SUMMER – 13 EXAMINATION

Subject Code:12241 **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 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. Any TEN (10 x 2)

a) It is the process of selection of the materials, shapes, sizes and arrangements of mechanical elements so that the resultant machine will perform the prescribed task.

OR

It is creation of new & better machines & improving the existing ones.

b) Dead or steady load, live or variable load, suddenly applied or shock loads, impact load.(½ each)

c) The loads, which vary in magnitude and / or direction with respect time, are known as fatigue.

e.g. fatigue failure begins with a small crack. The crack initials at a point (hole, keyway etc) of discontinuity in crank shaft and it propagates with the cyclic stress which result in fatigue failure of shaft.

d) It may be defined as the degree to which the theoretical expected effect of stress concentration is actually reached. It is ratio of Increase of actual stress over the nominal stress to Increase of theoretical expected stress over the nominal stress.

e) i) It is easy to remove the cotter & dismantle the joint.

ii) It ensures tightness of the joint in operation & prevents loosening of the parts.

f) It holds collar and prevent lifting or ejecting the knuckle pin from the joint.

g) (any two one each)

Shaft	Axle	Spindle
i) It is rotating element which transmits power from one place to another	It is a stationary m/c element used to support a rotating body.	It is a short shaft which imparts motion either to a cutting tool or work piece.
ii) e.g. propeller shaft	e.g front and rear axle of motor cycle	e.g. drill spindle
iii) subjected to torque, bending moment or axial force	Subject to bending moment	Subject to torque, B.M or axial force

h) Sun keys – Rectangular, square, parallel, Gib head, feather, woodruff key

Saddle keys – Hollow, flat, tangent keys, Kennedy keys, round keys, splines

i)

Coupling	Clutch
i) It is a device used to make permanent or semi-permanent connection.	i) It permits rapid connection or disconnection at the will of the operator
ii) It provides for the connection of shafts of two different units such as an electric motor & a machine.	ii) In automobiles, clutch is used to connect and disconnect the engine shaft to the driven shaft main shaft.

j) i) The relative position of input & output shafts, ii) speed ratio, iii) efficiency, iv) input speed,

v) power to be transmitted, vi) cost. (any 4, 1/2 each)

k) It is a translating screw are used to convert rotary motion into translating motion. It is a screw & nut system to transmit power.

l) i) High resilience, ii) Ductile, iii) High static strength, iv) High fatigue strength, v) Creep-resistant vi) Non-corrosive. (any 4,1/2 each)

m) i) Butt weld – square butt, single V-butt, single U-butt, Double V-butt, Double U-butt.

ii) Fillet weld or lap weld – parallel fillet weld, transverse fillet weld.

n) (Any four)

Colour	Meaning
Red	Danger, hot
Orange	Possible danger
Yellow	Caution
Green	Safe
Blue	Cold
Gray	Dull

Q. 2. Any FOUR

a. i) Depending upon the direction of load to be supported. – Redial bearing, thrust bearing

ii) Depending upon the nature of contact

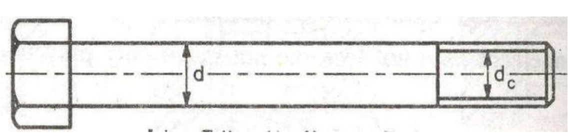
Sliding contact ,
- Hydrodynamic
- Hydrostatic
  - Guide
  - Journal
    - Full Journal
    - Partial
  - Thrust
    - Pivot
    - Collar

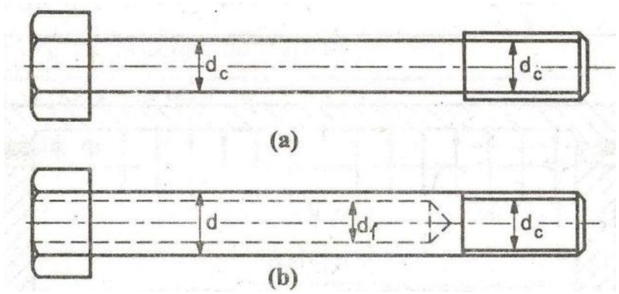
Rolling contact
- Ball Bearing
  - Single – Raw deep
  - Angular contact
- Roller Bearing
  - Cylindrical roller
  - Taper roller

b. i)Aesthetics: A set of principles of appreciation of beauty. It deals with the appearance of the product at any stage in the product life, the aesthetic quality cannot be separated from the product quality.(2)

ii) Ergonomics: It is the scientific study of the man-machine working environment relationship and the application of anatomical, physiological and psychological principles to solve the problems arising from this relationship. (2)

c. When a bolt is subjected to a shock loading as in case of a cylinder head bolt of an I.C. engine, the resilience of the bolt should be considered in order to prevent breakage at the thread.





(a)
(b)

If a shank diameter is reduced to a core dia. as shown in fig-(a) the stress become same throughout the length of the bolt. Hence impact energy is distributed uniformly throughout the bolt length, thus relieving the threaded portion of high stress. The bolt in this way becomes stronger & lighter. Another method of obtaining the bolt of uniform strength in shown in fig.(b). In this method instead of reducing the shank diameter, an axial hole is drilled through the head down the threaded portion such that the cross sectional area of the shank becomes equal to the core area of the threaded portion. (2+2 marks)

For bolt of uniform strength $\frac{\pi}{4} dc^{c} = \frac{\pi}{4}(d^{2} - d_{1})$

$$\therefore d_1 = \sqrt{d^2 - dc^2}$$

d. Mean diameter of the screw $d = do - \frac{P}{2} = 50 - \frac{8}{2} = 46mm$

$$\tan\alpha = \frac{P}{\pi d} = \frac{8}{\pi \times 46} = 0.055$$ (1mark)

$$\mu = \tan\phi = 0.15$$

Tangential force required at the circumference of the screw.

$$P = W\ \tan(\alpha \times \phi) = 100 \times 10^3 \left[\frac{0.055 + 0.15}{1 - 0.055x0.15}\right]$$ (1mark)

$$= 20.67 \times 10^3 N$$

Speed of the screw in revolutions per minute

$$N = \frac{Speed\ in\ mm/\min}{Pitch\ in\ mm} = \frac{360}{8} = 45\,rpm$$

Angular speed, $\varpi = \frac{2\pi N}{60} = \frac{2\pi \times 45}{60} = 4.71\, rad/\sec$ (1mark)

$\therefore$ Power of the motor $P = Tw = 475.42 \times 4.71 =$
$P = 2239.23W = 2.239kw$ (1mark)

e. Deflection of the spring $\delta = \frac{8WD^3 n}{Gd^4}$ $\frac{w}{\delta} = \frac{Gd^4}{8D^3 n}$ (1)

Since G, D & d are constant.

$\frac{Gd^4}{8D^3} = X$, a constant, $\frac{w}{\delta} = k = \frac{X}{n}$ (½)

$$X = k.n = 12k \quad (½)$$

$K_1 = \frac{X}{n_1} = \frac{12k}{5} = 2.4K$ (1)

$K_2 = \frac{X}{n_2} = \frac{12k}{7} = 1.7K$ (1)

f. General Design Procedure (4marks)

- Definition of problem
- Synthesis
- Analysis of forces
- Selection of material
- Determination of mode of failure
- Selection of factor of safety
- Determination of dimensions
- Modification of dimensions
- Preparation of drawings
- Preparation of design report

Q. 3. Any four (4 x 4)

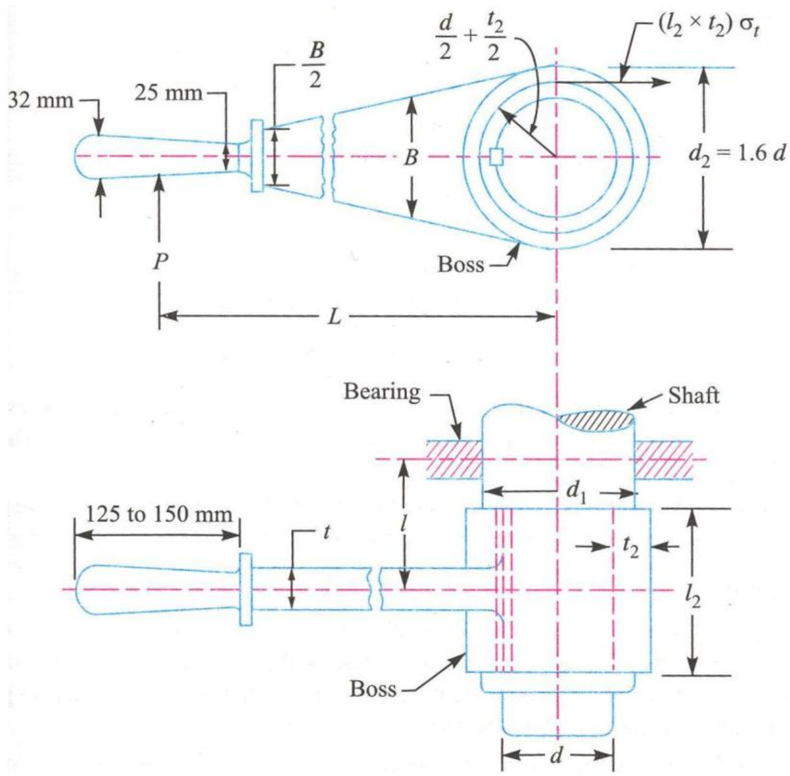
a. The diameter of the shaft is obtained by considering the shaft under pure torsion

$$T = P \times L$$

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

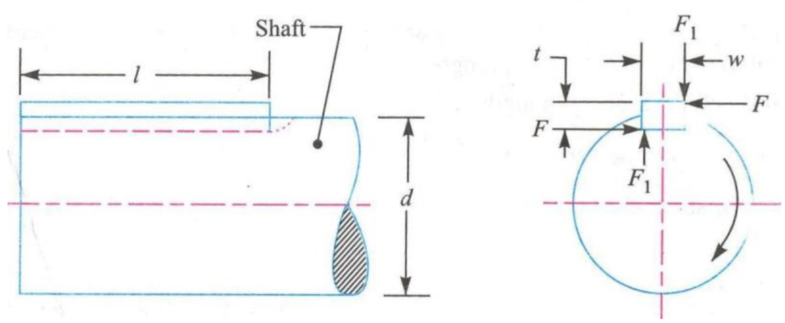
d = from this relation diagram of the shaft (d) may be obtained

The diameter of the boss (d_2) is taken as 1.6 d.



(2+2 marks)

b. Considering shearing of the key, the tangent shearing force acting at the circumference of the shaft (1+1+1+1)



F = area resisting shearing x shear stress = l x w x τ

$\therefore$ Torque transmitted by the shaft

$$T = F \times \frac{d}{2} = l \times w \times \tau \times \frac{d}{2} \qquad (1)$$

Considering crushing of the key, tangential crushing force

F = Area resistance crushing x crushing stress

$$F = l \times t/2 \times \sigma_c \times d/2 \qquad (2)$$

The key is equally strong in shearing & crushing if $l \times w \times \tau \times \frac{d}{2} = l \times \frac{t}{2} \times \sigma c \times \frac{d}{2}$

$$\frac{w}{t} = \frac{\sigma c}{2\tau}$$

$$\therefore \sigma c = 2\tau$$

$$w = t \quad for\ squre\ key$$

c. Torque transmitted by the shaft

$$T = \frac{P \times 60}{2\pi N} = \frac{25 \times 10^3 \times 60}{2\pi \times 250} = 954.93\, N - m$$

Torque transmitted by the shaft

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

$$954.93 \times 10^3 = \frac{\pi}{16} \times 42 \times d^3$$ (4 marks)

$$d = 48.75 \quad say\ 50mm$$

d. Length of the weld for static loading (2 marks)

Maximum load

$$P = 1.414.s \times l \times \tau$$

$$50 \times 10^3 = 1.414 \times 12.5 \times l \times 56 = 990l$$

$$l = 50.5mm$$

Adding 12.5 mm for starting & stopping of weld run

$$l = 50.5 + 12.5 = 63mm$$

Length of the weld for fatigue loading (2 marks)

Permissible shear stress $\tau = \frac{56}{2.7} = 20.74 N / mm^2$

Maximum load

$$P = 1.414 \times s \times l \times \tau$$

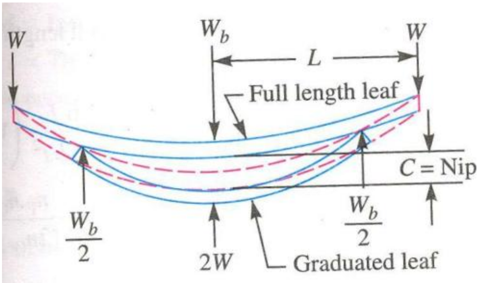
$$50 \times 10^3 = 1.414 \times 12.5 \times l \times 20.74$$

$$l = 136.2mm$$

Adding 12.5 for starting & stopping of weld run

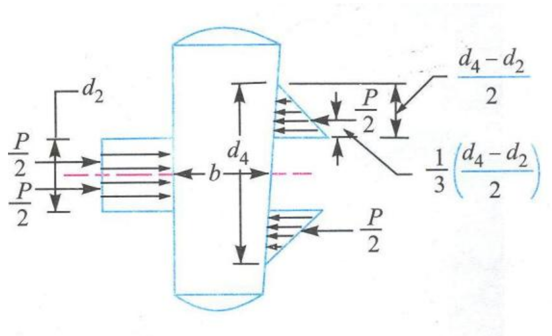
$$l = 136.2 + 12.5 = 148.7mm$$

e. Nipping : The stresses in full length are 50% grater than the stresses in graduated leaves; all the leaves should be equally stressed. This may be achieved by pre-stressing the leaves.



The pre-stressing of the spring can be done by giving greater radius of curvature to the full length leaves than the graduated leaves before assembly as shown in fig. The initial gap 'C' between full length leaf and graduated leaf before assembly in called nip. When the central bolt holding leaves together is tightened, the extra full length leaf will bend back as shown by dotted lines & have an initial stress in a direction opposite to that of normal load. This process of pre-stressing the spring by giving, different radii of curvature before assembly is known as nipping. (1+3marks)

f. In order to find out the bending stress included, 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) as shown in fig. The maximum bending moment occurs at the centre of the cotter and is given by



and section modulus of the cotter $z = \frac{tb^2}{6}$

$\therefore$ Bending stress induced in the cotter

$$\sigma_b = \frac{M \max}{2}$$ (2+2marks)

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

Q. 4.

a) Given : d = 120 mm, P = 24 mm, W = 20 kN, D_1 = 300 mm.

R_1 = 150 D_2 = 100 mm R_2 = 50mm l = 400 mm μ = tan ϕ = 0.15 μ_1 = 0.25

Let P_1 = force required at the end of the lever for a 2 shaft square thread

Lead = 2P = 2 x 24 = 48 mm

$$\tan\alpha = \frac{Lead}{\pi d} = \frac{48}{\pi \times 120} = 0.127$$

1) For lifting the load circumference

Tangential force required at the of the screw

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

(2 marks)

$$= 20\times10^3\left[\frac{0.127+0.15}{1-0.127\times0.15}\right] = 5648N$$

Mean radius of the collar

$$R = \frac{R1+R2}{2} = \frac{150+75}{2} = 112.5mm$$

Torque required at the end of the lever

$$T = P \times \frac{d}{2} + \mu, WR$$

$$= 5648 \times \frac{120}{2} + 0.25 \times 20 \times 10^3 \times 112.5$$ (2 marks)

$$= 901380\ N - mm$$

Also, torque required at the end of lever

$$T = P_1 \times l \quad \therefore 901380 = P_1 \times 400mm$$

Force required at end of lever

$$P_1 = 2253.45N$$ (2 marks)

2) For lowering the load

Tangential force at circumference of screw

$$P = w\tan(\theta - \alpha)$$

$$= w\left[\frac{\tan\theta - \tan\alpha}{1 - \tan\alpha\tan\theta}\right] = 451N$$

Torque required at the end of lever

$$T = P \times \frac{d}{2} + \mu_1\ WR$$

$$= 364560N$$

Also $T = P_1 \times l \therefore 364560 = P_1 \times 400$

Force read at the end of lever

$$P_1 = 911.60N$$ (2 marks)

b) Design procedure of knuckle joint

(2 Marks for sketch and 1 mark for each step max 6)

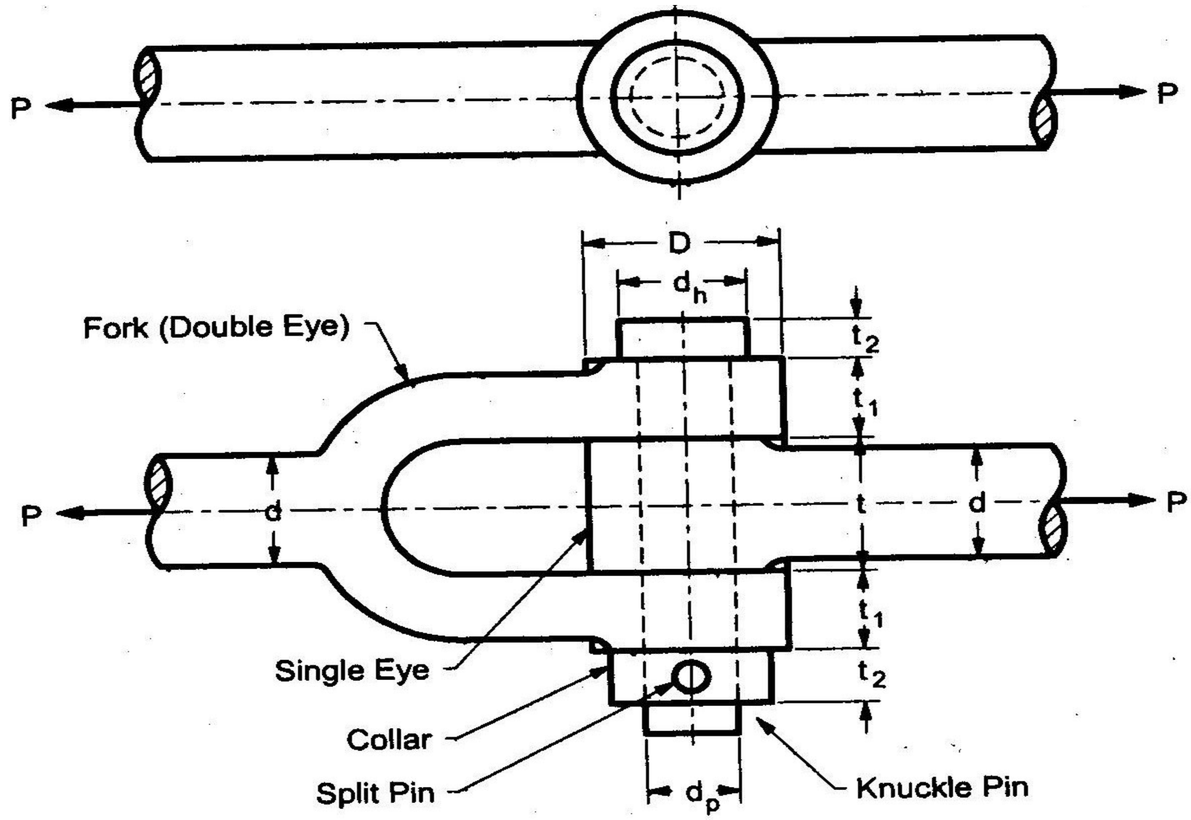


Fig. Shows a knuckle joint with dimensions of the parts. Following procedure is adopted by considering the various methods of failure.

1) Failure of solid rod in tension:- If d = diameter of rod 6_t = tensile stress of rod
   Load transmitted

   $$P = \frac{\pi}{4} \times d^2 \times 6t$$

   Diameter d is calculated for a given load
   The imperial dimensions formulated by experience are calculated as below.
   Dia. of knuckle pin = d_1 = d
   Outer dia of eye = d_2 = 2d.
   Dia. of knuckle pin head & collar
   d_3 = 1.5 d
   Thickness of single eye or rod end.
   t = 1.25 d
   Thickness of fork t_1 = 0.75 d
   Thickness of pin head t2 = 0.5 d
   All these checked by considering failures of various components.
2) Failure knuckle pin in shear : (double shear)

   $\therefore$ load of $P = 2 \times \frac{\pi}{4} \times d_1^2 \times \tau$

   Calculate τ, check for less than given value.
3) Failure of single eye or rod end in tension

Load $P = (d_2 - d_1) \times t \times 6t$

Check for $6t \angle$ given value.

4) Failure of single eye or rod end in shear

Load $P = (d2 - d1) \times t \times \tau$

Check for τ ∠ given value.

5) Failure of single eye or rod end for crushing

Load $P = d1 \times t \times 6c$ check for $6c \angle$ given value.

6) Failure of forked end in tension

Load $P = (d2 - d1) \times 2t_1 \times 6t$ check for $6t \quad \angle$ given value.

7) Failure of forked end in shear

Load $P = (d2 - d1) \times 2 \times t_1 \times \tau$ check for $\tau \leq$ given value (3 marks)

8) Failure of forked end in crushing.

Load $P = d_1 \times 2 \times t_1 \times 6c$ check for $6c \quad \angle$ given value.

c) Given : $m = 20\ tons = 20{,}000\,kg\ V = 9\ kmph = 9 \times \dfrac{1000}{3600} = 2.5\,m/s$

$D = 360\,mm\ \delta = 300\,mm\ \tau = 800\,MPa\ or\ N/mm^2$

Design of spring.

1) Diameter of spring wire : (d)

K.E. of wagon $= \dfrac{1}{2} mV^2$

$$= \frac{1}{2} \times 20{,}000 \times (2.5)^2 = 62500\,N - m \quad (i)$$

$$= 62500 \times 1000\,N - mm$$

Let w be equivalent load which when applied gradually on each spring, causes a deflection of 250 mm, since there are two springs.

Energy stored in springs

$$= \frac{1}{2} w \times \delta \times 2 = 300 \times w \qquad (ii)$$

Equate (i) & (ii) since K.E of wagon = energy stored in springs.

$\therefore 62500 \times 1000\ N - mm = 300 \times w$

$\therefore w = 208 \times 10^3\,N \qquad (2\ marks)$

But, torque transmitted by spring,

$$T = w \times \frac{D}{2}$$

$$= 208 \times 10^3 \times \frac{360}{2}$$

$$= 37.5 \times 10^6\,N - mm$$

Also torque transmitted

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

$$37.5 \times 10^6 = \frac{\pi}{16} \times 800 \times d^3$$

$$\therefore d^3 = 238.7 \times 10^3 \quad or \quad d = 62.005mm\ say\ 64mm \qquad (2\,marks)$$

2) No. of turns of the spring coil.

net n = no. of active turns of spring coil

we know deflection (δ)

$$300 = \frac{8WD^3n}{Gd^4} = \frac{8 \times 208 \times 10^3}{84 \times 10^3 \times (64)^4}$$

$$\therefore (Taking\ G = 84\,MPa = 84 \times 10^3\ N/mm^2$$

$$\therefore n = 5.45\ say\ 6$$

Assuming square and ground ends

Total No. of turns

$$n^1 = n + 2 = 6 + 2 = 08 \qquad (2\ marks)$$

3) Free length of the spring

$$L_F = n^1.d + \delta + 0.15\,\delta$$

$$= 0.8 \times 64 + 300 + 0.15 \times 300$$

$$= 857mm \qquad (2\,marks)$$

Q. 5.

Factors governing the selection of factor of safety. (any eight 4 marks)

1) The reliability of the properties of material and change of these properties during service.
2) Reliability of test results and accuracy of application of these results to actual machine parts.
3) Reliability of applied load.
4) The certainty as to exact mode of failure.
5) Extent of simplifying assumptions.
6) The extent of localized stresses.
7) The extent of initial stresses set up during manufacture.
8) The extent of loss of life if failure occurs.

b) Given : L = 1m =1000 mm, P = 800 N, τ = 70 Mpa = 70 N/mm^2.

Let d = Diameter of shaft

Twisting moment on the shaft (T)

T = P x L = 800 x 1000 = 800 x 10^3 N-mm

Also Twisting moment (T)

$$T = 800 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 \qquad \text{(4 marks)}$$

$$\therefore d = 38.8mm \quad or\ say\ 40mm$$

Diameter of boss

$d_2 = 1.6d = 1.6 \times 40 = 64mm$

Thickness of boss $t_2 = 0.3d = 0.3 \times 40 = 12mm$

Length of boss $l_2 = 1.25d = 1.25 \times 40 = 50mm$

Now considering the shaft under combined twisting and bending moment, diameter of the shaft at the centre of bearing (d_1) is given by

$$\frac{\pi}{16} \times \tau \times d_1^3 = P\sqrt{l^2 + L^2}$$

$$\frac{\pi}{16} \times 70 \times d_1^3 = 800\sqrt{100^2 + 1000^2}$$

$$where\ l = 2 \times l_2$$

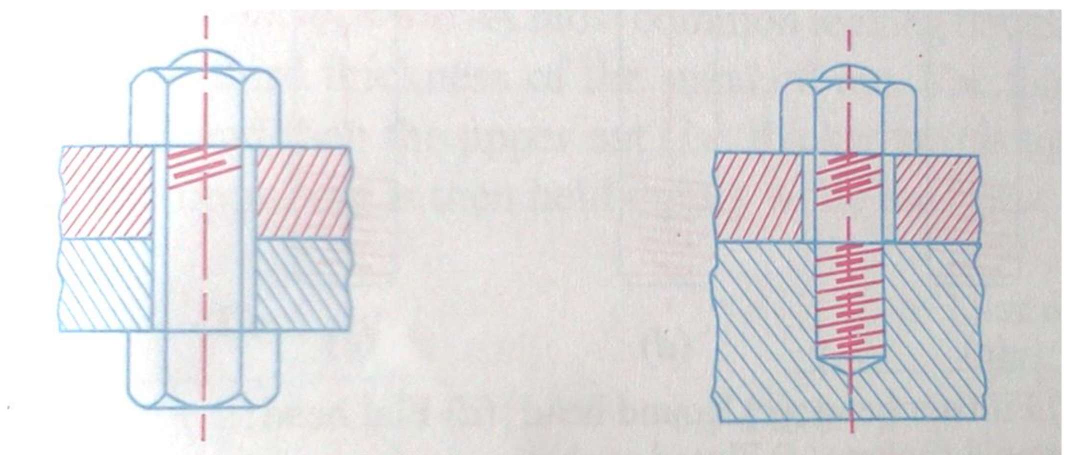
$$\therefore d_1 = 38.8\ or \quad 40mm$$

c) Properties of sliding contact bearing materials.

1) Compressive strength – The bearing material should have high compressive strength to with stand the maximum bearing pressure so as to prevent extrusion or other permanent deformation of the bearing.
2) Fatigue strength – The bearing material should lave sufficient fatigue strength so that can with stand repeated loads without developing surface fatigue cracks.
3) Embeddability – It is ability of the bearing material to accommodate or embed small particles of dust, grit etc, without scoring the material of bearing.
4) Bondability – Many high capacity bearings are made by bonding one or more thin layers of a bearing material to high strength steel shell strength of bond is important consideration while selecting bearing material.

(1 mark each)

d)



(a) Through bolt. (b) Tap bolt.

i) Through bolt – It is a cylindrical bar with the threads at one end for the nut and head at the other end. The cylindrical part is called shank it is passed easily through the drilled holes in the two parts to be fastened together and clamped them securely to each other as nut is screwed at the threaded end. The bolt is put under tension along it axis, when tightened. They are also known as machine bolts, carriage bolt, eye bolts. (2 marks)

ii) Tap bolts – A tap bolt suffers from through bolt since of has threads through out the length it is screwed into the tapped hole of one of the parts to be fastened.

(2 marks)

e) Give P = 50 kw = 50 x 10^3 w

N = 450 rpm

for shaft & key - $\tau = 40\,MPa\ 6c = 80\,MPa$

for muff – Assume $\tau c = 15\,N/mm^2$

1) Design of shaft –

Let d = diameter of shaft

Torque transmitted

$$T = \frac{P \times 60}{2\pi N} = \frac{50 \times 10^3 \times 60}{2\pi \times 450}$$

$$= 1.06 \times 10^3\ N - m$$

$$T = 1.060 \times 10^3 N - mm$$

Also Torque

$$1060 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$\therefore d^3 = \frac{1060 \times 10^3 \times 16}{40 \times \pi}$$

$$d = 49.3 \quad say \quad 52mm \qquad (1\,mark)$$

2) Design for sleeve

Outer diameter of muff

$$D = 2d + 13 = 2 \times 52 + 13 = 117 \quad say\,120mm$$

Length of muff

$$L = 3.5\,d = 182mm\ say\,185mm$$

Let us check for induced shear stress in the muff. Considering muff as hollow shaft

$$1060 \times 10^3 = \frac{\pi}{16} \times \tau_c \times \left[\frac{D^4 - d^4}{D}\right]$$

Torque, $= \frac{\pi}{16} \times \tau_c \left[\frac{(120)^4 - (52)^4}{120}\right]$ this value is less than the permissible shear stress.

$$\tau_c = 3.23\,N/mm^2$$

$\therefore$ design is safe (1 mark)

3) Design for key –

For shaft of 5.2 mm diameter $\left[Take\ w = \frac{d}{3}\right]$

$\therefore$ Wilder of key w = 18mm

Since crushing stress for key material is twice the shearing stress, a square key. Can be used $\therefore$ Thickness key t = w = 18mm

Length of key in each shaft

$$l = L/2 = 185/2 = 93mm$$

Let is check for induced shear and crushing stress in two key.

$\therefore$ Torque $1060 \times 10^3 = l \times w \times \tau \times \frac{d}{2}$

$$\therefore \tau = 24.5\ N/mm^2 < \quad given\ value$$

Also for crushing $\therefore$ Design is safe

Torque $1060 \times 10^3 = l \times \frac{t}{2} \times 6c \times \frac{d}{2}$

$$\therefore 6c = 49\ N/mm^2 given\ value$$

Design of safe (2 marks)

g. The effort required of the circumference of the screw to lower the load is P = W tan (P-α)

The torque required to lower the load

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

i) Overhauling : In the above equation if $\phi < \alpha$, than the torque required to lower the load will be negative. This means the load will be negative. This means the load will start moving downward without the application of any torque. This condition is known as overhauling.

(2 marks)

ii) Self locking : If in above equation, $\phi < \alpha$, the torque required to lower the load will be positive. This means come effort is required to lower the load such a condition is known as self locking. In such a case friction angle is greater than the helix angle.

(2 marks)

Q.6. Ball bearings : (2 marks each)

a) i) Basic static load rating:- The load carried by non-rotating member is called static load. The basic static load rating is the static radial load or axis load which corresponds to a total permanent deformation of ball (or roller) and race, at the most heavily stressed contact, equal to 0.0001 times the ball (or roller) diameter.

As per IS 3823 – 1984, for radial ball bearings basic static load rating is given by

Co = f_o x I z x D^2 cos α

i = no. of rows of balls

z = no. of balls per row

D = dia. of balls in mm

α = nominal angle of contact

f_o = a factor depending on type of bearing. (2 marks)

ii) Basic dynamic load rating : It is defined as constant stationary radial load or constant axial load which a group of apparently identical bearings with stationary outer ring can endure for rating life of one million revolutions with only 10% failure.

The basic dynamic load rating, as per IS 3824 – 1983 is given by $C = Fc\,(i \cos \alpha)^{0.7}\, Z^{2/3}\, D^{1.8}$

Where Fc = factor depending upon the geometry of the bearing component accuracy of manufacture x material used.

(2 marks)

b) Given W = 20 KN = 20 x 10^3 N L= 500 mm

L_1 = 50mm L_2 = 400 mm 6_t = 90 Mpa = 90 N/mm^2 n = 4.

Direct shear load on each bolt,

$$W_S = \frac{w}{n} = \frac{20\times10^3}{4} = 5000\ N = 5\,KN$$

Since the load 'w' will by to tilt a bracket in the clockwise direction about the lower edge, therefore bolts will be subjected to tensile load due to turning moment. Therefore upper side bolts no. 3 & 4 will be will be maximum loaded.

$\therefore$Maximum tensile load carried by bolts (upper)

$$W_t = \frac{w\times L\times L_2}{2\left[(L_1)^2+(L_2)^2\right]} = \frac{20\times500\times400}{2\left[50^2+400^2\right]}$$

$$=12.30\ KN$$

since the bolts are subjected to shear load as well as tensile load, therefore equivalent tensile load.

$$Wt_e = \frac{1}{2}\left[\sqrt{(W_t)^2+4(W_c)^2}+Wt\right]$$

$$=\frac{1}{2}\left[\sqrt{(12.3)^2+4\,(5)^2}+12.3\right] \qquad (02)$$

$$=\frac{1}{2}(15.85+12.3)=14.07KN$$

size of the bolt : dc = core diameter for. The equivalent tensile load Wt_e

$$Wt_e = \frac{\pi}{4}dc^2\times90 \qquad (2\text{ marks})$$

$$dc = 14.42\ mm$$

for course series and core diameter 14.42 take size of bolt as M 18. (02)

c) Comparison between rigid and flexible coupling. (01 mark each)

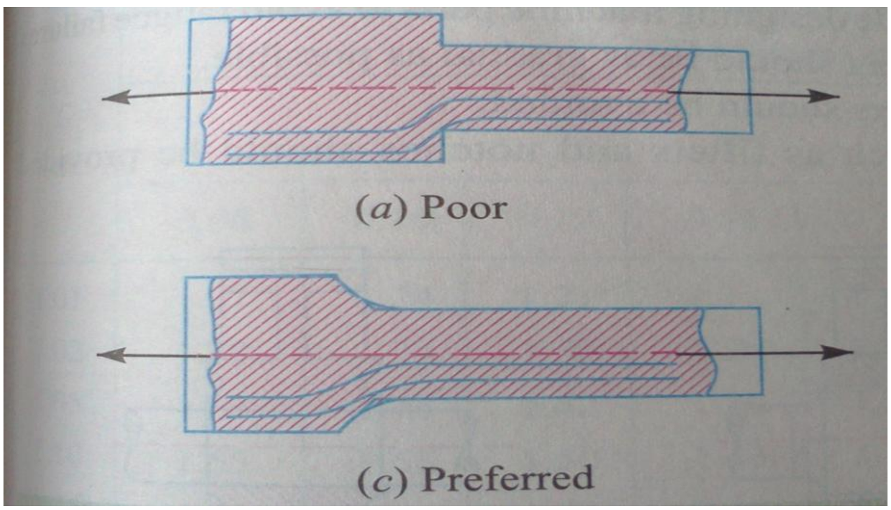
	Rigid coupling	Flexible coupling
Purpose	Connect two shaft without misalignment	Connect shafts with certain misalignment.
Alignment	Perfect alignment in lateral & angular direction is required	Both angular as well as lateral misalignment is allowed.
Deflection	No deflection takes place	Deflection takes place.
Cost	Less	More

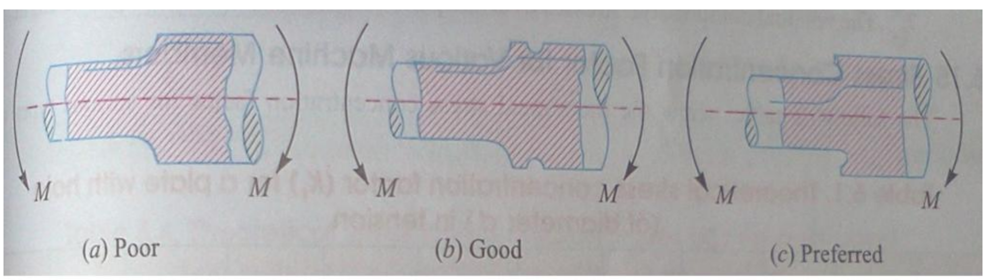
d) Stress concentration : Wherever a machine component changes the shape of it cross section the simple stress distribution no longer holds good and the nearby discontinuation is different. This irregularly in the stress distribution caused by abrupt changes of form is called as stress concentration. (02 marks)

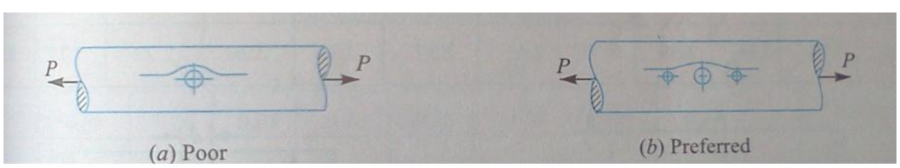
Remedial measures : whenever there is change in cross section such as shoulders, holes, threads, then stress concentration occurs.

To minimize the stress concentration, stress flow lines should maintain equal. Spacing as for possible.

1) A shaft with change in cross section as shown in fig (1), to reduce stress concentration fillets are provided.
2) Sometimes it is not possible to provide fillet, then notches may be cut as cylindrical part as shown in fig (2)
3) In case of holes in shaft, to reduce the stress concentration, small holes are drilled as bolts sides as shown is fig (3).
(02 marks)







e) P = 30 kN = 30 x 10^3 N, N = 300 rpm, w = 1000 N.

$L = 3\, m \quad \tau = 42\, MPa = 42\, N/mm^2$

$6 = 56\, MPa = 56\, N/mm^2$

for a suddenly applied load K_t = Km = 25.

1) Size of shaft :- d = dia. of shaft, mm

Torque transmitted by shaft

$$T = \frac{P \times 60}{2\pi\, N} = \frac{3\phi \times 10^3 \times 6\phi}{2\pi \times 3\phi} = 955\, N - m$$

$$= 955 \times 10^3\, N - mm$$

Maximum bending moment of a simply supported shaft carrying a central load

$$M = \frac{W \times L}{4} = \frac{1000 \times 3}{4}$$

$$= 750\ N - m = 750 \times 10^3\, N - mm$$

we known, Equivalent Twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(955 \times 10^3)^2 + (750 \times 10^3)^2}$$

$$= 1214 \times 10^3\, N - mm$$

Also equivalent twisting moment (T_e)

$$1214 \times 10^3 = \frac{\pi}{16} \tau \times d^3$$

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

$$\therefore d^3 = \frac{1214 \times 10^3 \times 16}{\pi \times 42}$$

$$d = 52.59\, mm$$

we know, equivalent bending moment

$$M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right]$$

$$= \frac{1}{2}\left[M + T_e\right] = \frac{1}{2}\left[750 \times 10^3 + 1214 \times 10^3\right]$$

$$= 982 \times 10^3$$

Also, equivalent bending moment (M_e)

$$982 \times 10^3 = \frac{\pi}{32} \times 6 \times d^3 = \frac{\pi}{32} \times 56 \times d^3$$

$$\therefore d^3 = \frac{982 \times 10^3 \times 32}{56 \times \pi} = 178.6 \times 10^3$$

$$d = 56.08\, mm$$

Taking larger value of the two values

$d = 56.08 \; say \; 58 mm$

(02 marks)

2) If the same shaft is subjected to suddenly applied, then diameter of shaft 'd' can be calculated as below.

We know that equivalent twisting moment

$$T_e = \sqrt{(Km \times M)^2 + (Kt \times T)^2}$$

$$= \sqrt{(2.5 \times 750)^2 + (2.5 \times 955 \times 10^3)^2}$$

$$= 2.5 \times 10^3 \sqrt{(750)^2 + (955)^2}$$

$$= 3035 \times 10^3 \; N - mm$$

Also, equivalent twisting moment (Te)

$$3035 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 42 \times d^3$$

$$\therefore d^3 = 368 \times 10^3$$

$$\therefore d = 71.35 \; mm$$

we know that equivalent bending moment.

$$Me = \frac{1}{2}\left[Km \times M + \sqrt{(Km \times M)^2 + (Kt \times T)^2} \right]$$

$$= \frac{1}{2}\left[Km \times M + T_e \right]$$

$$= \frac{1}{2}\left[2.5 \times 750 \times 10^3 + 3035 \times 10^3 \right]$$

$$= 2455 \times 10^3 \; N - mm$$

Also equivalent bending moment (Me)

$$2455 \times 10^3 = \frac{\pi}{32} \times 6_b \times d^3 = \frac{\pi}{32} \times 56 \times d^3$$

$$d^3 = 446.5 \times 10^3$$

$$d = 76.1$$

value larger than above two values

$$d = 76.1 \quad say \; 78 mm$$

This diameter is larger than the diameter under the normal loading.

(02 marks)

f) Theories of failure

1) Maximum principal or normal stress theory (Rankin's)
2) Maximum shear stress theory (Guest's Theory)
3) Maximum principal or normal strain theory (Saint verant theory)
4) Maximum strain energy theory (Haigh's Theory) (01 mark each)

-----The End-----