Important Instructions to examiners:

1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling errors should not be given more Importance (Not applicable for subject English and Communication Skills.
4) While assessing figures, examiner may give credit for principal components indicated in the figure. The figures drawn by candidate and model answer may vary. The examiner may give credit for any equivalent figure drawn.
5) Credits may be given step wise for numerical problems. In some cases, the assumed constant values 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. No.	Sub Q. No.	Answer	Marking Scheme
01.	(a)	**Define machine design.** **Machine design** is the process of selection of the materials, shapes, sizes and arrangements of mechanical elements so that the resultant machine will perform the prescribed task. OR **Machine Design** is the creation of new and better machines and improving the existing ones.	2
	(b)	**Give the composition of** **(i) FeE220:** Steel having yield strength of 220 N/mm^2. **(ii) 20C8** : Carbon steel containing 0.15 to 0.25 percent (0.2 percent on average) carbon and 0.60 to 0.90 percent (0.80 percent on average) manganese.	1 1
	(c)	**State four types of loads acting on machine elements.** (i) Dead or steady load (ii) Live or variable load (iii) Suddenly applied or shock load (iv) Impact load	½ ½ ½ ½
	(d)	**What do you mean by creep?** When a machine part is subjected to a constant stress at high temperature for a long period of time, it will undergo a slow and permanent deformation called 'creep'. This property is considered in designing internal combustion engines, boilers and turbines.	2

	(e)	**Define Ergonomics.** **Ergonomics** is defined as the scientific study of the **man – machine working environment** relationship and the application of anatomical, physiological, psychological principles to solve the problems arising from this relationship.	2
	(f)	**Give two applications of knuckle joint.** (i) A knuckle joint is used to connect two rods which are under the action of tensile loads. However, if the joint is guided, the rods may support a compressive load. (ii) Its use may be found **in the link of a cycle chain**, **tie rod joint of roof truss**, **valve rod joint with eccentric rod, pump rod joint, tension link in bridge structure and lever and rod connections of various types.**	**2 mark for any 2 application**
	(g)	**Define following terms of spring:** **(i) Spring rate**: The spring rate is defined as the load required per unit deflection of the spring. It is also known as spring stiffness or spring constant. Mathematically, Spring rate, $k = W / \delta$ Where, W = Load δ = Deflection of the spring **(ii) Spring index:** The spring index is defined as the ratio of the mean diameter of the coil to the diameter of the wire. Mathematically, Spring index, $C = D / d$ Where, D = Mean diameter of the coil d = Diameter of the wire	1 1
	(h)	**How do you express the life of bearings?** • The life of an individual bearing is defined as the total number of revolutions (or the number of hours at a given constant speed) which the bearing can complete before the evidence of fatigue failure develops on the balls or races. • The bearing life can be defined by **rating life.** • The rating life of a group of apparently identical bearing is defined as the number of revolutions (or the number of hours at a given constant speed) that 90 percent of a group of bearings will complete or exceed before the first evidence of fatigue failure develops. It is also known as **L_{10} life.**	1 1
	(i)	**Draw the different thread profiles used for power screws.(Draw any four thread profiles)**	1

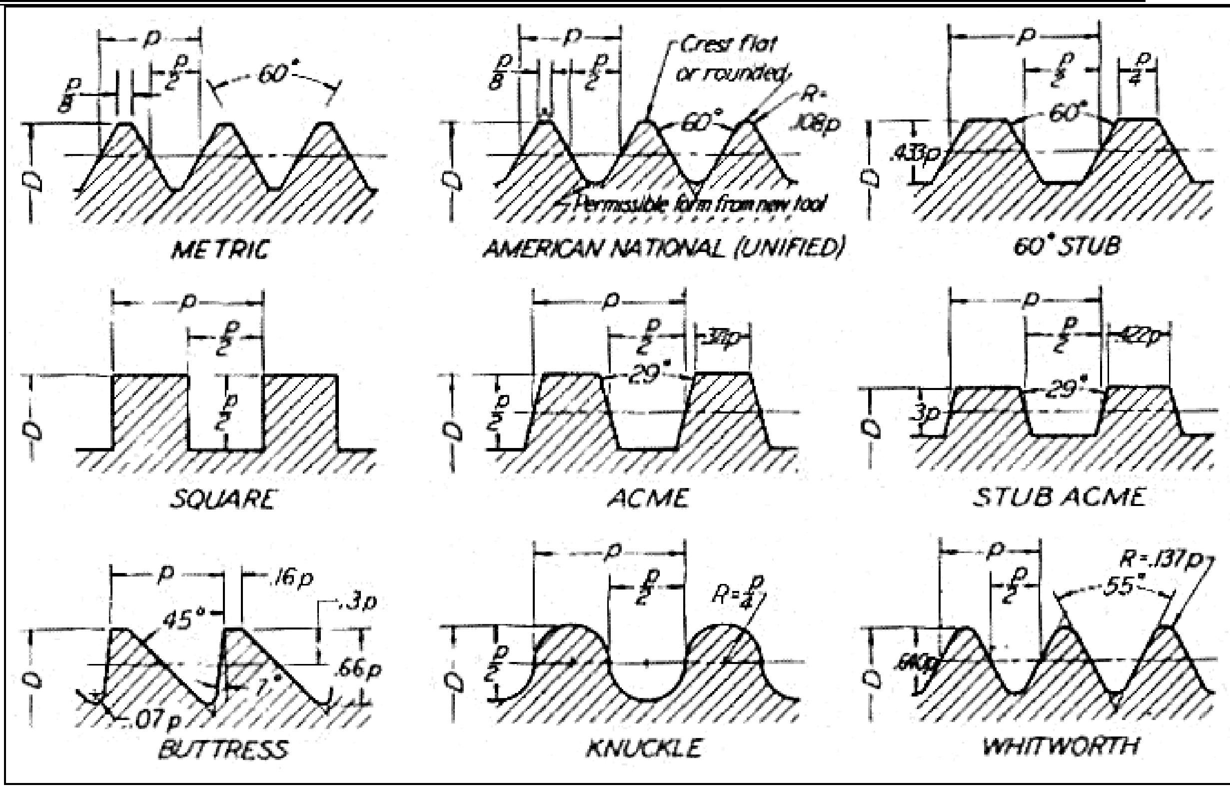


Fig. Types of thread profiles

½ mark for each

(j) State four types of keys.

(i) Sunk key: Rectangular sunk key, Square sunk key and Parallel sunk key
(ii) Gib-head key
(iii) Feather key
(iv) Woodruff key
(v) Saddle key: Flat saddle key, Hollow saddle key
(vi) Tangent key
(vii) Round key

Any four types of keys

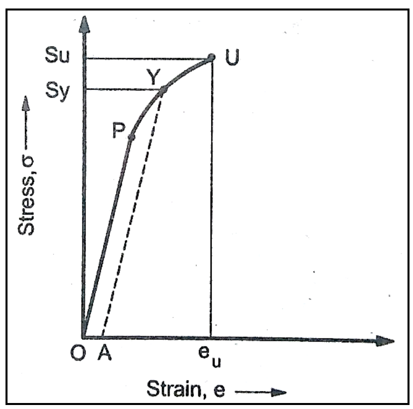
½ mark for each

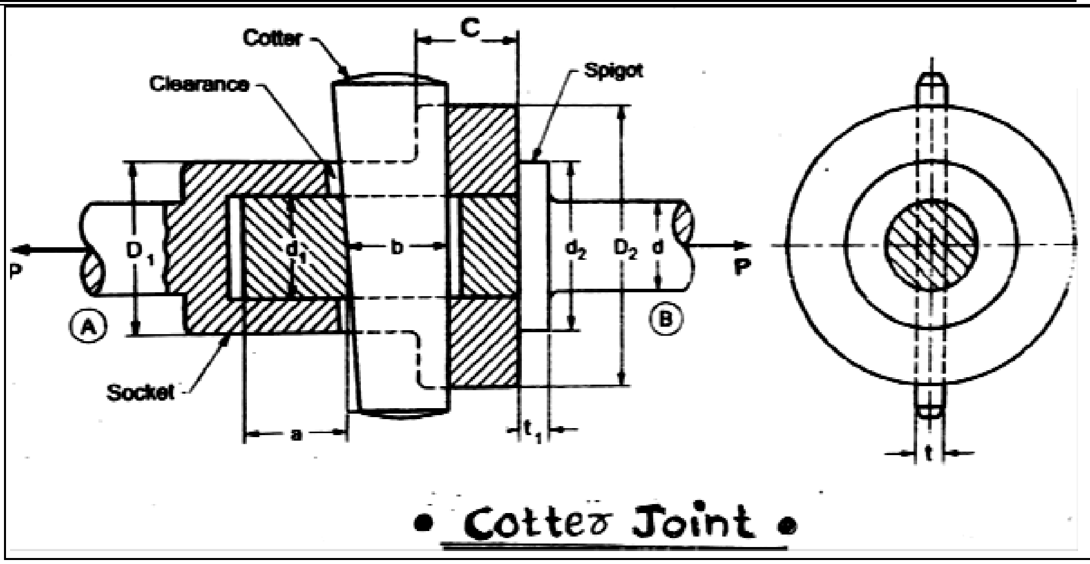
(k) Give two examples, where screwed joints are preferred over welded joints.

(i) Cylinder head of the engine.
(ii) Machine foundation.
(iii) Assembly of fan, couplings.
(iv) Connect two bogies of the train with the turn buckle.
(v) Structural bridges, pressure vessels, fly press
(vi) Assembly of crank shaft and connecting rod.

Any two examples

1mark for each

	(l)	**State any four applications of rolling contact bearings.** (i) Industrial and automotive gear boxes. (ii) Electric motors and machine tool spindles. (iii) Small size centrifugal pumps. (iv) Automobile front and rear axles.	**½ mark for each**
	(m)	**What are the requirement of a good coupling?(Any four)** A good coupling should have the following requirements: (i) It should be easy to connect and disconnect. (ii) It should transmit the full power from one shaft to another shaft without losses. (iii) It should hold the shafts in perfect alignment. (iv) It should reduce the transmission of shock loads from one shaft to another shaft. (v) It should have no projecting parts.	**½ mark for each**
	(n)	**Draw stress – strain diagram for brittle material.**  **Fig. Stress vs Strain diagram for Brittle materials**	**2**
02.	**(a)**	**Explain various failures to be considered in designing a cotter joint along with the necessary sketches and strength equations.**	

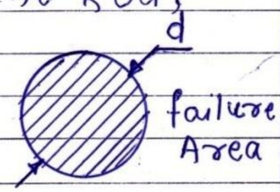
		 It consist of 3 elements: i. Socket ii. Spigot iii. Cotter Where, d= End diameter of rod d_1= Diameter of spigot/Inside diameter of socket d_2= Diameter of spigot collar D_1= Outer diameter of socket D_2= Diameter of socket collar C=Thickness of socket collar t_1= Thickness of spigot collar t= thickness of cotter b= Mean width of cotter a= Distance of end of slot to the end of spigot P= Axial tensile/compressive force σ_t, σ_c, τ= Permissible tensile, compressive, shear stress for the component material	**1 mark for fig**

Design Procedure

① Design of dia of rod (d)

Considering tensile failure of the Rod,

$$\boxed{\sigma_t = \frac{P}{A} = \frac{P}{\frac{\pi}{4}\times d^2}}$$



$\frac{1}{2}$ mark

½ mark

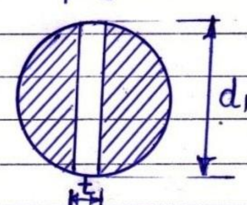
② Design of dia of Spigot (d_1) & thickness of cotter (t)

ⓐ By emperical Relation

$$\boxed{t = 0.3d}$$

ⓑ Considering tensile failure of spigot.

$$\boxed{\sigma_t = \frac{P}{\left[\frac{\pi}{4}d_1^2 - d_1 t\right]}}$$

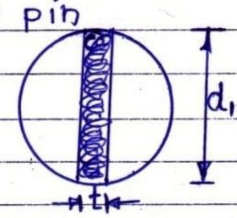


$\frac{1}{2}$ mark

½ mark

ⓒ Considering crushing failure of spigot area which is in connection with cotter pin

$$\boxed{\sigma_c = \frac{P}{d_1 t}}$$



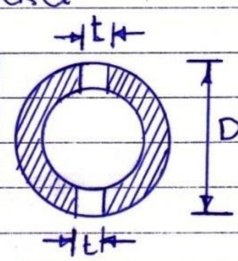
$\frac{1}{2}$ mark

½ mark

③ Design of outside diameter of socket (D_1)

Considering tensile failure of socket

$$\boxed{\sigma_t = \frac{P}{\left[\frac{\pi}{4}(D_1^2 - d_1^2) - (D_1 - d_1)\right]\times t}}$$



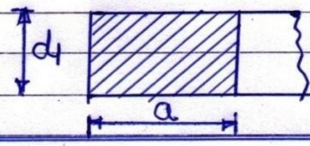
1 mark

1 mark

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. $\frac{1}{2}$ mark

½ mark

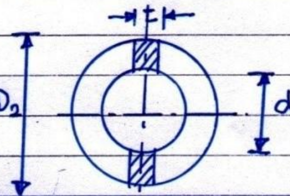
$$\boxed{\tau = \frac{P}{2d_1 a}}$$



⑤ Design of Dia. of socket collar (D_2)
Considering crushing failure of socket collar as shown in fig. 1 mark

1 mark

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

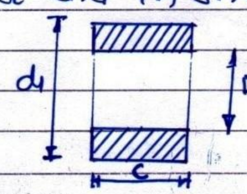


⑥ Design of thickness of socket collar (c)
Considering failure of socket end in shearing

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

1 mark

1 mark

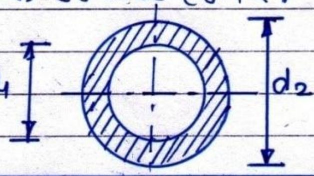


⑦ Design of Dia. of socket collar (d_2)
Considering crushing failure of spigot collar at the contact area between socket collar

1 mark

1 mark

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

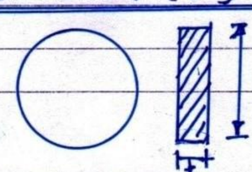


⑧ Design of thickness of spigot collar (t_1)

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

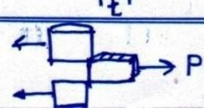
$\frac{1}{2}$ mar

½ mark



⑨ Design of Width of cotter (b)

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



$\frac{1}{2}$ mark

½ mark

In practice, sometimes the following proportions in terms of the diameter of the rod (d), are used when all components of the cotter joint are made of steel.

$d_1 = 1.21$ d; $d_2 = 1.5$ d;

$D_1 = 1.75$ d; $D_2 = 2.4$ d;

$t = 0.3$ d; $b = 1.6$ d;

$t_1 = 0.45$ d; $a = c = 0.75$ d

Knowing the dimensions, the various stresses induced in the components are calculated and ensured that all are within the permissible limits.

(b) **State the theories of elastic failure. Explain maximum normal stress theory and maximum shear stress theory with equations.**

The principal theories of failure for a member are as follows:(Any four)

2 marks

½ mark each

(i) Maximum principal or normal stress theory

(ii) Maximum shear stress theory

(iii) Maximum principal or normal strain theory

(iv) Maximum strain energy theory

(v) Maximum distortion energy theory

Maximum normal stress theory

- According to this theory, the elastic failure occurs when the greatest principal stress reaches the elastic limit value in a simple tension test irrespective of the value of other two principal stresses.
- Taking factor of safety (F. S.) into consideration, the maximum principal or normal stress (σ_t) is given by,

$$\sigma_t = \sigma_{yt} / F.S. \quad \textbf{(for ductile materials)}$$

3

$$\sigma_t = \sigma_u / F.S. \quad \textbf{(for brittle materials)}$$

where, σ_{yt} = Yield point stress in tension as determined from simple tension test

σ_u = Ultimate stress

- This theory ignores the possibility of failure due to shear stress, therefore it is not used for ductile materials.

- However, for brittle materials which are relatively strong in shear but weak in tension and compression, this theory is generally used.
- This theory is also known as **maximum principal stress theory** or **Rankine's theory.**

Maximum Shear Stress Theory

- According to this theory, the failure or yielding occurs at a point in a member when the maximum shear stress reaches a value equal to the shear stress at yield point in a simple tension test. Mathematically, **3**

$$\tau_{max} = \tau_{yt} / F.S.$$

where, τ_{max} = Maximum shear stress

τ_{yt} = Shear stress at yield point as determined from simple tension test

F. S = Factor of safety

- Since the shear stress at yield point in a simple tension test is equal to one half the yield stress in tension, therefore

$$\tau_{max} = \sigma_{yt} / (2 \times F.S.)$$

- This theory is mostly used for designing members of ductile materials.
- This theory is also known as **Guest's theory or Tresca's theory.**

(c) **(i) State and describe in brief about four ergonomic considerations in the designing of machine elements.**

The different areas covered under the ergonomics are:

1. Communication between the man (user) and the machine.
2. Working environment.
3. Human anatomy and posture while using the machine.
4. Energy expenditure in hand and foot operations.

1 mark for each cosideration

Communication between man and machine

- The machine has a display unit and a control unit.
- A man (user) receives the information from the machine display through the sense organs.
- He (or she) then takes the corrective action on the machine controls using the hands or feet.
- This man-machine closed loop system in influenced by the working environmental factors such

as: lighting, noise, temperature, humidity, air circulation, etc.

Working Environment

- The working environment affects significantly the man-machine relationship.
- It affects the efficiency and possibly the health of the operator.
- The major working environmental factors are: Lighting, Noise, Temperature, Humidity and air circulation.

Ergonomics Considerations in Design of Controls

- The control devices should be logically positioned and easily accessible.
- The control operation should involve minimum and smooth moments.
- The control operation should consume minimum energy.
- The controls should be painted in proper colour to attract the attention.

Ergonomics Considerations in the Design of Displays

- The scale should be clear and legible.
- The size of the numbers or letters on the scale should be taken appropriate.
- The pointer should have a knife-edge with a mirror in a dial to minimize the parallax error while taking the readings.
- The scale should be divided in a linear progression such as 0 – 10 – 20 – 30… and not as 0 – 5 – 25 – 45…..
- The number of subdivisions between the numbered divisions should be as less as possible.
- The numbering should be in clockwise direction on a circular scale, from left to right on a horizontal scale and from bottom to top on a vertical scale.

(ii) How will select bearing from manufacturer catalogue?

The following steps must be adopted in selecting the bearing from the manufacturer's catalogue:

1. Calculate the radial and axial load reaction (F_a and F_r) acting on the bearing.
2. Decide the diameter of the shaft on which the bearing is to be mounted.
3. Select the proper size of bearing suitable for given application, specified with speed and available space. **4 marks**
4. Find the basic static rating C_o of the selected bearing from the catalogue.
5. Calculate the ratio (F_a / VF_r) and (F_a / C_o).
6. Find the value of x and y i. e. radial and thrust factor from the catalogue. These values depend upon (F_a / VF_r) and (F_a / C_o).

7. Find the value of load factor or application factor 'K_a' from the catalogue.
8. Calculate the equivalent dynamic load by using relation,

$$P_e = (XVF_a + YF_a) K_a$$

9. Calculate the approximate bearing life in hours from the type of bearing, operation and type of machinery that depends upon application.
10. Calculate the required basic dynamic capacity for the bearing by using relation,

$$L_{10} = (C / P_e)^a$$

or

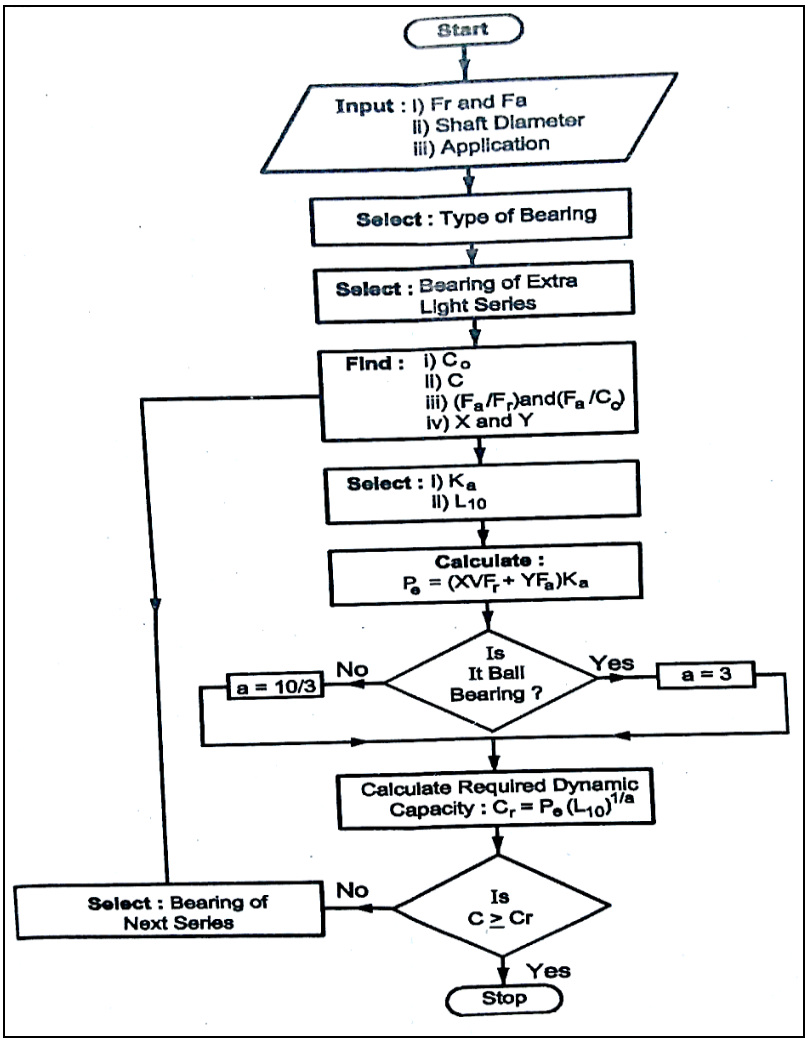
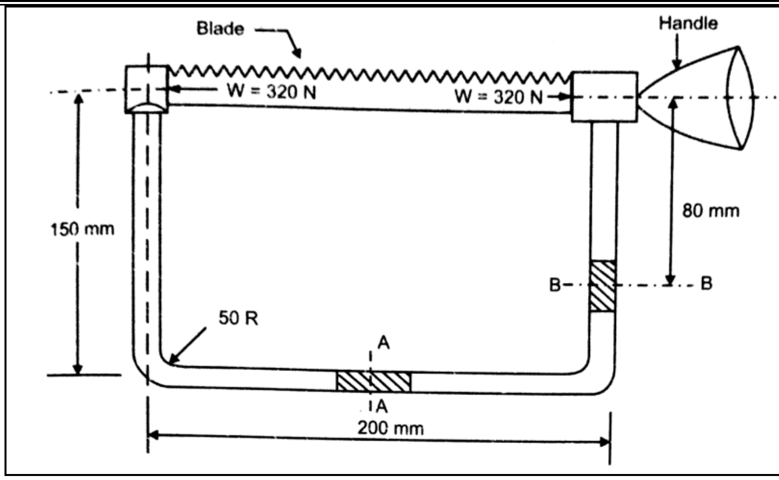


Fig. Procedure for selection of bearing from manufacturer's catalogue

4 marks

03.

a)



Solution :- Given data ⇒ $W = 320\ N$, $\sigma_{yt} = 360\ N/mm^2$, $FOS = 4$, $b = 2.5t$ **1**

$$\left[\begin{array}{c}\text{Allowable stress} \\ \text{in tension}\end{array}\right] = \frac{\text{Yield pt stress}}{FOS} = \frac{\sigma_{yt}}{FOS} = \frac{360}{4} = 90\ N/mm^2$$ **1**

At Section A – A :-

$$\text{Direct stress, } \sigma_o = \frac{W}{A} = \frac{320}{b \times t} = \frac{320}{2.5t \times t} = \frac{128}{t^2}$$ **1**

$$\text{Bending stress, } \sigma_b = \frac{M}{Z} = \frac{W \times e}{(1/6)tb^2} = \frac{320 \times 150 \times 6}{6.25t^3} = \frac{46080}{t^3} \quad \ldots [\because b = 2.5t]$$ **1**

Resultant stress,

$$\sigma_{tR} = \sigma_b + \sigma_o \geq 90$$

$$\therefore \frac{46080}{t^3} + \frac{128}{t^2} = 90$$

$$\therefore 90t^3 - 128t - 46080 = 0$$

$$\therefore t = 8.0592\ mm \approx 9\ mm\ (say)$$

$$\therefore b = 2.5t = 2.5(9) = 22.5\ mm$$ **1**

At section B – B :-

Frame is uniform in section throughout.
∴ Using same section at B – B, we should find the stresses induced at section B – B. These induced stresses should be within the permissible stress limit.

Use $b = 22.5\ mm$ & $t = 9\ mm$ **1**

Bending stress induced at section B – B.

$$\sigma_{b\ ind} = \frac{M}{Z} = \frac{W \times e}{\frac{tb^2}{6}}$$

$$\therefore \sigma_{b\ ind} = \frac{320 \times 80 \times 6}{9 \times (22.5)^2} = 33.71\ N/mm^2 < 90\ N/mm^2$$ **1**

		It is less than permissible stress. In addition to bending stress, there is transverse shear stress. We know, $\tau = 1.5\sigma_0$ $\therefore \tau = 1.5 \times \frac{W}{A} = 1.5 \times \frac{W}{(b \times t)} = 1.5 \times \frac{320}{(22.5 \times 9)} = 2.37\ N/mm^2$ which is less than 90 N/mm^2 Hence, design is safe $\therefore$ At section B - B :- Use, $b = 22.5\ mm$, $t = 9\ mm$	1
	b	Solution :- Given data $\Rightarrow$ $P = 20\ kW = 20 \times 10^3\ W$ $N = 700\ r.p.m$ $\tau_s = 40\ MPa = 40\ N/mm^2 = \tau_k = \tau_b$ $\sigma_{ckt} = 110\ N/mm^2$ $\tau = 10\ N/mm^2$, $\sigma_t = \sigma_{ck} = 100\ N/mm^2$, $n = 6$ The power transmitted by shaft, $P = 2\pi NT/60$ $\therefore$ Torque, $T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2\pi \times 700} = 272.84\ N\text{-}m$ $\therefore T = 272.84 \times 10^3\ N\text{-}mm$	1
		We know that, torque transmitted by shaft is given by, $T = \frac{\pi}{16} \times \tau_s \times d^3$ $\therefore 272.84 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$ $\therefore d = 32.6290\ mm \approx 33\ mm$ (say) $\therefore$ Diameter of shaft, $d = 33\ mm$	1
		Step I :- Design of Hub Usual proportions are, D = Outer diameter of hub $\therefore D = 2d = 2 \times 33 = 66\ mm$ L = Length of hub $= 1.5d = 1.5 \times 33 = 49.5\ mm$ $k = \frac{d}{D} = \frac{33}{66} = 0.5$	1
		Considering hub as a hollow shaft transmitting the same torque as that of the shaft, we have, $T = \frac{\pi}{16} \times \tau_{ci} \times D^3 (1 - k^4)$ $\therefore 272.84 \times 10^3 = \frac{\pi}{16} \times \tau_{ci} \times (66)^3 \times [1 - (0.5)^4]$ $\therefore \tau_{ci} = 5.15\ N/mm^2 < 10\ N/mm^2$ Thus, the induced shear stress in the cast iron hub is less than the given permissible shear stress. Hence, the design is safe.	1

Step - II :- Design of flange

Take, $t_f = \frac{d}{2} = \frac{33}{2} = 16.5\ mm$

While transmitting the torque, the flange is under shear.

$\therefore$ T = Circumference of hub × Thickness of flange × Shear stress × Radius of hub

$$\therefore T = (\pi \times D) \times t_f \times \tau_f \times \frac{D}{2}$$

$$\therefore 272.84 \times 10^3 = \pi \times 66 \times 16.5 \times \tau_f \times \frac{66}{2}$$

$$\therefore \tau_f = 2.42\ N/mm^2 < 10\ N/mm^2$$ 1

Thus, induced shear stress is less than given permissible shear stress for flange material. Hence, design is safe.

The other proportions are, $D_2 = 4d = 4 \times 33 = 132\ mm$

& thickness of protective circumferential flange, 1

$t_p = 0.25\,d = 0.25 \times 33 = 8.25\ mm$

Step - III :- Design of bolts

The bolts are subjected to shear stress due to torque transmitted.

$\therefore$ Load on each bolt $= \frac{\pi}{4} \times (d_c)^2 \times \tau_b$

$\therefore$ Total load on all bolts $= n \times \frac{\pi}{4} \times (d_c)^2 \times \tau_b$

$\therefore$ Torque transmitted = Load × radius $= n \times \frac{\pi}{4} \times (d_c)^2 \times \tau_b \times \frac{D_1}{2}$

Taking, $D_1 = 3d = 3(33) = 99\ mm$ 1

& $n = 6$ (given), the above equation becomes,

$$\therefore 272.84 \times 10^3 = 6 \times \frac{\pi}{4} \times (d_c)^2 \times 40 \times \frac{99}{2}$$

$$\therefore d_c = 5.407\ mm$$

We have, $d_c = 0.84 \times d_o$

$$\therefore d_o = \frac{d_c}{0.84} = \frac{5.407}{0.84} = 6.428\ mm \simeq 8\ mm\ (\text{say})$$

$\therefore$ We can use M8 bolts. For safety reasons, we can increase the size of bolts upto M16. 1

c. **Define stress concentration. What are the causes of stress concentration? State any four methods of reducing stress concentration with neat sketches.**

Stress concentration: Whenever a machine component changes the shape of its cross section, the simple stress distribution no longer holds good and the neighbourhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called 'stress concentration'.

1

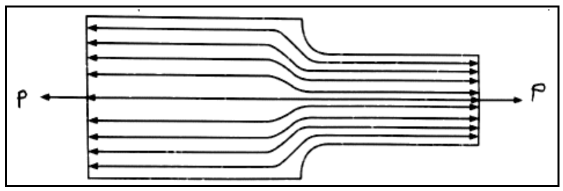


Fig. Stress concentration

Causes of stress concentration

The various causes of stress concentration are as follows:

(i) Abrupt change of cross section 1

(ii) Poor surface finish 1

(iii) Localized loading 1

(iv) Variation in the material properties

Methods of reducing 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.
   a) Use of multiple notches.
   b) Drilling additional holes. 2
2. Fillet radius, undercutting and notch for member in bending.
3. Reduction of stress concentration in threaded member.
4. Provide taper cross-section to the sharp corner of member.

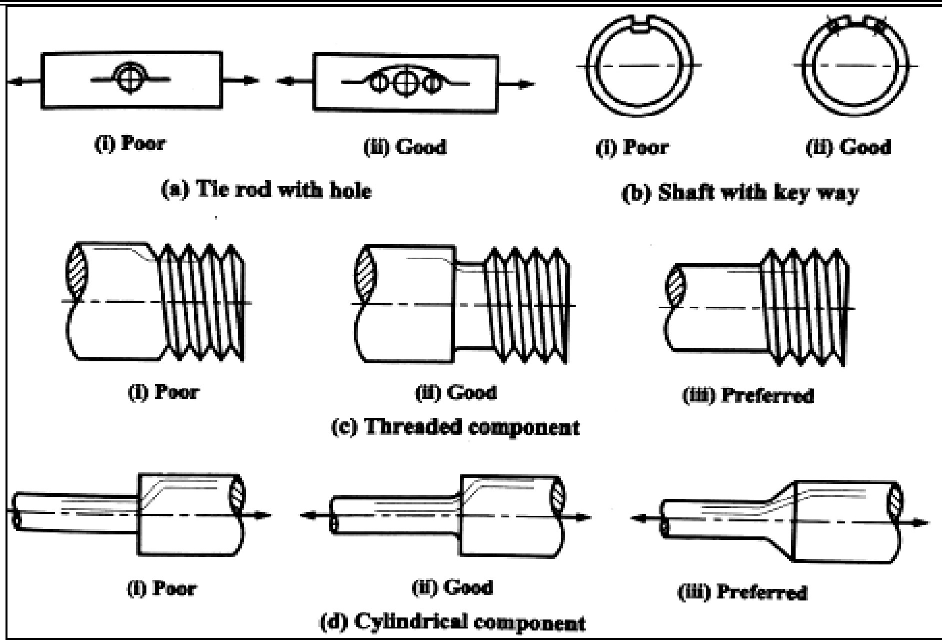


Fig. Methods of reducing stress concentration

2 for sketch

4

Solve any Two of the following

a) **Given Data:** Tension In slack side = T_2 = 5000 N , Tension In Tight side = T_1 = 10000 N

Safe stress σ_t =60 N/mm^2 ,L1= 60 mm , L2= 120 mm , e = 250 mm

Total Tension in pulley is acting downward direction = T= $T_1 + T_2$

=10000+5000=**15000 N…………………** 1M

Bracket will try to tilt about edge due to tension in belt

Tilting moment = M = T X e =15000 x 250 =**365 x 10^4 N-mm…………..**Eq(1) 1M

Let w be the load in each bolt per unit distance from tilting edge.

Assume bracket with4 Bolts

Total resisting moment = 2 w [L1^2 + L2^2] = 2 X W X [60^2 + 120^2] = 36000 w …………Eq(2) 1M

Equating eq (1) & (2)

365 x 10^4 N-mm =36000 w Therefor W= 101.39 N

The maximum tensile load will be in the bolt at a distance L2

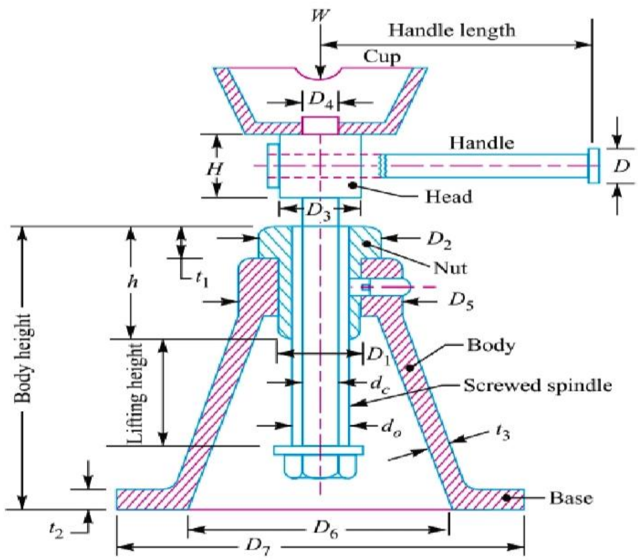
Wt $_{max}$ =w L2 = 101.39 N X 120 = 12.166 KN 2M

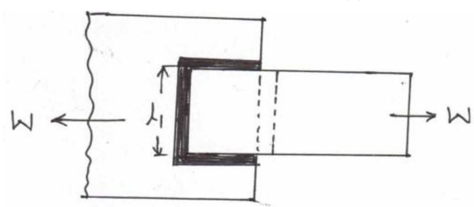
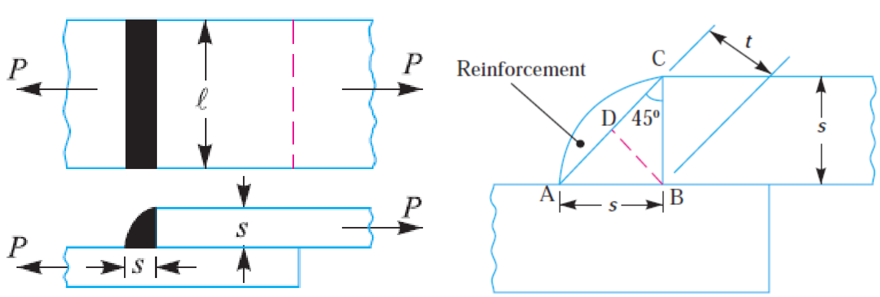
SIZE OF BOLT :

$6t = \frac{LOAD}{C/SAREA}$, $60\ N/mm2 = \frac{12.166 \times 10^3}{\frac{\pi}{4} dc^2}$ dc= 16.067 mm 2M

Do= 16.07/0.84 = 19.13 say 20 mm 1M

Size of bolt = M20

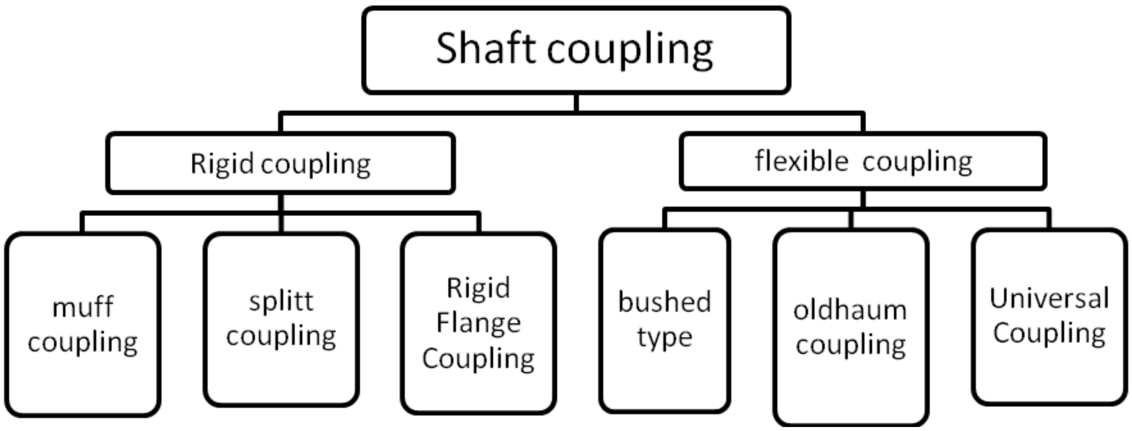
	b	**Design of spring** Given Data:LOAD W= 500N , $\delta = 25\ mm$ C= 8 τ =350 MPa = 350 N/m G= 85 x 10 ^{3} N/mm^{2} $Kw = \frac{4C-1}{4C-4} + \frac{0.615}{C} Kw = \frac{4x8-1}{4x\,8-4} + \frac{0.615}{8}$ = 1.184 $\tau = Kw\ \frac{8\ W\ C}{\pi\ d^2}$ $350 = 1.184\ \frac{8\ X500X8}{\pi\ d^2}$ $d = 5.87\ mm$ say 6 mm $\delta = \frac{8\ W\ C^3\ n}{G\ d}$, $25 = \frac{8\ x\ 500\ 8^3\ n}{85\ X\ 10^3\ X6}$, n =6.15 say 7 $Number\ of\ active\ turns\ of\ spring = 7$	2 M 2M 1 M 2M 1M
	c	**Design of screw jack**  **Design of Screw:** 1)Consider the screw under pure compression to find diameter of screw $\sigma c = \frac{W}{\frac{\pi}{4} X (dc)^2}$ As screw is subjected to twisting moment, higher value of screw is selected . Select The dimension of d_c w.r.t pitch Mean diameter d= do- p/2 2) Torque required to overcome the friction (T_1) Helix angle $\alpha = tan^{-1} \frac{p}{\pi x d}$ $\emptyset = tan^{-1}\mu$ Torque required lifting the load	Diagram 2 M 1M 1M

		T1= W. tan ($\alpha + \emptyset$) $\frac{d}{2}$ As collar friction is Neglecting, T_2=0 Total Torque required to lift the load = T_1 For Checking: Direct compressive stress in screw: $\sigma c = \frac{W}{\frac{\pi}{4} X (dc)^2}$, Torsional shear stress τ , $\tau = \frac{16\,T1}{\pi\, X\, (dc)^3}$ According to Maximum shear stress theory, the maximum shear stress in the screw $\tau_{max} = 1/2\sqrt{\sigma c^2 + 4\,\tau^2}$ Permissible shear stress for a screw τ = $\sigma c/2$ $\tau_{max} < a_{llowable}$, So screw is safe **Design of Nut:** The bearing pressure between the thread $Pb = \frac{W}{\frac{\pi}{4} X (do^2 - dc^2)\, n}$, Height of Nut: H= n X P Check: Shear stress induced in the screw thread $\tau = \frac{W}{\pi\, X\, (dc) X\, t\;\; n}$ as t= p/2 $\tau_{calculated} < \tau_{allowable}$, So screw is safe .	1 M 1M 1M 1M
5	A	**Equation: Parallel and transverse Weld**   Let *t* = Throat thickness (*BD*),*s* = Leg or size of weld, t = Thickness of plate, and *l* = Length of weld, , we find that the throat thickness, $t = s \times \sin 45° = 0.707\, s$	SKETCH 2 M 1M

		∴*Minimum area of the weld or throat area, A = Throat thickness ×Length of weld = $t \times l = 0.707\, s \times l$ … If σt is the allowable tensile stress for the weld metal, then the tensile strength of the joint for single fillet weld, P = Throat area × Allowable tensile stress = $0.707\, s \times l \times \sigma t$ …….. and tensile strength of the joint for double fillet weld, $P = 2 \times 0.707\, s \times l \times \sigma t = 1.414\, s \times l \times \sigma t$ …….. If τ is the allowable shear stress for the weld metal, then the shear strength of the joint for single parallel fillet weld, P = Throat area × Allowable shear stress = $0.707\, s \times l \times \tau$ and shear strength of the joint for **double parallel fillet weld**, $P = 2 \times 0.707 \times s \times l \times \tau = 1.414\, s \times l \times \tau$ …… The strength of the joint is given by the sum of strengths of **single transverse and double parallel fillet welds.** Mathematically, $P = 0.707 s \times l1 \times \sigma t + 1.414\, s \times l2 \times \tau$ ……	1M 1M 1M 1M 1M
	b)	**Power Screw: Given Data** Do= 100 mm , W =300 KN = 300 X 10^3 N, P=12 mm , μ= μ1= 0.15 Since,Screw is double start, Lead of screw = 2 p =2 x12 =24 mm dc= do-P =100-12 =88 Mean diameter d =(do+dc)/2 =(100+88)/2 =94 mm Tan $\alpha = \frac{Lead}{\pi d} = \frac{2p}{\pi d}$, $\alpha = tan^{-1}(\frac{2p}{\pi d})$ $\alpha = tan^{-1}\frac{24}{\pi x 94}$ = 4.64° $\emptyset = tan^{-1}\mu$ $=tan^{-1}x\ 0.15$ = 8.53° Torque Required to lift the load , T1= W.tan ($\alpha + \emptyset$) $\frac{d}{2}$ T1= 300 x 10^3x tan ($4.64° + 8.53°)\frac{94}{2}$ = 3301.15 x 10^3 N.mm Total Torque =Tt=T1+T2 =3301.15 x 10^3 + 0 = 3301.15 x 10^3 N.mm …………. **Efficiency of screw:** n = $\frac{tan \propto}{tan(\propto+\emptyset)}$ = $\frac{tan 4.64}{tan(4.64+8.53)}$ **= 0.347 i.e 34.71 %**	1M 1M 2M 1 M 1M 2M
	c	**Hollow shaft:** **Given data:** P= 20 kw = 20 x 10^3 W, N=200 rpm, σut = 360 Mpa, F.O.S = 8 , k=0.5 Shear stress σ = $\frac{\sigma ut}{fos}$ = $\frac{360}{8}$ =45 MPa =45 N/mm^2	2M

		Power transmitted = P= $\frac{2\pi N T}{60}$, 20 x 10^3 =$\frac{2\pi X 200 X T}{60}$ T=954.929 N.m =954.929 x 10^3 N.mm $Do^3 = \frac{\pi}{16}$ τdo^3 (1-0.5^4) do =48.66 mm**Outer Dia of Hollow shaft do =48.66 mm** $\frac{di}{do}$ = 0.5 **Inner Dia of Hollow shaft di= 0.5 x 48.66 = 24.33 mm**	2 M 2M 2M
6	a.i)	**Attempt any Two of the following** **<u>Effect of Keyway on strength of shaft</u>:** The keyway is a slot machined either on the shaft or in hub to accommodate the key.It is cut by vertical or horizontal milling cutter. A little consideration will show that the keyway cut into the **<u>shaft reduces the load carrying capacity of the shaft.</u>** This is due to the stress concentration near the corners of the keyway and **<u>reduction in the cross-sectional area of the shaft</u>**. It other words, the **<u>torsional strength of the shaft is reduced.</u>** The following relation for the weakening effect of the keyway is based on the experimental results by H.F. Moore.$e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$ where *e* = Shaft strength factor. *w* = Width of keyway, *d* = Diameter of shaft, and *h* = Depth of keyway =Thickness of key (t)/2 It is usually assumed that the **<u>strength of the keyed shaft is 75% of the solid shaft</u>**, which is somewhat higher than the value obtained by the above relation. In case the keyway is too long and the key is of sliding type, then the **<u>angle of twist is increasedin the ratio $K\theta$</u>**as given by the following relation $K\theta = 1 + 0.4\left(\frac{w}{d}\right) - 0.7\left(\frac{h}{d}\right)$ where *k*θ= Reduction factor for angular twist.	Correct Explanation 4 M
	ii)	**The different CAUSES of gear teeth failure:** 1. *Bending failure*. 2. *Pitting*. 3. *Scoring*. 4. *Abrasive wear*. 5. *Corrosive wear* **1. *Bending failure*.** Gear tooth behave like a cantilever beam subjected to repetitive bending stress. The tooth may crack due to repetitive bending stress In order to avoid such failure, the module and face width of the gear is adjusted so that the beam strength is greater than the dynamic load. **2. *Pitting*.**	ANY 4 CAUSES 2 Marks & Its Explanation 2M

		It is a surface fatigue failure due to repetitive contact stresses. Pitting starts when total load acting on the gear tooth exceeds the wear strength of the gear. In order to avoid the pitting, the dynamic load between the gear tooth should be less than the wear strength of the gear tooth. **3. *Scoring*.** It is lubrication failure. Inadequate lubrication along with high tooth load & poor surface finish results in breakdown of oil film and causes metal to metal contact. This type of failure can be avoided by properly designing the parameters such as speed, pressure and proper flow of the lubricant, so that the temperature at the rubbing faces is within the permissible limits. **4. *Abrasive wear*.** It is a surface damage caused by particles trapped in between the matting teeth surfaces. This type of failure can be avoided by providing filters for the lubricating oil or by using high viscosity	
		lubricant oil which enables the formation of thicker oil film and hence permits easy passage of such particles without damaging the gear surface. **5. *Corrosive wear*..** It is due to chemical action by the improper lubricant or sometimes it may be due to surrounding atmosphere which may be corrosive nature .In order to avoid this type of wear, proper anti-corrosive additives should be used.	
	b.i)	**Material & composition:**	
		A) X10C$_r$ 18 Ni9 Mo 4 Si 2 : High Alloy steel	1M
		having carbon 0.10% , chromium 18%, nickel 9 % ,Molybdenum 4% & silicon 2%	1M
		B) XT72W18Cr4V1: high speed tool steel	1M
		having carbon 0.72% ,chromium 4% , tungsten 18% , vanadium 1%	1M
	ii)	**Design consideration while designing the spur Gear** 1) The power to be transmitted 2) The velocity ration or speed of gear drive. 3) The central distance between the two shafts 4) Input speed of the driving gear. 5) Wear characteristics of the gear tooth for a long satisfactory life. 6) The use of space & material should be economical. 7) Efficiency & speed ratio 8) Cost	**:(Any FOUR)** **1 M EACH**
	c) i)	**Application of spring:** 1) To cushion, absorb or control energy to external load : Car springs, Railway buffers 2) To store Energy : Watches Toys 3) To Measure forces : Spring Balances, Gauges ,Engines 4) To provide clamping force in Jigs & fixtures. 5) To apply forces as in brakes, clutches & spring loaded valve.	**Any four 1M each**
	ii)	**Classification of shaft coupling :**	

		Shaft coupling Rigid coupling flexible coupling muff coupling splitt coupling Rigid Flange Coupling bushed type oldhaum coupling Universal Coupling	**4 marks**