MODEL ANSWER
Summer – 17 EXAMINATION

Subject Code: 17525

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 judgment 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 Que.	Answer	Marking Scheme
1	A	**Attempt any Three of the following**	12
1A	a	**Give Classification of design.**	04
	Ans.	Depending Upon the method used design may be classified as:(for any four, 4Marks) i) Industrial Design ii) Rational Design iii) Empirical Design iv) Optimum design v) System design vi) Elemental design vii) Computer aided design	04
1A	b	**Define the following terms.** **(i) Factor of safety (ii) Endurance Limit**	04
		(i) Factor of Safety: Factor of safety is defined as the ratio of the maximum stress (yield point stress for ductile material) to the working stress or design stress. In case of ductile materials- $\text{Factor of safety} = \dfrac{\text{Yield point stress}}{\text{Working or design stress}}$	02

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		(ii) 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). **OR** Endurance or fatigue limit can be defined as the magnitude of stress amplitude value at or below which no fatigue failure will occur, no matter how large the number of stress reversals are, in other words leading to an infinite life to the component or part being stressed. For most ferrous materials Endurance limit (Se) is set as the cyclic stress level that the material can sustain for 10 million cycles.	**02**
1A	**c**	**Enlist the applications of following joints:** **(i) Turn Buckle (ii) Knuckle Joint**	**04**
	Ans.	**i) Turn Buckle:** (*Any two applications – 1 mark each*) 1. Tie rod of steering system 2. To connect compartments of locomotives 3. Tie strings of electric poles 4. link rod of leaf springs in multi axle vehicles 5. linkages of gear shifter 6. Connection between brake pedal and master cylinder **ii) Knuckle joint:** (*Any two applications – 1 mark each*) 1. It is used in link of cycle chain 2. It is used in tie rod joints for roof truss 3. It is used in valve rod joint for electric rod 4. It is used in pump rod joint 5. It is used in tension link in bridge structure 6. It is used in lever and rod connection of various types	**02** **02**
1A	**d**	**Enlist types of keys and draw neat sketch of any two types of keys.**	**04**
		The following types of keys are **:(Any four types 2marks, and 1 mark for each sketch, any twotypes)** **1.** Sunk keys, (i) Rectangular Sunk Key (ii) Square sunk key (iii) Parallel sunk key (iv) Gib-headed key (v) Feather key (vi) Woodruff key **2.** Saddle keys, (i) Flat saddle key (ii) Hollow saddle key **3.** Tangent keys, **4.** Round keys, **5.** Splines.	**02**

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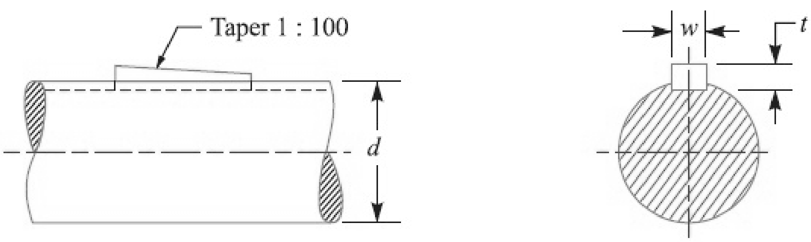
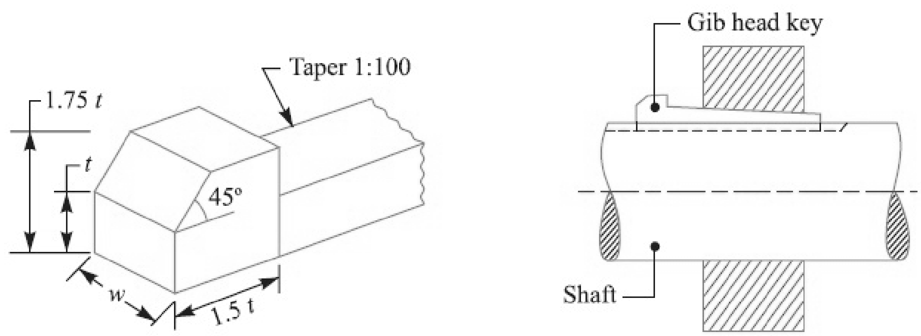
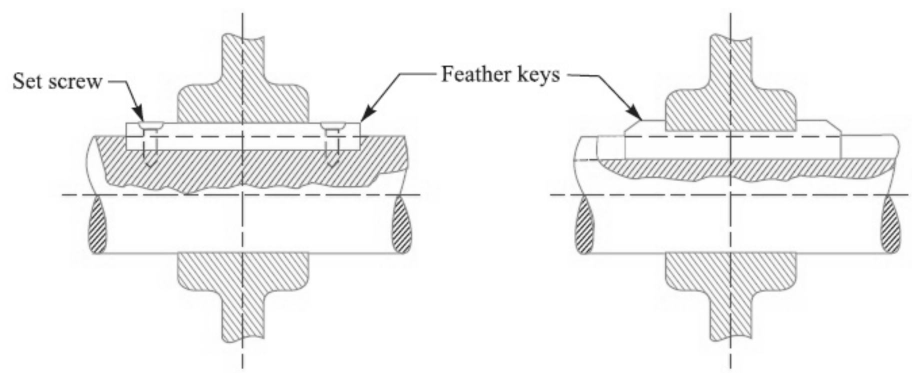


Rectangular Sunk Key

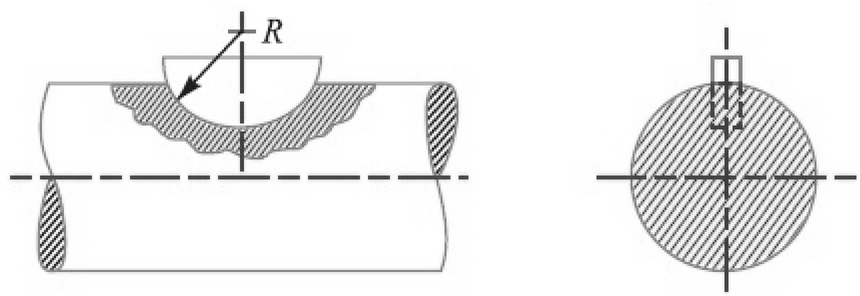


Gib Headed Key

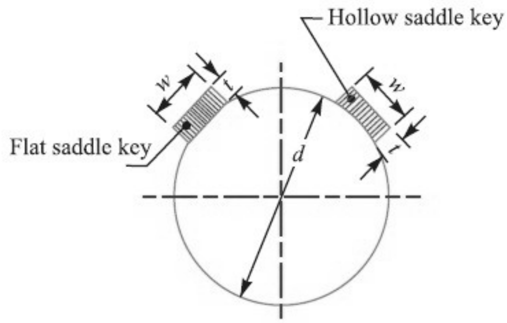
02



Feather Keys

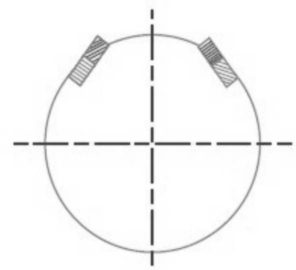


Woodruff Key

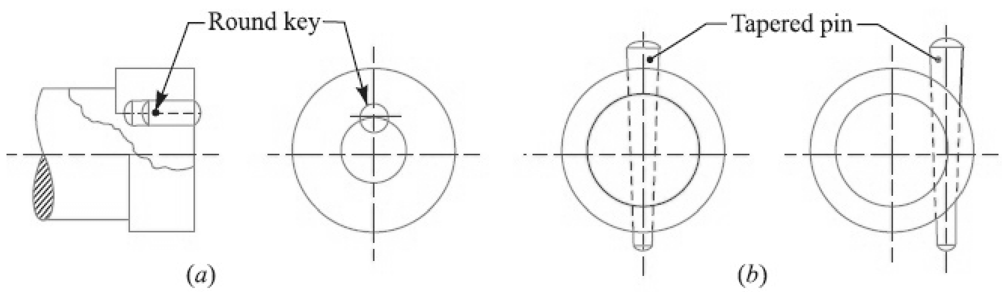


$t = w / 3 = d / 12$

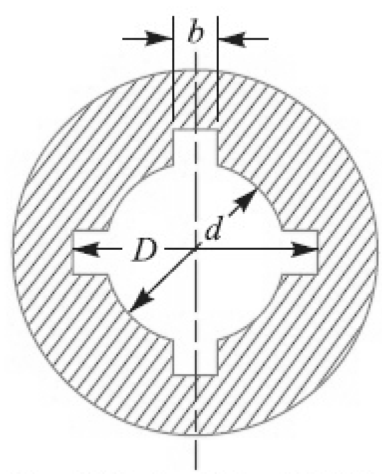
Saddle keys



Tangent Keys

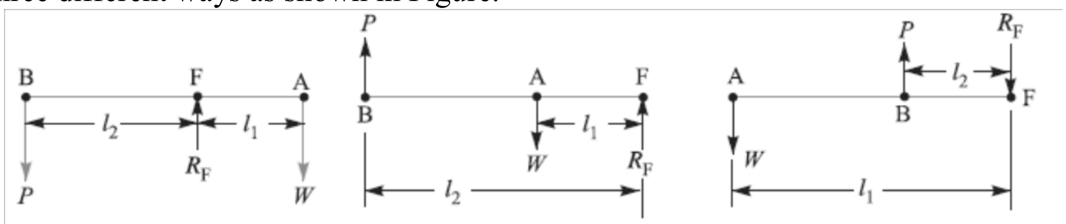


Round Keys



$D = 1.25\,d$ and $b = 0.25\,D$

Splines

1	**B**	**Attempt <u>any One</u> of the following;**	**06**
1B	**a**	**State any three types of levers with neat sketch. Also mention application of each.**	**06**
		Answer: *(Figure-3 marks, explanation with example-3 marks)* **Types of leaver:** First type, second type and third type levers shown in figure at (a), (b) and (c) respectively. The load W and the effort P may be applied to the lever in three different ways as shown in Figure.  (*a*) First type of lever. (*b*) Second type of lever. (*c*) Third type of lever. **First type lever:** 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 mechanical advantage obtained is more than one. **Examples:** Such type of levers are commonly found in bell cranked levers used in railway signaling arrangement, rocker arm in internal combustion engines, handle of a hand pump, hand wheel of a punching press, beam of a balance, foot lever etc. **Second type lever:** In the second type of levers, the load is in between the fulcrum and effort. In this case, the effort arm is more than load arm; therefore the mechanical advantage is more than one. **Examples:** It is found in levers of loaded safety valves. **Third type lever:** 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 the mechanical advantage is less than one. **Examples:** The use of such type of levers is not recommended in engineering practice. However a pair of tongs, the treadle of a sewing machine etc. are examples of this type of lever.	**03** **03**
1B	**b**	**State the design procedure for semi-elliptical leaf spring.**	**06**
	Ans.	Let $2W$ = Central load $2L$ = Span of spring b = Width of leaves t = Thickness of leaves n = Total number of leaves l = Length of central band n_f = Number of full length leaves n_g = Number of graduated leaves	

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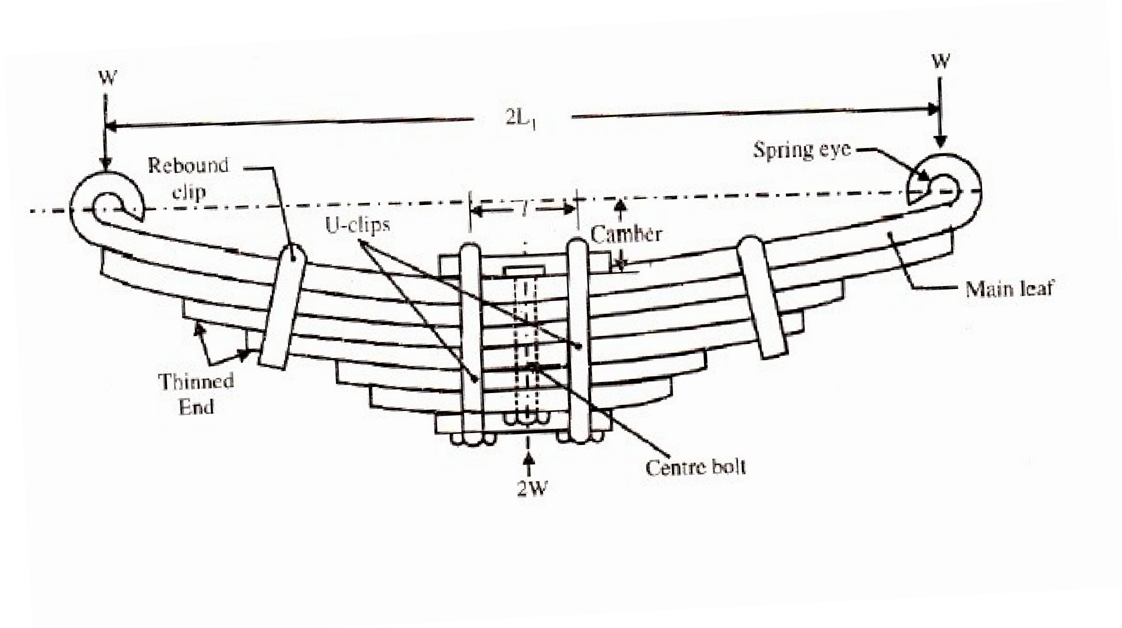


02

Design steps for calculating thickness of Leaf Spring: (any two steps - 1 marks each)

(1) Stress in leaf spring :

$$\sigma_b = \frac{6\,WL}{n\,b\,t^2}$$

where,

Effective length of spring $= 2L = 2L_1 - l$ (when central band is used)

$= 2L = 2L_1 - \frac{2}{3} \cdot l$ (when U-bolt is used)

(2) Deflection in leaf spring :

$$\delta = \frac{6\,WL^3}{n\,E\,b\,t^3}$$

02

(3) Stress in full length leaves :

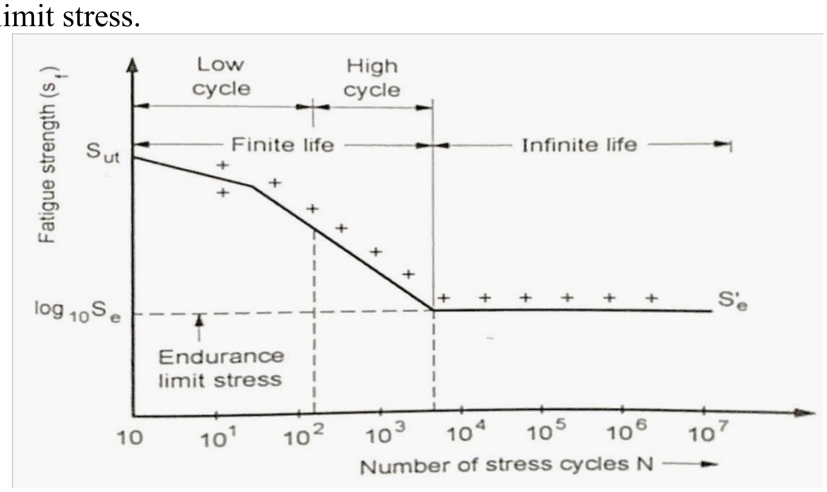
$$\sigma_F = \frac{18\,WL}{bt^2\,(2n_g + 3n_f)}$$

(4) Stress in graduated leaves :

$$\sigma_G = \frac{12\,WL}{bt^2\,(2n_g + 3n_f)}$$

(5) Deflection in full length and graduated leaves :

$$\delta = \frac{12WL^3}{E\,b\,t^3\,(2n_g + 3n_f)}$$

		Design steps for calculating length of Leaf Spring: (2 marks) **Length of smallest leaf = (L x 1)/ (n-1) + l** **Length of second smallest leaf = (L x 2)/ (n-1) + l** **Length of $(n-1)^{th}$ leaf = (L x (n-1))/ (n-1) + l** **Length of master leaf = $2L_1$ + (π (d+t) x 2** **Where d = diameter of Eye.** $d = (32M/ \pi\ \sigma_b)^{1/3}$	02
2		**Attempt <u>any Four</u> of the following**	**16**
2	a	**State the importance of S-N diagram for variable stresses.**	**04**
	Ans.	The S-N curve is the graphical representation of stress amplitude verses the number of stress cycles on a **log – log graph paper**. Each test on fatigue testing machine gives one failure point on the S – N diagram. To help determine when a specimen will fail, S-N diagrams are constructed to compare the fatigue strength (Sf) to the life cycle of the specimen N. For some materials, such as steel and iron, the S-N diagram will become horizontal at a certain point, this point is known as the endurance point S'e. Any fatigue failure when number of stress cycles are less than 1000, is called as low-cycle fatigue. If numbers of stress cycles are more than 1000, is called high-cycle fatigue. Components subjected to low cycle fatigue are designed on the basis of ultimate tensile strength or yield strength with suitable factor of safety. Components subjected to high-cycle fatigue are designed on the basis of endurance limit stress.	**03**
			**01(equi valent sketch)**

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2	**b**	**State the material used for following components with proper justification.** **(i) Piston (ii) Crankshaft**	**04**
	Ans.	**(i) Piston : Cast Aluminum alloys** Since a piston is subjected to very high temperature condition along with extreme and sudden compression and tensile forces on combustion as well as on thrust sides, it calls for a material which has very high strength to weight ratio and has very high heat conductivity in order to minimize thermal fatigue. Usually sand-cast / die-cast aluminium used for engines less than 10 inches of bore diameter. Gray iron is used for small engines and those with bore diameter more than 10 inches. **(iii) Crank Shaft: Medium carbon steel** The crankshaft wants to be as small as possible as much of the engines physical size is dependent on it, a smaller engine is better as it leaves more room for ancillaries and passenger space. The component can be machined and shaped easily, and then hardened afterwards which avoids the issue of machining hard materials.	**02** **02**
2	**c**	**Explain maximum principal stress theory.**	**04**
	Ans.	**Answer:** **Statement:** According to this theory, the failure occurs at a point in a member when the maximum normal stress in a bi-axial stress system reaches the limiting strength of the material in a simple tension test. The maximum or normal stress in a bi-axial stress system is given by, $\sigma_{t1} = \frac{\sigma_{yt}}{F.S.}$, for ductile materials $= \frac{\sigma_{u}}{F.S.}$, for brittle materials σ_{yt} = Yield point stress in tension as determined from simple tension test, and σ_{u} = Ultimate stress. **Brittle material** which are relatively strong in shear but weak in tension or compression, this theory are generally used.	**02** **01** **01**
2	**d**	**State the effect of keyway on transmission shaft strength**	**04**
	Ans.	**Effect of key way cut into the shaft:** The keyway cut into the shaft reduces the load carrying capacity of the shaft. This is due to the stress concentration near the corners of the keyway and reduction in the cross-sectional area of the shaft. It other words, the torsional strength of the shaft is reduced. The following relation for the weakening effect of the keyway is based on the experimental results by H.F. Moore. $e = 1 - 0.2\,(w/d) - 1.1\,(h/d)$	**02**

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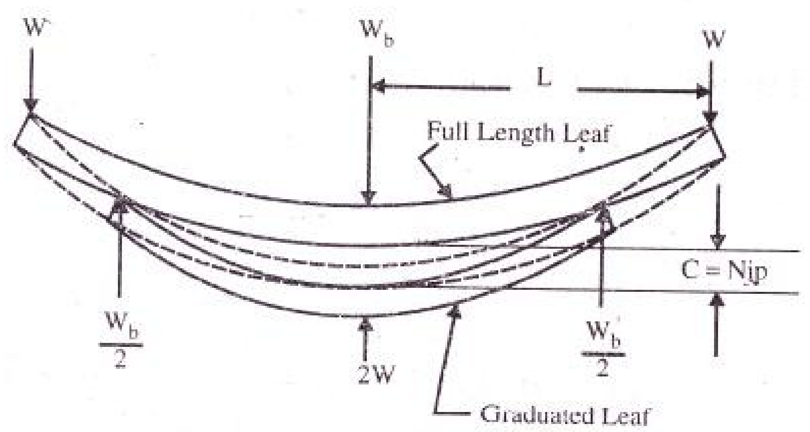
		where, e = Shaft strength factor, w = width of key way, d = diameter of shaft, and h = depth of keyway It is usually assumed that the strength of the keyed shaft is 75% of the solid shaft. In case the keyway is too long and the key is of sliding type, then the angle of twist is increased in the ratio Kθ as given by the following relation: $k_\theta = 1 + 0.4\left(\frac{w}{d}\right) + 0.7\left(\frac{h}{d}\right)$ k_θ = Reduction factor for angular twist.	01 01
2	e	**Explain the concept of hipping.**	04
		Answer: *(Figure- 2mark and explanation-2 mark)* **Nipping:**  The initial gap 'C' between the extra full length leaf and graduated length leaf before the assembly is called as 'Nip'. Such pre-stressing, achieved by a difference in radii of curvature is known as 'Nipping'. It is seen that, stress in full length leaves is 50% greater than the stress in graduated leaves. In order to make best use of material; it is necessary that all the leaves must be equally stressed. This can be achieved by in following two ways: i) By making full length leaves of smaller thickness than graduated leaves. In this way the full length leaves will induce a smaller bending stress due to small distance from neutral axis to edge of the leaf. ii) By giving a greater radius of curvature to the full length leaves than graduated leaves before leaves are assembled to form a spring. By doing so, gap or clearance will be left between the leaves.	02 02

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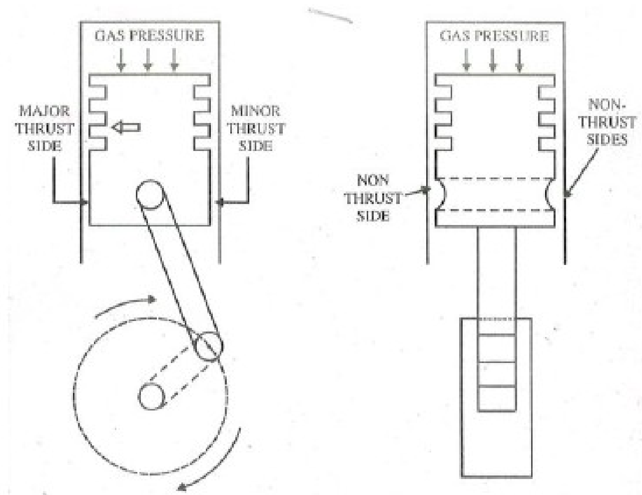
3		Attempt **any four** of the following**:**	**16**
	a)	With neat sketch, show the thrust side and non-thrust side of I.C. engine piston.	**04**
		Ans: *(Each view-2 mark)*  **Thrust side of I.C. Engine piston** **Non-thrust side of I.C. Engine piston**	**04**
3	**b)**	Determine the thickness of plain cylinder head for 300mm cylinder diameter. The maximum gas pressure is 3.2 N/mm^2. Take allowable tensile stress for cylinder cover is 42 N/mm^2 and constant is 0.1.	**04**
		Answer: **Given Data:** **D= 300mm** **P_{max}= 3.2 N/mm^2** σ_t **= 42 N/mm^2** **C= 0.1** **Thickness of plain cylinder head:** $t = D\sqrt{\dfrac{C \times P_{max}}{\sigma t}}$ $t = 300\sqrt{\dfrac{0.1 \times 3.2}{42}}$ $\therefore t = 26.18\ \text{mm}$ $\therefore t \cong 27\ \text{mm}$	**01** **03**

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3	c)	Explain concept of critical speed of shaft.	04
		Answer: **Critical speed of shaft: -** The speed at which the shaft tends to vibrate violently in transverse direction. All rotating shaft, even in the absence of external load, deflect during rotation. The combined weight of a shaft and wheel can cause deflection that will create resonant vibration at certain speeds, known as Critical Speed. **The magnitude of deflection depends upon the followings :-** i. stiffness of the shaft and it's support ii. total mass of shaft and attached parts iii. unbalance of the mass with respect to the axis of rotation iv. the amount of damping in the system Critical speed can be calculated by formula, $N_c = \frac{30}{\pi}\sqrt{\frac{g}{\delta_{st}}}$ Where, N_c= Critical speed g = gravity acceleration δ_{st}=total maximum static deflection	04
3	d)	Determine the dimensions of a rectangular sunk key. The shaft of 100mm diameter to resist a torque of 5000 Nm. The material for shaft key is mild steel. Shear stress is 50 N/mm^2 and crushing stress is 120 N/mm^2.	04
		Answer: **Given data:** **d = 100mm** **σ_{sk}= 50 N/mm^2** **σ_{ck}= 120 N/mm^2** **T = 5000 Nm= 5× 10^6 Nmm** Let, l = length of key, W = width of the key, t = thickness of key, P = tangential force acting at circumference of the shaft P = T / (d/2) = (5000 x 10^3) / (100/2) **P = 1 x 10^5 N** Assuming l = 1.57 d l = 1.57 x 100 **l = 157 mm**	 01 01

		Considering shearing of key, P = l x b x fs $1 \times 10^5 = 157 \times b \times 50$ b = 12.73 mm taking next higher value, **b=13 mm** Considering crushing of key, P = l x (t/2) x fc $1 \times 10^5 = 157 \times (t/2) \times 120$ t = 10.61 mm taking next higher value, **t = 11 mm**	01 01
3	e)	State the design procedure for single plate clutch on the basis of uniform pressure theory.	04

Answer: Answer:
Design procedure of single plate clutch using uniform pressure condition:-

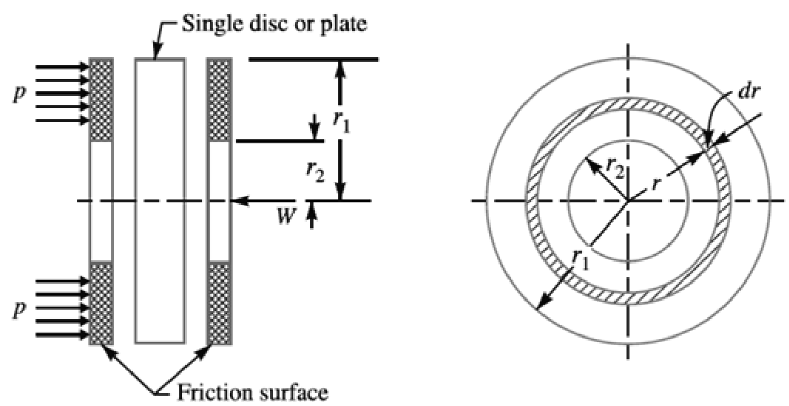

Fig. Forces on a single plate clutch

01

Consider two friction surfaces maintained in contact by an axial thrust (*W*) as shown in Fig.

Let,

W = Axial force/thrust
T = Torque transmitted by the clutch,
p = Intensity of axial pressure
r_1 and r_2 = External and internal radii of friction faces,
r = Mean radius of the friction face, and
μ = Coefficient of friction.
b=face width of frictional surface.

Consider an elementary ring of radius *r* and thickness *dr* as shown in Fig.

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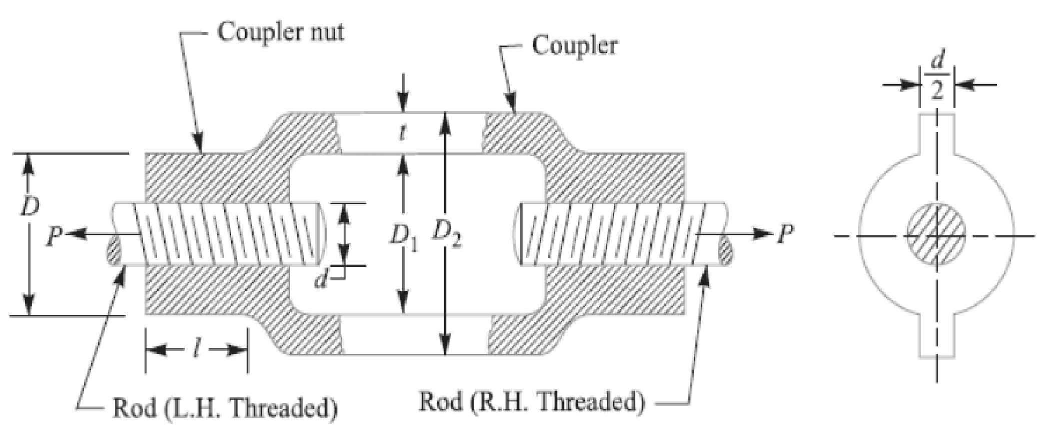
		We know that area of the contact surface or friction surface = ***2π.r.dr*** Therefore Normal or axial force on the ring, δW = Pressure × Area = $p \times 2\pi.r.dr$ and the frictional force on the ring acting tangentially at radius r, $Fr = \mu.\ \delta W = \mu.p \times 2\pi.r.dr$ Therefore Frictional torque acting on the ring, $Tr = Fr \times r = \mu.p \times 2\pi.r.dr \times r = 2\ \pi\mu.p.\ r^2.dr$ **Considering uniform pressure:** When the pressure is uniformly distributed over the entire area of the friction face as shown in Fig. , then the intensity of pressure, $$p = \frac{W}{\pi\left[(r_1)^2 - (r_2)^2\right]}$$ Where, W = Axial thrust with which the friction surfaces are held together. We have discussed above that the frictional torque on the elementary ring of radius r and thickness dr is $$T_r = 2\pi\ \mu.p.r^2.dr$$ Integrating this equation within the limits from r_2 to r_1 for the total friction torque. Total frictional torque acting on the friction surface or on the clutch, $$T = \int_{r_2}^{r_1} 2\pi\ \mu.p.r^2.dr = 2\pi\mu.p\left[\frac{r^3}{3}\right]_{r_2}^{r_1}$$ $$= 2\pi\ \mu.p\left[\frac{(r_1)^3 - (r_2)^3}{3}\right] = 2\pi\ \mu \times \frac{W}{\pi\left[(r_1)^2 - (r_2)^2\right]}\left[\frac{(r_1)^3 - (r_2)^3}{3}\right]$$ ... (Substituting the value of p) $$= \frac{2}{3}\mu.\ W\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right] = \mu\ .W.R$$ where $R = \frac{2}{3}\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right]$ = Mean radius of the friction surface.	03
4	(A)	Attempt **any four** of the following**:**	**12**
	a)	Enlist any four factors affects selection of factor of safety.	**04**
		Answer: *(Any four – 1Marks Each)* **Factors affects on selection of factor of safety:** 1. Degree of economy desired.	**04**

		2. The reliability of applied load and nature of load, 3. The reliability of the properties of material and change of these properties during service, 4. The reliability of test results & accuracy of application of these results to actual machine parts, 5. The certainty as to exact mode of failure, 6. The extent of simplifying assumptions, 7. The extent of localized stresses, 8. The extent of initial stresses setup during manufacture, 9. The extent of loss of property if failure occurs, 10. The extent of loss of life if failure occurs.	
4A	**b)**	State any four design considerations for design of piston.	**04**
		Answer: *(Any four – 1Marks Each)* **Following are the points are taken into consideration for design of the piston:** 1. It should have enormous strength to withstand the high gas pressure and inertia forces. 2. It should have minimum weight to minimize the inertia forces. 3. It should have good and quick dissipation of heat from crown to the rings and bearing area and then to the cylinder walls. 4. It should form an effective gas and oil sealing of the cylinder. 5. It should have sufficient rigid construction to withstand thermal and mechanical distortion. 6. It should provide sufficient bearing area to prevent undue wear. 7. It should have symmetrical design for even expansion under thermal loads, as free as possible from discontinuities. 8. It should have high speed reciprocation without noise. 9. It should have minimum work of friction. 10. It should have a little of no tendency towards corrosion or picking up.	**04**
4A	**c)**	State design procedure of propeller shaft.	**04**
		Answer: **Design procedure of propeller shaft:** The propeller shaft is designed on the basis of torsional shear stress. By using the torsional equation, $$\frac{T_p}{J_p} = \frac{\sigma_s}{r}$$ *Where,* T_p = Torque transmitted by propeller shaft. Tp = Te x G1 Te = Engine Torque.	**04**

		G1 = Maximum gear Ratio in Gear Box Jp = Polar moment of inertia. $= \pi/32 \times d^4$ ……….. …… (for Solid shaft) $= \frac{\pi}{32}(d_o^4 - d_i^4)$……….(for Hollow shaft) σ_s = Torsional shear stress. r = distance from neutral axis to outer most fibres. r = d/2 (for Solid shaft) r = d_o/2 (for Hollow shaft) After simplifying the equations, $T_p = \frac{\pi}{16}\,\sigma_s\,d^3$ …………………For solid shaft $T_p = \frac{\pi}{16}\,\sigma_s\,d_o^3\,(1-k^4)$……….. For hollow shaft $k = \frac{d_i}{d_o}$ d_i = Inner diameter of shaft d_o = Outer diameter of shaft From these equations, we can find out the diameter of propeller shaft.	
	d)	Draw neat labeled sketch of turn buckle.	**04**
		Answer:  **Fig. Turn Buckle**	**04**
	4(B)	Attempt **any one** of the following:	**06**
4B	**a)**	Explain the following terms:	**06**

		i) Concept of standardisation ii) Preferred number.	
		Answer: **i) Standardisation:** It is defined as obligatory norms, to which various characteristics of a product should conform. The characteristics include materials, dimensions and shape of the component, method of testing and method of marking, packing and storing of the product. **Advantages of Standardization:-** 1. Mass production is easy. 2. Rate of production increases. 3. Reduction in labour cost. 4. Limits the variety of size and shape of product. 5. Overall reduction in cost of production. 6. Improves overall performance, quality and efficiency of product. 7. Better utilization of labour, machine and time.	**03**
		ii) Preferred number: • Preferred numbers (also called preferred values) are standard guidelines for choosing exact product dimensions within a given set of constraints. • The system is based on the use of **geometric progressive** to develop a set of numbers. • Preferred numbers are used to specify the specification because a company may manufacture different models of same product. • There are **four basic series**, denoted by **R5, R10, R20, R40** which increases in steps of 58%, 26%, 12%, 6% respectively. Each series has its own series factor given below,	**03**

Series	**Series Factor**
R5	$10^{1/5} = 1.58$
R10	$10^{1/10} = 1.26$
R20	$10^{1/20} = 1.12$
R40	$10^{1/40} = 1.06$

4B	**b)**	State the design procedure of bell crank lever.	**06**
		Answer: **Design procedure of bell crank lever:**	**02**

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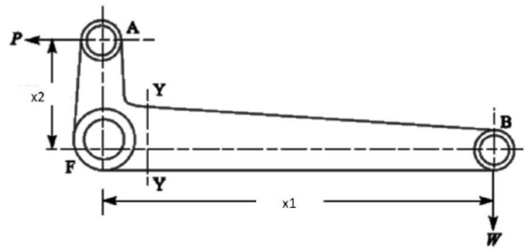
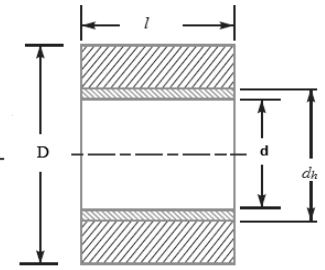


Fig.bell crank lever



Step I: Calculate reaction at the fulcrum pin 01

$$R_F = \sqrt{W^2 + P^2 - 2W \times P \times \cos\theta}$$

$$R_F = \sqrt{W^2 + P^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots cos\theta = 0$$

Step II: Design of fulcrum pin:

(a) Let d = Diameter of the fulcrum pin, and

l = Length of the fulcrum pin

= 1.25 d 01

Considering the bearing of the fulcrum pin. We know that load on the fulcrum pin (R_F),

$$\therefore \text{ Bearing pressure } = \frac{\text{Load}}{\text{Bearing area}} = \frac{R_F}{l \times d} = \frac{R_F}{1.25d \times d}$$

From here, l and d can be determined. 01

(b) Checking shear stress induced in the fulcrum pin. As the pin is in double shear,

$$\tau = \frac{R_F}{2 \times \left(\frac{\pi}{4} \cdot d^2\right)}$$

External diameter of the boss,

$$D = 2\,d$$

Internal diameter of the hole in the lever,

$$d_h = d + (2 \times 3)$$

check the induced bending stress for the section of the boss at the fulcrum

Step III: Design of cross-section of lever:

Bending moment at this section = $W \times X_1$

Section Modulus, 01

$$Z = \frac{l(D^3 - dh^3)}{6D}$$

Induced bending stress,

$$\sigma_b = \frac{M}{Z}$$

Q. No	Sub Q No	Answer	Marking Scheme
5		**Attempt any two of the following: (8 x2)**	**16**
5	a)	**Design a cotter joint to support a load varying from 30 KN in compression to 30 KN in tension. The material used is carbon steel. For which following allowable stresses may be used. The load is applied statically. Tensile stress=Compression stress=50 MPa, Shear stress=35 MPa, Crushing Stress= 90MPa**	**08**
	Ans	**Given:** $P=30x10^3$ N σ_t =50N/mm^2 τ =35N/mm^2 σc=90N/mm^2 Let- d= diameter of rod d1 = outer diameter of socket d2= outer diameter of spigot d3= diameter of spigot collar d4= diameter of socket collar a= distance between end of slot and end of spigot b=width of cotter c=width of socket collar e=width of socket neck t= thickness of cotter t1= thickness of spigot collar l= length of cotter 1.**<u>Find diameter of rod “d” considering failure in tension</u>**: $$P = \frac{\pi}{4}(d)^2 \times \sigma t$$ $30 \times 10^3 = \frac{\pi}{4}(d)^2 \times 50$ **<u>d=27.64 mm say 28 mm</u>** **2. <u>Find outside diameter of spigot “ d2” considering failure in tension :</u>** $$P = [\frac{\pi}{4}(d2)^2 - d2 \times t] \times \sigma t$$ $30 \times 10^3 = [\frac{\pi}{4}(d2)^2 - d2 \times \frac{d2}{4}] \times 50$ $(assuming\ \ t = \frac{d2}{4})$ $30 \times 10^3 = \frac{(d2)^2}{4}[\pi - 1] \times 50$ **<u>d_2=33.46 mm say 34 mm</u>** $t = \frac{d2}{4} = \frac{34}{4} = 8.5 = 9\ mm$ **<u>t=9 mm</u>**	1

		3.<u>Check the crushing stress considering failure at cotter in crushing :</u> $P = (d_2 \times t)\sigma_c$ $30 \times 10^3 = (34 \times 9)\ \sigma_c$ $\sigma_c = 98.03 \frac{N}{mm^2} > Permissible\ crushing\ stress\ 90\ \frac{N}{mm^2}$ So design is not safe.. Hence redesign the values of d_2 & t $P = (d_2 \times t)\sigma_c$ $30 \times 10^3 = (d_2 \times \frac{d_2}{4}) \times 90$ $d_2 = 36.51\ mm = 37\ mm$ $t = \frac{d2}{4} = \frac{37}{4} = 9.25 = 10\ mm$ **<u>d_2= 37 mm and t=10 mm</u>**	1
		4.<u>Find outside diameter of socket "d1 " considering failure socket in tension:</u> $P = [\frac{\pi}{4}({d_1}^2 - {d_2}^2) - (d_{1-}d_2)t] \times \sigma t$ $30 \times 10^3 = [\frac{\pi}{4}(d_1)^2 - \frac{\pi}{4}(37)^2 - (d_1 \times 10) + (37 \times 10)] \times 50$ $\frac{30 \times 10^3}{50} = 0.785(d_1)^2 - 1075.21 - 10d_1 + 370$ $0.785(d_1)^2 - 10d_1 - 1305.21 = 0$ $(d_1)^2 - 12.74d_1 - 1662.69 = 0$ For this quadratic equation- a=1,b=-12.74 & c=-1662.69 $d_1 = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$ $d_1 = \frac{-(-12.74) \pm \sqrt{(-12.74)^2 - 4 \times 1 \times (-1662.69)}}{2 \times 1}$ $d_1 = \frac{12.74 \pm \sqrt{162.30 + 6650.76)}}{2}$ $d_1 = \frac{12.74 + 82.54}{2}$ $d_1 = 47.64\ mm\ say\ 48\ mm$ **<u>d_1= 48 mm</u>**	1
		5.<u>Find the diameter of spigot collar considering failure in crushing :</u> $P = \frac{\pi}{4}({d_3}^2 - {d_2}^2) \times \sigma_c$ $30 \times 10^3 = \frac{\pi}{4}({d_3}^2 - 37^2) \times 90$ $\frac{30 \times 10^3}{90} = \frac{\pi}{4}(d_3)^2 - \frac{\pi}{4}(37)^2$ $333.33 = 0.785(d_3)^2 - 1075.21$ $d_3 = 42.35\ mm\ say\ 43\ mm$ **<u>d_3= 43 mm</u>**	1

		6.<u>Find diameter of socket collar considering failure in crushing :</u> $P = (d_4 - d_2) \times t \times \sigma_c$ $30 \times 10^3 = (d_4 - 37) \times 10 \times 90$ $d_4 = 70.33\ mm = 71\ mm$ **<u>d_4= 71 mm</u>**	
		7.<u>Find the width of cotter "b" considering failure in shear :</u> $P = 2 \times b \times t \times \tau$ $30 \times 10^3 = 2 \times b \times 10 \times 35$ $b = 42.85\ mm = 43\ mm$ **<u>b= 43 mm</u>**	**1**
		8.<u>Find the thickness of spigot collar " t1" by considering failure in shear:</u> $P = \pi \times d_2 \times t_1 \times \tau$ $30 \times 10^3 = \pi \times 37 \times t_1 \times 35$ $t_1 = 7.37\ mm = 8\ mm$ **<u>t_1= 08 mm</u>**	
		9.<u>Find the thickness of socket collar "c" by considering failure in shear :</u> $P = 2(d_4 - d_2) \times c \times \tau$ $30 \times 10^3 = 2(71 - 37) \times c \times 35$ $c = 12.60\ mm = 13\ mm$ **<u>c= 13 mm</u>**	**1**
		10. <u>Find the distance from cotter slot to end of spigot rod "a" by considering failure</u> <u>in shear</u> : $P = 2 \times d_2 \times a \times \tau$ $30 \times 10^3 = 2 \times 37 \times a \times 35$ $a = 11.58\ mm = 12\ mm$ **<u>a= 12 mm</u>**	
		11. <u>Find the length of cotter</u> : $L = 4 \times d$ $L = 4 \times 28$ $L = 112\ mm$ **<u>L= 112 mm</u>**	**1**
		12. <u>Find the thickness of socket of neck"e"</u> : $e = 1.2 \times d$ $e = 1.2 \times 28$ $e = 33.6\ mm = 34\ mm$ **<u>e= 34 mm</u>**	

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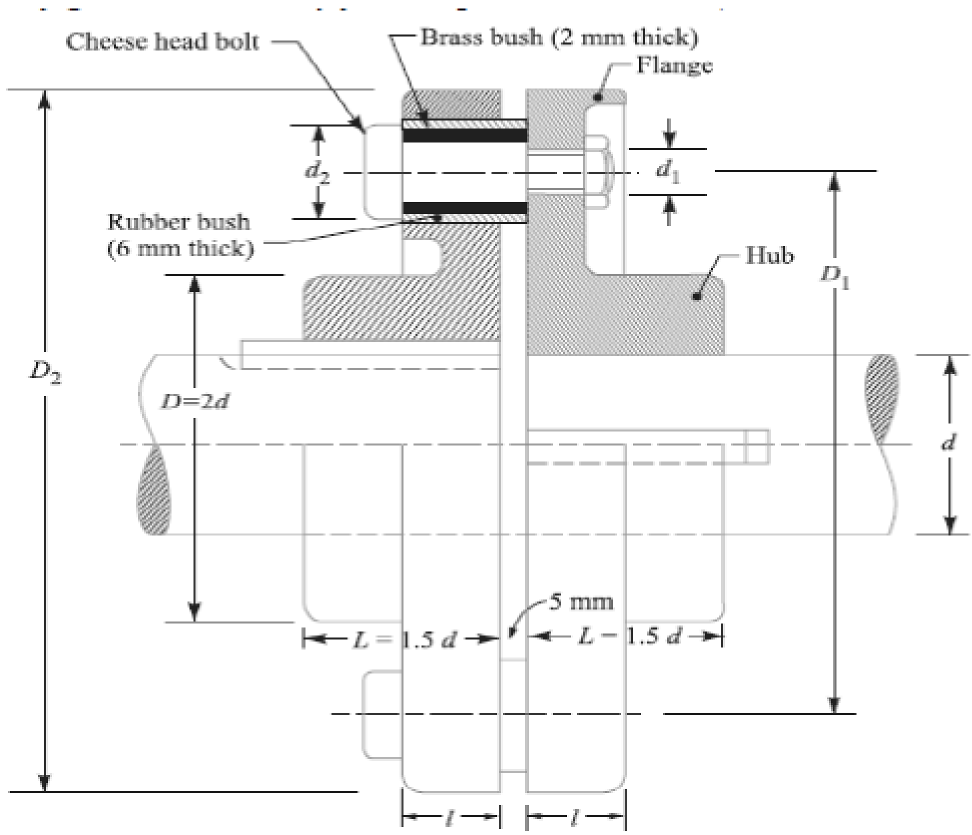
			1
5	**b)**	**State the design procedure for bush pin type flexible coupling.**	**08**
	Ans	***(Neat fig. – 2 Marks, Design procedure – 06 Mark)***  **Figure. Bushed Pin Flexible Coupling** **<u>Design Procedure:</u>** 1. **Design of shaft:** $P = 2\Pi NT/60$ $T = \Pi/16\ \tau\ d^3$ **2.Design of Pin:**	02

Let l = Length of bush in the flange,

d_2 = Diameter of bush,

p_b = Bearing pressure on the bush or pin,

n = Number of pins, and

$n = d/25 + 3$

Diameter of pin $d_1 = 0.5d/\sqrt{n}$

Dia. of pin in rubber bush $d_3 = 1.5d_1$

$d_2 = d_1 + 6$ mm

D_1 = Diameter of pitch circle of the pins.

$= 3d$

We know that bearing load acting on each pin,

$$W = p_b \times d_2 \times l$$

∴ Total bearing load on the bush or pins

$$= W \times n = p_b \times d_2 \times l \times n$$

and the torque transmitted by the coupling,

$$T = W \times n\left(\frac{D_1}{2}\right) = p_b \times d_2 \times l \times n\left(\frac{D_1}{2}\right)$$

Direct shear stress due to pure torsion in the coupling halves,

$$\tau = \frac{W}{\frac{\pi}{4}(d_1)^2}$$

maximum bending moment on the pin,

$$M = W\left(\frac{l}{2} + 5 \text{ mm}\right)$$

We know that bending stress,

$$\sigma = \frac{M}{Z} = \frac{W\left(\frac{l}{2} + 5 \text{ mm}\right)}{\frac{\pi}{32}(d_1)^3}$$ 3

Since the pin is subjected to bending and shear stresses, therefore the design must be checked either for the maximum principal stress or maximum shear stress by the following relations :

		Maximum principal stress $= \frac{1}{2}\left[\sigma + \sqrt{\sigma^2 + 4\tau^2}\right]$ and the maximum shear stress on the pin $= \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2}$ **3. Design of Hub:** The hub is designed by considering it as a hollow shaft, transmitting the same torque (T) as that of a solid shaft. $\therefore \quad T = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D}\right)$ The outer diameter of hub is usually taken as twice the diameter of shaft. Therefore from the above relation, the induced shearing stress in the hub may be checked. The length of hub (L) is taken as 1.5 d. **4. Design of Key:** For rectangular Key, w = d/4, t = d/6 For square key, w = d/4, t = d/4 $T = l \times w \times \tau \times \frac{d}{2}$ ... (Considering shearing of the key) $= l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$ ... (Considering crushing of the key) **5. Design of Flange:** The flange at the junction of the hub is under shear while transmitting the torque. Therefore, the troque transmitted, T = Circumference of hub × Thickness of flange × Shear stress of flange × Radius of hub $= \pi D \times t_f \times \tau_c \times \frac{D}{2} = \frac{\pi D^2}{2} \times \tau_c \times t_f$ The thickness of flange is usually taken as half the diameter of shaft. Therefore from the above relation, the induced shearing stress in the flange may be checked.	1 1 1
5	c)	**A four speed gear box is to be constructed for providing the ratios of 1.0, 1.46, 2.28 and 3.93 to 1 as nearly as possible. The module of gear is 3.25mm and the smallest pinion is to have at least 15 teeth. Determine the suitable number of teeth of different gear. Also calculate the distance between main shaft and lay shaft.**	08
	Ans	**<u>First gear ratio:</u>**	

2

$$G_1 = \frac{T_B}{T_A}\frac{T_D}{T_C} = 3.93$$

We have, $\frac{T_B}{T_A} = \frac{T_D}{T_C} = \sqrt{3.93} = 1.98$

Adopting $T_A = T_C = 15$ the lowest value given,

we get $T_B = T_D = 1.98 \times 15 = 29.7 = 30$

Thus actual ratio $= \frac{30}{15} \times \frac{30}{15} = 4:1$

and $T_A + T_B = T_C + T_D = T_E + T_F = T_G + T_H = 45.$

Second gear ratio:

$$G_2 = \frac{T_B}{T_A}\frac{T_F}{T_E} = 2.28$$ 2

or $\frac{T_F}{T_E} = 2.28\frac{T_A}{T_B} = 2.28 \times \frac{15}{30} = 1.14$

Hence, $T_E + T_F = 2.14\,T_E = 45$

The actual ratio $= \frac{30}{15} \times \frac{24}{21} = 2.286:1$

Third gear ratio:

$$G_3 = \frac{T_B}{T_A}\frac{T_H}{T_G} = 1.46$$ 2

or $\frac{T_H}{T_G} = \frac{1.46}{2} = 0.73$

But $T_H + T_G = 45.$

or $T_G = \frac{45}{1.73} = 26$ 1

Hence, $T_H = 45 - 26 = 19$

Actual ratio $= \frac{30}{15} \times \frac{19}{26} = 1.461:1.$ 1

Top gear ratio:

$G_4 = 1:1$

The distance between main shaft and lay shaft:

= (3.25*45) / 2

=73.125 mm

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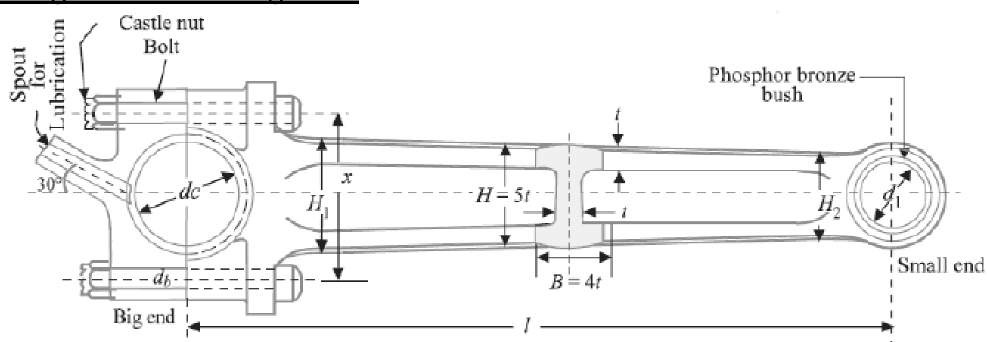
6		**Attempt any two of the following: (8 x2)**	**16**
6	**a)**	**Write design procedure for connecting rod.**	**08**
	Ans	**(Figure-2mark, Design procedure-6 marks)**	

Design of Connecting Rod:



2

1. **Dimensions of cross-section of the connecting rod:**
   According to Rankine's formula:

$$W_B = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{xx}}\right)^2}$$

Let-

WB = Buckling load (Maximum gas load)

σc = Crippling or Buckling stress or crushing stress.

A = Cross-sectional area of the connecting rod=11 t^2

a = Rankine's constant

L =Effective length of the connecting rod

(i.e distance between centres of crank pin and piston pin)

$k_{xx}{}^2 = 3.18\ t^2$ from this relation t (thickness of the flange and web of the section) can be determined.

Width of the section, *B* = 4 *t* and

Depth or height of the section, *H* = *5t*

The dimensions *B* = 4 *t* and *H* = 5 *t,* as obtained above by applying the Rankine's formula, are at the middle of the connecting rod.

The width of the section (*B*) is kept constant throughout the length of the connecting rod, but the depth or height varies.

The depth near the small end (or piston end) is taken as ***H*1 = 0.75 *H* to 0.9*H***

The depth near the big end (or crank end) is taken ***H*2 = 1.1*H* to 1.25*H***.

2. **Dimensions of the at the big end and small end of connecting rod:** 2

Maximum gas force:

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		$$F_L = \frac{\pi D^2}{4} \times p$$ Where- D = Cylinder bore or piston diameter in mm, and p = Maximum gas pressure in N/mm2 Let d_c = Diameter of the crank pin in mm, l_c = Length of the crank pin in mm, pb_c = Allowable bearing pressure in N/mm2, and d_p, l_p and pb_p = Corresponding values for the piston pin **load on the crank pin** = Projected area × Bearing pressure = dc . lc . pb_c (ii) **Similarly, load on the piston pin** = dp . lp . pbp (iii) Equating equation (i) and (ii), we have, FL = dc · lc · pbc Taking lc = 1.25 dc to 1.5 dc, the value of dc and lc are determined from the above expression. Again, equating equations **(i)** and **(iii)**, we have, FL = dp · lp · pbp Taking lp = 1.5 dp to 2 dp, the value of dp and lp are determined from the above expression **3. <u>Size of bolts for securing the big end cap</u>:** FI = Inertia load acting on bolts Let *dcb* = Core diameter of the bolt in mm, σt = Allowable tensile stress for the material of the bolts in MPa, and *nb* = Number of bolts. Generally two bolts are used. Force on the bolts: $$F_I = \frac{\pi}{4}(d_{cb})^2 \sigma_t \times n_b$$ From this expression, *dcb* is obtained. The nominal or major diameter (*db*) of the bolt is given by $$d_b = \frac{d_{cb}}{0.84}$$ **4. <u>Thickness of the big end cap:</u>** The thickness of the big end cap (*tc*) may be determined as below, Maximum bending moment acting on the cap will be taken as	2 1

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		$$M_C = \frac{F_I \times x}{6}$$ Where, x = Distance between the bolt centers. = Dia. of crankpin or big end bearing (dc) + 2 × Thickness of bearing liner(3 mm) + Clearance(3mm) Let, bc = Width of the cap in mm. It is equal to the length of the crankpin or big end bearing (lc), and σb = Allowable bending stress for the material of the cap in MPa. Section modulus for the cap, $$Z_C = \frac{b_c (t_c)^2}{6}$$ $\therefore$ Bending stress, $\sigma_b = \frac{M_C}{Z_C} = \frac{F_I \times x}{6} \times \frac{6}{b_c (t_c)^2} = \frac{F_I \times x}{b_c (t_c)^2}$ From this expression, the value of tc is obtain.	1
6	b)	**A four stroke diesel engine has following specifications:** **B.P.-=5KW, Speed=1200 rpm, Indicated mean effective pressure= 0.35 N/mm^2 and Mechanical efficiency 80% Determine: i) Bore and length of cylinder ii) Thickness of cylinder head.** **Assume: l = 1.5 D OR l = 1.08 D , Constant C=0.1 , Tensile stress for cylinder cover=52 N/mm^2**	08
	Ans	**(Note: Assume l = 1.5 D OR l = 1.08 D , Constant C=0.1 , Tensile stress for cylinder cover=52 N/mm^2)** **Given:** B.P. = 5kW= 5000 W N= 1200 r.p.m. n= N/2 = 600 Pm=0.35 N/mm^2 η_m=80%=0.8 for cylinder cover σ_t =52 N/mm^2 **1. <u>Bore and length of cylinder</u> :** Let D= Bore of the cylinder in mm A= across section area of the cylinder $= \pi/4d^2$ l= length of the stroke in m. = 1.5 D mm = 1.5 D/ 1000 m We know that the indicated power $I.P. = B.P./\eta_m = 5000/0.8 = 6250w$ We also know that the indicated power (I.P.)	1 1 1

		$6250=\frac{P_{m}.l.A.n}{60}=\frac{0.35\times1.5D\times\Pi D^2\times600}{60\times1000\times4}=4.12\times10^{-3}D^3$ $\therefore \quad D^3=6250/4.12\times10^{-3}=1517\times10^{-3}$ or $D=115$ mm $l=1.5D=1.5\times115=172.5$mm	1
		Taking a clearance on both sides of the cylinder equal to 15 % of the stroke therefore length of the cylinder. $L=1.15l=1.15\times172.5=198$ say 200 mm	1
		2. <u>Thickness of the cylinder head</u> : Since the maximum pressure (P) in the engine cylinder is taken as 9 to 10 times means effective pressure (Pm) therefore let us take $P=9P_m=9\times0.35=3.15N/mm^2$	1
		We know that thickness of the cylinder head $th = D\sqrt{\frac{C\times P}{\sigma_t}}$	1
		(Taking C= 0.1 and σ_t = 52 N/mm²) $th = 115\sqrt{(0.1\times3.15)/52}$ $th = 8.95\ mm\ say\ 09\ mm$	1
6	c)	**Design the piston pin with following data: i) Maximum pressure on the piston=4 N/mm2** **ii) Diagram of piston=70 mm iii) Allowable stresses due to bearing, bending and shear are 30 N/mm², 80 N/mm² stress 60 N/mm² respectively.**	08
	Ans	**Answer:** Given data, Dia. of piston = D = 70 mm. Max. pressure = Pmax = 4 N/mm2 Bearing pressure P_b = 30 N/mm2 Bending stress = σ_b = 80N/mm2 Shearing stress = τ = 60 N/mm2 **<u>Maximum gas load</u>**: $F=\frac{\pi}{4}(D)^2\times Pmax$ $F=\frac{\pi}{4}(70)^2\times4$ **<u>F = 15.3938 x 10³ N</u>**	1
		1. <u>Design the piston pin on the basis of bearing pressure</u>: Let, dpo = outer dia. of piston pin l_p = length of piston pin in small end of connecting rod l_p = 0.45xD = 0.45x70 l_p = 31.5 mm F = dpo x lp x Pb	

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$$dpo = \frac{15.3938 \times 10^2}{31.5 \times 30}$$

d_{po} = 16.29mm

d_{po} = 17mm — 2

2. Designing the piston pin on the basis of bending:

'Bending moment 'M' is calculated as

$$M = F \times \frac{D}{8}$$

$$M = \frac{15.3938 \times 10^3 \times 70}{8}$$

M = 134.69 x 10^3 N-mm

We know that,

$$M = \frac{\pi}{32} \times \sigma_b \times (d_{po})^3$$

$$134.69 \times 10^3 = \frac{\pi}{32} \times \sigma_b \times (17)^3$$

σ_b = 279.2589 N/mm² — 1

The induced bending stresses are greater than permissible bending stress 80N/mm^2 hence redesign is necessary. Now redesign value of dpo

$$M = \frac{\pi}{32} \times \sigma_b \times (d_{po})^3$$

$$134.69 \times 10^3 = \frac{\pi}{32} \times 80 \times (d_{po})^3$$

d_{po} =25.79 mm **d_{po} = 26 mm**

3. Designing piston pin on the basis of shear stress : — 1

$$F = \frac{2\pi}{4} \times (d_{po})^2 \times \tau$$

$$15.39 \times 10^3 = \frac{2\pi}{4} \times (26)^2 \times \tau$$

τ= 14.49 N/mm² — 2

The induced shear stresses are less than permissible shear stress. Hence Design is safe

4. The total length of piston is taken as : — 1

L_{pt} = 0.9 D = 0.9 x 70 = **63 mm**