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III/IV B. Tech (Regular/Supplementary) DEGREE EXAMINATION

November, 2019 — **Mechanical Engineering**

Fifth Semester — **DESIGN OF MACHINE ELEMENTS - I**

Time: Three Hours — **Maximum:** 60 Marks

Answer Question No.1 compulsorily. (1X12 = 12 Marks)

Answer ONE question from each unit. (4X12=48 Marks)

1. Answer all questions (1X12=12 Marks)
   a) What is the significance of preferred numbers in the design of machine elements?
   b) List the mechanical properties of materials to be considered while designing a machine element
   c) Which theory of static failure is best suited for ductile materials and why?
   d) List the factor which influences the endurance limit of a machine component.
   e) List the applications of cotter joints.
   f) Give the graphical representation of Goodman's equation
   g) What is the difference between caulking and fullering?
   h) What do you understand by the term 'efficiency of a riveted joint'?
   i) What are the assumptions made in the design of welded joint?
   j) Why are square threads preferable to V-threads in power transmission?
   k) Why self-locking of threads is necessary for power transmission?
   l) What are the advantages of threaded joints?

UNIT I

2. a) Describe the basic procedure of machine design 8M
   b) It is required to standardized eleven shafts from 100 to 1000 mm in diameter. Specify their diameters. 4M

(OR)

3. a) A bracket, made of steel FeE 200 (Syt=200 MPa) and is subjected to a force of 5 kN acting at an angle of 30^0 to the vertical is shown in Fig.1. The factor of safety is 4. Determine the dimensions of the cross-section of the bracket. 6M

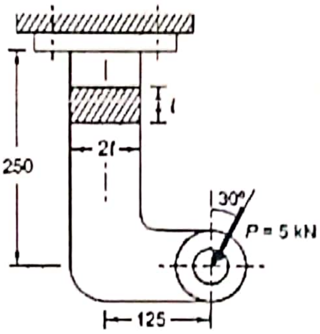


Fig.1.

   b) The stresses induced at a critical point in a machine component made of steel 45C8 are as follows: $\sigma_x = 100MPa;\ \sigma_y = 40\ MPa\ and\ \tau_{xy} = 80MPa$. Calculate the factor of safety by (i) maximum normal stress theory (ii) Maximum shear stress theory and (iii) the distortion energy theory 6M

UNIT II

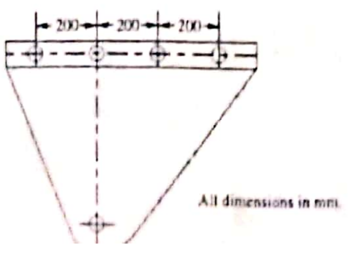
4. a) A hot rolled steel shaft is subjected to a torsional moment that varies from 330 N-m clockwise to 110 N-m counterclockwise and an applied bending moment at a critical section varies from 440 N-m to – 220 N-m. The shaft is of uniform cross-section and no keyway is present at the critical section. Determine the required shaft diameter. The material has an ultimate strength of 550 MN/m² and yield strength of 410 MN/m². Take the endurance limit as half the ultimate strength, the factor of safety of 2, size factor of 0.85 and a surface finish factor of 0.62. 8M
   b) Describe the S-N curve for steels. 4M

(OR)

5. It is required to design a cotter joint to connect two steel rods of equal diameters. Each rod is subjected to an axial force of 50kN. Design the joint and specify its main dimensions. Assume 20C8 material for all components. 12M

UNIT III

6. a) Find the value of P for the joint shown in Fig.2., based on working shear stress of 100 MPa for the rivets. The four rivets are equal, each of 20 mm diameter. 6M



b) A welded connection, as shown in Fig.3., is subjected to an eccentric force of 60 kN in the plane of welds. Determine the size of the welds if the permissible shear stress for the weld is 100 MPa. Consider all dimensions shown are in mm. 6M

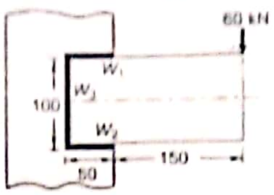


Fig.3.

(OR)

7. A steam boiler is to be designed for a working pressure of 2.5 N/mm^2 with its inside diameter 1.6 m. Give the design calculations for the longitudinal and circumferential joints for the following working stresses for steel plates and rivets: In tension equals to 75 MPa; In shear equals to 60 MPa; In crushing equals to 125 MPa. Draw the joints to a suitable scale. 12M

UNIT-IV

8. An electronic information sign for a rail network platform has the dimensions indicated in Fig.4. Due to wind pressure effects, a maximum horizontal load of 4kN is considered to apply at the centre of the sign. The vertical supporting column consists of a 200mm outside diameter pipe and is welded to a base plate. The base plate is in turn secured directly to the platform by means of six specially grouted ordinary bolts. Using a factor of safety of 1.6, determine the necessary diameter of the specially grouted black bolts. 12M

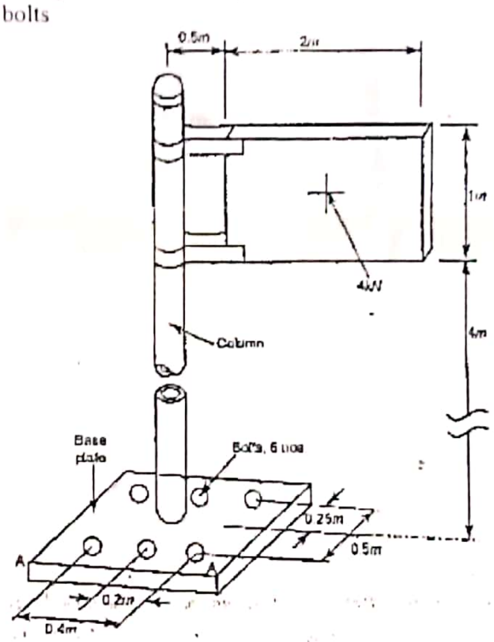
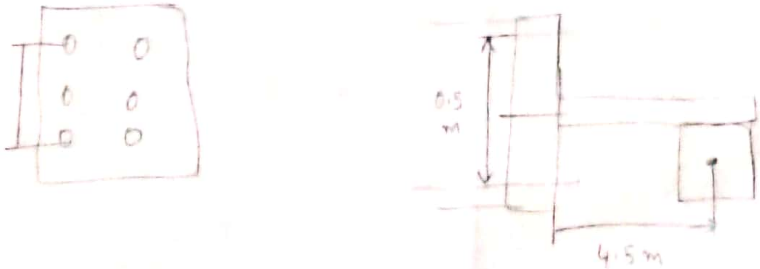


Fig.4.

(OR)

9. Design a screw jack for lifting a load of 50 kN through a height of 0.4 m. The screw is made of steel and nut of bronze. The coefficient of friction between the steel and bronze pair is 0.12. The dimensions of the swivel base may be assumed proportionately. The screw should have square threads. 12M



①

14ME502

III/IV B.Tech (Regular) Degree Examination

Mechanical Engineering

November, 2019 (Scheme of Evaluation) Design of Machine Elements-I

Fifth Semester Max. Marks - 60

Answer Question 1 Compulsorily $1 \times 12 = 12M$

1.

(a) What is the significance of preferred numbers in the design of machine

→ When there is a need of different ranges of a same m/c components manufacturer can fix the most suitable no of models in between the lower and upper capacity of a m/c component. This can be done by preferred numbers.

(b) List the mechanical properties of material to be considered while designing a machine element

→ Ductility, Toughness, brittleness, compressive strength, endurance strength, Tensile strength, creep strength etc.

(c) Which theory of static failure is best suited for ductile metals and why?

→ Distortion energy theory is best suited for ductile materials, because it deals with all maximum stresses and shear stresses i.e (σ_1, σ_2 and τ_{max})

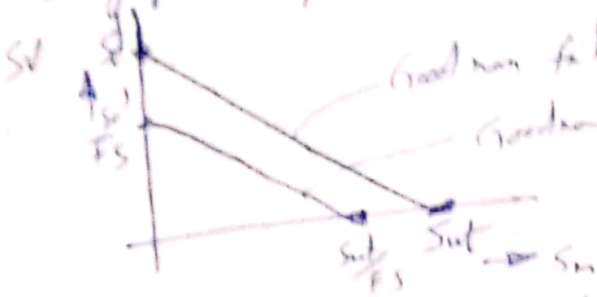
(d) List the factors which influence the endurance limit of a material

→ size factor, surface finish factor, Reliability factor, fatigue stress concentration factor

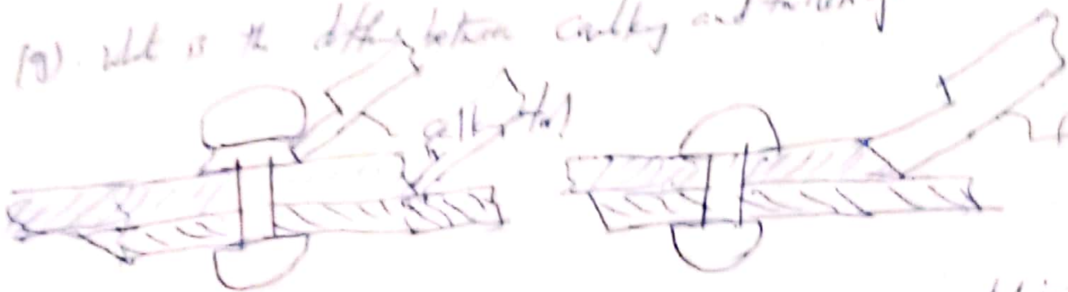
(e) List the applications of cotter joints

→ Where ever the two rods need to be connected, and the applied load is axial cotter joint can be used. eg: locomotive engines, (f). Give the graphical representation of Goodman's equation

→ Sd



(g). What is the difference between caulking and fullering?



(h). What do you understand by the term 'efficiency of a riveted joint'?

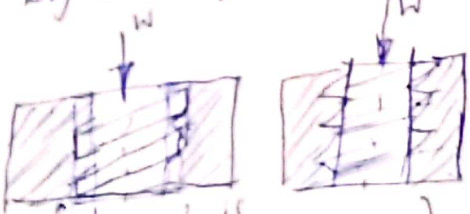
→ it is the ratio of strength of riveted joints to strength of unriveted plate

$$\eta = \frac{\text{least}\,(P_t, P_s, P_c)}{P \times p \times t}$$

(i). What are the assumptions made in the design of welded joints?

→ (i). Thread stress are negligible. (ii). the properties of weld metal is same as plates (iv) the load is applied on throat area.
(iii) there is no residual stress in the weld ... to be through in pure tension.

(j). Why are square threads preferable to V-threads?



friction angle is high as ... 30°
$F = \mu N$
hence f.F ...
power transmitted will be more

⌞ There is a friction angle and helix angle.
∴ the power transmitted will be less compared to square thread.

(k). Why self-locking of threads is necessary for power transmission?

→ Why lowering the load using threaded screws. If $\mu > \alpha$ then load will be lowered without any application of external load which is not the required condition. Thus self locking is necessary in power transmission.

(l). What are the advantages of threaded joints?

→ (i). threaded joints come under temporary joint. Threaded parts can be dis-assembled easily
(ii). They are ... as tensile members
(iii). no skill is required for the worker

UNIT-I. ②

2. (a). Describe the basic procedure of machine design.

Identification of problem
↓
Definition of problem
↓
Synthesis
↓
Analysis & optimization
↓
Evaluation
↓
Documentation.

diagram — 4M
explanation — 4M

2(b). 11 shafts between 100 mm to 1000 mm

$100\times\phi^0$ $\quad 100\phi^1 \ldots \quad 100\times\phi^{10} = 1000$

$$100\times\phi^{10} = 1000$$

$$\phi = \sqrt[10]{10} = 1.2589 \quad \text{— 2M}$$

∴ Diameters are
100, 125.89, 158.48, 199.516, 251.16,
316.19, 398.058, 501.116, 630.85,
794.18, 1000. — 2M

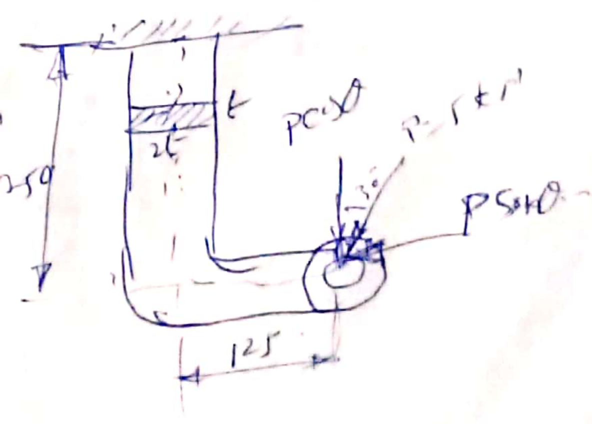
(OR)

3 (a). $S_{yt} = 200$ MPa ; F.S = 4

$\sigma_{t\,allow} = \frac{200}{4} = 50$

bending stress due to $P\sin\theta$.

$$\sigma_{b_x} = \frac{P\sin\theta \times e_y \times 2t/2}{I_{xx}}$$



direct tensile st

$$\sigma_t = \frac{P\cos\theta}{A} = \frac{5\times\cos 30}{2t\times t}$$

$$= \frac{2.165}{t^2}\ kN/mm^2$$

— 2M

Resultant bending Mt.

$$M_b = P\cos\theta\times 125 + P\sin\theta\times 250$$

$$= 5\times10^3\cos 30\times 125 + 5\times10^3\sin 30\times 250$$

$$= 1166.25\ kN\text{-}mm$$

bending stress

$$\sigma_b = \frac{M_b\times y}{I} = \frac{1166.25\times\left(\frac{2t}{2}\right)}{\frac{t\times(2t)^3}{12}}$$

$$= 1749.375\ /t^3\ kN/mm^2$$

— 2M

Total tensile stress σ_{max} =

$$\frac{1749.375\times10^3}{t^3} + \frac{2.165\times10^3}{t^2} = 50$$

$$50t^3 - 2.165\times10^3\,t - 1749.375\times10^3 = 0$$

$$\boxed{t = 33.148\ mm \simeq 34 \quad b = 68\ mm}$$

— 2M

3(b). 45C8

$S_{yt} = 380$ MPa

Soln

$\sigma_x = 100$; $\sigma_y = 60$; $\tau_{xy} = 80$.

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

$$= \frac{160}{2} + \sqrt{(20)^2 + 80^2}$$

$$= 155.44\ MPa$$

$$\sigma_2 = -15.4\ MPa$$

$$\tau_{max} = 85.44\ MPa$$

— 2M

(i) Max normal stress theory

$$\sigma_1 = \frac{S_{yt}}{F.S}$$

$$155.44 = \frac{380}{F.S}$$

$$\boxed{F.S = 2.44}$$

(ii) Max shear stress theory

$$\tau_{max} = \frac{S_{yt}}{2.F.S}$$

$$85.44 = \frac{380}{2\times F.S}$$

$$\boxed{F.S = 2.22}$$

— 2M

(iii). Max distortion energy theory

$$\sqrt{\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2} = \frac{S_{yt}}{F.S}$$

$$\sqrt{155.44^2 + (155.44\times15.4) + (-15.4)^2} = \frac{380}{F.S}$$

$$163.68 = \frac{380}{F.S}$$

$$\boxed{F.S = 2.32}$$

— 2M

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(3)

UNIT – II

4.(a).

$T_{max} = 330$ N-m

$T_{min} = -110$

$M_{b\,max} = 440$ N-m

$M_{b\,min} = 220$ N-m

$S_{ut} = 550$ MPa ; $S_{yt} = 410$ MN/m²

$S_e' = 0.5 S_{ut}$ F.S = 2 $K_a = 0.85$

$K_b = 0.62$

$T_m = \frac{T_{max} + T_{min}}{2} = 110$ N-m

$T_v = 220$ N-m

mean shear stress $\tau_m = \frac{16 T_m}{\pi d^3} = \frac{16 \times 110}{\pi d^3}$

$= 560/d^3$

variable stress $\tau_v = \frac{16 T_v}{\pi d^3} = \frac{1120}{d^3}$ —2M

$S_e' = 0.5 S_{yt} = 275 \times 10^6$ N/m²

$\tau_e' = 0.55 S_e' = 151.25 \times 10^6$

$\tau_y = 0.5 \times 410 \times 10^6 = 205 \times 10^6$ N/m²

equivalent shear stress

$\tau_{es} = \tau_m + \frac{\tau_v \cdot \tau_y \times K_{fs}}{\tau_e' \times K_a K_b}$

$\tau_{es} = 3440/d^3$ —2M

for bending Moment.

$M_m = 110$ N-m

$M_v = 330$ N-m

$S_{mb} = \frac{32 M_m}{\pi d^3} = \frac{1120}{d^3}$

$S_{vb} = \frac{32 M_v}{\pi d^3} = \frac{3360}{d^3}$

$S_{neb} = S_{ne} = S_m + \frac{S_v \cdot S_y \times k_{yl}}{K_a K_b \cdot S_e' b}$

$= \frac{10626}{d^3}$ —2M

for mean shear stress the

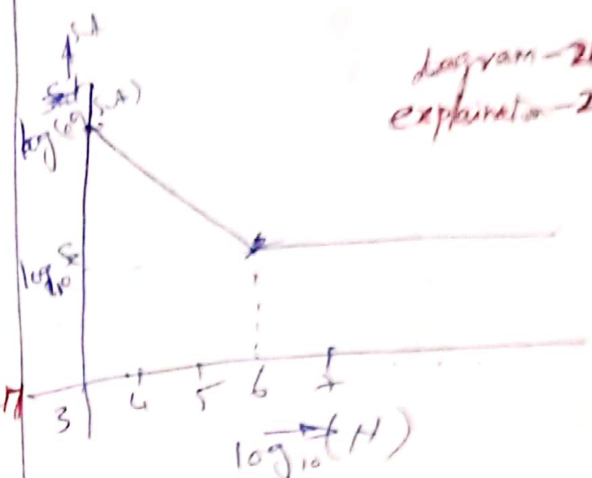
$(\tau_{es})_{max} = \frac{\tau_y}{F.S} = \frac{1}{2}\sqrt{(S_{ne})^2 + 4\tau_{es}^2}$

$\frac{205 \times 10^6}{2} = \frac{1}{2}\sqrt{\left(\frac{10626}{d^3}\right)^2 + 4\left(\frac{3440}{d^3}\right)^2}$

$d^3 = 0.0395$ m

$\boxed{d = 39.5 \text{ mm}}$ — 2M

4(b). Describe S-N Curve for steel



diagram—2M

explanation—2M

(5) Cotter joint

Two steel rods 20C8 material

in 20C8 $S_{yt} = 600$ MPa

Consider fs for spigot and socket as 6 and for cotter 4

we have $\sigma_t = \frac{S_{yt}}{FS} = \frac{600}{6} = 66.66$ N/mm²

$\sigma_c = \frac{S_{yc}}{FS} = \frac{2 S_{yt}}{FS} = \frac{2 \times 600}{6} = 133.33$

$\tau = \frac{S_{sy}}{FS} = \frac{0.5 S_{yt}}{FS} = 33.33$ MPa

for cotter

$\sigma_t = \frac{S_{yt}}{FS} = \frac{600}{6} = 100$ N/mm²

$\tau = \frac{S_{sy}}{FS} = \frac{0.5 S_{yt}}{FS} = 50$ N/mm² —2M

Calculate dimensions

step 1 $d = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4 \times 50 \times 10^3}{\pi\, 66.60}}$

$= 30.90 \simeq 32$ mm

step 2 $t = 0.31 d$ $t = 10$ mm

step 3 $P = \left[\frac{\pi}{4} d_2^2 - d_2 t\right] \sigma_t$

$d_2^2 - 4.73 d_2 - 954.88 = 0$

$d_2 = 37.90 \simeq 40$ mm —4M

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

$\Rightarrow d_1^2 - 12.73 d_1 - 2045.59 = 0$

$d_1 = 52.04 \simeq 55$ mm

step 5 $d_3 = 1.5 d = 48$ mm

$d_4 = 2.4 d = 76.8 \simeq 80$ mm —2M

step 6. $a = c = 0.75 d = 24$ mm

step 7 $b = \frac{P}{2\tau t} = 50$ mm

(or) $b = \sqrt{\frac{3P}{t \sigma_b}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]}$

$= 50$ mm

step 8 stress in spigot

$\sigma_c = \frac{P}{t d_2} = \frac{50 \times 10^3}{10 \times 40} = 125$ MPa

$\tau = \frac{P}{2 a d_2} = \frac{50 \times 10^3}{2 \times 24 \times 40} = 26.04$ MPa

step 9: stress in socket end

$\sigma_c = \frac{P}{(d_4 - d_2) t} = \frac{50 \times 10^3}{(80 - 40) 10} = 125$ MPa

$\tau = \frac{P}{2(d_4 - d_2) c} = \frac{50 \times 10^3}{2(80-40) 24}$

$= 26.04$ MPa —2M

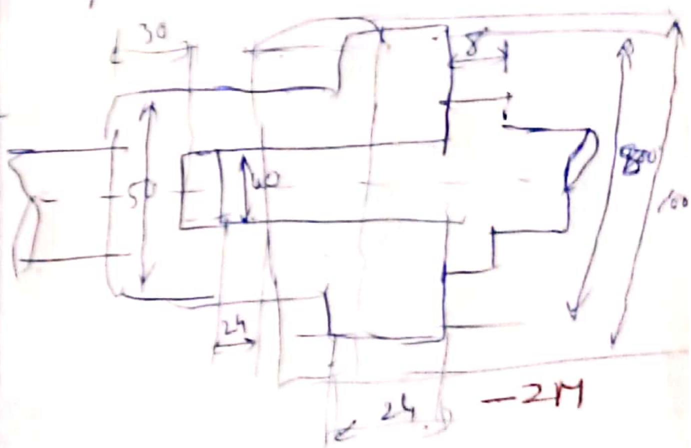
$\sigma_c < 133.33$ N/mm²

$\tau < 33.33$ MPa

Stress induced in spigot and socket are within the limit.

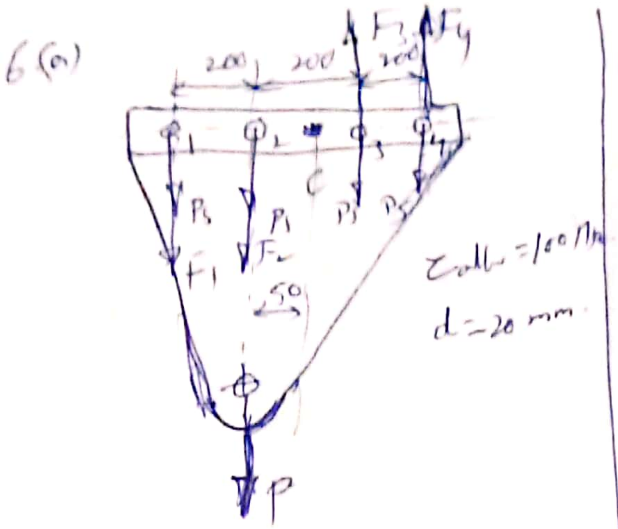
step 10: $t_1 = 0.45 d = 14.4 \simeq 15$ mm

taper of the cotter 1 in 32



—2M

6 (a)



$\tau_{all} = 100$ MPa

$d = 20$ mm

Line of action of force P is $\perp$ to axis of the rivet.

∴ Centroid of the system is $C(300, 0)$

$L_1 = 300$; $L_2 = 100$; $L_3 = 100$

$L_4 = 300$

eccentricity $e = 100$ — 2M

Primary shear $P_s = \frac{P}{n} = \frac{P}{4} = 0.25P$

max tensile force is acts at 1 and 4.

$P \times e = \frac{F_1}{l_1}\left[l_1^2 + l_2^2 + l_3^2 + l_4^2\right]$

$P \times 100 = \frac{F_1}{300}\left(2 \times 300^2 + 2 \times 100^2\right)$

$F_1 = 0.15P$ N
$= F_4$

$F_3 = F_2 = 0.05P$ N — 2M

Resultant shear is maximum on rivet 1

$R_1 = P_s + F_1 = 0.25P + 0.15P$
$= 0.4P$

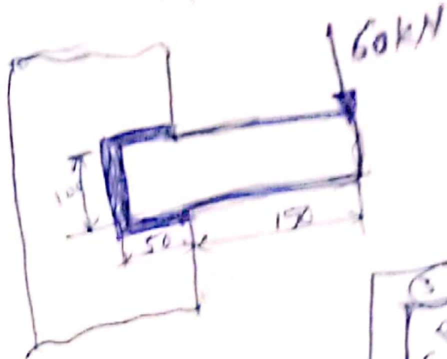
$\therefore \tau_{max} = \frac{R_1}{A}$

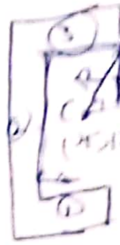
$100 = \frac{0.4P}{\frac{\pi}{4} \cdot 20^2}$

$\boxed{P = 78.55 \text{ kN}}$ — 2M

6 (b)

τ_{all} : 100 MPa





Total area

$A = A_1 + A_2 + A_3$
$= 200t$

Primary shear $P_s = \frac{P}{A} = \frac{60 \times 10^3}{200t}$
$= 300/t$ — 2M

$e = 150 + 25 = 175$ mm

radial distance of farthest point 'A' from centroid $R = 55.90$ mm

for the shape of welded $\sqsubset$

polar moment of inertia

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

$n = 50$
$m = 100$

$= t\left[666.66 \times 10^3 - 375{,}000\right]$

$\doteq 291{,}666\ t$

$F_s = \frac{T \cdot r}{J} = \frac{60 \times 10^3 \times 175 \times 55.90}{291{,}666 \cdot t}$

$F_s = 2012.4/t$ — 2M

$\tan\theta = \frac{50}{4}$ ⟹ $\theta = 63.43$

Resultant str. $\tau = \sqrt{P_s^2 + F_s^2 + 2P_sF_s\cos\theta}$

$= 2163.29/t$

$= \tau_{all} = 100$

$\boxed{\therefore t = 21.63 \text{ mm}}$

$s = \cancel{30.59}$ mm

$s = 0.707t$

$\boxed{s = 15.29 \text{ mm}}$ — 2M

7. steam boiler internal pressure $p = 2.5\ N/mm^2$ inside dia $D = 1.6\ m$

$\sigma_t = 75\ MPa$; $\tau = 60\ MPa$; $\sigma_c = 125\ MPa$.

Assume $\eta_l = 0.8$

Longitudinal joint

(i) thickness of the plate

$t = \frac{pD}{2\sigma_t \eta_l} + 1$

$= \frac{2.5 \times 1600}{2 \times 75 \times 0.8} + 1$

$= 34\ mm \simeq 35\ mm$

(ii). Dia of the rivet.

$d = 6.05\sqrt{t}$:

$= 35.79 \simeq 36\ mm$

$d_1 = 37.5$

(iii). pitch of the rivets

$P_t = (p - d)\, t\, \sigma_t$

$= (p - 37.5)\, 35 \times 75$

$P_s = 4 \times 1.875 \times \frac{\pi}{4} d^2 \tau + \frac{\pi}{4} d^2 \tau$

$= 4 \times 1.875 \times \frac{\pi}{4}\, 37.5^2 \times 60$

$+ \frac{\pi}{4}\, 37.5^2 \times 60$

$= 562.95\ kN$

$P_t = P_s$

$p - 37.5 \Rightarrow$

$p = 251.9$

According to IBR max pitch

$P_{max} = C \times t + 41.28\ mm$

for 5 rivets $C = 6$

$P_{max} = 6 \times 35 + 41.28$

$= 251.28\ mm$

pitch of the inner row $p' = \frac{251.28}{2}$

$= 125.64\ mm$

— 3M

(iv) Distance between rows of rivets

According to IBR distance between outer row to next row $= 0.2p + 1.15d$

$= 0.2 \times 252 + 1.15 \times 37.5$

$= 93.52\ mm$

distance b/w the inner rows for zigzag

$= 0.165p + 0.67d$

$= 0.165 \times 252 + 0.67 \times 37.5$

$= 42.83\ mm$

(v). Thickness of butt strap.

for wide strap $t_1 = 0.75t$

$= 0.75 \times 35 =$

$= 26.25$

for narrow butt strap $t_2 = 0.625t = 21.87\ mm$

(vi) Margin $m = 1.5d = 52.5$

efficiency of the joint.

$P_t = (252 - 37.5)\, 35 \times 75$

$= 563.\ kN$

$P_s = 562.95\ kN$.

$P_c = n \times d \times t \times \sigma_c$

$= 5 \times 37.5 \times 35 \times 125$

$= 820\ kN$.

$P_{t2} = (p - 2d)\, t\sigma_t + \frac{\pi}{4} d^2 \times \tau$

$= (252 - 2 \times 37.5)\, 75 \times 35 +$

$\frac{\pi}{4}\, 37.5^2 \times 60$

$= 530.859\ kN$

strength of riveted joint = 530.817 kN

strength of unriveted plate = $p \times t \times \sigma_t$

= 75 × 35 × 11

= 661.5 kN

$$\eta = \frac{530.817}{661.5} = 80.4\,\%$$

— 3M

for Circumferential joint:

(i) thickness same as longitudinal joint

$t = 35$ mm

$d_1 = 37.5$ mm

(ii) no. of rivets

shear resistance of rivet joint

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

Total shear load on the circumferential joint

$= \frac{\pi}{4} D^2 \times p$

$$n = \frac{\frac{\pi}{4} \times 1600^2 \times 2.5}{\frac{\pi}{4} \times 37.5^2 \times 60}$$

$n = 75.85 \simeq 76$

$\boxed{n = 76}$ — 3M

(iii) pitch of the joint.

Assume zig-zag double riveted

no. of rivets/row = $\frac{76}{2}$ = 38

pitch of the rivets $p_1 = \frac{\pi(D+t)}{\text{no. of rivets/row}}$

$= \frac{\pi(1600+35)}{38}$

$= 135.10 \simeq 140$ mm

(iv) efficiency of the joint

$$\eta_c = \frac{p_1 - d}{p_1} = \frac{140 - 37.5}{140}$$

$\eta_c = 73.2\,\%$

(v) distance between the rows of rivets

for zig-zag type $C_1 = 0.33p + 0.67d$

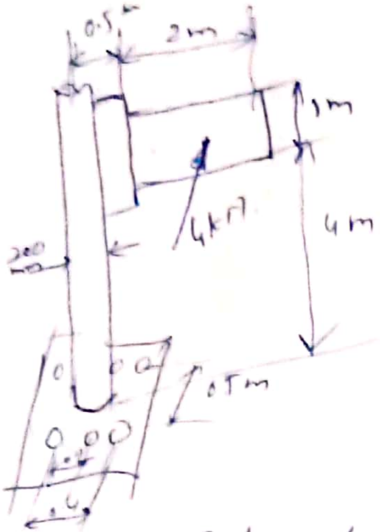
$= 0.33 \times 140 + 0.67 \times 37.5$

$= 71.325 \simeq 72$ mm

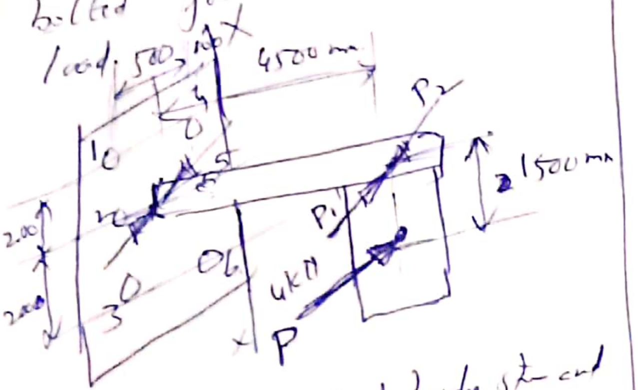
(vi) Margin $m = 1.5d$

$= 56.25$ mm — 3M

UNIT-IV



pole is fastened to the plate using welded joint. Assuming welded joint is rigid we have bolted joint with eccentric load.



P_1 creates direct bending stress and direct shear stress

direct shear stress

$$P_s = \frac{P}{N} = \frac{4\times10^3}{6} = 666.66$$ —2M

due to bending the tensile force created in bolts is given by

Assuming plate dimension 700×600 mm from bolting edge we have

$l_4 = l_5 = l_6 = 100$

$l_1 = l_2 = l_3 = 600$ mm

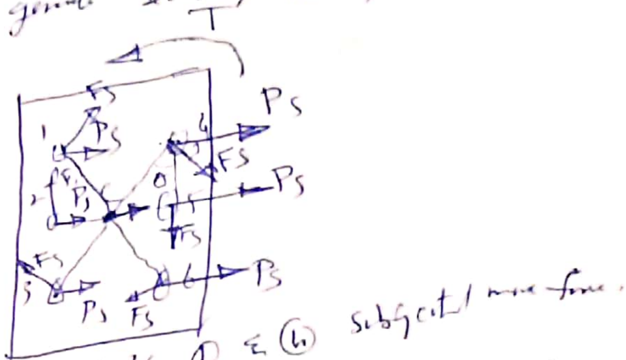
max tensile force $$F_1 = \frac{F\times e\times l_1}{(l_1^2+l_2^2+\dots+l_6^2)}$$

$$= \frac{4\times10^3\times 4500\times 600}{3(100^2)+3(600^2)}$$

$$= 9729.729\ N$$

$$\boxed{\sigma_t = \frac{F_1}{A_s} = \frac{9729.729}{A_s}}$$ —4M

The force P and P_2 create couple (or) twisting moment on the plate which in turn generate secondary shear force in bolts



bolts ① & ⑥ subjected to max force.

$r_1 = r_4 = \sqrt{(250^2+200^2)} = 320.156$ mm

$= r_3 = r_6$

$r_2 = r_5 = 250$

secondary shear force $$F_s = \frac{F\times 1500\times r_1}{(r_1^2+r_2^2+\dots+r_6^2)}$$

$$= \frac{4\times10^3\times1500\times320.156}{4\times320.156^2+2\times250^2}$$

$F_s = 3593.22\ N$ —2M

angle between P_s & F_s for bolt ⑥ is

$\theta_s = \frac{250}{200} = 51.34°$

Resultant shear force $R_s = \sqrt{P_s^2+F_s^2+2P_sF_s\cos\theta_s}$

$= 4043.33\ N$

Shear stress $\tau = \frac{R_s}{A_s} = \frac{4043.33}{A_s}$

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

$$= \frac{1}{A_s}\sqrt{\left(\frac{9729.729}{2}\right)^2+(4043.33)^2}$$

$\tau_{max} = 6325.77/A_s$

Assuming bolt is made of low carbon steel

$S_{yt} = 268$ MPa

$\tau_{max} = \frac{S_{yt}}{2}\Big/ F.S = \frac{268}{2\times1.6}$

$= 83.75$

$6325.77/A_s = 83.75$

$A_s = 75.53\ mm^2$

from data handbook $\boxed{M12\times1.75}$ —4M

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9.) load $W = 50\ kN$

Max length $H_1 = 0.4\ m = 400\ mm.$

Screw - steel $\sigma_{cs} = 80\ MPa$

$\tau_s = 45\ MPa.$

nut Bronze.

$\sigma_{tb} = 40\ MPa.$

$\sigma_b = 15\ MPa.$

bearing pressure $\tau = 25\ MPa.$

$\mu_{sb} = 0.12$

Screw square threaded

handle - steel bending stress ?

$\sigma_b = 150\ MPa$

(i). Design of screw for spindle:

d_c = Core dia of the screw

Screw is under compression $W = \frac{\pi}{4} d_c^2 \cdot \frac{\sigma_{cs}}{F.S}$ Let F.S = 2

$50 \times 10^3 = \frac{\pi}{4} \cdot d_c^2 \times \frac{80}{2}$

$d_c = 39.9\ mm.$

for square thread of normal series ; the dimen. of the screw are selected from data book

Core dia $d_c = 40\ mm$

$d_o = 48\ mm.$

pitch of the screw = 8 mm.

Area of core is 1257 mm²

for principal stress

$d = \frac{d_c + d_o}{2} = \frac{40+48}{2} = 44\ mm$

$\tan\alpha = \frac{p}{\pi d} = \frac{8}{\pi \times 44} =$

$\alpha = 3.31°$

$\mu = \tan\phi.$

$0.12 = \tan\phi$

$\phi = 6.84$

Torque reqd to rotate the screw in the nut

$T_1 = P \times \frac{d}{2} = W \tan(\alpha+\phi) \cdot \frac{d}{2}$

$= 50 \times 10^3 \tan(3.31 + 6.84) \cdot \frac{44}{2}$

$T_1 = 196.93\ N\text{-}m.$

Compression stress due to axial load

$\sigma_c = \frac{W}{A_c} = \frac{50 \times 10^3}{\pi/4 \cdot 40^2} = 39.80\ MPa$

Shear stress due to torque $\tau = \frac{16 T_1}{\pi d_c^3} = \frac{16 \times 196.93 \times 10^3}{\pi \times 40^3}$

$= 15.62\ MPa.$

Max principal stress $\sigma_{c\,max} = \frac{1}{2}\left[\sigma_c + \sqrt{\sigma_c^2 + 4\tau^2}\right]$

$= \frac{1}{2}\left[39.80 + \sqrt{39.80^2 + 4 \cdot 15.62^2}\right]$

$= 45.198$ MPa ... $< \sigma_{c_s}$ (80 MPa)

$\therefore$ screw is safe (4M)

(ii) Design of Nut:

n = no of thread in contact

h = length of the nut $= n \times p$

t = thickness of the screw $= \frac{p}{2} = 4$ mm

Assuming load is distributed uniformly over the C/s. area

bearing pr $P_b = 15 = \frac{W}{\frac{\pi}{4}(d_o^2 - d_c^2) \cdot n}$

$15 = \frac{80 \times 10^3}{\frac{\pi}{4} \cdot [48^2 - 40^2]\, n}$

$n = 6.03 \simeq 8$ thread

$h = n \times p = 8 \times 8 = 64$ mm.

stress induced $\tau_{(screw)} = \frac{W}{\pi n d_c t} = \frac{50 \times 10^3}{\pi \times 8 \times 40 \times 4}$

$= 12.44$ MPa. $< \tau_s$ (45)

Shear stress in nut $\tau_{(nut)} = \frac{W}{\pi n d_o t}$

$= \frac{50 \times 10^3}{\pi \times 8 \times 48 \times 4}$

$= 10.36$ MPa. $< \tau_n$ (25)

Let D_1 is the outside dia of the nut

D_2 = outside dia of the collar

t_1 = thickness of nut collar.

tearing strength of the nut $W = \frac{\pi}{4}[D_1^2 - d_o^2] \cdot \sigma_{t_n}$.

$50 \times 10^3 = \frac{\pi}{4}(D_1^2 - 48^2)\,\cancel{8}\,40$

$D_1 = 62.42$ mm $\simeq 65$ mm.

for crushing of the collar of the nut

$W = \frac{\pi}{4}[D_2^2 - D_1^2] \cdot \sigma_{cr}$.

$50 \times 10^3 = \frac{\pi}{4}(D_2^2 - 65^2) \cdot 40$

$\boxed{D_2 = 76.27}$

Stress of the collar $W = \pi D_1 t_1 \tau$

$$50\times10^3 = \pi\times65\times t_1\times 25$$

$$t_1 = 9.794 \text{ mm}$$ (4M)

(ii) Design of handle and cup:

Dia of the head (D_3) on the top of the screw rod is usually taken as $1.75 d_0$

$\therefore D_3 = 1.75\times\cancel{50}\,48 = \cancel{H}\,84$

Cup is fitted to the head with a pin of dia

$D_4 = \frac{D_3}{4} = \frac{84}{4} = 21 \text{ mm} \simeq 22 \text{ mm}$

~~Length of the cup = 5~~

Torque to overcome (T_2) friction at the top of the screw

$$T_2 = \frac{2}{3}\cdot\mu_1 W\left[\frac{R_3^3 - R_4^3}{R_3^2 - R_4^2}\right]$$

$$= \frac{2}{3}\cdot 0.12\times 50\times10^3\left[\frac{42^3 - 11^3}{42^2 - 11^2}\right]$$

$$= 177.132 \text{ N-m}$$

Total torque $T = T_1 + T_2 = 196.93 + 177.132 = 374.061$ N-m

Assuming force applied by person is 300N

Length of the handle $= T = P\times l$.

$l = T/P = 1246$ mm $\simeq 1250$ mm

bending moment of the handle $M =$ Force applied $\times$ length

$= 300\times1250 = 375\times10^3$ N-mm

Let D is Dia of the handle

$\sigma_b = 15 = \sigma_t = 150/2 = 75$ MPa.

$$375\times10^3 = \frac{\pi}{32}\times\sigma_b\times D^3 = \frac{\pi}{32}\cdot75\times D^3$$

$$D = 37.07 \text{ mm} \simeq 40 \text{ mm}$$

Length of the head (H) is taken as $2D$.

$H = 2\times D = 80$ mm (2M)

effective length of the buckling screw

$L = l_{eff} + \frac{1}{2}$ Height of nut $= H_1 + h/2$

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$L = 400 + 80/2 = 440$ mm

Critical load $W_{cr} = A_c \times \sigma_y \left[1 - \frac{\sigma_y}{4C\pi^2 E}\left(\frac{L}{k}\right)^2\right]$.

one end fixed and other end free $C = 0.25$

$k = 0.25 d_c = 0.25 \times 40 = 10$ mm.

$W_{cr} = \frac{\pi}{4} \cdot 40^2 \times 150 \left[1 - \frac{150}{4 \times 0.25 \times \pi^2 \times 210 \times 10^3}\left(\frac{440}{10}\right)^2\right]$

$= 161.975$ kN. $> 50 \times 10^3$

$\therefore$ screw design is safe.

(iv). ~~Design of body~~

Dia of body at the top $D_5 = 1.5 \cdot D_2 = 1.5 \times 76.27$

$= 114.405$ mm

thickness of the body $t_3 = 0.25 d_o = 0.25 \times 48 = 12$ mm

Inside dia at the bottom $D_6 = 2.25 D_2 = 2.25 \times 76.27$

$= 171.56 \simeq 180$ mm

outside dia of the bottom $D_7 = 1.75 \times D_6 = 315$ mm

thickness of the base $t_2 = 2t_1 = 2 \times 9.799 \simeq 19.598$ mm

Height of the body $=$ Max lift $+$ Height of nut $+ 100$ mm extra

$= 440 + 64 + 100 = 604$ mm

Torque req to rotate the screw without friction

$T_0 = W \tan\alpha \cdot \frac{d}{2} = 50 \times 10^3 \tan 3.31 \frac{44}{2}$

$= 63.618$ N-m

efficiency of screw jack $\eta = \frac{T_0}{T} = \frac{63.618}{374}$ (2M)

$= 0.17 \simeq 17.02\%$.

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