srivenkateshwaraa College of Engineering & Technology
(Approved by AICTE, New Delhi & Affiliated to Pondicherry University, Puducherry)
13-A, Pondy - Villupuram Main Road, Ariyur, Puducherry - 605 102.
ASPIRE TO EXCEL

ISO 21001

DEPARTMENT OF MECHANICAL ENGINEERING

SUBJECT NOTES

SUB NAME: DESIGN OF MACHINE ELEMENTS

SUBCODE : MET52 **YEAR & SEM :** III/V

PREPARED BY,

Mr. M.NATARAJ., B.E., M.E., MISTE.(Ph.D)

ASSISTANT PROFESSOR

SVCET

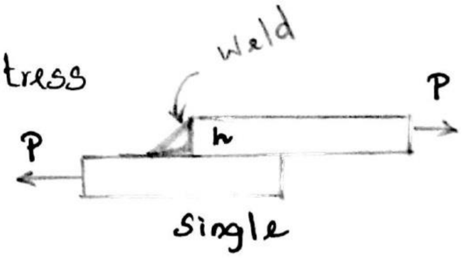
Welding.

Strength of Transverse Fillet Welded joints :

For Single Fillet Weld :

P = Throat Area × Allowable tensile stress

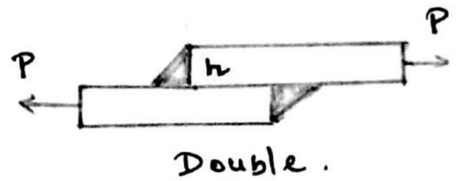
$$P = 0.707 h \times l \times \sigma_t$$



For Double Fillet Weld :

$$P = 2 \times 0.707 h \times l \times \sigma_t$$

$$P = 1.414 h \times l \times \sigma_t$$



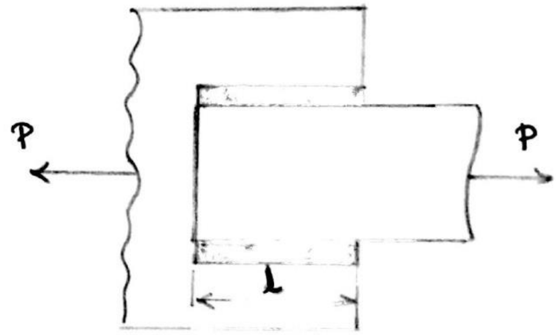
Strength of parallel fillet Welded joints :

For Single Parallel Weld :

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

For Double Parallel Weld :

$$P = 1.414 h \times l \times \tau$$



If there is Combination of single transverse and double parallel fillet Weld :

$$P = 0.707 h \times l_1 \times \sigma_t + 1.414 h \times l_2 \times \tau$$

※ In Welding We have to add 12.5 mm for starting and stopping of Weld run.

Stress Concentration factor for Welded joints.

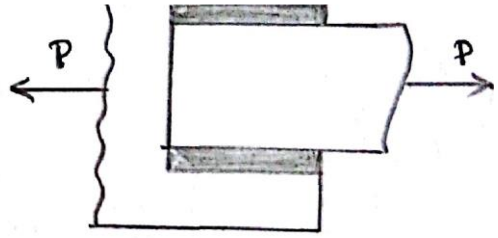
Type of joint	Stress Concentration Factor
Transverse fillet Welds	1.5
Parallel fillet Welds	2.7

A plate 100 mm Wide and 12.5 mm thick is to be Welded to another plate q,,j;q,t, ipitk abq jected to means of Parallel fillet Welds. The plates are the maximum Streload of 50 KN. Find the Length of the Weld so that under static loading and then under fatigue loading. the joint first

Given : Width (b) = * b

$P = 50$ KN = 50 x ‹b ' *_' ^' *'** ' ** *'** '

$P = 1.414\, h \times l \times \tau$



$50 \times 10^3 = 1.414 \times 12.5 \times l \times 56$

$l = \frac{50 \times 10^3}{qqb} =$

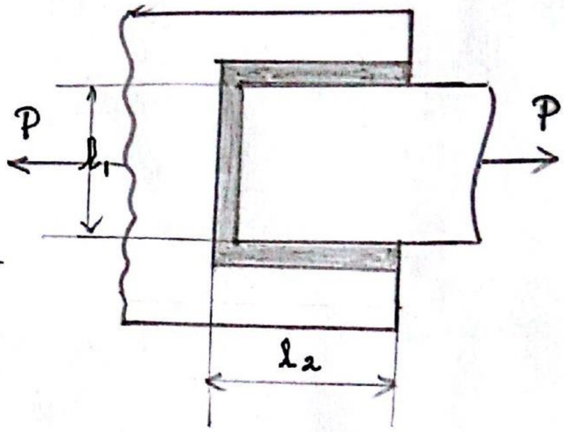
Ahd 12.5 mm for starting and stopping of Weld run

l 50.5 + 12.5

l 63 mm

A plate 75 mm wide and 12.5 mm thick is joined with another plate by a single transverse weld and a double parallel fillet weld as shown in fig. The maximum tensile and shear stresses are 70 MPa and 56 MPa respectively.

Find the length of each parallel fillet, if the joint is subjected to both static and fatigue loading.



Given:

Width (b) = 75 mm

Thickness (h) = 12.5 mm

σ_t = 70 MPa or N/mm^2

τ = 56 MPa or N/mm^2

Where

l_1 = Length of the transverse weld.

l_2 = Length of the parallel fillet weld.

For l_1 we have to subtract 12.5 mm from the width of the plate

l_1 = Width − 12.5 mm

l_1 = 75 − 12.5

l_1 = 62.5 mm.

Length of each parallel fillet for static loading.

W.K.T

$$P = P_1 + P_2$$

Where

P = Maximum load on the plate

P_1 = Load carried by transverse weld

P_2 = Load carried by parallel weld

$$\sigma = \frac{\text{Load}}{\text{Area}} = \frac{P}{A}$$

Area = Width × thickness

A = 75 × 12.5 = 937.5 mm^2

$P_1 \quad 0.707\, h \times l_1 \times \sigma_t$ → Single transverse Weld

$P_1 = 38664$ N

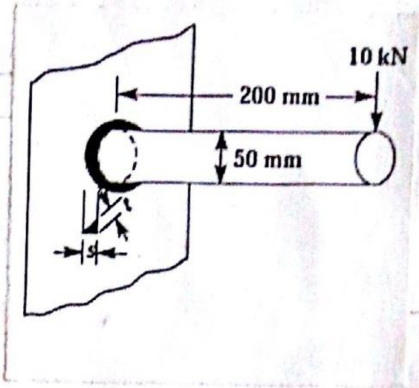
$P_2 \quad 1.414 \times h \times l_2 \times \tau$ → Double Parallel Wel

are 1.5 +

$P_1 \quad 0.707 \times 12.5 \times 62.5 \times 46.7 \quad 25795$ N

$P_2 \quad 1.414 \times 12.5 \times l_2 \times 20.74 \quad 366\, l_2$ N

A 50 mm diameter Solid shaft is Welded to a flat plate as shown in figure. If the Size of the Weld is 15 mm, find the maximum normal stress and shear stress in the Weld.



Given :

$D = 50$ mm

$h = 15$ mm

$L = 200$ mm

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

Maximum Normal stress :

$$\sigma_{max} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2}$$ Pag: No: 7.2

Maximum shear stress :

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2}$$ Pag: No: 7.2

$$\sigma_b = \frac{5.66\, M_b}{h D^2 \pi}$$ [M_b is Bending Moment] Pag: No: 11.3

$M_b = P \times L$

$$\sigma_b = \frac{5.66 \times 10 \times 10^3 \times 200}{15 \times 50^2 \times \pi}$$

$\sigma_b = 96$ N/mm² or MPa.

$$\tau = \frac{2.83\, M_t}{h \cdot D^2 \cdot \pi}$$ [M_t is twisting Moment] Pag: No: 11.3

$M_t = P \times R$ $\quad$ M_t = Load x Radius

$$\tau = \frac{2.83 \times 10 \times 10^3 \times 25}{15 \times 50^2 \times \pi}$$

$\tau = 6$ N/mm² or 6 MPa

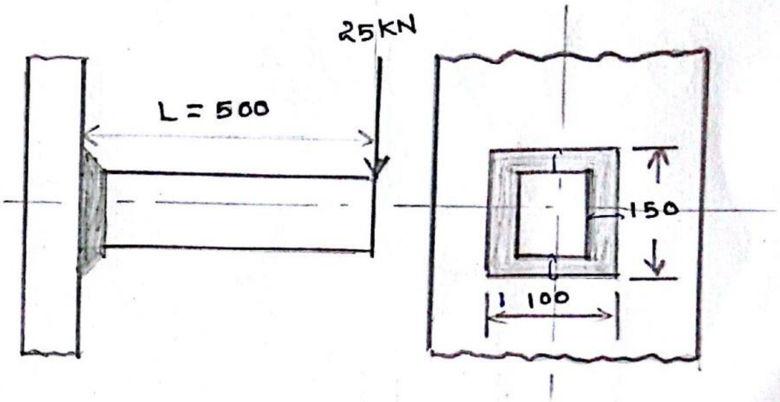
Max. Normal stress

$$\sigma_{max} = \frac{1}{2} \times 96 + \frac{1}{2}\sqrt{96^2 + 4 \times 6^2} = 96.4 \text{ MPa}$$

Max. shear stress

$$\tau_{max} = \frac{1}{2}\sqrt{96^2 + 4 \times 6^2} = 48.4 \text{ MPa}.$$

A rectangular Cross-section bar is Welded to a Support by means of fillet Welds as shown in figure. Determine the Size of the Welds, if the Permissible Shear stress in the Weld to 75 MPa



Given: $P = 25\ kN = 25 \times 10^3\ N$

$\tau_{max} = 75\ MPa$

$b = 100\ mm$

$d = 150\ mm$

$L = 500\ mm$

W.K.T

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

Pag:No: 7.2

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

$t = 0.707S$

Area $= t(2b + 2d)$

$= 0.707S(2 \times 100 + 2 \times 150)$

$[S = h]$

$A = 353.5\ S\ mm^2$

$$\tau = \frac{25 \times 10^3}{353.5S} = \frac{70.72}{S}$$

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

$M = P \times L$

Z is section Modulus

$$z = t\left[bd + \frac{d^2}{3}\right]$$ → Pag: No: 11.6

$$= 0.707S\left[150 \times 100 + \frac{150^2}{3}\right]$$

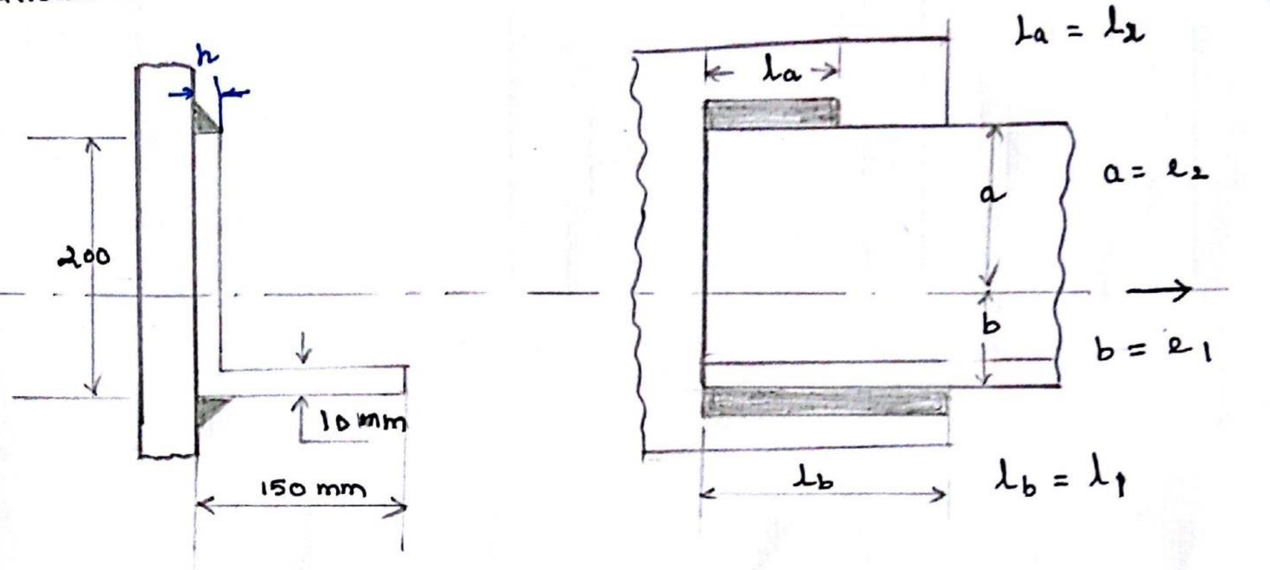
$= 15907.5\ S\ mm^3$

$$\sigma_b = \frac{25 \times 10^3 \times 500}{15907.5S} = \frac{785.8}{S}$$

$$75 = \frac{1}{2}\sqrt{\left(\frac{785.8}{S}\right)^2 + 4\left(\frac{70.72}{S}\right)^2} = \frac{399.2}{S}$$

h (or) $S = 5.32\ mm.$

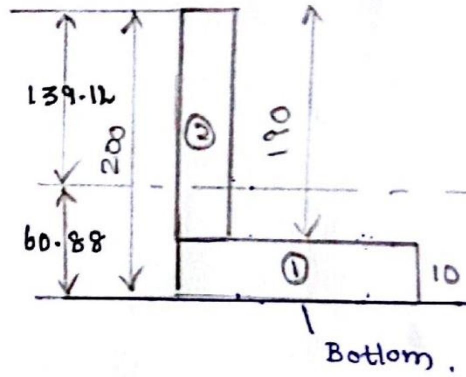
A 200 x 150 x 10 mm angle is to be Welded to a Steel Plate by fillet Welds as shown in figure. If the angle is subjected to a static Load of 200 KN, find the Length of Weld at the top and bottom. The allowable shear stress for Static Loading may be taken as 75 MPa.



Given ; $P = 200\ KN = 200 \times 10^3$

$\tau = 75\ MPa$

$$l_1 = \frac{1.414\, P\, e_2}{\sigma h b} \qquad l_2 = \frac{1.414\, P\, e_1}{\sigma h b}$$



Centroid

$$\bar{y}_{bottom} = \frac{A_1 y_1 + A_2 y_2}{A_1 + A_2}$$

$A_1 = 10 \times 150 = 1500$

$y_1 = \frac{10}{2} = 5$

$A_2 = 190 \times 10 = 1900$

$y_2 = \frac{190}{2} + 10 = 105$

$$\bar{y}_{bottom} = \frac{1500 \times 5 + 1900 \times 105}{1500 + 1900} = 60.88\ mm$$

$\bar{y}_{top} = 200 - 60.88 = 139.12$

$$l_1 = \frac{1.414 \times 200 \times 10^3 \times 139.12}{75 \times 10 \times 200} \qquad l_2 = \frac{1.414 \times 200 \times 10^3 \times 60.88}{75 \times 10 \times 200}$$

$l_1 = 262.28\ mm$ (or) l_b $\qquad$ $l_2 = 114.7\ mm$ (or) l_a

Design of Key. - Unit - III

A 15 kW, 960 rpm motor has a mild steel shaft of 40 mm diameter and the extension being 75 mm. The permissible shear and crushing stresses for the mild steel key are 56 MPa & 112 MPa. Design the keyway in the motor shaft extension. check the shear strength of the key against the normal strength of the shaft.

Given: $P = 15\ kW = 15 \times 10^3\ W$; $N = 960$ rpm; $d = 40$ mm; $l = 75$ mm

$\tau = 56$ MPa; $\sigma_c = 112$ MPa.

W.K.T

$$P = \frac{2\pi N T}{60} \quad ; T = \frac{P \times 60}{2 \times \pi \times N} = \frac{15 \times 10^3 \times 60}{2 \times \pi \times 960}$$

$$T = 149\ N.m$$

$$T = 149 \times 10^3\ N.mm.$$

Width of KeyWay or Key (b):

$$T = l \times b \times \tau \times \frac{d}{2}$$

$$149 \times 10^3 = 75 \times b \times 56 \times \frac{40}{2}$$

$$b = 1.8\ mm$$

∴ Width of the key is too small. The width should be atleast $\frac{d}{4}$

$$b = \frac{d}{4} = \frac{40}{4} = 10\ mm.$$

$\therefore h = 10$ mm ∴ { If $\sigma_c = 2 \times \tau$ Assume Square Key $h \rightarrow$ thickness.

Strength of the shaft with keyway

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

$$= \frac{\pi}{16} \times 56 \times 40^3$$

$$= 571844\ N \quad \text{(Normal)}$$

Shear strength of the key

$$= l \times b \times \tau \times d/2$$

$$= 75 \times 10 \times 56 \times 40/2$$

$$= 840000\ N$$

$$\therefore \frac{\text{Shear strength of Key}}{\text{Normal Strength of Shaft}} = \frac{840000}{571844} = \frac{1.47}{1}$$

Clamp Coupling

Design a clamp coupling to transmit 30 kW at 1000 rpm. The allowable shear stress for the shaft and key is 40 MPa and the

-tttg g,,,,,$t, o) boita Connecting the two halves are six. The permissible

h oaila 6lriv jar † ** bolt is 70 MPa. The coefficient of friction

b»tw»• +h° """il °^‘ °'$f surface are 0.3.

l¿j;,/m« §• zo x ia 4°tL , r\ * ° *7^' * " ***°

$T \quad \frac{30 \times 10^3 \times 60}{} \quad T \quad 2865 \text{ N.m}$

W.K.T Torque transmitted

Data book Page: No: 7.23

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

$,5 \times 10^3 \quad \frac{\pi}{16} \times 40 \times d^3$

$d \quad 71.4$ Say 75 mm

2. Design of Muff

W.K.T diameter of Muff $2 \times 75 + 13 = 163$ Say 165 mm

Total Length of the muff $3.5 \times 75 \quad 262.5$ mm

$L = 3.5\,d$

3. Design for Key

For shaft diameter $d = 75$ mm

Data book Pag: No: 5.21

b ** mm

mm

Design and make a neat dimensional sketch of a muff coupling which is used to conne two steel shafts, transmitting 40 kW at 350 rpm. The material for the shafts and key is plain carbon steel d crushing stresses may be taken.

as 40 MPa and 80 MPa respectively. The material for
ast iron ich the allowable shear stress may be assumed 15 MPa

Given : $P = 40 \times 10^3$ Watt ; $N = 350$ rpm ; $\tau_s = 40$ MPa or N/mm^2
$\sigma_{cs} = 80$ MPa or N/mm^2 ; $\tau_{cast} = \tau_c = 15\ N/mm^2$ or MPa

1. Design of Shaft

$\Rightarrow T \quad \frac{P \times 60}{}$

$T = \frac{,0 \times 10^3 \times \#0}{2 \times \pi \times 350}$ $\quad T \quad 1100$ N.m

\\ a 0 ” ”

!

$T \quad 1100 \times 10^S$ N.mm

$T \quad \frac{\pi}{} \tau_s d^3$ Page : No: 7.23

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

$d \quad 52$ say 55 mm

* *’ *'9*) r ,‘‘o’‘‘

Length of the muff

$L \quad 3.5d \quad 3.5 \times 55 \quad 192.5$ say 195 mm

$$T = \frac{\pi}{16} \times \tau_{cast} \left[\frac{D^4 - d^4}{D}\right]$$ Page : No : 7.21

Muff is in Hollow

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

$$\tau_{cast} = 2.97 \text{ N/mm}^2$$

Since τ_{cast} (Induced) $<$ τ_{cast} (permissible shear stress)

∴ Design of muff is Safe.

3. Design for Key:

Data book.

For shaft diameter $d = 55$ mm Select key Pag: No: 5.21

$$b = 18$$

Since Crushing stress for key material is twice the shear therefore a square key may be used

Pag: No: 5.21

$$h = 18$$

∴ W.K.T the Length of the key

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

Check for Induced shear + Crushing.

Shear $$T = l \times b \times \tau_s \times \frac{d}{2} = 97.5 \times 18 \times \tau_s \times \frac{55}{2}$$

$$\tau_s = \frac{1100 \times 10^3}{48.2}$$

$$\tau_s = 22.8 \text{ N/mm}^2 \text{ or MPa}$$

Crushing $$T = l \times \frac{h}{2} \times \sigma_{cs} \times \frac{d}{2}$$

$$1100 \times 10^3 = 97.5 \times \frac{18}{2} \times \sigma_{cs} \times \frac{55}{2}$$

$$\sigma_{cs} = 45.6 \text{ N/mm}^2 \text{ Or MPa}$$

Since Induced shear + crushing stresses are less than permissible stresses, ∴ Design of key is safe.

Design a Cast iron protective type flange Coupling to transmit 15 kW at 900 rpm. from an electric motor to a Compressor. The service factor may be assumed as 1.35. The following Permissible Stresses

* Shear Stress for shaft, bolt + Key material
* Crushing stress for bolt + Key = 80 MPa

i 4lu» dt›‹^^ d°› >^/ "°* = 8 MPa

Ket

Given P 15 x 10^3 Watt ; N 900 rpm ; Service Factor 1.35

τ_s τ_b τ_k 40 MPa ; σ_{cb} + σ_{ck} 80 MPa ; τ_{cast} = 8 MPa

T 15 x 10^3 x 60 N.m

T = 159.13 N.m

Tmax T x Service Factor

159.13 x 1.35

= 215 N.m

W.K.T

T - π τ_s d^3 Pag : No 7.23

215 x 10^3 π x 40 x d^3

~~484~~

«i -› »s«• ..»5 * *•

Length of the hub

Check for Induced stress for the hub

Data book Pag : No: 7.21

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

$$215 \times 10^3 = \frac{\pi}{16} \tau_{cast} \left[\frac{70^4 - 35^4}{70} \right]$$

$$\tau_{cast} = 3.4 \ N/mm^2 \text{ or } MPa$$

Induced (τ_{cast}) stress is Less than permissible stress

∴ Design of hub is Safe.

Design for Key :

For shaft diameter $d = 35$ mm

Width of Key $b = 12$

∴ W.K.t the Crushing stress is twice the shear stress

So thickness of Key $h = 12$

Square Key are used.

Length of the Key is taken equal to the length of the hub

$$l = L = 52.5 \text{ mm}$$

Check for Induced stress for Shear

$$T = l \times b \times \tau_K \times \frac{d}{2}$$

$$215 \times 10^3 = 52.5 \times 12 \ \tau_K \times \frac{35}{2}$$

$$\tau_K = 19.5 \text{ MPa or } N/mm^2$$

Check for Key Crushing

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

$$215 \times 10^3 = 52.5 \times \frac{12}{2} \times \sigma_{CK} \times \frac{35}{2}$$

$$\sigma_{CK} = 39 \text{ MPa or } N/mm^2$$

Since the induced stress are Less than the permissible stress.

∴ Design for key is ~~Kew~~ Safe.

Paaigp j»-]l‹»qa .

TnzcKruas o-} d!°'@ Al ' 0 s d

$t_f = 0.5\,d = 0.5 \times 35$ 17.5 mm.

is x\D ' * .°*x c•‹t ' ’* ”*
i

Induced stress is less than permissible stress

diameter

Pitch Circle diameter of b•\I»
d = 3 x 35 105 mm

Bolt Subjected to S

$$= \frac{\pi}{4}(d_1)^2 \tau_b \times n \times \frac{D_1}{2}$$

$$3 = \frac{\pi}{} d_1^2 \times 40 \times 3 \times \frac{105}{2}$$

Page : No: 5.42

$d_1 = 6.6$ mm

the bolt is M8

0°t• * the , y axss '°°"»

Thickness of protective Circumferential flange

Say

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

Given: $P = 90\ KW = 90 \times 10^3\ W$; $N = 250$ rpm; $\tau_{shaft} = 40$ MPa;

$\theta^\circ = 1^\circ = 1 \times \frac{\pi}{180} = 0.0175$ rad; $\tau_{bolt} = 30$ MPa or N/mm^2;

W.K.T $P = \frac{2\pi N T}{60} \Rightarrow T = \frac{P \times 60}{2\pi N} = \frac{90 \times 10^3 \times 60}{2\pi \times 250}$

$T = 3440$ N.m

$T = 3440 \times 10^3$ N.mm

W.K.T

$$\frac{T}{J} = \frac{C\theta}{L}$$

∵ $J = \frac{\pi}{32} \times d^4$

∵ Assume $C = 84\ kN/mm^2$

Given $L = 20d$.

$$\frac{3440 \times 10^3}{\frac{\pi}{32} \times d^4} = \frac{84 \times 10^3 \times 0.0175}{20d}$$

$$\frac{35 \times 10^6}{d^4} = \frac{73.5}{d}$$

$$d^3 = \frac{35 \times 10^6}{73.5}$$

$d = 78$ mm say 80 mm.

1. Design of Hub
2. Design for Key.
3. Design for flange
4. Design for bolts

Refer the Previous Problem. [Same Procedure].

A Vertical Screw With single start square threads of 50 mm mean diameter and 12.5 mm pitch is raised against a load of 10 KN by means of a hand Wheel, the boss of which is threaded to act as a nut. The axial Load is taken up by a thrust Collar which supports the Wheel boss and has a mean diameter of 60 mm. The coefficient of friction is 0.15 for the Screw and 0.18 for the Collar. If the tangential force applied by each hand to the Wheel is 100 N, find the Suitable diameter of the hand Wheel.

Given : $d_p = 50$ mm ; $S = 12.5$ mm ; $P = 10$ KN $= 10 \times 10^3$ N ;
$d_{red} = 60$ mm ; $\tan\rho = 0.15$; $\mu = 0.18$; $P_1 = 100$ N ;

Solution ;

We know that the total torque required to turn the hand Wheel

$$M_t = P \frac{d_p}{2} \tan(\beta+\rho) + M_f$$

Data book Pag: No: 7.87

$$M_f = P \frac{d_{red}}{2} \mu$$

$$\tan\beta = \frac{S}{\pi d_p}$$

Data book Pag: No: 7.87

$$\tan\beta = \frac{12.5 \text{ mm}}{\pi \times 50} = 0.08$$

$$\therefore \tan(\beta+\rho) = \left[\frac{\tan\beta + \tan\rho}{1 - \tan\beta \tan\rho}\right]$$

$$M_t = 10\times10^3 \times \frac{50}{2} \times \left[\frac{0.08+0.15}{1-0.08\times0.15}\right] + 10\times10^3 \times \frac{60}{2} \times 0.18$$

$$M_t = 58200 + 54000$$

$$M_t = 112200 \text{ N.mm}$$

Let D_1 = Diameter of the hand Wheel in mm

We know that the torque applied to the Hand wheel,

$$M_t = 2P_1 \times \frac{D_1}{2} = 2 \times 100 \times \frac{D_1}{2} = 100 D_1 \text{ N.mm}$$

$$\therefore \quad 112200 = 100 D_1$$

$$D_1 = 1122 \text{ mm} = 1.122 \text{ m}$$

An electric motor driven power screw moves a nut in a horizontal plane against a force of 75 kN at a speed of 300 mm/min. The screw has a single square thread of 6 mm pitch on a major diameter of 40 mm. The co-efficient of friction at screw threads is 0.1. Estimate power of the motor.

Given: $P = 75\ kN = 75 \times 10^3\ N$; $v = 300\ mm/min$; $s = 6\ mm$

$d_o = 40\ mm$; $\tan \rho = 0.1$

We know that mean diameter of the screw

$$d_p = d_o - s/2$$

$$= 40 - \frac{6}{2}$$

$$d_p = 37\ mm$$

$$\tan \beta = \frac{s}{\pi d_p} = \frac{6}{\pi \times 37}$$

$$\tan \beta = 0.0516$$

We know that

Data book Pag: No: 7.87

$$M_t = P \frac{d_p}{2} \tan(\beta + \rho) + M_f$$

∴ For this problem M_f is Zero = 0

$$M_t = 75 \times 10^3 \times \frac{37}{2} \left[\frac{0.0516 + 0.1}{1 - 0.0516 \times 0.1}\right]$$

$$M_t = 211.45 \times 10^3\ N.mm$$

$$M_t = 211.45\ N.m$$

Since the screw moves in a nut at speed of 300 mm/min and the pitch of the screw is 6 mm, ∴ Speed of the screw in r.p.m

$$N = \frac{\text{Speed in mm/min}}{\text{Pitch in mm}} = \frac{300}{6} = 50\ rpm$$

$$N = 50\ r.p.m$$

Angular speed

$$\omega = \frac{2\pi N}{60} = \frac{2 \times \pi \times 50}{60} = 5.24\ rad/sec$$

Power of the Motor $= M_t . \omega = 211.45 \times 5.24$

$$P = 1108\ W = 1.108\ kW$$

A Cutter of a broaching machine is pulled by square threaded screw of 55 mm external diameter and 10 mm pitch. The operating nut takes the axial load of 400 N. On a flat surface of 60 mm and 90 mm internal and external diameters respectively. If the coefficient of friction is 0.15 for all contact surfaces on the nut, determine the power required to rotate the operating nut when the cutting speed is 6 m/min. Also find the efficiency of the screw.

Given: $d_o = 55$ mm ; $S = 10$ mm ; $P = 400$ N ; $D_1 = 60$ mm
$D_2 = 90$ mm ; $\tan \rho = 0.15$; $\mu = 0.15$; $v = 6$ m/min

We know that mean diameter of the screw

$$d_p = d_o - S/2$$
$$= 55 - 10/2$$
$$d_p = 50 \text{ mm}$$

$$\tan \beta = \frac{S}{\pi d_p} = \frac{10}{\pi \times 50} = 0.0637$$

Data book Pag No: 7.87

$$d_{red} = \frac{D_1 + D_2}{2} = \frac{60 + 90}{2} = 75 \text{ mm}$$

Total Torque is

$$M_t = P \frac{d_p}{2} \tan(\beta + \rho) + M_f$$

Pag: No: 7.87

$$M_f = P \frac{d_{red}}{2} . \mu$$

$$M_t = 400 \times \frac{50}{2} \times \left[\frac{0.0637 + 0.15}{1 - 0.0637 \times 0.15}\right] + 400 \times \frac{75}{2} \times 0.15$$

$$M_t = 4410 \text{ N.mm}$$
$$M_t = 4.41 \text{ N.m}$$

We know that speed of the screw

$$N = \frac{\text{Cutting speed}}{\text{Pitch}} = \frac{6 \times 10^3 \text{ mm/min}}{10 \text{ mm}} = 600 \text{ rpm}$$

$$N = 600 \text{ rpm.}$$

Angular speed,

$$\omega = \frac{2\pi N}{60} = \frac{2 \times \pi \times 600}{60}$$

$$\omega = 62.84 \text{ rad/sec}$$

∴ power required to operate the nut

$$P = M_t \times \omega$$
$$= 4.41 \times 62.84$$
$$= 277 \text{ Watt}$$
$$= 0.277 \text{ KW}$$

Efficiency of the Screw

$$\eta = \frac{M_{to}}{M_t}$$

$$M_{to} = P \frac{d_p}{2} \tan \beta$$

$$= \frac{400 \times \frac{50}{2} \times 0.0637}{4410}$$

$$= 0.144$$

$$\eta = 14.4\%$$

A Vertical two start Square threaded Screw of a 100 mm mean diameter and 20 mm pitch supports a Vertical Load of 18 KN. The axial thrust on the Screw is taken by a Collar bearing of 250 mm outside diameter and 100 mm inside diameter. Find the force required at the end of a lever which is 400mm long in order to lift and lower the Load. The Co-efficient of firiction for the Vertical Screw and nut is 0.15 and that for Collar bearings is 0.20.

Given :- $d_p = 100$ mm ; $S = 20$ mm ; $P = 18 \times 10^3$ N ; $D_2 = 250$ mm

$D_1 = 100$ mm ; $l = 400$ mm ; $\tan \rho = 0.15$; $\mu = 0.20$

Solution :

Let F = Force required at the end of lever.

Since the screw is a two start Square threaded Screw

$$\therefore \tan \beta = \frac{2 \times S}{\pi \times 100}$$

2 is not be a two start Square thread.

$$\tan \beta = \frac{2 \times 20}{\pi \times 100} = 0.127$$

$$\tan \beta = 0.127$$

1. Force Required for raising the Load :

Torque required at the end of lever to lift

$$M_t = P \frac{d_p}{2} \tan(\beta + \rho) + M_f$$

Data book Pag:No: 7.87

$$M_f = P \frac{d_{red}}{2} \mu$$

$$d_{red} = \frac{D_1 + D_2}{2} = \frac{100 + 250}{2} = 175 \text{ mm}$$

$$\therefore \quad M_t = 18 \times 10^3 \times \frac{100}{2} \left[\frac{0.127 + 0.15}{1 - 0.127 \times 0.15} \right] + 18 \times 10^3 \times \frac{175}{2} \times 0.20$$

$$M_t = 569150 \text{ N.mm}$$

We Know that torque required at the end of lever to lift

$$M_t = F \times l$$

$$569150 = F \times 400$$

$$F = 1422.8 \text{ N}$$

2. Force Required for Lowering the Load :

$$M_t = P \frac{d_p}{2} \tan(\beta - \rho) + M_f$$

$$M_f = P \frac{d_{red}}{2} \mu$$

$$M_t = 18 \times 10^3 \times \frac{100}{2} \left[\frac{0.15 - 0.127}{1 + 0.127 \times 0.15} \right] + 18 \times 10^3 \times \frac{175}{2} \times 0.20$$

$$M_t = 335315 \text{ N.mm}$$

We Know that torque required at the end of lever to Lower

$$M_t = F \times l$$

$$335315 = F \times 400$$

$$F = 838.28 \text{ N}.$$

The Lead Screw of a Lathe has Acme threads of 50 mm outside diameter + 8 mm pitch. The screw must exert an axial pressure of 2500N in order to drive the tool carriage. The trust is carried on a collar 110 mm outside diameter + 55 mm inside diameter, the Lead Screw rotates at 30 r.p.m. Determine a) power required to drive the screw b) efficiency of the screw. Assume a coefficient of friction of 0.15 for the screw and 0.12 for the collar.

Given: $d_o = 50\,mm$; $S = 8\,mm$; $P = 2500\,N$; $D_1 = 55\,mm$,
$D_2 = 110\,mm$; $N = 300\, r.p.m$; $\tan\beta = 0.15$; $\mu = 0.12$

a) Power to drive the Screw ;

We know $d_p = d_o - S/2$
$d_p = 50 - 8/2 = 46\,mm$

$$\tan\beta = \frac{S}{\pi d_p} = \frac{8}{\pi \times 46} = 0.055$$

For ACME thread

α is always $14.5°$

$$\tan\rho = \frac{0.15}{\cos\alpha} = \frac{0.15}{\cos 14.5°}$$

$$\tan\rho = 0.155$$

Torque required

Pag: No: 7.87

$$M_t = P\frac{d_p}{2}\tan(\beta+\rho) + M_f$$

$$M_f = P\frac{d_{red}}{2}\mu$$

$$d_{red} = \frac{D_1 + D_2}{2} = \frac{110+55}{2} = 82.5\,mm$$

$$\therefore\ M_t = 2500 \times \frac{46}{2}\left[\frac{0.055+0.155}{1-0.055\times 0.155}\right] + 2500 \times \frac{82.5}{2} \times 0.12$$

$M_t = 24565\ N.mm$

$M_t = 24.565\ N.m$

Angular speed

$$\omega = \frac{2\times\pi\times N}{60} = \frac{2\times\pi\times 30}{60}$$

$\omega = 3.14\ rad/sec$

Power Required

$$P = 2 M_t \times \omega$$

$$= 24.565 \times 3.14$$

$$P = 77 \text{ Watt.}$$

b) Efficiency of the Lead Screw

$$\eta = \frac{M_{to}}{M_t} \qquad M_{to} = P \tan\beta \times \frac{d_p}{2}$$

$$\eta = \frac{2500 \times 0.055 \times \frac{46}{2}}{24565}$$

$$\eta = \frac{31625}{24565}$$

$$\eta = 0.13 \text{ or } 13\%$$

Stresses in power screw :

A power screw having double start square threads of 25 mm nominal diameter and 5 mm pitch is acted upon by an axial load of 10 kN. The outer & inner diameter of collar are 50 mm & 20 mm resp. The coefficient of thread + collar friction may be assumed as 0.2 & 0.15 resp. The screw rotates at 12 r.p.m. Assume uniform wear condition at collar & thread with pressure of 5.8 N/mm². Find 1. Torque required to rotate the screw 2. Stress in screw & 3. Number of threads of nut in engagement with screw.

Given: $d_o = 25$ mm ; $S = 5$ mm ; $P = 10 \times 10^3$ N ; $D_1 = 50$, $D_2 = 20$ mm

$\tan\rho = 0.2$, $\mu = 0.15$, $N = 12$ r.p.m , $P_b = 5.8$ N/mm^2

Solution:

W.K.T

$$d_p = d_o - \frac{S}{2}$$

$$= 25 - 5/2$$

$$d_p = 22.5 \text{ mm}$$

2 is for Double Start.

$$\tan\beta = \frac{2S}{\pi \times d_p} = \frac{2 \times 5}{\pi \times 22.5}$$

$$\tan\beta = 0.1414$$

1. Torque required to rotate the screw

$$M_t = P \frac{d_p}{2} \tan(\beta + \rho) + M_f$$ Pag: No: 7.87

$$M_f = P \frac{d_{red}}{2} \mu$$

$$d_{red} = \frac{D_1 + D_2}{2} = \frac{50+20}{2} = 35 \text{ mm}$$

$$M_t = 10 \times 10^3 \times \frac{22.5}{2} \left[\frac{0.1414 + 0.2}{1 - 0.1414 \times 0.2} \right] + 10 \times 10^3 \times \frac{35}{2} \times 0.15$$

$$M_t = 65771 \text{ N.mm}$$

$$M_t = 65.771 \text{ N.m}$$

2. Stress in the screw Pag: No: 7.87

$$\sigma_c = \frac{P}{A_c}$$

$$\sigma_c = \frac{10 \times 10^3}{\frac{\pi}{4} \times (20)^2} = 31.83 \text{ N/mm}^2$$

d_c or d_1

Area

$$\therefore A_c = \frac{\pi}{4} d_c^2 \text{ or } d_1^2$$

d_c is core diameter

$$d_1 \text{ or } d_c = d_o - P$$
$$= 25 - 5$$
$$d_1 \text{ or } d_c = 20 \text{ mm}$$

shear stress,

$$M_t (\text{or}) T = \frac{\pi}{16} \tau d_c^3 \text{ or } d_1^3$$

$$\tau = \frac{16 \times 65771}{\pi \times (20)^3} = 41.86 \text{ N/mm}^2$$

Pag: No: 7.87

W.K.T

Max shear stress

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_c^2 + 4\tau^2} = \frac{1}{2}\sqrt{(31.83)^2 + 4(41.86)^2}$$

$$\tau_{max} = 44.8 \text{ N/mm}^2$$

3. Number of threads of nut in engagement with screw

We know that

$$P_b = \frac{P}{A \times n} = \frac{P}{\pi d \times t \times n}$$

$$t = \frac{S}{2}$$

$\therefore$ thickness $= \frac{\text{pitch}}{2}$

$$5.8 = \frac{10 \times 10^3}{\pi \times 22.5 \times 2.5 \times n} = \frac{56.6}{n}$$

$$n = \frac{56.6}{5.8} = 9.76$$

$$n = 10$$

Unit-III

CoterJoint

- ✓ *Acoterisaflatwedgeshapedpieceofrectangularcross-sectionanditswidthis tapered(eitherononesideorbothsides)fromoneendtoanotherforaneasy adjustment.*
- ✓ *Thetapervariesfrom1in48to1in24anditmaybeincreasedupto1in8,ifa lockingdeviceisprovided.*
- ✓ *Thecoterisusua**l**ymadeofmildsteelorwroughtiron.*
- ✓ *Acoterjointisatemporaryfasteningandisusedtoconnectrigidlytwoco-axial rodsorbarswhicharesubjectedtoaxialtensileorcompressiveforces.*
- ✓ *Itisusua**lyusedin**connectingapistonrodtothecrossheadofareciprocating steamengine,apistonrodanditsextensionasatailorpumprod,strapendof connectingrodetc.*

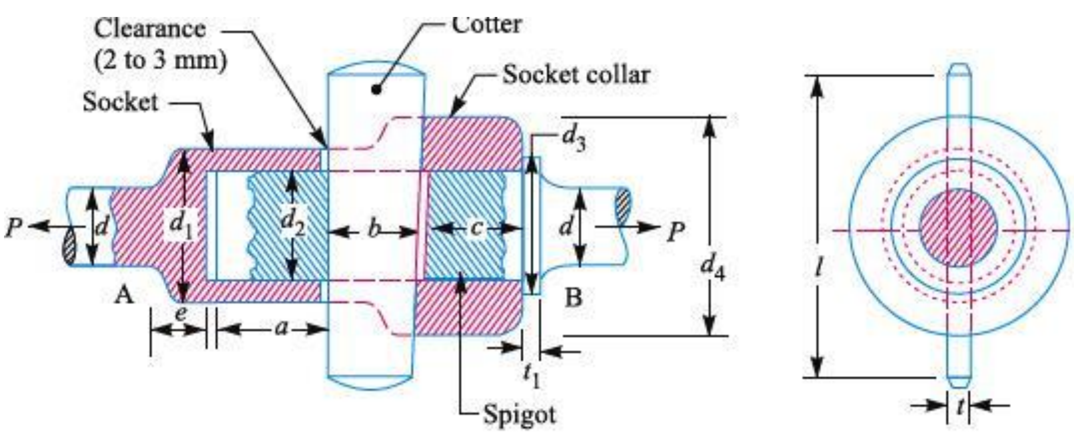
TypesofCoterJoints

*Fo**l**owingarethethreecommonlyusedcoterjointsoconnecttworodsbyacoter:*

- ✓ *Socketandspigotcoterjoint,*
- ✓ *Sleeveandcoterjoint,and*
- ✓ ***Gibandcoterjoint.***

SocketandSpigotCoterJoint

- ✓ *Inasocketandspigototerjoint,oneendoftherods(sayA)isprovidedwitha sockettypeofendasshowninFig.andtheotherendoftheotherrod(sayB)is insertedintoasockt.*



- ✓ *Theendoftherodwhichgoesintoasocketisalsoca**l**edspigot.Arectangular holeismadeinthesocketandspigot.*
- ✓ *Acoteristhendriventightlythroughaholeinordertomakethetemporary connectionbetweenthetworods.*

- ✓ *Theloadisusuallyactingaxially,butitchangesitsdirectionandhencethecoter jointmustbedesignedtocarrybooththetensileandcompressiveloads.The compressiveloadistakenupbythecollaronthespigot.*

DesignofSocketandSpigotCoterJoint

The socket and spigot cotter joint is shown in Fig. 12.1.

Let
P = Load carried by the rods,
d = Diameter of the rods,
d_1 = Outside diameter of socket,
d_2 = Diameter of spigot or inside diameter of socket,
d_3 = Outside diameter of spigot collar,
t_1 = Thickness of spigot collar,
d_4 = Diameter of socket collar,
c = Thickness of socket collar,
b = Mean width of cotter,
t = Thickness of cotter,
l = Length of cotter,
a = Distance from the end of the slot to the end of rod,
σ_t = Permissible tensile stress for the rods material,
τ = Permissible shear stress for the cotter material, and
σ_c = Permissible crushing stress for the cotter material.

1.Failureoftherodsintension

TherodsmayfailintnsionduetothetensileloadP.Weknowthat

Area resisting tearing

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

∴ Tearing strength of the rods,

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

Equating this to load (P), we have

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

Fromthisequation,diameteroftherods(d)maybedetermined.

2.Failureofspigotintensionacrosstheweakestsection(orslot):

Sincetheweakestsectionofthespigotisthatsectionwhichhasaslotinitfor thecoter,asshowninFig.therefore

$$= \frac{\pi}{4}(d_2)^2 - d_2 \times t$$

Arearesistingtearingofthespigotacrosstheslotandtearingstrengthofthe spigotacrosstheslot

Equatingthistoload(P),wehave

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

Fromthisequation,thediameterofspigotorinsidediameterofsocket(d2)may bedetermined.

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

Fromthisequation,thediameterofspigotorinsidediameterofsocket(d2)may bedetermined.

3.Failureoftherodorcoterincrushing

Weknowthattheareathatresistscrushingofarodorcoter

=d2×t

∴Crushingstrength=d2×t×σc

Equatingthistoload(P),wehave

P=d2×t×σc

Fromthisequation,theindued rushingstressmaybechecked.

4.Failureofthesocketintnsionacrosstheslot

Weknowthattheresistingareaofthesocketacrosstheslot,asshowninFig.

∴Tearingstrengthofthesocketacrosstheslot

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

Equatingthistoload(P),wehave

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

Fromthisequation,outsidediameterofsocket(d1)maybedetermined

5.Failureofcoterinshear

Consideringthefailureofcoterinshearsincethecoterisindoubleshear, *thereforeshearingareaofthecoter*

$$=2b\times t$$

andshearingstrengthofthecoter

$$=2b\times t\times \tau$$

Equatingthistoload(P),wehave

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

Fromthisequation,widthofcoter(b)isdetermined

6.Failureofthesocketcolarincrushing

Consideringthefailureofsocketcolarincrushingasshowninfig.Weknow thatareathatresistscrushingofsocketcolar

$$=(d_4-d_2)t$$

andcrushingstrength$=(d_4-d_2)t\times \sigma_c$

Equatingthistoload(P),wehave

$$P=(d_4-d_2)t\times \sigma_c$$

Fromthisequation,thediameterofsocketcolar(d4)maybeobtained.

7.Failureofsocketendinshearing

Sincethesocketendisindoubleshear,therefreareathatresistsshearingof socketcolar

$$=2(d_4-d_2)c$$

andshearingstrengthofsoketolar

$$=2(d_4-d_2)\times \tau$$

Equatingthistoload(P),w have

$$P=2(d_4-d_2)c\times \tau$$

Fromthisequation,thethicknessofsocketcolar(c)maybeobtained.

8.Failureofrodendinshear

Sincetherodendisindoubleshear,thereforetheareresistingshearoftherod end

$$=2a\times d_2$$

Andshearstrengthoftherodend

$$=2a\times d_2\times \tau$$

Equatingthistoload(P),wehave

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

Fromthisequation,thedistancefromtheendoftheslottotheendoftherod(a) maybeobtained.

9.Failureofspigotcolarincrushing

ConsideringthefailureofthespigotcolarincrushingWeknowthatareathat resistscrushingofthecolar

$$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]$$

Andcrushingstrengthofthecolar

$$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$$

Equatingthistoload(P),wehave

$$P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$$

Fromthisequation,thediameterofthespigotcolar(d3)maybeobtained.

10.Failureofthespigotcolarinshearing

ConsideringthefailureofthespigotcolarinsharingW knowthatareathat resistsshearingofthecolar

$=\pi d_2 \times t_1$

Andshearingstrengthofthecolar,

$=\pi d_2 \times t_1 \times \tau$

Equatingthistoload(P)we ave

$P=\pi d_2 \times t_1 \times \tau$

Fromthisequation,thethiknessofspigotcolar(t1)maybeobtained.

11.Failureofcoterinbending

ThemaximumbendingmomentoccursattheCentreofthecoterandisgivenby

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}\times\frac{d_4 - d_2}{2} + \frac{d_2}{2}\right) - \frac{P}{2}\times\frac{d_2}{4}$$
$$= \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{2} - \frac{d_2}{4}\right) = \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)$$

Weknowthatsectionmodulusofthecoter,

$Z=t\times b^2/6$

∴Bendingstressinducedinthecoter,

$$\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)}{t\times b^2/6} = \frac{P\,(d_4 + 0.5\,d_2)}{2\,t\times b^2}$$

Thisbendingstressinducedinthecotershouldbelessthanthealowablebending stressofthecoter.

12.Thelengthofcoter(l)istakenas4d.

13.Thetaperincotershouldnotexceed1in24.Incasethegreatertaperisrequired, thenalockingdevicemustbeprovided.

14.Thedrawofcoterisgeneralytakenas2to3mm.

Problems

1.*Designanddrawacoterjointtosupportaloadvaryingfrom30kNincompression to30kNintension.Thematerialusediscarbonsteelforwhichthefolowing alowablestressesmaybeused.Theloadisappliedstaticaly.Tensilestress= compressivestress=50MPa;shearstress=35MPaandcrushingstress=90MPa.*

Solution.*Given:P=30kN=30×10^3N;σ_t=50MPa=50N/mm^2;τ=35MPa=35N /mm^2;σc=90MPa=90N/mm^2*

1.Diameteroftherods

Letd=Diameteroftherods.

Consideringthefailureoftherodintension.Weknow haload(P),

$$30 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 50 = 39.3\, d^2$$

$$\therefore \quad d^2 = 30 \times 10^3 / 39.3 = 763 \quad \text{or} \quad d = 27.6 \text{ say } 28 \text{ mm} \quad \textbf{Ans.}$$

2.Diameterofspigotandthicknessofcoter

Letd2=Diameterofspigotorinsidediameterofsocket,and

t=Thicknessofcoter.Itmaybetakenasd2/4.

Consideringthefailureofspigotintensionacrosstheweakestsection.Weknowthat load(P),

Let d_2 = Diameter of spigot or inside diameter of socket, and

t = Thickness of cotter. It may be taken as $d_2 / 4$.

Considering the failure of spigot in tension across the weakest section. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right] 50 = 26.8\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 30 \times 10^3 / 26.8 = 1119.4 \quad \text{or} \quad d_2 = 33.4 \text{ say } 34 \text{ mm}$$

and thickness of cotter, $t = \frac{d_2}{4} = \frac{34}{4} = 8.5 \text{ mm}$

Let us now check the induced crushing stress. We know that load (P),

$$30 \times 10^3 = d_2 \times t \times \sigma_c = 34 \times 8.5 \times \sigma_c = 289\, \sigma_c$$

$$\therefore \quad \sigma_c = 30 \times 10^3 / 289 = 103.8 \text{ N/mm}^2$$

Sincethisvalueofσcismorethanthegivenvalueofσc=90N/mm2,thereforethe dimensionsd2=34mmandt=8.5mmarenotsafe.Nowletusfindthevaluesofd2 andtbysubstitutingthevalueofσc=90N/mm2intheaboveexpression,i.e.

$$30 \times 10^3 = d_2 \times \frac{d_2}{4} \times 90 = 22.5\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 30 \times 10^3 / 22.5 = 1333 \quad \text{or} \quad d_2 = 36.5 \text{ say } 40 \text{ mm } \textbf{Ans.}$$

$$\text{and} \quad t = d_2 / 4 = 40/4 = 10 \text{ mm } \textbf{Ans.}$$

3.outsidediameterofsocket

Let d1=Outsidediameterofsocket.

Consideringthefailureofthesocketintensionacrosstheslot.Weknowthatload(P),

$$30 \times 10^3 = \left[\frac{\pi}{4}\left\{(d_1)^2 - (d_2)^2\right\} - (d_1 - d_2)\,t\right]\sigma_t$$

$$= \left[\frac{\pi}{4}\left\{(d_1)^2 - (40)^2\right\} - (d_1 - 40)\,10\right]50$$

$$30 \times 10^3/50 = 0.7854\,(d_1)^2 - 1256.6 - 10\,d_1 + 400$$

$$\text{or} \quad (d_1)^2 - 12.7\,d_1 - 1854.6 = 0$$

$$\therefore \quad d_1 = \frac{12.7 \pm \sqrt{(12.7)^2 + 4 \times 1854.6}}{2} = \frac{12.7 \pm 87.1}{2}$$

$$= 49.9 \text{ say } 50 \text{ mm } \textbf{Ans.} \qquad \text{...(Taking +ve sign)}$$

4.Widthofcoter

Letb=Widthofcoter.

Consideringthefailureofthecoterinshear.Sincethecoterisindoubleshear,

thereforeload(P),

$$30 \times 10^3 = 2\,b \times t \times \tau = 2\,b \times 10 \times 35 = 700\,b$$

$$\therefore \quad b = 30 \times 10^3 / 700 = 43 \text{ mm } \textbf{Ans.}$$

5.Diameterofsocketcolar

*Letd4=Diameterofsocktco**l**ar.*

*Consideringthefailureofthesocketco**l**arandcoterincrushing.Weknowthatload(P),*

$$30 \times 10^3 = (d_4 - d_2)\,t \times \sigma_c = (d_4 - 40)\,10 \times 90 = (d_4 - 40)\,900$$

$$\therefore d_4 - 40 = 30 \times 103/900 = 33.3 \text{ or } d_4 = 33.3 + 40 = 73.3 \text{ say } 75 \text{ mm } \textbf{Ans.}$$

6.Thicknessofsocketcolar

*Letc=Thicknessofsocketco**l**ar.*

Consideringthefailureofthesocketendinshearing.Sincethesocketendisindouble

shear,thereforeload(P),

$$30 \times 10^3 = 2(d_4 - d_2)\,c \times \tau = 2(75 - 40)\,c \times 35 = 2450\,c$$

$$\therefore c = 30 \times 103/2450 = 12 \text{ mm } \textbf{Ans.}$$

7.Distancefromtheendoftheslottotheendoftherod

Leta=Distancefromtheendofslottotheendoftherod.

Consideringthefailureoftherodendinshear.Sincetherodendisindoubleshear, *therefore*

load(P),

$$30\times10^{3}=2a\times d_2\times\tau=2a\times40\times35=2800a$$

$\therefore a=30\times10^3/2800=10.7$ say 11 mm **Ans.**

8.Diameterofspigotcolar

Let d_3 *=Diameterofspigotcolar.*

Consideringthefailureofspigotcolarincrushing.Weknowthatload(P),

$$30\times10^3=\frac{\pi}{4}\left[(d_3)^2-(d_2)^2\right]\sigma_c=\frac{\pi}{4}\left[(d_3)^2-(40)^2\right]90$$

or $$(d_3)^2-(40)^2=\frac{30\times10^3\times4}{90\times\pi}=424$$

$$\therefore\quad (d_3)^2=424+(40)^2=2024 \quad\text{or}\quad d_3=45\text{ mm } \textbf{Ans.}$$

9.Thicknessofspigotcolar

Let t_1 *=Thicknessofspigotcolar.*

Consideringthefailureofspigotcolarinshearing.Weknowthatload(P),

$30\times10^{3}=\pi d_2\times t_1\times\tau=\pi\times40\times t_1\times35=4400t_1$ $\therefore t_1=30\times103/4400=6.8$ say 8mm **Ans.**

10. *Thelengthofcoter(l)istakenas4d.*

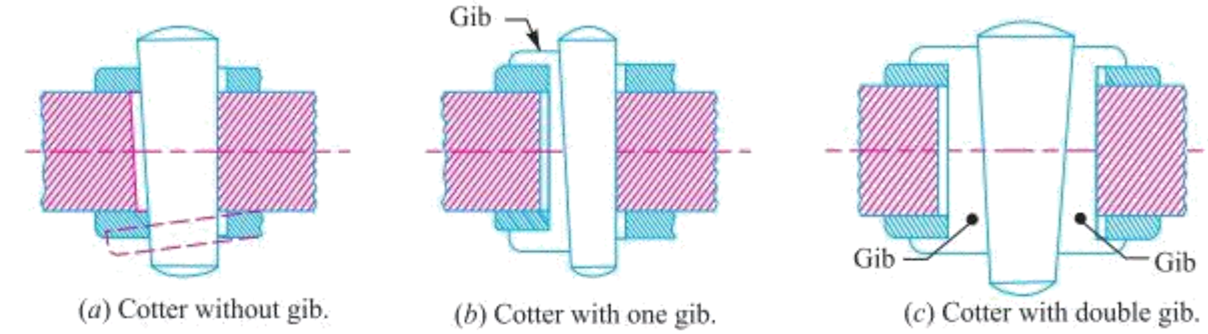
$\therefore l=4d=4\times28=112$ mm **Ans.**

11. *Thedimensioneistakenas1.2d.*

$\therefore e=1.2\times28=33.6$ say 34mm **Ans.**

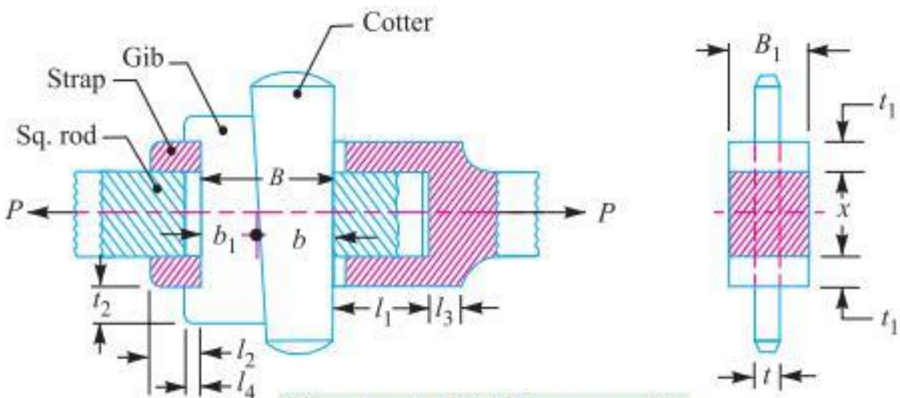
GibandCoterJoint

- ✓ *AGibandcoterjointisusuallyusedinstrapend(orbigend)ofaconnectingrod asshowninFig.below.*
- ✓ *whenthecoteralone(i.e.withoutGib)isdriven,thefrictionbetweenitsends andtheinsideoftheslotsinthestraptendstocausethesidesofthestrapto springopen(orspread)outwardsasshowndotedinFig.12.11(a).*
- ✓ *Inordertopreventthis,gibsasshowninFig.12.11(b)and(c),areusedwhich holdtogethertheendsofthestrap.Gibsprovidealargerbearingsurfaceforthe cotertoslideon,duetotheincreasedholdingpower.*
- ✓ *Thus,thetendencyofcotertoslackenbackowingtofrictionisconsiderably decreased.Thejib,also,enablesparallelholestobeused.*



DesignofGibandCoterJointforSquareRods

- ✓ *ConsideragibandcoterjointforsquarerodsasshowninFig.above.Therods maybesubjectedtoatensileorcompressiveload.*



Gib and cotter joint for square rods.

- ✓ *Alcomponentsofthejointareassumedtobeofthesamematerial*

Let P = Load carried by the rods,

x = Each side of the rod,

B = Total width of gib and cotter,

B_1 = Width of the strap,

t = Thickness of cotter,

t_1 = Thickness of the strap, and

σ_t, τ and σ_c = Permissible tensile, shear and crushing stresses.

In designing a gib and cotter joint, the following modes of failure are considered.

1. *Failure of the rod in tension*

The rod may fail in tension due to the tensile load P. We know that

Area resisting tearing = $x \times x = x^2$

$\therefore$ Tearing strength of the rod

$$= x^2 \times \sigma_t$$

Equating this to the load (P), we have

$$P = x^2 \times \sigma_t$$

From this equation, the side of the square rod (x) may be determined. The other dimensions are fixed as under :

Width of strap, B_1 = Side of the square rod = x

Thickness of cotter, $t = \frac{1}{4}$ width of strap $= \frac{B_1}{4}$

Thickness of gib = Thickness of cotter (t)

Height (t_2) and length of gib head (l_4)

= Thickness of cotter (t)

2. *Failure of the gib and cotter in shearing*

Since the gib and cotter are in double shear, therefore,

Area resisting failure $= 2\,B \times t$

and resisting strength $= 2\,B \times t \times \tau$

Equating this to the load (P), we have

$$P = 2B \times t \times \tau$$

From this equation, the width of gib and cotter (B) may be obtained. In the joint, as shown in Fig. 12.13, one gib is used, the proportions of which are

Width of gib, $b_1 = 0.55\,B$; and width of cotter, $b = 0.45\,B$

In case two gibs are used, then

Width of each gib $= 0.3\,B$; and width of cotter $= 0.4\,B$

3. *Failure of the strap end in tension at the location of gib and cotter*

Area resisting failure $= 2\,[B_1 \times t_1 - t_1 \times t] = 2\,[x \times t_1 - t_1 \times t]$ ... ($\because B_1 = x$)

$\therefore$ Resisting strength $= 2\,[x \times t_1 - t_1 \times t]\,\sigma_t$

Equating this to the load (P), we have

$$P = 2\,[x \times t_1 - t_1 \times t]\,\sigma_t$$

From this equation, the thickness of strap (t_1) may be determined.

4. *Failure of the strap or gib in crushing*

The strap or gib (at the strap hole) may fail due to crushing.

Area resisting failure $= 2\, t_1 \times t$

$\therefore$ Resisting strength $= 2\, t_1 \times t \times \sigma_c$

Equating this to the load (P), we have

$$P = 2\, t_1 \times t \times \sigma_c$$

From this equation, the induced crushing stress may be checked.

5. *Failure of the rod end in shearing*

Since the rod is in double shear, therefore

Area resisting failure $= 2\, l_1 \times x$

$\therefore$ Resisting strength $= 2\, l_1 \times x \times \tau$

Equating this to the load (P), we have

$$P = 2\, l_1 \times x \times \tau$$

From this equation, the dimension l_1 may be determined.

6. *Failure of the strap end in shearing*

Since the length of rod (l_2) is in double shearing, therefore

Area resisting failure $= 2 \times 2\, l_2 \times t_1$

$\therefore$ Resisting strength $= 2 \times 2\, l_2 \times t_1 \times \tau$

Equating this to the load (P), we have

$$P = 2 \times 2\, l_2 \times t_1 \times \tau$$

From this equation, the length of rod (l_2) may be determined. The length l_3 of the strap end is proportioned as $\frac{2}{3}$ rd of side of the rod. The clearance is usually kept 3 mm. The length of cotter is generally taken as 4 times the side of the rod.

Problem

DesignaGibandcoterjointasshowninFig.12.13,tocarryamaximumloadof35kN. AssumingthattheGib,coterandrodareofsamematerialandhavethefolowing alowablestresses:

$$\sigma_t = 20\ MPa\ ;\ \tau = 15\ MPa\ ;\ and\ \sigma_c = 50\ MPa$$

Solution. Given : $P = 35$ kN $= 35\,000$ N ; $\sigma_t = 20$ MPa $= 20$ N/mm^2 ; $\tau = 15$ MPa $= 15$ N/mm^2 ; $\sigma_c = 50$ MPa $= 50$ N/mm^2

1. *Side of the square rod*

Let x = Each side of the square rod.

Considering the failure of the rod in tension. We know that load (P),

$$35\,000 = x^2 \times \sigma_t = x^2 \times 20 = 20\, x^2$$

$\therefore$ $x^2 = 35\,000 / 20 = 1750$ or $x = 41.8$ say 42 mm **Ans.**

Other dimensions are fixed as follows :

Width of strap, $B_1 = x = 42$ mm **Ans.**

Thickness of cotter, $t = \dfrac{B_1}{4} = \dfrac{42}{4} = 10.5$ say 12 mm **Ans.**

Thickness of gib = Thickness of cotter = 12 mm **Ans.**

Height (t_2) and length of gib head (l_4)

= Thickness of cotter = 12 mm **Ans.**

2. Width of gib and cotter

Let B = Width of gib and cotter.

Considering the failure of the gib and cotter in double shear. We know that load (P),

$$35\ 000 = 2\,B \times t \times \tau = 2\,B \times 12 \times 15 = 360\,B$$

$$\therefore \quad B = 35\,000 / 360 = 97.2 \text{ say } 100 \text{ mm} \quad \textbf{Ans.}$$

Since one gib is used, therefore

Width of gib, $b_1 = 0.55\,B = 0.55 \times 100 = 55$ mm **Ans.**

and width of cotter, $b = 0.45\,B = 0.45 \times 100 = 45$ mm **Ans.**

3. Thickness of strap

Let t_1 = Thickness of strap.

Considering the failure of the strap end in tension at the location of the gib and cotter. We know that load (P),

$$35\ 000 = 2\,(x \times t_1 - t_1 \times t)\,\sigma_t = 2\,(42 \times t_1 - t_1 \times 12)\,20 = 1200\,t_1$$

$$\therefore \quad t_1 = 35\,000 / 1200 = 29.1 \text{ say } 30 \text{ mm} \quad \textbf{Ans.}$$

Now the induced crushing stress may be checked by considering the failure of the strap or gib in crushing. We know that load (P),

$$35\ 000 = 2\,t_1 \times t \times \sigma_c = 2 \times 30 \times 12 \times \sigma_c = 720\,\sigma_c$$

$$\therefore \quad \sigma_c = 35\,000 / 720 = 48.6 \text{ N/mm}^2$$

Since the induced crushing stress is less than the given crushing stress, therefore the joint is safe.

4. Length (l_1) of the rod

Considering the failure of the rod end in shearing. Since the rod is in double shear, therefore load (P),

$$35\ 000 = 2\,l_1 \times x \times \tau = 2\,l_1 \times 42 \times 15 = 1260\,l_1$$

$$\therefore \quad l_1 = 35\,000 / 1260 = 27.7 \text{ say } 28 \text{ mm} \quad \textbf{Ans.}$$

5. Length (l_2) of the rod

Considering the failure of the strap end in shearing. Since the length of the rod (l_2) is in double shear, therefore load (P),

$$35\ 000 = 2 \times 2\,l_2 \times t_1 \times \tau = 2 \times 2\,l_2 \times 30 \times 15 = 1800\,l_2$$

$$\therefore \quad l_2 = 35\,000 / 1800 = 19.4 \text{ say } 20 \text{ mm} \quad \textbf{Ans.}$$

Length (l_3) of the strap end

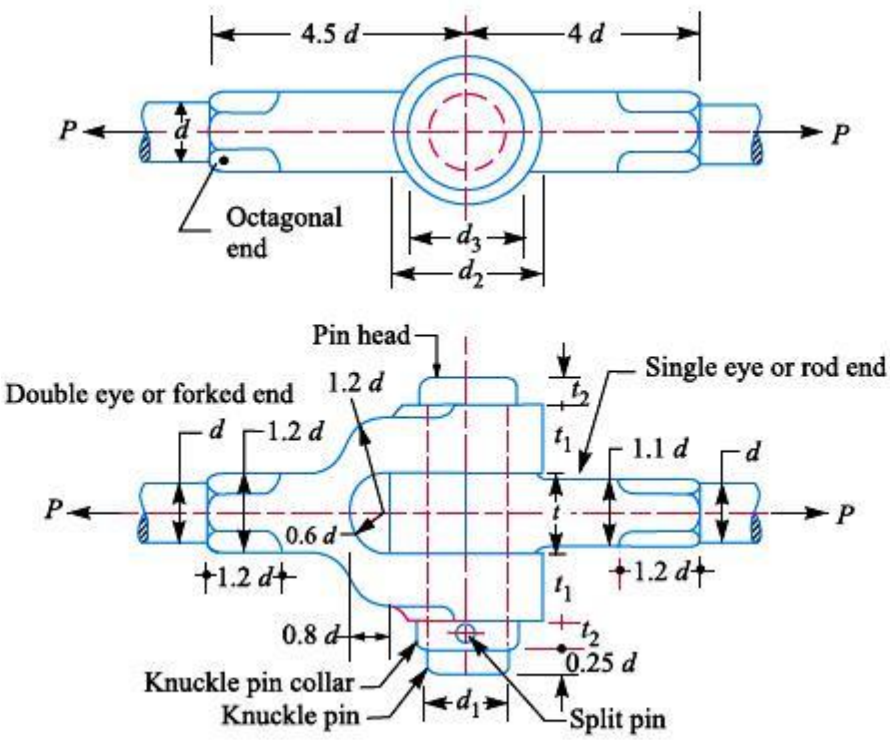
$$= \frac{2}{3} \times x = \frac{2}{3} \times 42 = 28 \text{ mm} \quad \textbf{Ans.}$$

and length of cotter $= 4\,x = 4 \times 42 = 168$ mm **Ans.**

KnuckleJoint

- ✓ *Aknucklejointisusedtoconnecttworodswhichareundertheactionoftensile loads.*

- ✓ *However,ifthejointisguided,therodsmaysupportacompressiveload.*
- ✓ *Aknucklejointmaybereadilydisconnectedforadjustmentsorrepairs.*
- ✓ *Itsusemaybefoundinthelinkofacyclechain,tierodjointforrooftruss,valve rodjointwitheccentricrod,pumprodjoint,tensionlinkinbridgestructureand leverandrodconnectionsofvarioustypes.*



DimensionsofVariousPartsoftheKnuckleJoint

- ✓ *Thedimensionsofvariouspartsoftheknucklejointarefixedbyempirical relationsasgivenbelow.*
- ✓ *Itmaybenotedthatallthepartsshouldbemadeofthesamemateriali.e.mild steelorwroughtiron.*

Ifdisthediameterofrod,thendiameterofpin,

$$d_1=d$$

Outerdiameterofeye,

$$d_2=2d$$

Diameterofknucklepinheadandcollar,

$$d_3=1.5d$$

Thicknessofsingleeyeorrodend,

t=1.25d

Thicknessoffork,t_1=0.75d

Thicknessofpinhead,t_2=0.5d

Otherdimensionsofthejointareshownabove.

MethodsofFailureofKnuckleJoint

ConsideraknucklejointasshowninFig.

Let P=Tensileloadactingontherod,

d=Diameteroftherod,

d1=Diameterofthepin,

d2=Outerdiameterofeye,

t=Thicknessofsingleeye,

t1=Thicknessoffork.

σ_t,τandσ_c=Permissiblestressesforthejointmaterialintension,shear andrushingrespectively.

Indeterminingthestrengthofthejointforthevariousm hodsoffailure,itisassumed that

1.Thereisnostressconcentration,and

2.Theloadisuniformlydistributedovereachpartofthejoint.

Folowingarethevariousmethodsoffailureofthejoint:

1. *Failure of the solid rod in tension*

Since the rods are subjected to direct tensile load, therefore tensile strength of the rod,

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

Equating this to the load (P) acting on the rod, we have

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

From this equation, diameter of the rod (d) is obtained.

2. *Failure of the knuckle pin in shear*

Since the pin is in double shear, therefore cross-sectional area of the pin under shearing

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

and the shear strength of the pin

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

Equating this to the load (P) acting on the rod, we have

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

From this equation, diameter of the knuckle pin (d_1) is obtained. This assumes that there is no slack and clearance between the pin and the fork and hence there is no bending of the pin. But, in

∴ Maximum bending (tensile) stress,

$$\sigma_t = \frac{M}{Z} = \frac{\frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)}{\frac{\pi}{32} (d_1)^3}$$

From this expression, the value of d_1 may be obtained.

3. *Failure of the single eye or rod end in tension*

The single eye or rod end may tear off due to the tensile load. We know that area resisting tearing $= (d_2 - d_1)\, t$

∴ Tearing strength of single eye or rod end

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

Equating this to the load (P) we have

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

From this equation, the induced tensile stress (σ_t) for the single eye or rod end may be checked. In case the induced tensile stress is more than the allowable working stress, then increase the outer diameter of the eye (d_2).

4. *Failure of the single eye or rod end in shearing*

The single eye or rod end may fail in shearing due to tensile load. We know that area resisting shearing $= (d_2 - d_1)\, t$

∴ Shearing strength of single eye or rod end

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

Equating this to the load (P), we have

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

From this equation, the induced shear stress (τ) for the single eye or rod end may be checked.

5. Failure of the single eye or rod end in crushing

The single eye or pin may fail in crushing due to the tensile load. We know that area resisting crushing $= d_1 \times t$

$\therefore$ Crushing strength of single eye or rod end

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

Equating this to the load (P), we have

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

From this equation, the induced crushing stress (σ_c) for the single eye or pin may be checked. In case the induced crushing stress in more than the allowable working stress, then increase the thickness of the single eye (t).

6. Failure of the forked end in tension

The forked end or double eye may fail in tension due to the tensile load. We know that area resisting tearing

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

$\therefore$ Tearing strength of the forked end

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

Equating this to the load (P), we have

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

From this equation, the induced tensile stress for the forked end may be checked.

7. Failure of the forked end in shear

The forked end may fail in shearing due to the tensile load. We know that area resisting shearing

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

$\therefore$ Shearing strength of the forked end

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

Equating this to the load (P), we have

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

From this equation, the induced shear stress for the forked end may be checked. In case, the induced shear stress is more than the allowable working stress, then thickness of the fork (t_1) is increased.

8. Failure of the forked end in crushing

The forked end or pin may fail in crushing due to the tensile load. We know that area resisting crushing $= d_1 \times 2\, t_1$

$\therefore$ Crushing strength of the forked end

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

Equating this to the load (P), we have

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

From this equation, the induced crushing stress for the forked end may be checked.

Problem

1.Designaknucklejointtotransmit150kN.Thedesignstressesmaybetakenas75MPa intension,60MPainshearand150MPaincompression.

Given: $P=150kN=150\times10^3 N; \sigma_t=75MPa=75N/mm^2; \tau=60MPa=60N/mm2; \sigma_c=150MPa=150N/mm^2$

Thejointisdesignedbyconsideringthevariousmethodsoffailureasdiscussed below:

1. *Failure of the solid rod in tension*

Let d = Diameter of the rod.

We know that the load transmitted (P),

$$150 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 75 = 59\, d^2$$

$\therefore$ $d^2 = 150 \times 10^3 / 59 = 2540$ or $d = 50.4$ say 52 mm **Ans.**

Now the various dimensions are fixed as follows :

Diameter of knuckle pin,

$$d_1 = d = 52 \text{ mm}$$

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

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\, d = 1.5 \times 52 = 78 \text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\, d = 1.25 \times 52 = 65 \text{ mm}$$

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

Thickness of pin head, $t_2 = 0.5\, d = 0.5 \times 52 = 26$ mm

2. *Failure of the knuckle pin in shear*

Since the knuckle pin is in double shear, therefore load (P),

$$150 \times 10^3 = 2 \times \frac{\pi}{4} \times (d_1)^2\, \tau = 2 \times \frac{\pi}{4} \times (52)^2\, \tau = 4248\, \tau$$

$\therefore \quad \tau = 150 \times 10^3 / 4248 = 35.3 \text{ N/mm}^2 = 35.3 \text{ MPa}$

3. *Failure of the single eye or rod end in tension*

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \sigma_t = (104 - 52)\, 65 \times \sigma_t = 3380\, \sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

4. *Failure of the single eye or rod end in shearing*

The single eye or rod end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \tau = (104 - 52)\, 65 \times \tau = 3380\, \tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

5. *Failure of the single eye or rod end in crushing*

The single eye or rod end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times t \times \sigma_c = 52 \times 65 \times \sigma_c = 3380\, \sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

6. *Failure of the forked end in tension*

The forked end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \sigma_t = (104 - 52)\, 2 \times 40 \times \sigma_t = 4160\, \sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

7. *Failure of the forked end in shear*

The forked end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \tau = (104 - 52)\, 2 \times 40 \times \tau = 4160\, \tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

8. *Failure of the forked end in crushing*

The forked end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times 2\, t_1 \times \sigma_c = 52 \times 2 \times 40 \times \sigma_c = 4160\, \sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 4180 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

From above, we see that the induced stresses are less than the given design stresses, therefore the joint is safe.

2.Designaknucklejointforatierodofacircularsectiontosustainamaximumpullof70 kN.Theultimatestrengthofthematerialoftherodagainsttearingis420MPa.The ultimatetensileandshearingstrengthofthepinmaterialare510MPaand396MPa respectively.Determinethetierodsectionandpinsection.Takefactorofsafety=6.

Solution. Given : $P = 70$ kN $= 70\,000$ N ; σ_{tu} for rod $= 420$ MPa ; *σ_{tu} for pin $= 510$ MPa $\tau_u = 396$ MPa ; $F.S. = 6$

We know that the permissible tensile stress for the rod material,

$$\sigma_t = \frac{\sigma_{tu} \text{ for rod}}{F.S.} = \frac{420}{6} = 70 \text{ MPa} = 70 \text{ N/mm}^2$$

and permissible shear stress for the pin material,

$$\tau = \frac{\tau_u}{F.S.} = \frac{396}{6} = 66 \text{ MPa} = 66 \text{ N/mm}^2$$

We shall now consider the various methods of failure of the joint as discussed below:

1. *Failure of the rod in tension*

Let d = Diameter of the rod.

We know that the load (P),

$$70\,000 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 70 = 55\,d^2$$

$\therefore$ $d^2 = 70\,000 / 55 = 1273$ or $d = 35.7$ say 36 mm **Ans.**

The other dimensions of the joint are fixed as given below :

Diameter of the knuckle pin,

$$d_1 = d = 36 \text{ mm}$$

Outer diameter of the eye,

$$d_2 = 2\,d = 2 \times 36 = 72 \text{ mm}$$

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\,d = 1.5 \times 36 = 54 \text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\,d = 1.25 \times 36 = 45 \text{ mm}$$

Thickness of fork, $t_1 = 0.75\,d = 0.75 \times 36 = 27$ mm

Now we shall check for the induced streses as discussed below :

2. *Failure of the knuckle pin in shear*

Since the knuckle pin is in double shear, therefore load (P),

$$70\,000 = 2 \times \frac{\pi}{4}\,(d_1)^2\,\tau = 2 \times \frac{\pi}{4}\,(36)^2\,\tau = 2036\,\tau$$

$\therefore$ $\tau = 70\,000 / 2036 = 34.4 \text{ N/mm}^2$

3. *Failure of the single eye or rod end in tension*

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$70\,000 = (d_2 - d_1)\,t \times \sigma_t = (72 - 36)\,45\,\sigma_t = 1620\,\sigma_t$$

$\therefore$ $\sigma_t = 70\,000 / 1620 = 43.2 \text{ N/mm}^2$

4. *Failure of the forked end in tension*

The forked end may fail in tension due to the load. We know that load (P),

$$70\,000 = (d_2 - d_1)\,2\,t_1 \times \sigma_t = (72 - 36) \times 2 \times 27 \times \sigma_t = 1944\,\sigma_t$$

$\therefore$ $\sigma_t = 70\,000 / 1944 = 36 \text{ N/mm}^2$

From above we see that the induced stresses are less than given permissible stresses, therefore the joint is safe.

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

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$70\ 000 = (d_2 - d_1)\, t \times \sigma_t = (72 - 36)\, 45\, \sigma_t = 1620\, \sigma_t$$

$$\therefore \quad \sigma_t = 70\ 000 / 1620 = 43.2 \text{ N/mm}^2$$

4. Failure of the forked end in tension

The forked end may fail in tension due to the load. We know that load (P),

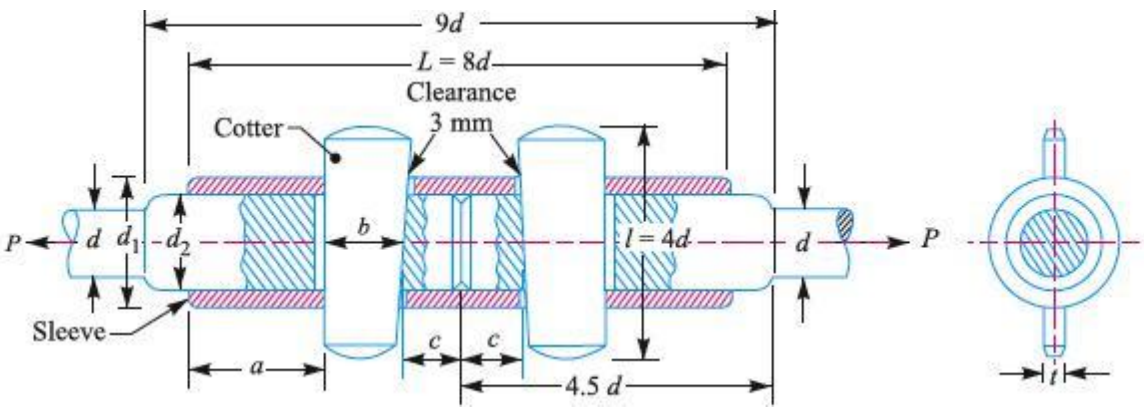
$$70\ 000 = (d_2 - d_1)\, 2\, t_1 \times \sigma_t = (72 - 36) \times 2 \times 27 \times \sigma_t = 1944\, \sigma_t$$

$$\therefore \quad \sigma_t = 70\ 000 / 1944 = 36 \text{ N/mm}^2$$

From above we see that the induced stresses are less than given permissible stresses, therefore the joint is safe.

Sleeveandcoterjoint

- ✓ *AsleeveandcoterjointasshowninFig.isusedtoconnecttworoundrodsor bars.*
- ✓ *Inthistypeofjoint,asleeveormuffisusedoverthetworodsandthentwo coters(oneoneachrodend)areinsertedinthehole providedfortheminthe sleeveandrods.*
- ✓ *Thetaperofcoterisusualy1in24.*
- ✓ *Itmaybenotedthatthetapersidesofthetw ctersshouldfaceeachotheras showninFig.below.*
- ✓ *Theclearanceissoadjustedthatwhenthecotersaredrivenin,thetworods comeclosertoeachothertusmakingthejointtight.*



Thevariousproportionsforthesleeveandcoterjointintermsofthediameterofrod(d) areasfolows:

Outsidediameterofsleeve,

d1=2.5d

Diameterofenlargedendofrod,

d2=Insidediameterofsleeve=1.25d

Lengthofsleeve,L=8d

Thicknessofcoter,t=d2/4or0.31d

Widthofcoter,b=1.25d

Lengthofcoter,l=4d

Distanceoftherodend(a)fromthebeginningtothecoterhole(insidethe sleeveend)=Distanceoftherodend(c)fromitsendtothecoterhole

=1.25d

DesignofSleeveandCoterJoint

1. *Failureoftherodsintension*

The rods may fail in tension due to the tensile load P. We know that

$$\text{Area resisting tearing} = \frac{\pi}{4} \times d^2$$

∴ Tearing strength of the rods

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

Equating this to load (P), we have

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

From this equation, diameter of the rods (d) may be obtained.

2. *Failureoftherodintensionacrosstheweakestsection(i.e.slot)*

Since the weakest section is that section of the rod which has a slot in it for the cotter, therefore area resisting tearing of the rod across the slot

$$= \frac{\pi}{4}(d_2)^2 - d_2 \times t$$

and tearing strength of the rod across the slot

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

Equating this to load (P), we have

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

From this equation, the diameter of enlarged end of the rod (d_2) may be obtained.

3. *Failureoftherodorcoterincrushing*

We know that the area that resists crushing of a rod or cotter

$$= d_2 \times t$$

$$\therefore \quad \text{Crushing strength} = d_2 \times t \times \sigma_c$$

Equating this to load (P), we have

$$P = d_2 \times t \times \sigma_c$$

From this equation, the induced crushing stress may be checked.

4. *Failureofsleeveintensionacrosstheslot*.

We know that the resisting area of sleeve across the slot

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

∴ Tearing strength of the sleeve across the slot

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

Equating this to load (P), we have

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

From this equation, the outside diameter of sleeve (d_1) may be obtained.

5.Failureofcoterinshear

Since the cotter is in double shear, therefore shearing area of the cotter

$$= 2b \times t$$

and shear strength of the cotter

$$= 2b \times t \times \tau$$

Equating this to load (P), we have

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

From this equation, width of cotter (b) may be determined.

6.Failureofrodendinshear

Sincetherodendisindoubleshear,thereforearearesistingshearoftherodend

$$=2a\times d2$$

andshearstrengthoftherodend

$$=2a\times d2\times\tau$$

Equatingthistoload(P),w have

$$P=2a\times d2\times\tau$$

Fromthisequation,distance(a)maybedetermined.

7.Failureofsleeveendinshear

Since the sleeve end is in double shear, therefore the area resisting shear of the sleeve end

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

and shear strength of the sleeve end

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

Equating this to load (P), we have

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

From this equation, distance (c) may be determined.

Problem:

Designasleeveandcoterjointtoresistatensileloadof60kN.Allpartsofthe jointaremadeofthesamematerialwiththefollowingallowablestresses:σt=60

MPa;τ=70MPa;andσc=125MPa.

Solution:Given:P=60kN=60×10^{3}N;σt=60MPa=60N/mm^2;τ=70MPa=70 N/mm2;σc=125MPa=125N/mm2

1.Diameteroftherods

Let d = Diameter of the rods.

Considering the failure of the rods in tension. We know that load (P),

$$60 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 60 = 47.13\, d^2$$

$$\therefore \quad d^2 = 60 \times 10^3 / 47.13 = 1273 \quad \text{or} \quad d = 35.7 \text{ say } 36 \text{ mm}$$

2.Diameterofenlargedendofrodandthicknessofcoter

Let d_2 = Diameter of enlarged end of rod, and

t = Thickness of cotter. It may be taken as $d_2 / 4$.

Considering the failure of the rod in tension across the weakest section (*i.e.* slot). We know that load (P),

$$60 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]60 = 32.13\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 60 \times 10^3 / 32.13 = 1867 \quad \text{or} \quad d_2 = 43.2 \text{ say } 44 \text{ mm}$$ **Ans.**

and thickness of cotter,

$$t = \frac{d_2}{4} = \frac{44}{4} = 11 \text{ mm}$$ **Ans.**

Let us now check the induced crushing stress in the rod or cotter. We know that load (P),

$$60 \times 10^3 = d_2 \times t \times \sigma_c = 44 \times 11 \times \sigma_c = 484\,\sigma_c$$

$$\therefore \quad \sigma_c = 60 \times 10^3 / 484 = 124 \text{ N/mm}^2$$

Since the induced crushing stress is less than the given value of 125 N/mm^2, therefore the dimensions d_2 and t are within safe limits.

3.Outsidediameterofsleeve

Letd$_1$=Outsidediameterofsleeve.

Considering the failure of sleeve in tension across the slot. We know that load (P)

$$60 \times 10^3 = \left[\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\,t\right]\sigma_t$$

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

$$\therefore \quad 60 \times 10^3 / 60 = 0.7854\,(d_1)^2 - 1520.7 - 11\,d_1 + 484$$

$$\text{or} \quad (d_1)^2 - 14\,d_1 - 2593 = 0$$

$$\therefore \quad d_1 = \frac{14 \pm \sqrt{(14)^2 + 4 \times 2593}}{2} = \frac{14 \pm 102.8}{2}$$

$$= 58.4 \text{ say } 60 \text{ mm}$$ **Ans.** ...(Taking +ve sign)

4.Widthofcoter

Let b = Width of cotter.

Considering the failure of cotter in shear. Since the cotter is in double shear, therefore load (P),

$$60 \times 10^3 = 2\, b \times t \times \tau = 2 \times b \times 11 \times 70 = 1540\, b$$

$\therefore$ $b = 60 \times 10^3 / 1540 = 38.96$ say 40 mm **Ans.**

5.Distanceoftherodfromthebeginningtothecoterhole(insidethesleeveend)

Let a = Required distance.

Considering the failure of the rod end in shear. Since the rod end is in double shear, therefore load (P),

$$60 \times 10^3 = 2\, a \times d_2 \times \tau = 2\, a \times 44 \times 70 = 6160\, a$$

$\therefore$ $a = 60 \times 10^3 / 6160 = 9.74$ say 10 mm **Ans.**

6.Distanceoftherodendfromitsendtothecoterhole

Let c = Required distance.

Considering the failure of the sleeve end in shear. Since the sleeve end is in double shear, therefore load (P),

$$60 \times 10^3 = 2\,(d_1 - d_2)\, c \times \tau = 2\,(60 - 44)\, c \times 70 = 2240\, c$$

$\therefore$ $c = 60 \times 10^3 / 2240 = 26.78$ say 28 mm **Ans.**

A 15 kW, 960 rpm motor has a mild steel shaft of 40 mm diameter and the extension being 75 mm. The permissible shear and crushing stresses for the mild steel key are 56 MPa & 112 MPa. Design the keyway in the motor shaft extension. Check the shear strength of the key against the normal strength of the shaft.

Given: $P = 15\ kW = 15 \times 10^3\ W$; $N = 960$ rpm; $d = 40$ mm; $l = 75$ mm

$\tau = 56$ MPa; $\sigma_c = 112$ MPa.

W.K.T

$$P = \frac{2\pi N T}{60} \quad ; T = \frac{P \times 60}{2 \times \pi \times N} = \frac{15 \times 10^3 \times 60}{2 \times \pi \times 960}$$

$$T = 149\ N.m$$

$$T = 149 \times 10^3\ N.mm.$$

Width of Keyway or Key (b):

$$T = l \times b \times \tau \times \frac{d}{2} \qquad \text{Pag : No : 7.133}$$

$$149 \times 10^3 = 75 \times b \times 56 \times \frac{40}{2}$$

$$b = 1.8\ mm$$

∴ Width of the key is too small. The width should be atleast $\frac{d}{4}$

$$b = \frac{d}{4} = \frac{40}{4} = 10\ mm.$$

$\therefore h = 10$ mm

∵ { If $\sigma_c = 2 \times \tau$ Assume Square key

$h \rightarrow$ thickness.

Strength of the shaft with keyway → Pag: No: 7.23

$$M_t \text{ (or) } T = \frac{\pi}{16} \tau \times d^3$$

$$= \frac{\pi}{16} \times 56 \times 40^3$$

$$= 571844\ N \quad \text{(Normal)}$$

Shear strength of the key Pag : No : 7.133

$$= l \times b \times \tau \times d/2$$

$$= 75 \times 10 \times 56 \times 40/2$$

$$= 840000\ N$$

$$\therefore \frac{\text{Shear strength of key}}{\text{Normal Strength of shaft}} = \frac{840000}{571844} = \frac{1.47}{1}$$

Clamp Coupling.

Design a clamp coupling to transmit 30 kW at 1000 rpm. The allowable shear stress for the shaft and key is 40 MPa and the the number of bolts connecting the two halves are six. The permissible tensile stress for the bolt is 70 MPa. The coefficient of friction between the muff and shaft surface are 0.3.

Given: $P = 30 \times 10^3$ Watt ; $N = 1000$ rpm ; $\tau_s = \tau_k = 40$ MPa ;

$n = 6$; $\sigma_{bolt} = 70$ MPa or N/mm^2 ; $\mu = 0.3$

1. Design of shaft

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

$$T = \frac{30 \times 10^3 \times 60}{2 \times \pi \times 1000} \qquad T = 2865 \text{ N.m}$$

$$T = 2865 \times 10^3 \text{ N.mm}$$

W.K.T Torque transmitted — Data book Page: No: 7.23

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

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

$$d = 71.4 \text{ say } 75 \text{ mm.}$$

Pag: No: 7.133

2. Design of Muff

W.K.T diameter of Muff

$$D = 2d + 13 = 2 \times 75 + 13 = 163 \text{ say } 165 \text{ mm}$$

Total Length of the muff

$$L = 3.5d = 3.5 \times 75 = 262.5 \text{ mm}$$

3. Design for Key:

For shaft diameter $d = 75$ mm — Data book Pag: No: 5.21

$b = 22$ mm

$h = 14$ mm

4. Design for bolts :

d_3 is Minor diameter or Core diameter of bolt

W.K.T

$$T = \frac{\pi^2}{16} \times \mu \times (d_3)^2 \times \sigma_b \times n \times d$$

$$2865 \times 10^3 = \frac{\pi^2}{16} \times 0.3 \times d_3^2 \times 70 \times 6 \times 75$$

$$d_3 = 22.2 \text{ mm} \quad \text{Say } 25 \text{ mm.}$$

See page : No : 5.42 in data book and select size of bolt for $d_3 = 25$ mm

Size of bolt is M30 × 3.5

Sleeve or Muff Coupling

Design and make a neat dimensional sketch of a muff coupling which is used to connect two steel shafts, transmitting 40 kW at 350 rpm. The material for the shafts 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 stress may be assumed 15 MPa.

Given: $P = 40 \times 10^3$ Watt ; $N = 350$ rpm ; $\tau_s = 40$ MPa or N/mm^2

$\sigma_{cs} = 80$ MPa or N/mm^2 ; $\tau_{cast} = \tau_c = 15\ N/mm^2$ or MPa

1. Design of Shaft

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

$$T = \frac{40 \times 10^3 \times 60}{2 \times \pi \times 350} \qquad T = 1100\ N.m$$

$$T = 1100\ N.m$$

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

W.K.T

Torque transmitted — Data book Page : No: 7.23

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

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

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

Page : No: 7.133.

2. Design for sleeve

We know that outer diameter

$$D = 2d + 13 = 2 \times 55 + 13 = 123 \text{ say } 125 \text{ mm}$$

Length of the muff

$$L = 3.5d = 3.5 \times 55 = 192.5 \text{ Say } 195 \text{ mm}$$

Check for Induced stress in muff

$$T = \frac{\pi}{16} \times \tau_{cast} \left[\frac{D^4 - d^4}{D}\right]$$ Page : No : 7.21

Muff is in Hollow

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

$$\tau_{cast} = 2.97 \ N/mm^2$$

Since τ_{cast} (Induced) $<$ τ_{cast} (permissible shear stress)

∴ Design of muff is safe.

3. Design for Key : Data book.

For shaft diameter $d = 55$ mm Select key Pag: No: 5.21

$$b = 18$$

Since Crushing stress for key material is twice the shear therefore a square key may be used Pag: No: 5.21

$$h = 18$$

∴ W.K.T the Length of the key

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

Check for Induced shear + Crushing. Pag : No : 7.135

Shear $$T = l \times b \times \tau_s \times \frac{d}{2} = 97.5 \times 18 \times \tau_s \times \frac{55}{2}$$

$$\tau_s = \frac{1100 \times 10^3}{48.2}$$

$$\tau_s = 22.8 \ N/mm^2 \text{ or MPa}$$

Crushing Pag : No : 7.135

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

$$1100 \times 10^3 = 97.5 \times \frac{18}{2} \times \sigma_{cs} \times \frac{55}{2}$$

$$\sigma_{cs} = 45.6 \ N/mm^2 \text{ Or MPa}$$

Since Induced Shear + Crushing stresses are less than permissible stresses, ∴ Design of key is safe.

Design a Cast iron protective type flange Coupling to transmit 15 kW at 900 rpm. from an electric motor to a Compressor. The service factor may be assumed as 1.35. The following permissible stresses

* Shear stress for shaft, bolt + Key material = 40 MPa
* Crushing stress for bolt & Key = 80 MPa
* Shear stress for Cast iron = 8 MPa

Draw a neat sketch of the Coupling

Given: $P = 15 \times 10^3$ Watt ; $N = 900$ rpm ; Service Factor $= 1.35$

$\tau_s = \tau_b = \tau_k = 40$ MPa ; $\sigma_{cb} + \sigma_{ck} = 80$ MPa; $\tau_{cast} = 8$ MPa

1. Design for Hub

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

$$T = \frac{15 \times 10^3 \times 60}{2 \times \pi \times 900} \text{ N.m}$$

$$T = 159.13 \text{ N.m}$$

$$T_{max} = T \times \text{Service Factor}$$

$$= 159.13 \times 1.35$$

$$= 215 \text{ N.m}$$

$$= 215 \times 10^3 \text{ N.mm}$$

W.K.T Pag: No: 7.23

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

$$215 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$d =$ ~~48.4 Say 50 mm~~ 30.1 Say 30 mm

Pag: No: 7.134

W.K.T Outer diameter of the hub

$D = 2d =$ ~~2 × 30 = 100 mm~~ $2 \times 35 = 70$ mm

Length of the hub

$L = 1.5 \times d = 1.5 \times$ ~~d50~~ 35 $= 52.5$ mm

Check for Induced stress for the hub

Data book Pag: No: 7.21 or Pag: No: 7.135

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

$$215 \times 10^3 = \frac{\pi}{16} \tau_{cast} \left[\frac{70^4 - 35^4}{70}\right]$$

$$\tau_{cast} = 3.4 \ N/mm^2 \text{ or MPa}$$

Induced (τ_{cast}) stress is Less than permissible stress

∴ Design of hub is Safe.

Design for Key:

For shaft diameter $d = 35$ mm — Pag: No:- 5.21

Width of Key $b = 12$

∴ W.k.t the Crushing stress is twice the shear stress

So thickness of Key $h = 12$

Square Key are used.

Length of the Key is taken equal to the length of the hub

$$l = L = 52.5 \text{ mm}$$

Check for Induced stress for Shear — Pag: No: 7.135

$$T = l \times b \times \tau_K \times \frac{d}{2}$$

$$215 \times 10^3 = 52.5 \times 12 \ \tau_K \times \frac{35}{2}$$

$$\tau_K = 19.5 \text{ MPa or } N/mm^2$$

Check for Key Crushing — Pag: No: 7.135

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

$$215 \times 10^3 = 52.5 \times \frac{12}{2} \times \sigma_{ck} \times \frac{35}{2}$$

$$\sigma_{ck} = 39 \text{ MPa or } N/mm^2$$

Since the induced stress are Less than the permissible stress.

∴ Design for key is ~~Key~~ Safe.

3. Design for flange :

Thickness of flange t_f is 0.5d

$t_f = 0.5\,d = 0.5 \times 35 = 17.5$ mm.

Check for Induced Stress (shear)

$T_{max} = \frac{\pi D^2}{2} \times \tau_{cast} \times t_f$ Page No: 7.135

$215 \times 10^3 = \frac{\pi \times 70^2}{2} \times \tau_{cast} \times 17.5$

$\tau_{cast} = 1.6$ MPa

∴ Induced stress is less than permissible stress.

Design of flange is safe.

4. Design for bolts

Let d_1 is the major or Nominal diameter

Since diameter of the shaft is 35 mm, therefore number of bolts

$n = 3$

Pitch Circle diameter of bolts

$D_1 = 3d = 3 \times 35 = 105$ mm

Bolt subjected to Shear Stress Page No: 7.135

$T_{max} = \frac{\pi}{4} (d_1)^2 \tau_b \times n \times \frac{D_1}{2}$

$215 \times 10^3 = \frac{\pi}{4} d_1^2 \times 40 \times 3 \times \frac{105}{2}$

$d_1 = 6.6$ mm Page : No: 5.42

∴ Size of the bolt is M8

Page No: 7.134

Outer diameter of the flange

$D_2 = 4d = 4 \times 35 = 140$ mm

Thickness of the protective Circumferential flange Page No: 7.134

$t_p = 0.25\,d = 0.25 \times 35$

$t_p = 8.75$ mm say 10 mm.

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

Given: $P = 90\ kW = 90 \times 10^3\ W$; $N = 250$ rpm; $\tau_{shaft} = 40$ MPa;

$\theta^\circ = 1^\circ = 1 \times \frac{\pi}{180} = 0.0175$ rad ; $\tau_{bolt} = 30$ MPa or N/mm^2;

W.K.T $P = \frac{2\pi N T}{60} \Rightarrow T = \frac{P \times 60}{2\pi N} = \frac{90 \times 10^3 \times 60}{2\pi \times 250}$

$T = 3440$ N.m

$T = 3440 \times 10^3$ N.mm

W.K.T

$\frac{T}{J} = \frac{C\theta}{L}$ } Pag: No: 7.1 $\because J = \frac{\pi}{32} \times d^4$

$\because$ Assume $C = 84\ kN/mm^2$

$\frac{3440 \times 10^3}{\frac{\pi}{32} \times d^4} = \frac{84 \times 10^3 \times 0.0175}{20d}$ Given $L = 20d$.

$\frac{35 \times 10^6}{d^4} = \frac{73.5}{d}$

$d^3 = \frac{35 \times 10^6}{73.5}$

$d = 78$ mm say 80 mm.

1. Design of Hub
2. Design for Key.
3. Design for flange
4. Design for bolts

] Refer the Previous Problem. [Same Procedure].

clutch

A Multiple clutch is to be designed to transmit a power of 50 kW at 500 rpm. Assuming Suitable Material & data.

Given: Power, (P) = 50 kW = 50,000 W

Speed, (N) = 500 rpm.

Solution: Designed power = $[M_t] \times K_W$ — Pag: No: 7.89, $[M_t] = T$

① $P = \frac{2\pi N T}{60}$

$T = \frac{50,000 \times 60}{2 \times \pi \times 500}$

$[M_t]$ or $T = 954.9$ N.m $= 954.9 \times 10^3$ N.mm

K_W is Service Factor. — Assume, Pag: No

$K_W = K_1 + K_2 + K_3 + K_4$

$K_W = 0.5 + 1.25 + 0.2 + 0.75$

$K_W = 2.7$

$K_1 = 0.5 \rightarrow 7.90$

$K_2 = 1.25 \rightarrow 7.91$

$K_3 = 0.2 \rightarrow 7.91$

$K_4 = 0.75 - 7.9$

Design Power $= 954.9 \times 10^3 \times 2.7$

$[M_t] = 2578230$ N.mm

To Find the diameter of clutch shaft: (d) — Pag: No: 7.23

$[M_t]$ or $T = \frac{\pi}{16} \tau d^3$

Assume $\tau = 550$ kgf/cm^2 or $\tau = 55$ N/mm^2

$2578230 = \frac{\pi}{16} \times 55 \times d^3$

$d = 62$ mm

② Inner Radius (r_{min}) + Outer Radius (r_{max}): — Pag: No: 7.90

$r_{min} = 2d = 2 \times 62 = 124$ mm

$r_{max} = 1.25\, r_{min} = 1.25 \times 124 = 155$ mm

} Pag: No: 7.90

$r_m \rightarrow$ Mean Radius $r_m = \frac{r_{max} + r_{min}}{2}$

$r_m = \frac{124 + 155}{2} = 139.5 = 140$ mm

③ Width and thickness of the plate (b + c) — Pag: No: 7.89 (Dimensions of plates)

$b = r_{max} - r_{min}$

$b = 31$ mm

$c = 1$ to 3 mm

$c = 3$ mm Assume Pag: No: 7.90

(4)

Number of friction surfaces:

$i = m_1 + m_2 - 1$

$$i_{min} = \frac{[M_t]}{2 \times \pi \times p_a \times b \times \mu \times r_m^2}$$

$R_{max} = 155$

$D = 310$ mm

$p_a = K \cdot p_b$ Where K = Speed Factor

$V = \frac{\pi \times D \times N}{60}$

$p_a = 0.65 \times 0.6$

K = 0.65

Pag: No: 7.90 (Graph)

$V = \frac{\pi \times 0.31 \times 500}{60}$

$p_a = 0.39$ N/mm^2

V = 8.1 m/sec

$p_b \rightarrow$ for wet running

$$i_{min} = \frac{2578230}{2 \times \pi \times .39 \times 31 \times 0.08 \times 140^2}$$

$p_b = 0.6$ N/mm^2 — Pag: No: 7.89

$\mu = 0.08$ — Pag: No: 7.92

$i = i_{min} = 21.6 \simeq 22$

Pag: No: 7.90

$m_1 = \frac{i}{2} = \frac{22}{2} = 11$; $m_2 = \frac{i}{2} + 1 \simeq \frac{22}{2} + 1 = 12$

Pag: No: 7.90

(5) Actual pressure b/w plates:

$$\sigma = \frac{[M_t]}{2\pi i b \mu r_m^2} = \frac{[2578230]}{2 \times \pi \times 22 \times 31 \times 0.08 \times 140^2} = 0.38 \text{ N/mm}^2$$

Axial Force $Q = \pi \sigma (r_{max}^2 - r_{min}^2)$

$= \pi \times 0.38 (155^2 - 124^2)$

$Q = 10,325$ N.

Force on the lever

Pag: No: 7.90

$$Q' = \frac{Q}{i_{el}}$$

$i_{el} = 3$ or 4

Assume = 4

$$Q' = \frac{10,325}{4} = 2581.25 \text{ N}$$

Knuckle Joint

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

Given: P = 150 kN = 150×10^3 N; σ_t = 75 MPa = 75 N/mm^2; τ = 60 MPa = 60 N/mm2; σ_c = 150 MPa = 150 N/mm^2

The joint is designed by considering the various methods of failure as discussed below:

1. *Failure of the solid rod in tension*

Let d = Diameter of the rod.

We know that the load transmitted (P),

$$150 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 75 = 59\, d^2$$

∴ $d^2 = 150 \times 10^3 / 59 = 2540$ or $d = 50.4$ say 52 mm **Ans.**

Now the various dimensions are fixed as follows :

Diameter of knuckle pin,

$$d_1 = d = 52 \text{ mm}$$

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

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\,d = 1.5 \times 52 = 78 \text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\,d = 1.25 \times 52 = 65 \text{ mm}$$

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

Thickness of pin head, $t_2 = 0.5\,d = 0.5 \times 52 = 26$ mm

2. *Failure of the knuckle pin in shear*

Since the knuckle pin is in double shear, therefore load (P),

$$150 \times 10^3 = 2 \times \frac{\pi}{4} \times (d_1)^2\, \tau = 2 \times \frac{\pi}{4} \times (52)^2\, \tau = 4248\, \tau$$

∴ $\tau = 150 \times 10^3 / 4248 = 35.3 \text{ N/mm}^2 = 35.3$ MPa

3. *Failure of the single eye or rod end in tension*

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \sigma_t = (104 - 52)\, 65 \times \sigma_t = 3380\, \sigma_t$$

∴ $\sigma_t = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4$ MPa

4. *Failure of the single eye or rod end in shearing*

The single eye or rod end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \tau = (104 - 52)\, 65 \times \tau = 3380\, \tau$$

∴ $\tau = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4$ MPa

5. *Failure of the single eye or rod end in crushing*

The single eye or rod end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times t \times \sigma_c = 52 \times 65 \times \sigma_c = 3380\, \sigma_c$$

∴ $\sigma_c = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4$ MPa

6. *Failure of the forked end in tension*

The forked end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,2\,t_1 \times \sigma_t = (104 - 52)\,2 \times 40 \times \sigma_t = 4160\,\sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 4160 = 36\text{ N/mm}^2 = 36\text{ MPa}$$

7. *Failure of the forked end in shear*

The forked end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,2\,t_1 \times \tau = (104 - 52)\,2 \times 40 \times \tau = 4160\,\tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 4160 = 36\text{ N/mm}^2 = 36\text{ MPa}$$

8. *Failure of the forked end in crushing*

The forked end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times 2\,t_1 \times \sigma_c = 52 \times 2 \times 40 \times \sigma_c = 4160\,\sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 4180 = 36\text{ N/mm}^2 = 36\text{ MPa}$$

From above, we see that the induced stresses are less than the given design stresses, therefore the joint is safe.

2. *Design a knuckle joint for a tie rod of a circular section to sustain a maximum pull of 70 kN. The ultimate strength of the material of the rod against tearing is 420 MPa.* ***The ultimate tensile and shearing strength*** *of the pin material are* ***510 MPa and 396 MPa*** *respectively. Determine the tie rod section and pin section. Take factor of safety = 6.*

Solution. Given : $P = 70\text{ kN} = 70\,000\text{ N}$; σ_{tu} for rod $= 420$ MPa ; *σ_{tu} for pin $= 510$ MPa $\tau_u = 396$ MPa ; $F.S. = 6$

We know that the permissible tensile stress for the rod material,

$$\sigma_t = \frac{\sigma_{tu}\text{ for rod}}{F.S.} = \frac{420}{6} = 70\text{ MPa} = 70\text{ N/mm}^2$$

and permissible shear stress for the pin material,

$$\tau = \frac{\tau_u}{F.S.} = \frac{396}{6} = 66\text{ MPa} = 66\text{ N/mm}^2$$

A helical Spring is made from a Wire of 6 mm diameter and has outside diameter of 75 mm. If the permissible shear stress is 350 MPa and Modulus of rigidity 84 kN/mm², find the axial Load which spring may carry and the deflection per active turn.

Given ; $d = 6$ mm; $D_o = 75$ mm; $\tau = 350$ MPa or N/mm²; $G = 84 \times 10^3$ N/mm²

W.K.T Mean diameter of the spring

$$D = D_o - d$$
$$= 75 - 6$$
$$D = 69 \text{ mm}$$

$\therefore$ Spring Index (C) $= \frac{D}{d} = \frac{69}{6}$ → Pag:No: 7.100

$$C = 11.5$$

To Find $W = ?$

$y/n = ?$

1. Neglecting the effect of Curvature. Pag:No: 7.100

$$K_s = 1 + \frac{1}{2C} = 1 + \frac{1}{2 \times 11.5} = 1.043$$

Max. Shear stress (τ)

$$\tau = K_s \times \frac{8PD}{\pi d^3} = 1.043 \times \frac{8 \times W \times 69}{\pi \times 6^3} = 0.848\,W$$

$$W = \frac{350}{0.848} = 412.7 \text{ N}$$

Deflection

$$y = \frac{8PD^3 n}{Gd^4}$$ → Pag: No: 7.100

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

Deflection per turn

$$\frac{y}{n} = 9.96 \text{ mm}$$

2. Considering the effect of Curvature.

Wahl Stress factor

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

Max. Shear Stress

$$\tau = K_s \frac{8PC}{\pi d^2}$$ → Pag: No: 7.100

$$= \frac{1.123 \times 8 \times W \times 11.5}{\pi 6^2}$$

$$350 = 0.913\, W$$

$$W = 383.4\ N$$

Deflection of the spring

$$y = \frac{8PC^3 n}{Gd} \quad \text{or} \quad \frac{8PD^3 n}{Gd^4}$$ → Pag: No: 7.100

$$\frac{y}{n} = \frac{8 \times 383.4 \times (11.5)^3}{84 \times 10^3 \times 6}$$

$$\frac{y}{n} = 9.26\ mm.$$

,., y,, \,,.t«c« x, ••=•»•u- b+' *ººº ^ ºvxv º ‹cab.

of Length 80 mm. The spring is to be enclosed in a casing of 25 mm diameter. The approximate number of turns is 30. The modulus of rigidity is 85 KN/mm². Also calculate the maximum shear stress induced.

Given; $W = P$ 1000 N y 80 mm ; $n = 30$; $G = 85 \times 10^3$ N/mm²

∵ Since the spring is to be enclosed in a casing of 25 mm diameter

∴ the outer diameter of the spring coil ($D_0 = D + d$) should be less

W.K.T $\quad ^3 . n \quad \frac{8PC^3 . n}{}$ → Pg : No: 7.100

$$80 \quad \frac{8 \times 1000 \times C^3 . 30}{85 \times 10^3 \times d} \quad \frac{240\,C^3}{85\,d}$$

$$C^3/d \quad 28.3$$

Let us assume that $d = 4$ mm

$= 113.2$

∴ Outer diameter of

$= 19.36 + 4$

$= 23.36$ mm

3.36 mm is Less than the casing diameter 25 mm

Max. shear stress

K_s · „c - i , t• s _tt.* «'»«y-i a v•» = i s•”-

$4C - 4 \quad C \quad (4 \times 4.84) - 4 \quad 4.84$

u.x'›

$\tau = K_s \times \frac{8PC}{} = 1.322 \times 8 \times 1000 \times 4.84$

'* v› • 4"

. l ie > * 1 >x " *•p ^º ' * "'

A Mechanism used in printing machinery Consists of a tension Spring assembled With a preload of 30N. The wire diameter of spring is 2 mm With a spring index of 6. The spring has 18 active coils. The spring Wire is hard drawn and oil tempered having Material properties;

Design shear stress = 680 MPa

Modulus of Rigidity = 80 KN/mm². Determine: 1. Initial Shear stress; 2. Spring rate and 3. force Cause the body of the spring to its yield strength.

Given: $P_i = 30\ N$; $d = 2\ mm$; $C = \frac{D}{d} = 6$; $n = 18$; $\tau = 680\ MPa$

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

1. Initial shear stress

$$\tau_i = K_s \times \frac{8 P_i C}{\pi d^2} \quad \rightarrow \text{Pag: No: 7.100}$$

$$K_s = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{(4 \times 6) - 1}{(4 \times 6) - 4} + \frac{0.615}{6} = 1.25$$

$$\tau_i = 1.25 \times \frac{8 \times 30 \times 6}{\pi \times 2^2}$$

$$\tau_i = 143.5\ MPa \text{ or } N/mm^2$$

2. Spring rate (or) Stiffness

$$q = \frac{G.d}{8 C^3 . n} \quad \rightarrow \text{Pag: No: 7.100}$$

$$q = \frac{84 \times 10^3 \times 2}{8 \times 6^3 \times 18}$$

$$q = 5.144\ N/mm$$

3. Force to cause the body of the spring to its yield strength

$$\tau = \frac{K_s \times 8 \times P \times C}{\pi d^2} \quad \rightarrow \text{Pag: No: 7.100}$$

$$680 = 1.25 \times \frac{8 \times P \times 6}{\pi \times 2^2} \Rightarrow 4.78\ P$$

$$P = 680/4.78$$

$$P = 142.25\ N$$

q<a-,j» a i«i.c=1 c•-9••••'»• *P•'•g 5°• • "•'xi=t,q »$ •q in» i4
jp, a d•}tsxI' on ‹•} ** ^"• "º'*fl *"^- *º ^^ ºt *P"•g ‘"xle› ax 5 lk»
maximum shear stress for spring wire is 420 MPa & Modulus of rigidity is

Given $P = 1000\ N$; $y = 25\ mm$; $C = 5$; $\tau = 420\ MPa$ or N/mm^2

$G = 84\ KN/mm^2 = 84 \times 10^3\ N/mm^2$.

W.K.T

$K_s \quad \frac{4C-1}{} + \frac{0.615}{} \qquad \frac{(4\times5)-1}{} + \frac{0.615}{} \qquad 1.31$

→

$\tau \quad K_s \times \frac{8PC}{\pi d^2} \ ; \quad 420 \quad \frac{1.31 \times 8 \times 1000 \times 5}{\pi \times d^2}$

Mean diameter of the Spring coil (D)

$D = C \cdot d \qquad \left[C \quad \frac{D}{d} \right]$

$= 5 \times 6.3$

Outer Diameter of the spring coil (D_o)

$D_o = D + d = 32 + 6.3$

2. Number of turns of coils

$y \quad \frac{8PC^3 n}{G \cdot d} \ ; \quad 25 \qquad \frac{1000\ y \quad y\ b,}{84 \times 10^3 \times 6.3} \qquad$ toy:bo•7•tDO

$n = 13.4$ say 14.

For squared & ground ends, the total number of turns

$n' = n + 2 = 14 + 2 = 16.$ → Page No: 7.101

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

$= 16 \times 6.3 + 25 + 0.15 \times 25$

$L_f \quad 131.2\ mm$

4. Pitch of the coil

$\text{Pitch} \quad \frac{\text{Free Length}}{n' - 1} \qquad \frac{131.2}{16 - 1}$

Design a close coiled helical compression spring for a service Load ranging from 2250 N to 2750 N. The axial deflection of the spring for Load range is 6 mm. Assume a spring index of 5. The permissible sh tress is 420 MPa + Modulus of rigidity $G = 84 \text{ kN/mm}^2$. Draw a fully dimensioned sketch of the spring, showing details of the coils.

Given: $P_1 = 2250$ N ; $P_2 = 2750$ N ; $y = 6$ mm ; $C = \frac{D}{d} = 5$; $\tau =$

the

$T = P_2 \times \left(\frac{D}{2}\right)$ → Radius

$2750 \times \frac{5d}{2}$

$6875\, d$

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

$5 = \frac{D}{d}$

$D = 5d$

I.*5

$T \;\; \frac{\pi}{} \tau \times d^3$

72

$D = 45.65$ mm

Outer diameter of the spring coil

$D_o \;\; D + d$

$= 45.65 + 9.13$

$=$ ~~36.52 mm~~ 54.78 mm

Inner diameter

$D_i \;\; D \;\; d$

9.13

2. Number of turns of the Spring coil

It is Given that deflection (y) for the Load range from 2250 N to 2750 N is 6 mm

i.e Range = 2750 - 2250

$P_{range} = 500$ N

$$y = \frac{8.P.C^3.n}{Gd} \rightarrow \text{Pag: No: 7.100}$$

$$6 = \frac{8 \times 500 \times 5^3 \times n}{84 \times 10^3 \times 9.13}$$

$n = 9.2$ Say 10

For squared and ground ends, the total number of turns

$n' = n + 2$ → Pag: No: 7.101

$= 10 + 2$

$n' = 12$

3. Free Length of the spring

Since For Load 500 N the deflection is 6 mm

For Max Load of 2750 the deflection is

$$\frac{6}{500} = \frac{y_{max}}{2750}$$

$$y_{max} = \frac{6}{500} \times 2750$$

$y_{max} = 33$ mm

$$L_f = n'd + y_{max} + 0.15\, y_{max}$$

$$= 12 \times 9.13 + 33 + 0.15 \times 33$$

$L_f = 147.51$ mm

4. Pitch of the Coil

$$\text{Pitch} = \frac{\text{Free Length}}{n' - 1} = \frac{147.51}{12 - 1}$$

$= 11.3$ mm

A rail Wagon of mass 20 tonnes is moving With a Velocity of 2 m/s. It is brought to rest by two buffers With Springs of 300 mm diameter. The maximum deflection of Springs is 250 mm. The allowable shear stress in Spring material is 600 MPa. Design the Spring for the buffers.

Given: $m = 20 \text{ tonnes} = 20 \times 10^3$ kg ; $v = 2$ m/s ; $D = 300$ mm ;

$y = 250$ mm ; $\tau = 600$ MPa or N/mm^2.

$$\text{Kg} \times \text{m}^2/\text{sec}^2$$

$$N = \text{kg.m/sec}^2$$

Diameter of the Spring Wire :

Kinetic energy of the wagon

$$= \frac{1}{2} m v^2$$

$$= \frac{1}{2} \times 20 \times 10^3 \times 2^2$$

$$= 40000 \text{ N.m}$$

$$= 40 \times 10^6 \text{ N.mm.} \quad \text{— (i)}$$

Pag:No: 7.100

Energy Stored in Spring

$$U = \frac{1}{2} P y = \frac{1}{2} \times W \times 250 \quad \text{— (ii)}$$

(P or W)

$$U = 250 \text{ W N.mm}$$

Equating (i) & (ii)

$$P \text{ or } W = \frac{40 \times 10^6}{250}$$

$$P \text{ (or) } W = 160 \times 10^3 \text{ N}$$

W.K.T Torque

R → Radius

$$T = P \times R$$

$$= 160 \times 10^3 \times \left(\frac{300}{2}\right)$$

$$T = 24 \times 10^6 \text{ N.mm}$$

W.K.T Torque

$$T = \frac{\pi}{16} \tau d^3 \quad \rightarrow \text{Pag:No: 7.231}$$

$$24 \times 10^6 = \frac{\pi}{16} \times 600 \times d^3$$

$$d^3 = 203.7 \times 10^3$$

$$d = 58.8 \text{ Say } 60 \text{ mm.}$$

Number of turns of the Spring coil.

Assume $G = 84$ kN/mm^2

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

$$250 = \frac{8 \times 160 \times 10^3 \times 300^3 . n}{84 \times 10^3 \times 60^4}$$

Pag:No: 7.101

$$n = 7.88 \text{ Say } 8.$$

Assuming Square and ground ends, total number of turns,

$$n' = n + 2$$

$$= 8 + 2$$

$$n' = 10$$

Free Length of the Spring

$$L_f = n'd + y + 0.15\,y$$

$$= 10 \times 60 + 250 + 0.15 \times 250$$

$$= 887.5 \text{ mm}$$

Pitch of the coil

$$p = \frac{\text{Free Length}}{n' - 1}$$

$$= \frac{887.5}{10 - 1}$$

$$p = 98.6 \text{ mm}$$

Leaf Spring (or) Laminated Spring

Design a leaf Springs for the following Specifications :

Total Load = 140 KN;

= 4 ; number of Leaves = 10 ; on of

Spring = 1000 mm ; permissible deflection = 80 mm

$E = 200\ KN/mm^2$ & allowable stress in Spring materials

as 600 MPa

,¿q, n ncr=t La•d= I4° KN ; No. of Spring

$2L = 1000$ mm or $L = 500$ mm ; $y = 80$ mm ; $E = 200 \times 10^3\ N/mm^2$;

$\sigma_b = 600\ MPa = 600\ N/mm^2$.

W.K.T Load on each Spring

$$= \frac{\text{Total Load}}{\text{No of Spring}} = \frac{140\ KN}{4} \quad 35\ KN$$

Load on each Spring 35 KN

2P 17.5 KN

b = Width of the Leaves.

Bending Stress In Data book

$$600 \quad \frac{6 \times 17500 \times 500}{\ }$$

b oo

< < T d›j;h=lio=] r'-• * P’'*§ d

In Data book

Pag: No: 7.104

$$y \quad \frac{6PL^3}{\text{Eo bt'}}$$

$$80 = \frac{6 \times 17500 \times 500^3}{n \times 200 \times 10^3 \times b \times t^3} = \frac{65.6 \times 10^6}{n.b.t^3}$$

$$n.b.t^3 = \frac{65.6 \times 10^6}{80} = 0.82 \times 10^6 \quad \text{---- (ii)}$$

Dividing equation (ii) by equation (i)

$$\frac{n.b.t^3}{n.b.t^2} = \frac{0.82 \times 10^6}{87.5 \times 10^3}$$

$$t = 9.37 \text{ say } 10 \text{ mm}$$

Now Subst t in eq (i)

$$b = \frac{87.5 \times 10^3}{n.t^2} = \frac{87.5 \times 10^3}{10 \times 10^2}$$

$$b = 87.5 \text{ mm}$$

& subst t in eq (ii)

$$b = \frac{0.82 \times 10^6}{n.t^3} = \frac{0.82 \times 10^6}{10 \times 10^3}$$

$$b = 82 \text{ mm}$$

Taking larger of two values

$$b = 87.5 \text{ Say } 90 \text{ mm}.$$

A truck spring has 12 number of leaves, two of which are full length leaves. The spring supports are 1.05 m apart & the central band is 85 mm wide. The central load is to be 5.4 KN with a permissible stress of 280 MPa. Determine the thickness & width of the steel spring leaves. The ratio of the total depth of the spring is 3. Also detrime the deflection of the spring.

Given: $n = 12$; $n_e = 2$; $2L_1 = 1.05 \text{ m} = 1050 \text{ mm}$; $l = 85 \text{ mm}$

$2P = 5.4 \times 10^3 \text{ N}$; $P = 2700 \text{ N}$; $\sigma_{be} = 280 \text{ MPa or N/mm}^2$

The ratio of total depth to width is

$$\frac{n \times t}{b} = 3$$

$$b = \frac{n \times t}{3} = \frac{12 \times t}{3} = 4t$$

W.K.T the effective Length of the Spring

$$2L = 2L_1 - l$$

$$= 1050 - 85 = 965 \text{ mm}$$

$$L = \frac{965}{2} = 482.5 \text{ mm}$$

Number of graduated Leaves

$$n = n_e + n_g$$

$$12 = 2 + n_g$$

$$n_g = 10$$

Bending stress of spring with extra full length Leaves

In data book Pag: No: 7.104

$$\sigma_{be} = \frac{18 PL}{bt^2 (3n_e + 2n_g)}$$

$$280 = \frac{18 \times 482.5 \times 2700}{4t \cdot t^2 (3 \times 2 + 2 \times 10)}$$

$$280 = \frac{225476}{t^3}$$

$$t = 9.3 \quad \text{Say } 10 \text{ mm}$$

W.K.T

$$b = 4 \times t$$

$$b = 4 \times 10$$

$$b = 40 \text{ mm.}$$

Deflection of the spring (y)

In data book Pag: No: 7.104

$$y = \frac{12 PL^3}{E\, bt^3 (3n_e + 2n_g)}$$

E is missing in data book.

$$y = \frac{12 \times 2700 \times 482.5^3}{2.1 \times 10^5 \times 40 \times 10^3 (3 \times 2 + 2 \times 10)}$$

$$y = 16.7 \text{ mm}$$

If E is not given take $E = 2.1 \times 10^5$ for steel Pag: No: 8.14

A Locomotive Semi-elliptical Laminated Spring has an overall length of 1 m and Sustains a Load of 70 KN at its Centre the spring has 3 full length leaves and 15 graduated leaves with ...tral band of 100 mm Width. All the leaves are to be stressed .. 400 MPa, When fully Loaded. The ratio of total spring depth to that of width is 2. E = 210 kN/mm² Determine;

1. thickness and width of the
2. The initial gap that should be provided between the full length and graduated leaves before the band load is applied
3. The Load exerted pn uiu bouA after the spring is assembled

Given $2L_1 = 1m = 1000$ mm $2P$ 70×10^3 N, P 35×10^3 N

Know that Total no of Leaves

$= 3 + 15$

the

b

$b \quad \frac{18 \times t}{2}$

1000 ifiO

$2L = 900$ mm

σ_{be}

$400 \quad \frac{18 \times 35 \times 10^3 \times 450}{9t \times t^2 (3 \times 3 + 2 \times 15)}$

$t = 12.6$ mm $\therefore b = 9 \times t$

Say 13 mm $b = 9 \times 13$

$b = 117$ mm

Initial Gap (c) : (or) Nip

W.K.T the Initial gap should be provided between full length Leaves and graduated Leaves before the band Load is applied

$$h \text{ (or) } C = \frac{2WL^3}{n \cdot E \cdot b \cdot t^3}$$ → Pag: No: 7.104

In Data book Pag: No: 7.104

$y = \frac{6PL^3}{Enbt^3}$ Insted of 6 use 2

$$h \text{ (or) } C = \frac{2 \times 35 \times 10^3 \times 450^3}{18 \times 210 \times 10^3 \times 117 \times 13^3}$$

$h \text{ (or) } C = 6.56$ mm

In data book (h)

$h \text{ (or) } C = 7$ mm

Load exerted on the band after the spring is assembled

$$P_b = \frac{2 \cdot n_e \cdot n_g \cdot P}{n(2n_g + 3n_e)}$$ Pag: No: 7.104

$$= \frac{2 \times 3 \times 15 \times 35 \times 10^3}{18(2 \times 15 + 3 \times 3)}$$

$P_b = 4487$ N

unit-v

Problem 1

A machine component is subjected to a flexural stress which fluctuates between + 300 MN/m2 and – 150 MN/m2. Determine the value of minimum ultimate strength according to Goodman relation; and Soderberg relation. Take yield strength = 0.55 Ultimate strength; Endurance strength = 0.5 Ultimate strength; and factor of safety = 2. Solution.

Given : $\sigma_1 = 300$ MN/m^2 ; $\sigma_2 = -150$ MN/m^2 ; $\sigma_y = 0.55\,\sigma u$; $\sigma_{-1} = 0.5\,\sigma_u$; $F.S. = 2$

Let σ_u = Minimum ultimate strength in MN/m^2.

We know that the mean or average stress,

$$\sigma_m = \frac{\sigma_1 + \sigma_2}{2} = \frac{300 + (-150)}{2} = 75 \text{ MN/m}^2$$

and variable stress,
$$\sigma_a = \frac{\sigma_1 - \sigma_2}{2} = \frac{300 - (-150)}{2} = 225 \text{ MN/m}^2$$

According to Goodman relation

We know that according to modified Goodman relation,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma_{-1}}$$

or
$$\frac{1}{2} = \frac{75}{\sigma_u} + \frac{225}{0.5\,\sigma_u} = \frac{525}{\sigma_u}$$

$\therefore$
$$\sigma_u = 2 \times 525 = 1050 \text{ MN/m}^2 \quad \textbf{Ans.}$$

According to Soderberg relation

We know that according to Soderberg relation,

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_{-1}}$$

or
$$\frac{1}{2} = \frac{75}{0.55\,\sigma_u} + \frac{255}{0.5\,\sigma_u} = \frac{586.36}{\sigma_u}$$

$\therefore$
$$\sigma_u = 2 \times 586.36 = 1172.72 \text{ MN/m}^2 \quad \textbf{Ans.}$$

Problem 2

A bar of circular cross-section is subjected to alternating tensile forces varying from a minimum of 200 kN to a maximum of 500 kN. It is to be manufactured of a material with an ultimate tensile strength of 900 MPa and an endurance limit of 700 MPa. Determine the diameter of bar using safety factors of 3.5 related to ultimate tensile strength and 4 related to endurance limit and a stress concentration factor of 1.65 for fatigue load. Use Goodman straight line as basis for design.

Solution. Given : $W_{min} = 200$ kN ; $W_{max} = 500$ kN ; $\sigma_u = 900$ MPa $= 900$ N/mm^2 ; $(F.S.)_u = 3.5$; $(F.S.)_e = 4$; $K_f = 1.65$; $\sigma_{-1} = 700$ MPa $= 700$ N/mm^2 ;

Let d = Diameter of bar in mm.

$\therefore$
$$\text{Area, } A = \frac{\pi}{4} \times d^2 = 0.7854\, d^2 \text{ mm}^2$$

We know that mean or average force,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{500 + 200}{2} = 350 \text{ kN} = 350 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{350 \times 10^3}{0.7854\, d^2} = \frac{446 \times 10^3}{d^2} \text{ N/mm}^2$$

$$\text{Variable force, } W_a = \frac{W_{max} - W_{min}}{2} = \frac{500 - 200}{2} = 150 \text{ kN} = 150 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Variable stress, } \sigma_a = \frac{W_a}{A} = \frac{150 \times 10^3}{0.7854\, d^2} = \frac{191 \times 10^3}{d^2} \text{ N/mm}^2$$

We know that according to Goodman's formula,

$$\frac{\sigma_a}{\sigma_{-1} / (F.S.)_e} = 1 - \frac{\sigma_m . K_f}{\sigma_u / (F.S.)_u}$$

$$\frac{\frac{191 \times 10^3}{d^2}}{700/4} = 1 - \frac{\frac{446 \times 10^3}{d^2} \times 1.65}{900/3.5}$$

$$\frac{1100}{d^2} = 1 - \frac{2860}{d^2} \quad \text{or} \quad \frac{1100 + 2860}{d^2} = 1$$

$$d^2 = 3960 \text{ or } d = 62.9 \text{ say } 63 \text{ mm} \quad \textbf{Ans.}$$

Problem 3

Determine the thickness of a 120 mm wide uniform pla for safe continuous operation if the plate is to be subjected to a tensile load that has a maximum value of 250 kN and a minimum value of 100 kN. The properties of the plate material are as follows: Endurance limit stress = 225 MPa, and Yield point stress = 300 MPa. The factor of safety based on yield point may be taken as 1.5.

Solution. Given : $b = 120$ mm ; $W_{max} = 250$ kN; $W_{min} = 100$ kN ; $\sigma_{-1} = 225$ MPa $= 225$ N/mm^2 ; $\sigma_y = 300$ MPa $= 300$ N/mm^2; $F.S. = 1.5$

Let $\quad t$ = Thickness of the plate in mm.

$$\therefore \quad \text{Area, } A = b \times t = 120\, t \text{ mm}^2$$

We know that mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{250 + 100}{2} = 175 \text{ kN} = 175 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{175 \times 10^3}{120t} \text{ N/mm}^2$$

$$\text{Variable load, } W_a = \frac{W_{max} - W_{min}}{2} = \frac{250 - 100}{2} = 75 \text{ kN} = 75 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Variable stress, } \sigma_a = \frac{W_a}{A} = \frac{75 \times 10^3}{120t} \text{ N/mm}^2$$

According to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_{-1}}$$

$$\frac{1}{1.5} = \frac{175 \times 10^3}{120t \times 300} + \frac{75 \times 10^3}{120t \times 225} = \frac{4.86}{t} + \frac{2.78}{t} = \frac{7.64}{t}$$

$$\therefore \quad t = 7.64 \times 1.5 = 11.46 \text{ say } 11.5 \text{ mm} \quad \textbf{Ans.}$$

Problem 4

Determine the diameter of a circular rod made of ductile material with a fatigue strength (complete stress reversal), σ_{-1} = 265 MPa and a tensile yield strength of 350 MPa. The member is subjected to a varying axial load from W_{min} = – 300 × 10^3 N to W_{max} = 700 × 10^3 N and has a stress concentration factor = 1.8. Use factor of safety as 2.0.

Solution. Given : σ_{-1} = 265 MPa = 265 N/mm^2 ; σ_y = 350 MPa = 350 N/mm^2 ; W_{min} = – 300 × 10^3 N ; W_{max} = 700 × 10^3 N ; K_f = 1.8 ; F.S. = 2

Let d = Diameter of the circular rod in mm.

$$\therefore \quad \text{Area, } A = \frac{\pi}{4} \times d^2 = 0.7854\, d^2 \text{ mm}^2$$

We know that the mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{700 \times 10^3 + (-300 \times 10^3)}{2} = 200 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{200 \times 10^3}{0.7854\, d^2} = \frac{254.6 \times 10^3}{d^2} \text{ N/mm}^2$$

$$\text{Variable load, } W_a = \frac{W_{max} - W_{min}}{2} = \frac{700 \times 10^3 - (-300 \times 10^3)}{2} = 500 \times 10^3 \text{ N}$$

$$\therefore \text{ Variable stress, } \sigma_a = \frac{W_a}{A} = \frac{500 \times 10^3}{0.7854\, d^2} = \frac{636.5 \times 10^3}{d^2} \text{ N/mm}^2$$

We know that according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a \times K_f}{\sigma_{-1}}$$

$$\frac{1}{2} = \frac{254.6 \times 10^3}{d^2 \times 350} + \frac{636.5 \times 10^3 \times 1.8}{d^2 \times 265} = \frac{727}{d^2} + \frac{4323}{d^2} = \frac{5050}{d^2}$$

$$\therefore \quad d^2 = 5050 \times 2 = 10\,100 \quad \text{or} \quad d = 100.5 \text{ mm}$$ **Ans.**

Problem 5

A circular bar of 500 mm length is supported freely at its two ends. It is acted upon by a central concentrated cyclic load having a minimum value of 20 kN and a maximum value of 50 kN. Determine the diameter of bar by taking a factor of safety of 1.5, size effect of 0.85, surface finish factor of 0.9. The material properties of bars are given by: ultimate strength of 650 MPa, yield strength of 500 MPa and endurance strength of 350 MPa.

Solution. Given : l = 500 mm ; W_{min} = 20 kN = 20 × 10^3 N ; W_{max} = 50 kN = 50 × 10^3 N ; F.S. = 1.5 ; K_{sz} = 0.85 ; K_{sur} = 0.9 ; σ_u = 650 MPa = 650 N/mm^2 ; σ_y = 500 MPa = 500 N/mm^2 ; σ_{-1} = 350 MPa = 350 N/mm^2

Let d = Diameter of the bar in mm.

We know that the maximum bending moment,

$$M_{max} = \frac{W_{max} \times l}{4} = \frac{50 \times 10^3 \times 500}{4} = 6250 \times 10^3 \text{ N-mm}$$

and minimum bending moment,

$$M_{min} = \frac{W_{min} \times l}{4} = \frac{20 \times 10^3 \times 500}{4} = 2550 \times 10^3 \text{ N-mm}$$

∴ Mean or average bending moment,

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{6250 \times 10^3 + 2500 \times 10^3}{2} = 4375 \times 10^3 \text{ N-mm}$$

and variable bending moment,

$$M_a = \frac{M_{max} - M_{min}}{2} = \frac{6250 \times 10^3 - 2500 \times 10^3}{2} = 1875 \times 10^3 \text{ N-mm}$$

Section modulus of the bar,

$$Z = \frac{\pi}{32} \times d^3 = 0.0982\, d^3 \text{ mm}^3$$

∴ Mean or average bending stress,

$$\sigma_m = \frac{M_m}{Z} = \frac{4375 \times 10^3}{0.0982\, d^3} = \frac{44.5 \times 10^6}{d^3} \text{ N/mm}^2$$

and variable bending stress,

$$\sigma_a = \frac{M_a}{Z} = \frac{1875 \times 10^3}{0.0982\, d^3} = \frac{19.1 \times 10^6}{d^3} \text{ N/mm}^2$$

We know that according to Goodman's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a \times K_f}{\sigma_{-1} \times K_{sur} \times K_{sz}}$$

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 650} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \quad \text{...(Taking } K_f = 1)$$

$$= \frac{68 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{139 \times 10^3}{d^3}$$

$$\therefore \quad d^3 = 139 \times 10^3 \times 1.5 = 209 \times 10^3 \text{ or } d = 59.3 \text{ mm}$$

and according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a \times K_f}{\sigma_{-1} \times K_{sur} \times K_{sz}}$$

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 500} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \quad \text{...(Taking } K_f = 1)$$

$$= \frac{89 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{160 \times 10^3}{d^3}$$

$$\therefore \quad d^3 = 160 \times 10^3 \times 1.5 = 240 \times 10^3 \text{ or } d = 62.1 \text{ mm}$$

Taking larger of the two values, we have $d = 62.1$ mm **Ans.**

Problem 6

A 50 mm diameter shaft is made from carbon steel having ultimate tensile strength of 630 MPa. It is subjected to a torque which fluctuates between 2000 N-m to – 800 N-m. Using Soderberg method, calculate the factor of safety. Assume suitable values for any other data needed.

Solution. Given : $d = 50$ mm ; $\sigma_u = 630$ MPa $= 630$ N/mm^2 ; $T_{max} = 2000$ N-m ; $T_{min} = -800$ N-m

We know that the mean or average torque,

$$T_m = \frac{T_{max} + T_{min}}{2} = \frac{2000 + (-800)}{2} = 600 \text{ N-m} = 600 \times 10^3 \text{ N-mm}$$

∴ Mean or average shear stress,

$$\tau_m = \frac{16\,T_m}{\pi d^3} = \frac{16 \times 600 \times 10^3}{\pi (50)^3} = 24.4 \text{ N/mm}^2 \quad \ldots \left(\because T = \frac{\pi}{16} \times \tau \times d^3\right)$$

Variable torque,

$$T_a = \frac{T_{max} - T_{min}}{2} = \frac{2000 - (-800)}{2} = 1400 \text{ N-m} = 1400 \times 10^3 \text{ N-mm}$$

∴ Variable shear stress, $\tau_a = \dfrac{16\,T_a}{\pi d^3} = \dfrac{16 \times 1400 \times 10^3}{\pi (50)^3} = 57 \text{ N/mm}^2$

Since the endurance limit in reversed bending (σ_e) is taken as one-half the ultimate tensile strength (*i.e.* $\sigma_e = 0.5\,\sigma_u$) and the endurance limit in shear (τ_e) is taken as $0.55\,\sigma_e$, therefore

$$\tau_e = 0.55\,\sigma_e = 0.55 \times 0.5\,\sigma_u = 0.275\,\sigma_u$$
$$= 0.275 \times 630 = 173.25 \text{ N/mm}^2$$

Assume the yield stress (σ_y) for carbon steel in reversed bending as 510 N/mm^2, surface finish factor (K_{sur}) as 0.87, size factor (K_{sz}) as 0.85 and fatigue stress concentration factor (K_{fs}) as 1.

Since the yield stress in shear (τ_y) for shear loading is taken as one-half the yield stress in reversed bending (σ_y), therefore

$$\tau_y = 0.5\,\sigma_y = 0.5 \times 510 = 255 \text{ N/mm}^2$$

Let $F.S.$ = Factor of safety.

We know that according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\tau_m}{\tau_y} + \frac{\tau_v \times K_{fs}}{\tau_e \times K_{sur} \times K_{sz}} = \frac{24.4}{255} + \frac{57 \times 1}{173.25 \times 0.87 \times 0.85}$$
$$= 0.096 + 0.445 = 0.541$$

∴ $F.S. = 1 / 0.541 = 1.85$ **Ans.**

A cantilever beam made of cold drawn carbon steel of circular cross-section as shown in Fig. 6.18, is subjected to a load which varies from – F to 3 F. Determine the maximum load that this member can withstand for an indefinite life using a factor of safety as 2. The theoretical stress concentration factor is 1.42 and the notch sensitivity is 0.9. Assume the following values:

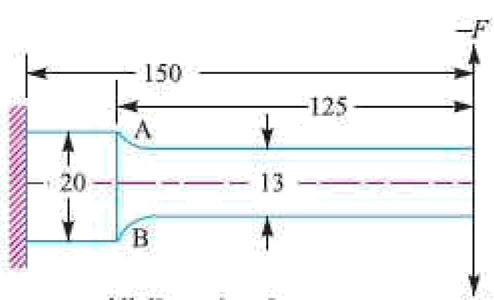
Ultimate stress = 550 MPa

Yield stress = 470 MPa

Endurance limit = 275 MPa

Size factor = 0.85

Surface finish factor= 0.89



All dimensions in mm.

Solution. Given : $W_{min} = -F$; $W_{max} = 3F$; $F.S. = 2$; $K_t = 1.42$; $q = 0.9$; $\sigma_u = 550$ MPa $= 550$ N/mm^2 ; $\sigma_y = 470$ MPa $= 470$ N/mm^2 ; $\sigma_{-1} = 275$ MPa $= 275$ N/mm^2 ; $K_{sz} = 0.85$; $K_{sur} = 0.89$

The beam as shown in Fig. is subjected to a reversed bending load only. Since the point A at the change of cross section is critical, therefore we shall find the bending moment at point A.

We know that maximum bending moment at point A,

$$M_{max} = W_{max} \times 125 = 3F \times 125 = 375\,F \text{ N-mm}$$

and minimum bending moment at point A,

$$M_{min} = W_{min} \times 125 = -F \times 125 = -125\,F \text{ N-mm}$$

$\therefore$ Mean or average bending moment,

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{375\,F + (-125\,F)}{2} = 125\,F \text{ N-mm}$$

and variable bending moment,

$$M_a = \frac{M_{max} - M_{min}}{2} = \frac{375\,F - (-125\,F)}{2} = 250\,F \text{ N-mm}$$

Section modulus, $$Z = \frac{\pi}{32} \times d^3 = \frac{\pi}{32}(13)^3 = 215.7 \text{ mm}^3 \quad ...(\because d = 13 \text{ mm})$$

$\therefore$ Mean bending stress, $$\sigma_m = \frac{M_m}{Z} = \frac{125\,F}{215.7} = 0.58\,F \text{ N/mm}^2$$

and variable bending stress, $$\sigma_a = \frac{M_a}{Z} = \frac{250\,F}{215.7} = 1.16\,F \text{ N/mm}^2$$

Fatigue stress concentration factor, $K_f = 1 + q\,(K_t - 1) = 1 + 0.9\,(1.42 - 1) = 1.378$

We know that according to Goodman's formula

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a \times K_f}{\sigma_{-1} \times K_{sur} \times K_{sz}}$$

$$\frac{1}{2} = \frac{0.58\,F}{550} + \frac{1.16F \times 1.378}{275 \times 0.89 \times 0.85}$$

$$= 0.001\,05\,F + 0.007\,68\,F = 0.008\,73\,F$$

$$\therefore \quad F = \frac{1}{2 \times 0.008\,73} = 57.3 \text{ N}$$

and according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a \times K_f}{\sigma_{-1} \times K_{sur} \times K_{sz}}$$

$$\frac{1}{2} = \frac{0.58\,F}{470} + \frac{1.16F \times 1.378}{275 \times 0.89 \times 0.85}$$

$$= 0.001\,23\,F + 0.007\,68\,F = 0.008\,91\,F$$

$$\therefore \quad F = \frac{1}{2 \times 0.008\,91} = 56 \text{ N}$$

Taking larger of the two values, we have $F = 57.3$ N **Ans.**

Shaft [Power Transmission]

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

Shaft Subjected to Twisting Moment Only.

$$T = \frac{\pi}{16}\tau d^3 \rightarrow \text{Solid shaft} \qquad T = \frac{\pi}{16}\tau\left[\frac{D_o^4 - D_i^4}{D_o}\right] \rightarrow \text{Hollow shaft}$$

Shaft Subjected to Bending Moment Only.

$$M_e = \frac{\pi}{32} \times \sigma_b \times d^3 \rightarrow \text{Pag:No: 7.24}$$

Shaft Subjected to Twisting and Bending Moment.

$$T_e = \sqrt{M^2 + T^2} \quad \& \quad T_e = \frac{\pi}{16}\tau d^3 \rightarrow \text{Twisiting}$$

+

Shaft Subjected to Bending

$$M_e = \frac{1}{2}[M + T_e] \quad \& \quad M_e = \frac{\pi}{32}\sigma_b d^3 \rightarrow \text{Bending}$$

Shafts Subjected to Fluctuating Loads (or Gradually applied Load.

$$T_e = \sqrt{(K_b \times M)^2 + (K_t \times T)^2} \quad \& \quad T_e = \frac{\pi}{16}\tau d^3$$

$$M_e = \frac{1}{2}\left[(K_b \times M) + T_e\right] \quad \& \quad M_e = \frac{\pi}{32}\sigma_b d^3$$

Belt drive:

* Torque

$$T = (T_1 - T_2)R$$

$$2.3\log\left(\frac{T_1}{T_2}\right) = \mu\theta$$

Gear Drive

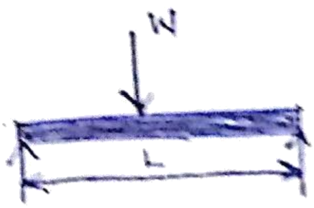
* Torque

$$T = F \times R$$

$$m \text{ (module)} = \frac{D}{T} = \frac{\text{Diameter (gear)}}{\text{No. of Teeth}}$$

Bending Moment for Beam.

$M = W \times L$



$M = \frac{W \times L}{4}$

Find the diameter of a solid steel shaft to transmit 20 kW at 200 r.p.m. The ultimate shear stress for the steel may be taken as 360 MPa and a factor of safety as 8.

If a hollow shaft is to be used in place of the solid shaft, find the inside and outside diameter when the ratio of inside to outside diameters is 0.5.

Given: $P = 20\,kW = 20 \times 10^3\,W$; $N = 200$ rpm ; $\tau_u = 360$ MPa or N/mm^2

$F.O.S = 8$; $\frac{D_i}{D_o} = 0.5 \Rightarrow D_i = 0.5 D_o$

W.K.T, the allowable shear stress

$$\tau = \frac{\tau_u}{F.O.S} = \frac{360}{8} = 45\,N/mm^2$$

Diameter of the solid shaft:

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

$$T = 955\,N.m$$

$$T = 955 \times 10^3\,N.mm.$$

W.K.T torque transmitted by solid shaft → Pag: No: 7.23

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

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

$$d^3 = \frac{955 \times 10^3}{8.84} \qquad d = 47.6 \text{ Say } 50\,mm.$$

Diameter of hollow shaft

W.K.T the torque transmitted by the hollow shaft → Pag: No: 7.21

$$T = \frac{\pi}{16} \tau \left[\frac{D_o^4 - D_i^4}{D_o}\right]$$

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

$$955 \times 10^3 = \frac{\pi}{16} \times 45 \times \frac{D_o^{4\,3}}{D_o} \left[1 - (0.5)^4\right]$$

$$D_o^3 = \frac{955 \times 10^3}{8.3} \; ; \quad D_o = 48.6 \text{ Say } 50\,mm$$

$$D_i = 0.5 D_o$$

$$= 0.5 \times 50$$

$$D_i = 25\,mm$$

A Solid Circular shaft is subjected to a bending moment of 3000 N·m and a torque of 10 000 N·m. The shaft is made of 45C8 steel having ultimate tensile stress of 700 MPa and a ultimate shear stress of 500 MPa. Assuming a factor of safety as 6, determine the diameter of the shaft.

Given : $M = 3000$ N·m $= 3000 \times 10^3$ N·mm ; $T = 10{,}000$ N·m $= 10{,}000 \times 10^3$ N mm

$\sigma_{tu} = 700$ MPa or N/mm² ; $\tau_u = 500$ MPa or N/mm².

W.K.T the allowable tensile stress,

$$\sigma_t \text{ or } \sigma_b = \frac{\sigma_{tu}}{F.O.S} = \frac{700}{6} = 116.7 \text{ N/mm}^2.$$

the allowable shear stress

$$\tau = \frac{\tau_u}{F.O.S} = \frac{500}{6} = 83.3 \text{ N/mm}^2.$$

Equivalent twisting Moment,

$$T_e = \sqrt{M^2+T^2} = \sqrt{(3 \times 10^6)^2 + (10 \times 10^6)^2} = 10.44 \times 10^6 \text{ N·mm}$$

We also know that

$$T_e = \frac{\pi}{16} \tau d^3 \quad \rightarrow \quad \text{Pag : No : 7.23}$$

$$10.44 \times 10^6 = \frac{\pi}{16} \times 83.3 \times d^3$$

$$d = 86 \text{ mm.}$$

According to equivalent bending moment,

$$M_e = \frac{1}{2}(M + T_e)$$

$$= \frac{1}{2}\left[3 \times 10^6 + 10.44 \times 10^6\right]$$

$$= 6.72 \times 10^6 \text{ N·mm}$$

We also know that the equivalent bending Moment

$$M_e = \frac{\pi}{32} \times \sigma_b \times d^3 \quad \rightarrow \quad \text{Pag : No : 7.24}$$

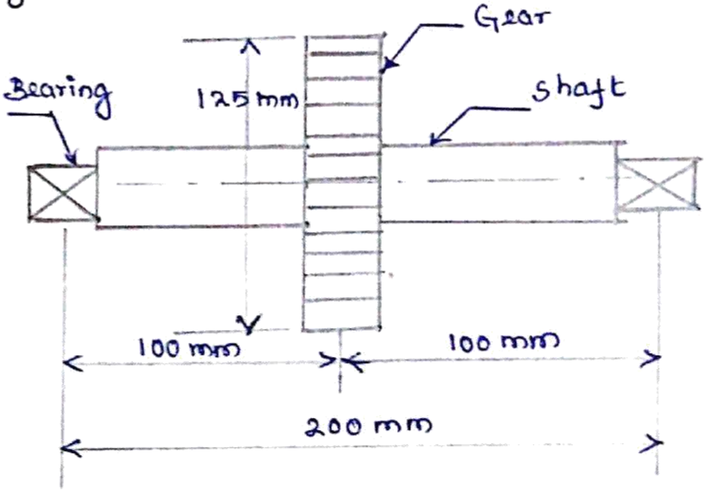
$$6.72 \times 10^6 = \frac{\pi}{32} \times 116.7 \times d^3$$

$$d = 83.7 \text{ mm}$$

∴ Take the larger of the two values, We have

$d = 86$ say 90 mm.

A shaft supported at the ends in ball bearings carries a straight tooth spur gear at its mid span and is to transmit 7.5 KW at 300 rpm. The pitch circle diameter of the gear is 150 mm. The distances between the centre line of bearings and gear are 100 mm each. If the shaft is made of steel and the allowable shear stress is 45 MPa, determine the diameter of the shaft. Show in a sketch how the gear will be mounted on the shaft; also indicate the ends where the bearings will be mounted? The pressure angle of the gear may taken as 20°



Given: $P = 7.5\ KW = 7.5 \times 10^3\ W$

$N = 300$ rpm

$D = 150$ mm

$L = 200$ mm

$\tau = 45$ MPa or N/mm^2

$\alpha = 20°$

W.K.T

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

$$T = \frac{7500 \times 60}{2 \times \pi \times 300} = 238.7\ N\cdot m = 238.7 \times 10^3\ N\cdot mm$$

∴ Tangential force on Gear

$$F_t = \frac{2T}{D} = \frac{2 \times 238.7 \times 10^3}{150} = 3182.7\ N$$

Normal Load acting on the tooth of the gear

$$W = \frac{F_t}{\cos\alpha} = \frac{3182.7}{\cos 20°} = \frac{3182.7}{0.9397} = 3387\ N$$

Since the gear is mounted at middle of the shaft, ∴ Max. bending moment at centre of the gear.

$$M = \frac{W.L}{4} = \frac{3387 \times 200}{4} = 169350\ N\cdot mm$$

W.K.T equivalent twisting Moment

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(238.7 \times 10^3)^2 + (169.4 \times 10^3)^2}$$

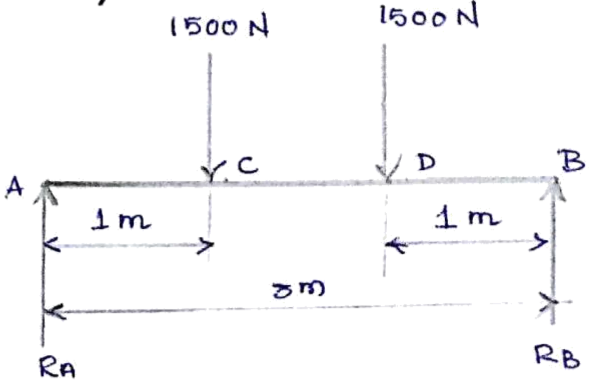
$$T_e = 292.7 \times 10^3\ N\cdot mm.$$

We also know that,

$$T_e = \frac{\pi}{16} \tau \times d^3 \Rightarrow 292.7 \times 10^3 = \frac{\pi}{16} \times 45 \times d^3$$

$d^3 = 33 \times 10^3$ $\quad d = 32$ say 35.

A shaft made of mild steel is required to transmit 100 kW at 300 rpm the supported Length of the shaft is 3 metres. It carries two Pulleys each weighing 1500 N. Supported at a distance of 1 metre from the ends respectively. Assuming the safe Value of stress, determine the diameter of the shaft.



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

$N = 300\ rpm.$

$L = 300\ mm$

$W = 1500\ N.$

W.K.T

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

$$T = \frac{100 \times 10^3 \times 60}{2 \times \pi \times 300}$$

$$T = 3183\ N.m = 3183 \times 10^3\ N.mm$$

The reaction at each support will be 1500 N

$$R_A = R_B = 1500\ N.$$

∴ Max Bending moment, Taking Moment at D

$$M = 1500 \times 1000$$
$$= 1500 \times 10^3\ N.mm$$

W.K.T equivalent twisting moment

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(1500 \times 10^3)^2 + (3183 \times 10^3)^2}$$

$$T_e = 3519 \times 10^3\ N.mm.$$

We also know that equivalent twisting moment → Pag: No: 7.23

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

$$3519 \times 10^3 = \frac{\pi}{16} \times 55 \times d^3 \qquad [\tau = 550\ kg/cm^2 = 55\ N/mm^2]$$

$$d^3 = 325.856 \times 10^3$$

$$d = 68.8 \text{ Say } 70\ mm$$

A line shaft is driven by means of a motor placed vertically below it. the pulley on the line shaft is 1.5 metre in diameter and has belt. tensions 5.4 KN and 1.8 KN on the tight side and slack side of the belt respectively. Both these tensions may be assumed to be vertical. If the pulley be overhang from the shaft, the distance of the centre line of the pulley from the centre line of the bearing being 400 mm, find the diameter of the shaft. Assuming max. allowable shear stress of 42 MPa.

Given: $D = 1.5\,m = 1.5 \times 10^3\,mm$

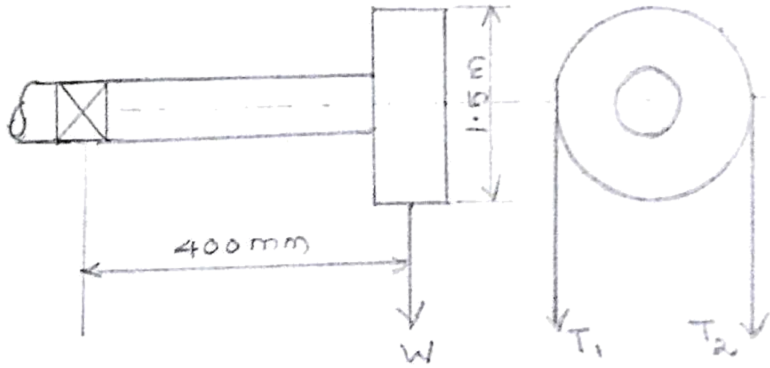
$R = 750\,mm$

$T_1 = 5.4\,KN = 5.4 \times 10^3\,N$

$T_2 = 1.8\,KN = 1.8 \times 10^3\,N$

$L = 400\,mm$

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



W.K.T torque transmitted by the shaft

$$T = (T_1 - T_2) R$$

$$= (5400 - 1800) \times 750$$

$$T = 2700 \times 10^3\ N\cdot mm$$

Neglecting the weight of shaft, Total vertical Load acting on the pulley.

$$W = T_1 + T_2 = 5400 + 1800$$

$$W = 7200\ N.$$

$\therefore$ Bending Moment, $M = W \times L$

$$= 7200 \times 400$$

$$= 2880 \times 10^3\ N\cdot mm$$

W.K.T equivalent Twisting moment

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(2880 \times 10^3)^2 + (2700 \times 10^3)^2}$$

$$T_e = 3950 \times 10^3\ N\cdot mm$$

We also know that

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

$$3950 \times 10^3 = \frac{\pi}{16} \times 42 \times d^3$$

$$d^3 = 479 \times 10^3$$

$$d = 78 \text{ say } 80\ mm.$$

Shaft Subjected to Fluctuating Loads.

A mild steel shaft transmits 23 KW at 200 rpm. It Carries a Central Load of 900 N and is simply supported between the bearing 2.5 metres apart. Determine the size of the shaft, if the allowable shear stress is 42 MPa and the maximum Tensile or compressive stress is not to exceeds 56 MPa. What is size of the shaft will be required, if it is subjected to Gradually applied Size.

Given: $P = 23\ KW = 23 \times 10^3\ W$; $N = 200\ rpm$

$W = 900\ N$; $L = 2.5\ m$; $\tau = 42\ MPa$

$\sigma_b = 56\ MPa$ or N/mm^2.

W.K.T

$$P = \frac{2\pi N T}{60} \Rightarrow T = \frac{P \times 60}{2\pi N} = \frac{23 \times 10^3 \times 60}{2 \times \pi \times 200} = 1098.16\ N.m$$

$$T = 1098.16 \times 10^3\ N.mm$$

W.K.T for Simply Support beam subjected to ~~Point~~ Central Load the Bending moment is $M = \frac{W \times L}{4}$

$$M = \frac{900 \times 2500}{4} = 562.5 \times 10^3\ N.mm.$$

For steady Load [stationary shafts]

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(562.5 \times 10^3)^2 + (1098.16 \times 10^3)^2}$$

$$T_e = 1233.69 \times 10^3\ N.mm.$$

We also know that $T_e = \frac{\pi}{16} \tau d^3$ → Twisting Pag:No: 7.23

$$1233.69 \times 10^3 = \frac{\pi}{16} \times 42 \times d^3$$

$$d = 53.07\ mm$$

W.K.T for Bending $M_e = \frac{1}{2}[M + T_e] = \frac{1}{2}[562.5 \times 10^3 + 1233.69 \times 10^3]$

$$M_e = 898.05 \times 10^3\ N.mm.$$

We also know that $M_e = \frac{\pi}{32} \times \sigma_b \times d^3$ → Bending Pag:No: 7.24

$$898.05 \times 10^3 = \frac{\pi}{32} \times 56 \times d^3$$

$$d = 54.66 \text{ say } 55\ mm$$

Taking Largest of two diameter $d = 55\ mm$

For Gradually applied [Rotating shaft]

$K_b = 1.5 \quad K_t = 1$

∴ Equivalent twisting Moment,

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

$$= \sqrt{(1.5 \times 562.5 \times 10^3)^2 + (1 \times 1098.16 \times 10^3)^2}$$

$$T_e = 1384.74 \times 10^3 \text{ N.mm}.$$

We also know that

$$T_e = \frac{\pi}{16} \tau d^3$$ → Pag: No: 7.23

$$1384.74 \times 10^3 = \frac{\pi}{16} \times 42 \times d^3$$

$$d = 55.16 \text{ mm}.$$

For Bending Moment

$$M_e = \frac{1}{2}\left[K_b \times M + T_e\right]$$

$$= \frac{1}{2}\left[(1.5 \times 562.5 \times 10^3) + 1384.74 \times 10^3\right]$$

$$= 1114.24 \times 10^3 \text{ N.mm}.$$

We also know that

$$M_e = \frac{\pi}{32} \sigma_b d^3$$ → Pag: No: 7.24

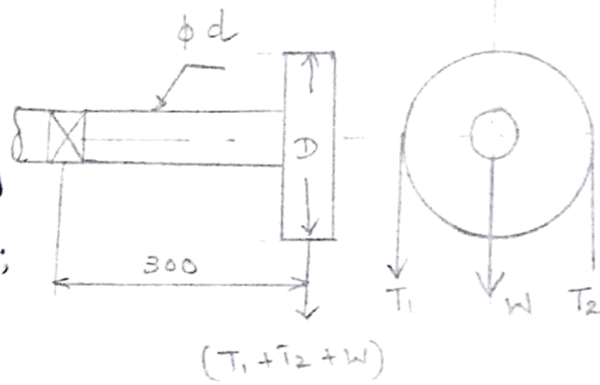
$$1114.24 \times 10^3 = \frac{\pi}{32} \times 56 \times d^3$$

$$d = 58.73 \text{ Say } 59 \text{ mm}.$$

Taking the largest of two diameter

$$d = 59 \text{ mm}.$$

Design a shaft to transmit power from an electric motor to a lathe head stock through a pulley by means of a belt drive. The pulley weighs 200 N and is Located at 300 mm from the Centre of the bearings. The diameter of the pulley is 200 mm and maximum power transmitted is 1 kW at 120 rpm. The angle of Lap of the belt is 180° and co-efficient of friction b/w belt + the Pulley is 0.3. The shock and fatigue factors for bending + twisting are 1.5 and 2.0 respectively. The allowable Shear stress in the shaft may be taken as 35 MPa.



Given : $W = 200 N$; $L = 300 mm$; $D = 200 mm$;

$R = 100 mm$; $P = 1 kW = 1 \times 10^3 W$;

$N = 120 rpm$; $\theta = 180°$; $\theta = \frac{\pi}{180} \times 180$ rad

$\theta = \pi$ rad; $\mu = 0.3$; $K_b = 1.5$; $K_t = 2$;

$\tau = 35$ MPa or N/mm^2

W.K.T

$$P = \frac{2\pi N T}{60} \Rightarrow T = \frac{P \times 60}{2\pi N} = \frac{1000 \times 60}{2 \times \pi \times 120} = 79.6 \text{ N·m} = 79.6 \times 10^3 \text{ N·mm}$$

T_1 & T_2 is Tension in Tight & Slack side of the belt

For belt
$$T = (T_1 - T_2) \times R$$
$$79.6 \times 10^3 = (T_1 - T_2) \times 100$$
$$T_1 - T_2 = 796 N \quad \text{— (i)}$$

W.K.T

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu\theta = 0.3 \times \pi$$

$$\log\left(\frac{T_1}{T_2}\right) = \frac{0.9426}{2.3} \Rightarrow \frac{T_1}{T_2} = 2.57 \quad \text{— (ii)}$$

From equation (i) + (ii)

$$T_1 = 1303 N \ \& \ T_2 = 507 N$$

Total Load

$$W_T = T_1 + T_2 + W$$
$$= 1303 + 507 + 200$$
$$W = 2010 N$$

Bending Moment acting on the shaft

$$M = W_T \times L$$

$$= 2010 \times 300$$

$$= 603 \times 10^3 \text{ N.mm}$$

∴ Equivalent twisting moment

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

$$= \sqrt{(1.5 \times 603 \times 10^3)^2 + (2 \times 79.6 \times 10^3)^2}$$

$$= 918 \times 10^3 \text{ N.mm.}$$

We also know that

$$T_e \text{ or } M_t = \frac{\pi}{16} \times \tau \times d^3$$ → Pag: No: 7.23

$$918 \times 10^3 = \frac{\pi}{16} \times 35 \times d^3$$

$$d = 51.1 \text{ mm say } 55 \text{ mm.}$$

A steel solid shaft transmitting 15 kW at 200 rpm. is supported on two bearings 750 mm apart and has two gears keyed to it. The pinion having 30 teeth of 5 mm module is located 100 mm to the left of the right hand bearing and delivers power vertically to the right. The gear having 100 teeth of 5 mm module is located 150 mm to the right of the Left hand bearing & receives power in a vertical direction from below. using an allowable stress of 54 MPa in shear, determine the diameter of the shaft.

Given: $P = 15\,kW = 15 \times 10^3\,W$;

$N = 200\,rpm$;

$AB = 750\,mm$;

$T_D = 30$;

$m_D = 5\,mm$;

$BD = 100\,mm$;

$T_C = 100$;

$m_C = 5\,mm$;

$AC = 150\,mm$;

$\tau = 54\,MPa$ or N/mm^2

W.K.T

$$P = \frac{2\pi NT}{60} \Rightarrow T = \frac{P \times 60}{2\pi N} = \frac{15 \times 10^3 \times 60}{2 \times \pi \times 200} = 716\,N \cdot m$$

$$T = 716 \times 10^3\,N \cdot m.$$

W.K.T Diameter of the Gear = No. of teeth × module

$D = T \times m$ → Pag: No: 8.22

Diameter of Gear C $D = 100 \times 5 = 500\,mm$.

Diameter of Pinion D $D = 30 \times 5 = 150\,mm$.

We also know that Torque for Gear

$T = F_C \times$ Radius of Gear

$$T = F_C \times \left(\frac{D_C}{2}\right) \rightarrow \text{Pag:No: 8.57}$$

$$\frac{716 \times 10^3 \times 2}{500} = F_C$$

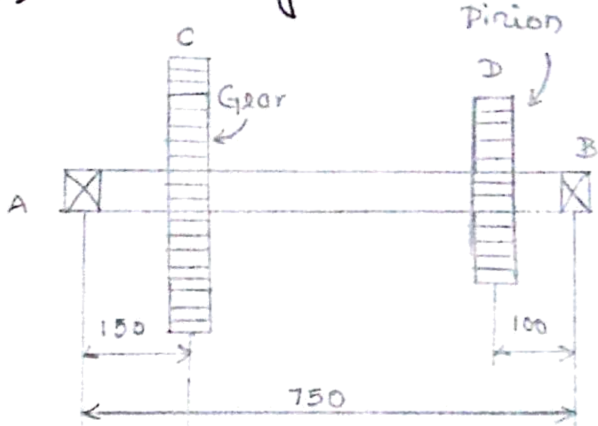
$F_C = 2864\,N$ → Vertically

III[ly] for Pinion

$$T = F_D \times R, \text{ or } \frac{D_D}{2}$$

$$716 \times 10^3 = F_D \times \frac{150}{2}$$

$$F_D = 9547 \text{ N.} \rightarrow \text{Horizontally.}$$



Make a ~~Vert~~ Horizontal to Vertical Load

$$R_A + R_B = 2864 + 9547$$

$$R_A + R_B = 12411$$

W.K.T

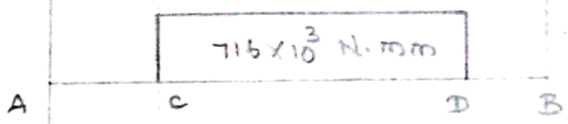
$$M_A = 0$$

$$M_B = 0$$

Taking Moment at A

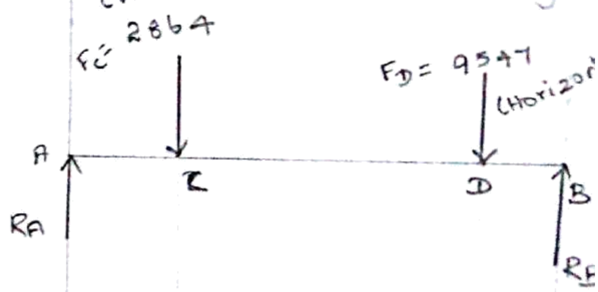
$$R_B \times 750 - 9547 \times 650 - 2864 \times 150 = 0$$

$$R_B = 8846.8 \text{ N}$$



$$R_A = 12411 - 8846.8$$

$$R_A = 3564 \text{ N}$$

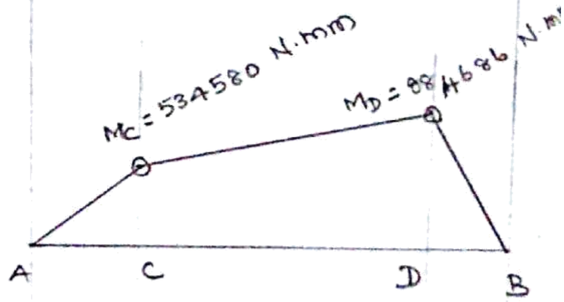


Now Find Max Bending Moment

$$M_D = 8846.8 \times 100 = 884686 \text{ N.mm}$$

$$M_C = 8846.8 \times 600 - 9547 \times 500$$

$$M_C = 534580 \text{ N.mm}$$



W.K.T the equivalent twisting Moment

$$T_e = \sqrt{M^2 + T^2}$$

use Max. Bending Moment

$$T_e = \sqrt{(884686)^2 + (716 \times 10^3)^2}$$

$$= 1138.12 \times 10^3 \text{ N.mm}$$

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

$$1138.12 \times 10^3 = \frac{\pi}{16} \times 54 \times d^3$$

$$\frac{1138.12 \times 10^3}{10} = d^3$$

$$d = 48 \text{ Say } 50 \text{ mm}$$

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

Given: $AB = 1\,m = 1000\,mm$; $D_C = 600\,mm$ or $R_C = 300\,mm$; $AC = 300\,mm$

$T_1 = 2.2\,kN = 2250\,N$; $D_D = 400\,mm$ or $R_D = 200\,mm$; $BD = 200\,mm$

$\theta = 180° = 180 \times \frac{\pi}{180}$ rad; $\mu = 0.24$; $\sigma_b = 63\,MPa$ or N/mm^2; $\tau = 42\,MPa$

Load on pulley C:

T_1 – Tight side $= 2250\,N$

T_2 – Slack side

W.K.T $\frac{T_1}{T_2} = e^{\mu\theta}$ $\quad \frac{T_1}{T_2} = e^{(0.24 \times \pi)}$ $\quad \frac{2250}{T_2} = e^{(0.24 \times \pi)}$

$T_2 = 1058\,N.$

Total Vertical Load on Pulley C is

$W_C = T_1 + T_2 = 2250 + 1058 = 3308\,N$

Total Horizontal Load on pulley C is zero.

W.K.T Torque acting on pulley C,

$T = (T_1 - T_2) R_C = (2250 - 1058) \times 300$

$T = 357600\,N.mm$

Total Horizontal Load on pulley D

$\frac{T_1}{T_2} = \frac{T_3}{T_4} = e^{\mu\theta}$ $\quad \frac{T_3}{T_4} = 2.125$

$T_3 = 2.125\,T_4$ ——— ①

W.K.T Torque for Both Pulley is equal

$(T_3 - T_4) \times R_D = 357600\,N.mm$

$T_3 - T_4 = \frac{357600}{200} = 1788$ ——— ②

Subt ① in ② we get

$2.125\,T_4 - T_4 = 1788$

$T_4 = 1589\,N$ $\quad T_3 = 2.125 \times 1589$

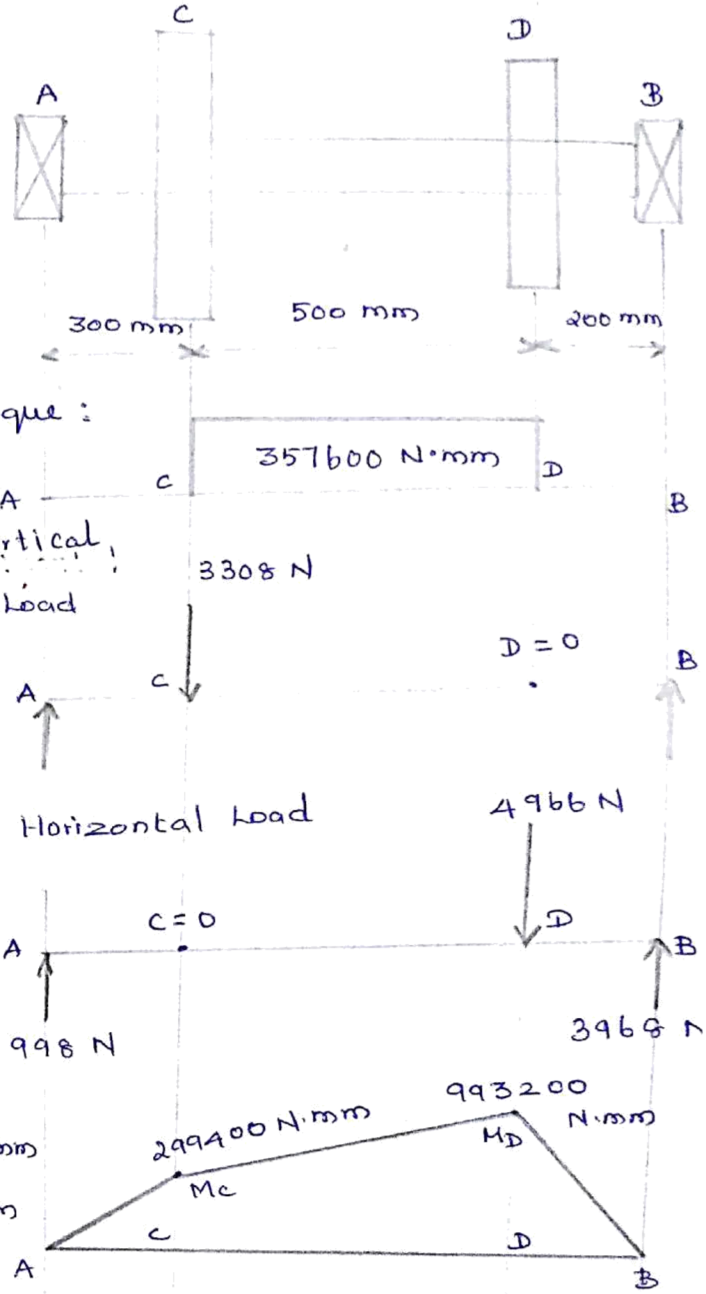
$T_3 = 3377\,N$

Total Horizontal Load on D

$W_D = T_3 + T_4 = 3377 + 1589$

$W_D = 4966\ N.$

Horizontal Load on C is Zero



Note :

Compare to Vertical Load, the horizontal Load is Maximum.

So, Go for horizontal Load

To find $R_A + R_B$

W.K.T $M_A = 0 + M_B = 0$

Taking Moment at A

$R_B \times 1000 - 4960 \times 800 = 0$

$1000\ R_B = 4960 \times 800$

$R_B = 3968\ N$

$R_A + R_B = 4966\ N$

$R_A = 4966 - 3968$

$= 998\ N$

To find Max. Bending Moment

$M_D = 4966 \times 200 = 993200\ N \cdot mm$

$M_C = 998 \times 300 = 299400\ N \cdot mm$

To find diameter – Torque.

$T_e = \sqrt{M^2 + T^2}$

$\frac{\pi}{16} \times 42 \times d^3 = \sqrt{(993200)^2 + (357600)^2}$

$\frac{\pi}{16} \times 42 \times d^3 = 1055615.46$

$d = 50\ mm.$

To find bending Moment.

$M_e = \frac{1}{2}(M + T_e)$

$= \frac{1}{2}(993200 + 1055615)$

$M_e = 1024407.5$

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

$\frac{\pi}{32} \times 63 \times d^3 = 1024407.5$

$d = 54.9 \simeq 55$

Unit-V

Amachinecomponentissubjectedtoaflexuralstresswhichfluctuatesbetween+300 MN/m2and – 150MN/m2.Determinethevalueofminimum ultimatestrength accordingtoGoodmanrelation;andSoderbergrelation.Takeyieldstrength=0.55 Ultimatestrength;Endurancestrength=0.5Ultimatestrength;andfactorofsafety=2.

Solution.

Given: $\sigma_1=300 MN/m^2$; $\sigma_2=-150 MN/m^2$; $\sigma_y=0.55\sigma_u$; $\sigma_{-1}=0.5\sigma_u$; $F.S.=2$

Let σ_u = Minimum ultimate strength in MN/m^2.

We know that the mean or average stress,

$$\sigma_m = \frac{\sigma_1+\sigma_2}{2} = \frac{300+(-150)}{2} = 75 \text{ MN/m}^2$$

and variable stress,

$$\sigma_a = \frac{\sigma_1-\sigma_2}{2} = \frac{300-(-150)}{2} = 225 \text{ MN/m}^2$$

According to Goodman relation

We know that according to modified Goodman relation,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma_{-1}}$$

or

$$\frac{1}{2} = \frac{75}{\sigma_u} + \frac{225}{0.5\,\sigma_u} = \frac{525}{\sigma_u}$$

$$\therefore \quad \sigma_u = 2 \times 525 = 1050 \text{ MN/m}^2 \quad \textbf{Ans.}$$

According to Soderberg relation

We know that according to Soderberg relation,

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_{-1}}$$

or

$$\frac{1}{2} = \frac{75}{0.55\,\sigma_u} + \frac{255}{0.5\,\sigma_u} = \frac{586.36}{\sigma_u}$$

$$\therefore \quad \sigma_u = 2 \times 586.36 = 1172.72 \text{ MN/m}^2 \quad \textbf{Ans.}$$

Abarofcircularcross-sectionissubjectedtoalternatingtensileforcesvaryingfroma minimumof200kNtoamaximumof500kN.Itistobemanufacturedofamaterial withanultimatetensilestrengthof900MPaandanendurancelimitof700MPa. Determinethediameterofbarusingsafetyfactorsof3.5relatedtoultimatetensile strengthand4relatedtoendurancelimitandastressconcentrationfactorof1.65for fatigueload.UseGoodmanstraightlineasbasisfordesign.

Solution. Given : $W_{min} = 200$ kN ; $W_{max} = 500$ kN ; $\sigma_u = 900$ MPa $= 900$ N/mm^2 ; $(F.S.)_u = 3.5$; $(F.S.)_e = 4$; $K_f = 1.65$; $\sigma_{-1} = 700$ MPa $= 700$ N/mm^2 ;

Let d = Diameter of bar in mm

$$\therefore \quad \text{Area, } A = \frac{\pi}{4} \times d^2 = 0.7854\, d^2 \text{ mm}^2$$

We know that mean or average force,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{500 + 200}{2} = 350 \text{ kN} = 350 \times 10^3 \text{ N}$$

$\therefore$ Mean stress, $\sigma_m = \frac{W_m}{A} = \frac{350 \times 10^3}{0.7854\, d^2} = \frac{446 \times 10^3}{d^2} \text{ N/mm}^2$

Variable force, $W_a = \frac{W_{max} - W_{min}}{2} = \frac{500 - 200}{2} = 150 \text{ kN} = 150 \times 10^3 \text{ N}$

$\therefore$ Variable stress, $\sigma_a = \frac{W_a}{A} = \frac{150 \times 10^3}{0.7854\, d^2} = \frac{191 \times 10^3}{d^2} \text{ N/mm}^2$

We know that according to Goodman's formula,

$$\frac{\sigma_a}{\sigma_{-1} / (F.S.)_e} = 1 - \frac{\sigma_m . K_f}{\sigma_u / (F.S.)_u}$$

$$\frac{\frac{191 \times 10^3}{d^2}}{700/4} = 1 - \frac{\frac{446 \times 10^3}{d^2} \times 1.65}{900/3.5}$$

$$\frac{1100}{d^2} = 1 - \frac{2860}{d^2} \quad \text{or} \quad \frac{1100 + 2860}{d^2} = 1$$

$d^2 = 3960$ or $d = 62.9$ say 63 mm **Ans.**

Determinethethicknessofa120mmwideuniformplateforsafecontinuousoperation iftheplateistobesubjectedtoatensileloadthathasamaximumvalueof250kNand aminimum valueof100kN.Thepropertiesoftheplatematerialareasfolows: Endurancelimitstress=225MPa,andYieldpointstress=300MPa.Thefactorof safetybasedonyieldpointmayb takenas1.5.

Solution. Given : $b = 120$ mm ; $W_{max} = 250$ kN; $W_{min} = 100$ kN ; $\sigma_{-1} = 225$ MPa = 225 N/mm² ; $\sigma_y = 300$ MPa = 300 N/mm²; $F.S. = 1.5$

Let t = Thickness of the plate in mm.

$\therefore$ Area, $A = b \times t = 120\,t$ mm²

We know that mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{250 + 100}{2} = 175 \text{ kN} = 175 \times 10^3 \text{ N}$$

$\therefore$ Mean stress, $\sigma_m = \dfrac{W_m}{A} = \dfrac{175 \times 10^3}{120t}$ N/mm²

Variable load, $W_a = \dfrac{W_{max} - W_{min}}{2} = \dfrac{250 - 100}{2} = 75 \text{ kN} = 75 \times 10^3$ N

$\therefore$ Variable stress, $\sigma_a = \dfrac{W_a}{A} = \dfrac{75 \times 10^3}{120t}$ N/mm²

According to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_{-1}}$$

$$\frac{1}{1.5} = \frac{175 \times 10^3}{120t \times 300} + \frac{75 \times 10^3}{120t \times 225} = \frac{4.86}{t} + \frac{2.78}{t} = \frac{7.64}{t}$$

$\therefore$ $t = 7.64 \times 1.5 = 11.46$ say 11.5 mm **Ans.**

Determinethediameterofacircularrodmadeofductilematerialwithafatiguestrength (completestressreversal),σ_{-1}=265MPaandatensileyieldstrengthof350MPa.The memberissubjectedtoavaryingaxialloadfromW_{min}=–300×10^3NtoW_{max}=700× 10^3Nandhasastressconcentrationfator=1.8.Usefactorofsafetyas2.0.

Solution. Given : $\sigma_{-1} = 265$ MPa = 265 N/mm² ; $\sigma_y = 350$ MPa = 350 N/mm² ; $W_{min} = -300 \times 10^3$ N ; $W_{max} = 700 \times 10^3$ N ; $K_f = 1.8$; $F.S. = 2$

Let d = Diameter of the circular rod in mm.

$\therefore$ Area, $A = \dfrac{\pi}{4} \times d^2 = 0.7854\, d^2$ mm²

We know that the mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{700 \times 10^3 + (-300 \times 10^3)}{2} = 200 \times 10^3 \text{ N}$$

$\therefore$ Mean stress, $\sigma_m = \dfrac{W_m}{A} = \dfrac{200 \times 10^3}{0.7854\, d^2} = \dfrac{254.6 \times 10^3}{d^2}$ N/mm²

Variable load, $W_a = \frac{W_{max} - W_{min}}{2} = \frac{700 \times 10^3 - (-300 \times 10^3)}{2} = 500 \times 10^3$ N

$\therefore$ Variable stress, $\sigma_a = \frac{W_a}{A} = \frac{500 \times 10^3}{0.7854\, d^2} = \frac{636.5 \times 10^3}{d^2}$ N/mm²

We know that according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a \times K_f}{\sigma_{-1}}$$

$$\frac{1}{2} = \frac{254.6 \times 10^3}{d^2 \times 350} + \frac{636.5 \times 10^3 \times 1.8}{d^2 \times 265} = \frac{727}{d^2} + \frac{4323}{d^2} = \frac{5050}{d^2}$$

$\therefore$ $d^2 = 5050 \times 2 = 10\,100$ or $d = 100.5$ mm **Ans.**

Acircularbarof500mmlengthissupportedfreelyatitstwoends.Itisacteduponbya centralconcentratedcyclicloadhavingaminimum valueof20kNandamaximum valueof50kN.Determinethediameterofbarbytakingafactorofsafetyof1.5,size effectof0.85,surfacefinishfactorof0.9.Thematerialpropertiesofbararegivenby: ultimatestrengthof650MPa,yieldstrengthof500MPaandendurancestrengthof350 MPa.

Solution. Given : l = 500 mm ; W_{min} = 20 kN = 20 × 10^3 N ; W_{max} = 50 kN = 50 × 10^3 N ; $F.S.$ = 1.5 ; K_{sz} = 0.85 ; K_{sur} = 0.9 ; σ_u = 650 MPa = 650 N/mm² ; σ_y = 500 MPa = 500 N/mm² ; σ_{-1} = 350 MPa = 350 N/mm²

Let d = Diameter of the bar in mm.

We know that the maximum bending moment,

$$M_{max} = \frac{W_{max} \times l}{4} = \frac{50 \times 10^3 \times 500}{4} = 6250 \times 10^3 \text{ N-mm}$$

and minimum bending moment,

$$M_{min} = \frac{W_{min} \times l}{4} = \frac{20 \times 10^3 \times 500}{4} = 2550 \times 10^3 \text{ N-mm}$$

$\therefore$ Mean or average bending moment,

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{6250 \times 10^3 + 2500 \times 10^3}{2} = 4375 \times 10^3 \text{ N-mm}$$

and variable bending moment,

$$M_a = \frac{M_{max} - M_{min}}{2} = \frac{6250 \times 10^3 - 2500 \times 10^3}{2} = 1875 \times 10^3 \text{ N-mm}$$

Section modulus of the bar,

$$Z = \frac{\pi}{32} \times d^3 = 0.0982\, d^3 \text{ mm}^3$$

$\therefore$ Mean or average bending stress,

$$\sigma_m = \frac{M_m}{Z} = \frac{4375 \times 10^3}{0.0982\, d^3} = \frac{44.5 \times 10^6}{d^3} \text{ N/mm}^2$$

and variable bending stress,

$$\sigma_a = \frac{M_a}{Z} = \frac{1875 \times 10^3}{0.0982\, d^3} = \frac{19.1 \times 10^6}{d^3} \text{ N/mm}^2$$

We know that according to Goodman's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a \times K_f}{\sigma_{-1} \times K_{sur} \times K_{sz}}$$

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 650} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \qquad \text{...(Taking } K_f = 1\text{)}$$

$$= \frac{68 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{139 \times 10^3}{d^3}$$

$\therefore \quad d^3 = 139 \times 10^3 \times 1.5 = 209 \times 10^3$ or $d = 59.3$ mm

and according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a \times K_f}{\sigma_{-1} \times K_{sur} \times K_{sz}}$$

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 500} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \qquad \text{...(Taking } K_f = 1\text{)}$$

$$= \frac{89 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{160 \times 10^3}{d^3}$$

$\therefore \quad d^3 = 160 \times 10^3 \times 1.5 = 240 \times 10^3$ or $d = 62.1$ mm

Taking larger of the two values, we have $d = 62.1$ mm **Ans.**

A50mmdiametershaftismadefromcarbonsteelhavingultimatetensilestrengthof 630MPa.Itissubjectedtoatorquewhichfluctuatesbetween2000N-mto–800N-m. UsingSoderbergmethod,calculatethefactorofsafety.Assumesuitablevaluesforany otherdataneeded.

Solution. Given : $d = 50$ mm ; $\sigma_u = 630$ MPa $= 630$ N/mm^2 ; $T_{max} = 2000$ N-m ; $T_{min} = -800$ N-m

We know that the mean or average torque,

$$T_m = \frac{T_{max} + T_{min}}{2} = \frac{2000 + (-800)}{2} = 600 \text{ N-m} = 600 \times 10^3 \text{ N-mm}$$

$\therefore$ Mean or average shear stress,

$$\tau_m = \frac{16\, T_m}{\pi d^3} = \frac{16 \times 600 \times 10^3}{\pi (50)^3} = 24.4 \text{ N/mm}^2 \qquad \ldots\left(\because\ T = \frac{\pi}{16} \times \tau \times d^3\right)$$

Variable torque,

$$T_a = \frac{T_{max} - T_{min}}{2} = \frac{2000 - (-800)}{2} = 1400 \text{ N-m} = 1400 \times 10^3 \text{ N-mm}$$

$\therefore$ Variable shear stress, $\tau_a = \dfrac{16\, T_a}{\pi d^3} = \dfrac{16 \times 1400 \times 10^3}{\pi (50)^3} = 57$ N/mm^2

Since the endurance limit in reversed bending (σ_e) is taken as one-half the ultimate tensile strength (*i.e.* $\sigma_e = 0.5\ \sigma_u$) and the endurance limit in shear (τ_e) is taken as $0.55\ \sigma_e$, therefore

$$\tau_e = 0.55\ \sigma_e = 0.55 \times 0.5\ \sigma_u = 0.275\ \sigma_u$$
$$= 0.275 \times 630 = 173.25\ \text{N/mm}^2$$

Assume the yield stress (σ_y) for carbon steel in reversed bending as 510 N/mm^2, surface finish factor (K_{sur}) as 0.87, size factor (K_{sz}) as 0.85 and fatigue stress concentration factor (K_{fs}) as 1.

Since the yield stress in shear (τ_y) for shear loading is taken as one-half the yield stress in reversed bending (σ_y), therefore

$$\tau_y = 0.5\ \sigma_y = 0.5 \times 510 = 255\ \text{N/mm}^2$$

Let $F.S.$ = Factor of safety.

We know that according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\tau_m}{\tau_y} + \frac{\tau_v \times K_{fs}}{\tau_e \times K_{sur} \times K_{sz}} = \frac{24.4}{255} + \frac{57 \times 1}{173.25 \times 0.87 \times 0.85}$$
$$= 0.096 + 0.445 = 0.541$$

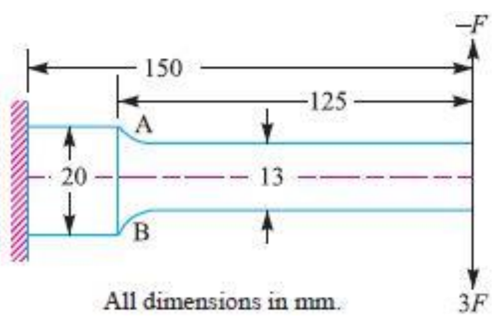
$\therefore$ $F.S. = 1 / 0.541 = 1.85$ **Ans.**

Acantileverbeammadeofcolddrawncarbonsteelofcircularcross-sectionasshown inFig.6.18,issubjectedtoaloadwhichvariesfrm–Fto3F.Determinethemaximum loadthatthismembercanwithstandforanindefinitelifeusingafactorofsafetyas2. Thetheoreticalstressconcentrationfactoris1.42andthenotchsensitivityis0.9. Assumethefolowingvalues:

Ultimatestress=550MPa

Yieldstress=470MPa

Endurance limit = 275 MPa



Sizefactor=0.85

Surfacefinishfactor=0.89

Solution. Given : $W_{min} = -F$; $W_{max} = 3\ F$; $F.S. = 2$; $K_t = 1.42$; $q = 0.9$; $\sigma_u = 550$ MPa $= 550$ N/mm^2 ; $\sigma_y = 470$ MPa $= 470$ N/mm^2 ; $\sigma_{-1} = 275$ MPa $= 275$ N/mm^2 ; $K_{sz} = 0.85$; $K_{sur} = 0.89$

The beam as shown in Fig. is subjected to a reversed bending load only. Since the point A at the change of cross section is critical, therefore we shall find the bending moment at point A.

We know that maximum bending moment at point A,

$$M_{max} = W_{max} \times 125 = 3F \times 125 = 375\,F \text{ N-mm}$$

and minimum bending moment at point A,

$$M_{min} = W_{min} \times 125 = -F \times 125 = -125\,F \text{ N-mm}$$

∴ Mean or average bending moment,

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{375\,F + (-125\,F)}{2} = 125F \text{ N-mm}$$

and variable bending moment,

$$M_a = \frac{M_{max} - M_{min}}{2} = \frac{375\,F - (-125\,F)}{2} = 250F \text{ N-mm}$$

Section modulus, $$Z = \frac{\pi}{32} \times d^3 = \frac{\pi}{32}(13)^3 = 215.7 \text{ mm}^3 \quad \ldots(\because d = 13 \text{ mm})$$

∴ Mean bending stress, $$\sigma_m = \frac{M_m}{Z} = \frac{125\,F}{215.7} = 0.58\,F \text{ N/mm}^2$$

and variable bending stress, $$\sigma_a = \frac{M_a}{Z} = \frac{250\,F}{215.7} = 1.16\,F \text{ N/mm}^2$$

Fatigue stress concentration factor, $K_f = 1 + q\,(K_t - 1) = 1 + 0.9\,(1.42 - 1) = 1.378$

We know that according to Goodman's formula

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a \times K_f}{\sigma_{-1} \times K_{sur} \times K_{sz}}$$

$$\frac{1}{2} = \frac{0.58\,F}{550} + \frac{1.16F \times 1.378}{275 \times 0.89 \times 0.85}$$

$$= 0.001\,05\,F + 0.007\,68\,F = 0.008\,73\,F$$

∴ $$F = \frac{1}{2 \times 0.008\,73} = 57.3 \text{ N}$$

and according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a \times K_f}{\sigma_{-1} \times K_{sur} \times K_{sz}}$$

$$\frac{1}{2} = \frac{0.58\,F}{470} + \frac{1.16F \times 1.378}{275 \times 0.89 \times 0.85}$$

$$= 0.001\,23\,F + 0.007\,68\,F = 0.008\,91\,F$$

∴ $$F = \frac{1}{2 \times 0.008\,91} = 56 \text{ N}$$

Taking larger of the two values, we have $F = 57.3$ N **Ans.**