
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.

(note : check once again marking scheme for all questions.Insert page no. to all pages)

Q.1- a) Attempt any THREE (3 x 4) **(12)**

i) List the steps involved in general design procedure. **(4marks)**

Ans. Definition of problem:- Define the problem giving all input parameters, output parameters & constraints. Synthesis:- It is the process of selecting or crating the mechanism for the machine and the shapes of the mechanical elements so as to get the desired output with given input.

Analysis of forces:- Find out the forces acting on each element by force analysis.

Selection of Material:- Select the suitable material for each element.

Determination of mode of failure:- Before finding out the dimensions of the element, it is necessary to know the type of failure by which the element will fail when put into the use.

Selection of factor of safety:- Based on the application, select the factor of safety, knowing factor of safety and material, determine the permissible stresses.

Determination of Dimensions:- Find the dimensions of each element of the machine by considering the forces acting on the element and the permissible stresses.

Modification of Dimensions:- Modify the dimensions of the element on the higher side if required based on past experience and standards.

Preparation of Drawings:- Prepare working drawing of each element or components with minimum into views showing details. Prepare assembly drawing giving part numbers, overall dimensions and part list. The component drawing is supplied to the shop flow for manufacturing purpose, while assembly drawing is supplied to the assembly shop.

Preparation of Design report:- Prepare design report containing details about step 1 to 8

ii) Name the different theories of elastic failure and explain any one.

Ans. 1) Maximum principal stress theory (Rankine's theory)

2) Maximum shear stress theory (Tresca & Guest theory)

3) Maximum strain energy theory (Haigh's theory)

4) Distortion energy theory (Von Mises & Hency theory)

5) Maximum principal strain theory (Saint venant's theory)

Explain of any one - 2 Marks

Name - 2 Marks

Maximum principal stress theory. States that the failure of the mechanical component subjected to biaxial or triaxial stresses occurs when the maximum principal stress reaches the ultimate or yield strength of the material.

According to this theory.

$6t_1 = \frac{6yt}{FS}$, for ductile materials

$= 6u/FS$, for brittle materials.

Maximum shear stress theory (Guest's theory):- According to this theory, the failure or yielding occurs at a point in a member when the maximum shear stress in a bi-axial stress system reaches a value equal to the shear stress at yield point in a simple tension test.

$$\tau \max = \frac{\tau yt}{FS}$$

Maximum principal strain theory

According to this theory, the failure or yielding occurs at a point in a member when the maximum principal strain in a bi-axial stress system reaches the limiting value of strain as determined from a simple tension test.

$$E \max = \frac{6t1}{E} - \frac{6t2}{mt}$$

$\therefore$ According to the above theory

$$E \max = \frac{6t1}{E} - \frac{6t2}{ME} = E = \frac{6yt}{EXF.S.}$$

$6t1 \,\&\, 6t2 =$ Maximum & minimum principal stresses in a bi-axial stress system E=Strain at yield point as determined from simple tension test.

1/m = Poisson's ratio
E = Young's modulus
FS = Factor of safety

Maximum strain energy theory:- According to this, the failure or yielding occurs at a point in a member when the strain energy per unit volume in a bi-axial stress system reaches the limiting strain energy per unit volume as determined from simple tension test. Strain energy per unit volume

$$U_1 = \frac{1}{2E}\left[(6t_1)^2 + (6t_2)^2 - \frac{26t_1 \times 6t_2}{m}\right]$$

limiting strain energy per unit volume for yielding

$$U_2 = \frac{1}{2E}\left(\frac{6yt}{FS}\right)^2$$

According to the above theory, $U_1 = U_2$

Used for ductile materials

Maximum Distortion Energy Theory:- Acc. To this theory, the failure or yielding occurs at a point in a member when the distortion strain energy per unit volume in a bi-axial stress system reaches the limiting distortion energy per unit volume as determined from a simple tension test.

$$(6t_1)^2 + (6t_2)^2 - 26t_1 \times 26t_2 = \left(\frac{6yt}{FS}\right)^2$$

Used for ductile materials

iii) What is the effect of keyway on the strength of shaft?

Ans. Keyway cut into the shaft reduces the load carrying capacity of the shaft. This is due to the stress concentration near the corner of the keyway and reduction in the cross sectional area of the shaft. Torsional strength of the shaft is reduced.

Weakening effect of the keyway is based on the experimental results by H.F. Moore

$$e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

e = shaft strength factor = it is the ratio of strength of shaft with keyway to the strength of the same shaft without keyway

w = width of keyway, d = dia of shaft

h = depth of keyway = $\frac{Thickness\ of\ key}{2}$

strength of keyed shaft is 75% of the solid shaft.

(4 marks)

iv) Explain the meaning of self locking and overhauling of screw.

Ans. Self locking of screw

Torque required to lower the load

$$T = P \times \frac{d}{2} = w\tan(\phi - \alpha)\frac{d}{2} \qquad (1)$$

In above expression, if $\phi > \alpha$, the toque required to lower the load will be positive indicating that an effort is applied to lower the load. For self-locking screw, friction angle is greater than the helix angle or coefficient of friction is greater than tangent of helix angle is μ or tan ϕ>tan α Efficiency of self-locking screw is less than ½ or 50%. **(2 marks)**

Overhauling of screw:- In equation if $\phi > \alpha$, then torque required to lower the load will be negative. The load will short moving downward without the application of any torque-such conditions are known as overhauling of screws efficiency is more than 50%. **(2 marks)**

b) Any one of the following

i) Design a 'c' clamp frame for a total clamping force of 20 kN. The cross section of the frame is 2. The distance between the load line & neural axis of rectangular section is 12 mm and gap between two factors is 180 mm. The frame is made of cast steel. The permissible tensile stress for cast steel is 100 N/mm^2.

Given:- P = 20 x 10^{3}N, b = 2t, e = 12mm, 6t = 100N/mm^2

Cross sectional area, A = b x t = 2t^2, mm^2

Direct tensile stress $6_0 = \frac{P}{A} = \frac{20\times10^3}{2t^2}$ ---------– **1 mark**

Bending moment due to the load P

M = P x e = 20 x 10^3 x 12 = 24 x 10^{4}Nmm

Section modulus $z = \frac{tb^2}{6} = \frac{t(2t)^2}{6} = \frac{4t^3}{6}$

$\therefore$ Bending stress $6_b = \frac{M}{Z} = \frac{24\times10^4}{4t^3} = \frac{36\times10^4}{t^3 N/mm^2}$ **1 mark**

The resultant stress is maximum at the inner most fibre

$$6t = \frac{P}{A} + \frac{M}{Z}$$

$$100 = \frac{20\times10^3}{2t^2} + \frac{36\times10^4}{t^3}$$ **2 mark**

$$100 = \frac{10\times10^3}{t^2} + \frac{36\times10^4}{t^3}$$

$$100t^3 - 10\times10^3 t + 36\times10^4$$

$$t^3 - 100t = 3600$$

Solved by trial and error

$$t = 17.5mm \text{ say } 18mm$$
$$b = 2t = 2 \times 18 = 36mm$$ **2 mark**

ii) A bracket carrying a vertical load of 25 kN as shown in figure 1. The load in taken up by 4 bolts for fixing the bracket. Determine the size of bolt for permissible tensile stress of 80 N/mm^2.

Ans. Direct tensile load carried by each bolt $wt_1 = \frac{W}{n} = \frac{25}{4} = 6.25kN$ and ----**1 mark**

load in a bolt per unit distance $w = \frac{wL}{2\left[(L_1)^2 + (L_2)^2\right]} = \frac{25 \times 200}{2\left[(40)^2 + (160)^2\right]}$ ------------**2 mark**

$$w = 0.092kN/mm$$

Heavily loaded bolt is at a distance of L_2 from the tilting edge, therefore load on the heavily loaded bolt $wt_2 = wL_2 = 0.092 \times 160 = 14.72kN$ -----**1 mark**

$\therefore$ maximum tensile load on the heavily loaded bolt $wt = wt_1 + wt_2 = 6.25 + 14.72 = 20.97kN$ -------

Maximum tensile load on the bolt

$$wt = \frac{\Pi}{4}(dc)^2 \times 6t$$
$$20.97 \times 10^3 = 0.7854(dc)^2 \times 80$$ ---**2 mark**
$$dc = 18.26mm$$

from table coarse series, the standard core diameter of the bolt is 18.933 mm and the corresponding size of the bolt is M22.

Q.2 Any Two (2 x8) (16)

a) Explain the design procedure of hand lever with suitable sketch.

Ans. 1) The diameter of the shaft (d) is obtained by considering the shaft under pure torsion. Twisting moment on the shaft T = P x L and resisting torque $T = \frac{\Pi}{16} \times \tau \times d^3$

From this relation the dia. of shaft may be obtained.

2) The diameter of boss (d_2) is taken as 1.6d and thickness of the boss t_2 as 0.3d.

3) The length of the boss (l_2) may be taken from d to 1.25d. for trial thickness (t_2)

$$P \times L = L_2 t_2 6t\left(\frac{d+t_2}{2}\right)$$

4) The diameter of the shaft at the centre of the bearing (d_1) is obtained by considering the shaft in combined bending and twisting. Bending moment on the shaft M = P x l & twisting moment T = P x L

$\therefore$ Equivalent twisting moment

$$Te = \sqrt{M^2 + T^2}$$

$$Te = \sqrt{(P \times l)^2 + (P \times L)^2} = P\sqrt{l^2 + L^2}$$

Equivalent twisting moment $Te = \frac{\Pi}{16} \times \tau (d_1)^3 \ or \ P\sqrt{l^2 + L^2} = \frac{\Pi}{16} \times \tau (d_1)^3$

$$l = 2l2$$

From above the value of d_1 may be det.

5) The key for the shaft is designed as usual for transmitting a torque of P x L.

6) The cross section of the lever near the boss may be determined by considering the lever in bending.

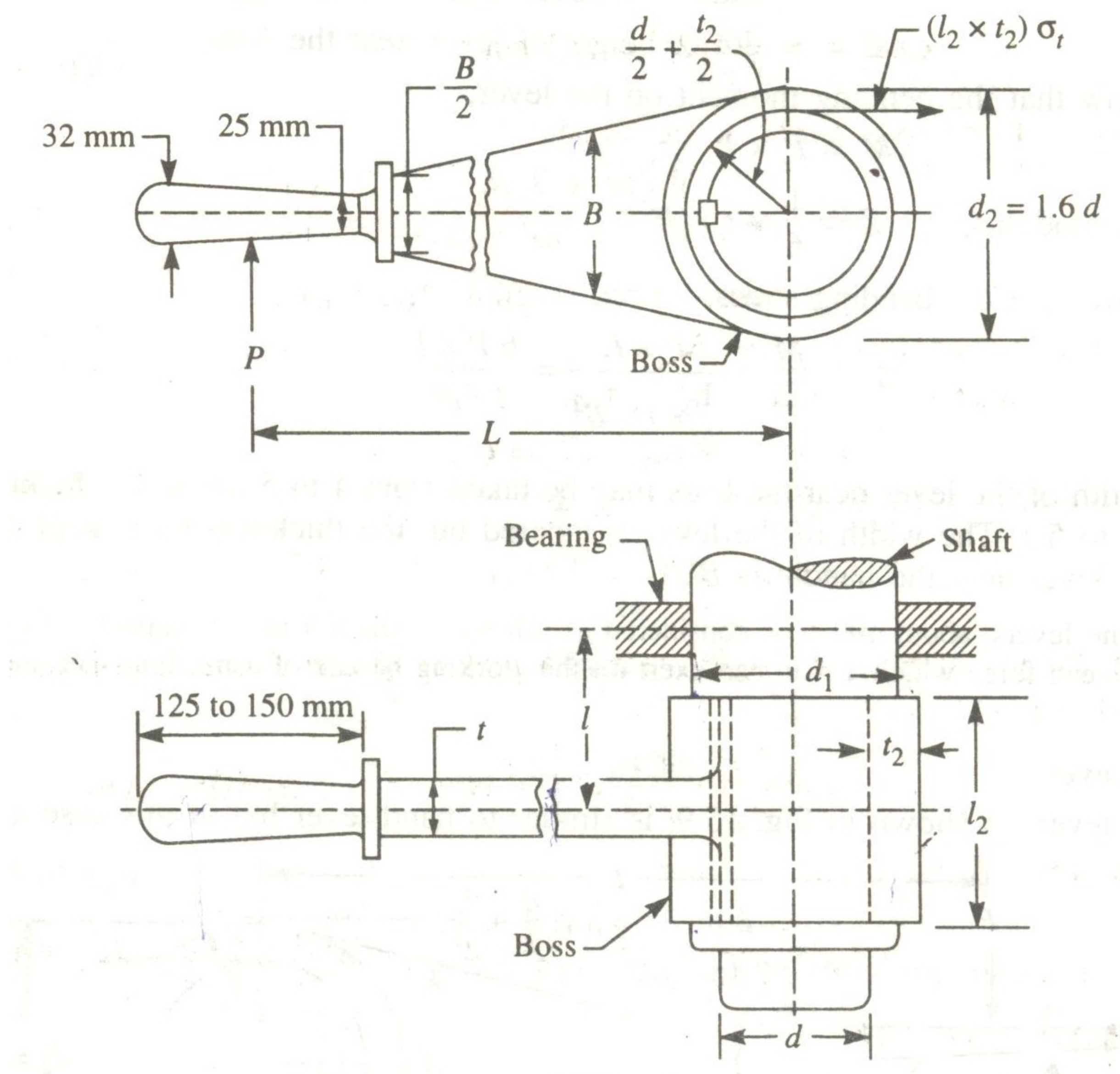
Bending moment on the lever M = P x L

Section modulus $z = \frac{1}{6} \times t \times B^2$

Bending stress, $6_b = \frac{M}{Z} = \frac{P \times L}{1/6tB^2} = \frac{6PL}{tB^2}$

B = 4 to 5 times the thickness of lower width of the lever near the handle is B/2.

sketch -2 marks,design procedure-6 marks **one mark per step**



Q.2 b)

Give d = 40 mm, t = 12 mm, 6t = 46 N/mm^2, τ = 35 N/mm^2, 6c = 70 N/mm^2

$$P = \frac{\Pi}{4} d^2 \times 6t = 0.7854 \times (40)^2 \times 46$$ **----2marks**

$$P = 57805.44N$$

1) Diameter of socket (d_1)

Failure of socket in tension across the slot

$$P = \left[\frac{\Pi}{4}\left\{(d_1)^2 - (d_2)^2\right\} - (d_1 - d_2)t\right]6t$$

Here $t = \frac{d2}{4}$ $or\ d_2 = 4 \times t = 4 \times 12 = 48mm$ ---------------1marks

and

$$\therefore 57805.44 = \left[\left\{0.7854(d_1)^2 - 0.7854 \times (48)^2 - (d_1 - 48) \times 12\right\}\right] \times 46$$

$$d_1 = 64.46mm\ say\ 65mm$$

----2marks

2) Diameter of socket collar.

Failure of the socket collar and cotter in cursing

$$P = (d4 - d2) \times t \times 6c$$
$$57805.44 = (d4 - 48) \times 12 \times 70$$
$$d4 = 116.82mm$$

----- 2marks

3) Thickness of socket collar, Failure of socket end in shearing since the socket end is in double shear

$$P = 2(d4 - d2) \times C \times \tau$$
$$57805.44 = 2(116.82 - 48) \times C \times 35$$
$$C = 12mm$$

----1 mark

Q.2 c)

P = 40 kW = 40 x 10^3w, N = 350 rpm, $\tau_s = \tau_k$ = 15Mpa, 6ck = 30Mpa

1) Design for shaft **(2 marks)**

Torque transmitted by the shaft, key and muff

$$T = \frac{P \times 60}{2\Pi N} = \frac{40 \times 10^3 \times 60}{2\Pi \times 350} = 1100N - M$$
$$= 1100 \times 10^3\ N - mm$$
$$T = \frac{\Pi}{16} \times \tau \times d^3 = \frac{\Pi}{16} \times 15 \times d^3$$
$$1100 \times 10^3 = \frac{\Pi}{16} \times 15 \times d^3$$
$$d = 72mm\ say\ 80mm$$

2) Design for sleeve **(2 marks)**

Outer diameter & the muff

$$D = 2d + 13mm = 2 \times 80 + 13 = 173mm$$

& length of the muff $L = 3.5d = 605.5mm$

check induced shear stress in the muff.

Muff is to be a hollow shaft

$$T = \frac{\Pi}{16} \times \tau_C \left(\frac{D^4 - d^4}{D} \right) = 1100 \times 10^3$$

$$1100 \times 10^3 = \frac{\Pi}{16} \tau_C \left(\frac{(173)^4 - (80)^4}{173} \right)$$

$$\tau_C = 1.13 N/mm^2$$

sinced τ_C induced < τ_C allowable $\therefore$ Design is safe

3) Design for key **(4 marks)**

consider shearing of the key

Torque transmitted (T)

$$T = l \times w \times \tau s \times \frac{d}{2}$$

$$1100 \times 10^3 = 1.75 \times d \times w \times 15 \times \frac{d}{2}$$

$$1100 \times 10^3 = 1.75 \times 80 \times w \times 15 \times \frac{80}{2}$$

$$w = 13.095$$

considering crushing of the key

Torque transmitted (T)

$$1100 \times 10^3 = l \times \frac{t}{2} \times 6cs \times \frac{d}{2}$$

$$= 1.75 \times d \times \frac{t}{2} \times 30 \times \frac{d}{2}$$

$$1100 \times 10^3 = 1.75 \times 80 \times \frac{t}{2} \times 30 \times \frac{80}{2}$$

$$t = 13.095 mm$$

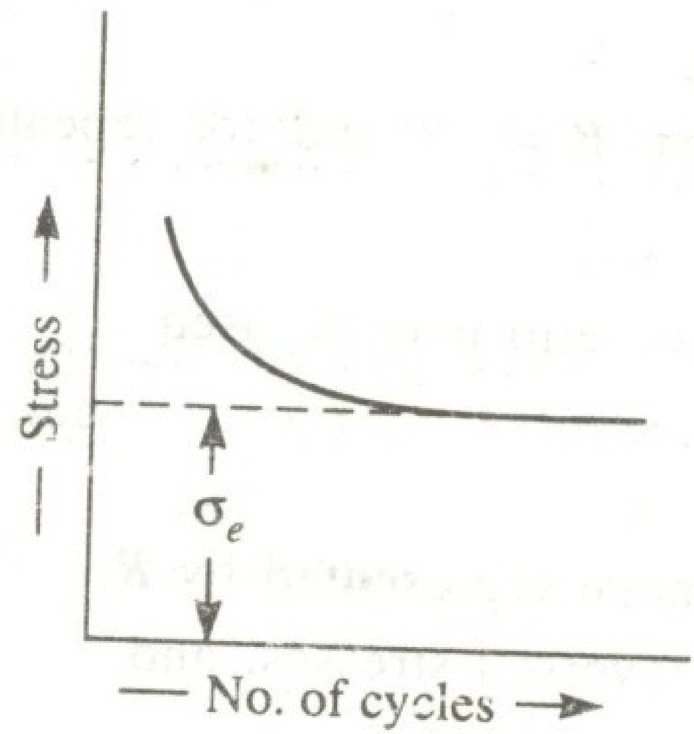
Q.3 **Any FOUR** **(4 x 4)** **(16)**

a) Explain the term endurance limit with 5-N curve.

Ans. If a component in subjected to repeated stresses below the yield point stresses such a type of failure of a material is known as fatigue failure. But if the stress in kept below the certain value as shown by dotted line in fig. the material will not fail whatever may be the no. of cycles. This stress as shown by dotted line is known as endurance limit or fatigue limit.

The term endurance limit is used for reversed bending only.

(Figure 1mark,explanation 3 marks)



b) Define stress concentration list any four methods to reduce it with neat sketches.

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

(1 mark)

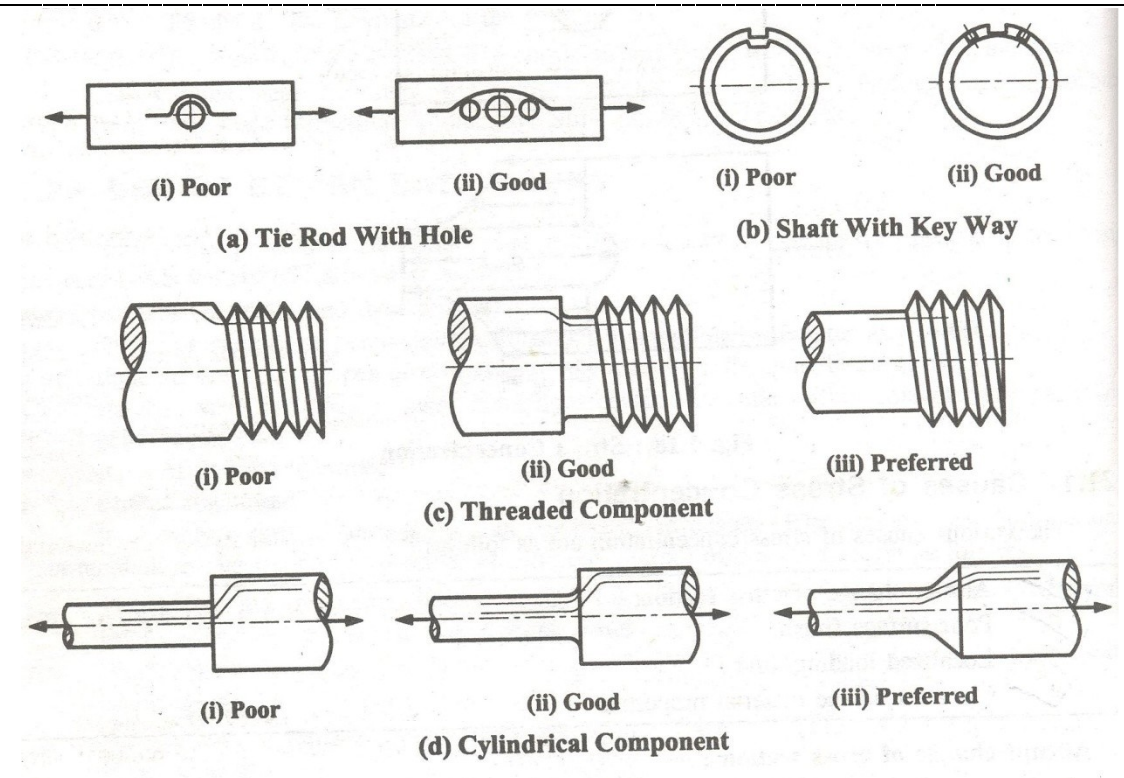
Methods of reducing stress concentration in **(1mark)**

i) Cylindrical members with holes.

ii) Cylindrical members with shoulders.

iii) Threaded component.

iv) Shaft with key way



Sketches **(2 marks)**

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

Ans. Lewis equation **(2 marks)**

$$WT = \sigma w.b.Pc.y = 6w.b.\pi m$$

Y is known as Lewis form factor or tooth form factor = $\frac{x^2}{6k}$

WT = Tangential load acting at the term

= Beam strength of the tooth

b = Width of the gear face

Pc = Circular pitch

m = Module

(Meaning of each term – 2 marks)

d) State the reasons for using hollow shaft rather than solid shaft for large power transmission.

Ans. The hollow shaft has higher torque transmitting capacity than the solid shaft of same weight or for required torque transmitting capacity hollow shaft is lighter in weight than the solid shaft.

The hollow shaft has higher torsional as well as lateral rigidity than the solid shaft of same weight or for the required rigidity hollow shaft is lighter in weight than the solid shaft.

Q.3 e) State the applications of the following bearings with suitable reasons.

i) Deep groove ball bearing –Automotive products, precision machines, electric tools. reason for light radial load with high rotational speed.

ii) Thrust roller bearing –pumps and compressors reason for high axial load carrying capacity and pure thrust loads.

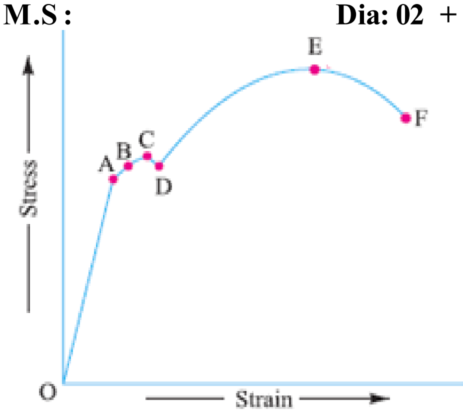
iii) Taper roller bearing –aerospace engg, automobile engg, high speed rolling machines reason for combined radial and axial load with medium speed. Can take thrust load in only one direction.

iv) Solid bush type journal bearing –turbomachinery, reason for shafts transmitting high power used in centrifugal pump & turbine shaft.

(applications –2 marks reasons –2 marks)

Qu.4 a) Attempt any three (3 x 4marks)

I) Stress –strain diagram for M.S : **Dia: 02 + lable/Points 02**



Point A: Proportional limit
Point B: Elastic limit
Point C: Upper yield point
Point D: Lower yield point
Point E: Ultimate tensile stress point
Point F: Breaking Stress point.

II) Material specification and practical application: (4 marks)

i) **FG 300:**

Specification: It is Grey cast iron having Min. U.T.S 300 N/mm^2 **01M**

Application: Gear, cylinder head, pump body, flywheel etc. **(any one) 01 M**

ii) X20Cr 18 Ni 2:

Specification: High alloy steel having Carbon 0.20% ,Chromium 18% and Nickel 2% **1M**

Application: used as sheet /strip for cold forming & press operation **01M**

iii. Distinguish between shaft and axle (4 marks) **1 mark each** point

Shaft	**Axle**
It is rotating element	It is stationary element
It transmit power from one place to another	It is used to support a rotating body
e.g Propeller shaft	e.g Front & rear axle
Subjected to torque, bending moment or axial force.	Subjected to bending moment only.

Iv) Wahl's Correction factor: (4 marks)

Wahl's correction factor or stress factor is modification of factor Ks in following equation

$$\tau = \frac{8\,W.D}{\pi\, d^3}\left(1+\frac{1}{2C}\right) = K_S \times \frac{8\,W.D}{\pi\, d^3}$$

Where, Ks= 1+1/2c

In the above equation, we have considered only torsion shear stress & direct shear stress due to direct load. **01 mark**

Here, We have neglected the stress due to curvature of wire.

When the wire bend, the stress concentration occurred at the inner fiber of coil and this curvature effect is considerably larger for high value of spring index. Therefore , Wahl's correction factor in introduced. **01 mark**

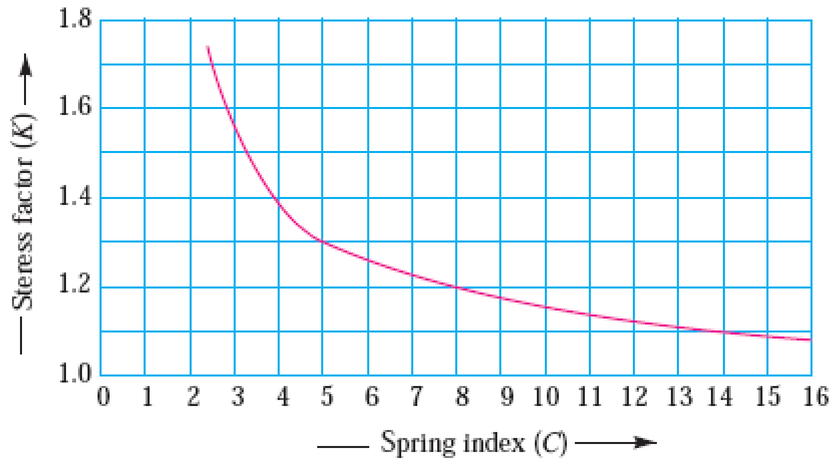
By considering, torsion shear, direct shear and curvature shear stress

∴ Maximum shear stress induced in the wire,

$$\tau = K \times \frac{8\,W.D}{\pi\, d^3} = K \times \frac{8\,W.C}{\pi\, d^2}$$

where $$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$ **01 mark**

Prof.A.M.Wahl's determined this factor hence it is known as wahl's factor.



01 mark

Wahl's stress factor increases very rapidly as the spring index decreases.

Qu.4 B) Attempt Any One (1 x 06marks)

Given Data:

n=2, do=100mm,
P=12mm, µ1 =0.15 ,
µ2 =0, W=300KN =300X10^3 N

To find: 1) Torque require to raise the load T
2) Efficiency of the screw

Core dia. dc=do- p =100 – 12 =88 mm
Mean dia of screw = d= (do + dc)/ 2 = (100+88)/2 =94 mm
Since the screw is a double start square threaded screw,
therefore lead of the screw, = 2 p = 2 × 12 = 24 mm **01 M**

$$\tan\alpha = \frac{\text{Lead}}{\pi d}$$

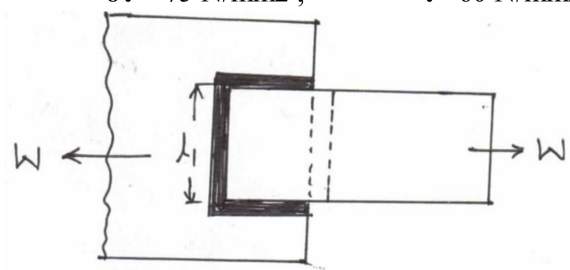
α =tan- (24/ Л x94) , **α = 4.64630**

tan ø= µ , **ø = tan- (0.15) =8.53070** **01 M**

The torque required to overcome the friction between nut & screw.
T1=P x d/2 = W (tan α + ø) x d/2
T1=300 x 10^3 x tan (4.6463 + 8.5307) x 94/2 = **3301.15 x 10^3 N.mm** **01 M**
As collar friction is neglected **T2=0**
Total torque T to raise the load = T= T1+T2
T=3301.15 X 10^3 + 0, **T= 3301.15 X 10^3 N.mm** **01 M**
2) Efficiency of screw
ή screw = tan α/ (tan α + ø)

η =tan (4.64463)/ tan (4.64463 + 8.5307)
Efficiency of screw = 0.3471 or **34.71 %** **02 M**

ii) Given data :
Width = 120 mm ; Thickness = 15 mm ;
στ = 75 N/mm2 ; τ =60 N/mm2.



W = Area × Stress = 120 × 15 × 75 = **135 × 10^3 N** **01M**

Assume, stress concentration factor for transverse weld is 1.5 and
For parallel fillet welds is 2.7.

Permissible tensile stress, $\sigma t = 75 / 1.5 = 50$ N/mm2

& permissible shear stress, $\tau = 60 / 2.7 = 22.22$ N/mm2 **01M**

The effective length of the weld run (l1) for a single transverse weld may be obtained by subtracting 12.5 mm from the width of the plate.

l_1= 120 – 12.5 = 107.5 mm

Let l_2 = Length of weld run for each parallel fillet, and
s = Size of weld = Thickness of plate = 15 mm

Load carried by single transverse weld,
W1 = 0.707 s × l_1 × σt = 0.707 × 15 × 107.5 × 50 = 57001.8 N
and load carried by double parallel fillet weld,
W2 = 1.414 s × l_2 × τ = 1.414 × 15 × l_2 × 22.22= 471.286 x l2 N **02 M**

Load carried by the joint (W), W=W1+W2

$$135 \times 10^3 = 57001.8 + 471.286\ l2$$

$$l_2 = \mathbf{165.50\ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

l_2 = 165.50 + 12.5 = **178 mm** **02 M**

Qu.5

a) Given data :-

$L = 1\,m = 1000\,mm$

$P = 15\,Kw = 15 \times 10^3\,W$

$N = 300\,rpm$

$\sigma_t = 70\,N/mm^2$

$\tau = 56\,N/mm^2$

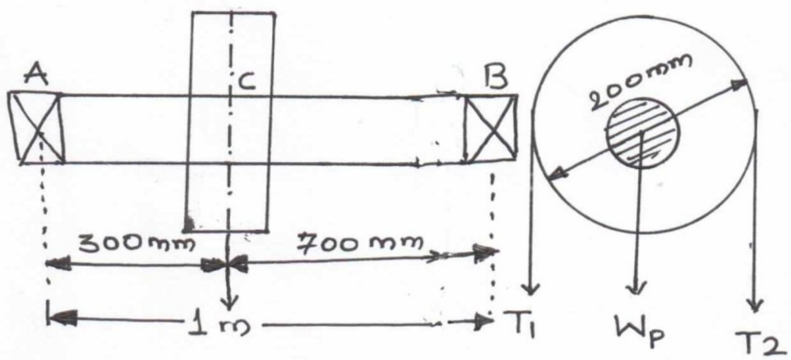
$D_{pulley} = 200\,mm$

$\therefore r_{pulley} = 100\,mm$

$\frac{T_1}{T_2} = \frac{2}{1} \quad \therefore T_1 = 2T_2$

$W_{pulley} = 500\,N$

Find : dia. of shaft $d = ?$



Power transmitted $P = \frac{2\pi N T}{60}$

$\therefore 15 \times 10^3 = \frac{2\pi \times 300 \times T}{60}$

$\therefore T = 477.46\,N\cdot m$

$\therefore T = 477.46 \times 10^3\,N\cdot mm$ —— 1M

Torque transmitted by pulley $T = (T_1 - T_2) \times R$

$\therefore 477.46 \times 10^3 = (2T_2 - T_2) \times 100$

$\therefore T_2 = 4774.6\,N.$

$T_1 = 2T_2 \quad \therefore T_1 = 2 \times 4774.6\,N.$

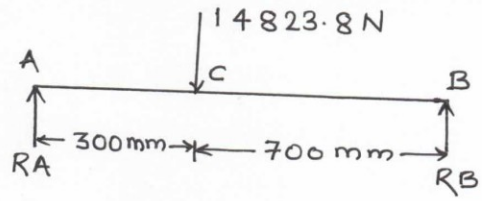
$\therefore T_1 = 9549.2\,N$

Total wt on pulley $W_p = W_P + T_1 + T_2$

$= 500 + 4774.6 + 9549.2$

$W = 14823.8\,N$ —— 1M

vertical load diagram :-



$\Sigma Fy = 0 \quad \therefore RA - 14823.8 + RB = 0$

$\therefore RA + RB = 14823.8 \text{ N}$

$\Sigma M_A = 0 \quad 14823.8 \times 300 - RB \times 1000 = 0$

$\therefore RB = 4447.140 \text{ N}$ —— $\frac{1}{2}$ M

$\therefore RA = 14823.8 - 4447.14$

$= 10376.66 \text{ N}$ —— $\frac{1}{2}$ M

Max. bending moment at Pulley (at c)

$\therefore M_{at\,C} \Rightarrow RA \times 300$ or $M_c = RB \times 700$

$= 10376.66 \times 300$ | $M_c = 4447.14 \times 700$

$= 3112998 \text{ N·mm}$ | $= 3112998 \text{ N·mm}$

—— 1 M

According to Max. shear stress Theory

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} d^3 \times \tau_{max}$$

$$\therefore T_e = \sqrt{(3112998)^2 + (477.46 \times 10^3)^2} = \frac{\pi}{16} d^3 \times 56$$

$$\therefore 3149400.67 = \frac{\pi}{16} d^3 \times 56$$

$$\therefore d = 65.91 \text{ mm}$$ —— $1\frac{1}{2}$ M

According to max. Principal Stress Theory

$$M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = \frac{\pi}{32} d^3 \times \sigma_b$$

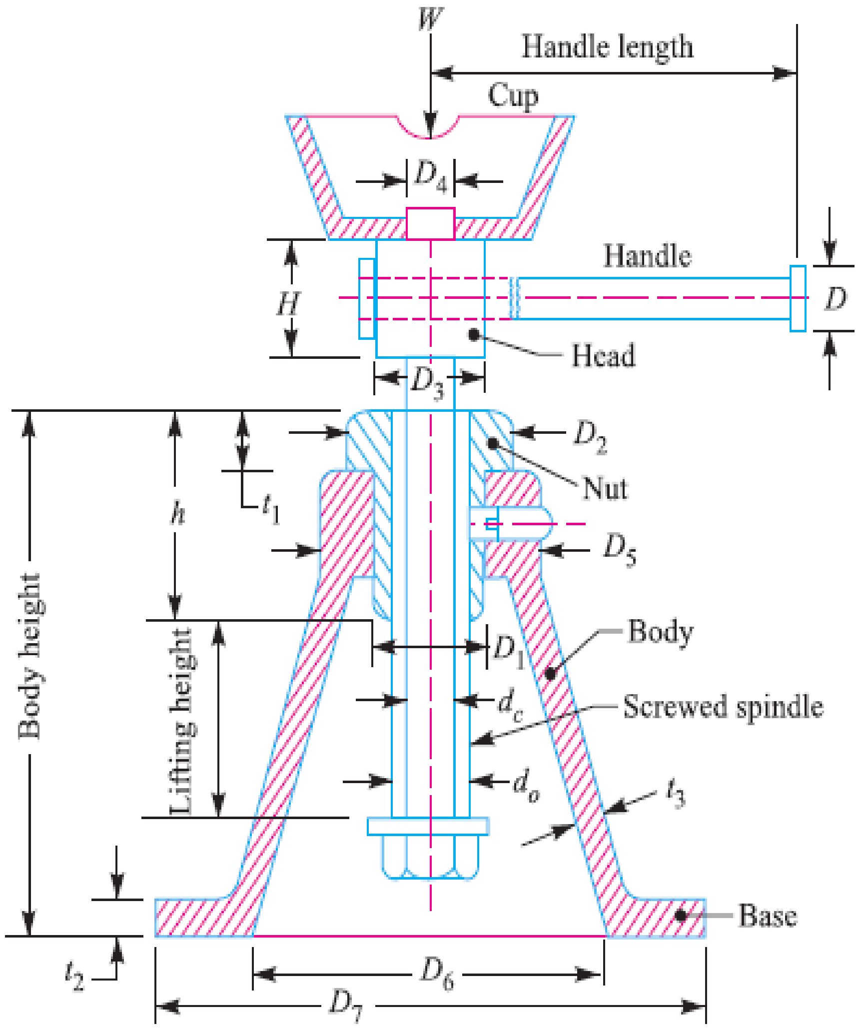
$$\Rightarrow \frac{1}{2}\left[3.113 \times 10^6 + 3149400.67\right] = \frac{\pi}{32} \times d^3 \times 70$$ —— $1\frac{1}{2}$ M

$$\therefore d = 76.95 \text{ mm}$$

Taking larger dia. of shaft $\therefore d = 76.95 \text{ mm}$

$d \simeq 77 \text{ mm}$ —— Selection 1 M

QU.5 b) Design procedure of screw & nut of a screw jack. **8 marks**



sketch: 02 marks

1) Find core dia. Dc : Consider screw is under pure compression

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

01 mark

2)Torque required to raise the load

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

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

d = Mean diameter of the screw.

01 mark

2. Shear stress due to torque T1

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

Find Direct compressive stress due to axial load

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

01 mark

3. Find the principal Stresses

Find the principal stresses as follows:

Maximum principal stress (tensile or compressive),

$$\sigma_{c(max)} = \frac{1}{2}\left[\sigma_c + \sqrt{(\sigma_c)^2 + 4\tau^2}\right]$$

and maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2}$$

These stresses should be less than the permissible stresses. **01 mark**

4.The height of nut can be calculated by considering bearing pressure

$$p_b = \frac{W}{\frac{\pi}{4}\left[(d_o)^2 - (d_c)^2\right]n}$$

where n = Number of threads in contact with screwed spindle.

$\therefore$ Height of nut, $h = n \times p$

where p = Pitch of threads. **01 mark**

5.Check the stresses in screw & nut:

Check the stressess in the screw and nut as follows :

$$\tau_{(screw)} = \frac{W}{\pi\, n.d_c.t}$$

$$\tau_{(nut)} = \frac{W}{\pi\, n.d_o.t}$$

where t = Thickness of screw $= p/2$

01 mark

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5 c)

Given data:-

$W = 800\ N$ $\quad C = 5$

$\delta = 25\ mm$ $\quad K_w = 1.3$

$\tau = 400\ MPa = 400\ N/mm^2$ $\quad G = 84\ KN/mm^2 = 84 \times 10^3\ N/mm^2$

Design of helical comp. spring

1) Mean dia. of spring coil :-

The max. shear stress induced

$$\tau = K_w \times \frac{8WD}{\pi d^3} \Rightarrow K_w \times \frac{8WC}{\pi d^2} \quad \text{—— } 0\tfrac{1}{2}\ M$$

$$\left(\because C = \frac{D}{d}\right)$$

$$\therefore d^2 = K_W \frac{8Wc}{\pi \times \tau} \Rightarrow 1.3 \times \frac{8 \times 800 \times 5}{\pi \times 400}$$

$$\therefore d = 5.75\ mm \quad \text{—— } 01\ M$$

& $D = C \times d = 5 \times 5.75$

$$D = 28.75\ mm \quad \text{—— } 01\ M$$

Dia. of spring wire $d = 5.75\ mm$

Mean dia. of spring coil $D = 28.75\ mm$

2) Number of turns :-

$$\delta = \frac{8WD^3 \times n}{G \times d^4} \quad \therefore n = \frac{\delta \times G \times d^4}{8WD^3} \quad \longrightarrow 0\tfrac{1}{2}\ M$$

$$\therefore n = \frac{25 \times 84 \times 10^3 \times (5.75)^4}{8 \times 800 \times (28.75)^3}$$

$$\therefore n = 15.09 \cong 16 \quad \longrightarrow 01\ M$$

Total no. of active turns $= n = 16$

$\therefore$ Total no. of turns $= n' = n + 2 = 18$

3) Solid length : $L_s = n' \times d = 18 \times 5.75 = 103.5\ mm \longrightarrow 1\ M$

4) Free length $L_f = n' \times d + \delta_{max} + 0.15\,\delta_{max}$

$= 18 \times 5.75 + 25 + 0.15 \times 25$

$= 132.25\ mm$ —— 01 M

5> Pitch of the coil $P = \frac{\text{Free length}}{n'-1}$

$P = \frac{132.25}{18-1}$

$= 7.78\ mm$ ———— 01 M

5> Outer dia. of coil $\Rightarrow D_o = D + d =$

$= 28.75 + 5.75 = 34.5\ mm$ — $\frac{1}{2}$ M

Inner dia. of coil $\Rightarrow D_i = D - d$

$= 28.75 - 5.75 = 23\ mm$ — $\frac{1}{2}$ M

(Note: Number of active turn may vary in the Design of spring and other dimensions accordingly.).

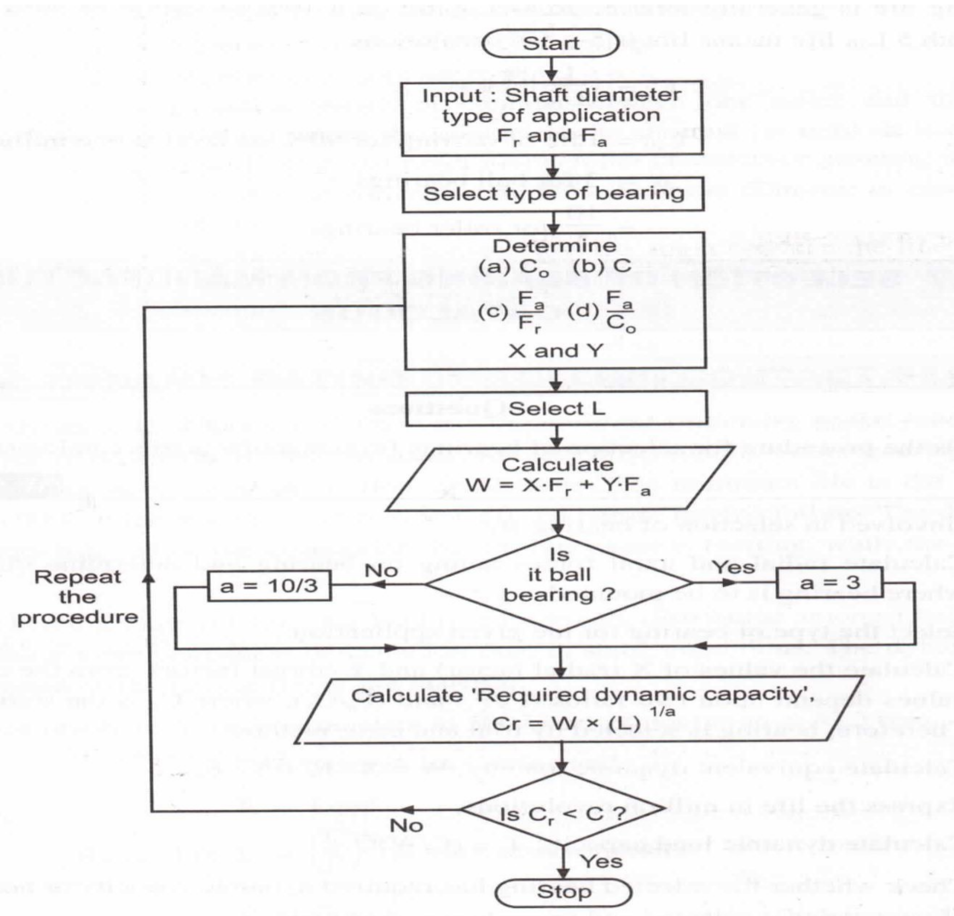
Qu.6 Attempt any four (4 x 4 M)

a) Procedure for selection of bearing from manufacturer's catalogue.

(Correct Procedure OR Flow chart - 4 Marks)

1) Calculate radial and axial forces and determine dia. of shaft.
2) Select proper type of bearing.
3) Start with extra light series for given diagram go by trial of error method.
4) Find value of basic static capacity (co) of selected bearing from catalogue.
5) Calculate ratios Fa/VFr and Fa/Co.
6) Calculate values of radial and thrust factors.(X & Y) from catalogue.
7) For given application find value of load factor Ka from catalogue.
8) Calculate equivalent dynamic load using relation. Pe = (XVFr + YFA) Ka.
9) Decide expected life of bearing considering application. Express life in million revolutions L10.
10) Calculate required basic dynamic capacity for bearing by relation.
11) Check whether selected bearing has req. dynamic capacity, IF it not select the bearing of next series and repeat procedure from step-4.

OR



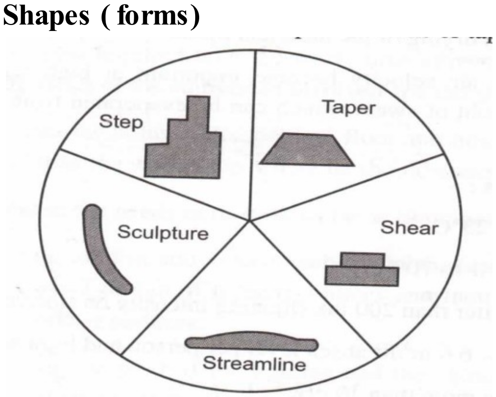
b) Importance of shape and size in Aesthetic design

The growing realization of the need of aesthetic consideration & tough competition, it is very essential for designer to create new forms & shapes which are aesthetically pleasant.

There are five basic forms of the products: 1) step 2) taper 3) streamline,4) shear 5) sculpture.

The external shape of any product is based on the combination of these basic shapes.

Shapes (forms)



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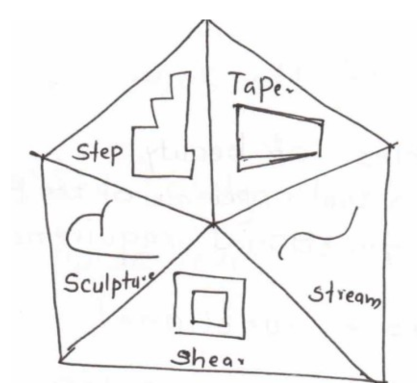


1)Stepped structure having vertical accent.e.g multistory building

2)Taper form: consist of tapered cylinder /blacks
3) Shear form: has square outlook. Suitable for free standing engg. product. stationery product
4) Streamline form: having smooth flow .for automobile and aero plane structure.
5) Sculpture form: consist of ellipsoids, paraboloids and hyperboloids. Mobile products.

In many cases, functional requirements results in shapes which are aesthetically pleasing.

The evolution of the streamlined shape of the Boeing is the result of studies in aerodynamics for effortless speed **03 Marks**

Sizes: **01 mark**

Due to advancement in electronics fields, designers can use previously unaccepted housing for integrated items, so freeing them from many of design constraints.

Now, design of telephone is an example of integrating the entire telephone circuitry in a single component providing good balance, proportional and ergonomic styling.

c) Advantages & disadvantages of V thread over square thread

Advantages: **Any 2 pt (1 Mark each)**

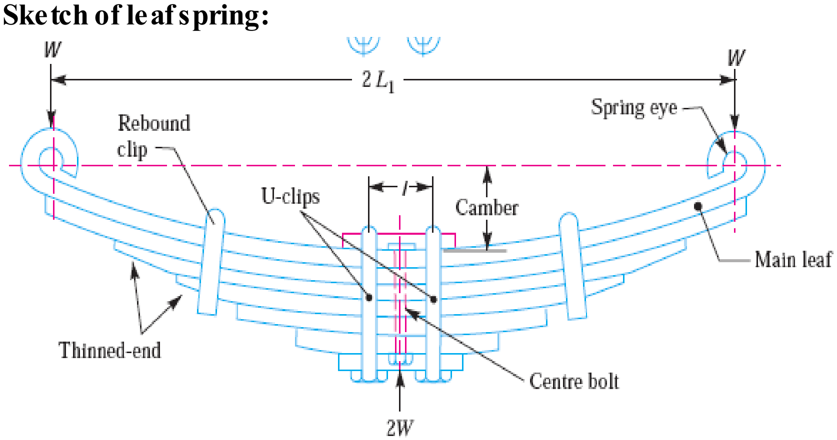
1) More Stronger due to more base thickness.
2) More strength than square thread
3) Load carrying capacity is more
4) Easy to manufacture
5) Low cost
6) easy availability.

Disadvantages: **Any 2 pt (1 Mark each)**

1) Efficiency is less
2) High friction
3) Radial pressure is more.

d) Construction of leaf spring with neat sketch

Sketch of leaf spring: **02 M**



Construction: **02 M**

It is built up of a number of plates (known as leaves). The leaves are usually given an initial curvature so that they will tend to straighten under the load.

The leaves are held together by means of a bolt passing through the centre.

The spring is clamped to the axle housing by means of U-bolts.

The longest leaf known as main leaf or master leaf has its ends formed in the shape of an eye through which the bolts are passed to secure the spring to its supports.

The other leaves of the spring are known as graduated leaves.

Rebound clips are located at intermediate positions in the length of the spring,

e) Function of spring: **(any 4) 01 mark each**

1. To provide cushion,
2. To absorb the shock
3. To apply forces, as in brakes, clutches and spring loaded valves.
4. To control motion by maintaining contact between two elements as in cams and followers.
5. To measure forces, as in spring balances and engine indicators.
6. To store energy, as in watches, toys, etc.