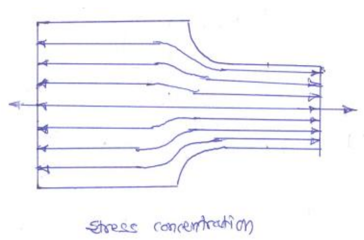
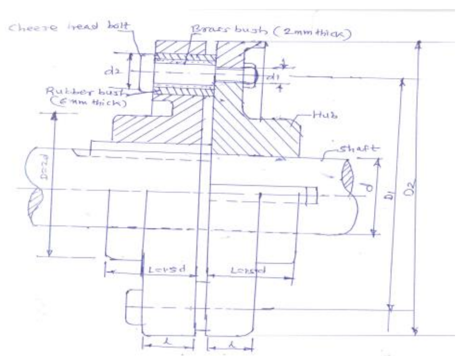
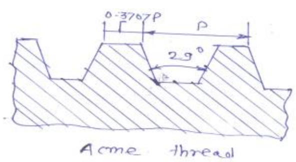
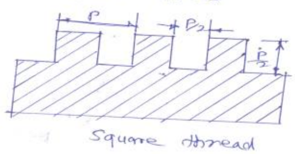
1Important Instructions to examiners:

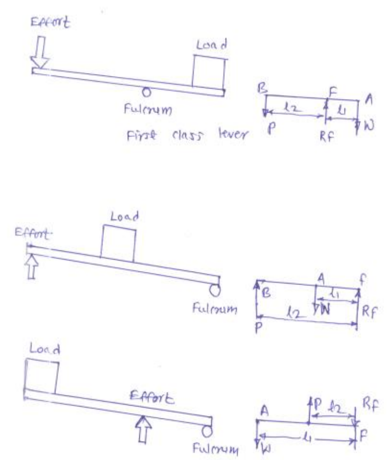
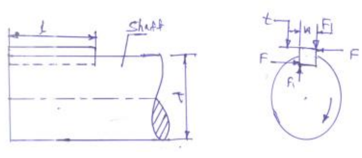
1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling 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 (i)	**Attempt any three** Whenever a machine component changes the shape of its cross section, the simple stress distribution no longer holds good. This irregularity in the stress distribution caused by abrupt changes of form is called as stress concentration  In most machine elements have some forms of discontinuity, namely sudden change in cross section, grooves ,holes, keyways and other changes in sections. these continuity in machine element alter the stress distribution in the neighborhood so that the elementary stress equations no longer described the actual state of stress in the part, such discontinuity is called stress raisers and in the region in which these occur is called the area of stress concentration. Internal cracks and flaws, cavities in welds, blowholes, and pressure in certain points are the common examples of stress raisers.	02 02
	ii.	**Design of Knuckle joint** Failure of rod in tension	

		Rod may fail in tension due to tensile load Tensile strength of rod , P = $\frac{\pi}{4}$xd^2xσ_t From this equation diameter of rod may obtained **Diameter of knuckle pin in shearing** Since the pin is in double shear, Shearing strength of pin P = $\frac{\pi}{4}$ x$d1^2$ xσ_t Value of d_1 can be found here d_1=d Fix the dimensions using empirical relations; Dia. Of pin = d_1 =d Outer dia. Of single or double eye = d_2 =2d Dia. Of knuckle pin head and collar =d_3 = 1.5d Thickness of single eye = t = 1.25d Thickness of fork =t_1 = 0.75d Thickness of collar pin =t_2 = 0.5d Checking the failure of single eye in tension σ_t = p/(d_2-d_1) x t Checking the failure of single eye in crushing σ_{ck} = p/d_1 x t Checking the failure of single eye in shear τ = p/(d_2-d_1) x t Checking the failure of double eye in tension σ_t= p/2(d_2-d_1) x t_1 Checking the failure of double eye in crushing σ_c= p/2d_1 x t_1 Checking the failure of double eye in shear τ = p/2(d_2-d_1) x t_1	Eight steps 04marks

	iii)		04
	iv		02
			
		Characteristics of Acme thread : (i)thread angle is 29^0 (ii) permit the use of split nut (iii)easy to manufacture (iv) max. bursting pressure on the thread Characteristics of Square thread : (i) zero profile thread angle (ii) minimum bursting pressure on the nut	02
b	i)	Main considerations in machine design Type of loads and stresses caused by the load: the load on a machine component, may act in several ways, due to which, the internal stresses are set up. Mechanism: the successful operation of any machine depends largely upon the simplest arrangement of the parts, which will give desired motion Selection of material: designer should know the deep knowledge of properties of materials and behavior under working conditions	Any 6 pts. 6 marks

		Convenient and economical features: the designed machine must be convenient to operate and cost wise economical for the customer	
		Use of standard part: reduced the overall cost	
		Safety of operation: to avoid accidental hazards, care should be taken by designer	
		Workshop facilities: a designer should be familiar with the limitations of his employer’s workshop, in order to avoid necessity of vendors	
		Number of machines to be manufactured	
		Cost of construction and assembly: designed machine should be cheap and easy to assem6ble	
		Frictional resistance and lubrication: designer should provide necessary lubrication to the parts, where there is a sliding, rolling and rotating motion	
	ii)	Given : P= 50 KW = 50000W	
		Speed = 600rpm	
		k=Di/do = 0.8	
		σ_{yt}= 380 N/mm2	
		Factor of safety= 4	
		Design stress $\sigma_t=\sigma_{yt}$/fos =380/4 =95	01
		Shear stress = $\tau = \sigma_t/2$ = 95/2 =47.5N/mm2	
		Torque transmitted by hollow shaft T = P x 60/2πN	01
		T = 50000 x 60/2π x600	
		T = 795.67 N-m	
		T= 795670 Nmm	
			01
		T= π/16 Xτ X $do^3(1-k^4)$	
		795670 =π/16 X 47.5 X do^3(1-0.8^4)	01
		Do^3=144529.313	
		Do = 53 mm say 55 mm	01
		Di = 0.8X 55 = 44mm	01

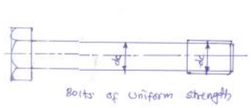
2	a.	Attempt any TWO	
		In the first type of levers, the fulcrum is in between the load and effort. In this case, the effort arm is greater than load arm, therefore M.A. obtained is more than 1	01
		Application: Bell crank levers used in railway signaling arrangement, rocker arm in I.C. Engines , handle of a hand pump, hand wheel of a punching press, beam of a balance, foot lever (any 1)	01
		In the second type of levers, the load is in between the fulcrum and effort. In this case, the effort arm is more than the load arm, therefore M.A. is more than 1.	01
		Application: levers of loaded safety valves, wheel barrow, nut cracker (any1)	01
		In the third type of levers, the effort is in between the fulcrum and load. Since the effort arm, in this case, is less than the load arm, therefore M.A. is less than 1	01
		Application: a pair of tongs, the treadle of sewing machine (any 1)	01
			02
	b.		02
		T= Torque transmitted by the shaft ,	
		F= tangential force acting at the circumference of the shaft,	

		d = dia. Of shaft,	
		l = length of key,	02
		w = width of key	
		t = thickness of key	
		τ and σ_c = shear and crushing stress for the material of key	
		Consider shearing of key, the tangential shearing force acting at the circumference of the shaft ,F = Area resisting shearing X shear stress = lxw x τ	01
		Torque transmitted by the shaft , T = F X d/2 = l x w x τ x d/2	
		Consider crushing of key, the tangential crushing force acting at the circumference of the shaft ,F = Area resisting crushing x crushing stress = lxt/2 x σ_c	01
		Torque transmitted by the shaft , T = F X d/2 = lxt/2 x σ_c x d/2	
		The key is equally strong in shearing and crushing ,if	
		lxw x τ x d/2= lxt/2 x σ_c X d/2	
		w/t =σ_c/2 τ	
		as, w = t	02
		$therefore\ \ \sigma_c=2\tau$	
	C i	(i) Applications of maximum shear stress theory : for ductile material , crank shaft, propeller shafts , c frames (ii) Applications of maximum principle normal stress theory : for brittle material , machine spindle, machine beds , c frames, overhang crank	02 02
	ii.	Applications of cotter joint: cotter foundation bolt, big end of the connecting rod of a steam engine, joining piston rod with cross head, joining two rods with a pipe	02
		Applications of knuckle joint: link of bicycle chain, tie bar of roof truss, link of suspension bridge, valve mechanism, fulcrum of lever, joint for rail shifting mechanism	02
3		Attempt any four	
	a.	30 Ni 16 Cr5 : alloy steel	One each
		carbon 0.3% of average, Nickel 16%, chromium 5%	
		40C8 : Plain carbon steel	
		carbon 0.4% of average, manganese 0.8%	
		FeE230 : Steel with yield strength of 230N/mm2	

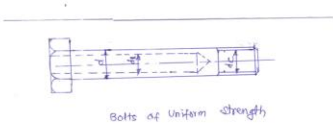
		X15Cr25Ni12 : high alloy steel carbon 0.15% of average, chromium 25%, Nickel 12%,	
	b.	Given : Load 200 KN= 200000N σ_t =75 MPa, τ= 50 MPa (i) **Dia of rod** $P= \pi/4\ xd^2\ x\ \sigma t$ $200000 = 0.7854x\ d^2\ x\ 75$ d = 58.27mm say 60 mm **failure of spigot in tension across the slot** $p= \pi/4\ (d_2)^2-d_2xt$ 200000 = 0.7854 xd2xd2 – d2x d2/4, $t= d_2/4 = 60/4=15$ $D_2^2= 200000/(0.7854-0.25)x\ 75$ D_2 =70.58mm **Failure of spigot end in shear** , P= 2Xaxd2x6s 200000 =2xax70.58x 50 a= 28.33mm **Failure of spigot collar in shear** $P= \pi\ d_2x\ t1\ x\ \tau$ 200000 = 3.142x 70.58xt1x 50 t1= 18.03mm **failure of socket in tension across the slot**, $P= \pi/4(d^2_1-d^2_2)- (d_1-d_2)x\ t\ x\ \sigma_t$ d_1xd_1 -19.09d1-7028.85 =0 solving by quadratic eq. method d_1= - (19.09)+- (-19,09x19.09- 4 x1x 7028.85)1/2/2 d_1 =84.925mm **failure of cotter in shearing** $P= 2xbxtx\tau$ 200000= 2xbx15x50 b =133.33mm	01 01 01 01
	c.	Lewis equation: $W_T = \sigma_w.b.\pi.m.y$, W_T= Tangential load acting at the tooth in N σ_w= bending stress in N/mm2 b= width of the gear face in mm m= module in mm y= lewis form factor.	02 02
	d.	bolts of uniform strength: if a shank dia.is reduced to a core dia.as shown in fig. the stress become same through out 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 and lighter. This type of bolt is known as bolt of uniform strength.Another method of obtaining the bolt of uniform strength is shown in fig.in this method, instead of reducing the shank dia.an axial hole is drilled through the head down to	02

the threaded portion such that the cross sectional area of the shank becomes equal to the area of the threaded portion

If bolts of uniform strength are not used a large portion of impact energy will be absorbed in the threaded portion and relatively a small portion of energy is absorbed by a shank. This uneven distribution of impact energy may lead to the fracture of the bolt in threaded portion .hence bolts of uniform strength are preferred.



02



e. F= tangential force acting at the circumference of the shaft,

d = dia. Of shaft,

l = length of key, 04

w = width of key

t = thickness of key

τ and σ_c= shear and crushing stress for the material of key

Consider shearing of key, the tangential shearing force acting at the circumference of the shaft ,F = Area resisting shearing X shear stress = lxw x 6s

Torque transmitted by the shaft , T = F X d/2 = lxw x 6s x d/2

Consider crushing of key, the tangential crushing force acting at the circumference of the shaft ,F = Area resisting crushing x crushing stress = lxt/2 x 6c

Torque transmitted by the shaft , T = F X d/2 = lxt/2 x 6c X d/2

The key is equally strong in shearing and crushing ,if

lxw x 6s x d/2= lxt/2 x σ_cX d/2

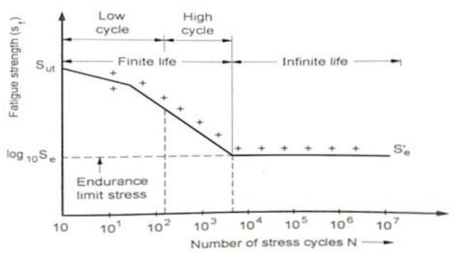
w/t =6c/2 τ

$\sigma_c=2\tau$

Commented [a1]: Please use the same symbol throughout .

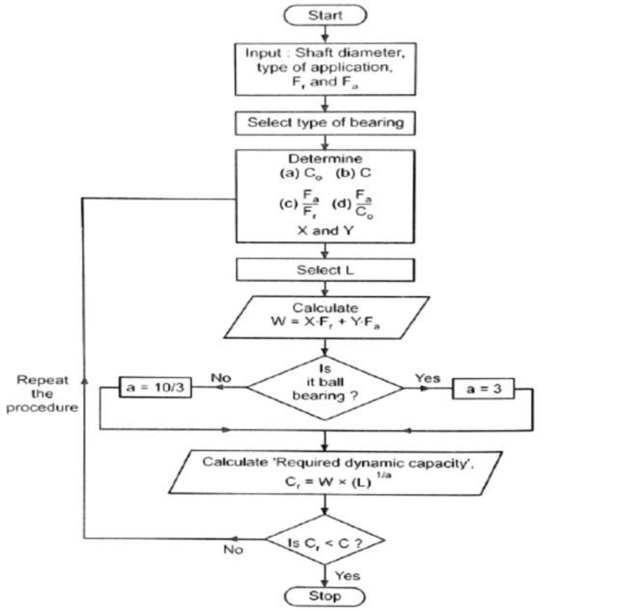
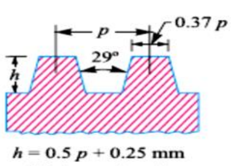
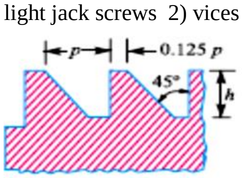
Commented [a2]: Please use the same symbol throughout .

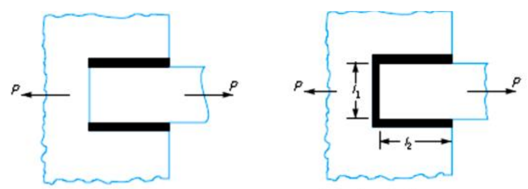
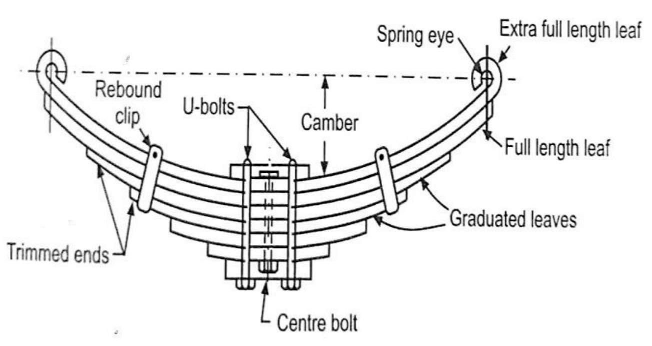
4	a.	**Attempt any THREE of the following**	
	i.	Ergonomics consideration in the design of Lathe machine Any 4 1) The controls on lathe should be easily accessible and properly positioned. 2) the control operation should involve minimum motions. 3) Height of lathe should be match with worker for operation 4)Lathe machine should make less noise during operation. 5) force& power capacity required in turning the wheel as per operation or human being can apply normally. 6) should get required accuracy in operation.	04
	ii.	Wahl's Factor Equation: $K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$ **S.I unit of Each Term:** C: Spring Index Unit: it is constant unit less term	02 02
	iii.	Modes of Gear Failure: ANY 4 modes **1. *Bending failure*.** Every gear tooth acts as a cantilever. If the total repetitive dynamic load acting on the gear tooth is greater than the beam strength of the gear tooth, then the gear tooth will fail in bending, **2. *Pitting*.** It is the surface fatigue failure which occurs due to many repetition of Hertz contactstresses. **3. *Scoring*.** The excessive heat is generated when there is an excessive surface pressure, high speed or supply of lubricant fails. **4. *Abrasive wear*.** The foreign particles in the lubricants such as dirt, dust or burr enter betweenthe tooth and damage the form of tooth. **5. *Corrosive wear*.** The corrosion of the tooth surfaces is mainly caused due to the presence ofcorrosive elements such as additives present in the lubricating oils.	One each (Total 4)
	iv.	**Four Disadvantages of screwed joints:** 1) Screwed joints are weaker than welded joint 2) Screwed joints weakens(due to holes) the parts that are to be joined. 3) Stress concentration in the threaded portion of screw makes them weak. 4) Locking arrangement is required in case of vibrations	04

	b.	Attempt any ONE of the following	
	i.	**Design procedure of Shaft on the Basis of torsional rigidity.** **Torsional rigidity.** The torsional rigidity is important in the case of camshaft of an I.C.engine where the timing of the valves would be affected. The permissible amount of twist should not exceed 0.25° per metre length of such shafts. For line shafts or transmission shafts, deflections 2.5 to 3 degree per metre length may be used as limiting value. The widely used deflection for the shafts is limited to 1 degree in a length equal to twenty times the diameter of the shaft. The torsional deflection may be obtained by using the **torsion equation**, Diameter of shaft on the basis of rigidity $\frac{T}{J} = \frac{G\theta}{L}$ Where ,**θ** = Torsional deflection or angle of twist **in radians,**	02
		T = Twisting moment or torque on the shaft, N.mm ***J*** = Polar moment of inertia of the cross-sectional area about the axis of rotation, L=Length of shaft in mm G=Modulus of rigidity in N/mm^2 $J = \frac{\pi}{32} \times d^4$-----------For solid shaft $J = \frac{\pi}{32} \times (do^4 - di^4)$-----------For Hollow shaft	02
		Two Applications: Propeller shaft of automobile, marine engine shaft and Shaft of pump and motor	02
	ii	S-N Curve:  	02

		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*** (σe).	02
		Need of Endurance Limit in Machine Design: *Endurance limit* is used to describe a property of materials: the amplitude (or range) of cyclic stress that can be applied to the material without causing ***fatigue*** failure.	02
5	a.	Attempt Any TWO	
		Design of power Screw: Given Data: $W = 15\ KN = 15 \times 10^3\ N$, $\sigma_{cnut} = 85\ N/mm^2$, $\tau_{nut} = 37\ N/mm^2$ $P_b = 5\ N/mm^2, \mu = 0.14$ Design of Screw: 1) Core Diameter of screw : Consider the screw under pure compression to find diameter of screw $\sigma c = \frac{W}{\frac{\pi}{4}X(dc)^2}$, $85 = \frac{15\,x(10)^3}{\frac{\pi}{4}X(dc)^2}$ $d_C = 14.99$ say 15 mm Do =Dc /0.84 = 15/0.84 = 17.86 Say 18 mm D = (do + dc)/2 = (15 +18) /2 = 16.5 mm P= do- dc = 18-15 = 3 mm	02
		ii)Length of Nut : The bearing pressure between the thread $Pb = \frac{w}{\frac{\pi}{4}X(do^2-dc^2)\,n}$, $5 = \frac{15\,x(10)^3}{\frac{\pi}{4}X(18^2-15^2)\,n}$, n= 38.60 i.e = 40 contacts Height of Nut: h=n x p =40 x 3 =120 mm Helix angle $\alpha = tan^{-1}\frac{\text{Lead}}{\pi x 16.5}$ = 3.31° $\emptyset = tan^{-1}\mu$ $= tan^{-1}x\ 0.12$ = 6.84° Torque required lifting the load $T_1 = W.\tan(\alpha + \emptyset)\frac{d}{2}$ $T_1 = 15x\ 10^3 \tan(3.31+6.84)\frac{16.5}{2}$ = 22159.13 N.mm As collar friction is Neglecting, $T_2=0$ Total Torque required to lift the load = T_1= 22159.13 N.mm III) Efficiency of power screw : $\tilde{n} = \frac{\text{W}.\tan(\alpha)\frac{d}{2}}{T}$	02

		$=\frac{(15x10^3 tan\,(3.31)16.5/2)}{22159.13}$ = 0.323 = 32 %	02
		IV)Shear stresses in threads of screw & nut : Shear stress induced in the screw thread $\tau = \frac{W}{\pi X(dc)Xtn}$ as t= p /2 $\tau = \frac{50 \text{ X } 10^3}{\pi X(15)X1.5\ X\,40}$ =5.30 N/mm^2 Shear stress induced in the Nut thread $\tau = \frac{W}{\pi X(do)Xtn}$ as t = p /2 $\tau = \frac{50 \text{ X } 10^3}{\pi X(18)X1.5\ X\,40}$ =4.42 N/mm^2	02
	b.	Design of Helical Compression Spring: Given Data: W=100 N , $\delta = 15\ mm$, τ =100 N/mm^2 , **G= 84 x 10^3 N/mm^2** (actually in question paper G= 4MPa is given , but it is not a correct value it could be a printing mistake) C= 12 C= D$_m$/d =12, Ks=1 + $\frac{1}{2C}$ = $1 + \frac{1}{2\ X\ 12}$ = 1.04**(Neglecting curvature effect)**	02
		$\tau = Ks\frac{8\ WC}{\pi d^2}$, 100 =1.04 x$\frac{8\ X\ 100\ X\ 12}{\pi X d^2}$ d=5.6 mm Say 6mm **ii.** **Spring diameter** : D= CX d = 12 X 6 = 72	02
		iii. **No of turns**: $$\delta = \frac{8\ WD^3 n}{Gd^4}$$ $15 = \frac{8\ X100\ \text{X}\ 72^3 n}{84\ x10^3 x\ 6^{\ 4}}$n= 5.47 i.e 6 turns	02
		Assuming squared & grounded ends ,total number of truns is given by n' = n +2 = 6+2 =8 **iv.** Stiffness of spring K= W/δ = 100/ 15 = 6.66 **(Note: spring is designed by considering G= 84 X10^3 N/mm^2 instead of 4 Mpa)**	02
	c.i	Steps Involved in selection of a proper ball bearing from Manufacture's Catalogue	

		 **Note:** It is not necessary to provide the above design procedure in a flow chart. Even if student enlistis the steps in a proper sequence, shall be given due credit.	04
	c.ii	Engg. Application of ACME Thread profiles : 1)screw cutting lathes,2) brass valves,3) cocks and 4) bench vices.	01
		 $h = 0.5\,p + 0.25$ mm (*b*) Acme thread.	01
		Buttress Thread profiles 1) light jack screws 2) vices	01
		 $h = 0.75\,p$	01

6		Attempt any FOUR of the following	
	a.	Derivation of strength equation for parallel fillet weld subjected to tensile load	
		The parallel fillet welded joints are designed for shear strength. Consider a double parallel fillet welded joint as shown in Fig.	
		(a) Double parallel fillet weld. (b) Combination of transverse and parallel fillet weld.	
		Minimum area of weld or the throat area, $A = 0.707\ s \times l$	01
		If τ is the allowable shear stress for the weld metal, then the shear strength of the joint for single parallel fillet weld,	
		P = Throat area × Allowable shear stress = $0.707\ s \times l \times \tau$	01
		and shear strength of the joint for double parallel fillet weld,	
		$P = 2 \times 0.707 \times s \times l \times \tau = 1.414\ s \times l \times \tau$	02
	b.	Application of Leaf spring (Any Two)	
		Bus/truck/Car suspension springs, diving board,	01
		Sketch of Leaf Spring of semi elliptical Type	03
			

c. **Given Data:**

D=250 mm , P=1.5 N/mm^2 , n =12 Nos. ,σt = 30 Mpa

The upward force on the cover =

$$P = \frac{\pi D^2 P}{4} , P = \frac{\pi X 250^2 X 1.5}{4} = 73631 \text{ N}$$ 01

Resisting force offered by 12 bolts

$$P = \frac{\pi \text{x dc}^2 \text{ x } \sigma\text{t x n}}{4}$$ 01

$$73631 = \frac{\pi \text{x dc}^2 \text{ x 30 x 12}}{4} , \text{ dc} = 16.13 \text{ mm}$$ 01

Do = dc /0.84 = 16.13/0.84 = 19.21 mm 01

Nominal diameter of bolts = 19.21 mm.=20 mm

d. Differentiate between sliding & rolling contact bearing Any 4

SR.NO	Parameter	Sliding contact bearing	Rolling contact bearing
1	**Size**	large	Small
2	**starting torque**	High	low
3	**noise**	Less noise	High noise
4	**Life**	Less life	Long life
5	**Cost**	Less cost	More costly
6	**Coeff. of friction**	High	less

04

e. **Given:**

C=26 KN ,L_{10h} = 8000 h , n=300 rpm

Bearing life (L_{10})

$$L10 = \frac{60\, n(L10h)}{10^6} , \quad L10 = \frac{60x \;300\, x\, 8000}{10^6} = 144 \text{ million rev.}$$ 02

Equivalent radial load

C= P (L_{10})$^{0.3}$, P =26000/(144)$^{0.3}$ =5854.16 N

Fr= P =5854.16 N 02