11819

17610

4 Hours / 100 Marks

Seat No. | | | | | | | | |

Instructions :
*(1) **All** questions are **compulsory**.*
*(2) Answer **each** next main question on a **new** page.*
*(3) Illustrate your answers with neat sketches **wherever** necessary.*
*(4) Figures to the **right** indicate **full** marks.*
*(5) Assume suitable data, if **necessary**.*
*(6) Use of Non-programmable Electronic Pocket Calculator is **permissible**.*
*(7) Mobile Phone, Pager and any other Electronic Communication devices are **not permissible** in Examination Hall.*

Marks

1. a) Attempt **any three** : **12**

i) State maximum principal shear stress theory and maximum shear stress theory.

ii) Define lever w.r. to (i) M.A. = 1 (ii) M.A. < 1 (iii) M.A. > 1. Define leverage.

iii) What is endurance limit ? Define fatigue failure.

iv) Explain :

i) Transverse shear stress

ii) Torsional shear stress with neat sketch.

b) Attempt **any one** : **6**

i) Define stress concentration with neat sketch. Explain with figures only the several ways of reducing the stress concentration in shafts and other cylindrical members with shoulders, holes and threads respectively.

ii) Design a shaft transmitting power of 60 KW at 600 rpm as shown in Fig. No. 1. The shaft is hollow having ratio of inner to outer diameter 0.5. The shaft carries pulley 'C' as shown. The angle of contact between belt and pulley is 180°. The belts are vertical as shown. The diameter and weight of pulley are 250 mm and 700 N respectively and the ratio of tight-side to slam side tension is 3. Take $K_m = 2$, $K_t = 1.5$. Design shaft if shear stress in material is not to exceed 52 N/mm^2.

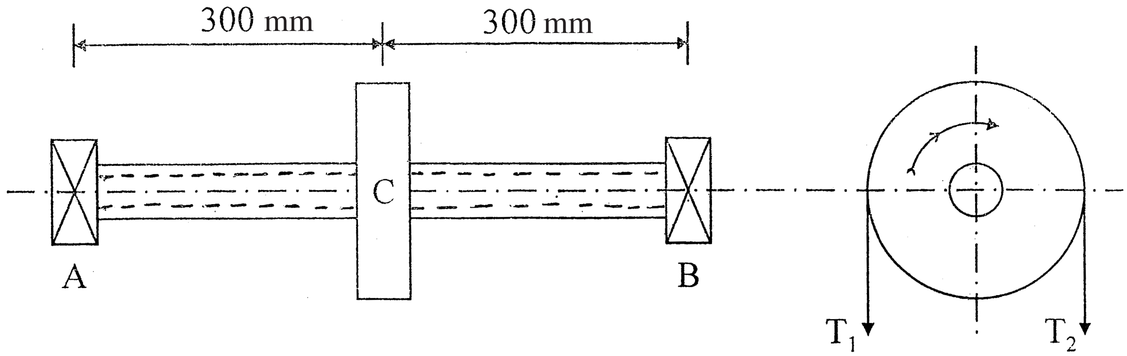


Fig. 1. No. 1 (b) (ii)

P.T.O.

Marks

2. Attempt **any two** : **16**

a) Write down the design procedure of power screw for nut and screw with diagram.

b) Explain design procedure of a flange coupling.

c) i) Define factor of safety for brittle material and ductile material. **4**

ii) Design a rectangle key for shaft of 50 mm diameter. The permissible stresses for key material are 40 N/mm^2 in shear and 70 N/mm^2 in crushing. **4**

3. Attempt **any four** : **16**

a) What is the effect of keyways on strength of shaft. Write the expression.

b) Write advantages and disadvantages of square thread over 'V' threads (two each).

c) Suggest suitable material for the following machine parts (i) Crank shaft (ii) Helical spring (iii) Bushes for Knuckle pin (iv) Lathe bed.

d) Suggest suitable couplings in the following cases :

i) Shaft having perfect alignment

ii) Shaft having both lateral and angular misalignment.

e) Explain the gear tooth failure modes :

i) Scoring ii) Pitting.

4. a) Attempt **any three** : **12**

i) Define the terms :

i) Solid length

ii) Spring index

iii) Free length

iv) Spring rate, w.r. to helical compression spring.

ii) Write any four ergonomic considerations which makes the job comfortable.

iii) Draw the graph of Wahl's stress factor Vs spring index for helical compression spring and state the effect of curvature of the coil on the stress distribution.

iv) Write only the equations for the conditions :

i) Self-locking

ii) Overhauling of a power screw and explain the terms used.

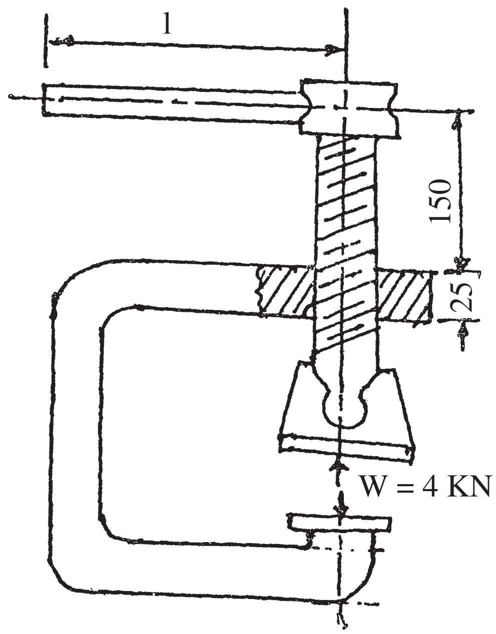
b) Attempt **any one** : **6**

i) State the strength equations for double parallel fillet weld and double transverse fillet weld with sketches.

ii) The C-clamp as shown in Fig. No. 2, has trapezoidal threads of 12 mm outside diameter and 2 mm pitch. The coefficient of friction for screw thread is 0.15 and for the collar is 0.28. The mean radius of the collar is 8 mm. If the force exerted by the operator at the end of handle is 80 N. Find :

i) The length of handle

ii) The bearing pressure on the threads.



All dimensions in mm

Fig. No. 2/No. 4 (b) (ii)

5. Attempt **any two** : **16**

a) A Knuckle joint is required to withstand a tensile load of 25 KN. Design the joint if, the permissible stresses are $f_t = 56$ N/mm^2, $f_s = 40$ N/mm^2, $f_c = 70$ N/mm^2.

b) i) Define the following terms related to bearing

a) Bearing modulus

b) Critical pressure

ii) State any four advantages and disadvantages of welded joint over screwed joint.

c) Design a helical compression spring for a maximum load of 1200 N and deflection 30 mm using the value of spring index as 5. The maximum permissible shear stress for spring wire is 420 MPa and modulus of rigidity is 84 KN/mm^2

Take Wahl's stress factor

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

Where C = spring index

Marks

6. Attempt **any four** : **16**

a) Write Lewis equation for strength of a gear tooth with usual notations.

b) A wall bracket as shown in Fig. No. 3 is fixed to a wall by means of four bolts. Find the size of the bolts. The safe stress in tension for the bolt may be assumed as 70 N/mm^2.

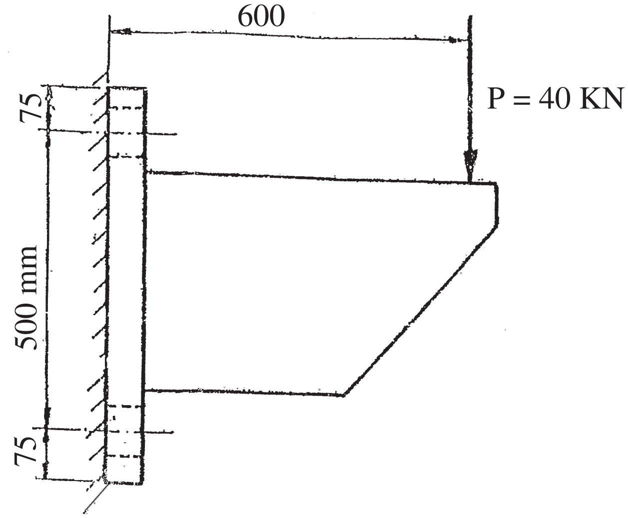


Fig. No. 3/No. 6 (b)

c) Compare sliding contact bearing and roller contact bearing on the basis of size, life, coefficient of friction and housing diameter.

d) Explain the methods of obtaining bolts of uniform strength.

e) Write down procedure for selection of bearing from manufacturer’s catalogue.

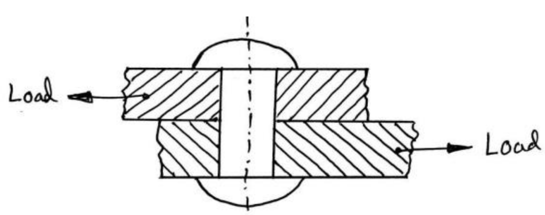
WINTER– 18 EXAMINATION

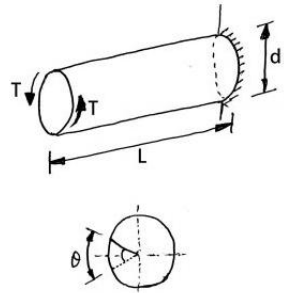
Subject Name: Design of Machine Elements **Model Answer** **Subject Code: 17610**

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. No.	Sub Q. N.	Answer	Marking Scheme
1	a)	Attempt any **THREE**	12 marks
	i)	**State maximum principal stress theory and maximum shear stress theory.** **Maximum Principal Stress Theory:** This theory states that failure occurs when the maximum principal stress from a combination of stresses equals or exceeds the value obtained for the direct stress at yielding in a simple tension test. **Maximum Shear Stress Theory:** This theory states that failure occurs when the maximum shear stress from a combination of stresses equals or exceeds the value obtained for the shear stress at yielding in the simple tensile test.	02 marks each
	ii)	**Define lever w.r.to (i) M.A.=1 (ii) M.A.<1 (iii) M.A.>1. Define leverage** **Mechanical Advantage (M.A.)=1** A rigid rod or bar pivoted at a point and capable for turning about the pivot point called fulcrum. In this case the length of the effort arm and the length of the load arm are **equal.** **length of the effort arm= length of the load arm**	01 marks each

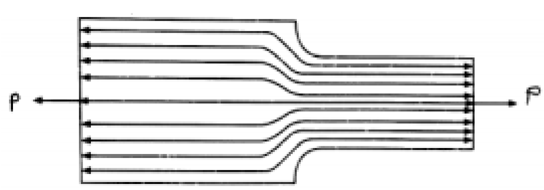
Q. No.	Sub Q. N.	Answer	Marking Scheme
		Mechanical Advantage (M.A.)<1 A rigid rod or bar pivoted at a point and capable for turning about the pivot point called fulcrum. In this case the length of the effort arm is less than the load arm. **length of the effort arm< length of the load arm** **Mechanical Advantage (M.A.)>1** A rigid rod or bar pivoted at a point and capable for turning about the pivot point called fulcrum. In this case the length of the effort arm is greater than the load arm. **length of the effort arm> length of the load arm** **Leverage**: The ratio of length of effort arm to the length of load arm is called leverage.	
	iii)	**What is endurance limit? Define fatigue failure.** **Endurance Limit:** It is defined as the maximum value of completely reversed bending stress that a material can withstand for a finite number of cycles (i.e. 10^7 cycles) without a fatigue failure. **Fatigue failure:** Fatigue failure refers to the fracturing of any given material due to the progressive cracking of its brittle surface under applied stresses of an alternating or cyclic nature.	02 marks each
	iv)	**Explain:** **i) Transverse shear stress:** When a mechanical component is subjected to two equal and opposite forces acting tangentially across the resisting area resulting in shearing off of the section, the stress induced in such a case is known as transverse shear stress.  **ii) Torsional shear stress with neat sketch:** When a machine component is subjected to the action of two equal and opposite couple acting in parallel plane (torque or twisting) then the machine component is subjected to torsion and the stress induced in such a case is known torsional shear stress.	02 marks each



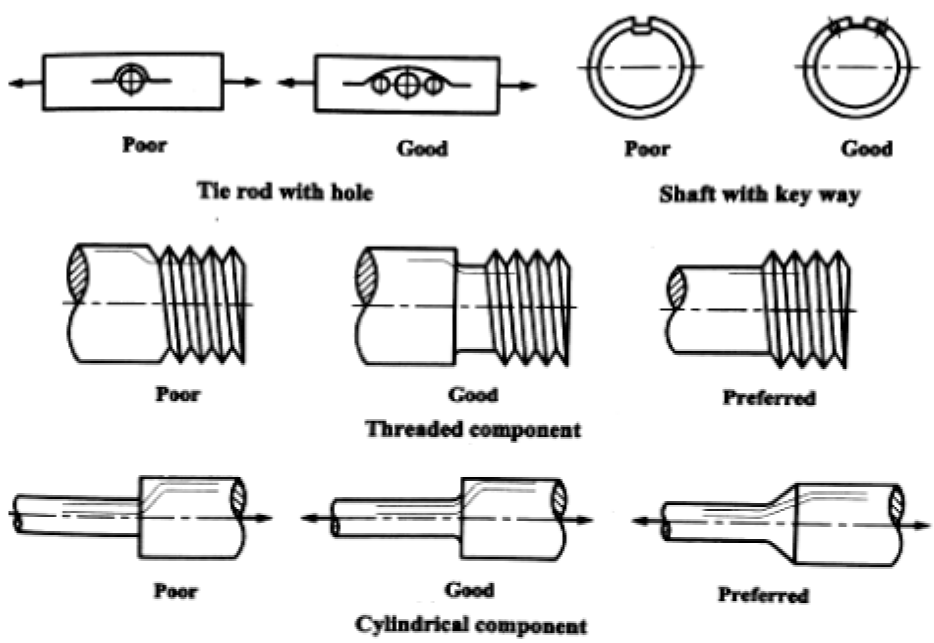
b) **Attempt any ONE:** — 06 marks

i) **Define stress concentration with neat sketch. Explain with figures only the several ways of reducing the stress concentration in shafts and other cylindrical members with shoulders, holes and threads respectively.**

Stress concentration: The stresses induced in the neighborhood of the discontinuities like key ways, threaded grooves, holes, notches are much higher than the stresses in the other parts of the stressed component. This concentration of high stresses due to discontinuities and abrupt changes in cross section is called stress concentration. **(01 mark for definition 01 mark for figure)**

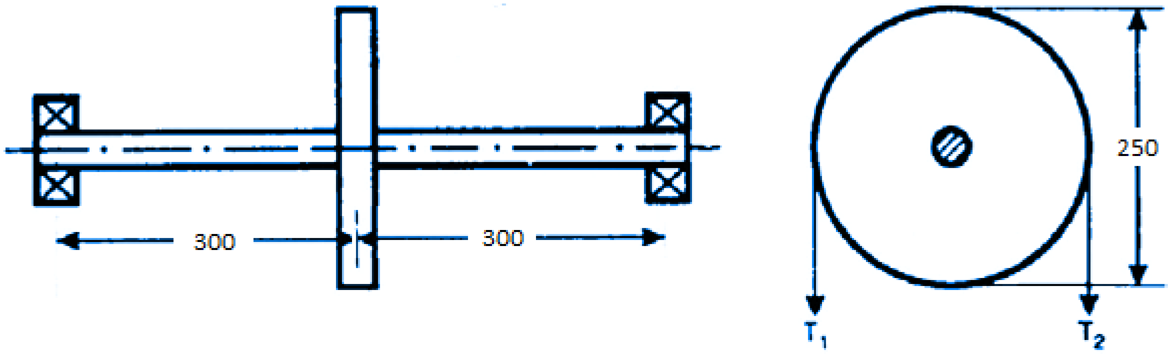


(04 marks for the figures of explaining stress reduction in shafts, cylindrical members with shoulders, holes, threads)



ii) **Design a shaft transmitting power of 60kW at 600 rpm as shown in Fig. No.1.** — 06 marks

(P- 01 mark, T -01 mark, T1,T2 -01 mark, M -01 mark and d-02 marks)



Given P = 60 kW = 60 x 10^3 W

N = 600 rpm

K = di/do = 1.5

Ө = 180°

D_p = 250 mm

W_p = 700 N

T_1/T_2 = 3

K_m = 2

K_t =21.5

T = 52 N/ mm^2

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

$$60\ x\ 10^3 = \frac{2\pi x 600 x T}{60}$$

$$T = 954 x 10^3 N - mm$$

$$T = (T_1 - T_2)R$$

$$954\ x\ 10^3 = (3\ T_2 - T_2)125$$

$$T_2 = 3816\ N$$

$$T_1 = 3816\ x\ 3 = 11448\ N$$

Maximum bending moment occurs at C, considering the simply supported beam

Bending moment, M = WL/4

$$M = \frac{(T_1 + T_2 + W_p) x\ 600}{4}$$

$$= \frac{(11448 + 3814 + 700) x 600}{4}$$

Put the value of T_2=3816

$$= 2394300\ N - mm$$

M=2394600 N-mm

The equivalent twisting moment,

$$T_e = \sqrt{(k_m x M)^2 + (k_t x T)^2}$$

$$= \sqrt{(2\ x\ 2394300)^2 + (1.5\ x\ 954\ x\ 10^3)^2}$$

$$= 4997415\ N - mm$$

=4998419 N-mm

$$T_e = \frac{\pi}{16} d^3 \tau_s [1 - k^4]$$

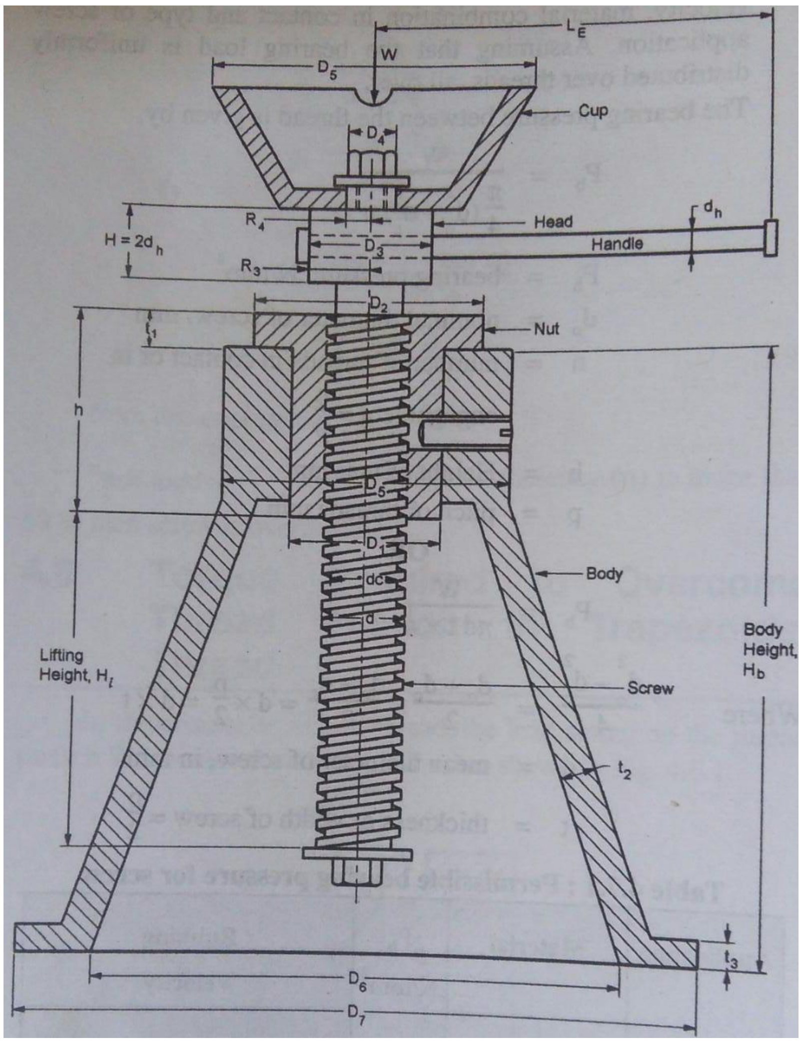
$$4997415 = \frac{\pi}{16} x\, d^3 x 52\, x\, [1 - 0.5^4]$$

$$d = 80.52\ mm$$

$$d \cong 82\ mm$$

2 **Attempt any TWO** — 16 marks

a) **Write down the design procedure of power screw for nut and screw with diagram. (02marks for diagram, 03 marks for design procedure of nut and 03 marks for that of the screw)** — 08 marks



1) Design of screw

i) Find the core diameter of screw (d_c) by considering the screw under pure compression. The direct compressive stress induced in the screw is given by

$$\sigma_c = \frac{W}{A} = \frac{W}{\frac{\pi}{4} d_c^2}$$

$$d_c = \sqrt{\frac{4W}{\pi \sigma_c}}$$

the core diameter d_c can be obtained

ii) Select the standard square thread for screw using the table, of normal series, select d_c, d_o and pitch OR. Take (d_c + 6) = d_c for design

$$d_o = \frac{d_c}{0.84}$$

$$d_c = d_o - p$$

$$main\ diameter, d = d_o - \frac{p}{2} = \frac{d_o - d_c}{2}$$

P = pitch of screw

iii) In addition to direct compressive load, the screw is subjected to twisting moment T_1, so for that the core diameter is increased and appropriately find the torque required to rotate the screw

$$T_1 = W \tan(\propto + \emptyset)\frac{d}{2}$$

a) Shear stress due to torque

$$\tau = \frac{16\,T_1}{\sigma d_c^3}$$

b) Direct compressive stress due to axial load

$$\sigma_c = \frac{4\,W}{\pi d_c^2}$$

Check the screw by,

i) Maximum principal stress theory,

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

ii) Maximum shear stress theory,

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

These stresses $\sigma_{c\ max}$ and τ_{max} should be less than permissible stresses.

2) (a) Design of nut

i) The height of nut 'h' can be found by considering the bearing pressure on Nut

$$P_b = \frac{W}{\frac{\pi}{4}(d_o^2 - d_c^2) x\, n}$$

Where, n is the number of threads in contact with screw spindle

Height of nut = h = n x p

Where p = pitch of the thread

ii) Check for shear stresses is screw and nut threads

$$Shear\ strss\ in\ screw\ \tau_s = \frac{W}{\pi d_c t\ x\ n}$$

$$Shear\ stress\ in\ nut\ \tau_n = \frac{W}{\pi d_o t\ x\ n}$$

For safety of the screw and nut thread

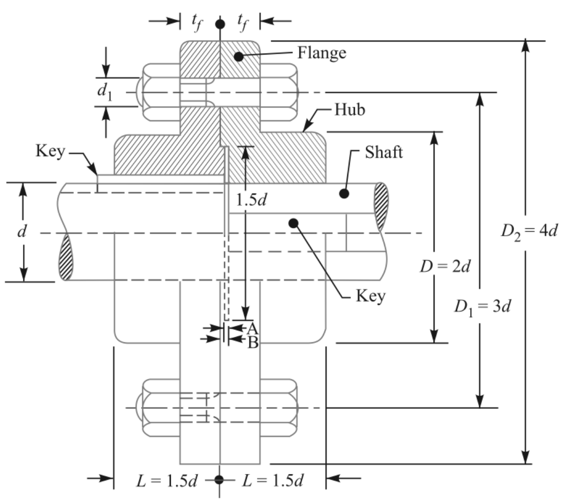
τ_s and $\tau_n \le \tau_d$ (allowable shear stresses)

b) Design of nut collar

To find the diameter of nut collar

Let D1 = outer diameter of Nut in mm

D2 = outer diameter of Nut collar in mm

		t1 = thickness of Nut collar in mm a) Considering tearing of the nut due to tensile stress $$\sigma_{t\,nut} = \frac{W}{\frac{\pi}{4}(D_1^2 - d_o^2)}; find\, D_1$$ b) Considering the crushing of the nut and collar $$\sigma_{c\,nut} = \frac{W}{\frac{\pi}{4}(D_2^2 - D_1^2)}; find\, D_2$$ c) Considering the shearing of the nut and collar $$\tau_{s\,nut} = \frac{W}{\pi D_1 t_1}; find\, t_1$$	
	b)	**Explain design procedure of a flange coupling.** **(02 marks for figure, 01 mark for shaft design, 02 marks for flange design, 02 marks for key design and 01 for bolts design)**  Let d= Diameter of the shafts D= Diameter Of hub D1= Pitch circle diameter D2= Outer diameter Of the flange L= Length of the flange tf=Thickness of the flange dc=Core diameter of bolt do= Outer diameter of bolt n=no of bolts t= thickness of the key w=width of the key l= length of the key **Step 1. To find torque acting on the shaft** $$T = \frac{60P}{2 \times \pi \times N} \times 10^6$$ Using this formula the Toque acting on the shaft is calculated.	8 marks

Step 2. To find diameter of shaft to transmit required torque

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

Use this equation to find the diameter of the shaft.

Step 3. Design of flanges

Emperical Relations

$$D = 2 \times d$$
$$D_1 = 3 \times d$$
$$D_2 = 4 \times d$$
$$l = 1.5 \times d$$
$$t_f = 0.5 \times d$$
$$t_p = 0.25 \times d$$

Check shear stress in hub

$$T = \frac{\pi}{16} \times \tau \times [D^3(1 - k^4)]$$

Find induced stress τ=induced shear stress. This should be less than allowable shear stress

Check Shear stress in the Flange

Force = area resisting Stress

$$\frac{T}{D/2} = \pi \times D \times t_f \times \tau$$

Find induced stress τ=induced shear stress .This should be less than allowable shear stress

Step 4. Design Of Key

Empirical relation

w=t=d/4 {if crushing stress is twice the shear stress}

w=d/4 and t=d/6 {if crushing stress is not twice the shear stress}

length of key = length of flange

check shear stress induced in key

T/(d/2)={w x l} τ

Find induced stress τ=induced shear stress. This should be less than allowable shear stress

check crushing stress induced in key

T/(d/2)={t/2 x l}σ_c

Find induced stress σ_c =induced crushing stress. This should be less than allowable shear stress

Step 5. Design Of Bolts

No of Bolts

n=4 if diameter of shaft is upto d<55 mm

n=6 if diameter is between 55 <d< 150 mm

		n=8 if diameter is above d>150mm. **Shear Failure of bolts** $T/(D1/2) = \{ n \times \pi/4 \times d_c^2 \}$ Dc= ….. $d_o=d_c/0.84$ The bolt size should be rounded to nearest even number..	
	c) i)	**Define factor of safety for brittle materials and ductile materials.** **(02 marks definition of factor of safety and 01 each for that of ductile and brittle materials)** While designing any mechanical component always there are certain areas of uncertainties such as variation and non uniformity in the mechanical strength etc. Hence in order to prevent failure of the component, designer assuming a value of design stress, which is very less as compared to the yield stress or ultimate stress. So factor of safety is defined as maximum stress to working stress or design stress. 1. For ductile materials: The factor of safety is defined as the ratio of yield point stress to design stress. $$Factor\ of\ safety\ (Nf) = \frac{Yield\ Stress}{Working\ of\ design\ stress}$$ 2. For brittle materials: The factor of safety is defined as the ratio of ultimate stress to design stress. $$Factor\ of\ safety\ (Nf) = \frac{Ultimate\ Stress}{Working\ of\ design\ stress}$$	04 marks
	ii)	**Design a rectangular key for shaft of 50 mm diameter. The permissible stresses for key material are 40 N/mm² in shear and 70 N/mm² in crushing.** **(for Torque 01 mark and 03 marks for key design)** Diameter of shaft, d = 50 mm τ_s =40 N/mm^2 σ_c = 70 N/mm^2 Torque transmitted, T = $\frac{\pi}{16}\tau_s d^3$ = $\frac{\pi}{16}\ x\ 40\ x\ (50)^3$ =981747.704 N-mm Given rectangular key, hence Thickness of the key, t = d/6 = 2w/3 t= 50/6 = 8.33 mm Considering crushing failure of key $$T = l\ x\ \frac{t}{2}\ x\ \sigma_{ck}\ x\ \frac{d}{2}$$	04 marks

$$981747.704 = l\ x\ \frac{8.33}{2}\ x\ 70\ x\ \frac{50}{2}$$

$$l = 134.69 \cong 135\ mm$$

$$\therefore the\ length\ of\ the\ key = 135\ mm$$

Considering shearing of the key

$$T = lwr\ x\ \frac{d}{2}$$

$$981747.704 = 135\ x\ w\ 40\ x\ \frac{50}{2}$$

$$w = 7.27 \cong 7.5\ mm$$

OR

put w=d/4=50/4=12.5mm

$$T = lwr\ x\ \frac{d}{2}$$

Find l, l=78.53mm

Taking the higher value ,l=135mm

3	a	Attempt any FOUR (Explanation -3, expression-1)	04

What is the effect of keyways on the strength of shaft. Write the expression

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 i.e. 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(w/d)-1.1(h/d)

Where,

e = Shaft strength factor. It is the ratio of the strength of the shaft with keyway to the strength of the same shaft without keyway

w = Width of keyway,

d = Diameter of shaft, and

h = Depth of keyway = Thickness of key / 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 is given by the following relation

	b	Write advantages and disadvantages of square thread over V thread(Two each) Advantages(2 marks) 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. Disadvantages (2 marks) i)The disadvantages are that most are not very efficient. ii) Due to the low efficiency they cannot be used in continuous power transmission applications. iii)They also have a high degree of friction on the threads, which can wear the threads out quickly.	04
	c	**Suggest suitable material for the following machine parts (1 mark each)** i) **Crankshaft** is usually made by steel. Generally medium-carbon steel alloys are composed of iron and contain a small percentage of carbon (0.25% to 0.45%), along with combinations of several alloying elements, the mix of which. ii) **Helical spring**-The most popular alloys include high-carbon (such as the music wire used for guitar strings), oil-tempered low-carbon, chrome silicon, chrome vanadium, and stainless steel. Other metals that are sometimes used to make springs are beryllium copper alloy, phosphor bronze, and titanium. iii) **Bushes for Knuckle pin-**carbon steel, alloy steel, stainless steel and aluminum. Finishes include zinc, nickel, mechanical plating, black oxide, passivated and anodized. iv) **Lathe bed**-Cast Iron.	04

	d	**Suggest suitable coupling in the following cases** i)shaft having perfect alignment- Rigid coupling-1) Muff or sleeve coupling 2) Split muff or clamp coupling 3) Rigid flange- Protected type, Unprotected type, Marine type flange coupling ii)shaft having both lateral and angular misalignment-Flexible coupling-1)Bush pin type 2) Oldham's coupling 3) Universal coupling	04
	e	**Explain the gear tooth failure modes (2 marks each)** i) SCORING: Scoring is due to combination of two distinct activities: First, lubrication failure in the contact region and second, establishment of metal to metal contact. Later on, welding and tearing action resulting from metallic contact removes the metal rapidly and continuously so far the load, speed and oil temperature remain at the same level. The scoring is classified into initial, moderate and destructive. (i) INITIAL SCORING Initial scoring occurs at the high spots left by previous machining. Lubrication failure at these spots leads to initial scoring or scuffing. Once these high spots are removed, the stress comes down as the load is distributed over a larger area. The scoring will then stop if the load, speed and temperature of oil remain unchanged or reduced. Initial scoring is non progressive and has corrective action associated with it. (ii) Initial scoring MODERATE SCORING After initial scoring if the load, speed or oil temperature increases, the scoring will Spread over to a larger area. The Scoring progresses at tolerable rate. This is called moderate scoring. DESTRUCTIVE SCORING After the initial scoring, if the load, speed or oil temperature increases appreciably, then severe scoring sets in with heavy metal torn regions spreading quickly throughout. Scoring is normally predominant over the pitch line region since elastohydrodynamic lubrication is	04

the least at that region. In dry running surfaces may seize.

ii)Pitting: This is a major cause of gear failure accounting for nearly 60% of the gear failures. Pitting is the formation of craters on the gear tooth surface. These craters are formed due to the high amount of compressive contact stresses in the gear surface occurring during transmission of the torque or in simple terms due to compressive fatigue on the gear tooth surface. There are two types of Pitting. They are

a) Micro Pitting: These are basically formed due to Inherent Errors in the gears Presence of water in the lubricant that is lubricating the gears . Wrong viscosity selection of the lubricant used. Visually, micro pitting is not so clearly visible at the first go. One has to study the surface of the gear tooth to identify the micro pitting. They appear as very small dots which one can feel when he runs his finger over the gear tooth. This sort of pitting normally tends to make the gear useless and damages the whole gear system.

4 A **Attempt any THREE**

(i) a)i) Define the terms(01 mark each) 04

(i)Solid length. When the compression spring is compressed until the coils

come in contact with each other, then the spring is said to be solid. The solid

length of a spring is the product of total number of coils and the diameter of

the wire. Mathematically,

Solid length of the spring,

LS = n'.d

where n' = Total number of coils, and

d = Diameter of the wire.

ii) Spring index. The spring index is defined as the ratio of

the mean diameter of the coil to the diameter of the

wire. Mathematically,

Spring index, C = D / d

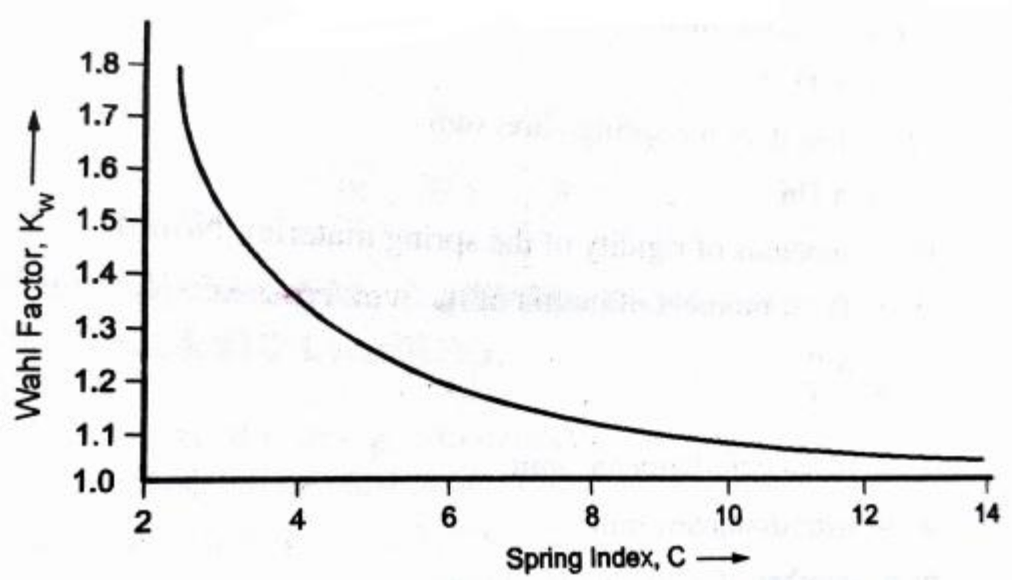
where D = Mean diameter of the coil, and

d = Diameter of the wire.

iii) Free length. The free length of a compression spring, is the length of the spring in the free or unloaded condition. It is equal to the solid length plus the maximum deflection or compression of the spring and the clearance between the adjacent coils (when fully compressed).

		iv) Spring rate. The spring rate (or stiffness or spring constant) is defined as the load required per unit deflection of the spring. Mathematically, Spring rate, $k = W / \delta$ where W = Load, and δ = Deflection of the spring.	
	ii	**ii)Ergonomics considerations any four** i)For assembly jobs, material should be placed in a position such that the worker's strongest muscles do most of the work. ii)For detailed work which involves close inspection of the materials, the workbench should be lower than for work which is heavy. iii)Hand tools that cause discomfort or injury should be modified or replaced. Workers are often the best source of ideas on ways to improve a tool to make using it more comfortable. For example, pliers can be either straight or bent, depending on the need. iv)A task should not require workers to stay in awkward positions, such as reaching, bending, or hunching over for long periods of time. v)Workers need to be trained in proper lifting techniques. A well designed job should minimize how far and how often workers have to lift. vi)Standing work should be minimized, since it is often less tiring to do a job sitting than standing. vii)Job assignments should be rotated to minimize the amount of time a worker spends doing a highly repetitive task, since repetitive work requires using the same muscles again and again and is usually very boring. viii)Workers and equipment should be positioned so that workers can perform their jobs with their upper arms at their sides and with their wrists straight.	04

iii Iii)Graph of Wahls stress factor- 2 marks , explanation- 2 marks 04



Curvature Effect

The curvature of the wire increases the stress on the inside of the spring, an

effect very similar to stress concentration but due to shifting of the neutral axis away from the geometric center, as could be observed in curved beams. Consequently the stress on the inside surface of the wire of the spring, increases but decreases it only slightly on the outside. The curvature stress is highly localized that it is very important only fatigue if is present.

This effect can be neglected for static loading, because local yielding with the first

application of the load will relieve it. The combined effect of direct shear and curvature

correction is accounted by Wahl's correction factor and is given as:

Whal's correction factor

The combined effect of direct shear and curvature correction is accounted by Wahl's

correction factor and is given as:

kw=4C- 1 /4C-4+ 0.615/C

iv **iv) write only the equations for the conditions (2 marks each)** 04

i)Self-locking:

T=Wdm/2xtan(φ−α)

When φ is greater than or equal to α, a positive torque is required to lower the load.

Under this condition, the load will not turn the screw and will not descend on its own unless an effort P is applied.

Screw will be self-locking if the co-efficient of friction is equal to or greater than the tangent of the helix angle, the screw is said to be self-locking.

Ii)Over hauling:

The torque required to lower the load can be given by the equation,

$T = Wdm/2 x \tan(\varphi - \alpha)$

It can be seen when $\varphi < \alpha$ the torque required to lower the load is negative.

It indicates a condition that no force is required to lower the load. The load itself will begin to turn the screw and descend down, unless a restraining torque is applied.

The condition is called overhauling of the screw. This condition is also called back driving of screw.

b **Attempt any ONE (strength equation – 2 marks each and fig. 1 each)** 06

i i)Strength equation of double parallel fillet weld= throat area x allowable shear stress

P= 2x 0.707x S x lx τ

=1.414 x S x lx τ

Where S= size or leg of the weld

l= legngth of the weld

τ = shear stress

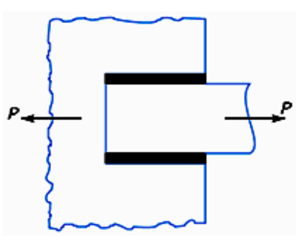


Fig. double parallel fillet weld

Strength equation of double transverse fillet weld

P =throat area x allowable tensile stress

P= 2x0.707x S x l x σ_t

=1.414 x S x lx σ_t

Where S= size or leg of the weld

l= legngth of the weld

σ_t = tensile stress

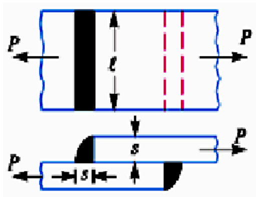


Fig.double transverse fillet weld

ii

1) **Length of handle (4 marks)**

Given data μ=0.15

Mean diameter of the screw =d= do-p/2= 12-2/2=11mm

Tan α= p/πd=2/πx11=0.058

Angle for trapezoidal thread is 2 = 30^0 or = 15^0 therefore virtual coefficient of friction, μ1= tanΦ_1= μ/cosβ=0.12/cos15=0.124 Put μ=0.15

μ1= tanΦ_1= μ/cosβ=0.15/cos15=0.155

Torque required to overcome friction at the screw,

T_1=Pxd/2=W tan (α + $\Phi_{1)}$ d/2 = 4000(0.058+0.124/(1-0.058x0.124)=4033Nmm

T_1=Pxd/2=W tan (α + $\Phi_{1)}$ d/2 = 4000(0.058+0.155/(1-0.058x0.155)d/2=4733Nmm

Assuming uniform wear the torque required to overcome friction at the coller,

T_2 = μ_2WR =0.28X 4000X 8 =8960 N-mm

Torque required at he handle , T = T1+ T2 = 4033+8960 =12993 N-mm

Torque required at he handle , T = T1+ T2 = 4733+8960 =13693 N-mm

Torque required at he handle = P_1x l =12993=80Xl

Torque required at he handle = P_1x l =13693=80Xl

=162.4125 mm. Ans.

=171 mm. Ans.

ii) the bearing pressure on the threads.(2 marks)

Height of the nut, h= nxp= 25

06

No. of threads in contact,

n =h/p =25/2 12.5

and thickness of threads t = p/2 = 2/2 =1 mm

The bearing pressure on the threads =P_b = W/π dtn =4000/3.142 x 11x 1 x12.5 **= 9.26 N/mm^2 Ans.**

5 a

Attempt any Two

(2X 8)

Given Data:

P = 25 KN = 25 X 10^3 N , f_t = 56 N/mm^2 , f_S = 40 N/mm^2 f_c =70 N/mm^2

Design a knuckle joint

STEP 1: Failure in rod in tension

The rod may fail in tension due to tensile load W

$$ft = \frac{p}{\frac{\pi}{4} x\, d^2}, \qquad 56 = \frac{25\ x 10^3}{\frac{\pi}{4} x\, d^2}, \qquad d^2=568.4 , \qquad d=23.84\approx 24 \text{ mm}$$

STEP II: Double Shearing in Pin:

$$fs = \frac{p}{2x\frac{\pi}{4}\, x\, dp^2}, 40 = \frac{25\ x10^3}{2x\frac{\pi}{4}\, x\, dp^2}, dp^2=397.88 \quad dp=19.94\approx 20 \text{ mm}$$ 1

STEP III: Thickness of single eye (t) :

Crushing stress induced in single eye 1

$$fc = \frac{p}{dpx\, t} = 70 = \frac{25\ x10^3}{20x\ t}, \quad t = 17.85 \text{ mm} \approx 18 \text{ mm}$$

STEP IV: Outside diameter of single eye (d_{oe})

$$ft = \frac{p}{(doe-dp)x\, t}, 56 = \frac{25\ x10^3}{(doe-20)x\, 18}, \quad doe = 44.80 \text{ mm} \approx 45 \text{ mm}$$ 1

2

$$fs = \frac{p}{2(doe-dp)x\, t/2}, \quad 40= \frac{25\ x10^3}{(doe-20)x\, 18}, \quad doe = 54.72 \text{ mm} \approx 55 \text{ mm}$$

The largest value of outside diameter of single eye is selected (doe)

(doe) =55 mm 1

Design of fork :

Step V :Thickness of double eye t_1 =

Double eye may fail in crushing due to tensile load

$$fc = \frac{p}{2\ x\ dpx\ t1} = 70 = \frac{25\ x10^3}{2\ x\ 20x\ t1}, \quad t1 = 8.9 \text{ mm} \approx 10 \text{ mm}$$

2

Step VI: Outside diameter of single eye (d_{of})

$$ft = \frac{p}{(dof-dp)x\ 2\ x\ t1}, \quad 56 = \frac{25\ x10^3}{(dof-20)x\ 2\ x\ 10}, \quad dof = 44.32 \text{ mm} \approx 45 \text{ mm}$$

		$fs = \frac{p}{2(dof-dp)x\,t1}$, $40 = \frac{25\,x10^{3}}{(dof-20)x\,10}$, doe = 82.5 mm ≈ 84 mm The largest value of outside diameter of double eye is selected (dof)………… (dof) =84 mm **Note: dimensions may vary according selection of final size. So give full credit to the correct method/ alternate method**	
	B i	**a)Bearing Modulus:** It is a ratio of (ZN)and P **. dimensionless Number** **Bearing modulus =**$\frac{ZN}{P}$ The factor *ZN/p* helps to predict the performance of a bearing. **b) 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, $$p = \frac{ZN}{4.75\times10^{6}}\left(\frac{d}{c}\right)^{2}\left(\frac{l}{d+l}\right)\text{N/mm}^{2}$$	02 02
	ii	**four advantages & disadvantages of welded joints over screwed joint** **Advantages:** 1)Permanent Joint 2) Longer life than screwed joint. 3)Welded joints are tight and leak proof. 4) More strength and rigidity. **Disadvantages:** 1)It is very difficult to inspect. 2) It can not be used to joint dissimilar material as in case of screwed joints. 3) Welded joints required skilled manpower 4)Assembly and disassembly not simple in welded joint . 5)Distortion and stresses may developed in due to heating.	½ EACH 2 M ½ EACH 2 M

c

Design of Helical compression spring:

a) Design of spring

Given Data: W=1200 N , $\delta = 30\ mm$, τ =420MPa , G= 84 x 10 ^{3} N/mm^2, C= 5

Wahl's stress correction factor is

K =$\frac{4C-1}{4C-4} + \frac{0.615}{C}$, K=$\frac{4\ X\ 5-1}{4\ X\ 5-4} + \frac{0.615}{5}$,K=1.3105 …………………..

1) Mean dia of spring coil

$\tau = KX\frac{8\ W\ D}{\pi\ d^3}$, $420 = 1.3105X\frac{8\ X\ 1200\ X\ 5}{\pi\ d^2}$,**d=6.9 mm..** 1

2) Mean dia of spring coil D_m =

C= Dm/d , Dm = 5 x 6.9 = 34.5 mm , **Dm = 34.5 mm…………………** 1

3)Number of turns:

$\delta = \frac{8\ W\ D^3\ n}{G\ d^4}$,30$=\frac{8\ X\ 1200\ X\ 34.5^3\ \ x\ n}{84\ x\ 10^3\ x\ 6.9^4}$**n= 14.49 i.e 16 turns**……..

Assuming squared & grounded ends ,total number of turns is given by 1

n' = n +2 = 16+2 =18 ……………………………..

3) Solid Length = Ls= n' x d =18 x 6.9 = **124.2 mm……………….** 1

4)Free Length = Lf = n' x d + $\delta max + 0.15\ \delta max$ 1

Lf = (18 x 6.9 + $30 + 0.15\ x30$ = **158.7 mm……….** 1

5)Pitch of coil = P=$\frac{free\ length}{n'-1}$ = $\frac{158.7}{18-1}$ = **9.33 mm** ……………..

Note: dimensions may vary according selection of final size. So give full credit to the correct method/ alternate method 1

1

6 a **Attempt Any Four**

Lewi's Equation for strength of a gear tooth with notations

$$W_T = \sigma w\ .b.\ Pc\ .y\ =\ \sigma w\ .b.\pi m.\ y$$

EQ: 2M

..(Pc = πm)

W_T = Tangential load acting at the term

σw = Beam strength of the tooth

Meaning of terms – 2 M

b = Width of the gear face

Pc = Circular pitch

m = Module **1/2**

Y is known as Lewis form factor or tooth form factor b

Given Data:

W= 40 x 10^3 N L = 650 mm , L=600mm L1 =75 mm, L2 = 575 mm $\sigma t =$ =70 N/mm^2

1) **Direct Shear load** on each bolt

$Ws = \frac{W}{n} = \frac{40}{4}$ = 10K N

2) **Max Tensile load**on upper bolts at a distance L2=575 mm

Put L=600mm

Max tensile $Load$ $wt = \frac{W\ x\ L\ x\ L2}{2(L1^2 + L2^2)} = \frac{40\ x\ 650\ x\ 575}{2(75^2 + 575^2)}$ =22.23KN…… **1**

=20.52 KN

When bolts are subjected to shear as well as tensile loads ,then equivalent tensile load **1**

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

Wte= ½ (21.95 + [(21.95)^2 + 4 (10)^2]$^{1/2}$ =25.82 KN……………

=24.58 KN…………… **1**

3) **Size of bolts:**

$$\sigma t = \frac{Wte}{\frac{\pi}{4} x\ dc^2},\ 70 = \frac{25.82\ x\ 10^3}{\frac{\pi}{4} x\ dc^2},$$

dc = 21.77 mm , …………………………………. **1**

do= 21.77/0.84 = 25.92≅ 26 mm

Bolt size may be M26 …………………………….

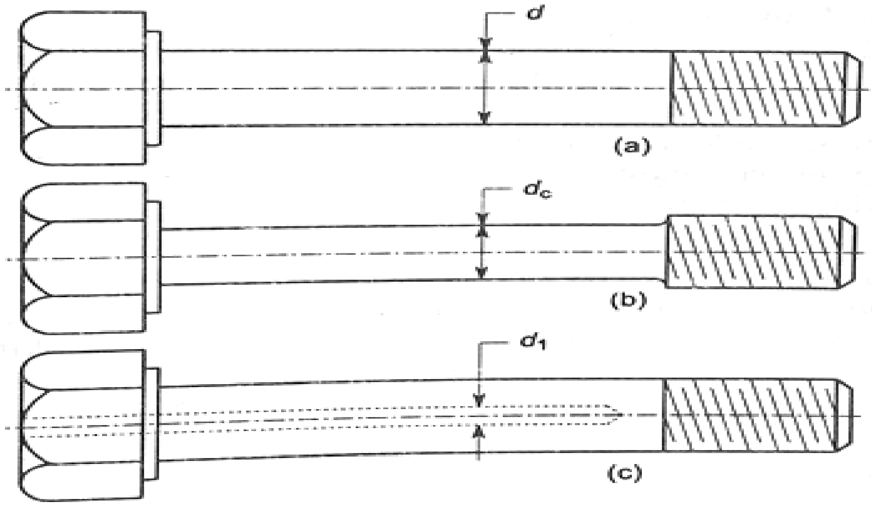
c

Comparison sliding contact bearing and roller contact bearing

SR. NO	Parameter	Sliding contact bearing	Rolling contact bearing
1	**Size**	Large	small
2	**Life**	Less life	Long life
3	**Coeff. of friction**	High	less
4	**Housing Diameter**	Less	Large

1 M

Each

d	**Method of obtaining bolts of uniform strength**  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	02 02
e	**Procedure for selection of bearing from manufacturer's Catalogue.** 1) Calculate radial and axial forces and determine dia. of shaft. 2) Select proper type of bearing. 3) Start with extra light series for given diagram go by trial of error method.	

4) Find value of basic static capacity (co) of selected bearing from catalogue.

5) Calculate ratios Fa/VFr and Fa/Co.

6) Calculate values of radial and thrust factors.(X & Y) from catalogue.

7) For given application find value of load factor Ka from catalogue.

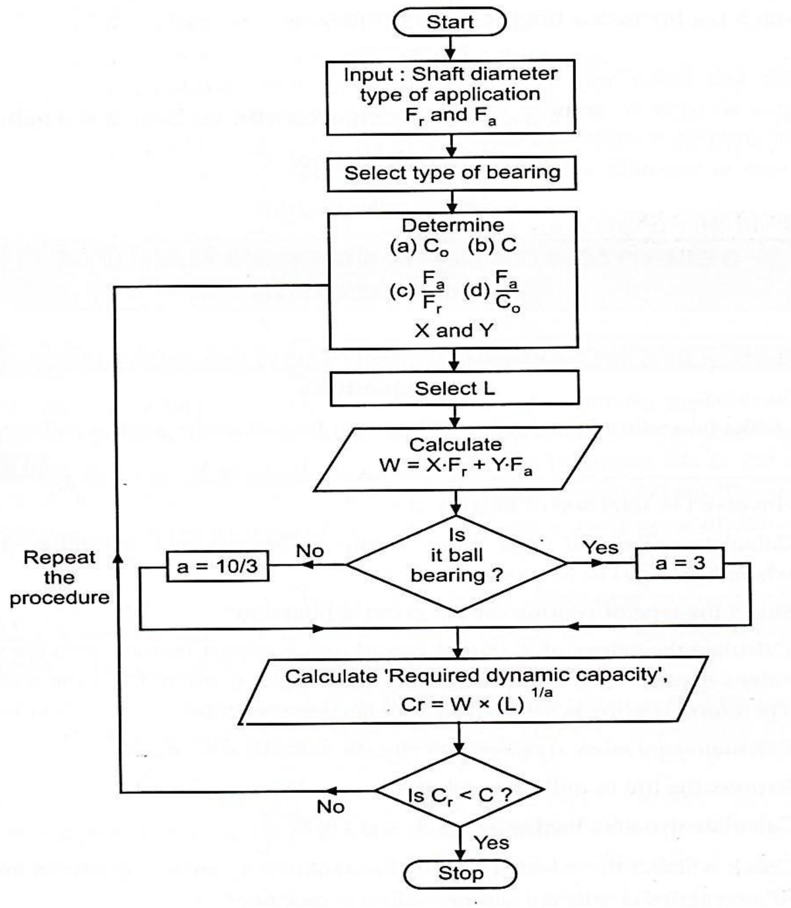
8) Calculate equivalent dynamic load using relation. Pe = (XVFr + YFA) Ka.

9) Decide expected life of bearing considering application. Express life in million revolutions L10.

10) Calculate required basic dynamic capacity for bearing by relation.

11) Check whether selected bearing has req. dynamic capacity, IF it not select the bearing of next series and repeat procedure from step-4.

OR (flowchart)



CORRECT STEPS

OR

FLOW

CHART

4 M