MA TRA STATE BOARD OF TECHNICAL EDUCATION
(Autonomous)
(ISO/IEC - 27001 - 2005 Certified)
SUMMER – 16 EXAMINATION

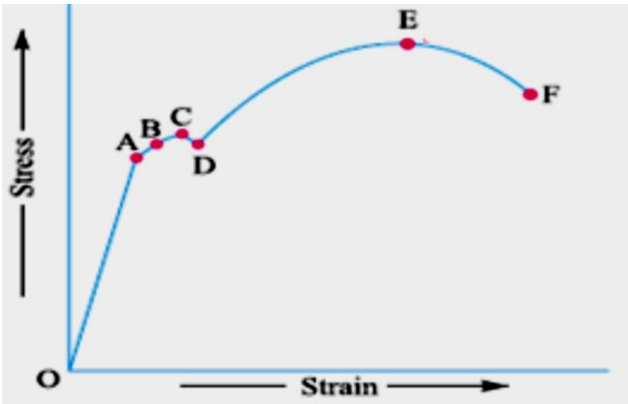
Subject Code :**17610** **Model Answer**

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 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 equivalent concept.

Q.1. (a)

i. Stress-Strain diagram for ductile material stating salient points
(1 Mark for diagram, 3 Marks for description)



Proportional limit (A): The stress is proportional to strain. Beyond point *A*, the curve slightly deviates from the straight line. It is thus obvious, that Hooke's law holds good up to point *A* and it *is known as* ***Proportional limit****.*

Elastic limit (B): If the load is increase between point A and B, the body will regain its original shape when load is removed; it means body possesses elasticity up to point B, known as **Elastic Limit.**

Upper yield point (C): If the material is stressed beyond point *B*, the plastic stage will reach and the material will start yielding known as **Upper Yield Point.**

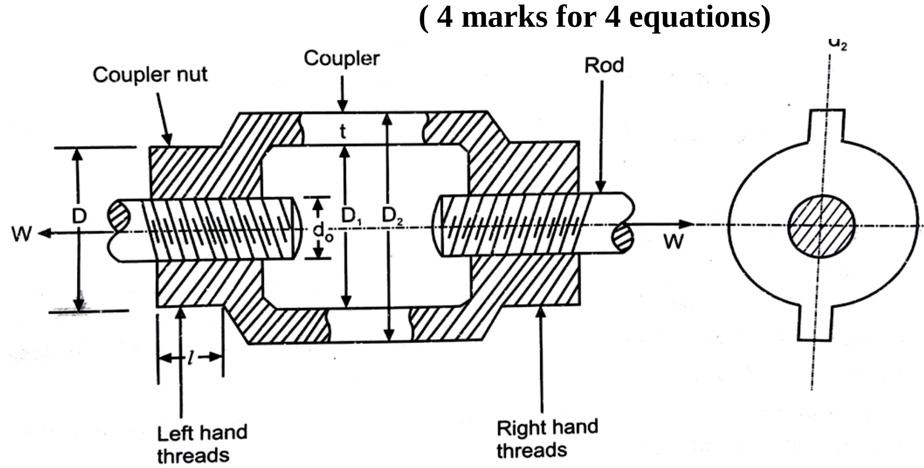
Lower yield point (D): Further addition of small load drops the stress-strain diagram to point D, as soon as the yielding start, this point 'D' is known as **Lower yield point.**

Ultimate stress point (E): After the end of yielding, if the load is increase beyond point 'D', there is increase in stresses up to point E and thus maximum value of stresses at point 'E' is called as **Ultimate Stress point.**

Breaking Stress point (F): After the specimen has reached the ultimate stress, a neck is formed, which decreases the cross-sectional area of the specimen. The stress corresponding to point *F* is known as **Breaking stress.**

Q.1. (a) ii. Write any four equation s in the design of turn buckle with relevant sketches

(4 marks for 4 equations)



Where,

W=design load =1.3 or1.4 times load carried by rods

τ=permissible shear stress in N/mm2 σ_t=permissible tensile stress in N/mm2

σ_c=permissible crushing stress in N/mm2 d_c=core diameter of rod in mm,

d_o=nominal diameter of rod in mm p=pitch of the thread in mm ,n=no threads,

l=length of coupler nut in mm D=diameter of coupler nut,

D_1=inside diameter of coupler D_2=outside diameter of coupler,

t=thickness of coupler

Step 1: Design of rod (d_c)

Considering tensile failure,

$$\sigma_t = \frac{W}{\frac{\pi}{4} \times d_c^2}$$ ——— 1 ma

after calculating dc, d_o & pitch can be determine from std. table.

Step 2: Design of coupler nut (l)

Considering shear failure,

$$\tau = \frac{W}{\pi d_c l}$$

Step 3: Checking of crushing stress induced in thread (σ_c)

$$\sigma_c = \frac{W}{\frac{\pi}{4}[d_o^2 - d_c^2] \times n \times l}$$

Wher, $n = \frac{1}{\text{Pitch}}$

Step 4: Design of coupler nut (D)

Considering tensile failure,

$$\sigma_t = \frac{W}{\frac{\pi}{4} \times [D^2 - d_o^2]}$$

Step 5: Design of outer dia. of coupler [D_2]

$$\sigma_t = \frac{W}{\frac{\pi}{4}[D_2^2 - D_1^2]}$$

Where, $D_1 = d_o + (6 \text{ to } 8\ mm)$

$t = 0.75\ d_o$

Q.1. (a)

iii. State any four factors to be consider while selecting the coupling.

(1 mark each for any four factors)

Following factors should be consider while selecting coupling

1) Cyclic operation
2) Duration or life
3) Misalignment of shafts
4) Required torque and desired speed

5) Direction of rotation
6) Protection against overload
7) Operating conditions

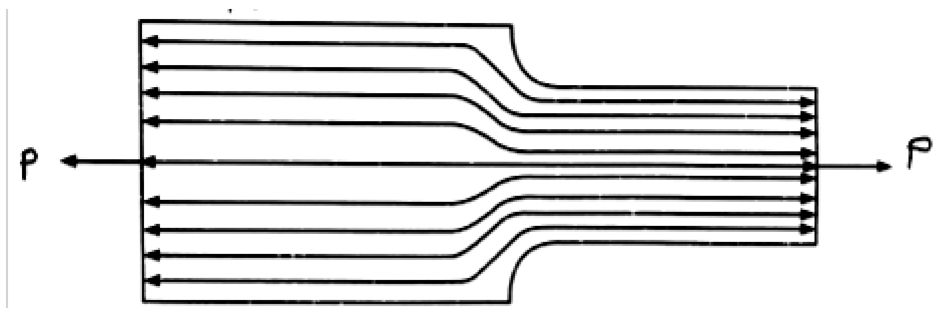
iv. Square threads are preferred over V-thread for power transmission because of following points. (1 mark each for any four factors)

1) Square thread has the greatest efficiency as its profile angle is zero.
2) It produces minimum bursting pressure on the nut.
3) It has more transmission efficiency due to less friction.
4) It transmits power without any side thrust in either direction.
5) It is more smooth and noiseless operation.

Q.1. (b) What is stresses concentration? State the remedial measures to control the effects of stress concentration with neat sketches.(2m for stresses conc., 4 m for remedial measure)

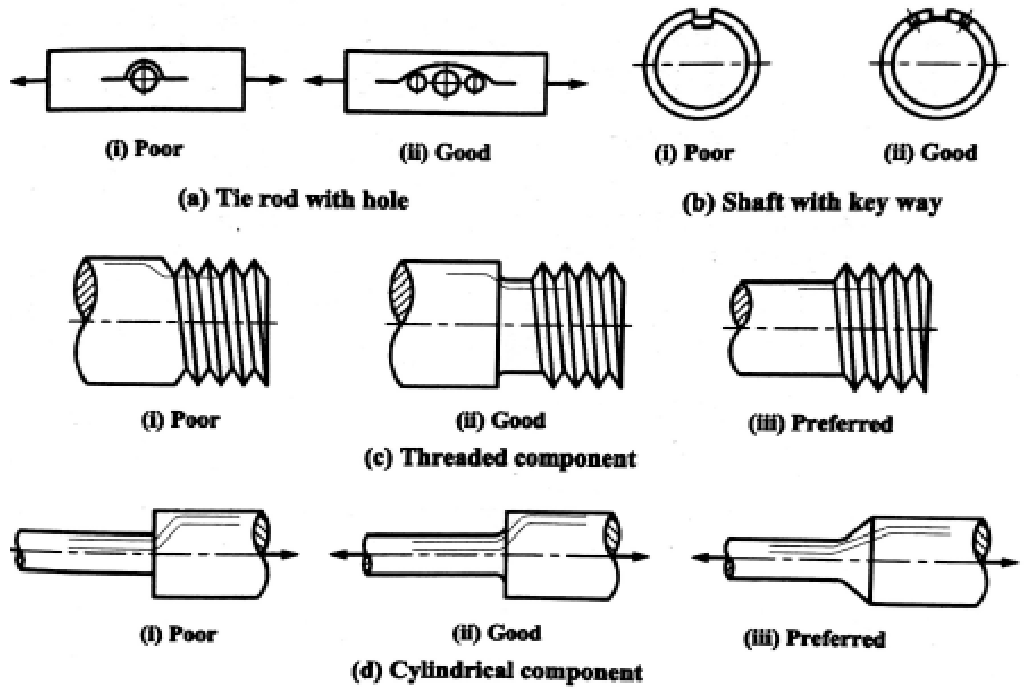
i. Stress Concentration:

Whenever a machine component changes the shape of its 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 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.



The presence of stresses concentration cannot be totally eliminated but it can be reduced, so following are the remedial measures to control the effects of stress concentration.

1. Provide additional notches and holes in tension members as shown in fig (a)
   a)Use of multiple notches.
   b)Drilling additional holes as shown in fig(b)
2. Fillet radius, undercutting and notch for member in bending.
3. Reduction of stress concentration in threaded members as shown in fig(c)
4. Provide taper cross-section to the sharp corner of member as shown in fig(d)



Q.1. (b)(ii)

Given Data:

N= 125 RPM

$$P = 440\ kW = 440 \times 10^3 Watt$$

τ = 55 N/mm²

Ɵ = 1°= $\frac{\pi}{180}$ Rad

L= 16d

G = 0.80 X 10^5 N / mm²

To find: Diameter of Shaft

Solution: i. To find T

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

$$440 \times 10^3 = \frac{2\pi \times 125 \times T}{60}$$

$$\boldsymbol{T = 33.613 \times 10^3 N.m}$$

$$\boldsymbol{T = 33.613 \times 10^6 N.mm}$$ ---------------------- **(1 Mark)**

ii. Diameter of shaft on the basis of strength

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

$$33.613 \times 10^6 = \frac{\pi}{16} \times d^3 \times 55$$

$$\boldsymbol{d = 146mm}$$ --------------------- **(2 Mark)**

iii. Diameter of shaft on the basis of rigidity

$\frac{T}{J} = \frac{G\theta}{L}$--------------------- **(1 Mark)**

$$\frac{33.61 \times 10^6}{J} = \frac{(\,0.80 \times 10^5) \times \frac{\pi}{180}}{16 \times d}$$

$$385.14 \times 10^3 d = J$$

$$385.14 \times 10^3 d = \frac{\pi}{32} \times d^4$$

$$\boldsymbol{d = 157.71mm}$$ --------------------- **(2 Mark)**

Q.2. (a)

i. **State any four factors that govern 'Factors of safety'(1 mark each for any four)**

a. Reliability of applied load.
b. The extent of simplifying assumptions.
c. The certainty as to exact mode of failure.
d. Reliability of properties of material and change in these properties during service.
e. Extent of stress concentration.
f. The reliability of test results to actual machine parts.
g. The extent of initial stresses set up during manufacturing.
h. The extent of loss of life, if failure occurs.

Q.2. (a)

ii. Why taper is provided on cotter? State recommended value of taper.

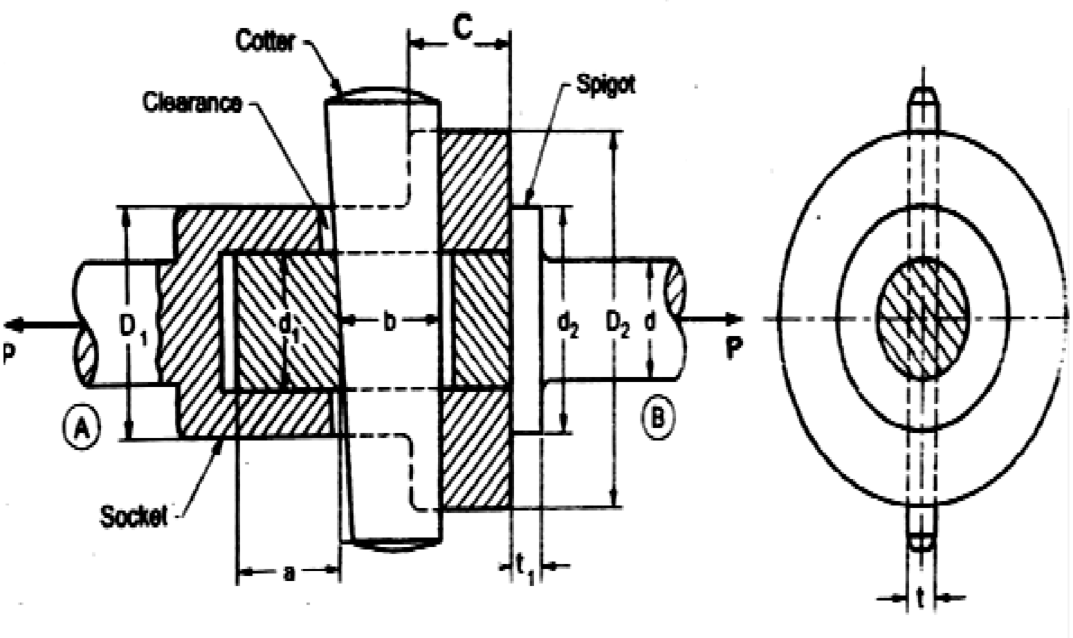
Taper is provided due to following two reasons.

a. When cotter is driven through the slots, it fit, fight due to wedge action. This ensures tightness of joint in operation and present loosening of the parts.
b. Due to taper, it is easy to remove the cotter and dismantle the joint.

The normal value of taper varies from 1 in 48 to 01 in 24 and it may increase to 1 in 8.**(2 Marks for reason, 2 Marks for taper value)**

Q.2. (b) Draw neat sketch showing the details of cotter joint.State strength equation for each component with suitable failure cross-sectional area.

(1 Mark for diagram, 7 Marks for design procedure with failure diagram)



• Cotter Joint •

It consist of 3 elements

i. Socket
ii. Spigot
iii. Cotter

Where,

d= End diameter of rod d_1= Diameter of spigot/ID of socket
d_2= Diameter of spigot collar D_1= Outer diameter of socket
D_2= Diameter of socket collar C=Thickness of socket collar
t_1= Thickness of spigot collar t= thickness of cotter
b= Mean width of cotter a= Distance of end of slot to the end of spigot
P= Axial tensile/compressive force
σ_t, σ_c, τ= Permissible tensile, compressive, shear stress for the component material

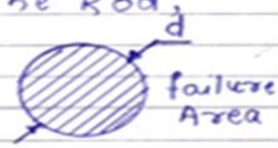
Note: Notations given for the cotter joint design may change….

Design Procedure

① Design of dia of rod (d)

Considering tensile failure of the Rod,

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



$\frac{1}{2}$ mark

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

ⓐ By emperical Relation

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

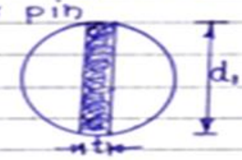
ⓑ Considering tensile failure of spigot.

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

$\frac{1}{2}$ mark

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

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

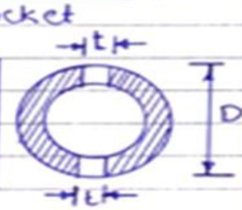


$\frac{1}{2}$ mark

③ Design of outside diameter of socket (D_1)

Considering tensile failure of socket

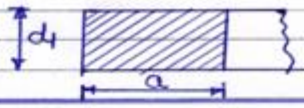
$$\boxed{\sigma_t = \frac{P}{\left[\frac{\pi}{4}(D_1^2 - d_1^2) - (D_1 - d_1)\right] \times t}}$$



1 mark

④ Design of distance from end of slot to the end of spigot (a)
Considering double shear failure along the two plane, as shown in fig. — $\frac{1}{2}$ mark

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



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

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

⑥ Design of thickness of socket collar (c)
Considering failure of socket end in shearing — 1 mark

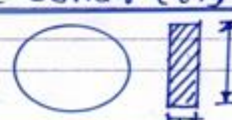
$$\boxed{\tau = \frac{P}{2[D_2 - d_1]\,c}}$$

⑦ Design of Dia. of socket collar (d_2)
Considering crushing failure of spigot collar at the contact area between socket collar — 1 mark

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

⑧ Design of thickness of spigot collar (t_1) — $\frac{1}{2}$ mark

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



⑨ Design of Width of cotter (b) — $\frac{1}{2}$ mark

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

Q.2. (c)Given Data:

Diameter of shaft= 90mm
N=300
w=20 mm
l=140 mm
$\tau = 40\ N/mm^2$
$\sigma_c = 100 N/mm^2$

To find:

Power Transmitted & Depth of Key Required

Solution:

To find torque T: $T = w \times l \times \tau \times \frac{d}{2}$

$$T = 20 \times 140 \times 40 \times \frac{90}{2}$$

$$T = 5.04 \times 10^6 N.mm$$

$\boldsymbol{T = 5.04 \times 10^3 N.m}$ --------------------- **(2 Mark)**

To find Power P: $P = \frac{2\pi NT}{60}$

$$P = \frac{2\pi \times 300 \times 5.04 \times 10^3}{60}$$

$$\boldsymbol{P = 158.33 \times 10^3 Watt}$$

$$\boldsymbol{P = 158.33 kW}$$ --------------------- **(2 Mark)**

To find depth of key h:

$$T = l \times \frac{t}{2} \times \frac{d}{2} \times \sigma_c$$

$$5.04 \times 10^6 = 140 \times \frac{t}{2} \times \frac{90}{2} \times 100$$

$$\boldsymbol{t = 16mm}$$ --------------------- **(2 Mark)**

$$Depthonkeywayonshaft(h) = \frac{t}{2}$$

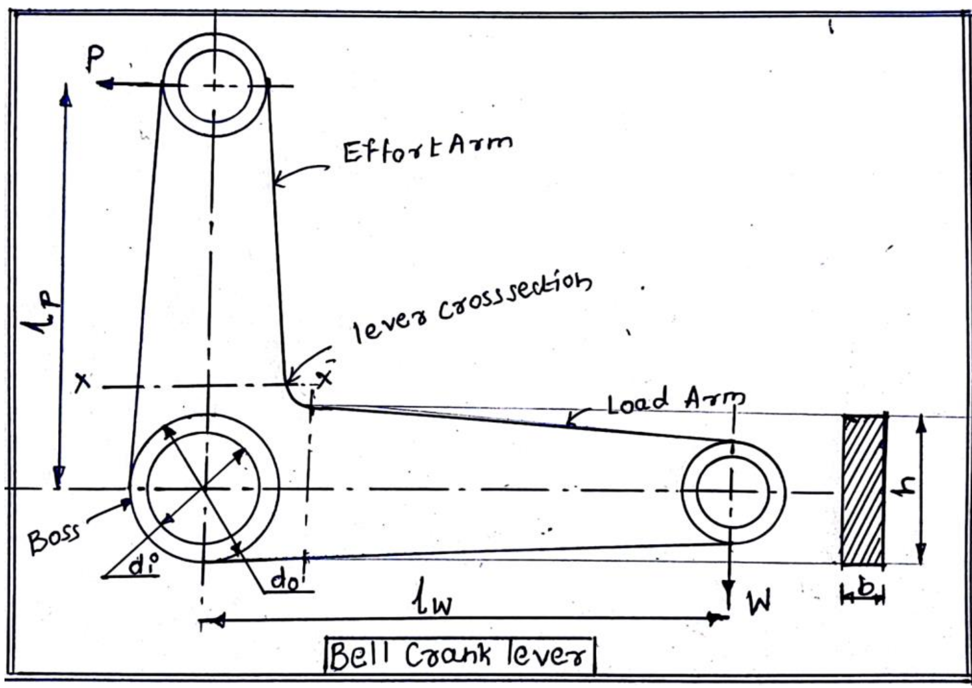
$$Depthonkeywayonshaft(h) = \frac{16}{2}$$

$$\boldsymbol{h = 8mm}$$ --------------------- **(2 Mark)**

Q.3. (a)Advantages of standardization: (Consider any four points, One mark each)

i. It saves effort of design of engineers to design and manufacture new machines, as standard components are readily designed by experts.
ii. It ensures certain minimum specified quality.
iii. It help in manufacturing the components on mass production.
iv. Easy and quick replacement of the components is possible.
v. Interchangeability of components is possible.
vi. It helps in manufacturing the components quickly and economically.
vii. Effective utilization of resources.
viii. It also contributes to ensure the safety.

Q.3. (b) Draw neat sketch of bell crank lever. Enlist steps in designing the bell crank lever. (1mark for diagram, 3 marks for design steps)



Where,

P=Effort ,W=Load

l_w= Length of load arm, l_p= Length of effort arm,

R_f= Fulcrum Reaction ,d= Diameter of pin

l_p= Length of fulcrum pin= 1.25dl_b= Length of boss= 1.25d

P_b= Bearing Pressure d_o= Outer diameter of boss

d_i= Inside diameter of boss

Consider a brass bush of 3mm thickness is fit into the boss

$d_i = d+(3\times2)$

$d_i=d+6$

M=Bending Moment

b= Width of lever cross-section h= Depth of lever cross-section

Note: Design steps are to be enlisted only .so if any student enlist the steps marks should be given not necessary to write the equations…

Design Procedure

① Determination of effort (P) — ½ mark

$$\boxed{W \times l_w = P \times l_p}$$

② Determination of fulcrum reaction (RF) — ½ mark

$$\boxed{R_F = \sqrt{W^2 + P^2}}$$

③ Design of fulcrum pin.

$$\boxed{l_p = l_b = 1.25\, d}$$

ⓐ Considering bearing pressure at fulcrum pin — ½ mark

$$\boxed{P_b = \frac{R_F}{l_b \times d}}$$

ⓑ considering double shear failure of pin — ½ mark

$$\boxed{\tau = \frac{R_F}{2 \times \frac{\pi}{4} \times d^2}} \longrightarrow \text{Checking of value of } '\tau'$$

④ Design of boss of lever

$\boxed{d_i = d + 6}$ – emperical relation

considering bending stress acting on the boss. — ½ mark

$$\boxed{\sigma_b = \frac{M \cdot Y}{I_{xx}}}$$

Boss cross section

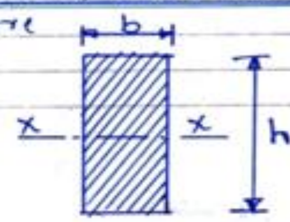
Where,

- $M = l_p \times P$, • $Y = \frac{d_o}{2}$
- $I_{xx} = \frac{1}{12} \times l_b \times [d_o^3 - d_i^3]$

⑤ Design of lever arm crossection near to boss

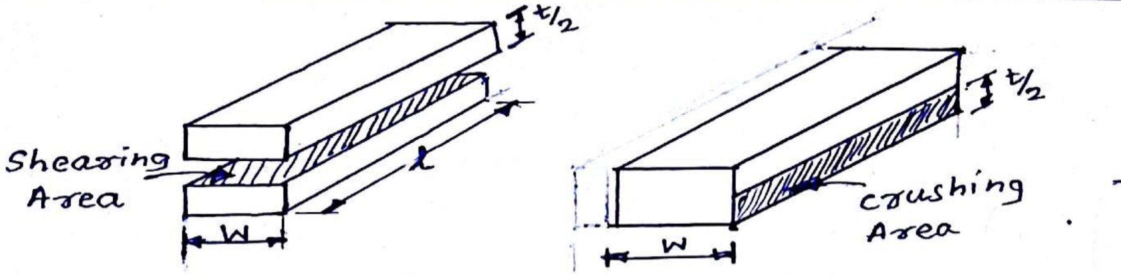
considering bending failure — ½ mark

$$\boxed{\sigma_b = \frac{M \cdot Y}{I_{xx}}}$$



- $M = P \times [l_p - \frac{d_o}{2}]$
- $Y = \frac{h}{2}$
- $I_{xx} = \frac{1}{12} \cdot b h^3$

Q.3. (c) Prove that, for square key, permissible crushing stress is twice the permissible shear stress



ⓐ Shearing failure ⓑ Crushing failure

1 mark

- Considering shearing failure of key, from fig ⓐ,

Torque Transmitted by key $= T = F \times \frac{d}{2}$ — ①

Where, F = Shearing force (tangential)

F = Shearing Area x Shearing Stress

$= l \times w \times \tau$, put in eqⁿ 1

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

1 mark

- Considering crushing failure of key from fig ⓑ,

F = tangential crushing force

= Crushing area X crushing stress

$= l \times \frac{t}{2} \times \sigma_{ck}$, put in eqⁿ 1

Torque transmitted by key $= T = F \times \frac{d}{2}$

$T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ — ③

1 mark

- If key is equally strong in shearing & crushing,

② = ③

$l \times w \times \tau \times \frac{d}{2} = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$

as, for square key $w = t$

$\boxed{\frac{\tau}{\sigma_{ck}} = \frac{1}{2} \text{ or } \sigma_{ck} = 2\tau}$

1 mark

Q.3. (d) Why a coupling should be placed as close to a bearing as possible?

(1 mark each for any four factors)

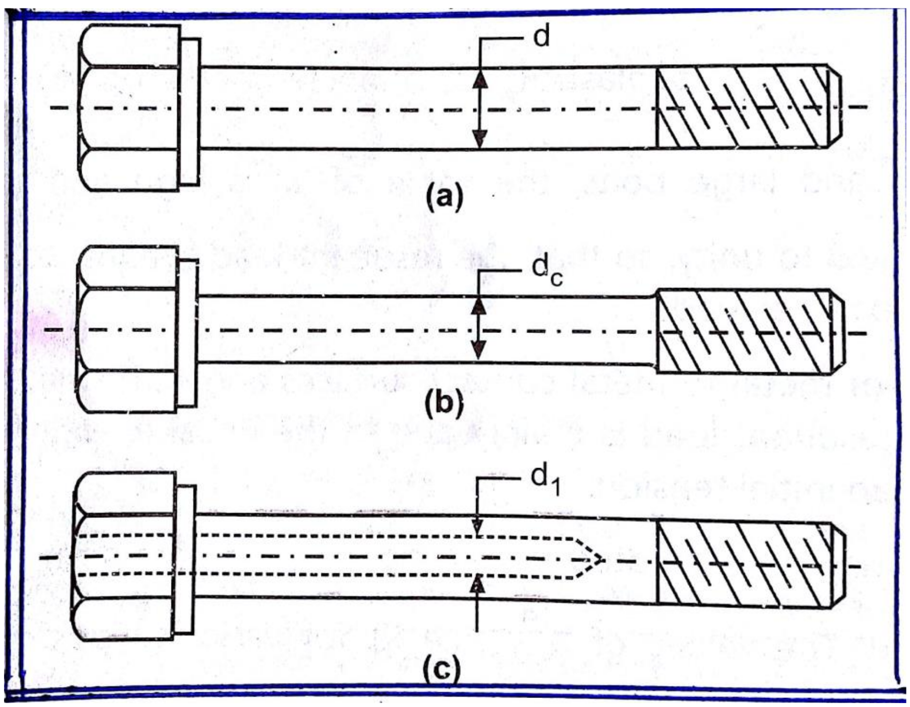
Coupling should be placed as close to a bearing because of following reasons

i. It gives minimum vibrations
ii. Bending load on the shaft can be minimized
iii. It increases power transmission stability.
iv. To avoid deflections of shaft.

Q.3. (e) Describe 'bolt of uniform strength' with neat sketch.

(1 mark for figure, 2 marks for methods, 1 mark for equation)

When an ordinary bolt of uniform diameter is subjected to shock load stress concentration across at the weakest part of the bolt i.e. threaded portion (as shown in figure a), it means that greater portion of energy will be absorbed at the region of threaded part and it may cause the failure of threaded portion.



There are two methods to achieve bolts of uniform strength

i. Turn down shank diameter of bolt equal or lesser than the core diameter of thread (d_c) as shown in figure (b) and it gives bolt of uniform strength.
ii. In this method an axial hole is drilled to the head as far as threaded portion such that area of shank become equal to the root area of thread as shown in figure (c).

Where, d_1 = Diameter of hole to be drill

d_o = Nominal diameter

d_c = Core diameter

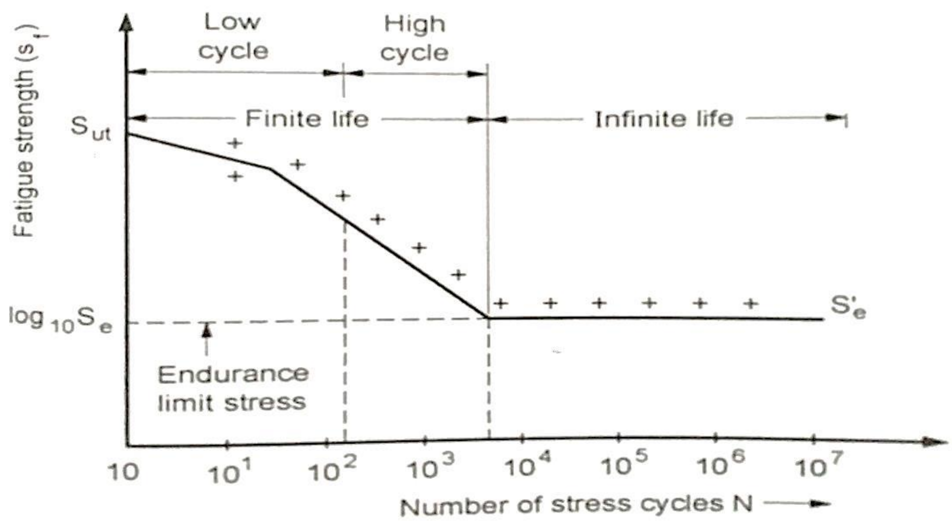
$$\frac{\pi}{4} \times {d_1}^2 = \frac{\pi}{4} \times \left({d_o}^2 - {d_c}^2\right)$$

$$d_1 = \sqrt{\left({d_o}^2 - {d_c}^2\right)}$$

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

i) **Endurance Limit:** It is defined as maximum value of the completely reversed bending stress which a polished standard specimen can withstand without failure, for infinite number of cycles (usually 107 cycles).It is known as ***endurance*** or ***fatigue limit*** (σ*e*). **………2M**

S-N Curve for steel:



2M

ii) **Effect of Keyway on strength of shaft:**

The keyway is a slot machined either on the shaft or in hub to accommodate the key.It is cut by vertical or horizontal milling cutter.

A little consideration will show that the keyway cut into the shaft reduces the load carrying capacity of the shaft.

This is due to the stress concentration near the corners of the keyway and reduction in the cross-sectional area of the shaft. It other words, the torsional strength of the shaft is reduced.

The following relation for the weakening effect of the keyway is based on the experimental results by H.F. Moore.

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

where*e* = Shaft strength factor.

w = Width of keyway,

d = Diameter of shaft, and
h = Depth of keyway =Thickness of key (t)/2

It is usually assumed that the strength of the keyed shaft is 75% of the solid shaft, which is somewhat higher than the value obtained by the above relation.
In case the keyway is too long and the key is of sliding type, then the angle of twist is increased in the ratio $K\theta$ as given by the following relation

$$K\theta = 1 + 0.4\left(\frac{w}{d}\right) - 0.7\left(\frac{h}{d}\right)$$

where$k\theta$= Reduction factor for angular twist**…………………Correct Explanation 4 M**

iii) Application of spring: **Any four …….1M each**

1) To cushion, absorb or control energy to external load : Car springs, Railway buffers
2) To store Energy : Watches Toys
3) To Measure forces : Spring Balances, Gauges ,Engines
4) To provide clamping force in Jigs & fixtures.
5) To apply forces as in brakes, clutches & spring loaded valve.

iv) Advantages& disadvantages of welded joints over riveted joints:

Advantages: **Any four …….1/2 M each**

1. The welded structures are usually **lighter** than riveted structures. This is due to the reason, that in welding, gussets or other connecting components are not used.
2. The welded joints provide **maximum efficiency (may be 100%)** which is not possible in case of riveted joints.
3. Alterations and additions can be **easily made** in the existing structures.
4. As the welded structure **is smooth in appearance**, therefore it looks pleasing.
5. In welded connections, the **tension members are not weakened** as in the case of riveted joints.
6. A welded joint has a **great strength**. Often a welded joint has the strength of the parent metal itself.
7. Sometimes, the members are of such a shape (*i.e.* circular steel pipes) that they afford difficulty for riveting. But they can be **easily welded**.
8. The welding provides **very rigid joints**. This is in line with the modern trend of providing rigid frames.
9. It is possible **to weld any part of a structure** at any point. But riveting requires enough clearance.
10. The process of welding **takes less time** than the riveting.

Disadvantages **Any four ……1/2 M each**

1. Since there is an uneven heating and cooling during fabrication, therefore the members may get **distorted or additional stresses** may develop.
2. It requires a **highly skilled labour and supervision**.
3. Since **no provision is kept for expansion** and contraction in the frame, therefore there is a **possibility of cracks** developing in it.
4. The inspection of welding work is **more difficult** than riveting work.

B)Attempt Any ONE:
1X6 =6

i) **Asthetic consideration in design related to shape ,colour& surface finish**

Regarding Shape: … ……….Any 2 pt : 1 M Each

1) The shape should not be like blocks but various forms like sculpture, streamlined, aerodynamic, taper should be used.
2) The component should be symmetrical at lean about one axis.
3)proper shape of a product help to make the product more attractive.
4) The shape of the product should be regular, even & proportionate

Regarding Colour: ……….Any 2 pt : 1 M Each

1) The colour and shape of component should be such that in should attract appeal and impress customer.
2) The colour should match with conventions, moods e.g. red for danger, gray for dull, yellow for cautions, green for safe etc.
3) Too bright colour should be avoided.
4) The colour should be compatible with conventional ideas of the operator.

Regarding Surface finish ……….Any 2 pt : 1 M Each

1) Products with better surface finish are always aesthetically pleasing'
2) The surface coating processes like spray painting, anodizing, electroplating etc greatly the aesthetic appeal of product.

ii) Design consideration while designing the spur Gear:(Any Six) .1 Marks for Each

1) The power to be transmitted
2) The velocity ration or speed of gear drive.
3) The central distance between the two shafts
4) Input speed of the driving gear.
5) Wear characteristics of the gear tooth for a long satisfactory life.
6) The use of space & material should be economical.
7) Efficiency & speed ratio
8) Cost

Qu.5 Attempt Any TWO 2X8 =16

a) **Design of screw jack**

Given Data:

W =50 KN = 50 X 10^3 N, σt_{screw} =100 N/mm^2 ,σc_{screw} =50 N/mm^2

P=8 mm, σt_{nut} =50 N/mm^2 ,σc_{nut} =45 N/mm^2, τ_{nut} =40 N/mm^2

P_b= 20 N/mm^2, μ =0.14

Design of Screw:

1)Consider the screw under pure compression to find diameter of screw

$$\sigma c = \frac{W}{\frac{\pi}{4} X (dc)^2} \quad , \quad 50 = \frac{50 \text{ X } 103}{\frac{\pi}{4} X (dc)^2} \quad d_{c=}35.68 \text{ Say } 36 \text{ mm} \quad \ldots \ \mathbf{1\ M}$$

As screw is subjected to twisting moment ,higher value of screw is selected .

The dimension of $d_{c=}$ 42 mm for P=8

Mean diameter d= do- p/2 = 50-8/2 = 46 mm

2) Torque required to overcome the friction (T_1)

Helix angle $\alpha = tan^{-1} \frac{8}{\pi x 46}$ = 3.17°

$\emptyset = tan^{-1}\mu$ $= tan^{-1} x\ 0.14$ = 7.97° ……………… **1 M**

Torque required lifting the load

T1= W. tan ($\alpha + \emptyset$) $\frac{d}{2}$

T1= 50 x 10 ^{3} tan (3.17+7.97) $\frac{46}{2}$ = 226416.5 N.mm ……………… **1 M**

As collar friction is Neglecting, T_2=0

Total Torque required to lift the load = T_1= 226416.5 N.mm ……….. **1 M**

For Checking:

Direct compressive stress in screw:

$$\sigma c = \frac{W}{\frac{\pi}{4} X (dc)^2} \quad , \quad \sigma c = \frac{50 \text{ X } 10^3}{\frac{\pi}{4} X (42)^2} \quad \sigma c = 36.09 \text{ N/ mm}^2$$

Torsional shear stress τ

$$\tau = \frac{16\ T1}{\pi\ X\ (dc)^3}, \quad \tau = \frac{16\ X\ 226416.5}{\pi\ X\ (42)^3}, \tau = 15.56 \text{ N/ mm}^2$$

According to Maximum shear stress theory, the maximum shear stress in the screw

$\tau_{max} = 1/2\sqrt{\sigma c^2 + 4\ \tau^2}$

$\tau_{max} = 1/2\sqrt{36.09^2 + 4\ (15.56)^2}$ = 23.83 N/mm^2

Permissible shear stress for a screw τ = $\sigma c/2$ =50/2 =25 N/mm^2

$\tau_{max} < a_{llowable}$, So screw is safe. ……………… **1 M**

Design of Nut:

The bearing pressure between the thread

$Pb = \frac{W}{\frac{\pi}{4} X (do^2 - dc^2)\, n}$, $20 = \frac{50 X 10^3}{\frac{\pi}{4} X (50^2 - 42^2)\, n}$, n= 4.32 i.e = 5 threads in contacts **1 M**

Height of Nut: h=n x p =5x8 =40 mm **… 1 M**

Check: Shear stress induced in the screw thread

$\tau = \frac{W}{\pi X (dc) X t n}$ as t= p/2

$\tau = \frac{50 X 10^3}{\pi X \frac{(42) X 8}{2} X 5}$ =18.95 N/mm^2 < 40 N/mm^2

$\tau_{calculated} < \tau_{allowable}$, So screw is safe **… 1 M**

b) Design of spring

Given Data: m=1500 kg , V= 1 M/s , $\delta = 150\ mm$,

τ =360 N/m , G= 8.4 x 10^4 N/mm^2 , C= 6

K.E=$\frac{1}{2}\ M\, V^2$ = $\frac{1}{2}\ 1500 x 1^2$ =750 N.m = 750 x 10^3 N.mm **…… 1 M**

Energy stored in spring = $\frac{1}{2}\ W\ \ \delta\ \ X\ 2$ (2 Buffer spring)

750 x 10^3 = W x 150 , W= 5 X 10^3 N **……………… 1 M**

Torque transmitted by spring

T=W X Dm/2 = 5 X 10^3 x (C x d)/2 = 5 X 10^3 x (6 x d)/2 =15x 10^3 d **……1 M**

T= π /16 x τ x d^3 =(16x15x10^3) / (π x 360)

d= 14.56 mm =15 mm **……………… 1 M**

C= Dm/d = 6= Dm/15 , Dm = 90 mm

$\delta = \frac{8\ W\ D^3\, n}{G\, d^4}$, $150 = \frac{8 X 5 X\ 10^3\ x\, 90^3\, n}{8.4\, x\, 10^4 x\, 15^{\ 4}}$ n= 21.87i.e 22 turns …. **1 M**

Assuming squared & grounded ends ,total number of truns is given by

n' = n +2 = 22+2 =24

Solid Length = Ls= n' x d = 24 x 6 = 144 mm **… 1 M**

Free Length = Fs = n' x d + $\delta max + 0.15\ \delta max$

Fs = 22 x 6 + $150 + 0.15\ x\ 150$ = 304.5 mm …. **1 M**

Pitch of coil = P=$\frac{free\ length}{n'-1}$ = $\frac{304.5}{24-1}$ = 13.24 mm **… 1 M**

c) i) **Efficiency of self-locking screw is less than 50 %**

Torque required to lower the load

T_1= P X $\frac{d}{2}$ = W. tan ($\emptyset - \alpha$) $\frac{d}{2}$**……………… 1 M**

If however ,Φ > α 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 self-locking

1) if the friction angle is greater than helix angle or coefficient of friction is greater than tangent of helix angle i.e μ or tan Φ > tan α… **1 M**

We know that the efficiency of screw,

$$\tilde{n} = \frac{\tan \Phi}{\tan(\emptyset + \alpha)}$$ ……………… **1 M**

Therefore, Efficiency for self-locking screws,

$$\eta \leq \frac{\tan \phi}{\tan(\phi + \phi)} \leq \frac{\tan \phi}{\tan 2\phi} \leq \frac{\tan \phi (1 - \tan^2 \phi)}{2 \tan \phi} \leq \frac{1}{2} - \frac{\tan^2 \phi}{2}$$

………..**1M**

From this expression we see that efficiency of self-locking screws is less than 1/2 or 50%.

c) ii) **Advantages of ball bearings over plain journal bearing: …….Any Four 1 M Each**

1) The ball bearings have a far smaller contact area and thus have a lower **frictional drag** coefficient.

2) Due to less frictional drag means better response and less power consumption.

3) The turbo can spool up much faster, which reduces turbo-lag and offers a major **performance advantage** over journal bearing turbochargers at lower to mid turbocharger speeds.

4) The reduced contact area of the ball bearings means that it requires far **less lubrication**, allowing for lower oil pressure feeds.

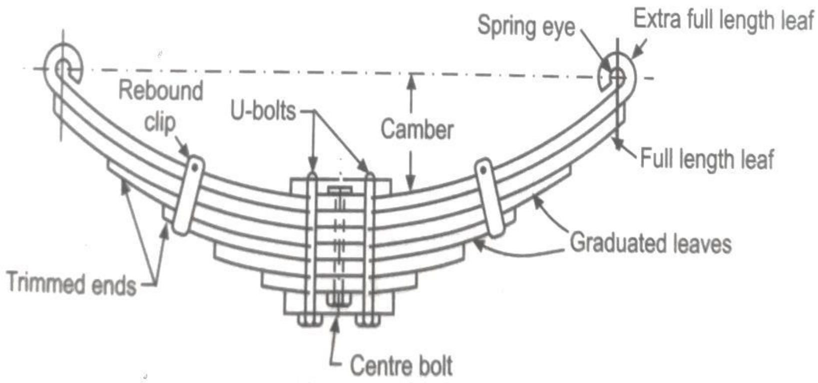
5) The ball bearing more reliable.

6) Less expensive.

7) Ball bearings generate less heat and require simple and inexpensive lubrication - by oil ring, oil mist, or oil bath method.

Qu.6 Attempt Any FOUR **4 X 4 =16 M**

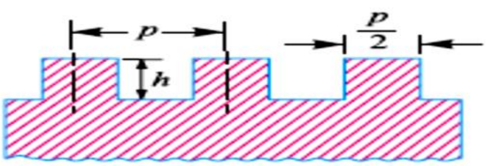
a) Sketch of Leaf Spring of semi elliptical Type ….Diagram+ Names : 2M+2M



b) Application of Square threads: ……………. 1M

1) Feed mechanisms of machine tools, 2) valves, 3)spindles, 4) screw jacks etc
Sketch

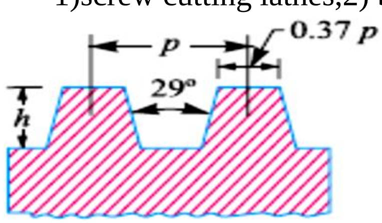
……..:1M



$h = 0.5\,p$
(*a*) Square thread.

Application of Acme threads : ………1M

1)screw cutting lathes,2) brass valves,3) cocks and 4) bench vices.



$h = 0.5\,p + 0.25$ mm
(*b*) Acme thread. ………1M

c)Given Data: Effective Diameter D =400 mm

p= 1.5 N/mm^2 , n=16 ,σt =25 Calculate: Bolt Size =?

The force acting on cylinder head is given by

$P = \frac{\pi}{4} D^2\, p$, $P = \frac{\pi}{4}\ x\ 400^2\ x 1.5$ = 188.49 x 10^3 N/mm^2…… **1 M**

Tensile stress $\sigma t = \frac{P}{\frac{\pi}{4} dc^2\ n}$, $25 = \frac{1.5}{\frac{\pi}{4} x\ dc^2\ x\ 16}$ dc=24.49 mm …… **1 M**

Nominal diameter do = dc /0.84 = 24.49 /0.84 =29.16 mm …… **1 M**

Bolt size will be M 30 or M32 **…… 1 M**

) Disadvantages of rolling bearing as compared to Journal Bearing:(Any Four , 4m)

1) Initial cost is very high
2) Noisy in normal operation.
3) Shock capacity is less.
4) Finite life due to failure by fatigue.
5) Dirt & metal chips can enter the bearing & may lead it to failure.
6) Occupies greater diametral space compared to journal bearing.

e) Application of bearings : **……1 mark for Each**

i) **Deep Groove Ball bearing :** Electric Motor
ii) **Taper roller bearing :** axle housing of automobile
iii) **Thrust collar bearing:** Clutch of automobile
iv) **Needle roller bearing:** Differential of automobile