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

WINTER– 16 EXAMINATION

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 TWO**	
	(i)	**Factor of safety:** It is defined as ratio of Maximum stress to the working stress (permissible /design stress Mathematically, Factor of safety = $\frac{\text{Maximum stress}}{\text{working stress / Designstress}}$ For Ductile Material, Factor of safety = $\frac{Yield\ stress}{working\ stress\ /\ Designstress}$ For Brittle material, Factor of safety = $\frac{Ultimate\ stress}{working\ stress\ /\ Designstress}$----------	03
		In design analysis, number of parameters which are difficult to evaluate accurately such as a) Variation in the properties of material like yield strength or ultimate strength. b) Uncertainty in magnitude of externals forces acting on the components. c) Variations in the dimensions of the components due to imperfect workmanship. In order to ensure the safety against such circumstances, factor of safety is useful in design.	**03**
	ii.	**Cotter Joint:** A cotter joint is temporary joint and used to connect two coaxial rods or bars which are subjected to axial tensile and or compressive forces. It consist of 1) spigot 2) socket 3) cotter **Application:**	02

		1) Lewis foundation bolt 2) connection of the piston rod to cross head of a reciprocating steam engine. 3) valve rod & its stem 4) piston rod to the trail end in an air pump. 5) Cycle pedal sprocket wheel.	04 (1 Each)
	iii	**Design of Hollow shaft:** Given Data: T =4750 N-m = 4750 X 10^3 N-mm , Ʈ = 50 N/mm^2 , K=Di/Do =0.4 The hollow shaft is designed on the basis of strength from the derived torsion equation. $T = \frac{\pi}{16}\ x\ Do^3\ x\ \tau\ x\ (\ 1 - K^4\ )$ 4750 X 10^3 N-mm $= \frac{\pi}{16}\ x\ Do^3\ x\ 50\ x\ (\ 1 - 0.4^4\ )$----	02
		Thus Do= 79.18 mm ≅ 80 mm (Say)	02
		Di = 0.4 x Do = 0.4 x 80 = 32 mm.	02
	b	**Attempt any ONE of the following**	
	i (1)	**Factors to be considered for selection of material for design of machine elements** a) Availability: Material should be available easily in the market. b) Cost: the material should be available at cheaper rate. c) Manufacturing Consideration: the manufacturing play a vital role in selection of material and the material should suitable for required manufacturing process. d) Physical properties: like colour, density etc. f) Mechanical properties: such as strength, ductility, Malleability etc. g) Corrosion resistance: it should be corrosion resistant.	03 (3 pts)
	i(2)	**a) Ductility:** the property of material which enables it to be drawn into thin wire under the action of tensile load is called as ductility.	01
		b) Toughness: The property which resists the fracture under the action of impact loading is called as toughness. Toughness is energy for failure by fracture.	01
		c) Creep: when a component is subjected to constant stress at a high temperature over a long period of time ,it will undergo a slow& permanent deformation called creep Or it is defined as “slow and progressive deformation of material with time under constant stress at elevated temperature. E.g : Bolts & pipes in thermal power plants	01

ii) **Bush type flange coupling**

Given Data: P= 40 KW = 40 X 10^3 W , N= 1000 rpm , d= 50 mm dp= 45 mm, τ_{ci}=15 N/mm^2

P_b=0.45 N/mm^2, τ=25 N/mm^2

1) Power Transmitted $P = \frac{2\pi NT}{60}$

$T = \frac{P\ X\ 60}{2\pi N} = \frac{40\ X\ 103;X\ 60}{2\pi\ X\ 1000}$ = 381.97 N.m =381.97 x 10^3 N.mm

Let Number of Pins = 6

01

2) Diameter of pin: d_1 =0.5d/$\sqrt{n}$ = 0.5 x 50/$\sqrt{6}$ =10.20

In order to permit the bending stress induced in the pin due to compressibility of brass bush .let us modify diameter of pin d1=20 mm .This diameter is threaded and secured Right hand half coupling .

01

Let us take, diameter of the enlarged portion in the left half coupling d1 =24 mm. A brass bush of 2mm is fitted over the enlarged portion of pin. also brass bush carries rubber bush of 6 mm.

Diameter of rubber bush =d_2 = d_1+2x2+2 x 6 =24+4+12 = 40 mm.

Diameter of pitch circle of pin =D1 = 2 x d + d2 + 2 x6 =100+40+12 = 152 mm.

3)Bearing load acting on each pin W = Pb x d_2 x l =0.45 x 40 x l = 18 x l

Total bearing load on all pins = n x W

Torque transmitted by coupling = T =n x W x D1/2

381.97 x 10^3 = 6 x 18 x l x 152/2

.l=46.54 mm 01

W=18 x l = 18 x 46.54 = 837.72 N-

4.Direct shear stress in coupling halves

$\tau = \frac{w}{\frac{\pi}{4}d1^2} = \frac{837.72}{\frac{\pi}{4}20^2}$ =2.67 N/mm^2

σ_b = (M / Z) , M= W X (l/2 +5)= 837.72 x(46.54/2 +5) Z = (π/32) 20^3

σ_b = [837.72 x(46.54/2 +5) / (π/32) 20^3] = 30.15 N/mm^2

Checking of maximum stress

According to Maximum shear stress theory 01

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

$\tau_{max} = \frac{1}{2}\sqrt{\sigma b^2 + 4\,\tau^2}$

$\tau_{max} = \frac{1}{2}\sqrt{30.15^2 + 4\,(2.676)^2}$ = 15.31 N/mm^2

According to Normal stress theory

$\sigma b\ max = \frac{1}{2}\sigma b + \frac{1}{2}\sqrt{\sigma b^2 + 4\,\tau^2}$

$\sigma b\ max = \frac{1}{2}30.15 + \frac{1}{2}\sqrt{30.15^2 + 4\,(2.67)^2}$ =30.38 N/mm^2

01

As above maximum stresses are within safe limit , design is safe.

5)Design of Hub:

Diameter of Hub: D = 2d = 100 mm

Length of Hub L = 1.5 d = 75 mm

Considering hub is hollow shaft

$$T = \frac{\pi}{16}\ x\ D^3\ x\ \tau ci\ \ x\ (\ 1 - K^4\ )$$

01

$381.97\ x\ 10 = \frac{\pi}{16}\ x\ \ 100^{\ 3}\ x\ \tau ci\ \ x\ (\ 1 - 0.5^{\ 4}\ )\ =$

τ_{ci} =2.59 N/mm^2

As it is within limit , design is safe.

2 **Attempt any TWO of the following**

a) **Design of knuckle joint:**

Step 1) Diameter of Rod: d : =?

Consider tensile failure of Rod

1. P =σ_t x A ,
   150 x 10^3 = 75 x π/4 xd^2 ,
   d = 50.4 mm ≅ **52 mm (say)**

Using Imperial relations

Diameter of Knuckle pin Outside

01

		$d_1 = d = 52$ mm Outer diameter of eye, $d_2 = 2d = 2 \times 52 = 104$ mm Diameter of knuckle pin head and collar, $d_3 = 1.5d = 1.5 \times 52 = 78$ mm Thickness of single eye or rod end, $t = 1.25d = 1.25 \times 52 = 65$ mm Thickness of fork, $t_1 = 0.75d = 0.75 \times 52 = 39$ say 40 mm Thickness of pin head, $t_2 = 0.5d = 0.5 \times 52 = 26$ mm	01
		2. Failure of the knuckle pin in shear Since the knuckle pin is in double shear, therefore load (P), $150 \times 10^3 = 2 \times \frac{\pi}{4} \times (d_1)^2 \tau = 2 \times \frac{\pi}{4} \times (52)^2 \tau = 4248\,\tau$	01
		$\tau = 150 \times 10^3 / 4248 = 35.31$ MPa	01
		3. Failure of the single eye or rod end in tension The single eye or rod end may fail in tension due to the load. We know that load (P), $150 \times 10^3 = (d_2 - d_1)\, t \times \sigma_t = (104 - 52)\, 65 \times \sigma_t = 3380\, \sigma_t$ $\therefore \sigma_t = 150 \times 10^3 / 3380 = 44.4$ N/mm^2 = 44.4 MPa	01
		4. Failure of the single eye or rod end in shearing The single eye or rod end may fail in shearing due to the load. We know that load (P), $150 \times 10^3 = (d_2 - d_1)\, t \times \tau = (104 - 52)\, 65 \times \tau = 3380\, \tau$ $\therefore \tau = 150 \times 10^3 / 3380 = 44.4$ N/mm^2 = 44.4 MPa	01
		5. Failure of the single eye or rod end in crushing The single eye or rod end may fail in crushing due to the load. We know that load (P), $150 \times 10^3 = d_1 \times t \times \sigma_c = 52 \times 65 \times \sigma_c = 3380\, \sigma_c$ $\therefore \sigma_c = 150 \times 10^3 / 3380 = 44.4$ N/mm^2 = 44.4 MPa	
		6. Failure of the forked end in tension The forked end may fail in tension due to the load. We know that load (P), $150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \sigma_t = (104 - 52)\, 2 \times 40 \times \sigma_t = 4160\, \sigma_t$ $\therefore \sigma_t = 150 \times 10^3 / 4160 = 36$ N/mm^2 = 36 MPa	01
			01

7. Failure of the forked end in shear

The forked end may fail in shearing due to the load. We know that load (*P*),

$$150 \times 10^3 = (d_2 - d_1)\,2\,t_1 \times \tau = (104 - 52)\,2 \times 40 \times \tau = 4160\,\tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 4160 = 36\ \text{N/mm}^2 = 36\ \text{MPa}$$

8. Failure of the forked end in crushing ---------

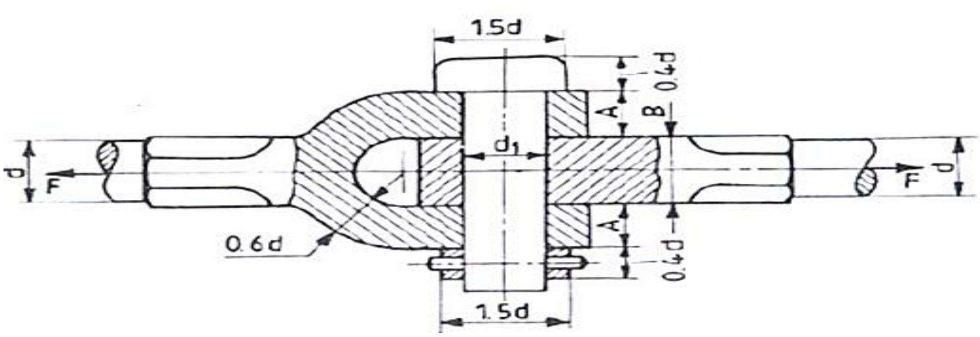
The forked end may fail in crushing due to the load. We know that load (*P*),

$$150 \times 10^3 = d_1 \times 2\,t_1 \times \sigma_c = 52 \times 2 \times 40 \times \sigma_c = 4160\,\sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 4180 = 36\ \text{N/mm}^2 = 36\ \text{MPa}$$

From above, we see that the induced stresses are less than the given design stresses, therefore

b) i



b) ii

For power transmission gears, the tooth form most commonly used the involute profile as

a)Involute gears can be manufactured easily: Since the rack in an involute system has straight sides and since the generating cutters usually have rack profile, these cutters can be easily manufactured. Involute gears can be produced more accurately and at a lesser cost.

b) The gearing has a feature that enables smooth meshing despite the misalignment of center distance to some degree.

c) For effective conjugate action i.e for maintaining a constant velocity ration, in case of involute gearing system, the center distance can be changed without affecting angular velocity ratio.

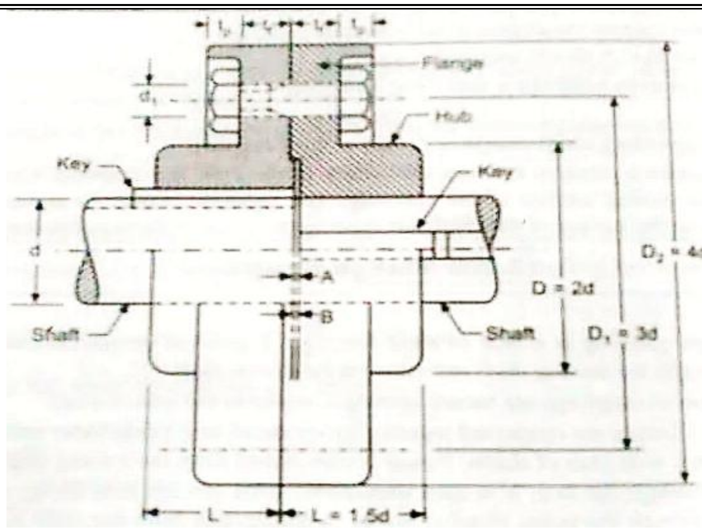
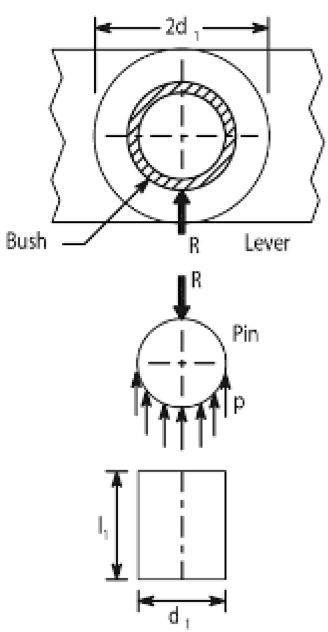
d)In involute gearing as the path of contact is a straight line and the pressure angle is constant .

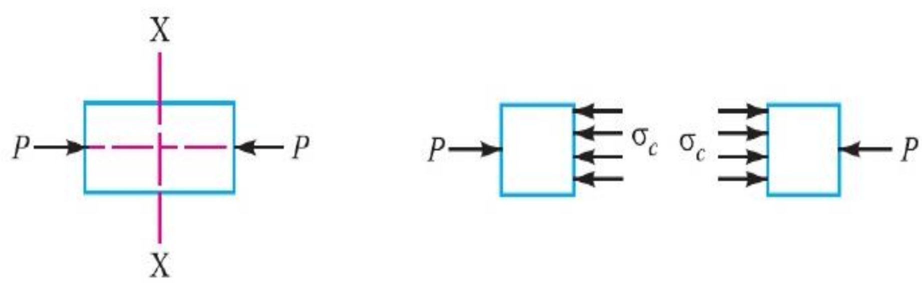
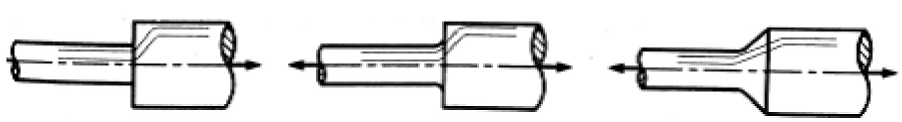
2 marks each

(2 reasons)

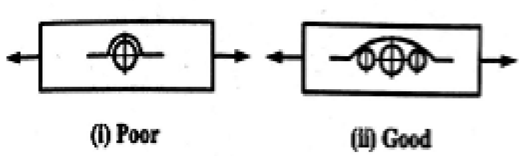
04

Sketch of Protected type flanged coupling with details :

	C) i		02
	c) ii	the forces acting on the boss of lever & the pin are equal & opposite .There is a relative motion between the pin & the lever and bearing pressure becomes design criteria. The projected area of the pin is d_1 x $l_1$therefore Reaction R= P (d_1 x l_1). A softer material like phosphorous bronze bush with 3 mm thick is fitted in eyes **to reduce the friction. & bear a bearing pressure upto5 to 10 N/mm$^{2.}$ Bushes are cheaper and can be easily replaceable.**  **Explanation of stresses :**	02 02

3	**a) i**	**a) Transverse shear stress:** When a section is subjected to two equal & opposite forces acting tangentially across the section such that it tends to shear off across the section. The stress is produced is called as transverse stress For Single shearing, Shear stress τ = W/A For Double shearing, Shear stress τ = W/2A **b)Compressive stress:** When a body is subjected to equal & opposite axial push forces, the stress produced is called as compressive stress. It is denoted by " σ_c" $\sigma_c = \frac{P}{A}$ N/mm^2  **c) Torsional shear stress:** When a machine member is subjected to the action of two equal and opposite couples acting in parallel planes (or torque or twisting moment), then the machine member is said to be subjected to torsion. The stress set up by torsion is known as torsional shear stress. It is zero at the centroid axis and maximum at the outer surface. $$\frac{T}{J} = \frac{\tau}{r} = \frac{G\theta}{L}$$	02 Dia.01 explanation01
	a)ii	**Attempt any FOUR of the following** **Methods of reducing stress concentration in cylindrical members with shoulders**  (i) Poor (ii) Good (iii) Preferred (d) Cylindrical component Stress concentration can be reduced in cylindrical members with shoulders by providing fillet at sharp corners of shoulders. Fig 1. Showing cylindrical member with shoulder having sharp corners i.e change in C/S is sudden and therefore stress distribution line get disturbed .so for fig 1, stress concentration is more. fig. 2 & 3 members shoulder having gradual change in C/S. so here stress line maintain spacing and therefore stress concentration is less.	Dia.01 explanation01

Methods of reducing stress concentration in cylindrical members with holes



(a) Tie rod with hole

b) Stress concentration can be reduced in cylindrical members with holes by providing additional holes in vicinity of holes as shown in fig. (ii).

Fig (i) Showing cylindrical member with hole at center having stress line in disturb manner at vicinity of hole and component will fail at hole so for fig (i) ,stress concentration is more . fig. (ii) members shoulder having additional hole in vicinity of hole and therefore stress line maintain spacing between them so here stress concentration is less.

Design of foot lever :

Given data: L=1 m =1000 mm , P=800 N , σ_t =70 N/mm^2 , τ =70 N/mm^2 , Assume B=3t

Step 1) Considering shaft is under pure torsion , therefore

$$T = \frac{\pi}{16} \, x \; d^3 \; x \, \tau$$ **01**

But Twisting Moment on shaft

T= P X L = 800 X 1000 = 800 X 10^3

$$800 \text{ X } 10^3 = \frac{\pi}{16} \, x \; d^3 \; x 70$$

d = 38.75 mm ≅ 40 mm (say)

Step 2) Using the imperical relation fix the other dimensions

d_2 =1.6 d =1 .6 x 40 = 64 mm,
t_2 = 0.3 X d =0.3 x 40 = 12 mm,
l_2 = 1.25 x d = 50 mm, l =2 x l_2 =100 mm 01

Step 3) Considering shaft supported at center of bearing under combined twisting & bending moment.

M=P X l = 800 x 100 = 80 x 10^3 N-mm

		$T = P \times L = 800 \times 1000 = 800 \times 10^3$ N-mm Equivalents twisting moments $Te = \sqrt{M^2 + T^2} = \sqrt{(80 \times 10^3)^2 + (800 \times 10^3)^2} = 804 \times 10^3 N - mm$ Also ,Equivalents twisting moments $Te = \frac{\pi}{16} x\ d1^3\ x\ \tau_{max}$ $804 \times 10^3 = \frac{\pi}{16} x\ d1^3\ x\ 70$, **d_1 = 38.81 mm** $\cong$ **44 mm** (***assume diameter more than* 40 *mm***) **Step 4) Design of key : Consider Key is rectangular** **W= d/4 =40/4 = 10mm t = d/6 =40/6 =6.67 mm** $T = W\ x\ l\ x\ \tau\ x\ \frac{d}{2}$ $800 \times \mathbf{10^3} = 10\ x\ l\ x 70\ x\ \frac{40}{2}$ $\boldsymbol{l = 57.14\ mm}$ Length of key l may be taken as boss length $l2 = 50\ mm$- **Step 5)** Considering bending failure of lever , we can determine cross section of lever. Bending moment on lever, $R_b = d_2/2 = 64/2 = 32$ mm $M = P \times [L - R_b] = 800 \times [1000 - 32] = 774.4 \times 10^3$ N.mm $\sigma_b = (M / Z)$, $Z = 1/6\ t\ B^2 = 1.5 \times t^3$ $73 = (774.4 \times 10^3 / 1.5 \times t^3)$, **t = 19.9 mm** $\cong$ **20 *mm* & B = 3t = 3 x 20 = 60 mm**	01M 01 M 02
	c)	**Consideration in design of key:**	Any 4
	d)	1) Power to be transmitted. 2) Tightness of fit 3) Stability of connection	Any 4 pts 4 marks

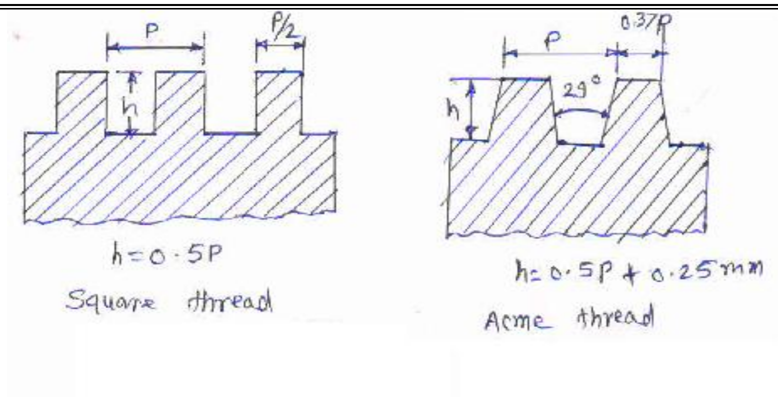
		4) Cost	
		5) Crushing failure of key:	
		6) shearing failure of key	
		7) Material of key ,shaft should be same but key should be weaker than shaft .	
	e)	**Comparison of welded joints with screwed joint.**	
		1) Welded Joint is rigid & permanent. Screwed joint is temporary.	
		2) Cost of welded assembly is lower than that of screwed joints.	1&1/2
		3) Strength of welded structure is more than screwed joints.	
		4) For welding joints, highly skilled worker are required	
		5) Welded joints are tight & leak proof as compared to Screwed joints.	
		6) Welded joint is very difficult to inspect compared to other joints.	1&1/2
		Design of Key	
		Data: d =30 mm ,τ_{Max}=80 Mpa , τ_{Key}=50 Mpa , $l_K = 4\ W_k$	01
		Torque transmitted by shaft is given by	
		$T = \frac{\pi}{16}\ x\ d^3\ x\ \tau$, $T = \frac{\pi}{16}\ x\ 30^3\ x\ 80$ = 424.115 x10 ^{3} N/mm^2 Considering shear failure of key,	
		$$T = W_k \times l_k \times \tau \times \frac{d}{2}$$	
		$$424.115\ \times 10^3 \frac{\mathrm{N}}{\mathrm{mm2}} = W_k \times 4\ \mathrm{W_k} \times 50 \times \frac{30}{2}$$ $$\boldsymbol{W_k = 11.89\ mm}$$	
		$l_K = 4\ W_k$, **l_K= 4 x 11.89 = 47.56 mm** if Key is Square then **$t_k = w_k$ = 11.89 mm** if Key is Rectangular then**t_k = 2/3 w_k = 2/3 x 11.89 mm= 7.92 mm**	

Q. No.	Sub Q. N.	Answer	Marking Scheme

4	a	Attempt any THREE	
	(i)	Composition in percentage 1) Carbon-0.3-0.4 %, manganese 0.5 % and molybdenum 2.8 % 2) Carbon-0.26-0.34, ,Nickel 1 % and Chromium 0.25 % 3) Carbon-0.2-0.3,chromium 0.75 % and molybdenum 5.5 %	4 marks for 3
	ii)	**Definition of** 1) Free length-it is a length of spring in unloaded condition 2) Solid height-it is a length of spring in fully loaded condition 3) Spring rate-load per unit deflection 4) Spring index- ratio of mean diameter of coil to diameter of wire	(1 each)
	iii)	**Effect of keyways** – when the keyways are cut on the shafts, material is removed at the skin, there by weakening the cross section of the shaft. Stress concentration effect is also serious at the corner of the keyways. Thus the shaft become weak. Type of key- Hollow saddle key or Tangent key (1 mark)	(3 marks)
	iv)	**Definition w.r.t. bolts** 1) Major dia.- dia. Of imaginary cylinder parallel with the crest of the thread ,it is the distance from crest to crest largest dia. of an external or internal thread 2) Minor dia.-dia. Of imaginary cylinder which just touches the roots of an external thread or smallest dia.of an external or internal screw thread 3) Pitch-distance from a point on one thread to the corresponding point on the next thread. 4) lead- distance between two corresponding points on the same helix	(1 each)
	b)	**Attempt any ONE**	(6marks)
	i)	**Causes-**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 remedies-module and face width of the gear is adjusted so that the beam strength is greater than the dynamic load 2)Pitting-surface fatigue failure which occurs due to many repetition of Hertz contact stresses , failure occurs when the surface contact stresses are higher than the endurance limit of the material. It starts with the formation of pits which continue to grow resulting in the rupture of the tooth surface. Remedies- dynamic gear tooth load the of gear tooth between the gear tooth should be less than the wear strength 3)Scoring-the excessive heat is generated when there is a excessive surface pressure,	

		high speed or supply of lubricant fails. it is stick-slip phenomenon in which alternate shearing and welding takes place rapidly at high spots. Remedies- by proper designing of the parameters such as speed, pressure and proper flow of lubricant, so that the temperature at the rubbing faces is within the permissible limits. 4)Abrasive wear- the foreign particles of lubricants such as dirt, dust or burr enter between the tooth and damage the form of tooth Remedies- by providing filters for the lubricating oil or using high viscosity lubricant oil which unable the formation of thicker oil film and hence permits easy passage of such particles with ought damage of gear tooth surface. 5)Corrosive wear- due to presence of corrosive elements such as additives present in the lubricating oils. Remedies- proper anti corrosive additives should be used.	
	b)ii)	**Importance of Aesthetic considerations in design –** Each product is to be design to perform a specific function or a set of functions to the satisfaction of customers. In a present days of buyer's market, with a number of products available in the market are having most of the parameters identical,the appearance of the product is often a major factor in attracting the customer. For any product, there exists a relationship between the functional requirement and the appearance of a product. The aesthetic quality contributes to the performance of the product, through the extent of contribution varies from product to product. The job of industrial designer is to create new shapes and forms for the product which are aesthetically appealing. For ex.(1) The chromium plating of automobile components improves the corrosion resistance along with the appearance.(2) the aerodynamic shape of the car improves the performance as well as gives the pleasing appearance	**(explanation 4 marks)** **(2 marks)**
5	a) i.	**Attempt any TWO (2X8)** (i) efficiency of screw $\eta = \tan\alpha / \tan(\alpha + \phi)$ And for self locking screws, $\phi \geq \alpha$ or $\alpha \leq \phi$ Efficiency $\leq \tan(\phi) / \tan(\phi + \phi)$ $\leq \tan\phi / \tan 2\phi$ $\leq \tan\phi / (2\tan\phi/(1-\tan^2\phi))$ $\leq \tan\phi \times (1-\tan^2\phi)/(2\tan\phi)$ $\leq ½ - \tan^2\phi/2$ From this expression efficiency of self locking screw is less than 50%	(4 marks)
	ii.	**self locking property of the threads**-if $\phi > \alpha$ the torque required to lower the load will	

		be positive, indicating that an effort is applied to lower the load. if friction angle is greater than the helix angle or coefficient of friction is greater than the tangent of helix angle applications- for very large use of screw in threaded fastener, screws in screw top container lids, vices, C-clamps and screw jacks	(3 marks) (1 mark)
	b) i.	(i) it is easier to overextend the extension spring. Compression springs will bottom out before the overextend. Also it seems like the tensile strength will be weaker at the attachment point for the extension spring, making it generally larger and more cumbersome to correct the deficiency	(4marks)
	ii.	**self locking property -** torque required to lower the load, T= Wtan(ϕ - α)xd/2 **self locking property** of the threads-if ϕ > α the torque required to lower the the load will be positive, indicating that an effort is applied to lower the load. if friction angle is greater than the helix angle or coefficient of friction is greater than the tangent of helix angle(2marks) **Over hauling of screws-** in the above expression, if ϕ< α,then the torque required to lower the load will be negative. The load will start moving downward without the application of any torque, such a condition is known as over hauling of screws.(2marks)	(2 marks) (2 marks)
	c) i.	**(i) definition of** (1) Basic static load rating-static radial load or axial load which corresponds to a total permanent deformation of the ball and race,at the most heavily stressed contact,equal to 0.001times the ball diameter. **(2) basic dynamic load rating**- the constant stationary radial load or a constant axial load which a group of of apparently bearings with stationary outer ring can endure for a rating life of one million revolutions with only 10% failure. **(3)Limiting speed-** it is the empirically obtained value for the maximum speed at bearings can be continuously operated without failing from seizure or generation of excessive heat.	(4 marks)
	ii.	**Physical characteristics of good bearing material**- compressive strength, fatigue strength, embeddability, bondability, corrosion resistant, thermal conductivity, thermal expansion, conformability	Any four (1 each)
6		**Attempt any four(4x4)**	
	a)	Acme thread is stronger-1 mark	

			3 marks
	b)	given load W= 135N Deflection δ =7.5mm Spring index c=10 Permissible shear stress Ʈ=480 MPa Modulus of rigidity G =80 KN/mm^2 Wahl's factor K =4C-1/4C-4 +0.615/C=4X10-1/4X10-4 +0.615/10=1.14 **(1)Mean dia. Of the spring coil (1 mark)** Maximum shear stress, Ʈ = Kx 8WC/π d^2 480 = 1.14x 8x135x10/3.142xd^2 d = 2.857mm from table we shall take a standard wire of size SWG 3 having diameters (d) =2.946mm mean dia. Of the spring coil D= CXd =10x2.946=29.46 mm outer dia. Of the spring coil Do =D+d=29.46+2.946=32.406mm **(2) number of turns of the spring coil (n) (1 mark)** Deflection δ= 8WC^{3}n/Gd 7.5 =8x135X10^{3x}n/ 80000xd n =1.64 say 2 For square and ground end n' =n+2=2+2=4 **(3) free length of spring (1 mark)** =Lf =n'd+ δ + 0.15 x δ=4x2.496+7.5+0.15xx7.5=18.609mm **(4) pitch of the coil (1 mark)** p= free length/n'-1=18.609/4-1=6.203mm	(1 each)

	c)	Horizontal component of 45 KN, W_H= 45Sin 60^0=45x 0.866=38971N and vertical component of 45 KN, W_v,= 45xcos60^0=45x0.5=22500N Direct tensile load in each bolt,W_{t1}= W_H /5=38971/5=7794.20N Turning moment due to W_H about G T_H = W_{Hx} 25=38971x25=974275N (anticlockwise) direct shear load on each bolt =Ws=Wv/5 =22500/5=4500N Turning moment due to Wv about edge of the bracket, Tv= W_vx175=22500x175=3937500N-mm(clockwise(clockwise) Net turning moment =3937500-974275=2963225N---------(I) total moment of the load on the bolts @ th tilting edge = 2w $x(L_1)^2$+ 2w $x(L_2)^2$=2xwx$(50)^2$ + 2xwx$(150)^2$= 50000 w N-mm-----(II) from equations (I) and(II) 2963225N=50000 w N- w= 592.645 N max. tensile load on each of the upper bolt, Wt_2= wL_2 =592.645x150=88896.75 N tensile load on each of the upper bolt, Wt = W_{t1}+ Wt_2 =7794.20+ 88896.75=96690.95N equivalent tensile load =Wte=1/2(Wt+ $\sqrt{}$ $(Wt)^2$+ $4(Ws)^2$ =1/2 (96690.95+97108.91)=96899.93 N Tensile load on each bolt = ∏/4$(dc)^2$ x 6t =0.7854x$(dc)^2$ x 70 dc = 41.98 mm from coarse series the standard core dia. Is 49.0177 mm and corresponding size of the bolt is M56 thickness of the arm of the bracket cross sectional area of the arm A = bXt =100x t	2 marks

section modulus of the arm, $Z = 1/6\ t\ (b)^2 = 1/6\ xtx(100)^2 = 1666.67\ xt$

direct tensile stress $6t_1 = W_H/A = 38971/100t = 389.71/t$

bending stress $6t_2 = M_H/Z = 208/t$

bending stress $6t_3 = Mv\ /Z = 2632.49/t$

net tensile stress, $6t_1 + 6t_2 + 6t_3 = 3230.20/t$ — 2 marks

max. tensile stress , 6t max. $6t/2 + \frac{1}{2}\sqrt{(6t)^2 + 4(\tau)^2} = 70$

$t = 46.36$ mm

d) Rolling contact bearing- contact between the surfaces is rolling ,it is antifriction bearing — (1 mark)

Advantages (any six) — ½ each

(1)low starting and running friction except at very high speed

(2) ability to withstand momentary shock loads

(3) accuracy of shaft alignment

(4) low cost of maintenance

(5) reliability of service

(6) easy to mount and erect

(7) cleanliness

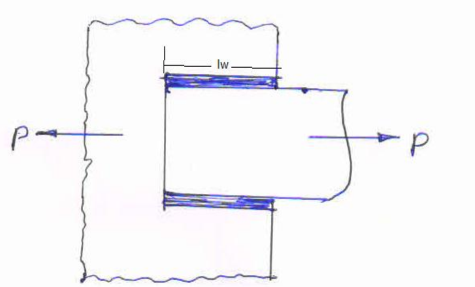
(8) small overall dimension

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

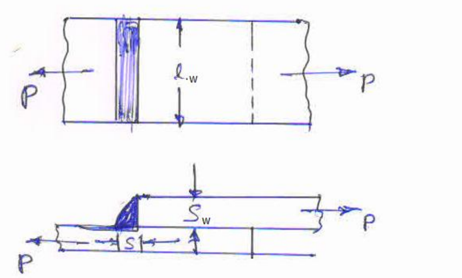
$P = 2x\ 0.707x\ S_w\ x\ l_w x\ \tau = 1.414\ x\ S_w\ x\ l_w x\ \tau$ **(1mark)**

Strength equation of single transverse fillet weld

P =throat area x allowable tensile stress $P = 0.707x\ S_w\ x\ l_w\ x\ \sigma_t$ **(1mark)**



Double parallel fillet weld



Single transverse fillet weld