WINTER– 2022 EXAMINATION

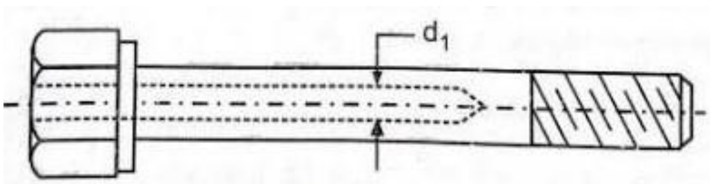
Model Answer

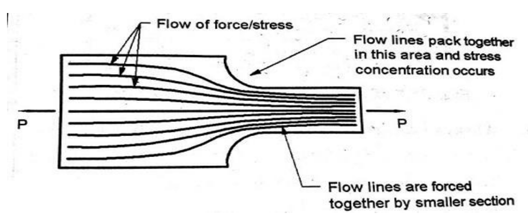
Subject Name: Elements of Machine Design **Subject Code:** 22564

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 somequestions 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.
8) As per the policy decision of Maharashtra State Government, teaching in English/Marathi and Bilingual (English + Marathi) medium is introduced at first year of AICTE diploma Programme from academic year 2021-2022. Hence if the students in first year (first and second semesters) write answers in Marathi or bilingual language (English +Marathi), the Examiner shall consider the same and assess the answer based on matching of concepts with model answer.

Q. No.	Sub Q. N.	Answer	Marking Scheme
Que.1		**Attempt any FIVE of the following**	**10 Marks**
	a)	**Give composition of 45Cr20Si2**	
	Sol.	Alloy Steel Containing 0.45% of carbon , 20% of chromium and 2% of silicon.	02 Mark
	b)	**Define creep**	
	Sol.	When a machine component is subjected to constant stress (load) at high temperature for a long period of time,it will undergo a slow and progressive permanent deformation called creep.	02 Mark
	c)	**State two application of Knuckle Joint**	
	Sol.	i. Tie rod of roof truss. ii. Link of bicycle chain. iii. Link of roller chain. iv. Tension link in bridge structure. v. Tie rod of jib crane. vi. Air braking arrangement of locomotive and track shifting mechanism of railway. vii. It is used in coupling trolley with railway. viii. It is used in structure for suspension link.	01 Mark for one application Any two expected

Q. No.	Sub Q. N.	Answer	Marking Scheme
	d)	**State any two examples where hollow shafts are used**	
	Sol.	i. It is used in machine tool spindle. ii. Marine applications. iii. Automobile. iv. Electric motor.	01 Mark for each Any two expected
	e)	**Explain "bolts of uniform strength"**	
	Sol.	A method of drillling an axial hole through the head of the bolt as far as the threaded portion; such that the area of the shank became equal to the root area of the thread. By achieving this the intensity of stress, that is stress in the shank portion and the threaded portion is same.This gives the bolt of uniform strength.  (c) Let, d1= Diameter of hole d_o = Outer diameter of thread dc = Core diameter of thread $\frac{\Pi}{4} d1^2 = \frac{\Pi}{4}(do^2 - dc^2)$ $d1 = \sqrt{(do^2 - dc^2)}$	02 Mark
	f)	**State any two applications of torsion spring.**	
	Sol.	i. Clocks. ii. Automobile. iii. Door Hinges. iv. Spring balance.	01 Mark for each Any two expected
	g)	**Define "Basic Static Load Rating" of rolling contact bearings.**	
	Sol.	The basic static load rating is defined as the static radial load which correponds to a permanent deformation of the ball and race at the most heavily stressed contact equal to 0.0001 times the ball diameter.	02 Mark

<table>
<tr><th>Q. No.</th><th>Sub Q. N.</th><th>Answer</th><th>Marking Scheme</th></tr>
<tr><td>Q.2</td><td></td><td>Attempt any <u>THREE</u> of the following:</td><td>12 Marks</td></tr>
<tr><td></td><td>a)</td><td>Explain the term stress concentration and remedies to reduce stress concentration with neat sketch (any four)</td><td></td></tr>
<tr><td></td><td>Sol.</td><td>
1. During the design of any machine component, discontinuities in any machine part are there due keyway, threaded grooves and steps are present on the component which is functional requirement to perform their functions.
2. Such type of discontinuity alters the stress distribution in the vicinity of the discontinuity and elementary stress equations no longer describe the state of stress in the component.
3. The stresses induced in the neighbourhood of the discontinuity are much higher than the stresses in the other part of the component.
4. This concentration of high stresses due to the discontinuities or abrupt change of cross-section is called stress concentration.


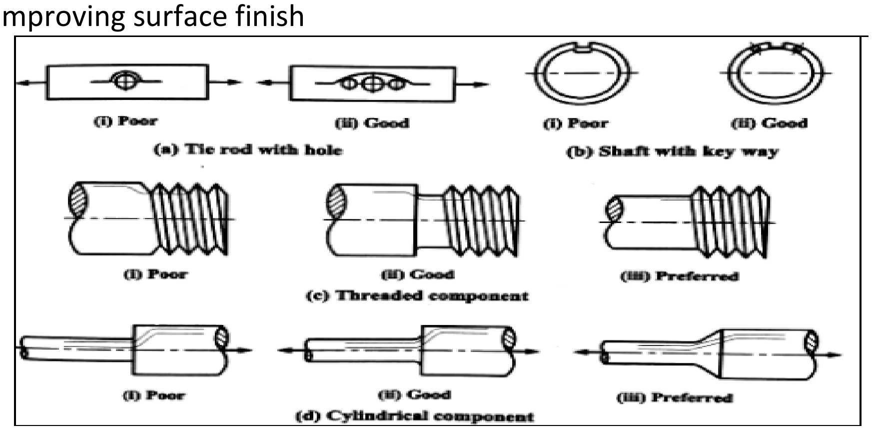
5. remedies to reduce stress concentration are as follows:
i. Additional notches and holes in tension member like use of use of multiple notches and drilling additional holes.
ii. Fillet radius,undercutting and notch for member in bending.
iii. Reduction of stress concentration in Threaded members.
iv. Change in cross-section should be gradual.
v. By improving surface finish


Fig. Methods of reducing stress concentration
</td><td>02 Marks

01 Mark

01 Mark</td></tr>
<tr><td></td><td>b)</td><td>Discuss the design procedure of socket and spigot cotter joint.</td><td></td></tr>
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Q. No.	Sub Q. N.	Answer	Marking Scheme

Sol.

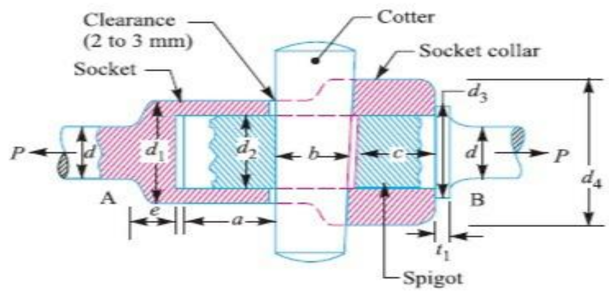


Fig: Socket spigot cotter joint

01 Mark for Fig & notation

Let, P = load carried by the rods
d = diameter of rod 01 Mark for figure
d1= outside diameter of socket
d2= inside diameter of socket/ diameter of spigot
d3= outside diameter of spigot collar
d4= outside diameter of socket collar
a = distance from the end of the slot to the end of the rod
b = mean width of cotter
c = thickness of socket collar
l = length of cotter
t = thickness of cotter
t1 = thickness of spigot collar
σt = permissible tensile stress for the rod material
σc = permissible crushing stress for the cotter material
τ = permissible shear stress for the cotter material

1/2 Mark

I) Design of Rod:

σt = $\frac{P}{A}$ = P /(π÷4) d^2 find 'd'

II) Design of Spigot:

i. Spigot failure in tension

σt = $\frac{P}{A}$ = P / [(π÷4)d^2 - d^2.t] always take t = $d2/4$ find 'd2' & 't'

1/2 Mark

ii. Spigot failure in crushing

σcrush = $\frac{P}{A}$ = P / (d2.t) Check σcrush and finalize d2& t

iii. Spigot failure in shear

τ = $\frac{P}{2A}$= P / 2 (d2.a) find 'a'

1/2 Mark

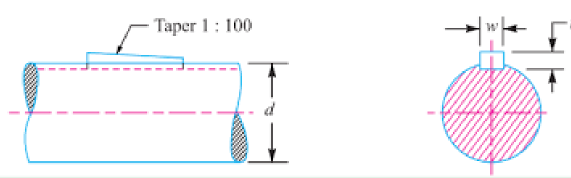
iv. Spigot collar failure in shear

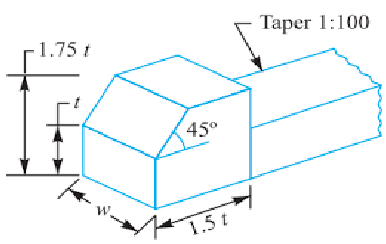
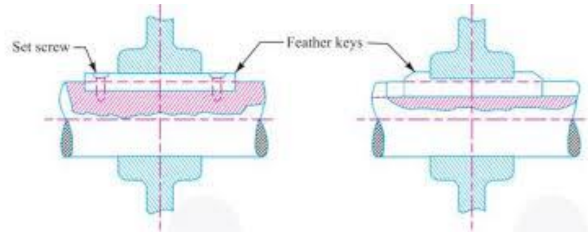
τ = $\frac{P}{A}$ = P / (π d2.t1) find 't1'

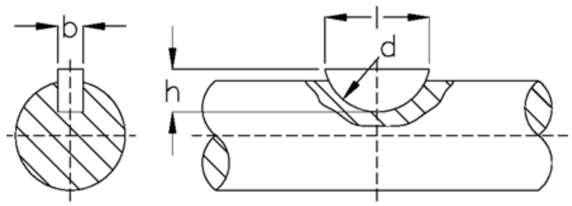
v. Spigot collar failure in crushing

1/2 Mark

σcrush = $\frac{P}{A}$ = P / [(π÷4)($d3^2 - d2^2$)] find 'd3'

Q. No.	Sub Q. N.	Answer	Marking Scheme
		III) <u>Design of Socket:</u> i) Socket failure in tension $\sigma t = \frac{P}{A} = P / [(\pi \div 4)\,(d1^2 - d2^2)] - [(d1 - d2)\,t]$ find 'd1' ii) Socket collar failure in crushing $\sigma crush = \frac{P}{A} = P / [(d4 - d2)\,t]$ find 'd4' iii) Socket collar failure in shear $\tau = \frac{P}{2A} = P / 2\,[(d4 - d2)\,c]$ find 'c' **IV) <u>Design of Cotter:</u>** $\tau = \frac{P}{2A} = P / 2\,(b.t)$ find 'b' Length of Cotter L = 4.d	1/2 Mark 1/2 Mark
	c)	**Explain with neat sketch different types of sunk keys.**	
		Different types of sunk keys are as follows: 1. Rectangular sunk key. 2. Square sunk key. 3. Parallel sunk key. 4. Gib headed key. 5. Feather key. 6. Woodruff key. **1. <u>Rectangular sunk key.</u>**  **Fig. 13.1.** Rectangular sunk key. i. The various proportions of this key are: ii. Width of key = w = $\frac{d}{4}$ iii. Thickness of key = t= $\frac{d}{6}$ iv. Where,d = diameter of the shaft v. The key has taper 1 in 100 on the top side only.	01 Mark 1/2 Mark

Q. No.	Sub Q. N.	Answer	Marking Scheme
	Sol.	**2. <u>Square sunk key.</u>** The only difference between a rectangular sunk key and square sunk key is that its width and thickness are equal. $W = t = \frac{d}{4}$ **3. <u>Parallel sunk key.</u>** i. The parallel key may be of rectangular or square cross-section uniform in width and thickness throught. ii. It may be noted that parallel key is taperless and is used where pulley, gear or other mating piece is required to slide along the shaft. **4. <u>Gib headed key.</u>**  i. It is a rectangular key with a head on one end known as gib head. ii. It is usually provided to facilitate the removal of key. **5. <u>Feather key.</u>**  i. A key attached to one member of a pair and which permits relative axial movement is known as feather key, ii. It is a special type of parallel key which transmits a turning moment and also permits axial moment, iii. It is fastened either to a shaft or hub, the key being a sliding fit in the keyway of the moving piece.	1/2 Mark 1/2 Mark 1/2 Mark 1/2 Mark

<table>
<tr><th>Q. No.</th><th>Sub Q. N.</th><th>Answer</th><th>Marking Scheme</th></tr>
<tr><td></td><td></td><td>6. <u>Woodruff key</u>.


i. The woodruff key is an easily adjustable key.
ii. It is a piece from a cylindrical disc having segmental cross-section in front view. This key is capable of tilting in a recess milled out in the shaft by a cutter having the same curvature as the disc from which the key is made.
iii. This key is largely used in machine tool and automobile construction.</td><td>1/2 Mark</td></tr>
<tr><td></td><td>d)</td><td>Explain self locking and overhauling property of screw.</td><td></td></tr>
<tr><td></td><td>Sol.</td><td><u>Self Locking Screw</u>
The effort required at the circumference of the screw to lower the load is given by equation,
P = W.tan (ø - α)

T = W.tan (ø - α). (dm/2)

In the above equation, if ø > α (i.e friction angle is greater than helix angle), the torque required to lower the screw is positive i.e. some effort is required to lower the load, such a screw is known as self locking.

<u>Over-hauling Screw</u>
The effort required at the circumference of the screw to lower the load is given by equation,
P = W.tan (ø - α)

T = W.tan (ø - α). (dm/2)

In the above equation, if ø < α (i.e friction angle is less than helix angle), the torque required to lower the screw is negative i.e. load will start moving downward without the application of any torque. Such condition is known as overhauling of screw.</td><td>02 Mark

02 Mark</td></tr>
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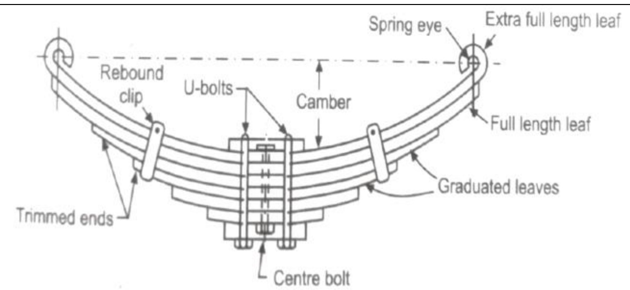
<tr><th>Q. No.</th><th>Sub Q. N.</th><th>Answer</th><th>Marking Scheme</th></tr>
<tr><td>Que.3</td><td></td><td>Attempt any <u>THREE</u> of the following</td><td>12 Marks</td></tr>
<tr><td></td><td>a)</td><td>Explain construction of leaf spring with neat sketch</td><td></td></tr>
<tr><td></td><td>Sol.</td><td>

Construction of leaf spring :
Semi-elliptical leaf springs are widely used for suspension in light and heavy commercial vehicle. In car these are used for rear suspension.
• The leaf springs are made of flat semi-elliptical plate.
• The advantage of leaf spring over helical spring is that the ends of the spring may be guided along the definite path as it deflects to acts as a structural member in addition to energy absorbing device.
• Thus, leaf spring carry lateral load, brake torque, driving thrust and shocks.
• It consists of number of semi-elliptical plates called blade or leaves.
• The leaves are given initially curvature or camber so that they tend to straighten under the load.
• The blades vary in length and are held together by a bolt passing through the center acting as a beam of uniform strength.
• The spring is clamped to the axle housing by means of ‘U’ bolts.
• The longest leave is known as master leaves, has its end formed in the shape of an eye through which the bolts are passed to secure the spring to its supports.
• The eyes are attached to shackle provided with bushing of anti-friction material such as bronze or rubber. The other leaves are graduated leaves.
• To prevent digging in the adjacent leaves, the ends of graduated leaves are trimmed in various forms.
• The master leaf has to withstand vertical bending load, side thrust and twisting moment due to presence of stresses caused by these loads so it is usual to provide two full length leaves and rest graduated leaves.
• Rebound clips are located at intermediate position in the length of the spring so that graduated leaves also shear the stresses induced in the full-length leaves when the spring rebound.
• Highly cambered spring provides a 50% suspension but they also increase tendency to jaw (movement about vertical axis). Flat spring reduces tendency of the vehicle to dip (pitching), when brake or accelerate suddenly.
• Generally rear spring are kept longer than the front spring. This causes them to vibrate at different frequencies, which prevent excessive bounce.</td><td>Diagram &lable:

1 & 1/2

Explanation :

2 & 1/2</td></tr>
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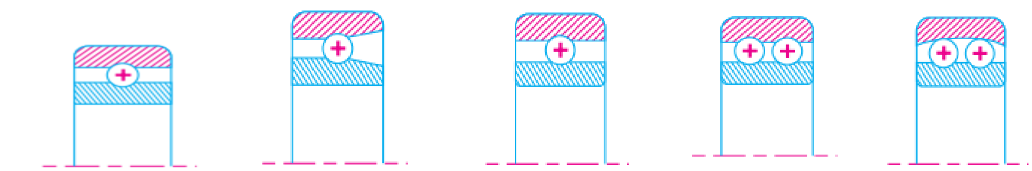
<tr><th>Q. No.</th><th>Sub Q. N.</th><th>Answer</th><th>Marking Scheme</th></tr>
<tr><td></td><td>b)</td><td>Explain with neat sketch different types of radial Ball bearings.</td><td></td></tr>
<tr><td></td><td>Sol.</td><td>Following are the various types of radial ball bearings:

(a) Single row deep groove. (b) Filling notch. (c) Angular contact. (d) Double row. (e) Self-aligning.
1. Single row deep groove bearing. : During assembly of this bearing, the races are offset and the maximum numbers of balls are placed between the races. The races are then centered and the balls are symmetrically located by the use of a retainer or cage.The deep groove ball bearings are used due to their high load carrying capacity and suitability for high running speeds. The load carrying capacity of a ball bearing is related to the size and number of the balls.
2. Filling notch bearing. : These bearings have notches in the inner and outer races which permit more balls to be inserted than in deep groove ball bearings. The notches do not extend to the bottom of the race way and therefore the balls inserted through the notches must be forced in position.
3. Angular contact bearing. These bearings have one side of the outer race cut away to permit the insertion of more balls than in a deep groove bearing but without having a notch cut into both races. This permits the bearing to carry a relatively large axial load in one direction while also carrying a relatively large radial load. The angular contact bearings are usually used in pairs so that thrust loads may be carried in either direction.
4. Double row bearing: These bearings may be made with radial or angular contact between the balls and races. The double row bearing is appreciably narrower than two single row bearings. The load capacity of such bearings is slightly less than twice that of a single row bearing
5. Self-aligning bearing: These bearings permit shaft deflections within 2-3 degrees. It may be noted that normal clearance in a ball bearing are too small to accommodate any appreciable misalignment of the shaft relative to the housing. If the unit is assembled with shaft misalignment present, then the bearing will be subjected to a load that may be in excess of the design value and premature failure may occur. Following are the two types of self-aligning bearings: (a) Externally self-aligning bearing, and (b) Internally self-aligning bearing.</td><td>1 Mark for each type

(Sketch & explanation)</td></tr>
</table>

Q. No.	Sub Q. N.	Answer	Marking Scheme
	c)	**Design a helical compression spring for a maximum load of 1000 N for a deflection of 25 mm using the value of spring index as 5. The maximum permissible shear stress for spring wire is 420 MPa and modulus of rigidity is 84 kN/mm^2.**	
	Sol.	Given : W = 1000 N ; δ = 25 mm ; C = D/d = 5 ; τ = 420 MPa = 420 N/mm2 ; G = 84 kN/mm2 = 84×10^3 N/mm^2 a) **Considering Curvature effect** : **1. Mean diameter of the spring coil** Let D = Mean diameter of the spring coil, and d = Diameter of the spring wire. We know that Wahl's stress factor, Wahl's factor K $= \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\ 5C-1}{4X5-4} + \frac{0.615}{5} = 1.31$ and maximum shear stress (τ), Maximum shear stress, $\tau = Kx\frac{8FC}{\pi d^2}$, $420 = 1.31x\frac{8x1000x5}{\pi d^2}$ d^2 = 16 677 / 420 = 39.7 or d = 6.3 mm ∴**Mean diameter of the spring coil, D** = C.d = 5 d = 5 × 6.3 = **31.5 mm Say 32 mm** **2. Number of turns of the coils** Let n = Number of active turns of the coils. We know that compression of the spring (δ), $\delta = \frac{8 \times F \times C^3 \times n}{G \times d}$, $25 = \frac{8 \times 1000 \times 5^3 \times n}{84 \times 10^3 \times 6.3} = 1.86$ n **n=25/1.86 = 13.44 ..Say 14 numbers of turns** Assuming square and grounded ends, total numbers of turns is given by**,** **n'=n+2=14+2=16 numbers of turns** **OR**	2 Mark 2 Mark

<table>
<tr><th>Q. No.</th><th>Sub Q. N.</th><th>Answer</th><th>Marking Scheme</th></tr>
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OR

b) **Without Considering Curvature effect** :

1 Mean diameter of the spring coil

Let D = Mean diameter of the spring coil, and d = Diameter of the spring wire.

and maximum shear stress (τ),

Maximum shear stress, $\tau = \frac{8\,FC}{\pi d^2}$, $420 = x\,\frac{8\,x1000\,x5}{\pi d^2}$

$d^2 = 12732.39/420 = 30.31$ or d = 5.5 mm say 6 mm

∴**Mean diameter of the spring coil, D** = C.d = 5 d = 5 × 6= **30 mm**

2 Number of turns of the coils

Let n = Number of active turns of the coils.

We know that compression of the spring (δ),

$\delta = \frac{8\times F\times C^3\times n}{G\times d}$, $25 = \frac{8\times1000\times5^3\times n}{84\,X\,10^3\times6} = 1.98$ n

n=25/1.86 = 12.62 ..Say 13 numbers of turns

</td><td>2 Mark

2 Mark</td></tr>
<tr><td></td><td>d)</td><td>

Explain selection of ball bearing using manufacturer's Catalogue.

</td><td></td></tr>
<tr><td></td><td>Sol.</td><td>

Procedure for selection of bearing from manufacturer's Catalogue.

The following procedure is followed in selecting the bearing from the manufacturer'scatalog.

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

</td><td>Correct Explanation /steps

4 Mark</td></tr>
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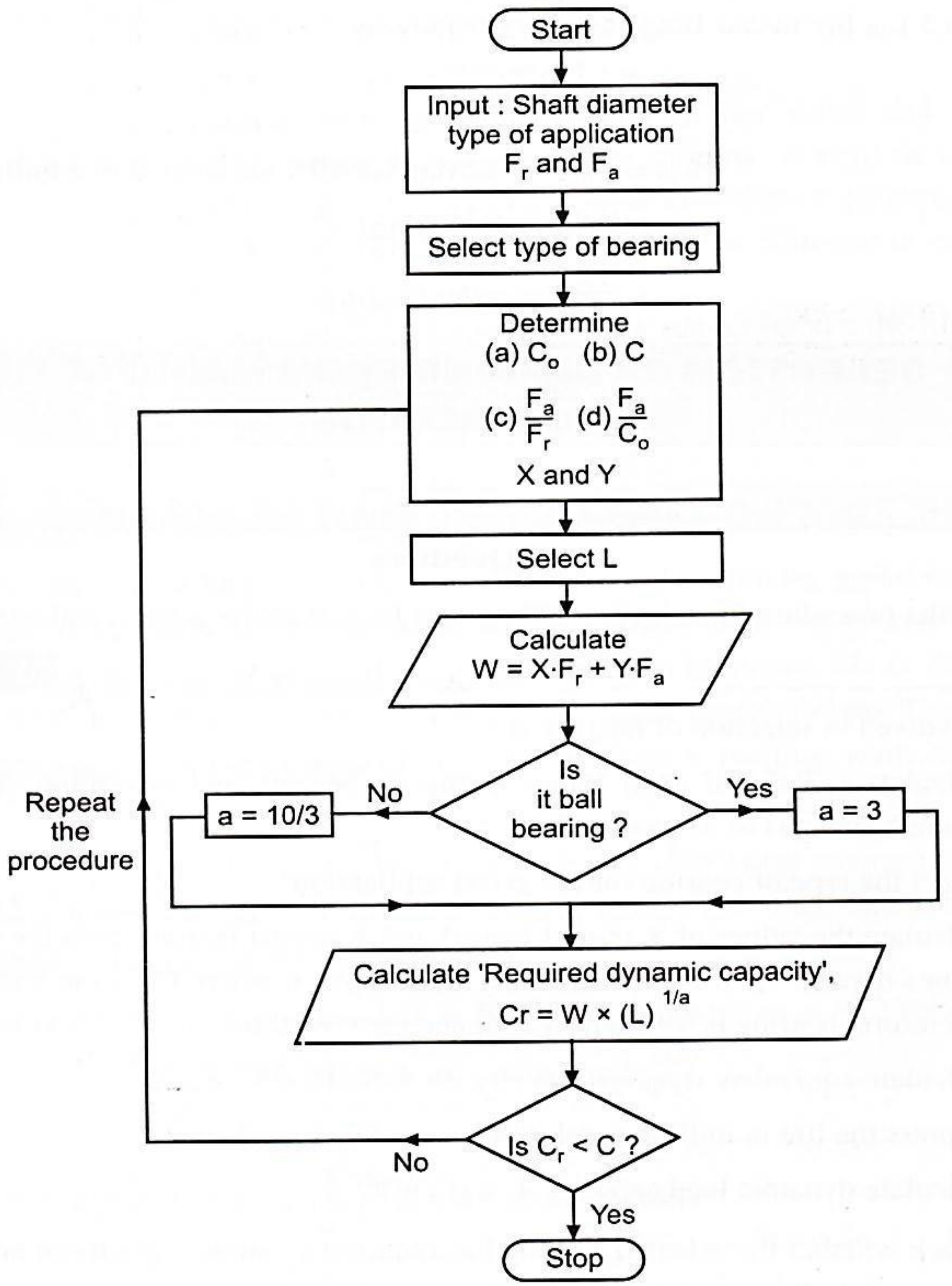
		of next series and repeat procedure from step-4.	
Q. No.	**Sub Q. N.**	**Answer**	**Marking Scheme**
		OR (flow chart) 	**OR** Flow Chart 4 Mark

Q. No.	Sub Q. N.	Answer	Marking Scheme
Que.4		**Attempt any TWO of the following**	**12 Marks**
	a)	**A screw jack is to lift a load of 80 kN through a height of 400 mm. The elastic strength of screw material in tension and compression is 200 MPa and in shear 120 MPa. The material for nut is phosphor bronze for which the elastic limit may be taken as 100 MPa in tension, 90 MPa in compression and 80 MPa in shear. The bearing pressure between the nut and the screw is not to exceed 18 N/mm^2. Design and draw the screw jack. Take FOS = 2**	
	Sol.	Solution. Given : W = 80 kN = 80 × 10 ^{3} N ; H1 = 400 mm = 0.4 m ; σet = σec = 200 MPa = 200 N/mm^2 ; τe = 120 MPa = 120 N/mm^2 ; σet(nut) = 100 MPa = 100 N/mm^2 ; σec (nut) = 90 MPa = 90 N/mm^2 ; τe(nut) = 80 MPa = 80N/mm^2 ; P_b = 18 N/mm^2 1. Design of screw for spindle Let d_c = Core diameter of the screw. Since the screw is under compression, ∴Load (W) $80 \times 10^3 = \frac{\pi}{4}(d_c)^2 \times \frac{\sigma_{ec}}{F.S.} = \frac{\pi}{4}(d_c)^2 \frac{200}{2} = 78.55\,(d_c)^2$ $(d_c)^2$ = 80 × 103 / 78.55 = 1018.5 or dc = 32 mm For square threads of normal series, dimensions of the screw selected is **Core diameter, dc = 38** mm Ans Nominal or outside diameter of spindle, do = 46 mm Ans. Pitch of threads, P = 8 mm Ans. Now let us check for principal stresses: We know that the mean diameter of screw d = (do + dc) / 2 = (46+38)/ 2 = 42 mm And tan α = p / π d = 8 / π x 42 = 0.0606 Assuming coefficient of friction between screw and nut, μ = tan ϕ = 0.14 ∴ Torque required to rotate the screw in the nut $T_1 = P \times \frac{d}{2} = W\tan(\alpha+\phi)\frac{d}{2} = W\left[\frac{\tan\alpha + \tan\phi}{1-\tan\alpha\tan\phi}\right]\frac{d}{2}$ $= 80\times10^3\left[\frac{0.0606+0.14}{1-0.0606\times0.14}\right]\frac{42}{2} = 340\times10^3$ N-mm	 Value dc : 1 Mark Calculation: 1 Mark

Q. No.	Sub Q. N.	Answer	Marking Scheme
		The given value of σ_c is equal to $\frac{\sigma_{ec}}{F.S.}$, i.e. $\frac{200}{2} = 100$ N/mm^2 We know that maximum shear stress, $\tau_{max} = \frac{1}{2}\left[\sqrt{(\sigma_c)^2 + 4\tau^2}\right] = \frac{1}{2}\left[\sqrt{(70.53)^2 + 4(31.55)^2}\right]$ $= \frac{1}{2} \times 94.63 = 47.315$ N/mm^2 The given value of τ is equal to $\frac{\tau_e}{F.S.}$, i.e. $\frac{120}{2} = 60$ N/mm^2. Since these maximum stresses are within limits, therefore design of screw for spindle is safe. **Design of Nut :** Let n = Number of threads in contact with the screwed spindle, h = Height of nut = $n \times p$, and t = Thickness of screw = $p/2 = 8/2 = 4$ mm Assume that the load is distributed uniformly over the cross-sectional area of nut. We know that the bearing pressure (p_b), $18 = \frac{W}{\frac{\pi}{4}\left[(d_o)^2 - (d_c)^2\right]n} = \frac{80 \times 10^3}{\frac{\pi}{4}\left[(46)^2 - (38)^2\right]n} = \frac{151.6}{n}$ $\therefore$ $n = 151.6/18 = 8.4$ say 10 threads **Ans.** and height of nut, $h = n \times p = 10 \times 8 = 80$ mm **Ans.** Now, let us check the stresses induced in the screw and nut. We know that shear stress in the screw, $\tau_{(screw)} = \frac{W}{\pi\, n.d_c.t} = \frac{80 \times 10^3}{\pi \times 10 \times 38 \times 4} = 16.15$ N/mm^2 ... ($\because t = p/2 = 4$ mm) and shear stress in the nut, $\tau_{(nut)} = \frac{W}{\pi\, n.d_o.t} = \frac{80 \times 10^3}{\pi \times 10 \times 46 \times 4} = 13.84$ N/mm^2 Since these stresses are within permissible limit, therefore design for nut is safe. Let D_1 = Outer diameter of nut, D_2 = Outside diameter for nut collar, and t_1 = Thickness of nut collar. First of all considering the tearing strength of nut, we have $W = \frac{\pi}{4}\left[(D_1)^2 - (d_o)^2\right]\sigma_t$ $80 \times 10^3 = \frac{\pi}{4}\left[(D_1)^2 - (46)^2\right]\frac{100}{2} = 39.3\left[(D_1)^2 - 2116\right]$ ... $\left[\because \sigma_t = \frac{\sigma_{et(nut)}}{F.S.}\right]$ or $(D_1)^2 - 2116 = 80 \times 10^3 / 39.3 = 2036$ $\therefore$ $(D_1)^2 = 2036 + 2116 = 4152$ or $D_1 = 65$ mm **Ans.**	Calculation of No. of Threads & Height of Nut : 1 Mark Dia& thickness of nut : 2 Mark

Q. No.	Sub Q. N.	Answer	Marking Scheme
		Now considering the crushing of the collar of the nut, we have $$W = \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right]\sigma_c$$ $$80 \times 10^3 = \frac{\pi}{4}\left[(D_2)^2 - (65)^2\right]\frac{90}{2} = 35.3\left[(D_2)^2 - 4225\right] \quad ...\left[\sigma_c = \frac{\sigma_{c(nut)}}{F.S.}\right]$$ or $(D_2)^2 - 4225 = 80 \times 10^3 / 35.3 = 2266$ $\therefore$ $(D_2)^2 = 2266 + 4225 = 6491$ or $D_2 = 80.6$ say 82 mm **Ans.** Considering the shearing of the collar of the nut, we have $$W = \pi D_1 \times t_1 \times \tau$$ $$80 \times 10^3 = \pi \times 65 \times t_1 \times \frac{80}{2} = 8170\, t_1 \quad ...\left[\tau = \frac{\tau_{c(nut)}}{F.S.}\right]$$ $\therefore$ $t_1 = 80 \times 10^3/8170 = 9.8$ say 10 mm **Ans.**	**01mark**
	b)	**Design a cast iron protective type flange coupling to transmit 15 KW at 900 rpm from an electric motor to compressor.** **use following permissible stress** **shear stress for shaft ,bolt and key = 40 Mpa** **crushing stress for Bolt and key= 80 Mpa** **shear stress for cast iron = 8 Mpa.**	
	Sol.	Given : P = 15 kW = 15 × 10^3 W ; N = 900 r.p.m. ; τs = τb = τk = 40 MPa = 40 N/mm2 ; σcb = σck = 80 MPa = 80 N/mm2 ; τc = 8 MPa = 8 N/mm^2 The protective type flange coupling is designed as discussed below : **1) Design for shaft** First of all, let us find the diameter of the shaft (d). We know that the torque transmitted by the shaft, **T = P x 60/2π N** = (15 x 10^3 x 60)/ (2 π x900) =159.15 Nm = 159.15 x 10^3Nmm T = π/16 x τ xd^3 , 159.15 x 10^3 = π/16 x 40 xd^3 , d^3=20263,**60 d=27.26 mm say 28 mm** **2) Design of Hub:** hub Outside dia. of hub **, D=2d =2x28=56mm** Length of Hub **l= 1.5 d = 1.5 x 28 = 42 mm** The shear stress induced in the hub is given by T = π/16 x τc x d^3 x (1-k^4) Where K = d/D = 0.5 159.15 x 10^3 = π/16 x τc x56 ^{3} x(1-0.5^4) τc =4.923 < 8 Mpa Hence, **Hub is safe against shear failure.**	Design of shaft : 2 Mark Design of Hub : 2 Mark

Q. No.	Sub Q. N.	Answer	Marking Scheme
		3)Design of Flange: Thickness of flange tf= 0.5 d = 0.5 x 28 = 14 mm Thickness of Protective flange Tp= 0.25 x d = 0.25 x 28 = 7 mm Dia of Bolt circle: D1 = 3 d = 3 x 28 = 84 mm Dia of Outer Falnge : D2 = 4d = 4 x 28 = 112 mm Dia of flange recess D3 = 1.1 D = 1.1 X 56 = 61.6 mm Direct stress induced in the flange at a junction with hub is Tf = (T/D/2) / (ΠxDxtf) $= 2T / (\Pi x D^2 x tf) = (2X\ 159.15 \times 10^3) / (\Pi \times 56^2 \times 14)$ $=2.3\ N/mm^2 < 8\ N/mm^2$ **Flange is safe against shear failure.**	Design of flange: 2 Mark
	c)	**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.**	
	Sol.	Outside diameter of hollow shaft (do) = Diameter of solid shaft (d) For same material: Density of solid = density of hollow shaft $L_S = L_H$,di = inside diameter of hollow shaft = 0.75 do $,k = \frac{di}{do}$ =0.5 **I)Comparison of weight:** We know that weight of a hollow shaft $W_{H=}$ Cross sectional area x Length x Density $= \pi/4[\ (do)^2 - (di)^2\ ]X$ Length x Density…………I and Weight of the solid shaft $W_{S=}$ Cross sectional area x Length x Density $= [\pi/4\ x\ \ (d)^2]\ X$ Length x Density……………….. Since both the shafts have the same material and length, therefore by dividing equation ***(i)*** by equation***(ii)***, we get	Comparison of weight: 2Mark

<table>
<tr><th>Q. No.</th><th>Sub Q. N.</th><th>Answer</th><th>Marking Scheme</th></tr>
<tr><td></td><td></td><td>

$$\frac{WH}{WS} = \frac{= [\,(do)^2 - (di)^2\,]}{(d)^2}$$

As K= di/do

$\frac{WH}{WS} = 1 - K^{2}$ =1- $(0.5)^2$ =0.75

W_H=0.75 W_S …………………Ans

II) Comparison of Strength:

Strength of hollow shaft $T_H = \frac{\pi}{16}\tau\, do^3\, x\,(1-k^4)$

Strength of Solid shaft $Ts = \frac{\pi}{16}\tau\, do^3$

$$\frac{TH}{TS} = \frac{[\,(do)^3\,(1-K^4)\,]}{do^3}$$

$=(1-k^4) = =(1-0.5^4)$

T_H=0.93 T_S …………………. Ans

III) Comparison of Stiffness:

Stiffness of hollow shaft $S_H = \frac{G}{L} X \frac{\pi}{32}(\,do^4 - di^4)$

Stiffness of Solid shaft $Ss = \frac{G}{L} X \frac{\pi}{32} do^4$

$$\frac{SH}{SS} = \frac{(\,do^4 - di^4)}{do^4}$$

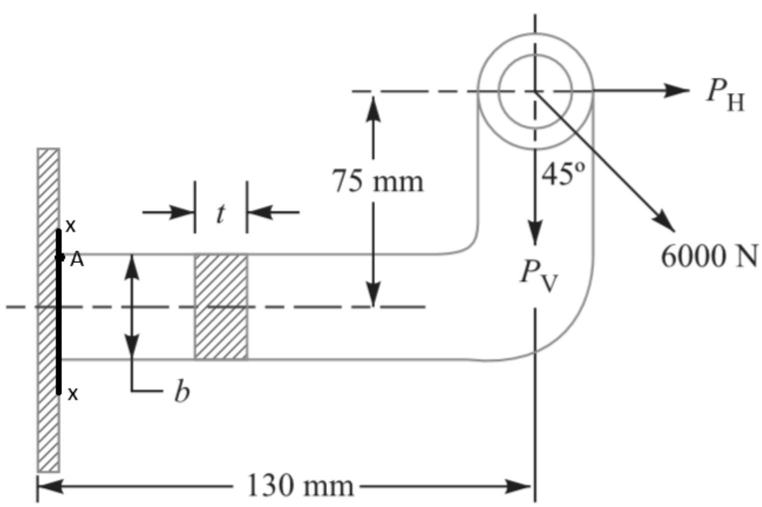
$=(1-k^4) = =(1-0.5^4)$

S_H=0.93 S_S ……………… Ans

</td><td>Comparison of strength: 2 Mark

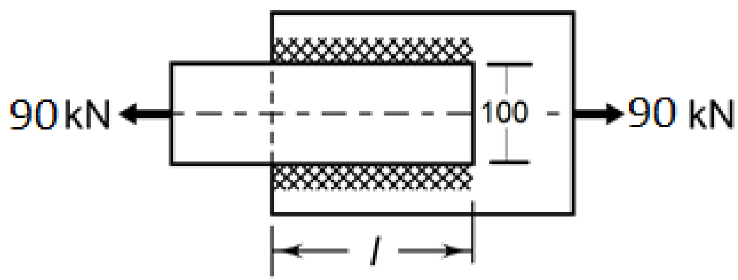
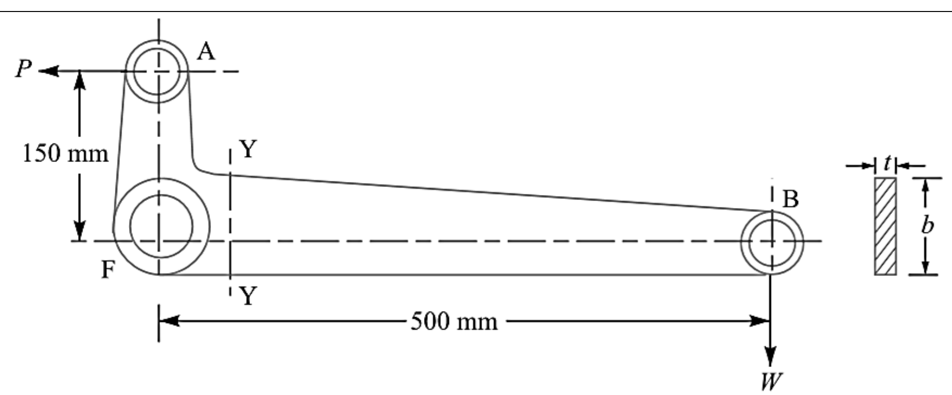
Comparison of stiffness: 2 Mark</td></tr>
<tr><td>Que.5</td><td></td><td>Attempt any <u>TWO</u> of the following</td><td>12 Marks</td></tr>
<tr><td></td><td>a)</td><td>Explain general considerations in Machine Design.</td><td></td></tr>
</table>

Q. No.	Sub Q. N.	Answer	Marking Scheme
	Sol.	**1. <u>Type of load and stresses caused by the load.</u>** The load, on a machine component, may act in several ways due to which the internal stresses are set up. : a) Compression- Applying forces to both ends b) Tension- Forces applied in the opposite direction c) Shear- Sliding forces that are applied in the opposite direction d) Bending- Force off-centered e) Torsional- Twisting force f) Combination – Combination of any loads **2. <u>Motion of the parts or kinematics of the machine.</u>** The successful operation of any machine depends largely upon the simplest arrangement of the parts which will give the motion required. The motion of the parts may be: a) Rectilinear motion which includes unidirectional and reciprocating motions. b) Curvilinear motion which includes rotary, oscillatory, and simple harmonic. c) Constant velocity d) Constant or variable acceleration. **3. <u>Selection of materials</u>** The following factors should be considered while selecting the material: a) Availability b) Cost c) Mechanical properties d) Manufacturing Considerations **4. <u>Form and size of the parts</u>**: Form and size of the parts can decide manufacturing process and cost of the product. **5. <u>Frictional resistance and lubrication:</u>** There is always a loss of power due to frictional resistance and it should be noted that the friction of starting is higher than that of running friction. It is, therefore, essential that careful attention must be given to the matter of lubrication of all surfaces which move in contact with others, whether in rotating, sliding, or rolling bearings. **6. <u>Ergonomic Considerations</u>**: User friendly machine operations, ease of control, proper force required to operate and safety for operating. **7. <u>Aesthetic considerations</u>**: Considerations related to the beauty of the product to attract the customer. **8. <u>Standardization</u>**: Use of standard parts makes product economical for design and maintenance with improved quality. **9. <u>Safety of operation</u>**: Designer should always provide safety devices for the safety of the operator. **10. <u>Manufacturing capability</u>:** Best manufacturing processes can provide quality but adds to the cost. Therefore appropriate manufacturing processes and available manufacturing set up must be used to the best possible effect so that cost of manufacturing must be limited. **11. <u>The number of components to be manufactured</u>:** Suitability of design for job production, batch production and mass production. **12. <u>Cost of manufacturing:</u>** The aim of the design engineer under all conditions should be to reduce the manufacturing cost to the minimum with optimum quality. **13.<u>Design for assembly</u>:** Designer must consider the ease of assembly and disassembly of the product to be manufactured.	Any 6 considerations 1 Mark each

Q. No.	Sub Q. N.	Answer	Marking Scheme
	b)	**Define factor of safety and state the factors affecting on selection of factor of safety for ductile and brittle material.**	
	Sol.	**Factor of Safety:** It isdefined as the ratio of failure stress to allowable or working stress. OR it is the ratio of failure load to allowable or working load. – Allowable stress $\boldsymbol{\sigma} = \frac{Syt}{fs}$ for ductile material & – Allowable stress $\boldsymbol{\sigma} = \frac{Sut}{fs}$ for brittle material **Selection of factor of safety (*fs*) depends upon following factors-** i) **Effect of failure**: Time, finance, danger to human life decides the value of fs. ii) **Type of load** : Static load– low *fs*, Dynamic load and Impact Load-high *fs*, iii) **Degree of accuracy in force analysis**: Accurate - low *fs,* iv) **Material of component**: Homogeneous material or ductile low *fs,* v) **Reliability of component**: *fs*increases for higher reliability of component. vi) **Cost of component**: Cost of component is directly proportional to value of factor of safety. vii) **Testing of machine element**: Actual testing conditions known - low *fs* viii) **Service conditions**: Operating conditions like temp, corrosion, humidity add to value of factor of safety. ix) **Quality of Manufacture:** High Mfg. quality leads to low *fs* **Selection of value of factor of safety is normally low for ductile materials and high for brittle materials.** It is essentially a compromise between the associated additional cost and weight and the benefit of increased safety or/and reliability.	1 Mark Any 5 factors 1 Mark each
	c) Sol.	 	

Q. No.	Sub Q. N.	Answer	Marking Scheme
		P = 6000 N , Depth of cross section b= 2 t where t= thickness of cross section ***Step I*** Permissible stress $\sigma_{max} = Permissible\ stress\ of\ material = 60\ MPa$	1 Mark
		Step II ***Calculation of direct and bending stress*** The stress is maximum at point A in the section XX. The point is subjected to combined bending and direct tensile stresses. The force P is resolved into two components – horizontal component P_H and vertical component P_v $P_H = Horizontal\ component\ of the\ load = P\sin 45^0 = 6000\times\sin 45^0 =$ 4242.64 N $P_v = Vertical\ component\ of the\ load = P\cos 45^0$ = $6000\times\cos 45^0 = 4242.64$ N	1 Mark
		The bending moment at section XX is given by M_b=$P_H\times 75 + P_v\times 130 = 4242.64\times 75 + 4242.64\times 130$ =318198 +551543.2 **M_b= 869741.2 N-mm**	1 Mark
		$\sigma_b = \frac{M_b y}{I} == \frac{869741.2\times(t)}{\left(\frac{t(2t)^3}{12}\right)} = \frac{1304611.8}{(t^3)}\ N/mm^2$ The direct tensile stress due to horizontal component P_H is given by $\sigma_t = \frac{P_H}{A} == \frac{4242.64}{(2t^2)} = \frac{2121.32}{(t^2)} N/mm^2$	1 Mark
		Step III ***Calculation of dimensions of cross section*** *The resultant tensile stress* $\sigma_{max} = \sigma_b + \sigma_t$ $60 = \frac{1304611.8}{(t^3)} + \frac{2121.32}{(t^2)}$ $t^3 - 35.35\,t - 21743.53 = 0$ *by trial and error method* thickness **= 28.4 mm = 30 mm**	1 Mark
		depth **b= 2t = 60 mm……………………..Ans.**	1 Mark

Q. No.	Sub Q. N.	Answer	Marking Scheme
Que.6		**Attempt any TWO of the following**	**12 Marks**
	a)	Diameter of wire = d= 10 mm , number of turns = N= 10, D = 120 mm, Axial Load P = 200 N, G = 80000 N/mm^2 **i) Shear stress (τ) induced in spring neglecting effect of stress concentration** $\tau = \left(1 + \frac{d}{2D}\right)\frac{8PD}{\pi d^3} \quad \therefore \quad \tau = \left(1 + \frac{10}{2 \times 120}\right)\frac{8 \times 200 \times 120}{\pi \times 10^3} = 61.115 N/mm^2$ **ii) Deflection (δ) in the spring** $\delta = \frac{8PD^3N}{Gd^4} = \frac{8 \times 200 \times 120^3 \times 10}{80000 \times 10^4} = \mathbf{34.56\ mm}$ **iii) Stiffness (k) of the spring** $k = \frac{P}{\delta} = \frac{200}{34.56} = \mathbf{5.8\ N/mm}$ **iv) Strain energy stored (U) in the spring** $U = \frac{PD}{2} = \frac{200 \times 34.56}{2} = 3456$ N-mm = **3.456 N-m**	1 Mark 1.5Mark 1.5Mark 1Mark 1Mark
	b)	i) **Stiffness / Spring rate** The stiffness of the spring (k) is defined as the force required to produce unit deflection. Therefore $k = \frac{P}{\delta}$ where, k = stiffness of the spring (N/mm) P = axial spring force (N) δ = axial deflection of the spring corresponding to the force P (mm)	1.5 Mark 1.5 Mark

Q. No.	Sub Q. N.	Answer	Marking Scheme
		ii) Ans: Figure shows the weld arrangement  **Given:** P = 90 KN, permissible shear stress τ = 60 N/mm^2, size of the weld= plate thickness h= 10 mm Length of the weld (l) , Maximum load (P) that double parallel fillet weld can carry, is given by P = 1.414 h.l.τ. $90 \times 10^3 = 1.414 \times 10 \times l \times 60$ $l = \frac{90 \times 10^3}{1.414 \times 10 \times 60} = 106.20\ mm = 107\ mm$ Adding 12.5 mm of length for starting and stopping of the weld run, the length of the weld is given by, $l = 107 + 12.5 = \mathbf{119.5\ mm}$**……………………………………………Ans**	1Mark 1Mark 1Mark
	c)	 	

Q. No.	Sub Q. N.	Answer	Marking Scheme
		Given: FB = 500 mm, FA= 150 mm, Load W= 5000 N $\sigma_t = 80 N/mm^2$ $\tau = 60 N/mm^2$ $P_b = 10 N/mm^2$	1Mark
		i) Effort applied P: Take moments about fulcrum F, we have W×500 = P ×150 $P = \frac{5000 \times 500}{150} = 16666.67\ N$ And reaction at fulcrum pin F is given by $\boldsymbol{R = \sqrt{W^2 + P^2} = \sqrt{5000^2 + 16666.67^2} = 17400.51\ N}$	1Mark
		ii) Design of fulcrum pin: let d = diameter of fulcrum pin, l = length of the fulcrum pin and let length of pin l=1.25 d Considering bearing pressure at fulcrum pin $17400.51 = d \times l \times P_b$ $17400.51 = d \times 1.25d \times 10$ $d = 37.31\ mm = 38\ mm$ Checking fulcrum pin for shear failure, $\tau = \frac{17400.51}{2 \times \frac{\pi}{4} \times 38^2}$ $\therefore\ \boldsymbol{\tau = 7.67\ N/mm^2}$ Since shear stress is induced in fulcrum pin is less than given value 60 MPa, hence fulcrum pin is safe in shear. A brass bush of 3 mm thickness is pressed into boss of fulcrum as a bearing. Diameter of hole in the lever = d + 2× $3 = 38 + 2 \times 3 = 44\ mm$	2Marks

Q. No.	Sub Q. N.	Answer	Marking Scheme
		ii) <u>Dimensions of lever:</u> Considering weakest section of failure at Y-Y Let t = thickness of lever b = width depth of lever at YY and b= 3t Maximum Bending Moment at Y-Y $$M_b = 5000(500-50) = 225000\ \ N-mm$$ $$\sigma_b = \frac{M_b \times y}{I} = \frac{2250000 \times 1.5\,t}{\frac{t(3t)^3}{12}}$$ $$t^3 = \frac{1500000}{80} \therefore\ \ \boldsymbol{t} = 26.56 = \mathbf{28\ mm}$$ **Therefore depth of cross section b= 3 t =**$28 \times 3 = \mathbf{84\ mm}$	2 Marks