deflection for this load ranges is 6 mm. Use a spring index of 5. Take the yield shear strength as 700 N/mm^2 and modulus of rigidity as 8×10^4 N/mm^2. Also factor of safety is not be less than 1.3. Also check the spring for buckling given data:

$P_{min} = 2.5 \text{ kN} = 2.5 \times 10^3 \text{ N}$

$P_{max} = 3 \text{ kN} = 3 \times 10^3 \text{ N}$

$y = 6 \text{ mm}$ (For this load ranges. ie 500 N)

$C = 5 \Rightarrow \frac{D}{d} = 5 \Rightarrow \boxed{D = 5d}$

$\tau_y = 700 \text{ N/mm}^2$

$G = 8 \times 10^4 \text{ N/mm}^2$

Factor of safety $= 1.3$

W.K.T $\quad F.O.S = \frac{\tau_y}{\tau_{max}}$

$$1.3 = \frac{700}{\tau_{max}}$$

$$\boxed{\tau_{max} = 539 \text{ N/mm}^2}$$

step 1 diameter of the wire

From PSG DB P. no 7.100

$$\tau_{max} = K_s \frac{8 P_{max} C}{\pi d^2}$$

P no 7.100

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

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

$$\boxed{K_s = 1.3105}$$

$$539 = 1.3105 \times \frac{8 \times 3 \times 10^3 \times 5}{\pi d^2}$$

$$\boxed{d = 9.64 \text{ mm}}$$

From PSGDB P. no 13.1

Suitable diameter of wire

$$\boxed{d = 10.16 \text{ mm}}$$

In that $D = 5d = 5 \times 10.16$

$$\boxed{D = 50.8 \text{ mm}}$$

Step 2: No of turns (n) & Total no of turns (n')

From PSGDB P. no 7.100

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

$$6 = \frac{8 \times 500 \times 5^3 \times n}{8 \times 10^4 \times 10.16}$$

$$n = 9.75$$

$$\boxed{n \simeq 10}$$

From PSGDB P. no 7.101

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

$$\boxed{n' = 12}$$

Step 3: Solid length (L_s)

From PSGDB P. no 7.101

$$L_s = nd + 2d$$

$$= (10 \times 10.16) + (2 \times 10.16)$$

$$\boxed{L_s = 121.92 \text{ mm}}$$

Step 4: Free length (L_f)

$$L_f = L_s + y_{max} + 0.15\, y_{max}$$

$$y_{max} = \frac{6}{500} \times 3\times10^{3}$$

$$\boxed{y_{max} = 36\,mm}$$

$$L_f = L_s + y_{max} + 15\%\text{ of } y_{max}$$

$$= 121.92 + 36 + (15\% \times 36)$$

$$= 121.92 + 36 + (0.15 \times 36)$$

$$\boxed{L_f = 163.32\,mm}$$

Step 5: pitch of the coil (p)

$$p = \frac{L_f}{n'-1} = \frac{163.32}{12-1}$$

$$\boxed{p = 14.85\,mm}$$

Step 6: check to avoid buckling

P. NO 7.101

$$\frac{L_f}{D} = \frac{163.32}{50.8} = 3.21 > 3$$

∴ The spring must be suitably guided.

---

(3) A helical compression spring made of oil tempered carbon steel is subjected to a load of which varies from 400 N to 1000 N. The spring index as 6 and the design factor of safety is 1.25. If the yield stress in shear is 770 MPa and endurance stress is shear is 350 MPa Find (i) size of the spring wire

x mean diameter of the spring (ii) No of turns in the spring (iii) free length of the spring

The compression of the spring at the maximum load is 30mm. The modulus of rigidity for the spring material may be taken as 80 kN/mm².

Given data:

$P_{min} = 400\ N$

$P_{max} = 1000N$

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

$F.O.S = 1.25$

$\tau_y = 770\ N/mm^2$

$\tau_e = 350\ N/mm^2$

$y = 30mm$ [for max load, ie 1000 N]

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

step 1 diameter of wire (d) x mean diameter (D)

From PSGDB Pn No 7.102

$$\tau_a = \frac{8K_s P_a C}{\pi d^2}$$

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

$$= \frac{4(6)-1}{4(6)-4} + \frac{0.615}{6}$$

$$\boxed{K_s = 1.2525}$$

$$P_a = \frac{P_{max} - P_{min}}{2} = \frac{1000 - 400}{2}$$

$$\boxed{P_a = 300N}$$

$$\tau_a = \frac{8 \times 1.2525 \times 300 \times 6}{\pi d^2}$$

$$\boxed{\tau_a = \frac{5741}{d^2}}$$

From PSGDB P. no 7.102

$$\tau_m = \frac{8\ K_{sh}\ P_m\ C}{\pi d^2}$$

From PSGDB P. no 7.102

~~Ksh~~ For $C = 6$

$$\boxed{K_{sh} = 1.15}$$

$\therefore$

$$P_m = \frac{P_{max} + P_{min}}{2} = \frac{1000 + 400}{2}$$

$$\boxed{P_m = 700\ N}$$

$$\tau_m = 8 \times \frac{1.15 \times 700 \times 6}{\pi d^2}$$

$$\boxed{\tau_m = \frac{12300}{d^2}}$$

From PSGDB P. no 7.102

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

$$\frac{1}{1.25} = \frac{\left(\frac{12300}{d^2} - \frac{5741}{d^2}\right)}{770} + \frac{2 \times \left(\frac{5741}{d^2}\right)}{350}$$

$$\frac{1}{1.25} = \frac{6559}{770 d^2} + \frac{11482}{350 d^2}$$

$d = 7.19\,mm$

From PSGDB P. No 13.1

Standard dia of wire $\boxed{d = 7.62\,mm}$

Given that

$$D = 6d = 6 \times 7.62$$

$$\boxed{D = 45.72\,mm}$$

Step 2: No of turns (n)

From PSGDB P. No 7.100

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

$$300 = \frac{8 \times 1000 \times 6^3 \times n}{80 \times 10^3 \times 7.62}$$

$$n = 10.58$$

$$\boxed{n \simeq 11}$$

From PSGDB P. No 7.101

Total no of coils $n' = n + 2$

$= 11 + 2$

$$\boxed{n' = 13}$$

Step 3: Solid length of spring (L_s)

From PSGDB P. No 7.101

$$L_s = nd + 2d$$

$$= (11 \times 7.62) + 2(7.62)$$

$$\boxed{L_s = 99.06\,mm}$$

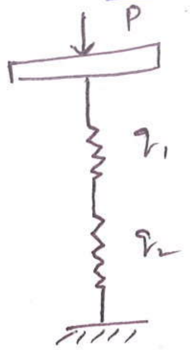
Step 4: Free length of spring (L_f)

$$L_f = L_s + y_{max} + 0.15\, y_{max}$$

$$= 99.06 + 30 + (0.15 \times 30)$$

$$\boxed{L_f = 133.56\,mm}$$

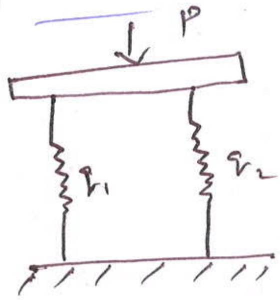
Springs are connected in series



$$\frac{1}{q} = \frac{1}{q_1} + \frac{1}{q_2}$$

$$y = y_1 + y_2$$

Springs are connected in parallel



$$q = q_1 + q_2$$

$$P = P_1 + P_2$$

where $q \rightarrow$ spring rate $= \frac{P}{y} = \frac{\text{load}}{\text{deflection}}$

LEAF (or) LAMINATED SPRING

From PSG DB P. no 7.104

① Bending stress $\sigma_b = \frac{6PL}{nbt^2}$

$P \rightarrow$ Load on the spring

$L \rightarrow$ length of the spring

$n \rightarrow$ Number of leaves

$b \rightarrow$ width of leaf

$t \rightarrow$ thickness of leaf.

(2) Deflection $y = \dfrac{6PL^3}{Enbt^3}$

3 Spring with extra full length leaves

$$\sigma_{bg} = \frac{12PL}{bt^2(3n_e + 2n_g)}$$

$$\sigma_{be} = \frac{18PL}{bt^2(3n_e + 2n_g)}$$

$$y = \frac{12PL^3}{bt^3(3n_e + 2n_g)}$$

Effective length $2L$ = Total length − central band width.

(1) Design a leaf spring for the following specifications. Total load = 140 kN; No of spring supporting the load = 4; Maximum Number of leaves = 10; span of the spring = 1000 mm, permissible deflection = 80 mm. Take E = 200 kN/mm² and allowable stress in spring material as 600 MPa.

(2) A truck spring has 12 number of leaves two of which are full length leaves. The spring supports are 1.05 m apart and central band 85 mm wide. The central load is to be 5.4 kN with a permissible stress of 280 MPa. Determine the thickness, width of steel spring leaves. The ratio of total depth to width of the spring is '3'. Also determine the deflection of the spring

FLY WHEEL

* It is a device which stores energy during the Period when supply of energy is more than the requirement and release it during the period when requirement of energy is more than supply.

COEFFICIENT OF FLUCTUATION OF SPEED (K_S)

* It is defined as the ratio of the maximum fluctuation speed to the mean speed

From PSGDB P. no 7.120

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

$N_1 \rightarrow$ Maximum speed
$N_2 \rightarrow$ Minimum speed
$N \rightarrow$ mean speed

Maximum fluctuation of energy (ΔE)

* It is defined as the difference b/w the maximum energy and minimum energy.

Coefficient of fluctuation of energy (C_E)

* It is defined as the ratio of maximum fluctuation of energy to the Workdone per cycle.

$$* \quad C_E = \frac{\Delta E}{W.D/cycle.}$$

where $W.D/cycle = T_{mean} \times \theta.$

$\theta = 2\pi$; steam engine, Two stroke
$= 4\pi$; Four stroke

$$W.D/cycle = \frac{60P}{n}$$

$P \rightarrow$ Shaft power in watts
$n \rightarrow N$; steam engine, Two stroke
$N/2$; Four stroke.

Design Procedure of flywheel.

step 1 mass of the flywheel.

From PSGDB P. no 7.120

$$\Delta E = I K_s \omega^2$$

where $I \rightarrow$ moment of Inertia $[I = mk^2]$

$m \rightarrow$ mass of the flywheel

$k \rightarrow$ radius of gyration [P.no 7.120]

$\omega \rightarrow$ Angular speed $\left[\omega = \frac{2\pi N}{60}\right]$

From the above relation find 'm'

step 2: cross sectional dimensions of the flyahead

From PSGDB P.no 7.120

$$W = \pi D b h \gamma$$

$$\boxed{m = \pi D b h \rho}$$

From the above relation find 'b' & 'h'

step 3: Diameter and Length of hub

$$P = \frac{2\pi N T_{mean}}{60}$$

From the above relation find T_{mean}

If service factor is not given, then

$$T_{max} = 2 T_{mean}$$

$$T_{max} = \frac{\pi}{16} \tau d_1^3$$

From the above relation find 'd_1'

Diameter of the hub $d = d_1$

Length of the hub $L = 2d_1$

Step 4. Cross-sectional dimension of the elliptical Arm

From PSGDB P. no 7.120

$$\text{Bending stress } \sigma_b = \frac{T(D-d)}{n Z_{yy} D}$$

σ_b is selected from P. No 7.120

$n \rightarrow$ no of Arms (P. no 7.120)

From the above relation, Find dimension of elliptical Arm

(1) ~~Step~~ The turning moment diagram for a petrol engine is drawn to the following scales: Turning moment 1mm = 5N.m ; crank angle 1mm = 1°. The turning moment diagram repeats itself at every half revolution of the engine and the areas above and below the mean turning moment line taken in order are 295, 685, 40, 340, 960, 270 mm². Determine the mass of 300mm diameter fly wheel rim, when the coefficient of fluctuation of speed is 0.3% and the engine runs at 1800 rpm. Also determine the cross section of the rim. when the width of the rim is twice the thickness. Assume density of the rim material as 7250 kg/m³

Gn data:

$D = 300\text{ mm} = 0.3\text{ m}$

$K_S = 0.3\% = 0.003$

$N = 1800\text{ rpm}$

$b = 2h$

$\rho = 7250\text{ kg/m}^3$

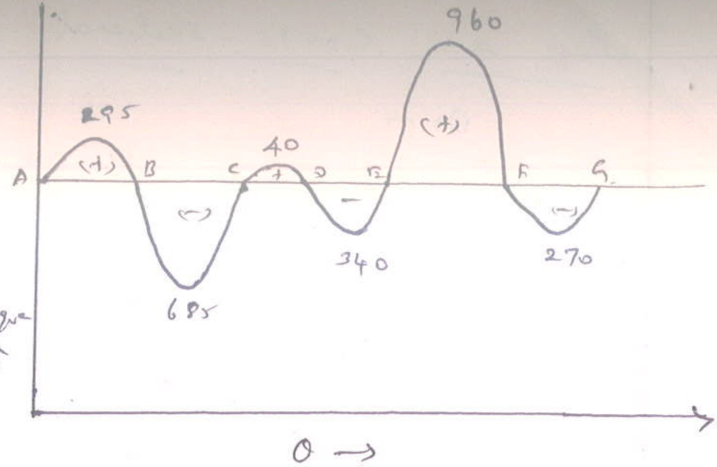
scale: $1\text{ mm} = 5\text{ N-m}$; $1\text{ mm} = 1^\circ$

$1\text{ mm} = 1 \times \frac{\pi}{180}$

$$1\text{ mm}^2 = 5 \times \frac{\pi}{180} = 0.087\text{ N-m}$$

$$\boxed{1\text{ mm}^2 = 0.087\text{ N-m}}$$

(1) mass of fly wheel.



Energy at A = E

Energy at B = E + 295

Energy at C = E + 295 − 685

= E − 390

Energy at D = E − 390 + 40

= E − 350

Energy at E = E − 350 − 340

= E − 690

Energy at F = E − 690 + 960

= E + 270

Energy at G = E + 270 − 270

= E

maximum energy = E + 295

Minimum energy = E − 690

ΔE = Maximum energy − Minimum energy

= (E + 295) − (E − 690)

= 985 mm^2

$\Delta E = 985 \times 0.087$

$\boxed{\Delta E = 86 \text{ N-m}}$

$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 1800}{60} = 188.5$ rad/sec

From PSGDB P. no 7.120

$\cancel{\Delta E} = I \cancel{K_s} \omega^2$

$\Delta E = m k^2 K_s \omega^2$

From PSGDB P. no 7.120

$k^2 = \frac{D^2}{4} = \frac{0.3^2}{4} = 0.0225$

$\boxed{k^2 = 0.0225}$

From PSGDB P. No 7.120

$\Delta E = I \, k_s \, \omega^2$

$\Delta E = m k^2 \, k_s \, \omega^2$

$86 = m \times 0.0225 \times 0.003 \times 188.5^2$

$\boxed{m = 35.86 \text{ kg}}$

(ii) Dimension of the rim

P. No 7.120

$W = \pi D b h \gamma$

$m = \pi D b h \rho$

$35.86 = \pi \times 0.3 \times 2h \times h \times 7250$

$h = 0.051 \text{ m}$

$\boxed{h = 51 \text{ mm}}$

Given that $b = 2h = 2 \times 51$

$\boxed{b = 102 \text{ mm}}$

(2) A single cylinder double acting steam engine develops 150 kW at a mean speed of 80 rpm. The coefficient of fluctuation of energy is 0.1 and fluctuation of speed is $\pm 2\%$ of the mean speed. If the mean diameter of the flywheel rim is 2m and the hub & spokes provide 5% of the rotational inertia of the flywheel, find the mass of the flywheel and cross sectional area of the rim. Assume density of the flywheel material as 7200 kg/m³

given data:

$P = 150 \text{ kW} = 150 \times 10^3 \text{ W}$

$N = 80 \text{ rpm}$

$C_E = 0.1$

$K_s = \pm 2\% = 4\% = 0.04$

$D = 2m$

Energy spend $\left.\begin{matrix}\text{to hub} \\ \text{\& spokes}\end{matrix}\right] = 5\%$

$\rho = 7200 \, kg/m^3$

(i) mass of the flywheel.

W.K.T

$$W.D/cycle = \frac{60P}{n}$$

For steam engine. $n = N = 80\,rpm.$

$$W.D/cycle = \frac{60 \times 150 \times 10^3}{80}$$

$$\boxed{W.D/cycle = 112500 \, N\text{-}m}$$

W.K.T

$$\Delta E = C_E \times W.D/cycle$$

$$= 0.1 \times 112500$$

$$= 11250 \, N\text{-}m$$

$$(\Delta E)_{Net} = 95\% \; \Delta E$$

$$= 0.95 \times 11250$$

$$\boxed{(\Delta E)_{net} = 10688 \, N\text{-}m}$$

W.K.T

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 80}{60}$$

$$\boxed{\omega = 8.38 \, rad/sec}$$

From PSGDB P. no 7.120

$$k^2 = \frac{D^2}{4} = \frac{2^2}{4}$$

$$\boxed{k^2 = 1}$$

From PSGDB P. no 7.120

$$(\Delta E)_{net} = I \; k_s \; \omega^2$$

$$(\Delta E)_{net} = m \; k^2 \; k_s \; \omega^2$$

$$10688 = m \times 1 \times 0.04 \times 8.38^2$$

$$\boxed{m = 3805 \text{ kg}}$$

(ii) C-S dimension of rim

From PSGDB P. no 7.120

$$m = \pi D b h \rho$$

$$m = \pi D A \rho \qquad \left[\begin{array}{l} \text{C-S area} \\ A = bh \end{array}\right]$$

$$3805 = \pi \times 2 \times A \times 7200$$

$$\boxed{A = 0.084 \text{ m}^2}$$

(3). The turning moment diagram of a multi cylinder engine, is drawn with a scale of (1 mm = 1°) on the abscissa and (1 mm = 250 N-m) on the co-ordinate. The intercepted area b/w the torque developed by the engine and mean resisting torque of the machine taken in order from one end are −350, +800, −600, +900, −550, +450 & −630 mm². The engine running at a mean speed of 750 rpm. and the coefficient of speed fluctuation is limited to 0.02. A rimmed flywheel made of grey C.I FG200 ($\rho = 7100\ kg/m^3$) is provided. The spokes, hub & shaft are assumed to contribute 10% of of the required amount of Inertia. The rim has rectangular c-s and ratio of width to thickness is 1.5. Determine the dimensions of the rim.

(4) A rim fly wheel is to be designed to store 5 KN-m of energy and to keep the speed within 395 rpm and 405 rpm. The mean rim diameter is limited to 1m. Design the rims and arms of the fly wheel which is used in an engine developing 20KW.

Given data

$\Delta E = 5 KN\text{-}m = 5 \times 10^{3} N\text{-}m$

$N_{min} = 395 rpm$

$N_{max} = 405 rpm$

$D = 1m = 1000 mm$

$P = 20KW = 20 \times 10^{3} W$

mean speed $N = \frac{N_{max} + N_{min}}{2} = \frac{405 + 395}{2}$

$\boxed{N = 400 rpm}$

coefficient of fluctuation speed $K_s = \frac{N_{max} - N_{min}}{N}$

$= \frac{405 - 395}{400}$

$\boxed{K_s = 0.025}$

step 1 mass of the fly wheel

From PSG DB P. NO 7.120

$\Delta E = I K_s \omega^2$

$\Delta E = m k^2 K_s \omega^2$

From PSG DB P.no 7.120 $k^2 = \frac{D^2}{4} = \frac{1^2}{4} = 0.25$

W.K.T $\omega = \frac{2 \pi N}{60} = \frac{2 \pi \times 400}{600}$

$\boxed{\omega = 41.89 \text{ rad/sec}}$

$$\Delta E = m K^2 K_s \omega^2$$

$$5 \times 10^3 = m \times 0.25 \times 41.89^2 \times \cancel{0.0025}\ 0.025$$

$\boxed{\cancel{m = 11.4\,kg}}$ $\boxed{m = 456\,kg}$

<u>step 2</u> <u>C-S dimension of rim</u>

From PSGDB P.no 7.120

$$m = \pi D b h \rho$$

Assume flywheel material is C.I & For C.I

$$\rho = 7200\ kg/m^3$$

From PSGDB P.no 7.120

$$\frac{b}{h} = 2 \implies b = 2h$$

$$m = \pi D b h \rho$$

$$456\ \cancel{kg} = \pi \times 1 \times 2h \times h \times 7200$$

$$h = \cancel{0.0153\,m}\ 0.1004\,m$$

$\boxed{\cancel{h = 15.3\,mm}}$ $\boxed{h = 100.4\,mm}$

W.K.T $b = 2h = 2 \times \cancel{15.3}\ 100.4$

$\boxed{\cancel{b = 30.6\,mm}}$ $\boxed{b = 200.8\,mm}$

<u>step 3</u> <u>Diameter & length of hub</u>

W.K.T $$P = \frac{2\pi N T_{mean}}{60}$$

$$20 \times 10^3 = \frac{2\pi \times 400 \times T_{mean}}{60}$$

$$T_{mean} = 477.5$$

Assume $T_{max} = 2T_{mean} = 2 \times 477.5$

$$T_{max} = 955\ N\text{-}m$$

$$\boxed{T_{max} = 955 \times 10^3\ N\text{-}mm}$$

$$T_{max} = \frac{\pi}{16} \tau d_1^3$$

$$\left[\begin{array}{c} \text{for C.I} \\ \tau = 40 \text{ N/mm}^2 \end{array}\right]$$

$$955 \times 10^3 = \frac{\pi}{16} \times 40 \times d_1^3$$

$$d_1 = 49.54$$

$$\boxed{d_1 = 50 \text{ mm}}$$

diameter of hub $d = d_1$

$$\boxed{d = 50 \text{ mm}}$$

length of hub $l = 2d_1 = 2 \times 50$

$$\boxed{l = 100 \text{ mm}}$$

<u>step 4:</u> <u>C-S</u> <u>dimension</u> <u>of</u> <u>the</u> <u>arm</u>

From PSG DB P. no 7.120

$$\sigma_{b_1} = \frac{M_t (D-d)}{n \, Z_{yy} \, D}$$

For C-I $\sigma_{b_1} = 130 \text{ kgf/cm}^2$ (PSG DB. P. no 7.120)

$$= 13 \text{ N/mm}^2$$

$$13 = \frac{955 \times 10^3 \times (1000 - 50)}{6 \times Z_{yy} \times 1000}$$

$$\boxed{Z_{yy} = 11631 \text{ mm}^3}$$

From PSG DD P. no 7.120

$$a = \sqrt[3]{\frac{64 Z_{yy}}{\pi}} = \sqrt[3]{\frac{64 \times 11631}{\pi}}$$

$$\boxed{a = 62 \text{ mm}}$$

$$c = \frac{a}{2} = \frac{62}{2} = 31 \text{ mm}$$

$$\boxed{c = 31 \text{ mm}}$$

CONNECTING ROD

Design Procedure of connecting rod

Step 1 Load due to gas or steam Pressure

From PSG DB P. No 7.122

$$F_G = \frac{\pi}{4} d^2 p$$

where $d \rightarrow$ diameter of the piston

$p \rightarrow$ steam pressure.

From the above relation Find F_G

Step 2 Inertia force due to reciprocating Parts

From PSG DB P. No 7.122

$$F_I = \frac{R}{g} \omega^2 r \left[\cos\theta + \frac{\cos 2\theta}{(l/r)} \right]$$

$$F_I = m \omega^2 r \left[\cos\theta + \frac{\cos 2\theta}{(l/r)} \right]$$

where $m \rightarrow$ mass of the connecting rod.

$\omega \rightarrow$ Angular velocity

$r \rightarrow$ radius of crank. [stroke length $L = 2r$, $r = \frac{L}{2}$]

$l \rightarrow$ length of connecting rod

$\theta \rightarrow$ crank angle [If it is not given assume $\theta = 0^\circ$]

From the above relation Find F_I

Step 3 Effective force (or) Max. Force on the connecting rod

$$F_{max} = \text{Max of} \left[F_G, F_I \right]$$

Step 4 Size of connecting rod

From PSGDB P.No 6.8

By using Rankines formula.

$$P_c = \frac{a\,\sigma_c}{1 + C\left(\frac{L}{K}\right)^2}$$

where $P_c \rightarrow$ Buckling load $= F_{max} \times$ F.O.S

$a \rightarrow$ Area of C-R [P.No. 7.122]

$\sigma_c \rightarrow$ crushing stress [For mild steel $\sigma_c = 320\ N/mm^2$]

$C \rightarrow$ Rankines constant [P.No. 6.8]

$L \rightarrow$ Length of column [$L = l$]

$K \rightarrow$ Radius of gyration [P.No 7.122]

From the above relation Find size of connecting rod.

Step 5 Bending stress due inertia force

From PSGDB P.No 7.122

$$\sigma_{b_{max}} = \frac{\gamma a l^2 \omega^2 r}{9\sqrt{3}\, g\, Z_{xx}}$$

$$\sigma_{b_{max}} = \frac{\rho a l^2 \omega^2 r}{9\sqrt{3}\, Z_{xx}}$$

where $\rho \rightarrow$ Density of C-R material

$a \rightarrow$ Area of C-R

$l \rightarrow$ length of C-R

$\omega \rightarrow$ Angular velocity

$r \rightarrow$ radius of crank

$Z_{xx} \rightarrow$ section modulus [P. NO 7.122]

From the above relations find $(\sigma_b)_{max}$

---

(1) Determine the dimensions of an I section connecting rod for a petrol engine from the following data. Diameter of the piston = 110 mm, mass of the reciprocating parts = 2 kg, length of connecting rod from centre to centre = 325 mm, stroke length = 150 mm, RPM = 1500 with possible overspeed of 2500, compression ratio = 4 : 1, Maximum explosion pressure = 2.5 N/mm²

given data:

$d = 110\ mm = 0.11 m$

$m = 2 kg$

$l = 325\ mm = 0.325 m$

stroke length $L = 150\ mm \Rightarrow L = 2r$

$150 = 2r$

$r = 75\ mm$

$\boxed{r = 0.075 m}$

$N = 2500$ rpm.

$p = 2.5\ N/mm^2$

step 1 Load due to gas or steam pressure

From PSGDB P. NO 7.122

$$F_G = \frac{\pi}{4} d^2 p$$

$$= \frac{\pi}{4} \times 110^2 \times 2.5$$

$$\boxed{F_G = 23760\ N}$$

Step 2 Inertia force due to reciprocating Parts

From PSG DB P. no 7.122

$$F_I = \frac{R}{g}\omega^2 r\left[\cos\theta + \frac{\cos 2\theta}{(l/r)}\right]$$

$$F_I = m\omega^2 r\left[\cos\theta + \frac{\cos 2\theta}{(l/r)}\right]$$

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 2500}{60} = 261.8 \text{ rad/sec.}$$

θ is not given $\therefore$ $\theta = 0^\circ$

$$F_I = 2 \times 261.8^2 \times 0.075\left[\cos 0^\circ + \frac{\cos(2\times 0^\circ)}{\left(\frac{0.325}{0.075}\right)}\right]$$

$$\boxed{F_I = 12672 \text{ N}}$$

Step 3: Effective force (or) Max. force on C-R

$$F_{max} = \text{Max of } [F_G, F_I]$$

$$\boxed{F_{max} = 23760 \text{ N}}$$

Step 4 Size of connecting rod

From PSG DB P. no 6.8.

$$P_c = \frac{a\,\sigma_c}{1 + c\left(\frac{L}{K}\right)^2}$$

$$P_c = F_{max} \times F.o.s \qquad \left[\begin{array}{l}\text{P. no 7.122} \\ F.o.s = 6\end{array}\right]$$

$$= 23760 \times 6$$

$$\boxed{P_c = 142560 \text{ N}}$$

From PSGDB P.no 7.122 $a = 11t^2$

From PSGDB P.no 6.8 Assume for mild steel

Rankines constant $C = \frac{1}{7500}$

σ_c for mild steel $= 320 \text{ N/mm}^2$

$L \rightarrow$ length of column $= l = 325 \text{ mm}.$

From PSGDB P.no 7.122

$$K^2 = 3.18t^2$$

$$P_c = \frac{a\sigma_c}{1 + C\left(\frac{L}{K}\right)^2}$$

$$P_c = \frac{a\sigma_c}{1 + C\left(\frac{L^2}{K^2}\right)}$$

$$142560 = \frac{11t^2 \times 320}{1 + \frac{1}{7500}\left(\frac{325^2}{3.18t^2}\right)}$$

$$t^4 - 40.5t^2 - 179.8 = 0$$

Assume $t^2 = x$

$$x^2 - 40.5x - 179.8 = 0$$

$$\boxed{x = 44.55}$$

$x = 44.5 \Rightarrow t^2 = 44.5$

$$\boxed{t = 6.8 \text{ mm}}$$

$\therefore$ Height $= 5t = 5 \times 6.8 = 34$ mm

width $= 4t = 4 \times 6.8 = 27.2$ mm.

step 5 Bending stress due to Inertia force

From PSGDB P. No 7.122

$$\sigma_{b_{max}} = \frac{\gamma a l^2 \omega^2 r}{9\sqrt{3} g Z_{xx}} = \frac{\rho a l^2 \omega^2 r}{9\sqrt{3} Z_{xx}}$$

From P. No 7.122, $a = 11t^2 = 11 \times (6.8 \times 10^{-3})^2$

$$\boxed{a = 5.09 \times 10^{-4} m^2}$$

$l = 325 = 0.325$ m.

$\omega = 261.8$ rad/sec

$r = 0.075$ m

W.K.T $Z_{xx} = \frac{I_{xx}}{y} = \frac{\frac{419}{12} t^4}{(5t/2)}$

$Z_{xx} = \frac{419}{12} t^4 \times \frac{2}{5t}$

$Z_{xx} = 14t^3 = 14 \times 0.0068^3$

$$\boxed{Z_{xx} = 4.4 \times 10^{-6} m^3}$$

$$\sigma_{b_{max}} = 7800 \times \frac{5.09 \times 10^{-4} \times 0.325^2 \times 261.8^2 \times 0.075}{9\sqrt{3} \times 4.4 \times 10^{-6}}$$

$= 31428514 N/m^2$

$= 31.4 \times 10^6 N/m^2$

$= 31.4 N/mm^2$

$$\boxed{\sigma_{b_{max}} = 31.4 \text{ MPa}}$$

UNIT – V

BEARING:

* It is the part of the machine which supports the rotating element.

classification of bearings:

* sliding contact bearing
* Rolling contact bearing.

sliding contact bearing

* sliding contact takes place b/w the moving element and fixed element.

Types.

* Journal bearing
* Full Journal bearing
* Partial Journal bearing
* clearance bearing

Rolling contact bearing.

* Rolling motion takes place b/w the moving element and fixed element with aid of ball or roller.

Types of ~~ro~~ Rolling contact bearing

* Ball bearing
* Roller bearing.

Design Procedure of Journal bearing

Step 1 Length of bearing (L)

From PSGDB P. no 7.31 for suitable L/D ratio. From the machinery select $\frac{L}{D}$ ratio. From the $\frac{L}{D}$ ratio find length of bearing (L)

check bearing Pr. (p)

Pressure developed $p = \frac{W}{LD}$

From the above relation Find 'p' check with given range in P. no 7.31

Step 2: Viscosity of oil

From PSGDB P. no 7.31 select suitable $\frac{Zn}{p}$ value & from the relation find viscosity of oil

$Z \rightarrow$ absolute viscosity in cp

$n \rightarrow$ speed in rpm

$p \rightarrow$ Bearing pressure in kgf/cm²

Step 3 coefficient of friction:

From PSGDB P. no 7.34

$$\mu = \frac{33.25}{10^{10}} \left(\frac{Zn}{p}\right) \left(\frac{D}{c}\right) + K$$

$Z \rightarrow$ Absolute viscosity in cP.

$n \rightarrow$ speed in rpm.

p → Bearing pressure in kgf/cm²

D → Diameter of Journal in mm

C → Diametral clearance [P. no 7. 32]

K → constant [From graph in P. no 7. 34]

From the above relation find 'μ'

Step 4 **Heat generated** (H_g)

From PSG DB P. no 7. 34

$$H_g = \mu W V$$

where V → Velocity $\left[V = \frac{\pi D N}{60}\right]$

W → Load in N.

Step 5 **heat dissipated:** (H_d)

From PSG DB P. no 7. 34

$$H_d = \frac{(\Delta t + 18)^2 L D}{(K/1600)}$$

Δt → change in temp $\left(\Delta t = \frac{1}{2}(T_o - T_a)\right)$ (P. no. 7.35)

T_o → operating temperature

T_a → Ambient temperature

L → Length of bearing

D → Diameter of Bearing

K → constant for heat dissipation

step 6 check for Artificial cooling

$H_g < H_d \Rightarrow$ No Artificial cooling is required

$H_g > H_d \Rightarrow$ Artificial cooling is required

mass of oil required for artificial cooling

$$Q = m\, C_p\, dt$$

$Q \rightarrow H_g$ or H_d

$m \rightarrow$ mass of oil

$C_p \rightarrow$ specific heat (1840 to 2100)

(1) Design a journal bearing for a centrifugal pump with the following data : Diameter of the journal = 150 mm ; load on the bearing = 40 kN, speed of the journal = 900 rpm.

given data:

$D = 150$ mm

$W = 40\text{ kN} = 40 \times 10^3$ N

$n = 900$ rpm

step 1 Length of bearing (L)

From PSGDB P. No 7.31, for centrifugal pump

$$\frac{L}{D} = 1 \text{ to } 2$$

$$\frac{L}{D} = 1.5$$

$$\frac{L}{150} = 1.5$$

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

Check bearing Pressure

WKT $p = \frac{W}{LD} = \frac{40 \times 10^3}{225 \times 150}$

$p = 1.185\ N/mm^2$

~~$p = 0.1185\ kgf/cm^2$~~

$p = 11.85\ kgf/cm^2 <$ given range $p = 7 - 14$

Step 2 viscosity of oil.

From PSGDB P.no 7.31, for centrifugal pump

$$\frac{Zn}{p} = 2844.5$$

$$\frac{Z \times 900}{11.85} = 2844.5$$

$$Z = 37.45\ cP.$$

$$\boxed{Z \simeq 40\ cP}$$ [P.no 7.41]

Step 3 coefficient of friction (μ)

From PSGDB P.no 7.34

$$\mu = \frac{33.25}{10^{10}} \left(\frac{Zn}{p}\right)\left(\frac{D}{C}\right) + k$$

From PSGDB P.no 7.32 for centrifugal pump,

Diametral clearance $C = 75$ to 150 microns

$$C = 100 \times 10^{-6}\ m$$

$$\boxed{C = 100 \times 10^{-3}\ mm}$$

From PSGDB P.no 7.34 for $\frac{L}{D} = 1.5$

$$\boxed{k = 0.0024}$$

$$\mu = \frac{33.25}{10^{10}} * \left(\frac{40 \times 900}{11.85}\right) \times \left(\frac{150}{100 \times 10^{-3}}\right) + 0.0024$$

$$\boxed{\mu = 0.0176}$$

Step 4: Heat generated (H_g)

PSGDB P. no 7.34

$$H_g = \mu W V$$

WKT

$$V = \frac{\pi D n}{60} = \frac{\pi \times 150 \times 10^{-3} \times 900}{60}$$

$$\boxed{V = 7.07 \text{ m/sec}}$$

$$H_g = 0.0176 \times 40 \times 10^{3} \times 7.07$$

$$\boxed{H_g = 4977.28 \text{ watts}}$$

Step 5 Heat dissipated: (H_d)

from PSGDB P. no 7.34

$$H_d = \frac{(\Delta t + 18)^2 LD}{(K/1600)}$$

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

$$= \frac{1}{2}(65 - 30)$$

$$\Delta t = 17.5^\circ c$$

From PSGDB P. no 7.35

$K = 775$ for light constructions

$$H_d = \frac{(17.5+18)^2 \times 0.225 \times 0.15}{(775/1600)}$$

$= 88$ watts

step 6 check for Artificial cooling

$H_g > H_d \Rightarrow$ Artificial cooling is required

mass of oil required for Artificial cooling:

$Q = m\, C_p\, dt$

$Q = H_g - H_d = 4977 - 88$

$\boxed{Q = 4889}$

$cp = 1840$ to 2100

$\boxed{C_p = 2000}$

$\boxed{dt = 10^\circ C}$

$Q = m\, C_p\, dT$

$4889 = m \times 2000 \times 10$

$\boxed{m = 0.245\ kg}$

(2) Design a journal bearing for a 49.9 mm diameter journal. It is ground and hardened and is rotating at 1500 rpm in a bearing of diameter and length 50 mm. The inlet temperature of the oil is 65°c. Determine (a) max radial load that the journal can carry (b) power loss.

Gn data:

$d = 49.9\,mm$

$N = 1500\,rpm$

$L = 50\,mm$

$t_0 = 65°c$

step 1 maximum radial load

Given bearing is ground and hardened.

~~machinery~~ Bearing is spindle

∴ from PSGDB p.no 7.31

$p = 0.07\,kgf/cm^2$

$= 0.007\,N/mm^2$

WKT
$$p = \frac{W}{LD}$$

$$0.007 = \frac{W}{50 \times 49.9}$$

$$\boxed{W = 17.465\,N}$$

step 2 viscosity of oil

from PSGDB p.no 7.31,

$$\frac{Zn}{p} = 142231$$

$$\frac{Z \times 1500}{0.07} = 142231$$

$\boxed{Z = 12\ cP}$

Step 3 coefficient of friction (μ)

P.no 7.34.

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

For spindle from PSGDB P.no 7.32

Diameter clearance $C = 50$ to 75 microns

$C = 65 \times 10^{-6}$ m

$C = 65 \times 10^{-3}$ mm

For $\frac{L}{D} = \frac{50}{49.9} = 1$ from PSGDB P.no 7.34

$$\boxed{K = 0.0023}$$

$$\mu = \frac{33.25}{10^{10}} \times \left(\frac{12 \times 1500}{0.07}\right) \times \left(\frac{49.9}{65 \times 10^{-3}}\right) + 0.0023$$

$$\boxed{\mu = 0.66}$$

Step 4 Heat generated (H_g)

From PSGDB P.no 7.34,

$$H_g = \mu W V$$

$$V = \frac{\pi d n}{60} = \frac{\pi \times 49.9 \times 10^{-3} \times 1500}{60}$$

$$\boxed{V = 3.92\ m/sec}$$

Heat generated $H_g = MWV$

$$H_g = 0.66 \times 17.465 \times 3.92$$

$$\boxed{H_g = 45 \text{ Watts}}$$

Step 5 heat dissipated (H_d)

From PSGDB P. no 7.34

$$H_d = \frac{(\Delta t + 18)^2 \, L D}{(K/1600)}$$

$$\Delta t = \frac{1}{2}(t_o - t_a) = \frac{1}{2}(65 - 35)$$
$$= 15^\circ C$$

From PSGDB P.no 7.35 for light construction

$$\boxed{K = 775}$$

$$H_d = \frac{(15 + 18)^2 \times 0.05 \times 0.0499}{(775/1600)}$$

$$\boxed{H_d = 5.61 \text{ watts}}$$

Step 6: check for artificial cooling

Artificial cooling is required.

$H_g > H_d \Rightarrow$

Power lost due to friction $= H_g - H_d$
$= 45 - 5.6$
$= 39.4$ watts.

(3) Following data is given for a 360° hydrodynamic bearing: Radial load = 3.2 kN, Journal speed = 1490 rpm, L/d ratio = 1, Unit bearing pressure = 1.3 MPa, Radial clearance = 0.05 mm, viscosity of oil = 25 cP. Assume that the total heat generated in the bearing is carried by the total oil flow in the bearing, calculate the journal diameter, power lost in friction and the temperature rise.

Gn data:

$W = 3.2\ kN = 3.2 \times 10^3\ N$

$n = 1490\ rpm$

$L/d = 1 \Rightarrow \boxed{L = d.}$

$p = 1.3\ MPa = 1.3\ N/mm^2 = 13\ kgf/cm^2$

$C = 0.05\ mm$

$Z = 25\ cP$

Gn that $H_g = H_d$.

(i) Journal diameter:

WKT $\quad p = \dfrac{W}{Ld}$

$$1.3 = \frac{3.2 \times 10^3}{d \times d}$$

$d = 49.6$

$\boxed{d = 50\ mm}$

(ii) Power lost in friction (or) Heat generated

From PSGDB p.no 7.34 $\quad \frac{L}{d} = 1$

$\boxed{K = 0.0023}$

P no 7.34

$$\mu = \frac{33.25}{10^{10}}\left(\frac{Zn}{p}\right)\left(\frac{D}{c}\right) + K$$

$$= \frac{33.25}{10^{10}} \times \left(\frac{25 \times 1490}{13}\right) \times \left(\frac{50}{0.05}\right) + 0.0023$$

$$\boxed{\mu = 0.0118}$$

WKT $V = \dfrac{\pi d n}{60} = \dfrac{\pi \times 0.05 \times 1490}{60}$

$$\boxed{V = 3.9 \text{ m/sec}}$$

From PSG DB p. no 7.34

$$H_g = \mu W V$$

$$= 0.0118 \times 3.2 \times 10^3 \times 3.9$$

$$\boxed{H_g = 147.26 \text{ watts}}$$

(iii) Temperature rise

Wkt $H_g = H_d$

$$147.26 = \frac{(\Delta t + 18)^2 L D}{K/1600}$$

$$147.26 = \frac{(\Delta t + 18)^2 \times 0.05 \times 0.05}{775/1600}$$

Rolling contact bearing

Design procedure of rolling contact bearing

Step 1 dynamic equivalent load (P)

From PSGDB P.no 4.2

$$P = (X F_r + Y F_a) S$$

[from PSGDB P.no 4.2]

(i) where $S \rightarrow$ service factor

(ii) Find X & Y

From PSGDB 4.12 to 4.36

From the given diameter select C_0 value.

then find $\frac{F_a}{C_0}$ value.

By the use $\frac{F_a}{C_0}$ value Find 'e'

from the value of $\frac{F_a}{F_r}$ & e, Find X & Y

(iii) Find $P = \cancel{XF_r}$ from the formula

$$P = (X F_r + Y F_a) S$$

Step 2 calculate dynamic load capacity (C)

(i) From PSGDB P.no 4.6 & 4.7 select $\frac{C}{P}$ ratio with the aid of speed in rpm and Life in hrs

(ii) From $\frac{C}{P}$ ratio, find 'c' check with the tabulated value.

(1) Select a single row deep groove ball bearing for a radial load of 4000 N and an axial load of 5000 N, operating at a speed of 1600 rpm. for an average life of 5 years at 10 hours per day. Select from series 64

Given data:

$F_r = 4000\ N$

$F_a = 5000\ N$

$N = 1600\ rpm$

Life = 5 years 10 hrs per day

$L_H = 5 \times 300 \times 10$

$\boxed{L_H = 15000\ hrs}$

1 yr = 300 days
1 yr = 52 weeks

SKF 6415

min 60

Step 1 dynamic equivalent load

From PSGDB p. no 4.2

$$P = (X F_r + Y F_a) S$$

(i) Service factor (S)

Assume Rotary m/c with no impact

$S = 1.1$ to 1.5

$\boxed{S = 1.2}$

(ii) Find X & Y

From PSGDB p. no 4.15, given that series 64

(2) Select a suitable ball bearing for a drilling machine spindle of diameter 40mm rotating at 3000 rpm.. It is subjected to a radial load of 2000N and axial thrust of 1000N. It is to work for 45 hrs a week for one year.

Given data:

$\boxed{SKF\,6308}$

$d = 40\,mm$

$N = 3000\,rpm$

$F_r = 2000\,N$

$F_a = 1000\,N$

$L_H = 1 \times 52 \times 45$

$\boxed{L_H = 2340\,hrs}$

Step 1 dynamic equivalent load (P)

From PSG DB P.no 4.2

$P = (X F_r + Y F_a) S$

(i) Service factor (S)

From PSG DB p.no 4.2

For spindle → Rotary m/c $S = 1.1$ to 1.5

$\boxed{S = 1.2}$

(ii) Find X & Y

From PSG DB P.no 4.12 to 4.36

For $d = 40\,mm$ $C_0 = 9800\,N$, $C = 13200\,N$

$\frac{F_a}{C_0} = \frac{1000}{9800} = 0.10$; $\frac{F_a}{F_r} = \frac{1000}{2000} = 0.5 > e.$

$\therefore e = 0.29$

$\therefore X = 0.56,\ Y = 1.5$

$P = (0.56 \times 2000 + 1.5 \times 1000) \times 1.2$

$\boxed{P = 3144\,N}$

Step 2 Calculate dynamic load capacity

From PSGDB P.no 4.6

For N = 3000 rpm, L_H = 2340 hrs

$$\frac{C}{P} = 7.81$$

$$\frac{C}{3144} = 7.81$$

$$= 24555 \, N > 13200 \, N.$$

(3). A ball bearing is operating on a work cycle consisting of three parts a radial load of 3000 N at 1440 rpm for one quarter cycle, a radial load of 5000 N at 720 rpm for one half cycle and radial load of 2500 N at 1440 rpm for the remaining cycle. The expected life of the bearing is 10000 hrs. calculate the dynamic load carrying capacity of the bearing

(4) Select a single row deep groove ball bearing with operating cycle listed below, which will have a life of 15000 hrs.

Fraction of cycle	Type of load	Radial (N)	Thrust (N)	Speed (rpm)	Service factor (S)
1/10	Heavy shock	2000	1200	400	3.0
1/10	light shock	1500	1000	500	1.5
1/5	moderate shock	1000	1500	600	2.0
3/5	no shock	1200	2000	800	1.0