

MAHARASHTRA STATE BOARD OF TECHNICAL EDUCATION  
(Autonomous)  
(ISO/IEC - 27001 - 2005 Certified)  
**SUMMER-15 EXAMINATION**

Subject Code: **17610**

**Model Answer**

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**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.1- A) Attempt any THREE**

**(3 x 4) (12)**

**a) What is factor of safety? Define it for brittle & ductile loading.**

**Ans:** It is the ratio of the maximum stress to the working stress.

In order to prevent the failure of the component, designer assuming a value of design stress which is very less as compared to ultimate or yield stress

.....1 Mark

Mathematically,

$$\text{Factor of safety} = \frac{\text{Maximum stress}}{\text{Working stress or design stress}}$$

.....1 Mark

For Brittle : It is the ratio of the Ultimate stress to the working or design stress

$$\text{Factor of safety} = \frac{\text{Ultimate stress}}{\text{Working stress or design stress}}$$

..... 1 Mark

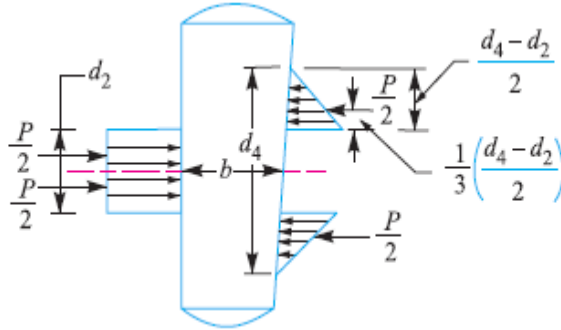
For Ductile: It is the ratio of the Yield stress to the working stress

$$\text{Factor of safety} = \frac{\text{Yield stress}}{\text{Working stress or design stress}}$$

.... 1 Mark

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**b) State with neat diagram about “Failure of Cotter In bending”**



..... sketch 1&1/2 Mark

It is assumed that the load is uniformly distributed over the various cross-sections of the joint. But in actual practice, this does not happen and the cotter is subjected to bending. In order to find out the bending stress induced,

it is assumed that the load on the cotter in the rod end is uniformly distributed while in the socket end it varies from zero at the outer diameter ( $d_4$ ) and maximum at the inner diameter ( $d_2$ ), as shown in Fig.

The maximum bending moment occurs at the center of the cotter and is given by

$$M_{max} = \frac{P}{2} \left( \frac{1}{3} \times \frac{(d_4 - d_2)}{2} + \frac{P}{2} \times \frac{d_2}{4} \right)$$

$$M_{max} = \frac{P}{2} \left( \frac{d_4 - d_2}{6} + \frac{d_2}{2} - \frac{d_2}{4} \right)$$

$$M_{max} = \frac{P}{2} \left( \frac{d_4 - d_2}{6} + \frac{d_2}{4} \right)$$

We know the section modulus of the cotter

$$Z = t \times \frac{b^2}{6}$$

Bending stress induced in the cotter

$$\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2} \left( \frac{d_4 - d_2}{6} + \frac{d_2}{4} \right)}{t \times \frac{b^2}{6}}$$

This bending stress induced in the cotter should be less than the allowable bending stress of the cotter.

**Explanation.... 2 &1/2 marks**

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**c) Prove that for a square key, permissible crushing stress is always twice the permissible shearing stress.**

Ans: Let,

T = Torque transmitted by the shaft

F = Tangential force acting at the circumference of the shaft

d = Diameter of shaft

l = Length of key

w = Width of key

t = Thickness of key and

$\tau$  &  $\sigma_c$  = shear and crushing stresses for the material of key.

A little consideration will show that due to the power transmitted by the shaft, the key may fail due to shearing or crushing.

Considering shearing of the key, the tangential shearing force acting at the circumference of the shaft,

$$F = \text{Area resisting shearing} \times \text{Shear stress} = l \times w \times \tau$$

Therefore, Torque transmitted by the shaft,

$$T = F \times \frac{d}{2} = l \cdot w \cdot \tau \cdot \frac{d}{2} \dots\dots\dots \mathbf{1 \text{ mark}}$$

Considering crushing of the key, the tangential crushing force acting at the circumference of the shaft,

$$F = \text{Area resisting crushing} \times \text{Crushing stress} = l \cdot \frac{t}{2} \cdot \sigma_c$$

Therefore, Torque transmitted by the shaft

$$T = F \times \frac{d}{2} = l \cdot \frac{t}{2} \cdot \sigma_c \cdot \frac{d}{2} \dots\dots\dots \mathbf{II \dots\dots\dots 1 \text{ mark}}$$

The key is equally strong in shearing and crushing, If equ. I = equ. II

$$l \cdot w \cdot \tau \cdot \frac{d}{2} = l \cdot \frac{t}{2} \cdot \sigma_c \cdot \frac{d}{2}$$

As key is square,  $w=t=d/4$

$$l \cdot \frac{d}{4} \cdot \tau \cdot \frac{d}{2} = l \cdot \frac{d}{8} \cdot \sigma_c \cdot \frac{d}{2}$$

For the usual key material crushing stress is twice the shear stress

$$\sigma_c = 2 \cdot \tau \dots\dots\dots \text{Proved} \dots\dots\dots \mathbf{2 \text{ marks}}$$

**d) Explain the various types of screw threads used in power screws**

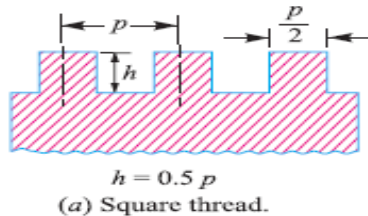
Ans:

Following are the three types of screw threads mostly used for power screws:

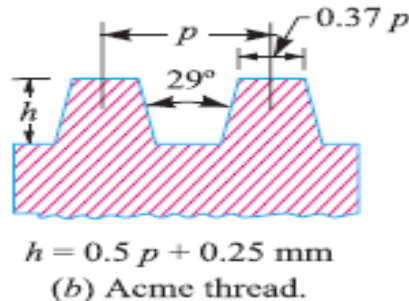
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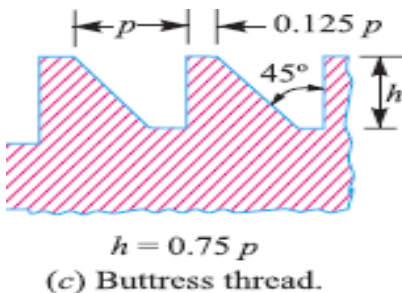
**1. Square thread.** A square thread, as shown in Fig (a) .It is adapted for the transmission of power in either direction. This thread results in maximum efficiency and minimum radial or bursting pressure on the nut. It is difficult to cut with taps and dies. It is usually cut on a lathe with a single point tool and it cannot be easily compensated for wear. The square threads are employed in screw jacks, presses and clamping devices.



**2. Acme or trapezoidal thread.** An acme or trapezoidal thread, as shown in Fig. (b), is a modification of square thread. The slight slope given to its sides lowers the efficiency slightly than square thread and it also introduce some bursting pressure on the nut, but increases its area in shear. It is used where a split nut is required and where provision is made to take up wear as in the lead screw of a lathe. Wear may be taken up by means of an adjustable split nut. An acme thread may be cut by means of dies and hence it is more easily manufactured than square thread.



**3. Buttress thread.** A buttress thread, as shown in Fig. (c), is used when large forces act along the screw axis in one direction only. This thread combines the higher efficiency of square thread and the ease of cutting and the adaptability to a split nut of acme thread. It is stronger than other threads because of greater thickness at the base of the thread. The buttress thread has limited use for power transmission. It is employed as the thread for light jack screws and vices.



.....Correct dia. &Explanation. 4 marks

**B) Attempt any ONE**

**1x 06 Marks**

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- a) State and describe in brief about any six ergonomics considerations in the designing of machine elements.

**Ergonomics consideration in control design:**

- 1) The control should be accessible and logically positioned.
- 2) The shape of control component which come in contact with hands should be comfortable with anatomy of hand.
- 3) Control should be painted in red color with grey background of machine tool for attention.
- 4) The control operation should involve Minimum motions.

**Ergonomics consideration in Display design:**

- 5) The scale on the dial indicator should be divided in suitable linear progression such as 0-10-20-30.
- 6) Number of subdivisions between divisions should be minimum.
- 7) Vertical figures should be used for stationary dials and radially oriented figures should be used for rotating dials.
- 8) The height of letters or numbers on display should be greater or equal to reading distance /200.
- 9) The pointer should have knife edge with a mirror to minimize parallax error.
- 10) The numbering should increase in clockwise direction on circular scale, rightward on a horizontal scale and upwards on vertical scale.

**Lighting:**

- 11) The surrounding area including walls, ceiling, floor and other object should be bright and more colored than workplace. The light should be match the needs of the task as far as illumination is concerned.

**Noise:**

- 12) If the noise level is too high, it can be stopped at source by better maintenance of equipment, placing vibration isolating material, plug in ears and providing sound insulating walls.

**Temperature:**

- 13) In order to get efficiency to perform the task, the operator should feel neither too hot nor too cold but comfortable.
- 14) When the heavy work is done, the temperature should be lowered and when the office work is done, it should be little higher.

**Humidity and Air circulation:**

- 15) Low humidity may cause discomfort through drying of the nose and throat.
- 16) Air humidity and air velocity become important at high temp. because they influence the amount of sweat, which can be evaporated from body surface to produce cooling effect

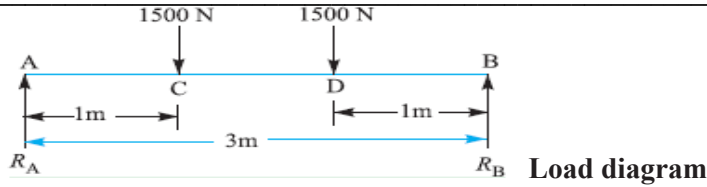
.....any 6 , 1 mark each

- b) A shaft made of mild steel is required to transmit 100 kW at 300 rpm. The supported length of the shaft is 3 meters. It carries two pulleys each weighing 1500 N supported at a distance of 1 meter from the ends respectively. Determine the diameter of the shaft. Take  $f_s=60 \text{ N/mm}^2$  &  $f_b=90 \text{ N/mm}^2$

Given :  $P = 100 \text{ kW} = 100 \times 10^3 \text{ W}$  ;  $N = 300 \text{ r.p.m.}$  ;  $L = 3 \text{ m}$  ;  $W = 1500 \text{ N}$

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..... **1 mark**

We know that the torque transmitted by the shaft,

$$T = \frac{P \times 60}{2\pi n}$$

$$T = \frac{100 \times 10^3 \times 60}{2\pi \times 300} = 3183 \text{ N-m}$$

..... **1 mark**

The shaft carrying the two pulleys is like a simply supported beam as shown in Fig. The reaction at each support

will be 1500 N, *i.e.*  $R_A = R_B = 1500 \text{ N}$

..... **1 mark**

A little consideration will show that the maximum bending moment lies at each pulley *i.e.* at C and D. Maximum bending moment,

$$M = 1500 \times 1 = 1500 \text{ N-m}$$

..... **1 mark**

Let  $d$  = Diameter of the shaft in mm.

We know that equivalent twisting moment,

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

$$T_e = \sqrt{(1500^2 + 3183^2)} = 3519 \text{ N-m} = 3519 \times 10^3 \text{ N.mm}$$

..... **1 mark**

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

$$3519 \times 10^3 = \frac{\pi}{16} \times 60 \times d^3$$

$$d^3 = \frac{33519 \times 10^3}{11.8} = 298 \times 10^3$$

$$d = 66.8 \text{ say } 70 \text{ mm}$$

Diameter of shaft  $d = 66.8 \text{ say } 70 \text{ mm}$

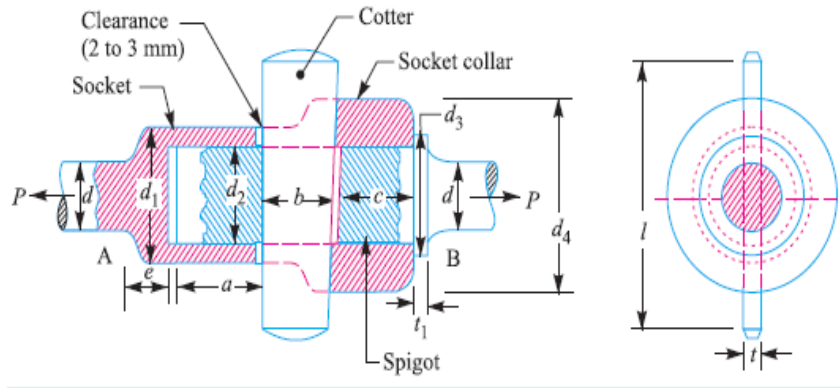
..... **1 mark**

**Q.2 Attempt any TWO :**

a) State the design procedure of a cotter joint with neat diagram.

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Let  $P$  = Load carried by the rods,  
 $d$  = Diameter of the rods,  
 $d_1$  = Outside diameter of socket,  
 $d_2$  = Diameter of spigot or inside diameter of socket,  
 $d_3$  = Outside diameter of spigot collar,  
 $t_1$  = Thickness of spigot collar,  
 $d_4$  = Diameter of socket collar,  
 $c$  = Thickness of socket collar,  
 $b$  = Mean width of cotter,  
 $t$  = Thickness of cotter,  
 $l$  = Length of cotter,  
 $a$  = Distance from the end of the slot to the end of rod,  
 $\sigma_t$  = Permissible tensile stress for the rods material,  
 $\tau_c$  = Permissible shear stress for the cotter material, and  
 $\sigma_c$  = Permissible crushing stress for the cotter material.

.....sketch 1 mark & procedure 7 marks

### 1. Failure of the rods in tension

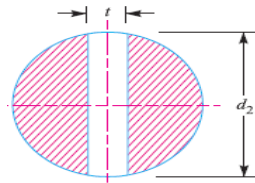
The rods may fail in tension due to the tensile load  $P$ . We know that

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

From this equation, diameter of the rods ( $d$ ) may be determined.

### 2. Failure of spigot in tension across the weakest section (or slot)

$$P = \left[ \frac{\pi}{4} \times (d_2)^2 - d_2 \times t \right] \sigma_t$$



From this equation, the diameter of spigot or inside diameter of socket ( $d_2$ ) may be determined

### 3. Failure of the rod or cotter in crushing

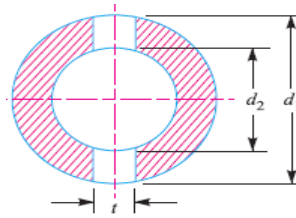
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$P = d_2 \times t \times \sigma_c$  this equation, the induced crushing stress may be checked

**4. Failure of the socket in tension across the slot**

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



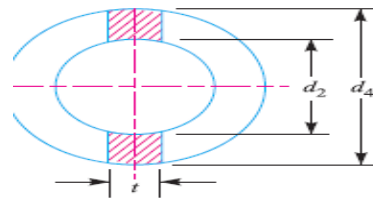
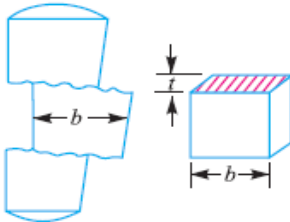
From this equation, outside diameter of socket ( $d_1$ ) may be determined

**5. Failure of cotter in shear**

Considering the failure of cotter in shear as shown in Fig. Since the cotter is in double shear,

$$P = 2 b \times t \times \tau$$

From this equation, width of cotter ( $b$ ) is determined.



**6. Failure of the socket collar in crushing**

Considering the failure of socket collar in crushing.

$$P = (d_4 - d_2) t \times \sigma_c$$

From this equation, the diameter of socket collar ( $d_4$ ) may be obtained.

**7. Failure of socket end in shearing**

Since the socket end is in double shear,

$$P = 2 (d_4 - d_2) c \times \tau$$

From this equation, the thickness of socket collar ( $c$ ) may be obtained

**8. Failure of rod end in shear**

Since the rod end is in double shear,  $P = 2 a \times d_2 \times \tau$

From this equation, the distance from the end of the slot to the end of the rod ( $a$ ) may be obtained.

**9. Failure of spigot collar in crushing**

Considering the failure of the spigot collar in crushing,

$$P = \frac{\pi}{4} [(d_3)^2 - (d_2)^2] \sigma_c$$

From this equation, the diameter of the spigot collar ( $d_3$ ) may be obtained.

**10. Failure of the spigot collar in shearing**

Considering the failure of the spigot collar in shearing

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$$P = \pi d_2 \times t_1 \times \tau$$

From this equation, the thickness of spigot collar (t1) may be obtained.

**11.** The length of cotter (l) is taken as 4 d.

- b) Compare the weight, strength and stiffness of a hollow shaft of the same external diameter as that of solid shaft. Inside diameter of hollow shaft is 0.75 times the external diameter. Both shafts have the same material & length.**

Given :

Outside diameter of hollow shaft (do) = Diameter of solid shaft (d)

For same material: Density of solid = density of hollow shaft

$$L_S = L_H, \quad d_i = \text{inside diameter of hollow shaft} = 0.75 d_o, \quad k = \frac{d_i}{d_o} = 0.75 \quad \dots 2 \text{ Marks}$$

**I) Comparison of weight:**

We know that weight of a hollow shaft

$$W_H = \text{Cross sectional area} \times \text{Length} \times \text{Density}$$

$$= \pi/4 [(d_o)^2 - (d_i)^2] \times \text{Length} \times \text{Density} \dots \dots \dots \text{I}$$

and Weight of the solid shaft

$$W_S = \text{Cross sectional area} \times \text{Length} \times \text{Density}$$

$$= \pi/4 (d)^2 \times \text{Length} \times \text{Density} \dots \dots \dots \text{II}$$

Since both the shafts have the same material and length, therefore by dividing equation (i) by equation (ii), we get

$$\frac{W_H}{W_S} = \frac{[(d_o)^2 - (d_i)^2]}{(d)^2}$$

$$\text{As } K = d_i/d_o$$

$$\frac{W_H}{W_S} = 1 - K^2 = 1 - (0.75)^2 = 0.44$$

$$W_H = 0.44 W_S \quad \dots \dots \dots \text{Ans} \dots \dots \dots 2 \text{ Marks}$$

**II) Comparison of Strength:**

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

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Strength of Solid shaft  $T_S = \frac{\pi}{16} \tau d o^3$

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

$$=(1 - k^4) = (1 - 0.75^4)$$

$$T_H = 0.68 T_S \dots\dots\dots \text{Ans} \dots\dots\dots \mathbf{2 \text{ Marks}}$$

**II) Comparison of Stiffness:**

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

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

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

$$=(1 - k^4) = (1 - 0.75^4)$$

$$S_H = 0.68 S_S \dots\dots\dots \text{Ans} \dots\dots\dots \mathbf{2 \text{ Marks}}$$

**c) Identify the material and its composition**

i) **a) X 10 Cr18Ni9Mo4Si2 :**

Material: High alloy steel ----- (1mark)

Composition: Avg. 0.1% Carbon, 18% chromium, 9% nickel, 4% Molebdenum, 2% Silicon. - --- **(1 mark)**

**b) XT 72 W 18 Cr 4 V 1 :**

Material: High speed tool steel ----- (1 mark)

Composition: average carbon content 0.72 %, tungsten 18 %, chromium 4 % and vanadium 1 %.--  
**(1mark)**

**NOTE: ( In Q paper X772 W 18 Cr 4 V 1 is asked instead of XT 72 W 18 Cr 4 V 1 . Marks can be awarded if attempted. )**

**ii) Explain with neat diagram about “ failure of knuckle pin in bending”**

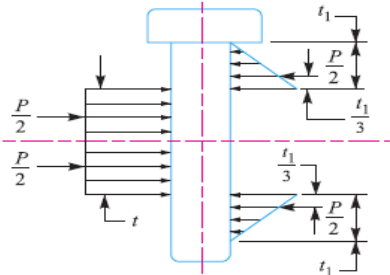
Ans: Actual practice, the knuckle pin is loose in forks in order to permit angular movement of one with respect to the other, therefore the pin is subjected to bending in addition to shearing. By making the diameter of knuckle

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pin equal to the diameter of the rod (*i.e.*,  $d_1 = d$ ), a margin of strength is provided to allow for the bending of the pin.



.....sketch 1 & 1/2 Marks

In case, the stress due to bending is taken into account, it is assumed that the load on the pin is uniformly distributed along the middle portion (*i.e.* the eye end) and varies uniformly over the forks as shown in Fig. Thus in the forks, a load  $P/2$  acts through a distance of  $t/3$  from the inner edge and the bending moment will be maximum at the center of the pin. The value of maximum bending moment is given by

$$\begin{aligned} M &= \frac{P}{2} \left( \frac{t_1}{3} + \frac{t}{2} \right) - \frac{P}{2} \times \frac{t}{4} \\ &= \frac{P}{2} \left( \frac{t_1}{3} + \frac{t}{2} - \frac{t}{4} \right) \\ &= \frac{P}{2} \left( \frac{t_1}{3} + \frac{t}{4} \right) \\ Z &= \frac{\pi}{32} (d_1)^3 \end{aligned}$$

$\therefore$  Maximum bending (tensile) stress,

$$\sigma_t = \frac{M}{Z} = \frac{\frac{P}{2} \left( \frac{t_1}{3} + \frac{t}{4} \right)}{\frac{\pi}{32} (d_1)^3}$$

From this expression, the value of  $d_1$  may be obtained. ....**Explanation** .....2& 1/2 Marks

**Q.3. Attempt any FOUR**

( 4 x 4marks)

**a) Write down the names of any four theories of elastic failure**

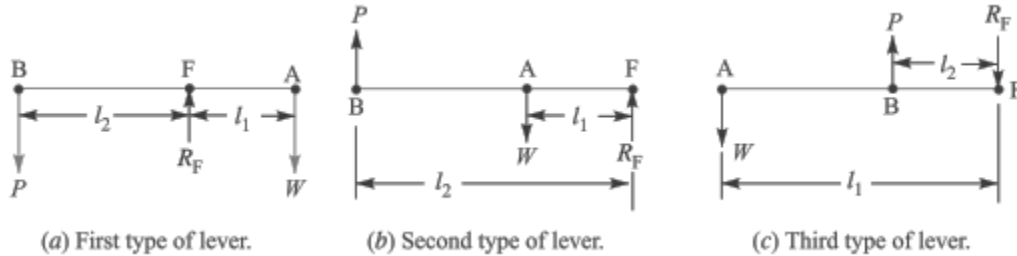
- 1) Maximum principal stress theory (Rankine's theory)
- 2) Maximum shear stress theory (Tresca & Guest theory)
- 3) Maximum strain energy theory (Haigh's theory)
- 4) Distortion energy theory (Von Mises & Hency theory)
- 5) Maximum principal strain theory (Saint venant's theory)

.....(any 4) 1 mark each

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b) State the three different ways of applications of levers in engineering practice

Ans:



**First type** of levers, the fulcrum is in between the load and effort. In this case, the effort arm is greater than load arm, therefore mechanical advantage obtained is more than one. Such type of levers are commonly found in bell cranked levers used in railway signalling arrangement, rocker arm in internal combustion engines, handle of a hand pump, hand wheel of a punching press, beam of a balance, foot lever etc.

**Second type** of levers, the load is in between the fulcrum and effort. In this case, the effort arm is more than load arm, therefore the mechanical advantage is more than one. The application of such type of levers is found in levers of loaded safety valves.

**Third type** of levers, the effort is in between the fulcrum and load. Since the effort arm, in this case, is less than the load arm, therefore the mechanical advantage is less than one. The use of such type of levers is not recommended in engineering practice. However a pair of tongs, the treadle of a sewing machine etc. are examples of this type of lever.

.....4 marks

c) State the classification of shaft couplings

Ans: Shaft couplings are divided into two main groups as follows:

1. **Rigid coupling.** It is used to connect two shafts which are perfectly aligned.

Rigid coupling further **classified** in

(a) Sleeve or muff coupling.

(b) Clamp or split-muff or compression coupling, and

(c) Flange coupling.

.....2 Marks.

2. **Flexible coupling.** It is used to connect two shafts having both lateral and angular

misalignment.

(a) Bushed pin type coupling,

(b) Universal coupling, and

(c) Oldham coupling.

.....2

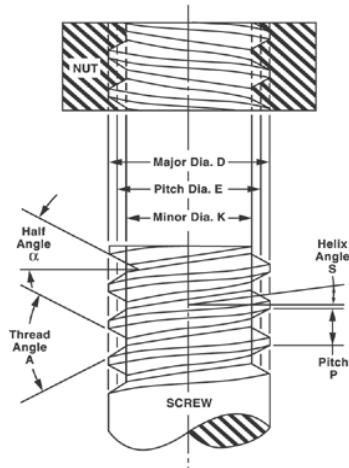
Marks.

d) Explain with neat sketch the following terms used in screw threads:

i) Major diameter ii) Minor diameter iii) Pitch iv) Crest

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**Major Diameter:** The largest diameter of the thread of the screw or nut. The term "major diameter" replaces the term "outside diameter" as applied to the thread of a screw and also the term "full diameter" as applied to the thread of a nut.

**Minor Diameter :** The smallest diameter of the thread of the screw or nut. The term "minor diameter" replaces the term "core diameter" as applied to the thread of a screw and also the term "inside diameter" as applied to the thread of a nut.

**Pitch:** The distance from a point on a screw thread to a corresponding point on the next thread measured parallel to the axis.

**Crest:** The surface of the thread corresponding to the major diameter of the screw and the minor diameter of the nut.

.....(Each term with fig. 1 mark)..... **4 Marks**

e) **Design a rectangular key to be used for a shaft of diameter 50 mm. Both are having the same material.**

**Take  $f_s=42 \text{ Mpa}$  &  $f_c=70 \text{ Mpa}$**

Given: Dia. of shaft  $d = 50 \text{ mm}$ ,  $f_s=42 \text{ Mpa} = 42 \text{ N.mm}^2$  &  $f_c=70 \text{ Mpa} = 70 \text{ N.mm}^2$

$$\text{Torque transmitted } T = \frac{\pi}{16} f_s d^3$$

$$T = \frac{\pi}{16} \times 42 \times 50^3$$

$$T = 1030835.089 \text{ N-mm}$$

.....

**1 mark**

As rectangular key is given ,

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**Model Answer**

Thickness of key  $t = d/6 = 50/6 = 8.33 \text{ mm}$

Width of key :  $w = d/4 = 50/4 = 12.5 \text{ mm}$

..... 1 mark

Consider, Key is under shear failure  $T = l \cdot w \cdot f_s \cdot d/2$

$$1.03 \times 10^6 = l \times (12.5) \times (42) \times 50/2$$

$$\text{Length of key } l = 78.47 \text{ mm}$$

... 1 mark

Consider, Key is under Crushing failure  $T = l \cdot t/2 \cdot f_c \cdot d/2$

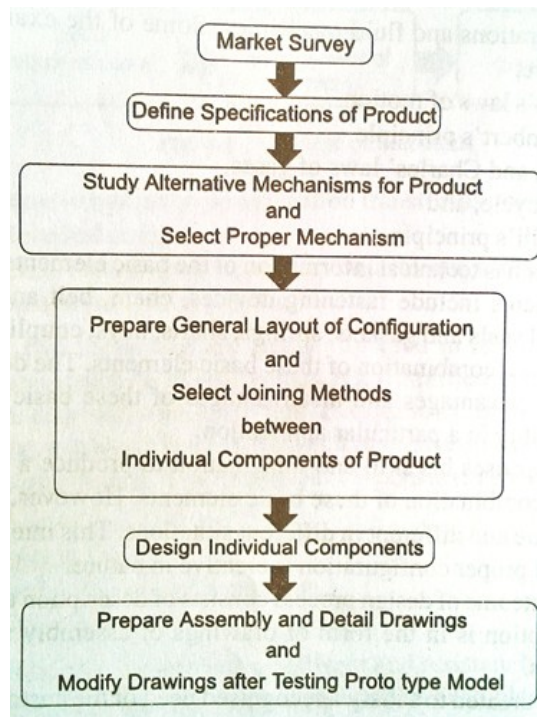
$$1.03 \times 10^6 = l \times (12.5) \times (8.33/2) \times (70) \times 50/2$$

$$\text{Length of key } l = 141.26 \text{ mm}$$

Taking larger value of two values of length of key  $= l = 141.26 \text{ mm} \approx 142 \text{ mm}$  ..... 1 mark

**Q.4 (A) Attempt any THREE..... 04 Marks  $\times$  03 = 12 Marks**

a) **Steps involved in General Design Procedure are as given below.**



**Note:** It is not necessary to provide the above design procedure in a flow chart. Even if student enlistis the steps in a proper sequence, shall be given due credit. ---4 Marks(Stepwise)

b) **Four areas of application of springs.**

**(Any four applications of the following.1 mark each)**

i) To absorb or control energy in automobiles suspension springs, vibration

dampers, railway buffers, for

- ii) To apply forces in brakes, clutches, valves of IC engines
- iii) To store the energy in watches and toys.
- iv) To measure forces in spring balances, gauges
- v) To provide clamping force in toolings like jigs and fixtures etc.
- vi) To control motion by maintaining contact between two elements.

**c) Four causes of gear tooth failure.**

**(Any four causes of the following with a statement providing the correct information. 1 mark each.)**

**1. Bending failure.**

Every gear tooth acts as a cantilever. If the total repetitive dynamic load acting on the gear tooth is greater than the beam strength of the gear tooth, then the gear tooth will fail in bending

**2. Pitting.**

It is the surface fatigue failure which occurs due to many repetition of Hertz contact stresses. The failure occurs when the surface contact stresses are higher than the endurance limit of the material.

**3. Scoring.**

The excessive heat is generated when there is an excessive surface pressure, high speed or supply of lubricant fails. It is a stick-slip phenomenon in which alternate shearing and welding takes place rapidly at high spots.

**4. Abrasive wear.**

The foreign particles in the lubricants such as dirt, dust or burr enter between the tooth and damage the form of tooth.

**5. Corrosive wear.**

The corrosion of the tooth surfaces is mainly caused due to the presence of corrosive elements such as additives present in the lubricating oils.

**d) Bolts of uniform strength.**

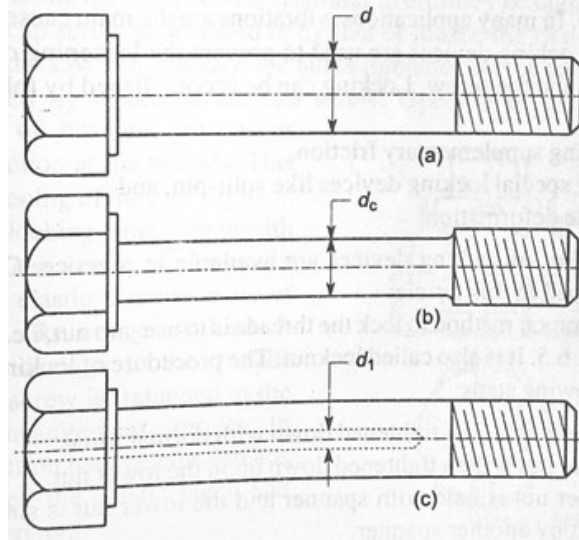
**(Explanation including two methods of making bolt of uniform strength accompanied with figure. 2 marks each)**

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

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

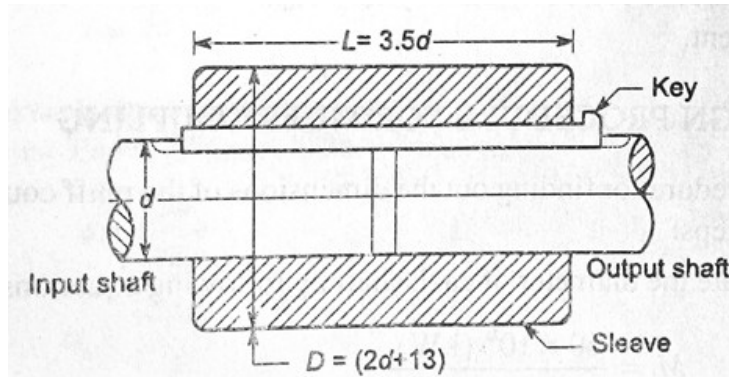
A second alternative method of obtaining the bolts of uniform strength is shown in **Fig. (c)**. An axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread.

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**Q.4 (B) Attempt any ONE..... 06 Marks × 01 = 06 Marks**

**a) Design Procedure of a sleeve or muff coupling is as given below.**



**Fig. Sleeve or muff coupling .....(1 mark)**

**i) Design for sleeve**

The sleeve is designed by considering it as a hollow shaft.

Let T = Torque to be transmitted by the coupling, and

$\tau_c$  = Permissible shear stress for the material of the sleeve which is cast iron.

We know that torque transmitted by a hollow section,

$$T = \frac{\pi}{16} \times \tau_c \left( \frac{D^4 - d^4}{D} \right) = \frac{\pi}{16} \times \tau_c \times D^3 (1 - k^4)$$

From this expression, the induced shear stress in the sleeve may be checked.....(2 marks)

**ii) Design for key**

The width and thickness of the coupling key is obtained from the proportions.

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The length of the coupling key is at least equal to the length of the sleeve (i.e. 3.5 d). The coupling key is usually made into two parts so that the length of the key in each shaft,

$$l = \frac{L}{2} = \left( \frac{3.5d}{2} \right) \dots\dots\dots(1 \text{ marks})$$

After fixing the length of key in each shaft, the induced shearing and crushing stresses may be checked. We know that torque transmitted,

$$T = l \times w \times \tau \times \frac{d}{2} \dots\dots\dots(1 \text{ marks})$$

$$T = l \times w \times \sigma_c \times \frac{d}{2} \dots\dots\dots(1 \text{ marks})$$

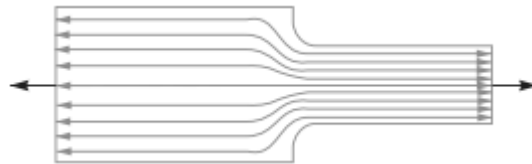
**Note:**

The depth of the keyway in each of the shafts to be connected should be exactly the same and the diameters should also be same.

**b) 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. It occurs for all kinds of stresses in the presence of fillets, notches, holes, keyways, splines, surface roughness or scratches etc. ....(1 mark)



**Fig. Stress Concentration** ....(1 mark)

**In static loading,** stress concentration in ductile materials is not so serious as in brittle materials, because in ductile materials local deformation or yielding takes place which reduces the concentration. In brittle materials, cracks may appear at these local concentrations of stress which will increase the stress over the rest of the section.

It is, therefore, necessary that in designing parts of brittle materials such as castings, care should be taken. In order to avoid failure due to stress concentration, fillets at the changes of section must be provided.

.....(2 marks)

**In cyclic loading,** stress concentration in ductile materials is always serious because the ductility of the material is not effective in relieving the concentration of stress caused by cracks, flaws, surface roughness, or any sharp discontinuity in the geometrical form of the member.

If the stress at any point in a member is above the endurance limit of the material, a crack may develop under the action of repeated load and the crack will lead to failure of the member.

.....(2 marks)

**Q.No.5 Attempt any TWO.....**

**08 Marks × 02= 06 Marks**

**(a)**

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**Solution.** Given :  $d_o = 25$  mm ;  $p = 5$  mm ;  $W = 10$  kN =  $10 \times 10^3$  N ;  $D_1 = 50$  mm or  $R_1 = 25$  mm ;  $D_2 = 20$  mm or  $R_2 = 10$  mm ;  $\mu = \tan \phi = 0.2$  ;  $\mu_1 = 0.15$  ;  $N = 12$  r.p.m. ;  $p_b = 5.8$  N/mm<sup>2</sup>

**1. Torque required to rotate the screw**

We know that mean diameter of the screw,

$$d = d_o - p / 2 = 25 - 5 / 2 = 22.5 \text{ mm}$$

Since the screw is a double start square threaded screw, therefore lead of the screw,

$$= 2 p = 2 \times 5 = 10 \text{ mm}$$

$$\therefore \tan \alpha = \frac{\text{Lead}}{\pi d} = \frac{10}{\pi \times 22.5} = 0.1414 \quad \dots\dots\dots 1 \text{ mark}$$

We know that tangential force required at the circumference of the screw,

$$P = W \tan (\alpha + \phi) = W \left[ \frac{\tan \alpha + \tan \phi}{1 - \tan \alpha \tan \phi} \right]$$

$$= 10 \times 10^3 \left[ \frac{0.1414 + 0.2}{1 - 0.1414 \times 0.2} \right] = 3513 \text{ N} \quad \dots\dots\dots 1 \text{ mark}$$

and mean radius of the screw collar,

$$R = \frac{R_1 + R_2}{2} = \frac{25 + 10}{2} = 17.5$$

$\therefore$  Total torque required to rotate the screw,

$$T = P \times \frac{d}{2} + \mu_1 W R = 3513 \times \frac{22.5}{2} + 0.15 \times 10 \times 10^3 \times 17.5 \text{ N-mm}$$

$$= 65\,771 \text{ N-mm} = 65.771 \text{ N-m} \quad \text{Ans.} \quad \dots\dots\dots 2 \text{ marks}$$

**2. Stress in the screw**

We know that the inner diameter or core diameter of the screw,

$$d_c = d_o - p = 25 - 5 = 20 \text{ mm}$$

$\therefore$  Corresponding cross-sectional area of the screw,

$$A_c = \frac{\pi}{4} (d_c)^2 = \frac{\pi}{4} (20)^2 = 314.2 \text{ mm}^2$$

We know that direct stress,

$$\sigma_c = \frac{W}{A_c} = \frac{10 \times 10^3}{314.2} = 31.83 \text{ N/mm}^2$$

and shear stress,  $\tau = \frac{16 T}{\pi (d_c)^3} = \frac{16 \times 65\,771}{\pi (20)^3} = 41.86 \text{ N/mm}^2 \quad \dots\dots\dots 1 \text{ mark}$

We know that maximum shear stress in the screw,

$$\tau_{max} = \frac{1}{2} \sqrt{(\sigma_c)^2 + 4\tau^2} = \frac{1}{2} \sqrt{(31.83)^2 + 4(41.86)^2}$$

$$= 44.8 \text{ N/mm}^2 = 44.8 \text{ MPa} \quad \text{Ans.} \quad \dots\dots\dots 1 \text{ mark}$$

**3. Number of threads of nut in engagement with screw**

Let  $n$  = Number of threads of nut in engagement with screw, and

$t$  = Thickness of threads =  $p / 2 = 5 / 2 = 2.5$  mm

$$5.8 = \frac{W}{\pi d \times t \times n} = \frac{10 \times 10^3}{\pi \times 22.5 \times 2.5 \times n} = \frac{56.6}{n}$$

$$\therefore n = 56.6 / 5.8 = 9.76 \text{ say } 10 \quad \text{Ans.} \quad \dots\dots\dots 2 \text{ marks}$$

(b)

**Solution.** Given : No. of springs = 10 ;  $W_1 = 75 \text{ kN} = 75\,000 \text{ N}$  ;  $W_2 = 15 \text{ kN} = 15\,000 \text{ N}$  ;  
 $h = 50 \text{ m} = 50\,000 \text{ mm}$  ;  $d = 50 \text{ mm}$  ;  $C = 6$  ;  $n = 20$  ;  $G = 80 \text{ kN/mm}^2 = 80 \times 10^3 \text{ N/mm}^2$

We know that net weight of the falling load,

$$P = W_1 - W_2 = 75\,000 - 15\,000 = 60\,000 \text{ N}$$

Let

$W$  = The equivalent static (or gradually applied) load on each spring which can produce the same effect as by the falling load  $P$ .

We know that compression produced in each spring,

$$\delta = \frac{8 W . C^3 . n}{G . d} = \frac{8 W \times 6^3 \times 20}{80 \times 10^3 \times 50} = 0.008\,64 W \text{ mm} \dots\dots\dots 2 \text{ marks}$$

Since the work done by the falling load is equal to the energy stored in the helical springs which are 10 in number, therefore,

$$P (h + \delta) = \frac{1}{2} W \times \delta \times 10$$

$$60\,000 (50\,000 + 0.008\,64 W) = \frac{1}{2} W \times 0.008\,64 W \times 10$$

$$3 \times 10^9 + 518.4 W = 0.0432 W^2$$

$$\text{or } W^2 - 12\,000 W - 69.4 \times 10^9 = 0$$

$$\therefore W = \frac{12\,000 \pm \sqrt{(12\,000)^2 + 4 \times 1 \times 69.4 \times 10^9}}{2} = \frac{12\,000 \pm 527\,000}{2} \dots\dots\dots 3 \text{ marks}$$

$$= 269\,500 \text{ N} \dots\dots\dots \text{... (Taking +ve sign)}$$

We know that Wahl's stress factor,

$$K = \frac{4C - 1}{4C - 4} + \frac{0.615}{C} = \frac{4 \times 6 - 1}{4 \times 6 - 4} + \frac{0.615}{6} = 1.25$$

and maximum stress induced in each spring,

$$\tau = K \times \frac{8 W . C}{\pi d^2} = 1.25 \times \frac{8 \times 269\,500 \times 6}{\pi (50)^2} = 2058.6 \text{ N/mm}^2 \dots\dots\dots 2 \text{ marks}$$

$$= 2058.6 \text{ MPa Ans.}$$

As the determined maximum shear stress i.e.2058 MPa is less than the permissible shear stress i.e.2800 MPa,design of the spring is safe.....1 mark

(c) (i) Define the following terms related to bearings ( 1 mark each)

**(a) Bearing characteristics Number**

The factor  $ZN / p$  is termed as bearing characteristic number and is a dimensionless number.

$Z$  = Absolute viscosity of the lubricant, in kg / m-s,

$N$  = Speed of the journal in r.p.m.,

$p$  = Bearing pressure on the projected bearing area in N/mm,

= Load on the journal /  $l \times d$

$d$ =diameter of the journal

$l$ =length of the bearing

$c$ =diametral clearance

The factor  $ZN/p$  helps to predict the performance of a bearing.

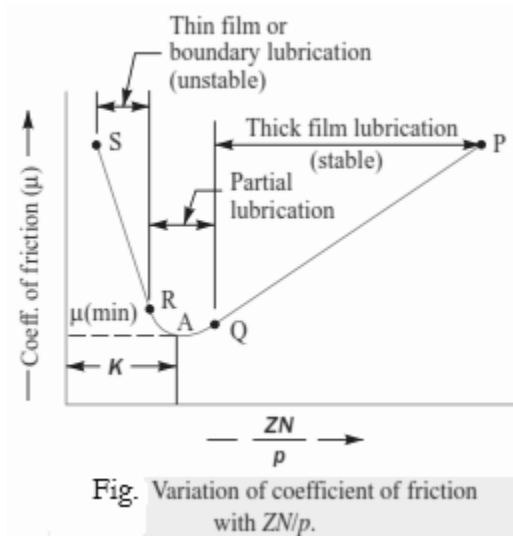
**(b) Bearing modulus.**

From Fig., we see that the minimum amount of friction occurs at value of  $ZN / p$  is known as bearing modulus

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operated at this value of bearing modulus, because a slight decrease in speed or slight increase in pressure will break the oil film and make the journal to operate with metal to metal contact.



**(c) Critical Pressure.**

The pressure at which the oil film breaks down so that metal to metal contact begins, is known as **critical pressure** or the **minimum operating pressure** of the bearing. It may be obtained by the following empirical relation, *i.e.*

Critical pressure or minimum operating pressure,

$$p = \frac{ZN}{4.75 \times 10^6} \left( \frac{d}{c} \right)^2 \left( \frac{l}{d+l} \right) \text{ N/mm}^2 \quad \dots (\text{when } Z \text{ is in kg / m-s})$$

**(d) Sommerfeld number**

The Sommerfeld number is also a dimensionless parameter used extensively in the design of journal bearings. Mathematically,

$$\text{Sommerfeld number} = \frac{ZN}{p} \left( \frac{d}{c} \right)^2$$

For design purposes, its value is taken as follows :

$$\frac{ZN}{p} \left( \frac{d}{c} \right)^2 = 14.3 \times 10^6 \quad \dots (\text{when } Z \text{ is in kg / m-s and } p \text{ is in N / mm}^2)$$

**(ii) Meaning of “Overhauling and self locking”**

The effort required at the circumference of the screw to lower the load is

$$P = W \tan (\phi - \alpha)$$

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and the torque required to lower the load.

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

In the above expression,  $\Phi$  is friction angle and  $\alpha$  is helix angle.....2 marks

If  $\Phi < \alpha$ , then torque required to lower the load will be negative. i.e., the load will start moving downward without the application of any torque. Such a condition is known as **over hauling** of screws.  
.....1 mark

If  $\Phi > \alpha$ , 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 .....1 mark

**Q.6 Attempt any FOUR..... 04 Marks × 04= 16 Marks.**

(a) Let  $t$  = Throat thickness (BD), mm

$s$  = Leg or size of weld, mm

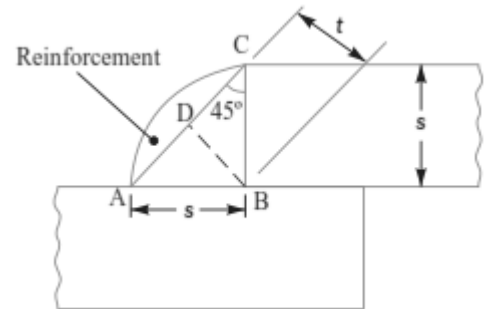
= Thickness of plate, and

$l$  = Length of weld, mm

Throat thickness,  $t = s \times \sin 45^\circ = 0.707 s$

∴ Minimum area of the weld or throat area,

$$\begin{aligned} A &= \text{Throat thickness} \times \text{Length of weld} \\ &= t \times l = 0.707 s \times l \text{ mm}^2 \end{aligned}$$



If  $\sigma_t$  is the allowable tensile stress for the weld metal in MPa,  
then the tensile strength of the joint for single fillet weld,

$$P = \text{Throat area} \times \text{Allowable tensile stress} = 0.707 s \times l \times \sigma_t \dots\dots\dots 1 \text{ mark}$$

If  $\tau$  is the allowable shear stress for the weld metal in MPa,  
then the shear strength of the joint for single parallel fillet weld,

$$\begin{aligned} P &= \text{Throat area} \times \text{Allowable shear stress} \\ &= 0.707 s \times l \times \tau \end{aligned}$$

and shear strength of the joint for double parallel fillet weld,

$$P = 2 \times 0.707 \times s \times l \times \tau = 1.414 s \times l \times \tau \dots\dots\dots 1 \text{ mark}$$

Combination of single transverse and double parallel fillet welds as shown in Fig. ,  
then the strength of the joint is given by the  
sum of strengths of single transverse and  
double parallel fillet welds.

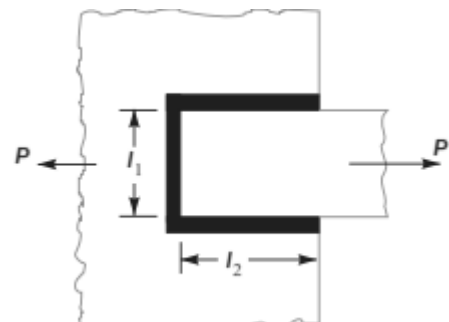


Fig. Single transverse and Double Parallel Fillet weld.....1 mark

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Mathematically,

$P = 0.707 s \times l_1 \times \sigma_t + 1.414 s \times l_2 \times \tau$ .....**Ans.**.....**1 mark**  
where  $l_1$  is normally the width of the plate.

**(b)**

**Solution.** Given :  $n = 12$  ;  $n_1 = 5$  ;  $n_2 = 7$

We know that the deflection of the spring,

$$\delta = \frac{8 W . D^3 . n}{G . d^4} \quad \text{or} \quad \frac{W}{\delta} = \frac{G . d^4}{8 D^3 . n}$$

Since  $G$ ,  $D$  and  $d$  are constant, therefore substituting

$$\frac{G . d^4}{8 D^3} = X, \text{ a constant, we have } \frac{W}{\delta} = k = \frac{X}{n}$$

or  $X = k . n = 12 k$  .....**2 marks**

The spring is cut into two springs with  $n_1 = 5$  and  $n_2 = 7$ .

Let  $k_1$  = Stiffness of spring having 5 turns, and

$k_2$  = Stiffness of spring having 7 turns. ....**1 mark**

$$\therefore k_1 = \frac{X}{n_1} = \frac{12 k}{5} = 2.4 k \quad \text{Ans.}$$

and  $k_2 = \frac{X}{n_2} = \frac{12 k}{7} = 1.7 k \quad \text{Ans.}$  .....**1 mark**

**(c)**

**Solution.** Given :  $W = 30 \text{ kN}$  ;  $\sigma_t = 60 \text{ MPa} = 60 \text{ N/mm}^2$  ;  $L_1 = 80 \text{ mm}$  ;  $L_2 = 250 \text{ mm}$  ;  
 $L = 500 \text{ mm}$

We know that the direct tensile load carried by each bolt,

$$W_{t1} = \frac{W}{n} = \frac{30}{4} = 7.5 \text{ kN}$$

and load in a bolt per unit distance,

$$w = \frac{W . L}{2 [(L_1)^2 + (L_2)^2]} = \frac{30 \times 500}{2 [(80)^2 + (250)^2]} = 0.109 \text{ kN/mm} \quad \text{.....1 mark}$$

Since the heavily loaded bolt is at a distance of  $L_2$  mm from the tilting edge, therefore load on the heavily loaded bolt, .....**1 mark**

$$W_{t2} = w . L_2 = 0.109 \times 250 = 27.25 \text{ kN}$$

$\therefore$  Maximum tensile load on the heavily loaded bolt, .....**1 mark**

$$W_t = W_{t1} + W_{t2} = 7.5 + 27.25 = 34.75 \text{ kN} = 34750 \text{ N}$$

Let  $d_c$  = Core diameter of the bolts.

We know that the maximum tensile load on the bolt ( $W_t$ ),

$$34750 = \frac{\pi}{4} (d_c)^2 \sigma_t = \frac{\pi}{4} (d_c)^2 60 = 47 (d_c)^2$$

$$\therefore (d_c)^2 = 34750 / 47 = 740$$

$$\text{or } d_c = 27.2 \text{ mm}$$

From (coarse series), we find that the standard core diameter of the bolt is 28.706 mm and the corresponding size of the bolt is M 33..... **Ans.** .....**1 mark**

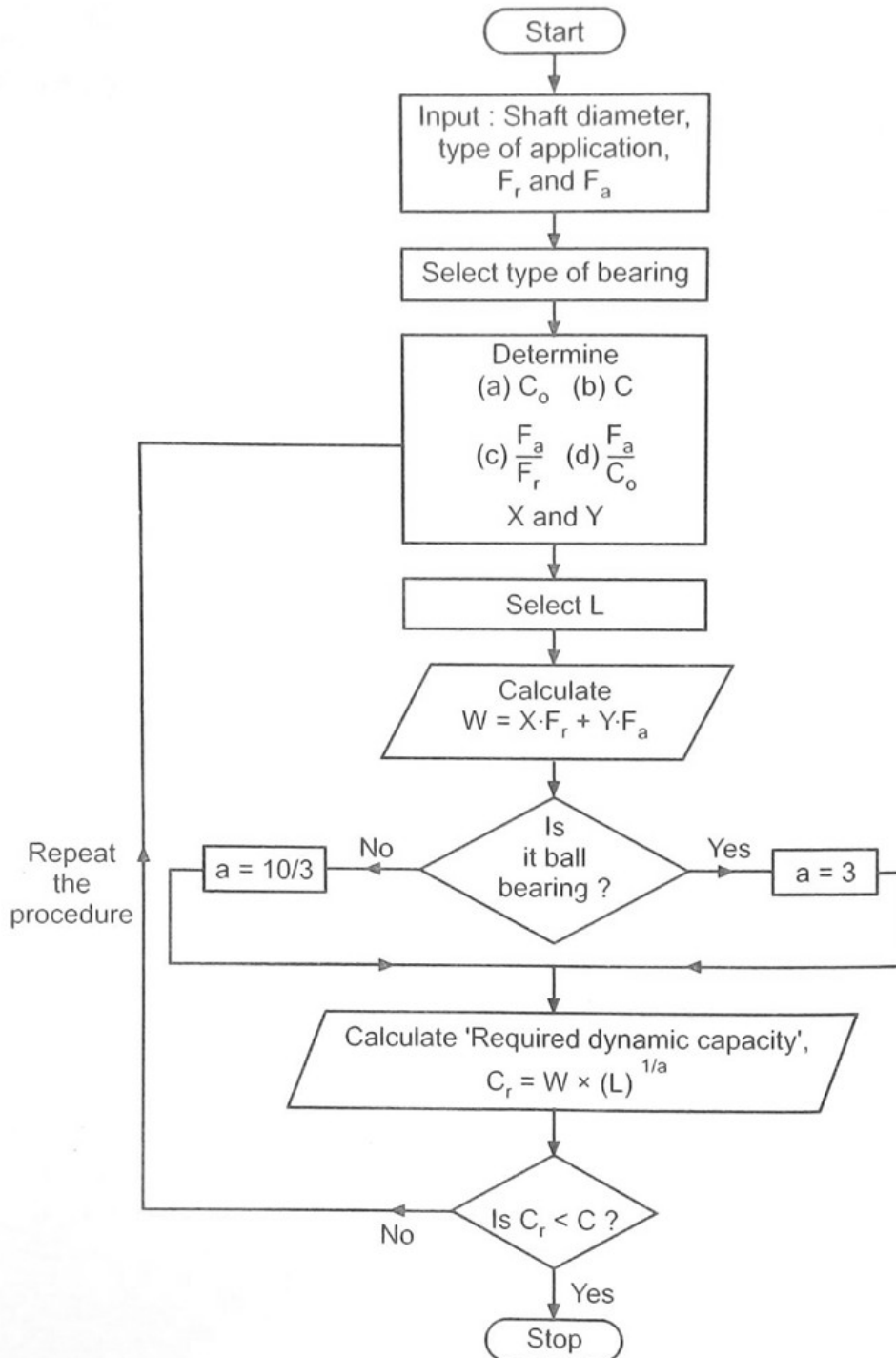
**(d)**

**Properties of bearing materials (Any four. 1 mark each)**

- 
- 1. Compressive strength.** The maximum bearing pressure is considerably greater than the average pressure obtained by dividing the load to the projected area. Therefore the bearing material should have high compressive strength to withstand this maximum pressure so as to prevent extrusion or other permanent deformation of the bearing.
  - 2. Fatigue strength.** The bearing material should have sufficient fatigue strength so that it can withstand repeated loads without developing surface fatigue cracks. It is of major importance in aircraft and automotive engines.
  - 3. Conformability.** It is the ability of the bearing material to accommodate shaft deflections and bearing inaccuracies by plastic deformation (or creep) without excessive wear and heating.
  - 4. Embed ability.** It is the ability of bearing material to accommodate (or embed) small particles of dust, grit etc., without scoring the material of the journal.
  - 5. Bendability.** Many high capacity bearings are made by bonding one or more thin layers of a bearing material to a high strength steel shell. Thus, the strength of the bond i.e. bondability is an important consideration in selecting bearing material.
  - 6. Corrosion resistance.** The bearing material should not corrode away under the action of lubricating oil. This property is of particular importance in internal combustion engines where the same oil is used to lubricate the cylinder walls and bearings. In the cylinder, the lubricating oil comes into contact with hot cylinder walls and may oxidise and collect carbon deposits from the walls.
  - 7. Thermal conductivity.** The bearing material should be of high thermal conductivity so as to permit the rapid removal of the heat generated by friction.
  - 8. Thermal expansion.** The bearing material should be of low coefficient of thermal expansion, so that when the bearing operates over a wide range of temperature, there is no undue change in the clearance.

(e) **Procedure for selection of ball bearing from manufacturer's catalogue.**

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