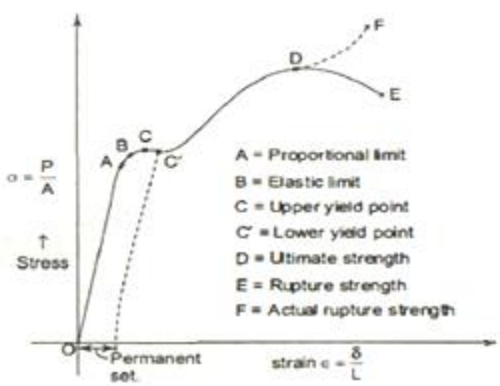
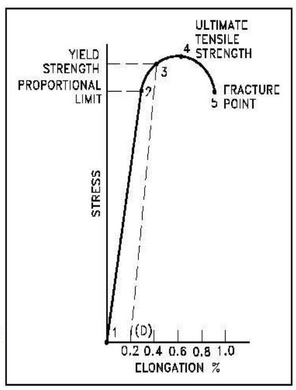
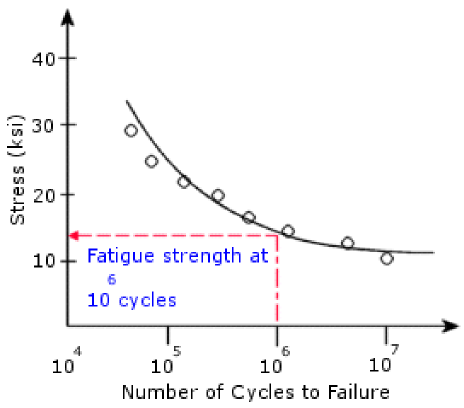
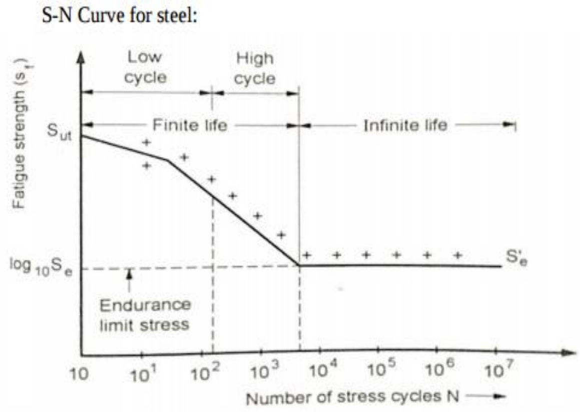
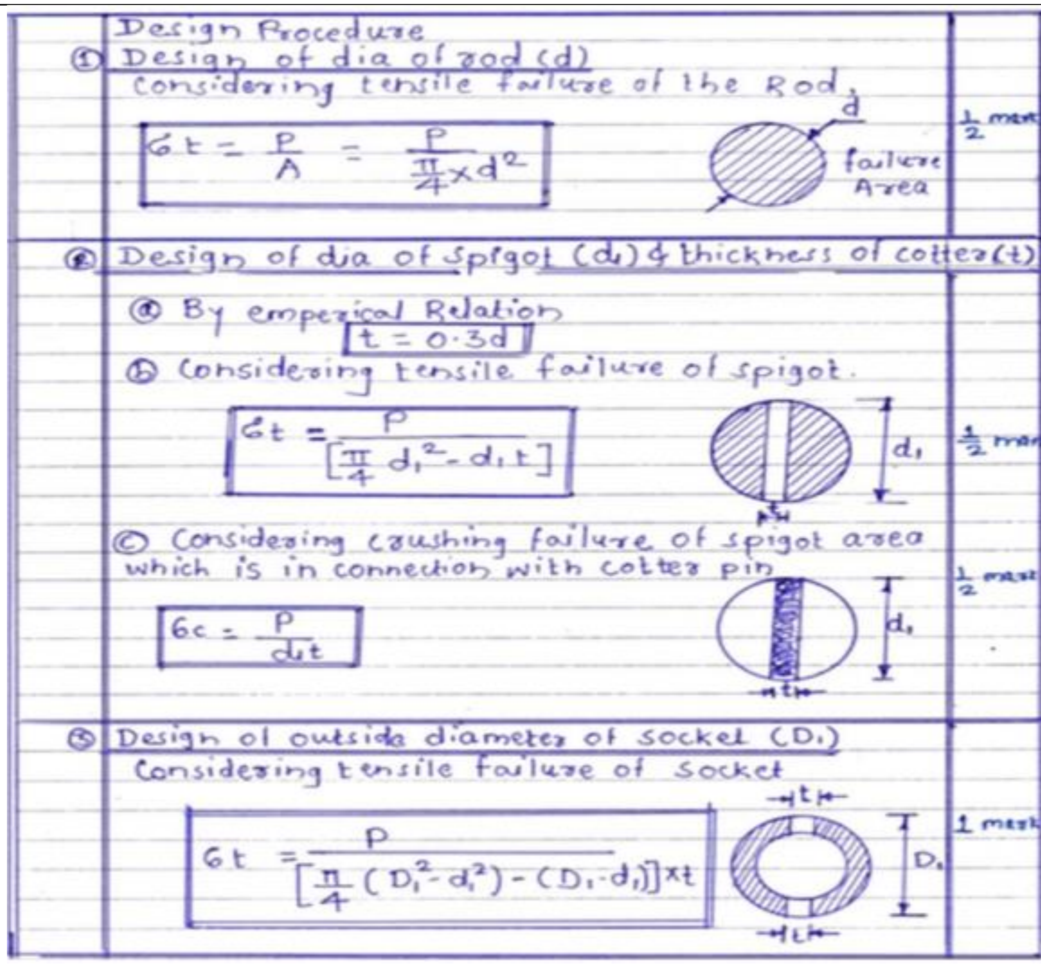
WINTER– 19 EXAMINATION

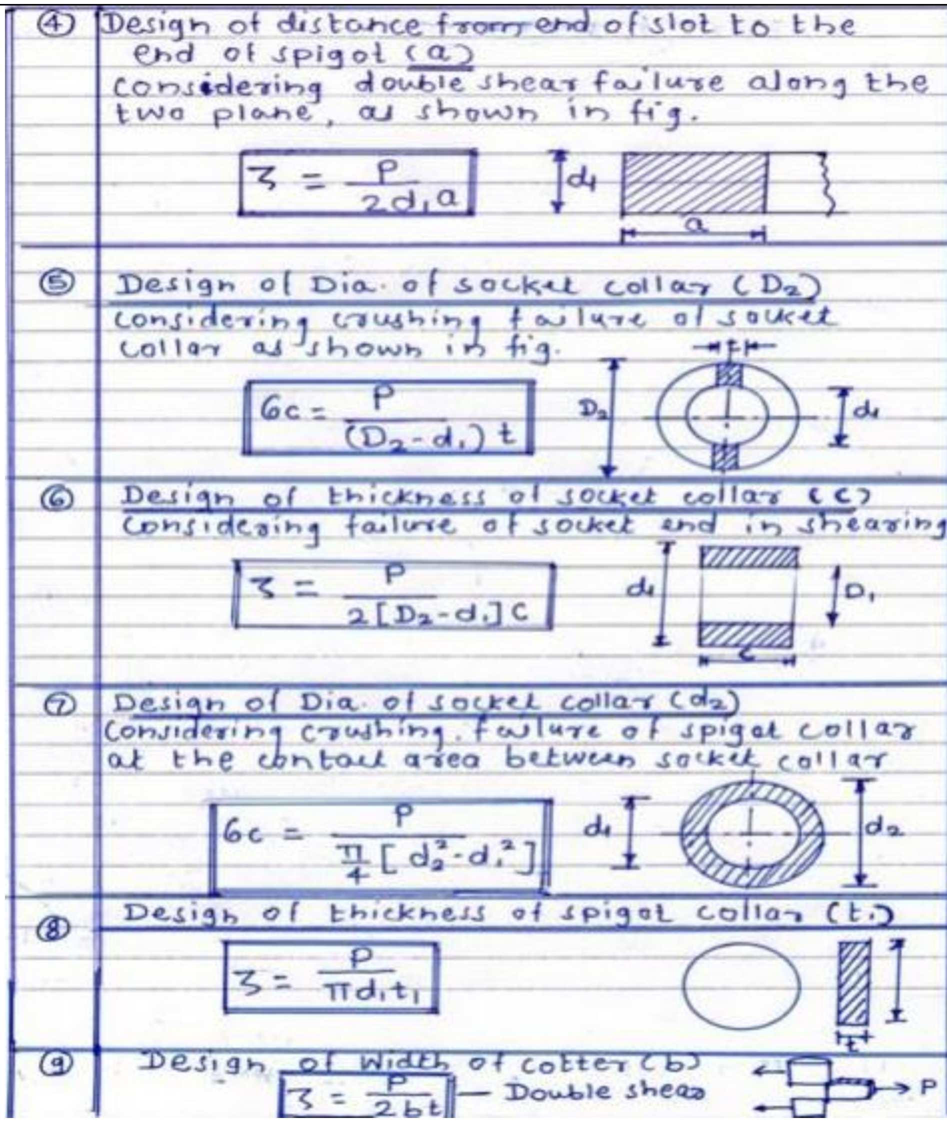
SubjectName:DesignofMachine Elements **Model Answer** **SubjectCode: 17610**

Important Instructions to the examiners:

1) Theanswersshouldbeexaminedbykeywordsandnotasword-to-wordasgiven inthemodelanswer scheme.
2) Themodelanswerandtheanswerwrittenbycandidatemayvarybuttheexaminermay trytoassessthe understandinglevel ofthe candidate.
3) Thelanguageerrorssuchasgrammatical,spellingerrorsshouldnotbegivenmoreImportance(Not applicable forsubjectEnglish andCommunication Skills.
4) Whileassessingfigures,examinermaygivecreditforprincipalcomponentsindicatedinthefigure.The figures drawn bycandidate andmodel answermayvary. Theexaminer maygive credit foranyequivalent figure drawn.
5) Creditsmaybegivenstepwisefornumericalproblems.Insomecases,theassumedconstantvalues may varyand theremaybesomedifferenceinthe candidate'sanswersand model answer.
6) Incaseofsomequestionscreditmaybegivenbyjudgementonpartofexaminerofrelevantanswer basedoncandidate'sunderstanding.
7) Forprogramminglanguagepapers,creditmaybegiventoanyotherprogrambasedonequivalent concept.

Q.1.(A)		Attempt any **THREE** of the following: (3X4)	12 Marks
	a)	**Draw stress-strain diagram for i) ductile material ii) brittle material**	
	Ans	  **Figure:** Stress- Strain diagram for ductile material **Figure:** Stress- Strain diagram for Brittle material	**02 marks for each diagram**
	b)	**Define endurance or fatigue limit and draw S-N curve for the steel.**	
	Ans	Endurance strength is defined as the maximum value of completely reversed bending stress that a material can withstand for a finite number of cycles without a fatigue failure. Endurance limit, Se, for the stress below which failure never occurs, even for an indefinitely large number of loading cycles, as in the case of steel; and fatigue limit or fatigue strength, Sf, for the stress at which failure occurs after a specified number of loading cycles, such as 500 million,	**02 marks for definition** **02 marks for**

		 **OR**  	**any one diagram**
	c)	**Write the design procedure for the socket and spigot joint with the strength equations (any four) with neat sketches.**	
	Ans	 Design Procedure ① Design of dia of rod (d) Considering tensile failure of the Rod, $\sigma_t = \frac{P}{A} = \frac{P}{\frac{\pi}{4} \times d^2}$ d, failure Area — ½ mark ② Design of dia of Spigot (d_1) & thickness of cotter (t) ⓐ By emperical Relation $t = 0.3d$ ⓑ Considering tensile failure of spigot. $\sigma_t = \frac{P}{[\frac{\pi}{4} d_1^2 - d_1 t]}$ d_1, t — ½ mark ⓒ Considering crushing failure of spigot area which is in connection with cotter pin $\sigma_c = \frac{P}{d_1 t}$ d_1, t — ½ mark ③ Design of outside diameter of socket (D_1) Considering tensile failure of socket $\sigma_t = \frac{P}{[\frac{\pi}{4}(D_1^2 - d_1^2) - (D_1 - d_1)] \times t}$ t, D_1 — 1 mark	**01 mark each with sketches any 04**



④ Design of distance from end of slot to the end of spigot (a)
Considering double shear failure along the two plane, as shown in fig.

$$\tau = \frac{P}{2 d_1 a}$$

⑤ Design of Dia. of socket collar (D_2)
Considering crushing failure of socket collar as shown in fig.

$$\sigma_c = \frac{P}{(D_2 - d_1) t}$$

⑥ Design of thickness of socket collar (c)
Considering failure of socket end in shearing

$$\tau = \frac{P}{2[D_2 - d_1] c}$$

⑦ Design of Dia of socket collar (d_2)
Considering crushing failure of spigot collar at the contact area between socket collar

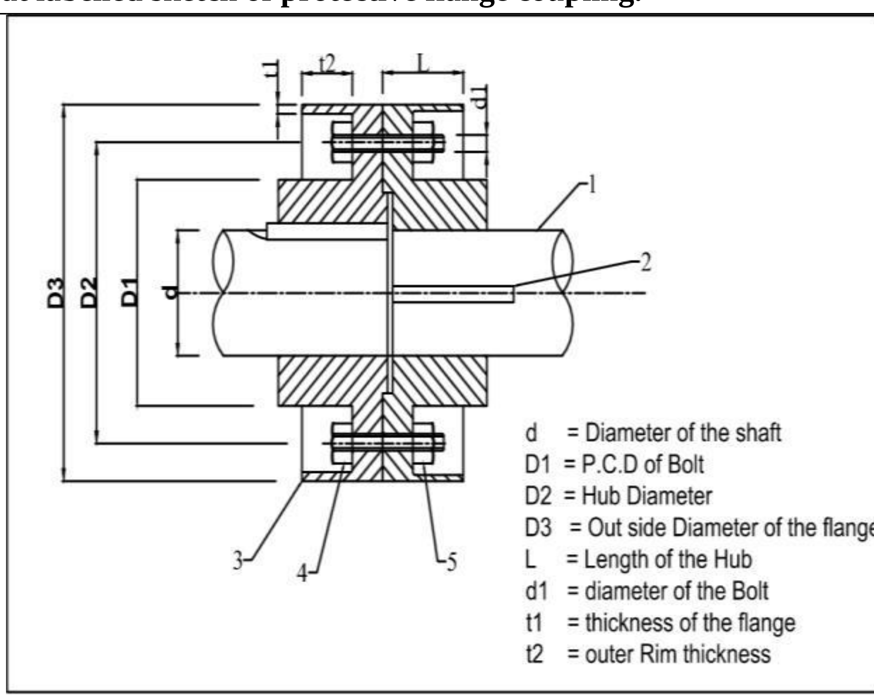
$$\sigma_c = \frac{P}{\frac{\pi}{4}[d_2^2 - d_1^2]}$$

⑧ Design of thickness of spigot collar (t_1)

$$\tau = \frac{P}{\pi d_1 t_1}$$

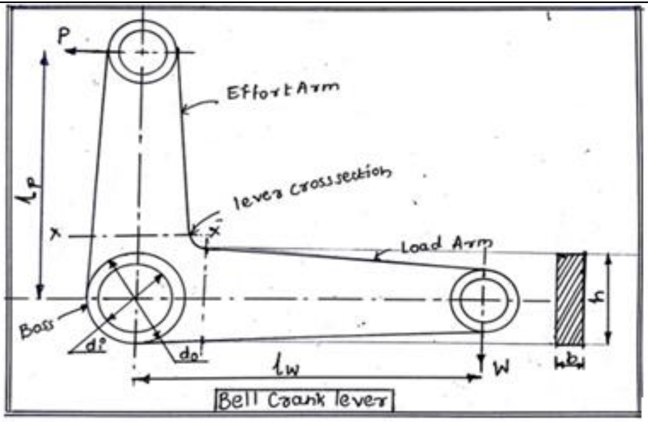
⑨ Design of width of cotter (b)

$$\tau = \frac{P}{2bt}$$ — Double shear

	d)	**Draw a neat labelled sketch of protective flange coupling.**	
	Ans		**03 marks for diagram and** **01 mark for labelling**

Q.1.(B)		Attempt any ONE of the following: (1X6)	06 Marks
	a)	**Write the general procedure of bell crank lever.**	

Ans



Where,
P=Effort ,W=Load
l_w= Length of load arm, l_p= Length of effort arm,
R_f= Fulcrum Reaction ,d= Diameter of pin
l_p= Length of fulcrum pin= 1.25dl_b= Length of boss= 1.25d
P_b= Bearing Pressure d_o= Outer diameter of boss
d_i= Inside diameter of boss
Consider a brass bush of 3mm thickness is fit into the boss
d_i= d+(3×2)
d_i=d+6
M=Bending Moment
b= Width of lever cross-section
h= Depth of lever cross-section

01 mark for diagram and

01 mark for labelling

Design Procedure

① Determination of effort (P)

$$W \times l_w = P \times l_p$$

② Determination of fulcrum reaction (Rf)

$$R_F = \sqrt{W^2 + P^2}$$

01 mark

③ Design of fulcrum pin.

$$l_p = l_b = 1.25\,d$$

ⓐ Considering bearing pressure at fulcrum pin

$$P_b = \frac{R_F}{l_b \times d}$$

01 mark

ⓑ Considering double shear failure of pin

$$\tau = \frac{R_F}{2 \times \frac{\pi}{4} \times d^2}$$ → Checking of value of 'τ'

④ Design of boss of lever

$d_i = d+6$ — emperical relation

considering bending stress acting on the boss.

$$\sigma_b = \frac{M \cdot Y}{I_{xx}}$$

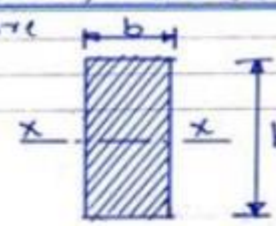
Where,
• $M = l_p \times P$, • $Y = \frac{d_o}{2}$
• $I_{xx} = \frac{1}{12} \times l_b \times [d_o^3 - d_i^3]$

01 mark

⑤ Design of lever arm crossection near to boss

considering bending failure

$$\sigma_b = \frac{M \cdot Y}{I_{xx}}$$



• $M = P \times [l_p - \frac{d_o}{2}]$
• $Y = \frac{h}{2}$
• $I_{xx} = \frac{1}{12} \cdot bh^3$

01 marks

	b)	**Determine the diameter of hollow shaft having inside diameter 0.6 of outside diameter. The shaft is driven by 900 mm overhung pulley placed vertically. The weight of the pulley is 600 N. The overhung is 250 mm and the tensions in tight and slack side are 2900 N and 1000 N respectively. Assume Fs = 80 N/mm^2.**	
	Ans	$T = (T_1-T_2)XR = (2900-1000)X900/2 = 855000$ N-mm Total vertical load acting on the pulley $W_v = T_1+T_2+$weight of pulley $=2900+1000+600 = 4500N$ B.M. $M = W_v x l = 4500X250 = 112500$ Nmm Equivalent twisting moment $Te = (M^2+T^2)^{0.5}$ $= [(112500)^2+ (855000)^{2]0.5}$ $=862369.55$ Nmm $Te = \pi /16\ Fs\ do^3 (1-k^4)$ $862369.55 = \pi /16 \times 85 \times do^3(1-0.6^4)$ **do= 39.01 mm say 40 mm and di=24 mm**	**02 marks** **02 marks** **02 marks**
Q.2.		**Attempt any <u>TWO</u> of the following: (2X8)**	**16 Marks**
	a)	**Design a knuckle joint to transmit 150kN. The design stresses are σ_t = 75 MPa, σ_c= 150 MPa, τ_{shear} =60 MPa.**	
	Ans	Given : $P = 150\ kN = 150 \times 10^3$ N $\sigma_t = 75$ MPa $= 75$ N/mm^2 , $\tau = 60$ MPa $= 60$ N/mm^2 $\sigma_c = 150$ MPa $= 150$ N/mm^2 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), $P= \pi /4\ d^2 x\ \sigma_t$ $d^2 = 150 \times 10^3 / 59 = 2540$ d = 50.4 say 52 mm Now the various dimensions are fixed as follows: **Diameter of knuckle pin, d_1 = d = 52 mm** **Outer diameter of eye, d_2 = 2d = 2 × 52= 104 mm** **Diameter of knuckle pin head and collar, d_3 = 1.5d = 1.5 × 52 = 78 mm** **Thickness of single eye or rod end, t= 1.25 d =1.25 × 52 =65 mm** **Thickness of fork,t_1 = 0.75 d= 0.75 × 52= 39 say 40 mm**	

		Thickness of pin head,t_2 = 0.5 d= 0.5 × 52= 26 mm	**01 marks**
		2. Failure of the knuckle pin in shear	
		Since the knuckle pin is in double shear,	
		therefore load (P),= 150 × 10^3 / 4248=35.3 N/mm^2 = 35.3 MPa	
		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 × 10^3 = (d_2 – d_1) t × σ_t= (104 – 52) 65 × σ_t=3380 σ_t	
		σ_t = 150 × 103 / 3380= 44.4 N / mm^2 = 44.4 MPa	**01 mark**
		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 × 10^3 = (d_2 – d_1) t × τ =(104 – 52) 65 × τ=3380 τ= 150 × 103 / 3380=44.4 N/mm^2 = 44.4 MPa	**01 mark**
		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 × 10^3 = d_1 × t × σ_c=52 × 65 × σc=3380 σ_c	
		σ_c = 150 × 103 / 3380=44.4 N/mm^2 = 44.4 MPa	**01 mark**
		Failure of the forked end in tension	
		The forked end may fail in tension due to the load. We know that	
		load (P), 150 × 10^3 = (d_2 – d_1) 2 t_1 × σ_t= (104 – 52) 2 × 40 × σ_t=4160 σ_t	
		σ_t = 150 × 103 / 4160=36 N/mm^2 = 36 MPa	**01 mark**
		Failure of the forked end in shear	
		The forked end may fail in shearing due to the load. We know that	
		load (P), 150 × 10^3 = d_1 × t × σ_c=52 × 65 × σc=3380 σ_c	**01 mark**
		σ_c = 150 × 103 / 3380=44.4 N/mm2 = 44.4 MPa	
		Failure of the forked end in tension	
		The forked end may fail in tension due to the load. We know that	
		load (P), 150 × 10^3 = (d_2 – d_1) 2 t_1 × σ_t=(104 – 52) 2 × 40 × σ_t=4160 σ_t	
		σ_t = 150 × 103 / 4160=36 N/mm^2 = 36 MPa	
		Failure of the forked end in shear	
		The forked end may fail in shearing due to the load. We know that	**01 mark**

		load (P),150×10^3 = $(d2 – d_1)\ 2\ t_1 \times\ \tau$=(104 – 52) 2 × 40 × τ =4160 τ= 150 × 10^3 / 4160=36 N/mm^2 = 36 MPa **Failure of the forked end in crushing** The forked end may fail in crushing due to the load. We know that load (P), 150 × 10^3 = $d_1 \times 2t_1 \times \sigma_c$=52 × 2 × 40 × σ_c=4160 σ_c σ_c = 150 × 103 / 4180= 36 N/mm^2 = 36 MPa From above, we see that the induced stresses are less than the given design stresses, therefore the joint is safe.	**01 mark**
	b)	**Compare the weight, strength and stiffness of a hollow shaft of the same external diameter as that of solid shaft. Inside diameter of hollow shaft is half of the external diameter. Both shafts have the same material & length.**	
	Ans	**Comparison of weight** We know that weight of a hollow shaft, W_H= Cross-sectional area × Length × Density=π/4$(d_0)^2$-$(di)^2$× Length × Density...(i) and weight of the solid shaft, W_S=π/4 d^2x Length × Density..(ii) Since both the shafts have the same material and length, therefore by dividing equation (i) by equation (ii), we get W_H/W_S= $(d0)^2$-$(di)^2$/d^2 = 1 – k^2 = 1 – $(0.5)^2$ = 0.75 Ans. **Comparison of strength** We know that strength of the hollow shaft,T_H=)π/16×τd_0^3x(1-k^4)–...(iii)and strength of the solid shaft,T_S=)π/16×τ$d^{3---------}$iv) Dividing equation (iii) by equation (iv), we $T_{H/}$ T_S =)π/16×τ$d0^3$x(1-k^4)/ π/16×τd^3–...(iii) = 1 – $(0.5)^4$ = 0.9375	**04 marks** **04 marks**
	c)	**A bracket as shown in fig.no.1 is fixed to the wall by means of four bolts.Find the size of the bolts if σ_t = 70 N/mm^2 for bolt material.**	

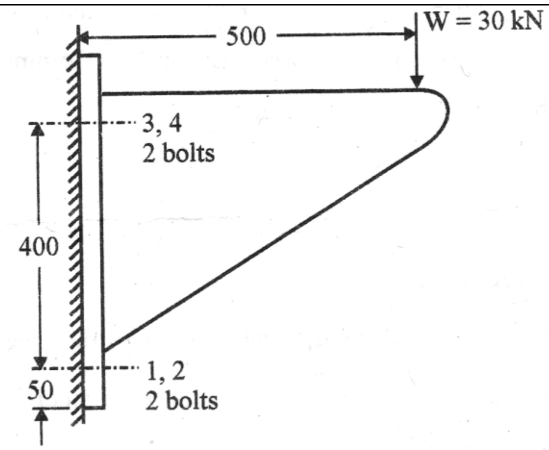


Fig. No. 1

Ans

Given : W = 30 kN = 30 × 10^3 N ;

L = 500 mm ; L_1 = 50 mm ; L_2 = 450 mm ; σ_t= 70 MPa = 70 N/mm^2 ; n = 4

We know that

Direct shear load on each bolt,**Ws=30/4=W/n= 30/4= 7.5 kN**

Since the load W will try to tilt the bracket in the clockwise direction about the lower edge, therefore the bolts will be subjected to tensile load due to turning moment.

The maximum loaded bolts are 3 and 4 because they lie at the greatest distance from the tilting edge A–A(i.e. lower edge). **02 marks**

We know that maximum tensile load carried by bolts 3 and 4,Wt=W.L.L_2/ [2(L_1)^2+(L_2)^2]

=30 × 10^3 X500X450/2[(50)^2+(450)^2] = **16463.41kN** **01mark**

Since the bolts are subjected to shear load as well as tensile load, therefore

equivalent tensile load, **Wte=1/2[Wt+(Wt)^2+4(Ws)^2]$^{0.5}$** **01 mark**

=1/2[16463.41+(16463.41)^2+4(7500)^2]$^{0.5}$ **01 mark**

= 19367.72 N

Size of the bolt Let dc= Core diameter of the bolt. **01 mark**

We know that the equivalent tensile load **(Wte), 19367.72 = π /4x(dc)^2**

= 19367.72 / 70 = 276.68

Or **dc = 16.63 mm**

From table **dc= 16.933 mm and the corresponding size of bolt is bolt is M 20.** **02 marks**

Q.3.	**Attempt any <u>FOUR</u> of the following: (4X4)**	**16 Marks**
a)	**Define factor of safety with respect to mild steel and cast iron.**	
Ans	While designing any mechanical component always there are certain areas of uncertainties such as variation and non uniformity in the mechanical strength etc. Hence in order to prevent failure of the component, designer assuming value of design	

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stress, which is very less as compared to the yield stress or ultimate stress.
So **factor of safety is defined as a ratio of maximum stress to working stress or design stress**.

i) **For ductile materials(Mild steel)**: The factor of safety is defined as the ratio of yield point stress to design stress.

$$factor\ of\ safety = \frac{Yield\ Stress}{Working\ or\ Design\ stress}$$

ii) **For brittle materials(Cast iron)**:The factor of safety is defined as the ratio of ultimate stress to design stress.

$$factor\ of\ safety = \frac{Ultimate\ Stress}{Working\ or\ Design\ stress}$$
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02 marks

02 marks
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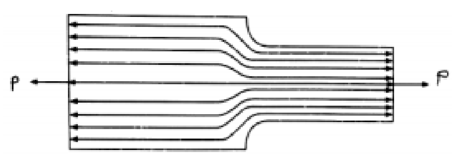
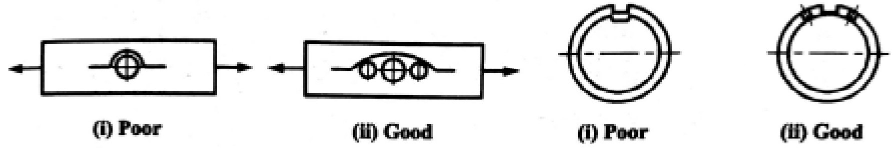
<tr><td></td><td>**b)**</td><td>**What is stress concentration? Illustrate methods to reduce it with sketches.**</td><td></td></tr>
<tr><td></td><td>**Ans**</td><td>
Stress concentration: The stresses induced in the neighborhood of the discontinuities like keyways, threaded grooves, holes, notches are much higher than the stresses in the other parts of the stressed component. This concentration of high stresses due to discontinuities and abrupt changes in cross section is called stress concentration.

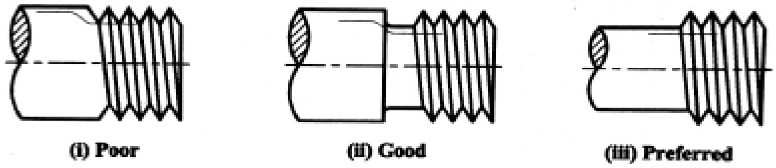


The presence of stresses concentration cannot be totally eliminated but it can be reduced, so following are the remedial measures to control the effects of stress concentration.

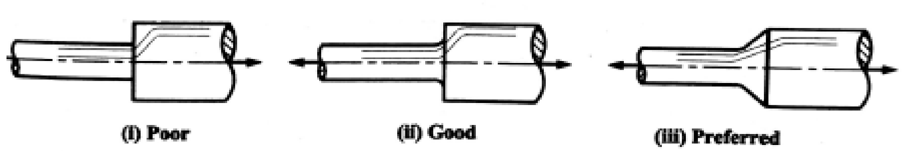
1. Provide additional notches and holes in tension members as shown in fig (a)
a)Use of multiple notches.
b)Drilling additional holes as shown in fig(b)
2. Fillet radius, undercutting and notch for member in bending.
3. Reduction of stress concentration in threaded members as shown infig(c)
4. Provide taper cross-section to the sharp corner of member as shown in fig(d)



(a) Tie rod with hole (b) Shaft with key way



(c) Threaded component



(d) Cylindrical component
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01 mark Definition

03 mark for 3 methods
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	c)	**State the following material specifications.** **(i) FeE 230 (ii) FG 200 (iii) 35C8 (iv) X20Cr18Ni12**	
	Ans	i) **FeE 230 -**steel(Steel having yield strength of 230 N/mm^2) with minimum tensile strength of 230 N/mm^2 **ii) FG 200-** Grey cast iron with minimum tensile strength of 200 N/mm^2 iii) **35C8** Means a carbon steel containing avg. percentage of carbon is 0.35 and avg. percentage of manganese is 0.8. iv) **X20Cr18Ni12 –**Means alloy steel with average percentage of carbon is 0.20 average percentage of chromium is 25 average percentage of nickel is 12	**01 mark each**
	d)	**State applications of maximum shear stress theory and principal normal stress theory.**	
	Ans	**Applications of maximum shear stress theory :** Designing the machine components made of ductile material. **Examples:** Crank shaft, Propeller shafts , springs, keys, **Applications of maximum principle normal stress theory :** Designing the machine components made of brittle material. **Examples:** spindle of Screw Jack, machine beds , c frames, overhang crank	**02 marks** **02 marks**
	e)	**What are the advantages and disadvantages of muff coupling (02 each) ?**	
	Ans	**Advantages :** ▪ It is simple, it has only two parts a sleeve and a key ▪ Since it has no projecting parts hence it is safe to use ▪ It has compact construction ▪ It is cheaper compared to other types of couplings **Disadvantages:** ▪ It is difficult to assemble or dismantle. ▪ Since it is a rigid coupling so it cannot accommodate any misalignment. ▪ Due to absence of flexible elements it cannot absorbs shocks and vibrations	**02 marks** **02 marks**
Q.4.(A)		**Attempt any THREE of the following: (3X4)**	**12 Marks**
	a)	**Write the equation with Wahl's factor, used for design of helical coil spring. State the SI units of each term in the equation.**	
	Ans	$$\tau = K\frac{8PD}{\pi d^3}$$ **Where τ =** shear strength of spring material in N/mm^2, **K**= Wahl's Stress Correction factor, **P**= Load on spring causing the deflection in N, **D**= Mean coil diameter of spring in mm, **d**= wire diameter of spring in mm.	**02 marks** **02 marks**
	b)	**A helical compression spring carries a load of 500 N with a deflection of 25 mm. The spring index may be taken as 8. Assume permissible =350 MPa. Modulus of rigidity N = 84 kN/mm. Wahl's factor as $\frac{4C-1}{4C-4}+\frac{0.615}{C}$ where C is spring index. Find the number**	

of active turns of spring.

Ans

Given : –

Axial load $P = 500\ N$, Deflection $\delta = 25\ mm$,

Spring Index $C = 8$, $\tau = 350\ N/mm^2$.

$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$, Modulus of rigidity $G = 84\ kN/mm^2$

Find — Number of active turns

I. Mean dia. of spring coil

Let D – Mean diameter of spring coil, and

d = Diameter of spring wire

We know that

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4(8)-1}{4(8)-4} + \frac{0.615}{8}$$

$$K = 1.1071 + 0.0768 = 1.184$$

$$K = 1.184$$

Maximum shear stress (τ)

$$\tau = K\frac{8PD}{\pi d^3} = K\frac{8PC}{\pi d^2} = 1.184 \times \frac{8 \times 500 \times 8}{\pi d^2}$$

$$d^2 = \frac{12058.56}{350} = 34.453$$

$$\boxed{d = 5.869 \text{ or } 6\ mm}$$

02 mark

II. ∴ Mean coil diameter = $C \times d = 8 \times 6 = 48\ mm$

II. Number of active turns (N) :

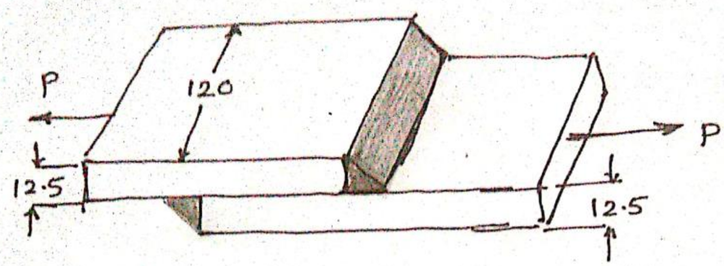
$$\delta = \frac{8PD^3N}{Gd^4} \quad \text{or} \quad 25 = \frac{8(500)(48)^3 N}{(84000)(6)^4}$$

$$\boxed{\therefore N = 6.15 \text{ or } 7 \text{ turns.}}$$ .... Ans

02 mark

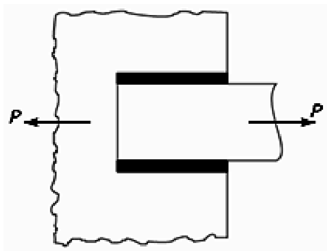
c) **A 45 mm diameter shaft is made of steel with yield strength of 400 N/mm^2. A key of size 14 mm wide and 9mm thick made of steel with yield strength of 340 N/mm^2 is to be used. Find the required length of key, if the shaft is loaded to transmit the maximum permissible torque. Use maximum shear stress theory and assume a factor of safety as 2.**

	Ans	Given :- diameter of shaft = 45 mm , S_{yt} = 400 N/mm² for shaft breadth or width of key b = 14 mm Height or depth of key d = 09 mm Yield strength for key S_{yt} = 340 N/mm² Let l = length of the key Maximum allowable shear stress for shaft is, $\tau_{max} = \frac{(S_{yt})_{shaft}}{2 \times f.s.} = \frac{(400)}{2 \times 2} = 100\ N/mm^2$ and maximum shear stress for key is. $\tau_{key} = \frac{(S_{yt})_{key}}{2 \times f.s.} = \frac{340}{2 \times 2} = 85\ N/mm^2$ Max^m torque transmitted by shaft and key $M_t = \frac{\pi}{16} \times \tau_{max} \times d^3 = \frac{\pi}{16} \times 100 (45)^3 = 1.8 \times 10^6\ N\text{-}mm$ Considering shear failure of key $\tau_{key} = \frac{2 M_t}{d b l} = \frac{2 \times 1.8 \times 10^6}{14 \times 45 \times l}$ $l = \frac{2 \times 1.8 \times 10^6}{45 \times 14 \times 85} = 67.2\ mm$ Considering crushing failure of key $\sigma_{cr} = \frac{4 M_t}{d h l} = \frac{4 \times 1.8 \times 10^6}{45 \times 9 \times l}$ let $\sigma_{cr} = \frac{(S_{yt})_{key}}{2} = \frac{340}{2} = 170\ N/mm^2$ $\therefore\ l = \frac{4 \times 1.8 \times 10^6}{45 \times 9 \times 170} = 104.6\ mm$ Selecting larger of two value of length, we have l = 104.6 say 105 mm ....Ans.	**01 mark** **1.5 Marks** **1.5 marks**
	d)	**Two steels plates 120 mm wide and 12.5 mm thick are to be connected together by double transverse filled weld. The maximum tensile stress for the plate and welding**	

		Material is not to exceed 70 N/mm^2. Find the length of weld required for maximum static loading.	
	Ans	Given: Width of plate = 120 mm, Thickness of plate = 12.5 Maximum tensile stress in plate & weld = 70 N/mm^2 Find length of weld (l) = ? 	**01 mark**
		Max^m load the plate can carry is P = Area × stress $= (120 \times 12.5) \times 70$ $P = 105000$ N	**01 mark**
		Load carried by double transverse fillet weld $P = 2(0.707 s \times l \times \sigma_t)$ $105000 = 2(0.707 \times 12.5 \times l \times 70)$ $l = \frac{105000}{1.414 \times 12.5 \times 70} = 84.86$ mm	**01 mark**
		$l = 84.86$ Adding 12.5 mm for starting and stopping of weld rum, we have $l = 84.86 + 12.5 = 97.36$ mm --- Ans.	**01 mark**
Q.4.(B)		**Attempt any ONE of the following: (1X6)**	**06 Marks**
	a)	**State the strength equation of double parallel fillet weld and double transverse fillet weldwith neat sketches.**	

Ans

01 mark for each figure



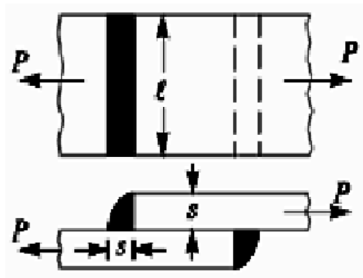


Figure: Double parallel fillet weld **Figure:** Double transverse fillet weld

i) Strength equation of double parallel fillet weld — **02 mark**

P =throat area x allowable shear stress

P =2 x0.707x Sx l x τ

=1.414x Sx l xτ

where S=size or leg of the weld, l=length of the weld, τ=shear stress

ii) Strength equation of double transverse fillet weld — **02 mark**

P= throat area x allowable tensile stress

P= 2 x 0.707x Sx lx σ_t

=1.414x Sx l xσ_t

where S=size or leg of the weld l=length of the weld σ_t=tensile stress

b) State and describe in brief any six ergonomics considerations in design of machine elements.

Ans — **01 mark each (any six considerations)**

- Ergonomics is defined as the scientific study of the man-machine-working environment relationship and the application of anatomical, physiological and psychological principles to solve the problems arising from the relationship.
- Ergonomics is related to the comfort between the man and machine while operating the machine.
- The objective of ergonomics is to make the machine fit for user rather than to make the user adapt himself or herself to the machine.
- From design consideration, the topics of ergonomics studies are as follows:

1. Anatomical factors in the design of driver's seat:

The design of driver's seat of an automobile is such that it is adjustable and comfortable to the end user.

2. Layout of instrument dials and display panels for accurate perception by the operators:

The basic objective behind the design of displays is to minimize the fatigue to the operator, who has to observe them continuously. The ergonomic considerations in the design of displays are as follows:

i) The scale on the dial indicator should be divided into suitable numerical divisions like 0-5-10-15 OR 0-10-20-30 and not 0-5-25-35
ii) The number of subdivisions between numbered divisions should be minimum.
iii) C. The size of letter or number on indicator is given as Height of letter or number ≥ $\frac{Reading\ distance}{200}$
iv) Vertical figures should be used for stationery dials, while radially oriented figures are used for rotating dials.
v) The pointer should have a knife edge with a mirror in the dial to minimize Parallex Error.

3. Design of hand levers and hand wheels:

The controls used to operate the machines consist of levers, hand wheels, knobs, switches, push buttons and pedals. Most of them are hand operated. When a large force is required to operate the controls, levers and hand wheels are used. When the operating forces are light, push buttons or knob are used. The ergonomic considerations in the design are as follows:

i) The controls should be easily accessible and logically positioned.
ii) The shape of the control component, which comes in contact with the hands, should be in conformity with anatomy of human hands.
iii) Proper colour produces beneficial psychological effects. The controls should be painted with grey background of machine tools to call for the attention.

4. Lighting, noise and climatic conditions in machine environment:

The working environment affect significantly the man-machine relationship. It affects the efficiency and possibly the health of the operator. The major working environmental factors are:

I. Lighting:

- The amount of light that is required to enable a task to be performed effectively depends upon the nature of the task, the cycle time, the reflective characteristics of the equipment involved and the vision of the operator.
- The intensity of light in the surrounding area should be less than that at the task area. This makes the task area the focus of attention.
- Operators will become less tired if the lighting and colour schemes are arranged so that there is a gradual change in brightness and colour from the task area to the surroundings. The task area should be located such that the operator can occasionally relax by looking away from the task area towards a distinct object or surface. The distinct object or surface should not be so bright that the operator's eyes takes time to adjust to the change when he or she again looks at the task.

II. Noise:

- The noise at the work place cause annoyance, damage to hearing and reduction of work efficiency. Noise caused by equipment that a person is using is less annoying than that caused by the equipment being used by another person, because the person has the option of stopping the noise caused by his own equipment. If the noise level is too high, it should be reduced at the source by maintenance, by the use of silencers and by placing vibrating equipment on isolating mounts. If required, ear plugs should be provided to the operators to reduce the effect of noise.

III. Temperature:

- For an operator to perform task efficiently, he should neither feel hot nor cold. When heavy work is done, the temperature should be relatively lower and when the light work is done, the temperature should be relatively higher.

IV. Humidity and Air circulation:

- At high temperatures, the low humidity may cause discomfort due to drying of throat and nose and high humidity may cause discomfort due to sensation of stuffiness and over sweating in a ill-ventilated or crowded room
- The proper air circulation is necessary to minimize the effect of high temperature and humidity.

Q.5.		**Attempt any <u>TWO</u> of the following: (2×8)**	**16 Marks**
	a)	**Explain self-locking and overhauling of power screw. State the reasons for using square threads over 'V' threads for power transmission.**	

Ans

Self-locking:

- The torque required to lower the load can be given by the equation,

  $T = W\ dm/2 x \tan(\phi - \alpha)$

- When ϕ is greater than or equal to α, a positive torque is required to lower the load. Under this condition, the load will not turn the screw and will not descend on its own unless an effort P is applied.
- Screw will be self-locking if the co-efficient of friction is equal to or greater than the tangent of the helix angle, the screw is said to be self-locking.
- A screw will be self-locking

  1)if the friction angle is greater than helix angle or coefficient of friction is greater than tangent of helix angle i.e μ or tan Ø > tan ά

  2) its efficiency is less than 50 % i.e $\eta < 50\%$

(03 marks)

Ii)Over hauling:

- The torque required to lower the load can be given by the equation,

  $T = W\ dm/2 x \tan(\phi - \alpha)$

- when $\phi < \alpha$ the torque required to lower the load is negative.
- It indicates a condition that no force is required to lower the load. The load itself will begin to turn the screw and descend down, unless a restraining torque is applied.
- The condition is called overhauling of the screw. This condition is also called back

		driving of screw. • A screw will be Overhauling: if the friction angle is less helix angle or coefficient of friction is less than tangent of helix angle. • i.e μ or tan Ø < tan ά its efficiency will be Greater than 50 % i.e η > 50% **Reason for using Square threads over V threads:** 1) It has maximum efficiency. 2) Ability to carry heavy loads. 3) Square threads are of self locking type 4) Minimum radial or brusting pressure on nut 5) High velocity ration Accuracy of motion.	**(03 marks)** **(02marks)** **Any 4 Reasons (1/2 mark each)**
	b)	**Design a close coiled helical spring for service load ranging from 2250 N to 2750 N, the axial deflection of the spring of the load range is 6 mm. Assume a spring index of 5. The permissible shear stress intensity is 420 N/mm² and modulus of rigidity, G=84 kN/mm². Take design stress 25% of permissible stress for severe condition and intermittent operation.**	
	Ans	Given: F min = 2250 N , F max = 2759 N, δ = 6 mm , C=5 , τ= 420 N/mm² , G= 84 X 10³ N/mm², **for severe condition and intermittent operation. Take design stress 25% excess of permissible stress** τ design= 1.25 X 420 N/mm² = 525 N/mm² Wahl's factor $K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\ 5C-1}{4X5\ -4} + \frac{0.615}{5} = 1.31$ **(1)Mean dia. Of the spring coil** Maximum shear stress, $\tau = Kx\frac{8\ FC}{\pi d^2}$, $525 = 1.31x\frac{8\ x2750\ x5}{\pi d^2}$ **d=9.34 mm say 10 mm** mean dia. Of the spring coil D= CXd =5 x 10=50 mm outer dia. Of the spring coil Do =D+d=50+ 10=60 mm **Step no 2-**Numbers of turns **(n) for 6 mm deflection load = (2750 -2250) =500** $\delta = \frac{8\times F\times D^3\times n}{G\times d^4}$, $6 = \frac{8\times 500\times 5^3\times n}{84\ X\ 10^3\times 10}$, n= 10.08 **n=10.08…. ..Say 11 numbers of turns** Assuming square and grounded ends, total numbers of turns is given by**,** **n'=n+2=11+2=13numbers of turns** **Step no 3-Solid length (Ls)** **Ls=n'×d= 13×10 =130 mm**	 **1 Mark** **2 Marks** **1 Mark** **1 Mark** **1 Mark**

Step no 3-Free length (Lf)

δ_{max} = **(2750 X 6) /500 =33 mm** — **1 Mark**

Lf=n'×d×+δ_{max}+0.15× δ_{max} = 130+ 33+(0.15x 33) = 167.95 mm

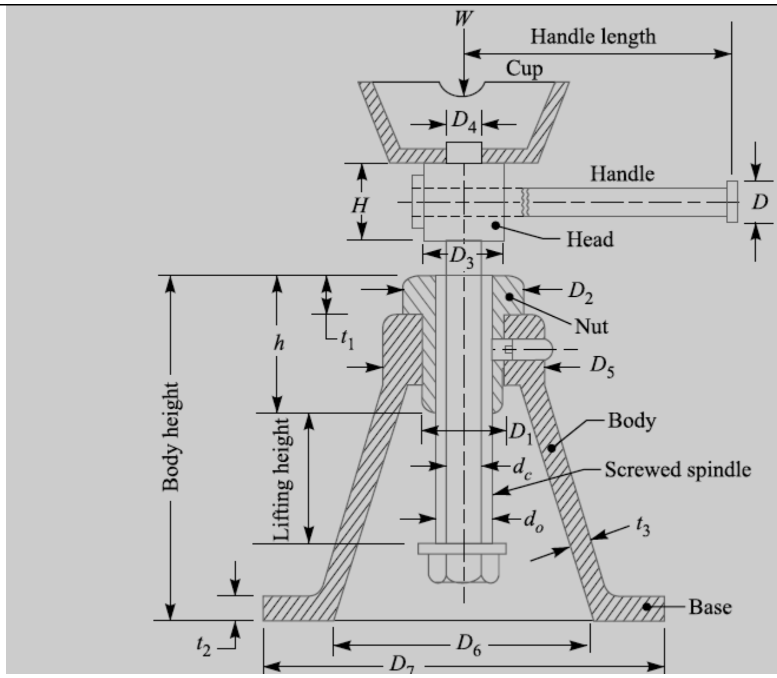
Lf=167.95**mm**

Step no 3-Pitch of the coil (p) — **1 Mark**

p = (Free length)/ (n'-1) = 167.95 /(13-1) = 13.99 mm **say 14mm**

c) Give the design procedure of screw and nut of a screw jack with the neat sketch.

Ans



Sketch 2 M

1. First of all, find the core diameter (*dc*) by considering that the screw is under pure compression,

$$W = \sigma_c \times A_c = \sigma_c \times \frac{\pi}{4}(d_c)^2$$

2. Find the torque (*T*1) required to rotate the screw and find the shear stress (τ) due to this torque.

We know that the torque required to lift the load,

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

1 Mark

P = Effort required at the circumference of the screw, and

d = Mean diameter of the screw.

∴ Shear stress due to torque *T*1,

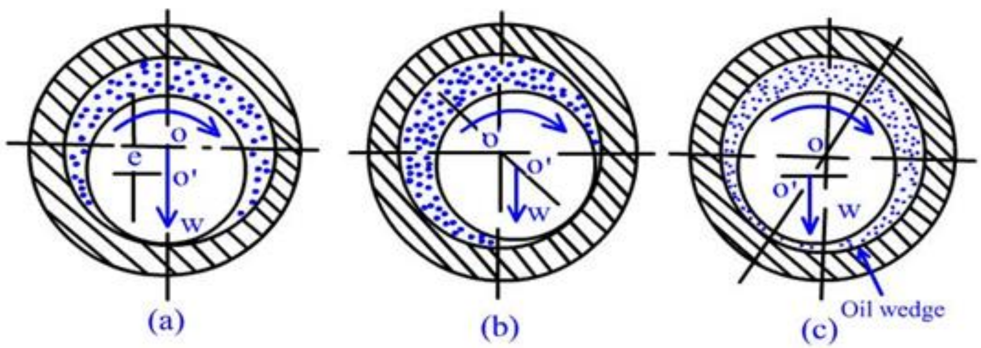
$$\tau = \frac{16\,T_1}{\pi (d_c)^3}$$

Also find direct compressive stress (σ*c*) due to axial load, *i.e.*

$$\sigma_c = \frac{W}{\frac{\pi}{4}(d_c)^2}$$

1 mark

		3. Find the principal stresses as follows: Maximum principal stress (tensile or compressive), $\sigma_{c(max)} = \frac{1}{2}\left[\sigma_c + \sqrt{(\sigma_c)^2 + 4\tau^2}\right]$ and maximum shear stress, $\tau_{max} = \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2}$ These stresses should be less than the permissible stresses. **4.** Find the height of nut (*h*), considering the bearing pressure on the nut. We know that the bearing pressure on the nut, $p_b = \frac{W}{\frac{\pi}{4}\left[(d_o)^2 - (d_c)^2\right] n}$ where*n* = Number of threads in contact with screwed spindle. ∴ Height of nut, $h = n \times p$ where*p* = Pitch of threads. **5.** Check the stressess in the screw and nut as follows : $\tau_{(screw)} = \frac{W}{\pi\, n.d_c.t}$ $\tau_{(nut)} = \frac{W}{\pi\, n.d_o.t}$ **6.** Find inner diameter (*D*1), outer diameter (*D*2) and thickness (*t*1) of the nut collar. The inner diameter (*D*1) is found by considering the tearing strength of the nut. We know That $W = \frac{\pi}{4}\left[(D_1)^2 - (d_o)^2\right]\sigma_t$ The outer diameter (*D*2) is found by considering the crushing strength of the nut collar. We know that $W = \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right]\sigma_c$ The thickness (*t*1) of the nut collar is found by considering the shearing strength of the nut collar. We know that $W = \pi\, D1.t1.\tau$	**1 Mark** **1 Mark** **1 Mark** **1 Mark**
Q.6.		**Attempt any FOUR of the following: (4× 4)**	**16 Marks**
	a)	**Explain gear tooth failures (i) Scoring (ii) Pitting**	
	Ans	i) **SCORING**: • Scoring is due to combination of two distinct activities: First, lubrication failure in the contact region and second, establishment of metal to metal contact. • Later on, welding and tearing action resulting from metallic contact removes the metal rapidly and continuously so far the load, speed and oil temperature remain at the same level. • The scoring is classified into initial, moderate and destructive. **ii)Pitting:**	**02 mark**

		• This is a major cause of gear failure accounting for nearly 60% of the gear failures. • Pitting is the formation of craters on the gear tooth surface. These craters are formed due to the high amount of compressive contact stresses in the gear surface occurring during transmission of the torque or in simple terms due to compressive fatigue on the gear tooth surface. • The pitting starts when total load acting on the gear tooth exceeds the wear strength of the gear tooth.	**02 mark**
	b)	**State any six design considerations while designing the spur gear.**	
	Ans	i) The power to be transmitted ii) The velocity ration or speed of gear drive. iii) The central distance between the two shafts iv) Input speed of the driving gear. v) Wear characteristics of the gear tooth for a long satisfactory life. vi) The use of space & material should be economical. **vii)** Efficiency & speed ratio **viii)** Cost	**Any four 01 mark Each**
	c)	**Explain the principle of working of hydrodynamic formal bearing with a neat sketch.**	
	Ans	 **Working principal :** in hydrodynamic bearing, the load supporting high pressure fluid film is created due to shape and relative motion between the two surfaces the moving surface pulls the lubricants into a wedge shaped zone at a velocity sufficiently high to create the high pressure film necessary to separate the two surfaces against the load. Fig a) initially when a shaft is at rest ,it makes contact with the bearing at its lowest point due to load W When the shaft start rotating in clockwise direction it will climb the bearing surface and contact is made at point as in fig (b) As the speed of the journal is further increased ,the lubrication is pulled into the wedge shaped region and forces the journal to the other side, as in fig c) Thus in the hydrodynamic bearing , it is not necessary to supply lubricant under pressure and only requirement is to ensure sufficient and conditions supply of lubricants	**02 mark** **02 mark**
	d)	**Give the classification of bearings.**	
	Ans	**Classification of bearing** **1.** ***Depending upon the direction of load to be supported.*** The bearings under this group are classified as: a) Radial bearings and ***(b)*** Thrust bearings.	**02 mark**

		2. Depending upon the nature of contact. The bearings under this group are classified as: **(*a*)** Sliding contact bearings, and **(*b*)** Rolling contact bearings	**02 mark**
	e)	**Write the design steps involved in selection of bearing from manufacturer's catalogue.**	
	Ans	**Procedure for selection of bearing from manufacturer's Catalogue.** 1) Calculate radial and axial forces and determine dia. of shaft. 2) Select proper type of bearing. 3) Start with extra light series for given diagram go by trial of error method. 4) Find value of basic static capacity (co) of selected bearing from catalogue. 5) Calculate ratios Fa/VFr and Fa/Co. 6) Calculate values of radial and thrust factors.(X & Y) from catalogue. 7) For given application find value of load factor Ka from catalogue. 8) Calculate equivalent dynamic load using relation. Pe = (XVFr + YFA) Ka. 9) Decide expected life of bearing considering application. Express life in million revolutions L10. 10) Calculate required basic dynamic capacity for bearing by relation. 11) Check whether selected bearing has req. dynamic capacity, IF it not select the bearing of next series and repeat procedure from step-4. **OR (flowchart**)	**Correct steps** **(04marks)** **OR** **Flow Chart**