17610

21718

4 Hours / 100 Marks Seat No.

Instructions : (1) ***All*** *questions are* ***compulsory****.*
(2) *Answer* ***each*** *next main question on a* ***new*** *page.*
(3) *Illustrate your answers with neat sketches* ***wherever*** *necessary.*
(4) *Figures to the* ***right*** *indicate* ***full*** *marks.*
(5) *Assume suitable data, if* ***necessary****.*
(6) *Use of Non-programmable Electronic Pocket Calculator is* ***permissible****.*

Marks

1. a) Attempt **any three** : **12**

a) Define Endurance or fatigue limit and draw typical S-N curve for steel.

b) Show that efficiency of square threaded power screw is never greater than 50%.

c) Define stress-concentration. Explain any 4 methods to reduce it with neat sketches.

d) Write general equation for :

i) Bending moment

ii) Torsion Equation and explain the various terms used in it.

b) Attempt **any one** : **6**

a) Explain ergonomics and aesthetics in automobile design.

b) Define :

i) Resilience

ii) Modulus of resilience. Show modulus of resilience on stress-strain diagram for ductile material.

2. Attempt **any two** : **16**

a) i) Why square thread is preferred over 'v' thread for power transmission ? **4**

ii) Differentiate key and cottor. Also explain why taper is provided on cottor. Give recommended value of taper. **4**

b) A helical spring is made from a wire of 8 mm diameter and has outside diameter 90 mm; if the permissible shear stress is 350 N/mm^2 and modulus of rigidity 84 kN/mm^2, find the axial load which the spring can carry and the deflection per active turn.

i) Neglecting the effect of curvature.

ii) Considering the effect of curvature.

P.T.O.

Marks

c) The pull in the tie rod of an iron roof truss is 50 kN. Design a suitable turn bucule (adjustable screwed joint). The permissible stress are 75 MPa in tension, 37.5 MPa in shear and 90 MPa in crushing. (Use the following data for ISO screw threads for screws, bolts and nut for course series).

Designation	Pitch (mm)	Major or nominal diameter Nut & Bolt d = D (mm)	Effective or pitch diameter Nut & Bolt (dp) mm	Minor or core diameter (dc) mm Bolt	Nut	Depth of thread (bolt) mm	Stress area (mm^2)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Course Series							
M36	4	36.000	33. 402	31.093	31.670	2.454	817
M39	4	39.000	36.402	34.093	34.670	2.454	976
M42	4.5	42.000	39.077	36.416	37.129	2.760	1104

3. Attempt **any four** : **16**

a) A hollow shaft is to be designed to transmit 600 kW at 110 rpm. The maximum torque being 20% greater than the mean. The shear stress is not to exceed 63 MPa and angle of twist in a length of 3 mts not to exceed 1.4 degree. Find external diameter of the shaft if the internal diameter to external diameter is 3/8. Take modulus of rigidity 84 GPa.

b) Prove that for a square key the permissible crushing stress is twise the permissible shear stress.

c) Give the composition of :

i) X_2O Cr18 Ni_2,

ii) $35C_8$

iii) Fe E 230

iv) FG 200

d) State and explain following theories :

i) Maximum principle stress theory.

ii) Maximum shear stress theory.

e) Differentiate between rolling contact and sliding contact bearing on the basis of :

i) Size

ii) Life

iii) Coefficient of friction

iv) Resistance to shock

Marks

4. a) Attempt **any three** : **12**

a) Write Lewis equation for the strength of gear tooth. Give the meaning of each term.

b) Define the terms :

i) Solid length

ii) Free length

iii) Spring index

iv) Pitch, w.r. to helical compression spring.

c) Draw symbolic representation of following types of weld :

i) Double V butt joint

ii) Double 'U' butt joint

iii) Single level butt

iv) Spot

d) A plate 100 mm wide and 10 mm thick is to be welded by another weld by means of double parallel fillet welds. The plates are subjected to a static load of 80 kN.

(Take τ permissible = 55 N/mm^2).

b) Attempt **any one** : **6**

a) Design a Knuckle joint to transmit 150 kN, the design stress are 75 MPa, 60 MPa and 150 MPa in tension, shear and compression respectively.

b) Explain the following modes of failure of gear tooth :

i) Pitting

ii) Scoring

iii) Abrasive wear

5. Attempt **any two** : **16**

a) A closed coil helical spring is used for front suspension of an automobile. The spring has stiffness 90 N/mm with square and ground ends. The load on the spring causes a total deflection of 8.5 mm. By taking permissible shear stress of material as 450 MPa. Find :

i) Spring wire diameter

ii) Length of spring

Assume spring index = 6 and G = 80×10^3 N/mm^2.

b) Give the design procedure of screw and nut of a screw jack.

c) Write the general design procedure of a flange coupling (unprotected type).

Marks

6. Attempt **any four** : **16**

a) Explain :

i) self locking

ii) overhauling of a power screw.

b) Explain with neat sketch, the bolt of uniform strength.

c) The spindle of a drilling machine is subjected to a maximum load of 10 kN. Determine the diameter of solid C.I. column of the machine, if tensile stress is limited to 40 N/mm^2. The distance between axis of spindle and axis of column is 330 mm; also find the direct stress and stress due to bending in the column. (Ref. Fig. No: 1)

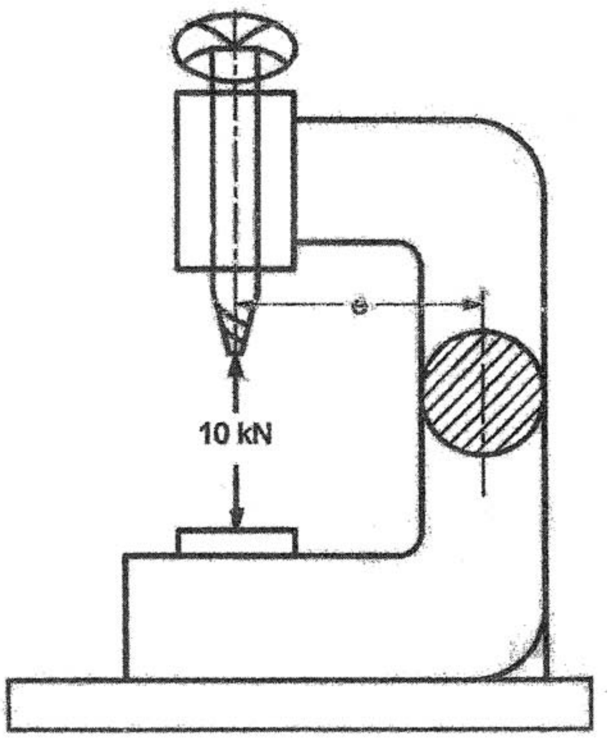


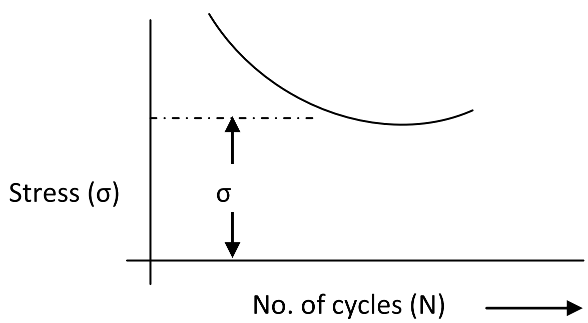
Fig. No. 1 [Q. No. 6(c)]

d) Write down the procedure for selection of bearing from manufacturer's catalogue.

e) State any four advantages and disadvantages of welded joints over screwed joint.

Important Instructions to examiners:

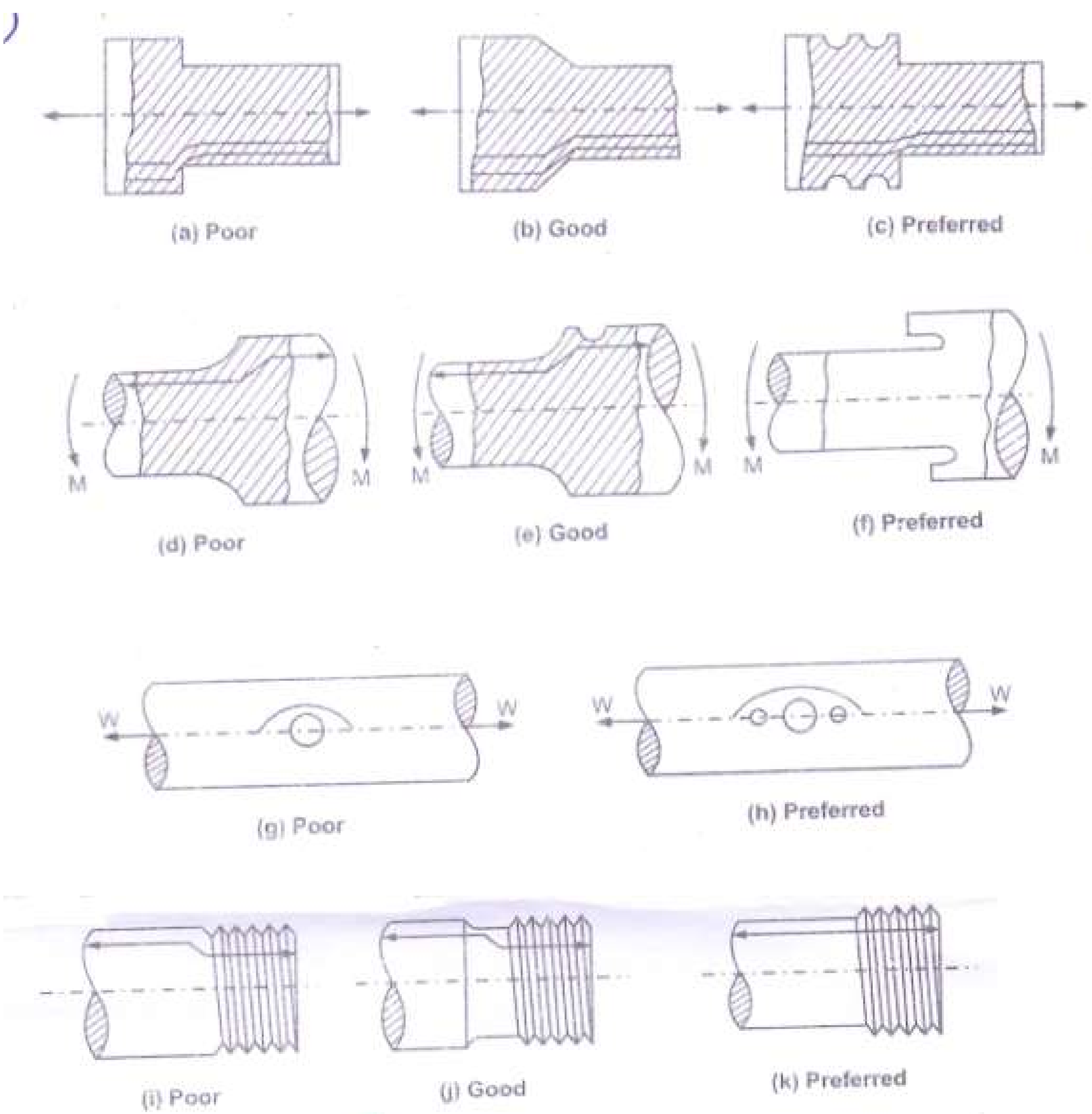
1) The answers should be examined by key words and not as word-to-word as given in themodel answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may tryto assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling errors should not be given moreImportance (Not applicable for subject English and Communication Skills.
4) While assessing figures, examiner may give credit for principal components indicated in thefigure. The figures drawn by candidate and model answer may vary. The examiner may give credit for anyequivalent figure drawn.
5) Credits may be given step wise for numerical problems. In some cases, the assumed constantvalues may vary and there may be some difference in the candidate's answers and model answer.
6) In case of some questions credit may be given by judgement on part of examiner of relevant answer based on candidate's understanding.
7) For programming language papers, credit may be given to any other program based on equivalent concept.

Q. 1	**a.**	**Attempt any THREE** Endurance limit or fatigue limit:- Maximum value of completely reversed bending stress, which a standard specimen can with store without failure, for infinite number of cycles of loads. 	2 Marks 2 Marks
	b.	Efficiency of Screw $\eta = \dfrac{\tan\alpha}{\tan(\alpha+\phi)}$ and for self locking screws, ø≥α OR α≤ø $\eta \le \dfrac{\tan\phi}{\tan(\phi+\phi)} \le \dfrac{\tan\phi}{\tan 2\phi}$ $\therefore$ Efficiency of self locking screws, $\le \dfrac{\tan\phi(1-\tan^2\phi)}{2\tan\phi}$ $\le \dfrac{1}{2} - \dfrac{\tan^2\phi}{2}$ from this expression efficiency of square threaded screw is never greater than 50%	1 Mark 1 Mark 1 Mark 1 Mark

c. Stress Concentration:- Whenever a machine component changes the shape of its cross-section the simple stress distribution no longer holds 900A. This irregularity in the stress distribution caused by abrupt changes of form is called as stress concentration. — 1 mark

Explanation:- To reduce stress concentration, various methods are adopted such as

i) By fillets, undercutting & notches
ii) Additional notches & holes
iii) Reducing stress-concentration in case of threaded members
iv) Cylindrical members with holes



3 marks

	d.	(i) Bending moment $\frac{M}{I}=\frac{\sigma b}{y}=\frac{E}{R}$ M = Bending moment acting at the given section I = M.I. of the cross section about the neutral axis σb= Bending stress y = Distance from the neutral axis to the extreme fibre E = Young's modulus of the material of the beam R = Radius of curvature of the beam	2 Marks
		(ii) Torsional equation $\frac{T}{J}=\frac{\tau}{r}=\frac{C\theta}{l}$ T = Torque or twisting moment J = Polar m.I τ = Torsinal shear stress or maximum shear stress r = Radius of the shaft c = Modulus of rigidity l = length of the shaft θ = Angle of twist in radians on a length l	2 Marks
	b)	**Attempt any ONE** (a) Explain ergonomics & aesthetics in automobile design Ergonomics in automobile design - - Anatomical factors in design of driver's seat - Layout of instrument dials & displays pannels for accurate perception by the operators - Design of hand levers & hand wheels - Lighting, noise and climatic conditions in machine environment. The purpose of applying ergonomics information to design situations is to ensure that, the environments provided and the design prepared offer the man the greatest comfort, advantages and safety.	3 Marks

- Designer should design the machine to suit the man's health, happiness and effectiveness.

Aesthetics in automobile Design -

The appearance should contribute to the performance of the product, thought the extent of contribution varies from product to product for example chromium plating of automobiles components improves the corrosion resistance along with the appearance. 3 Marks

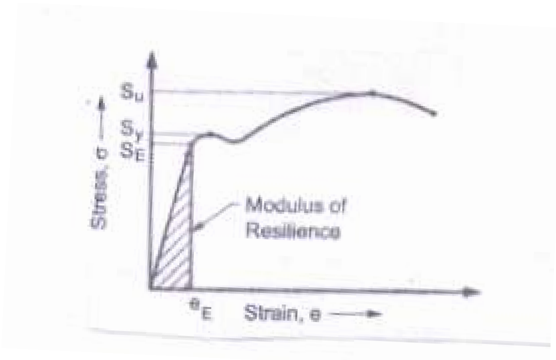
Similarly the aerodynamic shape of the car improve the performance the performance as well as gives the pleasing appearance lesser air resistance resulting in the lesser fuel consumption. The appearance should reflect the function of the product for example. The aerodynamic shape of the car creases the speed.

b) Define –

(b) i) Resilience : The property of a material to absorb strain energy while resisting shock or impact loads. This strain energy will be given up, when the load is removed. 2 Mark

The maximum stain energy stored in a member per unit volume, when loaded within elastic limit. 2 Mark

ii) Modulus of resilience – The maximum energy which can be stored in a body up to the elastic limit is called proof resilience. The proof resilience per unit volume of a maternal is known as modulus of resilience.



2 Marks

Q. 2

Attempt any TWO

a. (i) Why square trend is prepared over 'V' thread for power transmission. — 4 Marks

i) The efficiency of power screws, depends on the profile angle. The square thread how the greatest efficiency as profile angle is zero.
ii) They produce minimum bursting pressure on the nut.
iii) It can transmit power without any side thrust in either direction.
iv) More power transmission efficiency due to less friction.
v) Smooth and noiseless in operation.

(ii) Differentiate key land cotter. Also explain why taper is provided on cotter. Give recommended value of taper.

Key	Cotter
i) Key is driven parallel to the axis of shaft.	i) Cotter is normally driven at right angles to the axis of connected parts.
ii) Key is subjected to torsional shear stress & crushing stress.	ii) Cotter is subjected to crushing stress and shear stress.
iii) Key resists shear over longitudinal section.	iii) Cotter resists shear over transverse sections.

2 Marks

Any 2 points

Reasons – It helps the easy removal — 1 Mark

- Due to it, cutter remains in its position

- It provides maximum frictional area

Recommended valve of taper – 1 in 48 to 1 in 24 — 1 Mark

b. Mean diagram of the spring

$$D = Do - d = 90 - 8 = 82mm$$

$$\therefore Spring\ index\ C = \frac{D}{d} = \frac{82}{8} = 10.25mm$$

1 Mark

(i) Neglecting the effect of curvature

Shear stress factor

$$ks = 1 + \frac{1}{2c} = 1 + \frac{1}{2 \times 10.25} = 1.048$$

1 mark

And maximum shear stress induced in the wire (τ)

		$$\tau = ks \times \frac{8wD}{\pi d^3}$$ $$350 = 1.048 \times \frac{8 \times W \times 82}{\pi \times (8)^3}$$ $$W = \frac{350}{0.4273} = 819.096N$$ Deflection of the spring, $$\delta = \frac{8wD^3 n}{Gd^4}$$ ∴ Deflection per active turn $$\frac{\delta}{n} = \frac{8wD^3}{Gd^4} = \frac{8 \times 819.096 \times (82)^3}{84 \times 10^3 \times (8)^4}$$ $$\frac{\delta}{n} = \frac{3612986584}{344064000}$$ $$= 10.50mm$$	1 Mark 1 Mark
		Considering the effect of curvature Wahl's stress factor $k = \frac{4c-1}{4c-4} + \frac{0.615}{C}$ $$k = \frac{4 \times 10.25 - 1}{4 \times 10.25 - 4} + \frac{0.615}{10.25}$$ $$k = \frac{40}{37} + 0.06\ k = 1.141$$ Maximum shear stress induced in the wire (τ) $$\tau = k \times \frac{8 \times w \times C}{\pi d^2}$$ $$350 = \frac{1.141 \times 8 \times W \times 10.25}{\pi (8)^2}$$	1 Mark 1 Mark

$$W = \frac{70380.8}{93.562} = 752.24\ N$$

And deflection of the spring

$$\delta = \frac{8wD^3n}{ad^4}$$

1 Mark

$\therefore$ Deflection per active turn $\frac{\delta}{n} = \frac{8wD^3}{Gd^4} = \frac{8\times752.24\times(82)^3}{84\times10^3\times(8)^4}$

$$\frac{\delta}{n} = \frac{3318088512}{344064000} = 9.64\ mm$$

1 Mark

c.

Solution Given : $P = 50kN = 50 \times 10^3 N; \sigma_t = 75\ MPa = 75\ N/mm^2;\ \tau = 37.5\ MPa$

$$= 37.5\ N/mm^2$$

We know that the design load for the threaded section.

$$P_d = 1.3\ P = 1.3 \times 50 \times 10^3 = 65 \times 10^3 N$$

1 mark

An adjustable screwed joint, , is suitable for the given purpose. The various dimensions for the joint are determined as discussed below :

1. Diameter of the tie rod

Let $d = Diameter\ of\ the\ tie\ rod, and$

$d_c = Core\ diameter\ of\ threads\ on\ the\ tie\ rod.$

Considering tearing of the threads on the tie rod at their roots.

We know that design load (P_d),

$$65 \times 10^3 = \frac{\pi}{4}(d_c)^2\sigma_t = \frac{\pi}{4}(d_c)^2 75 = 59(d_c)^2$$

$$\therefore (d_c)^2 = 65 \times 10^3/59 = 1100\ or\ d_c = 33.2\ mm$$

From table for coarse series, we find that the standard core diameter is 34.093 mm and the corresponding nominal diameter of the threads or diameter of tie rod,

$$d = 39mm\ Ans.$$

1 Mark

2. Length of the coupler nut

Let $l = Length\ of\ the\ coupler\ nut.$

Considering the shearing of threads at their roots in the coupler nut. We know that design load (P_d),

$$65 \times 10^3 = (\pi d_c . l)\tau = \pi \times 34.093 \times l \times 37.5 = 4107\ l$$

$$\therefore l = 65 \times 10^3/4017 = 16.2mm$$

1 Mark

Since the length of the coupler nut is taken from d to 1:25 d, therefore we shall take

$$l = d = 39\ mm\ Ans.$$

We shall now check the length of the coupler nut for crushing of threads.

From table for coarse series, we find that the pitch of the threads is 4 mm. Therefore the number of threads per mm length.

$$n = 1/4 = 0.25$$

We know that design load (P_d),

$$65 \times 10^3 = \frac{\pi}{4}[(d)^2 - (d_c)^2]n \times l \times \sigma_c$$

1 Mark

$$= \frac{\pi}{4}[(39)^2 - (34.093)^2]0.25 \times 39 \times \sigma_c = 2750\sigma_c$$

$$\therefore \sigma_c = 65 \times 10^3/2750 = 23.6\ N/mm^2 = 23.6MPa$$

Since the induced crushing stress in the threads of the coupler nut is less than the permissible stress, therefore the design is satisfactory.

3. Outside diameter of the coupler nut

Let $D = Outside\ diameter\ of\ the\ coupler\ nut.$

Considering tearing of the coupler nut. We know that axial load (P),

$$50 \times 10^3 = \frac{\pi}{4}(D^2 - d^2)\sigma_t$$

1 Mark

$$= \frac{\pi}{4}[D^2 - (39)^2]75 = 59[D^2 - (39)^2]$$

or $$D^2 - (39)^2 = 50 \times \frac{10^3}{59} = 848$$

$$\therefore D^2 = 848 + (39)^2 = 2369 \ or \ D = 48.7 \ say \ 50 \ mm \ Ans.$$

Since the minimum outside diameter of coupler nut is taken as $1.25\,d$ ($i.e.\ 1.25 \times 39 = 48.75\ mm$), therefore the above value of d is satisfactory.

4. Outside diameter of the coupler

Let $D_2 =$ Outsider diameter of the coupler, and

$D_1 =$ Inside diameter of the coupler $= d + 6\ mm = 39 + 6 = 45mm$

1 Mark

Considering tearing of the coupler. We know that axial load (P),

$$50 \times 10^3 = \frac{\pi}{4}[(D_2)^2 - (D_1)^2]\sigma_t = \frac{\pi}{4}[(D_2)^2 - (45)^2]75 = 59[(D_2)^2 - (45)^2]$$

$$\therefore (D_2)^2 = 50 \times 10^3/59 + (45)^2 = 2873 \ or \ D_2 = 53.6\ mm$$

Since the minimum outside diameter of the coupler is taken as $1.5\,d$($i.e.\ 1.5 \times 39 =$ *58.5 say 60 mm,* therefore we shall take

$$D_2 = 60\ mm\ Ans.$$

1 Mark

5. Length of the coupler between nuts,

$$L = 6\,d = 6 \times 39 = 234\ mm\ Ans.$$

6. Thickness of the coupler,

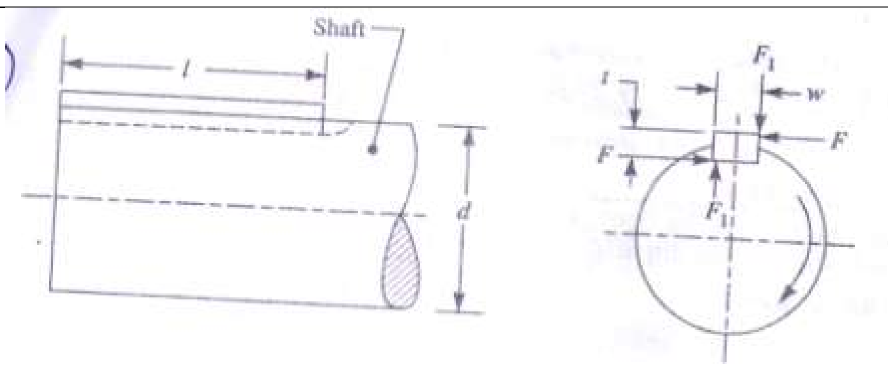
$$t_1 = 0.75\,d = 0.75 \times 39 = 29.25\ say\ 30\ mm\ Ans.$$

1 Mark

and thickness of the coupler nut,

$$t = 0.5\,d = 0.5 \times 39 = 19.5\ say\ 20\ mm\ Ans.$$

Q .3	a.	**Attempt any FOUR** Given $P = 600\ kw = 600 \times 10^3\ watts$ $N = 110\ rpm$ $Tmax = 1.20 \times Tmean$ $\tau = 63\ MPa$ $\theta = 1.4^0 = 1.4 \times \frac{\pi}{180} = 0.024\ radian$ $l = 3m$ $do = \delta$ $\frac{di}{do} = \frac{3}{8} = k$ $G = 84\ GPa = 84 \times 10^9\ N/m^2 = 84 \times 10^3 N/mm^2$ Toque transmitted by hollow shaft $Tmean = \frac{P \times 60}{2\pi N} = \frac{600 \times 10^3}{2\pi \times 110} \times 6 = 52087.07Nm$ $Tmax = 1.2Tmean = 1.2 \times 52087.07 = 62504.48\ N.m$ (i) Torque transmitted by the hollow shaft $Tmax = \frac{\pi}{16} \tau\ d_0^3 (1 - k^4)$ $(62504.48x1000) = \frac{\pi}{16} \times 63 \times (d_o)^3 (1 - 0.375^4)$ $62504.48 \times 10^3 = 12.13(do)^3$ $(do)^3 = 5152883.759$ $\therefore do = 174\ mm$	1 Mark 1 Mark

		(ii) Consider stiffness $$\frac{T}{J} = \frac{G\theta}{L}$$ $$J = \frac{\pi}{32} \times [(do)^4 - (di)^4] = \frac{\pi}{32}(do)^4[1 - k^4]$$ $$J = \frac{\pi}{32} \times do^4(1 - 0.375^4)$$ $$J = 0.0962\,(do)^4$$ $$\frac{Tmax}{J} = \frac{G\theta}{L}$$ $$\frac{(62504.46) \times 10^3}{0.0962(do)^4} = \frac{84 \times 10^9 \times 0.024 \times 10^{-6}}{3 \times 1000}$$ =966867141.9 (do) =176mm Taking larger of the two values do = 176mm	1 Mark 1 Mark
b)		 Considering a square key connecting the shaft hub. Let, T = Torque transmitted by shaft F = Tangential force acting at the circumference of d = Diameter of shaft	1 Mark

		l = Length of key w = Width of key t = Thickness of key for square key, we have, w = t Let τ and σ_{ck} be the permissible shear and crushing stress for key material respectively. A little consideration will show that due to power transmitted by the shaft the key may fail either due to shearing or crushing. Considering shearing of key, the tangential shearing force acting on the shaft is F = Area resisting shearing x Shear stress = l.w.r Also, Torque transmitted = $T = F \times \frac{d}{2} = l.w.\tau.\frac{d}{2}$ Considering crushing key, the tangential crushing force acting on the shaft is, F= Area resisting crushing x Crushing stress $= l.\frac{t}{2}.\sigma_{ck}$ ∴ Torque transmitted = $T = F \times \frac{d}{2} = l.\frac{t}{2}.\sigma_{ck}.\frac{d}{2}$ If key is equally strong in shearing and crushing then on equating equation and we get, $l.w.\tau.\frac{d}{2} = l.\frac{t}{2}\sigma_{ck}.\frac{d}{2}$ $\frac{\tau}{\sigma_{ck}} = \frac{t}{2w}$ But, for square key, $w = t$ $\frac{\tau}{\sigma_{ck}} = \frac{1}{2}$ or $\sigma_{ck} = 2\tau$	1 Mark 1 Mark 1 Mark
c)	c.	(i) x 20 Cr 18 Ni 2 High alloy steel having Carbon = 0.20 %,	1

		Chromium = 18 % Nickel = 2 %	Mark
		(ii) 35 C8 Plain carbon steel having Carbon = 0.35 % Manganese = 0.8 %	1 Mark
		(iii) fe E 230 Steel with minimum yield strength of 230 N/mm^2	1 Mark
		(iv) FG 200 Grey C.I. with tensile strength 200 MPa	1 Mark
d)		(i) Maximum principal or Normal stress theory or Rankine's Theory: According to this theory, "the failure of machine part occurs, when maximum principal stress in a biaxial stress system reaches limiting stress of the material in a simple tension test therefore, factor of safety is taken into consideration". According to this theory, we have maximum principal stress as, $\sigma_{t\,max} = \frac{\sigma_{yt}}{F.O.S.}$ (For ductile material) Where $\sigma_{yt} = Yield\ point\ tensile\ stress$ F.O.S. = Factor of safety Also $\sigma_{t\,max} = \frac{\sigma_{ut}}{F.O.S.}$ (For brittle material) Where $\sigma_{ut} = Ultimate\ tensile\ stress$ For ductile material, limiting stress is yield point tensile stress. For brittle material, limiting stress is ultimate stress.	2 Marks

This theory preferred for brittle material, as it considers possibility of failures only either in tension or compression. It ignores the possibility of failure due to sharing.

(ii) Maximum shear stress theory of guest's theory:

- According to this theory, "the failure of a machine part occurs, when the maximum shear stress in a biaxial stress system reaches to a value equal to shear stress at yield point in a simple tension test".

- According to this theory, considering factor of safety, we have,

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

2 Marks

Where, τ_{max} = Maximum shear stress

τ_y = Yield point shear stress

But shear stress at yield point is equal to half of yield point stress in tension.

$$\therefore\ \tau_y = \frac{1}{2}\sigma_{yt}$$

Where, σ_{yt} = Yield point stress in tension

Therefore equation (1.1) becomes,

$$\therefore\ \tau_{max} = \frac{\frac{1}{2}\sigma_{yt}}{F.O.S.} = 0.5 \times \frac{\sigma_{yt}}{F.O.S.}$$

$$\therefore \tau_{max} = 0.5 \times \sigma_{tmax} \qquad \because\ \sigma_{tmax}=\frac{\sigma_{yt}}{F.O.S.}$$

This theory is preferred for ductile materials.

e)

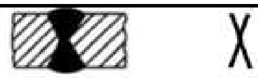
Rolling contact bearing	Sliding contact bearing
i) Size: Rolling contact bearing requires considerable radial space.	i) Sliding contact bearing requires more axial space
ii) Life: to fluctuating loads, the life of	ii) Life of sliding contact bearing is more.

		rolling contact bearing is limited.	
		iii) Coefficient of friction – Low starting friction - high running friction due to metal to metal contact	iii) High starting friction due to metal to metal contact & low running friction due to oil film separating the journal & bearing surfaces.
		iv) Resistance to shock – Poor in taking shock loads	iv) It can take shock load.

1 Mark each

Q.4	**a)**	**Attempt Any Three**	
	a)	Write Lewis equation for the strength of gear tooth. Give the meaning of each term.	
	Ans:	Lewis equation of Beam strength of gear tooth $S_b = \pi m.b.\sigma_b.Y$ where, S_b = beam strength of gear tooth (N)	Equation 2 marks Meaning

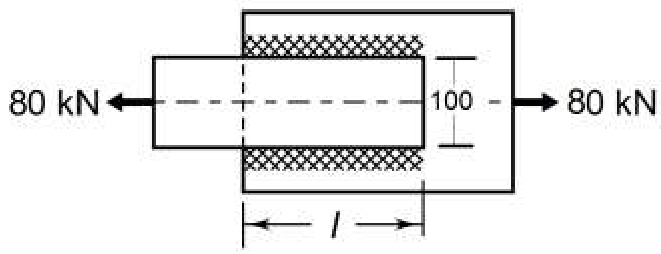
		σ_b = permissible bending stress (N/mm^2) m = module of gear b = face width = width of gear Y = Lewis form factor $\left(\frac{t^2}{6hm}\right)$	of terms 2 marks
	b) Ans	Define the terms: **i) Solid length** Solid length is defined as the axial length of the spring which is so compressed that the adjacent coils touch each other such that no further compression is possible. The solid length is given by, Solid length = Nt. D where, Nt = total number of coils and d= diameter of wire. **ii) Free length** Free Length is defined as the axial length of an unloaded helical compression spring. In this case, no external force acts on the spring. Free length is given by, free length = compressed length + δ = solid length + total axial gap + δ **iii) Spring Index** The spring index is defined as the ratio of mean coil diameter to wire diameter. Or, Spring Index $C = \frac{D}{d}$ **iv) Pitch** The pitch of the coil is defined as the axial distance between adjacent coils in uncompressed state of spring. It is denoted by p. It is given by, $$p = \frac{Free\ Length}{N_t - 1}$$	1 mark for Each definition
	c) Ans	Draw symbolic representation of following types of weld:	1 mark for each symbol

<table>
<tr><td></td><td></td><td>i) Double V Butt Weld 
ii) Double U Butt Weld 
iii) Single Bevel Butt Weld 
iv) Spot Weld </td><td></td></tr>
<tr><td></td><td></td><td></td><td></td></tr>
<tr><td></td><td>d)

Ans</td><td>A plate 100 mm wide and 10 mm thick is to be welded by another plate by means of double parallel fillet welds. The plates are subjected to a static load of 80 kN. (Take permissible shear stress = 55 N/mm^2.Determine the length of the weld.

Given: P = 80 KN, permissible shear stress τ = 55 N/mm^2, plate thickness h= 10 mm

Figure shows the weld arrangement



Length of the weld (l) ,

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

$80\times10^3 = 1.414 \times10\times l\times55$

$$l = \frac{80 \times 10^3}{1.414 \times 10 \times 55} = 102.86\ mm$$
Adding 15 mm of length for starting and stopping of the weld run, the length of the weld is given by,

$$l = 102.86 + 15 = 157.86\ mm$$</td><td>1 mark for figure

1 mark Equation

1 mark Answer

1 mark answer of</td></tr>
</table>

			/

Q.4 b) Attempt **ANY ONE**

a) Design a Knuckle joint to transmit 150 kN, the design stresses are 75 MPa, 60MPa and 150 MPa in tension, shear and compression respectively.

Ans

D = diameter of each rod (mm)

D_1 = enlarged diameter of each rod (mm)

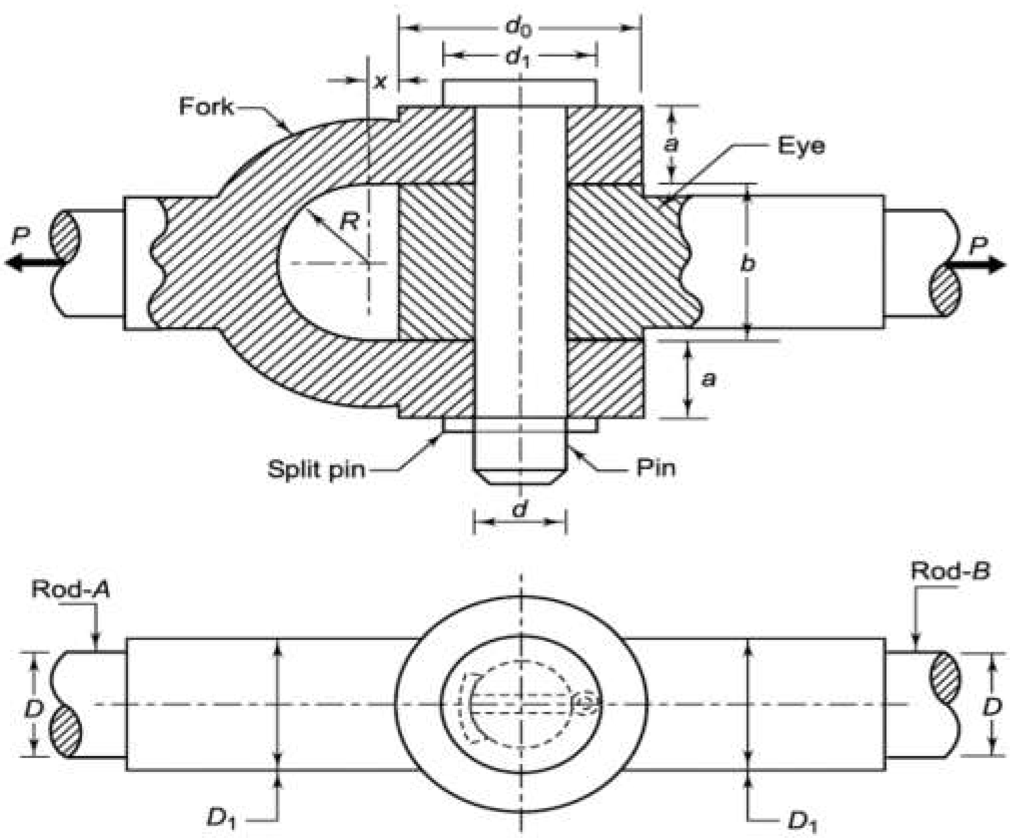
d = diameter of knuckle pin (mm)

d_0 = outside diameter of eye or fork (mm)

a = thickness of each eye of fork (mm)

b = thickness of eye end of rod-B (mm)

d_1 = diameter of pin head (mm)



.

1 mark Figure

Allowable stresses or permissible stresses.

$$\sigma_t = 75\ MPa \quad , \quad \tau = 60\ MPa \quad , \quad \sigma_c = 150\ MPa$$

Step 1: Tensile failure of Rod

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

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

$$D^2 = \frac{150\times10^3}{\frac{\pi}{4}\times75} = \frac{150\times10^3}{58.91} = 2546.14$$

$$\boldsymbol{D = 50.42\ mm = 52\ mm}$$

Step 2: Calculate the enlarged diameter of each rod by empirical relation

D_1= 1.1 D = 1.1× 52 = $57.2\ mm = 60\ mm$

Step 3: Calculate dimensions of a and b by empirical relations — 1 mark For D

a= 0.75 D = 0.75 × 52 = $39\ mm = 40\ mm$

b= 1.25 D=1.25 × 52 = $65\ mm$

Step 4: Shear failure of Pin

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

$$d^2 = \frac{150 \times 10^3}{(2)\frac{\pi}{4} \times 60} = \frac{150 \times 10^3}{58.91} = 3182.69$$

1 mark for D_1,a,b

Diameter of pin **d =39.89mm=say 40mm**

Bending failure of pin.

$$d = \sqrt[3]{\frac{32}{\pi\sigma_b} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

$$d = \sqrt[3]{\frac{32}{\pi \times 75} \times \frac{150 \times 10^3}{2}\left[\frac{65}{4} + \frac{40}{3}\right]}$$

$$d = \sqrt[3]{10184.5958[29.5833]} = 67.03 \text{ mm} = 68 \text{ mm}$$

Selecting the larger diameter of the pin d = 68 mm

2 marks for finding pin diameter using shear and bending criterion

Step 5: Calculate d_o and d_1 by empirical relations

d_o = 2 d = 2 × 68 = 136 mm

d_1 = 1.5 d = 1.5 × 68 = 102 mm

Step 6: Check for stresses in eye

Tensile failure of eye. $\sigma_t = \frac{P}{b(d_0-d)} = \frac{150\times10^3}{65(136-68)} = 33.93$ N/mm^2

Crushing failure of eye $\sigma_c = \frac{P}{bd} = \frac{150\times10^3}{65\times68} = 33.93$ N/mm^2

Shear failure of eye. $\tau = \frac{P}{b(d_0-d)} = \frac{150\times10^3}{65(136-68)} = 33.93$ N/mm^2

1 mark for

		Step 7 Check for stresses in fork Tensile failure of fork. $\sigma_t = \frac{P}{2a(d_0-d)} = \frac{150\times10^3}{2\times40(136-68)} = 27.57$ N/mm^2 Crushing Failure of Fork $\sigma_c = \frac{P}{2ad} = \frac{150\times10^3}{2\times40\times68} = 27.57$ N/mm^2 Shear failure of fork. $\tau = \frac{P}{2a(d_0-d)} = \frac{150\times10^3}{2\times40(136-68)} = 27.57$ N/mm^2 Tensile, crushing and shear stresses induced in eye and fork is within permissible limits hence the design of knuckle joint is safe. • Designed dimensions of knuckle joint are as follows	stresses in eye 1 mark for stresses in fork

D	Diameter of each rod	**52 mm**
D_1	Enlarged diameter of each rod	**60 mm**
d	Diameter of knuckle pin	**40 mm**
d_0	Outside diameter of eye or fork	**136 mm**
a	Thickness of each eye of fork	**40 mm**
b	Thickness of eye end of rod-B	**65 mm**
d_1	Diameter of pin head	**102 mm**

	b)	Explain the following modes of failure of gear tooth: **i) Pitting:** The initial or corrective pitting is a localized phenomenon, characterized by small pits at high spots. Such high spots are progressively worn out and the load is redistributed. Initial pitting is caused by the errors in tooth profile, surface irregularities and misalignment. The remedies against initial pitting are precise machining of gears, adjusting the correct alignment of gears so that the load is uniformly distributed across the full face width, and reducing the dynamic loads. **ii) Scoring:** Excessive surface pressure, high surface speed and inadequate supply of lubricant result in the breakdown of the oil film. This results in excessive frictional heat and overheating of	2 marks for each failure explaining phenom-enon and causes, remedies

		the meshing teeth. Scoring is a stick-slip phenomenon, in which alternate welding and shearing takes place rapidly at the high spots. Here, the rate of wear is faster. Scoring can be avoided by selecting the parameters, such as surface speed, surface pressure and the flow of lubricant in such a way that the resulting temperature at the contacting surfaces is within permissible limits. The bulk temperature of the lubricant can be reduced by providing fins on the outside surface of the gear box and a fan for forced circu1ation of air over the fins. **iii) Abrasive wear** Foreign particles in the lubricant, such as dirt, rust, weld spatter or metallic debris can scratch or brinell the tooth surface. Remedies against this type of wear are provision of oil filters, increasing surface hardness and use of high viscosity oils. A thick lubricating film developed by these oils allows fine particles to pass without scratching.	
Q.5	a)	Attempt **ANY TWO**: A closed coil helical spring is used for the front suspension of an automobile. The spring has stiffness 90 N/mm with square and ground ends. The load on the spring causes a total deflection of 8.5 mm. by taking permissible shear stress of material as 450 MPa Find: i) Spring wire diameter ii) Length of spring Assume spring index =6 and G=80000 N/mm^2. Given : k = 90 N/mm, δ = 8.5 mm, τ = 450 MPa, C = 6, G = 80000 N/mm^2 **i) <u>Spring wire diameter d</u>** As $k=\frac{P}{\delta}$ $\quad \therefore P = k \times \delta = 90 \times 8.5 = 765\ N$ Wahl's Correction factor K = $\frac{4C-1}{4C-4}+\frac{0.615}{C} = \frac{4(6)-1}{4(6)-4}+\frac{0.615}{6} = 1.2525$ $\tau = K\frac{8PD}{\pi d^3} \quad \therefore \quad d^2 = K\frac{8PC}{\pi\tau} = 1.2525 \times \frac{8(765)(6)}{3.142 \times 450} = 32.528$ $\therefore d = 5.7\ mm = 6\ mm$ **ii) <u>Length of the spring</u>** Mean coil diameter D = C × d = 6 × 6 = 36 mm.	1 mark for **P** 1 mark for **K** 1 mark for **d**

		Number of active coils $\delta = \frac{8PD^3N}{Gd^4} = \frac{8PC^3N}{Gd} \therefore N = \frac{\delta \times Gd}{8PC^3} = \frac{8.5 \times 80000 \times 6}{8 \times 765 \times 6^3} = 3.0864 = 4\ coils$ Total number of coils for squared and ground ends $N_t = N + 2 = 4 + 2 = 6$ coils Actual deflection of the coil is $\delta = \frac{8PD^3N}{Gd^4} = \frac{8PC^3N}{Gd} = \frac{8(765) \times 6^3 \times 4}{80000 \times 6} = 11.016\ mm$ Solid length of spring = N_t d = 6(6) = 36 mm Free length = Solid length + Total axial gap + δ = 36 + 0.15(δ) + δ = 36+ 0.15(11.016) +11.016 = 48.6684 mm or 49 mm	1 mark for **N** 1 mark for **N_t** 1 mark for **actual** δ 1 mark for **solid length** 1 mark for **Free length**

Q.5	b)	Give the design procedure of screw and nut of screw jack.	
	Ans	**i) Design of Screw** • Direct compressive stress $\sigma_c = \frac{W}{\frac{\pi}{4}(d_c{}^2)}$ Where $\sigma_c = Permissible\ compressive\ stress\ of\ screw\ material$ $d_c = core\ diamter\ of\ power\ screw\ thread$ From above equation calculate d_c There are additional stresses due to the collar friction torque. At this stage, it is not possible to calculate the additional stresses in the lower and upper parts of the screw. To account for these additional stresses, the diameter should be increased. • Find the total torque required to raise the load$(M_t)_t$ $(M_t)_t$ = Torque required to raise the load + Torque to overcome collar friction $(M_t)_t = M_t + (M_t)_c$ Torque required to raise the load M_t can be calculated as $M_t = \frac{Wd_m}{2} tan(\varphi + \alpha)$ Torque to overcome collar friction $(M_t)_c$ can be calculated as given below Using uniform pressure theory $(M_t)_c = \frac{\mu_c W}{3} . \frac{D_0^3 - D_i^3}{D_0^2 - D_i^2}$ or Using uniform wear theory $(M_t)_c = \frac{\mu_c W}{4} .(D_0 + D_i)$ Where D_0 and Di are the outer and inner diameter of the collar. • Check the torsional shear stress induced in the screw which is given by, Torsional shear stress $\tau = \frac{16(M_t)_t}{(\pi d_c{}^3)}$ • Using Maximum Shear Stress Theory determine maximum shear stress induced,	1 mark 1 mark 1 mark

		Principal shear stress is given by, $\tau_{max} = \sqrt{\left(\frac{\sigma_c}{2}\right)^2 + \tau^2}$ τ_{max} *must be less than permissible shear stress for screw material.* Otherwise increase the d_c • Screw threads engaged with nut and experiences transverse shear stress which is given by, $\tau_{sceww} = \frac{W}{(\pi\, d_c\, t\, z)}$ Where, τ_{screw} = transverse shear stress at root dia. of screw t = thread thickness at core diameter (mm) = 0.5 pitch z = number of threads in engagement with nut. *Determine the number of threads in contact* (z) using above equation **ii) <u>Design of nut</u>** • <u>Bearing pressure between contacting surface of screw and nut</u> Bearing area = $\left[\frac{\pi}{4}(d^2 - {d_c}^2)\right]$ $S_b = \frac{W}{\left[\frac{\pi}{4}(d^2 - {d_c}^2)z\right]}$ Where, S_b = unit bearing pressure in N/mm^2 *Determine the number of threads in contact* (z) using above equation Determine the height of the nut using relation H = z p • <u>Check the threads of the nut for the transverse shear stress</u> Screw threads engaged with nut and experiences transverse shear stress which is given by, $\tau_{nut} = \frac{W}{(\pi\, d\, t\, z)}$ Where, τ_{nut} =transverse shear stress at nominal dia. of nut t = thread thickness at core diameter (mm) z = number of threads in engagement with nut.	1 mark 1 mark 2 marks 1 mark
Q.5	c)	Write the general design procedure of flange coupling (unprotected type)	

Ans:

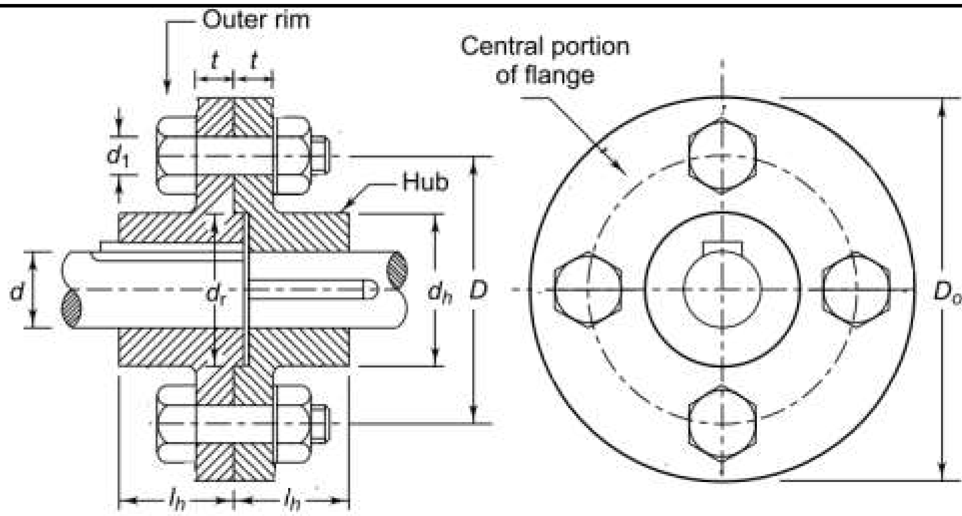


Figure 1 mark

The basic procedure for finding out the dimensions of the rigid flange coupling consists of the following steps:

i) Shaft Diameter

Calculate the shaft diameter d by using the following two equations:

$$\text{Twisting moment } M_t = \frac{60 \times 10^6 (kW)}{2\pi n} \text{ N.mm}$$

$$\boldsymbol{\tau} = \frac{16\, M_t}{(\pi d^3)}$$

Determine the shaft diameter from above equation.

1 Mark for shaft diameter **d**

ii) Dimensions of Flanges

Calculate the dimensions of the flanges by the following empirical equations:

Diameter of hub $d_h = 2d$

Length of the hub $l_h = 1.5d$

Pitch Circle Diameter of bolts D = 3d

Thickness of the flange t= 0.5 d

Diameter of spigot and recess dr = 1.5d

Outside diameter of flange Do = 4d

The torsional shear stress in the hub can be calculated by considering it as a hollow shaft subjected to torsional moment Mt. The inner and outer diameters of the hub are d and d_h respectively.

1 mark

The torsional shear stress in the hub is given by,

$$\tau = \frac{M_t r}{J}, \quad J = \frac{\pi(d_h^2 - d^2)}{32} \quad ,r = \frac{d_h}{2}$$

		The flange at the junction of the hub is under shear while transmitting the torsional moment Mt. $$M_t = \frac{1}{2}\pi . d_h^2 . t . \tau$$ Check the shear stress of CI flange within safe limits **iii) Diameter of Bolts Decide the number of bolts using the following guidelines:** N = 3 for d < 40 mm N = 4 for 40 ≤ d < 100 mm N = 6 for 100 ≤ d < 180 mm Determine the diameter of the bolt by Eq. Rearranging the equation, $$d_1^2 = \frac{8M_t}{\pi . D . N . \tau}$$ where τ is the permissible shear stress for the bolt material. The compressive stress in the bolt can be determined by referring to again. Crushing area of each bolt = $d_1 . t$ Crushing area of all bolts = $N.d_1 . t$ Compressive force = $N.d_1 . t. \sigma_c$ Then, $$M_t = \frac{D}{2} . \mathrm{N.d1.t.} \sigma_c$$ $$\sigma_c = \frac{2M_t D}{\mathrm{N.d1.t}}$$ Above equation is used to check the compressive stress in the bolt. **iv) Dimensions of Keys** Determine the standard cross-section of flat key. The length of the key in each shaft is l_h. Therefore, $l = l_h$ With these dimensions of the key, check the shear and compressive stresses in the key by $\sigma_c = \frac{2M_t}{\mathrm{d.b.l}}$ and $\sigma_c = \frac{4M_t}{\mathrm{d.h.l}}$	1 mark 1 mark 1 mark 2 marks
Q.6	a)	Attempt **Any Four** Explain : **i) Self-Locking**	

Ans

The torque required to lower the load can be obtained by equation

$$M_t = \frac{W d_m}{2} tan(\varphi - \alpha)$$

Where W – Load to be lifted,

d_m= Mean diameter of thread of screw

φ is the friction angle

and α is the helix angle

- When $\boldsymbol{\varphi} \geq$, 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. In this case, the screw is said to be 'self-locking'.
- A self-locking screw will hold the load in place without a brake.
- Can be achieved by optimum lubrication and less lead of screw thread.
- Application: Screw-jack

2 marks

ii) Overhauling of a power screw

- Using the same equation of torque required to lower the load, it can be seen that when; $\boldsymbol{\varphi < \alpha}$ 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. This condition is called overhauling of the screw.
- Caused due to excessive lubrication (Minimizing friction) or high lead of power screw thread.
- Applications- Yankee screwdriver , Power steering (Recirculating Ball type Screw)

2 marks

b) Explain with neat sketch the bolt of uniform strength.

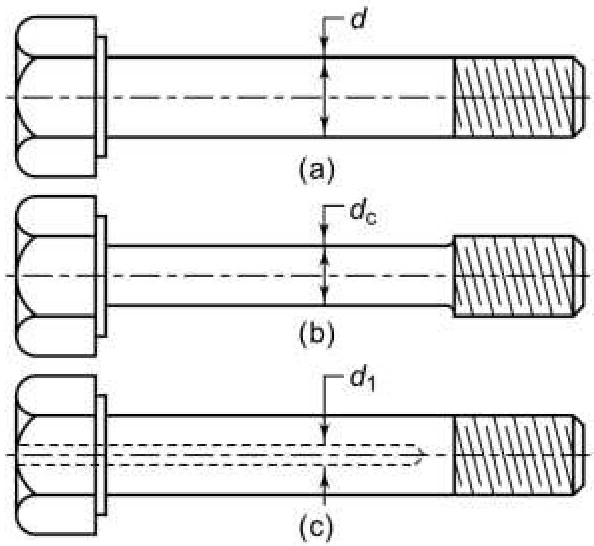
(Explaination including two methods of making bolt of uniform strength accompanied with figure. 2 marks each)

In an ordinary bolt shown in **Fig. (a)**, the effect of the impulsive loads applied axially is concentrated on the weakest part of the bolt i.e. the cross-sectional area at the root of the threads. In other words, the stress in the threaded part of the bolt will be higher than that in the shank. Hence a great portion of the energy will be absorbed at the region of the threaded part which may fracture the threaded portion because of its small length.

If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (d) as shown in **Fig. (b)**, then shank of the bolt will undergo a higher stress. This means that a shank will absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us bolts of uniform strength. The resilience of a bolt may also be increased by increasing its length.

A second alternative method of obtaining the bolts of uniform strength is shown in **Fig. (c).** An axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread.

2 marks



Bolts of Uniform Strength

2 marks

c) The spindle of a drilling machine is subjected to a maximum load of 10 kN. Determine the diameter of solid C.I. column of machine if tensile stress is limited to 40 N/mm2. The distance between axis of spindle and axis of column is 330mm. Also find the direct stress and stress due to bending in the column. (Refer figure No.1)

Ans:

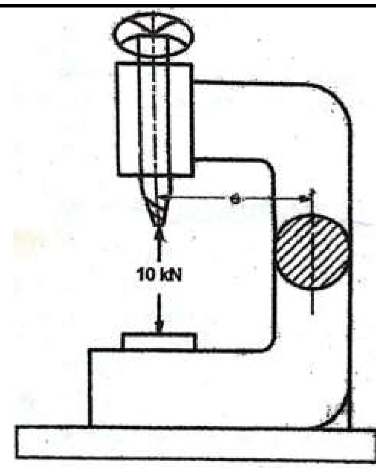


Fig. No. 1 [Q. No. 6(c)]

Given: P = 10000 N, e = 330 mm, $\sigma_t = 40\ N/mm^2$ — 1 mark

i) Direct tensile stress on the cross section is given by, — 1mark

$$\sigma_o = \frac{4\,P}{\pi d^2} = \frac{4\times 10000}{\pi d^2} = \frac{12730.75}{d^2} N/mm^2$$

ii) Bending Stress at the inner fiber of the circular column is given by,

$$\sigma_b = \frac{M_b \cdot y}{I} = \frac{M_b\left(\frac{d}{2}\right)}{\frac{\pi d^4}{64}} \quad or\sigma_b = \frac{32M_b}{\pi d^3} = \frac{32(P\times e)}{\pi d^3} = \frac{32(10000\times 330)}{\pi d^3} = \frac{33609166.140}{d^3} N/mm^2$$

2 marks

Summation of above two stresses must be equal to the permissible tensile stress.

Permissible tensile stress $\sigma_t = \sigma_o + \sigma_b$

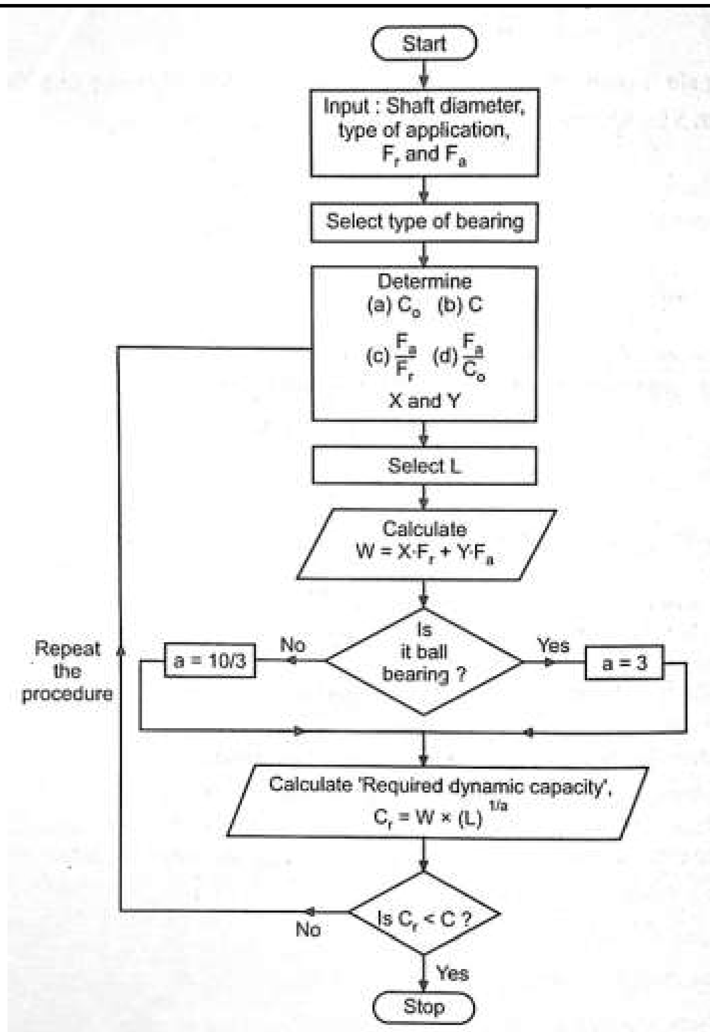
$$40 = \frac{12730.75}{d^2} + \frac{33609166.140}{d^3}$$

d = 95.49 mm = 96 mm……….(by trial and error method)

d) Write down the procedure for selection of bearing from manufacturer's catalogue.

Ans

Note: It is not necessary to provide the above design procedure in a flow chart.
Even if student is enlistis the steps in a proper sequence, shall be given due credit.



4 marks

e) State any four advantages and disadvantages of welded joints over screwed joints.

Ans

Advantages:

(i) Screwed joints require additional elements like washers and a large number of nut bolts, which increase the weight. Since there are no such additional parts, welded assembly results in lightweight construction. Welded steel structures are lighter compared to threaded joints.

(ii) Due to the elimination of these components, the cost of welded assembly is lower than that of screwed joints.

(iii) The design of welded assemblies can be easily and economically modified to meet the changing product requirements. Alterations and additions can be easily made in the existing structure by welding.

(iv) Welded assemblies are tight and leak-proof as compared with screwed assemblies.

		(v) The production time is less for welded assemblies. (vi) When two parts are joined by the nut and bolts, holes are drilled in the parts to accommodate the nut and bolts. The holes reduce the cross-sectional area of the members and result in stress concentration. There is no such problem in welded connections. (vii) A welded structure has smooth and pleasant appearance. The projection of screw head adversely affects the appearance of the threaded joint structure. (viii) The strength of welded joint is high. Very often, the strength of the weld is more than the strength of the plates that are joined together. (ix) Almost any shaped machine components can be easily welded which is difficult using screwed joints. (x) Efficiency of assembly is higher. **Disadvantages**: (i) The capacity of welded structure to damp vibrations is poor. (ii) Welding results in a thermal distortion of the parts, thereby inducing residual stresses. In many cases, stress-relieving heat treatment is required to relieve residual stresses. (iii) The quality and the strength of the welded joint depend upon the skill of the welder. It is difficult to control the quality when a number of welders are involved. (iv) The inspection of the welded joint is more specialized and costly compared with the threaded structures.	Any four Adv. 2 marks 2 Marks