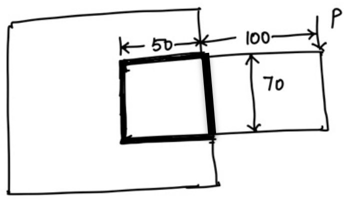
USN

Internal Assessment Test 3 –Jan. 2022

Sub:	Design of Machine Elements - 1				Sub Code:	18ME52	Branch:	Mech
Date:	25.01.2022	Duration:	90 min’s	Max Marks: 50	Sem/Sec:	V/A&B		OBE

Answer All the Questions
Usage of Design data handbook is permitted

		MARKS	CO	RBT
1.	Design a screw jack with a lift of 300 mm to lift a load of 50 kN	[20]	CO5	L3
2.	Design a rigid flange coupling to transmit 18 kW at 1440 rpm. The allowable shear stress for CI flange is 4 MPa. The shaft, bolts and key are made up of annealed steel having allowable shear stress of 93 MPa. Allowable crushing stress for key is 186 MPa.	[15]	CO5	L3
3.	A bracket supporting a load P = 3000 N is welded to a vertical member by means of 4 fillet welds as shown. Calculate the size of the weld, if the stress in the throat section is not to exceed 85 MPa.	[15]	CO5	L3



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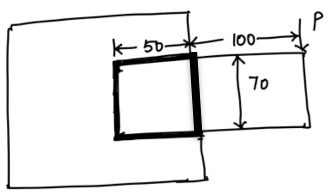

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Scheme of Evaluation

Prepared by: Prof.Manikandan H

Q.No	Scheme	Marks
1.	Design of Screw and check for screw	4
	Design of collar	4
	Design of nut	2
	Design of handle	2
	Design of cup	2
	Efficiency of screw	2
	Check for overhauling	2
	Design of Body	2
2.	Torque calculation	1
	Design of shaft	1
	Bolt circle diameter	1
	Design of hub	3
	Design of flange and check for flange	3
	Design of bolts and check for bolt	3
	Design of key and check for key	3
3.	Free body diagram	3
	CG of weld	2
	Finding r, cos θ and e	2
	Direct shear load per unit length (Pd)	2
	Torsional load per unit length (Pn)	2
	Resultant load per unit length (Pr)	2
	Size of the weld (w)	2

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Solutions

1.

$\therefore$ Allowable or Permissible stress $\sigma = \dfrac{\sigma_y}{FOS} = \dfrac{328.6}{3} = 109.53\ N/mm^2$

Allowable shear stress $\tau = 0.5\,\sigma = 0.5 \times 109.53 = 54.77\ N/mm^2$

Assume an over load of 25%

$\therefore$ Core area of the thread $A_c = \dfrac{\text{Load to be lifted} \times \text{Load factor}}{\text{Allowable stress}} = \dfrac{50 \times 10^3 \times 1.25}{109.53} = 570.62\ mm^2$

Assume the screw to be single start square thread

$\therefore$ From Table 18.8 (DDHB) for normal series square thread.

Std. core area $A_c = 707\ mm^2$

Major diameter $d = 36\ mm$

Minor diameter $d_1 = 30\ mm$

Pitch $p = 6mm$

Pitch diameter $d_2 = \dfrac{d + d_1}{2} = \dfrac{36 + 30}{2} = 33\ mm$

Major diameter of nut $d_{nut} = 36.5\ mm$

ii. Check for the screw

Direct compressive stress on the screw $\sigma_c = \dfrac{W}{A_c} = \dfrac{50 \times 10^3}{707} = 70.72\ N/mm^2$

Torsional moment of the screw $M_{t_s} = W\left[\dfrac{d_2}{2}\left(\dfrac{\tan\alpha + \mu}{1 - \mu\tan\alpha}\right)\right]$ ---- 18.29 (DDHB)

Assuming, Heavy machine oil $\therefore$ From Table 18.4 (DDHB), $\mu = 0.14$

For single start, lead = pitch $\therefore\ l = 6\ mm$

$\therefore\ \tan\alpha = \dfrac{l}{\pi d_2} = \dfrac{6}{\pi \times 33} = 0.05787$ ---- 18.26 (DDHB)

$\therefore\ M_{t_s} = 50 \times 10^3 \left[\dfrac{33}{2}\left(\dfrac{0.05787 + 0.14}{1 - 0.14 \times 0.05787}\right)\right] = 164576.12\ Nmm$

Also $M_{t_s} = \dfrac{\pi}{16} d_1^3 \tau$

i.e., $164576.12 = \dfrac{\pi}{16} \times 30^3 \times \tau$

$\therefore$ **Torsional shear stress on the screw** $\tau = 31.044\ N/mm^2$

$\therefore$ **Maximum principal normal stress** $\sigma_{max} = \dfrac{1}{2}\left[\sigma_c + \sqrt{\sigma_c^2 + 4\tau^2}\right] = \dfrac{1}{2}\left[70.72 + \sqrt{70.72^2 + 4 \times 31.044^2}\right]$

$= 82.42\ N/mm^2 < 109.53\ N/mm^2$

Maximum shear stress $\tau_{max} = \dfrac{1}{2}\left[\sqrt{\sigma_c^2 + 4\tau^2}\right]$

$$= \frac{1}{2}\left[\sqrt{70.72^2 + 4 \times 31.044^2}\right] = 47.05 \text{ N/mm}^2 < 54.77 \text{ N/mm}^2$$

Since the maximum induced normal and shear stress in the screw are less than their permissible values, the design of screw is safe.

iii. Design of screw head or collar (Fig 8.18)

Height of collar $H_1 = 1.5d = 1.5 \times 36 = 54$ mm. Outside diameter of collar $d_{c_o} = 2d = 2 \times 36 = 72$ mm. Diameter of pin above the head $D = 0.5d = 0.5 \times 36 = 18$ mm.

Assuming uniform pressure condition,

$$\text{Mean diameter of collar } d_c = \frac{2}{3}\left[\frac{d_{c0}^3 - D^3}{d_{c0}^2 - D^2}\right] = \frac{2}{3}\left(\frac{72^3 - 18^3}{72^2 - 18^2}\right) = 50.4 \text{ mm} \approx 51 \text{mm}$$

Assume the collar material as cast iron

From Table 18.5 [Old DDHB ; Table 18.5a (New DDHB)] for hardened steel on C.I. collar friction $\mu_c = 0.147$

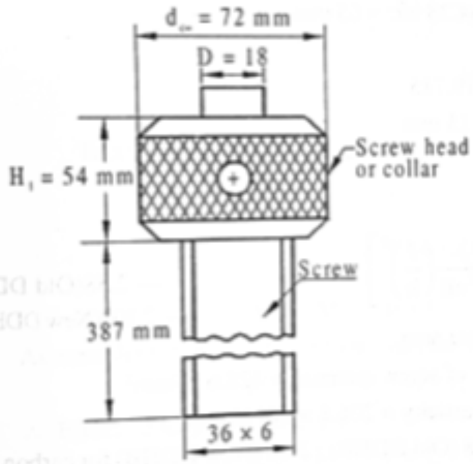


Fig. 8.18

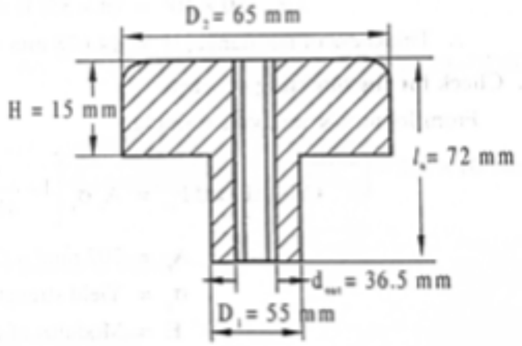


Fig. 8.19

iv. Design of nut (Fig 8.19)

Assume phosphor bronze as nut material. From Table 18.6 (DDHB) Allowable bearing pressure for steel screw and phosphor bronze nut (Jack screw) σ'_b = 10.8 to 17.2 MPa, take, σ'_b = 14 MPa

$$\therefore \text{ Number threads in the nut } i = \frac{4W}{\sigma'_b \pi(d^2 - d_1^2)} = \frac{4 \times 50 \times 10^3}{14 \times \pi(36^2 - 30^2)} = 11.48 \text{ ---- } 18.40 \text{ (DDHB)}$$

$\therefore$ Number of threads i = 12

Length of nut l_n = i.p = 12 × 6 = 72 mm.

Length of screw = Lift + Length of nut + Margin = 300 + 72 + 15 = 387 mm

For ductile material $\sigma_{ut} \approx \sigma_{uc}$. Assume FOS = 6 for ultimate strength and FOS = 3.5 for yield strength

$$\therefore\ \sigma_{c_{nut}} = \frac{345}{6} = 57.5\ N/mm^2$$

$$\sigma_{t_{nut}} = \frac{138}{3.5} = 39.43\ N/mm^2$$

$$\tau_{nut} = 0.5\,\sigma_{t_{nut}} = 0.5 \times 39.43 = 19.715\ N/mm^2$$

Tearing strength of nut $W = \frac{\pi}{4}\left(D_1^2 - d_{nut}^2\right)\sigma_{t_{nut}}$

i.e., $50 \times 10^3 = \frac{\pi}{4}\left(D_1^2 - 36.5^2\right) \times 39.43$

$\therefore$ Outside diameter of nut at the bottom $D_1 = 54.28\ mm \approx 55\ mm$

Crushing strength of nut $W = \frac{\pi}{4}\left(D_2^2 - D_1^2\right)\sigma_{c_{nut}}$

i.e., $50 \times 10^3 = \frac{\pi}{4}\left(D_2^2 - 55^2\right) \times 57.5$

$\therefore$ Outside diameter of nut flange at the top $D_2 = 64.28\ mm \approx 65\ mm$

Shearing of nut collar $W = (\pi D_1)\, H \times \tau_{nut}$

i.e., $50 \times 10^3 = (\pi \times 55)\, H \times 19.715$

$\therefore$ Thickness of nut flange $H = 14.678\ mm \approx 15\ mm$

vi. Design of handle or Tommy bar (Fig 8.20)

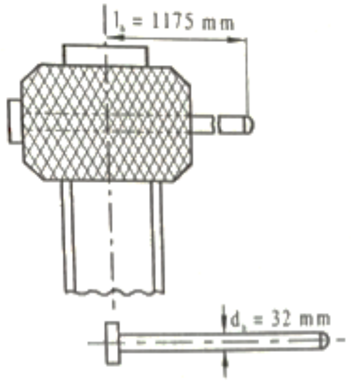


Fig. 8.20

Total frictional torque $M_t = M_{t_s} + M_{t_c} = W\left[\frac{d_2}{2}\left(\frac{\tan\alpha + \mu}{1 - \mu\tan\alpha}\right) + \frac{\mu_c d_c}{2}\right]$ ---- 18.29 (DDHB)

$$= 50 \times 10^3\left[\frac{33}{2}\left(\frac{0.05787 + 0.14}{1 - 0.14 \times 0.05787}\right) + \frac{0.147 \times 51}{2}\right] = 352 \times 10^3\ Nmm$$

Also $M_t = F \times l_h$

Assume the force applied on to the handle by a person is 300 N

$\therefore\ 352 \times 10^3 = 300 \times l_h$

$\therefore$ Effective length of handle $l_h = 1173.33\ mm \approx 1175\ mm$

Also $M_b = F \times l_h = 352 \times 10^3\ Nmm$

We have $\frac{M_b}{I} = \frac{\sigma_b}{c}$ where $I = \frac{\pi}{64} d_h^4;\ c = \frac{d_h}{2}$

Assume the handle material is C40 steel $\therefore\ \sigma_b = 109.53\ N/mm^2$

$$\therefore\ \frac{352 \times 10^3}{\frac{\pi}{64} d_h^4} = \frac{109.53}{\frac{d_h}{2}}$$

$\therefore$ Diameter of handle $d_h = 31.98\ mm \approx 32\ mm$

vii. Design of cup (Fig 8.21)

Diameter of cup at bottom $D_3 \approx d_{co} = 72\ mm$

Diameter of cup at top $D_4 \approx 2d_{c_o} = 2 \times 72 = 144\ mm$

Height of cup $H_2 \approx 60$ mm

Thickness of cup $t \approx 10$ mm

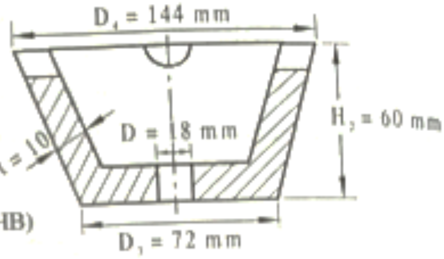


Fig. 8.21

viii. Efficiency of the screw

$$\eta = \frac{d_2 \tan\alpha}{\left(\frac{\tan\alpha + \mu}{1 - \mu\tan\alpha}\right) d_2 + \mu_c d_c} \quad \text{---- 18.33(DDHB)}$$

$$= \frac{33 \times 0.05787}{\left(\frac{0.05787 + 0.14}{1 - 0.14 \times 0.05787}\right) 33 + 0.147 \times 51} = 0.1367 = 13.67\%$$

ix. Check for overhauling

Condition for overhauling is.

$$\tan\alpha \geq \frac{\mu d_2 + \mu_c d_c}{d_2 - \mu\mu_c d_c} \quad \text{---- 18.37 (DDHB)}$$

$$\text{i.e., } 0.05787 \geq \frac{0.14 \times 33 + 0.147 \times 51}{33 - 0.14 \times 0.147 \times 51} \geq 0.3792$$

As 0.05787 is less than 0.3792, no overhauling. $\therefore$ It is a self locking screw.

x. Design of body (Fig 8.22)

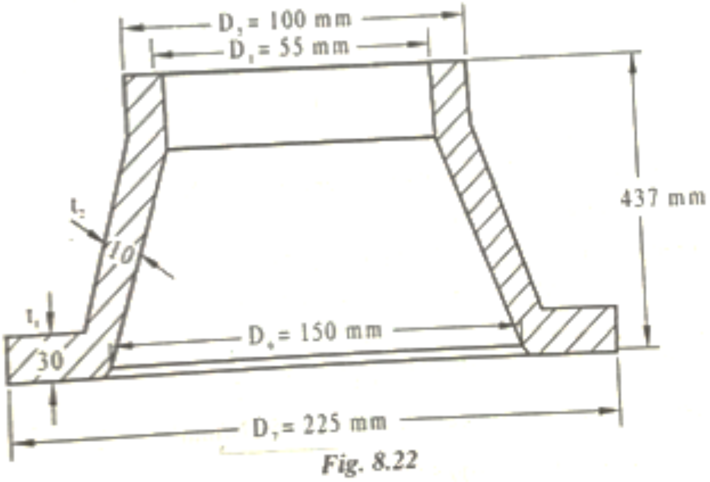


Fig. 8.22

(i) Length of screw = Lift + Length of nut + Margin = 300 + 72 + 15 = 387 mm

(ii) Height of body = Length of screw + Clearance = 387 + 50 = 437 mm

(iii) Diameter of body at the top $D_5 \approx 1.5\ D_2 = 1.5 \times 65 = 97.5$ mm $\approx$ 100 mm.

(iv) Diameter of body at the bottom inside $D_6 = 2.25\ D_2 = 2.25 \times 65 = 146.25$ mm $\approx$ 150 mm

(v) Outside diameter of body at the bottom $D_7 \approx 1.5\ D_6 = 1.5 \times 150 = 225$ mm.

(vi) Thickness of base $t_1 \approx 2H = 2 \times 15 = 30$ mm.

(vii) Thickness of body $t_2 \approx 0.25\ d = 0.25 \times 36 = 9$ mm $\approx$ 10 m

2. Flange coupling

$N = 18 kW \quad n = 1440 rpm$

$\tau_f = 47 MPa \; ; \quad \tau_s = \tau_k = \tau_b = 93 MPa$

$\sigma'_{bk} = 186 MPa.$

1. Maximum torque transmitted by coupling

$$M_{t_{max}} = 9550 \times 1000 \times \frac{18}{1440} =$$

$$M_{t_{max}} = 119.375 \times 10^3 \; N\text{-}mm$$

2. Dia of shaft

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

Assume $\eta = 0.75$

$$119.375 \times 10^3 = \frac{\pi}{16} \times 93 \times d^3 \times 0.75$$

$$d = 20.58 mm$$

Adopt standard dia = 22 mm.

3. Bolt circle dia

$D_1 = 2d + 50 = 94 mm$ (19.12b Eqn)

4. Design of hub

(i) Hub diameter $D_2 = 1.5d + 25$ Eqn 19.13b

$= 58 mm$

(ii) Length of Hub $l = 1.25d + 18.75$ Eqn 19.14d

$= 46.25 mm$

5. Design of Flange

(i) Outer dia of Flange $D = 2.5d + 75$ Eqn 19.13b

$= 130mm$

(ii) Thickness of Flange $t = 0.5d$

$= 11mm$

6. Check for Flange.

$$M_t = t(\pi D_2)\tau_f \frac{D_2}{2} \quad (\text{Eqn } 19.16)$$

$$119.375 \times 10^3 = 11(\pi \times 58) \times \tau_f \times \frac{58}{2}$$

$\tau_{f_{ind}} = 2.053\ N/mm^2 < 4\ N/mm^2$ Design safe

6. Design of bolts.

(i) No: of Bolts $i = 0.02d + 3$

$i = 3.44 \simeq 4$ bolts

(ii) Dia of Bolt

$$M_t = i\left\{\frac{\pi d_1^2}{4}\right\} \cdot \tau_b \left\{\frac{D_1}{2}\right\}$$

$$119.375 \times 10^3 = 4\left\{\pi \frac{d_1^2}{4}\right\} \times 93 \times \left\{\frac{94}{2}\right\}$$

$d_1 = 2.94$ mm.

Adopt standard dia. 3 mm x 0.5 pitch.

Check for the bolt

Crushing strength of Bolt

$$M_{t_{max}} = i\, d_1 t\, (\sigma_c)_b \frac{D_1}{2}$$

$$119.375 \times 10^3 = 4 \times 3 \times 11 \times (\sigma_c)_b \times \frac{94}{2}$$

$(\sigma_c)_{b_{ind}} = 19.24\ N/mm^2 < 186\ N/mm^2$

Design Safe

7. Design of key:

For $d = 22mm$ ~~and hubs~~

Key dimensions $b = 10mm$ (Taper key)

$h = 8mm$

For $b = 10mm$ and $h = 8mm$, Choose the prefered length

$l = 50mm$ (Choose $l >$ Hub length)

Check for shear strength

$$\tau_{d_2 (ind)} = \frac{2 M_t}{b \cdot l \cdot d} = \frac{2 \times 119.375 \times 10^3}{10 \times 50 \times 22}$$

$$\tau_{d_2 (ind)} = 21.70\ N/mm^2 < 93\ N/mm^2$$

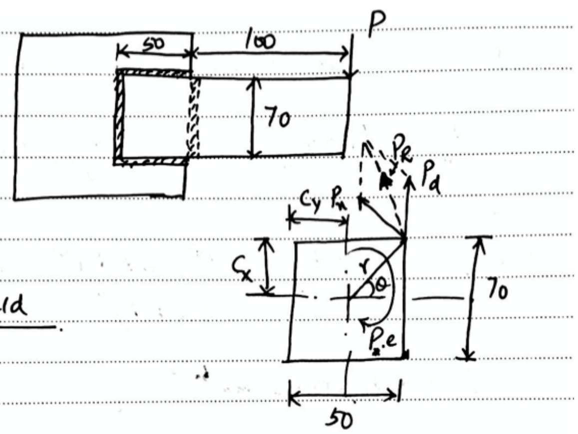
Safe design

Check for Crushing strength

$$\sigma_b' = \frac{4 M_t}{h \cdot l \cdot d} = \frac{4 \times 119.375 \times 10^3}{8 \times 50 \times 22}$$

$$\sigma_{b (ind)}' = 54.26\ N/mm^2 < 186\ N/mm^2$$

3.



Soln

1. Find Cg of weld

Table 12.3.

$$C_x = \frac{70}{2} = 35mm$$

$$C_y = \frac{50}{2} = 25mm$$

2. To Find r, e, $\cos\theta$

$$e = 100 + 25 = 125\,mm$$

$$r = \sqrt{25^2 + 35^2} = 43.01\,mm$$

$$\cos\theta = \frac{25}{43.01} = 0.581$$

3. Direct shear load per unit length of the weld

$$P_d = \frac{P}{l} = \frac{3000}{240} = 12.5\,N/mm$$

$$l = 70+70+50+50 = 240\,mm$$

4. Load due to torsional moment per unit length of weld

$$P_n = \frac{P \cdot e \cdot r}{J} \longrightarrow \text{Eqn } 12.20.$$

$$J = J_w = \frac{(b+d)^3}{6} = \frac{(50+70)^3}{6}$$

Table 12.3 DDB

$$J = 288000\,mm^3.$$

$$P_n = \frac{3000 \times 125 \times 43.01}{288000} = 56\,N/mm.$$

5. Resultant load per unit length of weld

$$P_R = \sqrt{P_d^2 + P_n^2 + 2P_d \cdot P_n \cos\theta}$$

$$P_R = 64.075\,N/mm$$

6. Size of the weld

$$\text{Allowable shear stress } \tau = \frac{P_R}{0.707\,w\,l}$$

$$85 = \frac{\cancel{3000}\;64.075}{0.707 \times w \times 1}$$

$$\Rightarrow w = 1.066\,mm \simeq 1.5\,mm$$