**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 creditfor anyequivalent figure drawn.
5) Credits may be given step wise for numerical problems. In some cases, the assumed constantvalues may vary and there may be some difference in the candidate's answers and model answer.
6) In case of some questions credit may be given by judgement on part of examiner of relevant answer based on candidate's understanding.
7) For programming language papers, credit may be given to any other program based on equivalentconcept.

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

**a)State the steps involved in general design procedure. (4marks)**

**1. Recognition of need**. First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed.

**2. Synthesis (Mechanisms).** Select the possible mechanism or group of mechanisms which will give the desired motion.

**3. Analysis of forces**. Find the forces acting on each member of the machine and the energy transmitted by each member.

**4. Material selection.** Select the material best suited for each member of the machine.

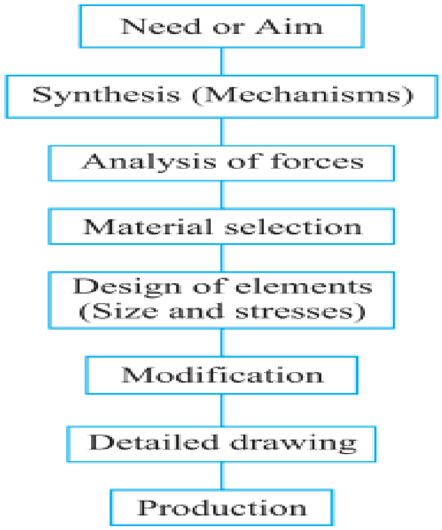
**5. Design of elements (Size and Stresses).** Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform than the permissible limit.

**6. Modification**. Modify the size of the member to agree with the past experience and judgment to facilitate manufacture. The modification may also be necessary by consideration of manufacturing to reduce overall cost.

**7. Detailed drawing**. Draw the detailed drawing of each component and the assembly of the machine with complete specification for the manufacturing processes suggested. 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

**8. Production.** The component, as per the drawing, is manufactured in the workshop.

**OR Each Step : ½ Mark )**




1. **A .b) 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. **2 M**

$$\boxed{\tau_{max} = \frac{\tau_{yt}}{FS}}$$

where $\tau_{max}$ = Maximum shear stress in a bi-axial stress system,

$\tau_{yt}$ = Shear stress at yield point as determined from simple tension test, and

$F.S.$ = Factor of safety. **1M**

Use: This theory is mostly used for designing members of ductile materials.**1M**

**1. A. c )List different types of keys:** **Any 8 types : 4M**

1) Rectangular sunk key 2) Square sunk key.3)Parallel sunk key.

4) Gib-head key 5)Feather key. 6)Woodruff key

7). Saddle keys, 8) Tangent keys, 9)Round keys and

10) Splines. **(Eachtype : ½ Mark )**

1. **A. d )Self locking of screw:**

Torque required to lower the load

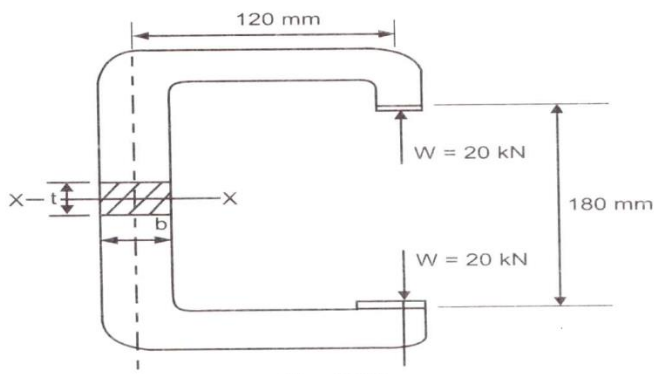
$$T = P \times \frac{d}{2} = W \tan(\phi - \alpha)\frac{d}{2}$$

If however , $\phi > \alpha$, the torque required to lower the load will be positive,indicating that an effort is applied to lower the load,such a screw is known as self locking screw.

A screw will be selflocking
1)if the friction angle is greater thanhelix angle or coefficient of friction is greater than tangent of helix angle $i.e.\ \mu$ or $\tan\phi > \tan\alpha$.
2) if the frequency efficiency is less than 50 % i.e$\eta$< 50% **( Correct Ans: 04 M )**

**1. B a) Design of C- Clamp:**

**Solution :**




Sketch : 1M

**Fig. 2.81**

**Given data :** $W = 20\text{ kN} = 20 \times 10^3\text{ N},\ b = 2t$

$e = 120\text{ mm},\ \sigma_t = 100\text{ N/mm}^2$

**At section X-X :**

**Step I : Direct stress :** $\sigma_o = \frac{W}{A} = \frac{20 \times 10^3}{b \times t} = \frac{20 \times 10^3}{2t^2} = \frac{10 \times 10^3}{t^2}$ 1 M

**Step II : Bending stress :** $\sigma_b = \frac{M}{Z} = \frac{W \times e}{\frac{1}{6} tb^2}$

$$\sigma_b = \frac{20 \times 10^3 \times 120 \times 6}{t \times (2t)^2} = \frac{3.6 \times 10^6}{t^3}$$

**Step III : Resultant stress :** 1 M

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

$$\therefore \quad \frac{10 \times 10^3}{t^2} + \frac{3.6 \times 10^6}{t^3} = 100$$

$$\therefore \quad \frac{10 \times 10^3 \times t + 3.6 \times 10^6}{t^3} = 100$$

$$\therefore \quad 10 \times 10^3\, t + 3.6 \times 10^6 = 100t^3$$

$$\therefore \quad 100t^3 - 10 \times 10^3\, t = 3.6 \times 10^6$$ 2 M

Divide the equation by 100,

$\therefore \quad t^3 - 100\, t = 3.6 \times 10^4$; Using trial and error method, we ge

**t= 34 mm**

**b= 2t =2 x 34 = 68 mm** **1 M**

**1.B. b) Wall Bracket Bolt size:** **6 Marks**

Given data :-

$W = 40\,kN = 40 \times 10^3\,N$

$L = 500\,mm$

$L_1 = 50\,mm$

$L_2 = 450\,mm$

$\sigma_t = 70\,N/mm^2$ $n = 4$

Direct shear load on each bolt

$$W_{s_1} = \frac{W}{n} = \frac{40 \times 10^3}{4} = 10 \times 10^3\,N$$ — 1 M

Max. tensile load carried by bolt

$$W_{t_1} = \frac{W \times L \times L_2}{2(L_1^2 + L_2^2)}$$

$$= \frac{40 \times 10^3 \times 500 \times 450}{2(50^2 + 450^2)}$$

$W_{t_1} = 21951\,N$ — 1 M

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

Equivalent tensile load

$$W_{te} = \frac{1}{2}\left[W_t + \sqrt{W_t^2 + 4(W_s)^2}\right]$$

$$= \frac{1}{2}\left[21951 + \sqrt{21951^2 + 4(10 \times 10^3)^2}\right]$$

$$= 25823.45\,N$$ — 2 M

For determining, size of bolt

$$\sigma_t = \frac{W_{te} =}{\frac{\pi}{4} dc^2} \Rightarrow 70 = \frac{25823.45}{\frac{\pi}{4} dc^2}$$

$$\therefore dc^2 = 469.70 \text{ m}$$

$$\therefore dc = 21.67 \text{ mm}$$ ——— 1M

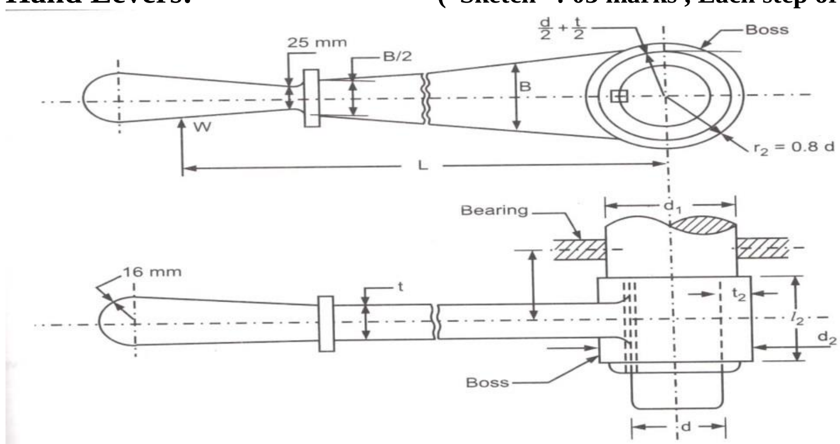
Nominal diameter $do = \frac{dc}{0.84}$

$$do = \frac{21.67}{0.84} = 21.67 \text{ mm}$$

$d_o = \frac{21.67}{0.84} = 25.79\text{mm}$ = M26 01 m

**Qu. 2) Attempt any TWO :Each : 08 Marks**

**a) Hand Levers:** **( Sketch : 03 marks , Each step or equation: 1M )**




Let
P = Force applied at the handle,
L = Effective length of the lever,
$\sigma t$ = Permissible tensile stress, and
$\tau$ = Permissible shear stress.

**Step I :** Considering shaft is under pure torsion, we have,

$$T = \frac{\pi}{16} \cdot \tau \cdot d^3$$

But, twisting moment on shaft,

$$T = P \times L$$

$\therefore$ On equating, we have,

$$P \times L = \frac{\pi}{16} \cdot \tau \cdot d^3$$

$\therefore$ Diameter of shaft (d) may be obtained.

**Step II :** Using the empirical relations, fix the other dimensions as,

$$d_2 = 1.6d$$
$$t_2 = 0.3d$$
$$l_2 = d \text{ to } 1.2d$$
$$l = 2 \times l_2$$

**Step III :** Considering the shaft supported at the centre of the bearing under combined twisting and bending moment, we have,

$$M = P \times l \text{ and } T = P \times L$$

$\therefore$ Equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(P \times l)^2 + (P \times L)^2} = P\sqrt{l^2 + L^2}$$

Also, equivalent twisting moment,

$$T_e = \frac{\pi}{16} \cdot \tau_{max} \cdot d^3$$

$$\therefore \quad P\sqrt{l^2 + L^2} = \frac{\pi}{16} \cdot \tau_{max} \cdot d^3$$

From here, value of $d_1$ = diameter of shaft supported at centre of bearings can be determined.

**Step IV : Design of key :** Let $t_1$ = thickness of key, w = width of key and $l_1$ = length of key.

After finding diameter of shaft (d), we can fix the dimensions for key as,

$$w = \frac{d}{4} \text{ and } t_1 = \frac{d}{6}$$

Considering shear failure of key,

We have, $$T = (w \cdot l_1 \cdot \tau) \times \frac{d}{2}$$

$\therefore$ Length $l_1$ can be determined.

Also, length $l_1$ may be taken as length of boss i.e. $l_2$.

**Step V :** Considering bending failure of lever, we can determine the cross-section of leve near the boss.

Let t = thickness of lever near the boss and B = width of height of lever near the boss.

We have,

Bending moment on the lever = $M = P \times (L - r_b)$

and Section modulus = $Z = \frac{1}{6} \cdot t \cdot B^2$

where, $r_b$ = Radius of boss = $\frac{d_2}{2}$

$$\therefore \quad \text{Bending stress, } \sigma_b = \frac{M}{Z} = \frac{P \cdot (L - r_b)}{\frac{1}{6} \cdot tB^2}$$

Width of lever may be taken as, B = 4t to 5t.

From this equation, values of t and B can be determined.

**( Sketch : 03 marks , Each step or equation: 1M )**

**Q.2. b) Design of Spigot Cotter joint:**

Given :-

$P = 100\ kN = 100 \times 10^3\ N$

$\sigma_t = 80\ N/mm^2$

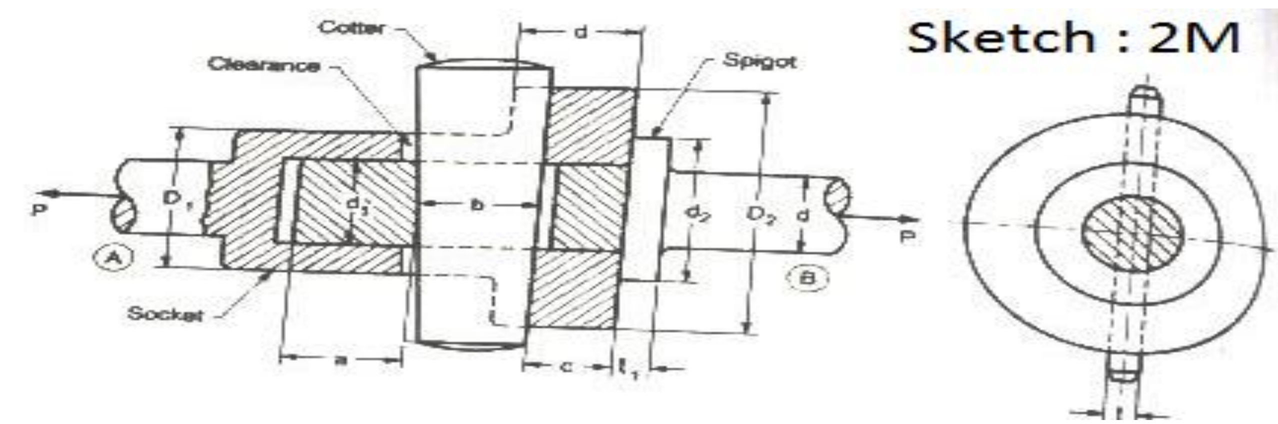
$\sigma_c = 100\ N/mm^2$

$\tau = 35\ N/mm^2$

The design load is 30 % overload

$\therefore$ Design load $= 1.3 \times P$

$P_d = 1.3 \times 100 = 130 \times 10^3\ N$




1> Dia. of rod ( d ) $\Rightarrow$

Tensile stress induced in rod

$$\sigma_t = \frac{P_d}{\frac{\pi}{4} d^2} \Rightarrow 80 = \frac{130 \times 10^3}{\frac{\pi}{4} d^2}$$

$\therefore d = 45.48 \simeq 46\ mm$ ———— 1 M

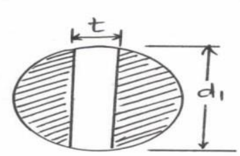
2> Thickness of cotter $= 0.3 \times d$

$= 0.3 \times 46 = 13.6 \simeq 14\ mm$

3> Dia. of spigot end :-

consider failure of spigot end in tension

$$\sigma_t = \frac{P_d}{\frac{\pi}{4} d_1^2 - d_1 t}$$




$$80 = \frac{130 \times 10^3}{\frac{\pi}{4} d_1^2 - d_1 \times 14}$$

$$\therefore \frac{\pi}{4} d_1^2 - 14\ d_1 = 1625$$

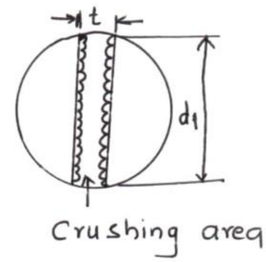
$d_1 = 50 \text{ mm}$

Crushing stress induced in a cottor

$$\sigma_c = \frac{P_d}{d_1 \times t}$$

$$100 = \frac{130 \times 10^3}{d_1 \times 14}$$

$$\therefore d_1 = 92.86 \text{ mm}$$




Select higher value of $d_1$

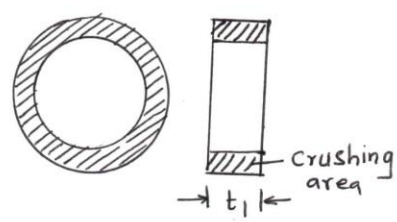
$$\therefore d_1 = 92.86 \text{ mm} \cong 94 \text{ mm}$$ ——— 1M

4) Dia. of Spigot collar ($d_2$) ⇒

Area of contact betn the spigot collar & socket collar will be subjected to crushing stress

$$\sigma_c = \frac{P_d}{\frac{\pi}{4}(d_2^2 - d_1^2)}$$

$$\therefore 100 = \frac{130 \times 10^3}{\frac{\pi}{4}(d_2^2 - 94^2)}$$

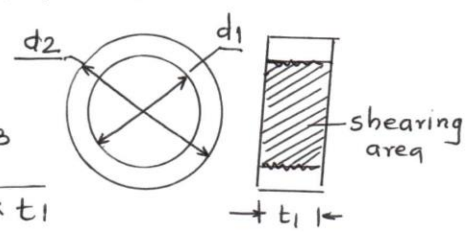



$$\therefore d_2 = 102.43 \text{ mm} \cong 104 \text{ mm}$$ ——— 1M

5) Thickness of Spigot collar $t_1$ :-

$$\tau = \frac{P_d}{\pi \cdot d_1 t_1}$$

$$\therefore 35 = \frac{130 \times 10^3}{\pi \times 94 \times t_1}$$




$$\therefore t_1 = 12.57 \text{ mm} \cong 14 \text{ mm}$$ ——— 1M

6) Distance from end of slot to end of Spigot (a) :-

Spigot end is subjected to double shearing

$$\tau = \frac{P_d}{2 \cdot d_1 \times a} \Rightarrow 35 = \frac{130 \times 10^3}{2 \times 94 \times a}$$

$$a = 19.75 \text{ mm} \simeq 20 \text{ mm} \longrightarrow 1\text{M}$$

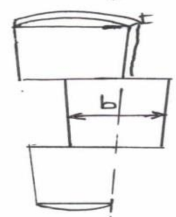
7) Width of cottor :-

cottor is subjected to double shear

$$\tau = \frac{P_d}{2bt}$$

$$\therefore 30 = \frac{130 \times 10^3}{2 \times b \times 14}$$

$$\therefore b = 132.65 \text{ mm} \simeq 134 \text{ mm}$$




——— 1 M

1 mark for Spigot diameter $d_1$=50mm

Q.2 c) Design of muff Coupling: 8 M

Given data :-

Design of muff coupling

$P = 40 kw = 40 \times 10^3 W$

$N = 350$ r.p.m

$\tau_s = \tau_{key} = 40 N/mm^2$

$\sigma_{c_{shaft,key}} = 80 N/mm^2$

$\tau_{muff} = 15 N/mm^2$

Torque transmitted by shaft,

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

$$\therefore 40 \times 10^3 = \frac{2\pi \times 350 \times T}{60}$$

$$\therefore T = 1091.35 \ N.m$$

$$T = 1091.35 \times 10^3 \ N.mm \text{ ———— } 1M$$

Diameter of shaft :-

$$T = \frac{\pi}{16} d^3 \times \tau_{shaft}$$

$$1091.35 \times 10^3 = \frac{\pi}{16} \times d^3 \times 40$$

$$d^3 = 138954.99$$

$$\therefore d = 51.79 \ mm \simeq 52 \ mm \text{ ———— } 1M$$

Dia. of Sleeve or muff :-

Outside diameter of Sleeve

$$D = 2d + 13$$

$D = (2 \times 52) + 13$

$D = 117\ mm$ ———— 1M

Length of muff $L = 3.5 \times d$

$= 3.5 \times 52$

$= 182\ mm$ ———— 1M

Checking $\tau_{muff} = 15\ N/mm^2$

$T = \frac{\pi}{16} \tau_{muff} \times D^3 (1 - k^4)$

$k = \frac{d}{D} = 0.44$

$1091.35 \times 10^3 = \frac{\pi}{16} \times \tau_{muff} \times 117^3 (1 - 0.44^4)$

$\therefore \tau_{muff} = 3.6\ N/mm^2 < 15\ N/mm^2$

$\therefore$ Design of sleeve is safe.. ———— 1M

Design of Key :-

Width of key $w_k = \frac{d}{4} = \frac{52}{4} = 13\ mm$

$t_k = \frac{d}{4} = 13\ mm$ ———— 1M

length of key $L_K = \frac{L}{2} = \frac{182}{2} = 91\ mm$ ———— 1M

Shearing strength of key :-

$T = w_k \times l_k \times \tau_{key} \times \frac{d}{2}$

$1091.35 \times 10^3 = 13 \times 91 \times \tau_{key} \times \frac{52}{2}$

$$\tau_{key} = 35.48 \ N/mm^2 < 40 \ N/mm^2$$
(calculated)

Thus, the induced shear stress in key is less than permissible stress $\therefore$ Design of key is safe.

Crushing strength of key

$$T = l_K \times \frac{t_K}{2} \times \sigma_{c\,key} \times \frac{d}{2}$$

$$1091.35 \times 10^3 = 91 \times \frac{13}{2} \times \sigma_{c\,key} \times \frac{52}{2}$$

$$\therefore \sigma_{c\,key} = 70.96 \ N/mm^2 < 80 \ N/mm^2$$

$\therefore$ Design of key is safe. ——— 1 M

**Qu. 3. Attempt any Four** **Each 4 Marks**

a) **Following are the factors which govern the factor of safety.**

i) The extent of loss of life it failure occurs.
ii) The extent of loss of property if failure occurs.
iii) The reliability of properties of material.
iv) The extent of assumption made in design process.
v) The reliability of applied load.
vi) The extent of stress concentration.
vii) Types of loading i.e whether static,fatigue,impact etc.
viii)service conditions
ix) quality of manufacturing **( any Four points : 4 M)**

b) **Causes of stress concentration are as under**.
i) Abrupt changes in cross-section like in keyway, steps, grooves, threaded holes results instress concentration. **1M**

ii) Poor surface finish – The surface irregularities is also one of the reason for stress concentration. **1M**

iii) Localized loading – Due to heavy load on small area the stress concentration occurs inthe vicinity of loaded area. **1 M**

iv) Variation in material properties – Particularly defects like internal flaws, voids, cracks, airholes, cavities also results in stress concentration. **1M**

C)**Lewis equation for strength of gear tooth**

$$\boxed{W_T = \sigma_w \,.\, b \,.\, p_c \,.\, y = \sigma_w \,.\, b \,.\, \pi m \,.\, y \qquad ...(\because p_c = \pi m)}$$

$W_T$ = Tangential load acting at the term tooth**( Equation :2 marks)**
= Beam strength of the tooth
b = Width of the gear face
Pc = Circular pitch
**m** = Module

Y is known as Lewis form factor or tooth form factor **(Meaning of terms – 2 marks)**

d)

1) 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.

**2)** 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.

**Correct Answer : 04 M**

**e) Advantages of rolling contact bearing:**

1.Low starting and running friction except at very high speeds.
2. Ability to withstand momentary shock loads.
3. Accuracy of shaft alignment.
4. Low cost of maintenance, as no lubrication is required while in service.
5. Small overall dimensions.
6. Reliability of service.
7. Easy to mount and erect.
8. Cleanliness. **Any 4 advantages: 04 M**

**Q.4. (A) a) Define Following**

**i) Transverse Shear stress: -** When a section is subjected to two equal and opposite forces acting tangentially across the section such that it tends to shear off across the section. The stress produces is called as transverse shear stress.

**……………………01 mark**

From figure

Mathematically transverse shear stress is represented as,

$$\tau = \frac{F}{A}$$

Where,

F = Tangential force applied

A = Area of cross section = $\frac{\pi}{4} d^2$

d = Diameter of rivet.

**……………………01 mark**

**ii) Torsional Shear stress:** When a machine component is under the action of two equal and opposite couples i.e. twisting moment or torque, then component is said to be torsional and the stresses set up due to torsion are called as torsional shear stress.

**……………………01 mark**

Consider a component of circular cross-section. ‘d’ in diameter, subjected to torque T, Torsional shear stress is given by, basic torsion equation

$$\frac{T}{J} = \frac{\tau}{r}$$

$$\tau = \frac{T.r}{J}$$

Where,

r = distance of outer fibre from neutral axis = d/2

J = Polar moment of inertia of cross- section = $\frac{\pi}{64} d^4$

**……………………01 mark**

**Q.4. (A) b) Designation of following material**

**i) Fe 500:- S**teel with minimum tensile strength of 500 N/mm$^2$.

**……………………02mark**

**ii) Fe X20Cr18Ni2:-** Alloy steel having 0.2 %Carbon, 18% Chromium and 2% Nikel

**……………………02mark**

**Q.4. (A) c) Design of Shaft to combine bending moment and twisting moment by maximum shear stress theory**

We know that for maximum shear stress theory

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4(\tau)^2} \qquad \text{Equation 01}$$

We know that, **T** = $\frac{\pi}{16}.\tau.d^3$

$$\tau = \frac{16T}{\pi d^3}$$

**…....01 mark**

Also M = $\frac{\pi}{32}.\sigma_b.d^3$

$$\sigma_b = \frac{32M}{\pi d^3}$$

**…....01 mark**

Put the values of $\tau$ and $\sigma_b$ in equation 01

$$\tau_{max} = \frac{1}{2}\sqrt{\left(\frac{32M}{\pi d^3}\right)^2 + 4\left(\frac{16M}{\pi d^3}\right)^2}$$

$$\tau_{max} = \frac{1}{2}\left(\frac{32}{\pi d^3}\right)\sqrt{M^2 + T^2}$$

**…....01 mark**

$$\sqrt{M^2 + T^2} = \left(\frac{\pi}{16}\right)\tau_{max} \cdot d^3$$

The expression $\sqrt{M^2 + T^2}$ is called as equivalent twisting moment and denoted by $T_e$.

By limiting the maximum shear stress $\tau_{max}$ equal to allowable shear stress $\tau$

$T_e = \sqrt{M^2 + T^2} = \left(\frac{\pi}{16}\right) \tau \, . \, d^3$ …....01 mark

**Q.4. (A) d) Define the following terms related to spring (01 mark each)**

**i)** **Solid length : -** When compression spring is compressed until the coils come in contact with each other, then spring is said to be solid. The solid length of spring is a product of total number of coils and diameter of wire.

**ii)** **Free length: -** it is length of spring when the spring is in free or unloaded condition.

**iii)** **Spring rate: -** it is defined as the load required per unit deflection of the spring.

iv) **Spring index: -** It is define as the ratio of mean diameter of coil to the diameter of spring wire.

**Q.4. (B) a) Derivation of efficiency of self-locking screw is less than 50%**

Efficiency of screw is expressed as,

$$\eta = \frac{tan\alpha}{\tan(\phi+\alpha)}$$

**……………….01mark**

We know that condition for self- locking is $\varphi \geq \alpha$

**……………….01mark**

Assume $\varphi = \alpha$ and put in equation no. 01

$$\eta = \leq \frac{tan\phi}{\tan 2\phi}$$

**……………….01mark**

$$= \leq \frac{tan\phi}{\frac{2\tan\phi}{1-tan^2\phi}}$$

**……………….01mark**

$$= \leq \frac{1-tan^2\phi}{2}$$

$$= \leq \frac{1}{2} - \frac{tan^2\phi}{2}$$

**……………….02mark**

This means that efficiency is less than 50%

If efficiency is more than 50%, the screw is said to be overhauling.

**Q.4. (B) b) Given data: -**

Effective diameter of cylinder = D = 300 mm.

Steam pressure = P = 0.6 N/mm$^2$.

Nominal diameter of bolt = $d_o$ = 20 mm.

Working stress for bolt (i.e. Tensile stress σt) = 20 N/mm$^2$.

Core diameter of bolt ($d_c$) = 0.84 $d_0$ = 0.8 X 20 = 16.8 mm

**……………….01mark**

Total Force acting on the cylinder.

$F_n$ = Pressure inside the cylinder X Area of cylinder

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

= 0.6 X $\frac{\pi}{4}300^2$

= 42.4115 X 10$^3$ N

**……………….02mark**

This load is taken up by n number of bolts.

Force on each bolt, F = $\frac{F_n}{n} = \frac{42.411\ X\ 10^3}{n}$

**……………….01mark**

Consider the failure of bolt in tension

$$\sigma_t = \frac{F}{\frac{\pi}{4}(d_c)^2}$$

$$= \frac{42.411\ X\ 10^3}{\frac{\pi}{4}(d_c)^2\ .\ n}$$

$$n = \frac{42.411\ X\ 10^3\ X\ 4}{\pi\ \sigma_t\ (d_c)^2}$$

n = $\dfrac{42.411 \, X \, 10^3 \, X \, 4}{\pi \, 20 \, (16.8)^2 \, .}$

**………………..02mark**

n = 9.566 ≈ 10.

Number of bolts required = 10

**Q.5. (A) Given data: -**

Weight of pulley = W= 400N

Diameter of pulley = $D_p$ = 270 mm

Radius of pulley R = $\frac{270}{2}$ = 135 mm

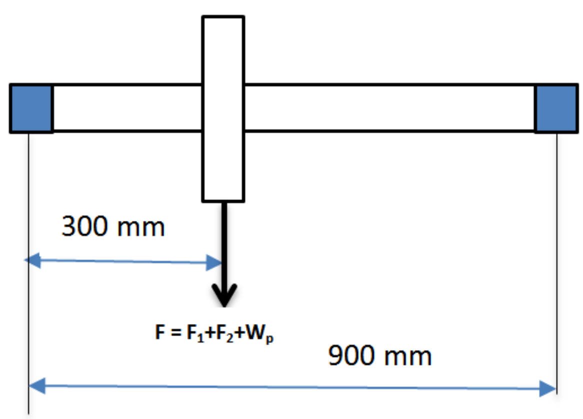
Power = P = 5Kw = 5 X$10^3$ watt.

Speed of shaft = N= 200 rpm.

Tension ratio = $\dfrac{F_1}{F_2}$ = 2.5 i.e. $F_1 = 2.5\ F_2$

Shear stress = 40 $N/mm^2$

**(Diagram 01 mark)**




**Solution: -** Calculate Torque by using following equation

P = $\dfrac{2\pi NT}{60}$

T = $\dfrac{60P}{2\pi N}$

= $\dfrac{60 \, X \, 5 \, X \, 10^3}{2\pi \, X \, 200}$

T = 238.7 N-mm

T =238.7 X $10^3$ N-mm

**(01 mark)**

Calculate tight and slack side tension

$F_1 = 2.5\ F_2$

Torque (T) = $(F_1 - F_2)$ X R

Put value of F2 in above equation we get

T = $(2.5F_2 - F_2)$ X R

238.7 X $10^3$ = 1.5 $F_2$ X 135

$$F_2 = \frac{238.7\ X\ 10^3}{1.5\ X\ 135} = 1178.76 \text{ N}$$

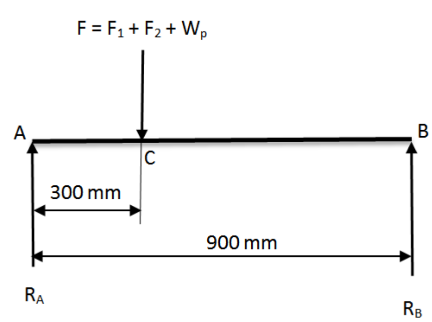
$F_1 = 2.5\ F_2 = 2.5$ X 1178.76 = 2946.9 N

**(01 mark)**

Calculate Bending moment

Total Force acting at point C ( F) = $F_1 + F_2 + W_p$ = 2946.9 + 1178.76 + 400 = 4525.66 N

**(Diagram 01 mark)**




From Fig. Take moment at point A and equate it to zero

Moment about A = 0

F X 300 – $R_B$ X 900 = 0

4525.66 X 300 = $R_B$ X 900

$$R_B = \frac{4525.66\ X\ 300}{900} = 1508.55 \text{ N}$$

**(01 mark)**

Take summation of all force = 0

$R_A + R_B - F = 0$

$R_A = F - R_B = 4525.66 - 1508.55 = 3017.11$ N

Take moment at point C

$M = R_A \times 300 = 3017.11 \times 300 = 905133$ N-mm

**(01 mark)**

Equivalent Torque = $T_e = \sqrt{M^2 + T^2}$

$$T_e = \sqrt{(905133)^2 + (238.7X10^3)^2}$$

$$T_e = 936.078 \times 10^3 \text{ N-mm}$$

**(01 mark)**

We know that

Shear stress for shaft (τ) = $\frac{16\,T_e}{\pi d^3}$

$d^3 = \frac{16\,T_e}{\pi \tau}$

$= \frac{16 \, X \, 936.078 \, X \, 10^3}{\pi \, X \, 40} = 119.185 \text{ X } 10^3$

d = 49.21 mm

**Diameter of shaft = 50 mm**

**(01 mark)**

**Q. 5. (b) Given data: -**

Outside diameter of lead screw ($d_0$) = 60 mm.

Pitch (p) = 8mm.

Axial load = 300kg

Axial load in newton (W) = 300 *9.81 = 2943 N.

Bearing collar outside diameter $D_1$ = 12 cm = 120mm.

Bearing collar outside Radius $R_1$ = 6 cm = 60mm.

Bearing collar Inside diameter $D_2$ = 6 cm = 60mm.

Bearing collar outside Radius $R_2$ = 3 cm = 30mm.

Rotation of screw (N) = 50 rpm.

Coefficient of friction for screw thread (μ) = 0.18

Coefficient of friction for collar ($\mu_2$) = 0.14

**Solution: -**

We know that mean diameter of screw

$$d = d_0 - \frac{p}{2} = 60 - \frac{8}{2} = 56\text{mm.}$$

$$\tan\alpha = \frac{p}{\pi d} = \frac{8}{56\pi} = 0.04547.$$

**(01 mark)**

Since the angle for ACME threads is $2\beta = 29^0$ or $\beta = 14.5^0$, therefore virtual coefficient of friction

$\mu_1 = \tan\phi_1$

$$= \frac{\mu}{cos\beta} = \frac{0.18}{cos 14.5} = \frac{0.18}{0.9681} = 0.18592.$$

**(01 mark)**

We know that the force required to overcome friction at screw,

$$F = W \tan(\alpha + \phi_1) = W \frac{tan\alpha + tan\Phi_1}{1 - tan\alpha * tan\Phi_1} =$$

$$= 2943 \times \frac{0.04547 + 0.18592}{1 - 0.04547 * 0.18592}$$

$$F = 686.89 \text{ N}$$

**(01 mark)**

Torque required to overcome friction at screw.

$$T_1 = F * \frac{d}{2} = 686.89 * \frac{56}{2} = 19232.92 \text{ N-mm.}$$

**(01 mark)**

We know that mean radius of collar.

$$R = \frac{R_1 + R_2}{2} = \frac{60+30}{2} = 45 \text{ mm.}$$

Assuming uniform wear, torque required to overcome friction at collars,

$T_2 = \mu_2$ W R = 0.14 * 2943 * 45 = 18540.9 N-mm.

**(01 mark)**

Total torque required to overcome friction,

$T = T_1 + T_2$ = 19232.92 + 18540.9 = 20473.82 N-mm. = 20.47382 N-m

We know that power required to drive the screw = $\frac{2\pi NT}{60} = \frac{2\pi * 50 * 20.47382}{60} =$

107.201Kw

Power required to drive screw = 107.201 Kw

**(01 mark)**

Efficiency of the screw

We know that the torque required to drive the screw with no friction,

$T_0$ = W tanα $\frac{d}{2}$ = 2943 * 0.04547 * $\frac{56}{2}$ = 3746.9099 N-mm = 3.74691 N-m

**(01 mark)**

Efficiency of the lead screw

$\eta = \frac{T_0}{T} = \frac{3.74691}{20.47382}$ = 0.183 = 18.3%

**(01 mark)**

**Q. 5. (C) Given data: -**

Pressure (P) = 1.2 N/mm$^2$.

Diameter of valve (D) = 65 mm.

Maximum lift of valve ($\delta_2$) = 17.5 mm.

Spring index (C) = 6.

Initial compression ($\delta_1$) = 30 mm.

Maximum permissible shear stress in spring material (τ) = 450 N/mm$^2$.

**Solution: -**

**I) Calculate diameter of spring wire(d)**

We know that maximum load acting on the valve when it just begins to blow off.

$F_1$ = Area of valve X Maximum pressure

$= \frac{\pi}{4} D^2 X 1.2 = \frac{\pi}{4} 65^2 X 1.2$

$F_1$ = 3981.9687 N

**(01 mark)**

Maximum compression of spring ($\delta_{max}$) = $\delta_1 + \delta_2$ = 30 + 17.5 = 47.5 mm

**(01 mark)**

Since a load of 3981.9687 N keeps the valve on its seat by providing initial compression of 35 mm. therefore the maximum load on the spring when the valve is open (i.e. for maximum compression of 47.5 mm)

$F = \frac{F_1}{\delta 1} X \delta_{max}$

$= \frac{3981.9687}{30} X 47.5$

F = 6304.7838 N

**(01 mark)**

We know that Wahl's stress factor ($K_w$) = $\frac{4C-1}{4C-4} + \frac{0.615}{C}$

$= \frac{4*6-1}{4*6-4} + \frac{0.615}{6}$

$K_w$ = 1.2525

**(01 mark)**

We also know that the maximum shear stress ($\tau$)

$\tau = K_w \frac{8 F C}{\pi d^2}$

$500 = 1.2525 X \frac{8* 6304.7838*6}{\pi d^2}$

$d^2 = 1.2525 X \frac{8* 6304.7838*6}{500\pi} = 241.30$

d = 15.53 mm

**(01 mark)**

**ii) Calculate mean coil diameter (D)**

Mean coil diameter = D = C*d= 6*15.53 = 93.14 mm

**iii) Calculate Number of active turns**

We know that maximum compression of the spring (δ),

$$\delta_{max} = \frac{8\ F\ C^3\ n}{G\ d}$$

$$n = \frac{G d \delta_{max}}{8\ F\ C^3}$$

$$n = \frac{0.84 * 10^5 * 15.53 * 47.5}{8 * 6304.7838 * 6^3} = 5.687 \text{ say } 6$$

**(01 mark)**

Taking the ends of the coil as squared and ground, the total number of turns

n' = n+2 = 6+2 = 8

**iV) Calculate pitch of spring**

We know that free length of the spring = $L_f = n'd + \delta_{max} + 0.15\delta_{max}$

$= 8 * 15.53 + 47.5 + 0.15*47.5$

$= 178.865$ mm

**(01 mark)**

Pitch of the coil (P) = $\frac{Free\ lemgth}{n'-1} = \frac{178.865}{8-1} = 25.55$ mm

**(01 mark)**

**Q.6. a) Difference between Sliding contact and rolling contact bearing (Any four and 01 mark each)**

| Sr. No. | Sliding Contact Bearing | Rolling Contact Bearing |
|---|---|---|
| 1 | it requires more Axial space | It requires Considerable radial space |

| | | |
|---|---|---|
| **2** | It is linearly proportional to the speed. | It has fixed load carrying capacity depending upon its size |
| **3** | It is suitable for high load – high speed condition | It is independent of speed and load carrying capacity is given in manufacturer's catalogue |
| **4** | it's Starting torque is comparatively high | it's Starting torque is comparatively low |
| **5** | it's noise is comparatively low | it's noise is comparatively high |
| **6** | it's housing design is simple | it's housing design is complicated |
| **7** | High initial cost | Low initial cost |
| **8** | Axes of journal moves eccentrically with respect to axis of the bearing | Axes of journal and bearing are co-liner. |

**Q.6. b) Importance of following in aesthetic design. (02 marks each)**

**i) Shape: -**

1. There are five basic shapes (form) of the product, viz., step, taper, shear, streamline and sculpture, as shown in fig.
2. The external shape of any product is based on one or combination of these basic shapes.

**a. Step form: -** The step form is a stepper structure having vertical accent. It is similar to the shape of multistory building.

**b. Taper form: -** The taper form consists of tapered blocks or tapered cylinders.

**c. Shear form: -** The shear form has a square outlook.

**d. Streamline form: -** The streamline form has a streamline shape having a smooth flow as seen in automobile and aeroplane structures.

**e. Sculpture form: -** The sculpture form consists of ellipsoids , paraboloids and hyperboloids.

**ii) Size: -**

**1.** Due to advancements in electronic fields, designers can use previously unaccepted housing for integrated items, so freeing them from many of design constraints.

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

**Q.6. c) Advantages of "V" thread over square thread (Any four, 01 marks each)**

1. These are used to tighten the parts together in bolts, studs, nuts, tap bolts etc.
2. V- threads offer greater frictional resistance of motion than square thread hence better suitable for fastening purpose
3. These are stronger than square thread.
4. These are cheaper than Square threads.
5. These thread are easy to cut by die or on machine than square thread.

**Q.6. d) Sketch of Semi- elliptical Leaf spring (02 mark sketch, 02 marks labeling)**

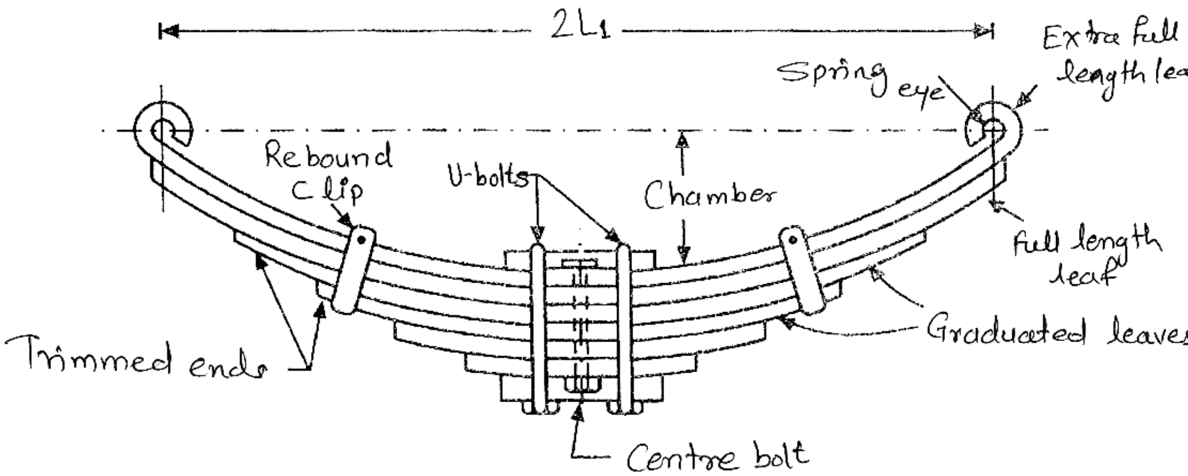



Fig. Semi-elliptic leaf spring

**Q.6. e) Function of spring (Any four, 01 marks each)**

1. To cushion, absorb or control energy due to either shock or vibration as in car spring, railway buffers, shock absorbers and vibration dampers.
2. To apply forces as in brakes, clutches and spring loaded valve
3. To measure forces, as in spring balance and engine.
4. To store energy, as in watches, toys.
5. To control motion by maintaining contact between two elements.
6. To change the vibration characteristics of a member as in flexible mounting of motor.

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