SUMMER– 18 EXAMINATION

Model Answer 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 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. N	Sub Q.N	Answer	Marking Scheme
1	A	**Attempt any THREE**	12
	a)	**Explain Ergonomics aspects of machine design.**	04
	Ans.	**Ergonomic aspects of machine design:** The word 'ergonomics' is coined from two Greek words ergon = work and nomos = natural laws. Ergonomics means the natural laws of work. Anthropometry, Physiology and psychology are the components of ergonomics. **Anthropometry:** With the help of anthropometry dimensions of the components are finalized so that they can be easily handled by operator without fatigue and with consistence efficiency for e.g. diameter of steering wheel, distance from chair to pedals. **Physiology:** With the help of physiology components are designed to be operated by hand or foot force. For e.g. Gear shifting, Steering wheel are designed to be operated by hand because they require speed and accuracy which is imparted by hand and brake pedal clutch pedal etc. are designed to be operated by foot force because they require great amount of force is require than accuracy. **Psychology:** Psychology affects mode of operation for e.g. size, colour and push operation of emergency stops button of any machine. The size of emergency control is made large and painted in red so that they can be easily identified and always they are push operated. All these components make design of automobile components user friendly.	04
	b)	**List the stresses induced in cotter with the stress equation. Also write any two applications of the joint.**	04
		The stresses induced in cotter with the stress equation: Emperical Relation $t = 0.31 \times d$ **Cotter in double shear stress:** $P = 2 \times b \times t \times \sigma_t$ **Cotter in Crushing**: $P = (d2Xt)\sigma_c$	02

<table>
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Bending failure:

$$\sigma_b = \frac{P \times [d_4 + 0.5 d_2]}{2tb^2}$$

OR

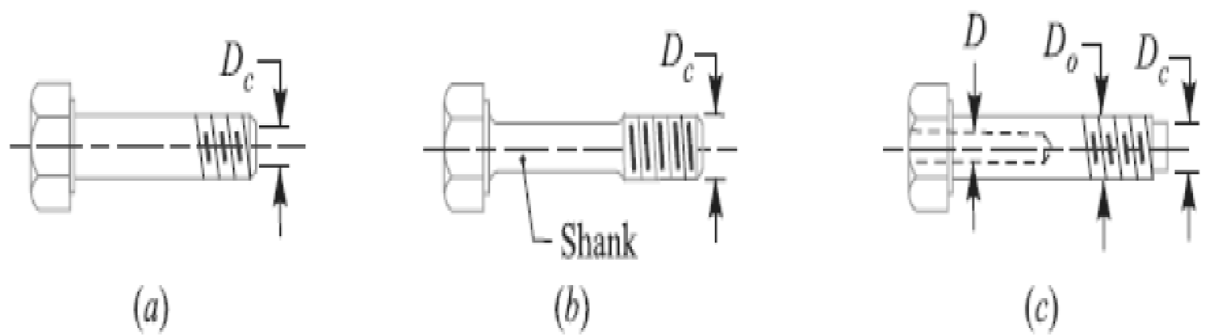
$$b = \sqrt{\frac{3P}{t \times \sigma_b}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]}$$

Applications of Cotter joint:

1. Connecting a piston rod to cross head of steam engine
2. Joining a tail rod with piston rod of an air pump
3. Valve rod and its stem.

</td><td>**02**</td></tr>
<tr><td></td><td>**c)**</td><td>**Describe the concept of "Bolts of uniform strength".**</td><td>**04**</td></tr>
<tr><td></td><td>**Ans.**</td><td>

Bolts of uniform strength:



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

In order to make the bolt of uniform strength, the shank of the bolt is reduced in diameter. the shank diameter can be reduced in following two manners:

1. If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (Dc) as shown in Fig. (b), then shank of the bolt will undergo a higher stress. This means that a shank will absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us bolts of uniform strength. The resilience of a bolt may also be increased by increasing its length.
2. A second alternative method of obtaining the bolts of uniform strength is shown in Fig. (c). In this method, an axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread.

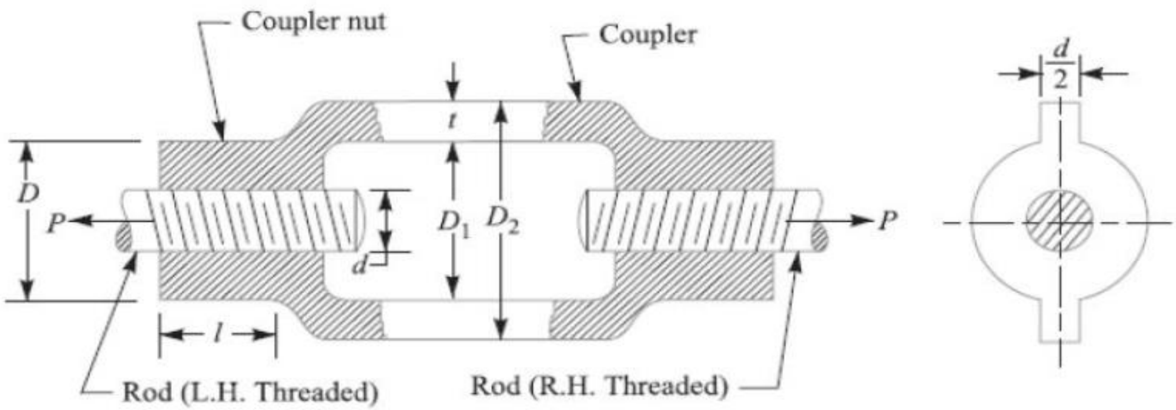
Let D = Diameter of the hole.
D_o = Outer diameter of the thread, and
D_c = Root or core diameter of the thread.

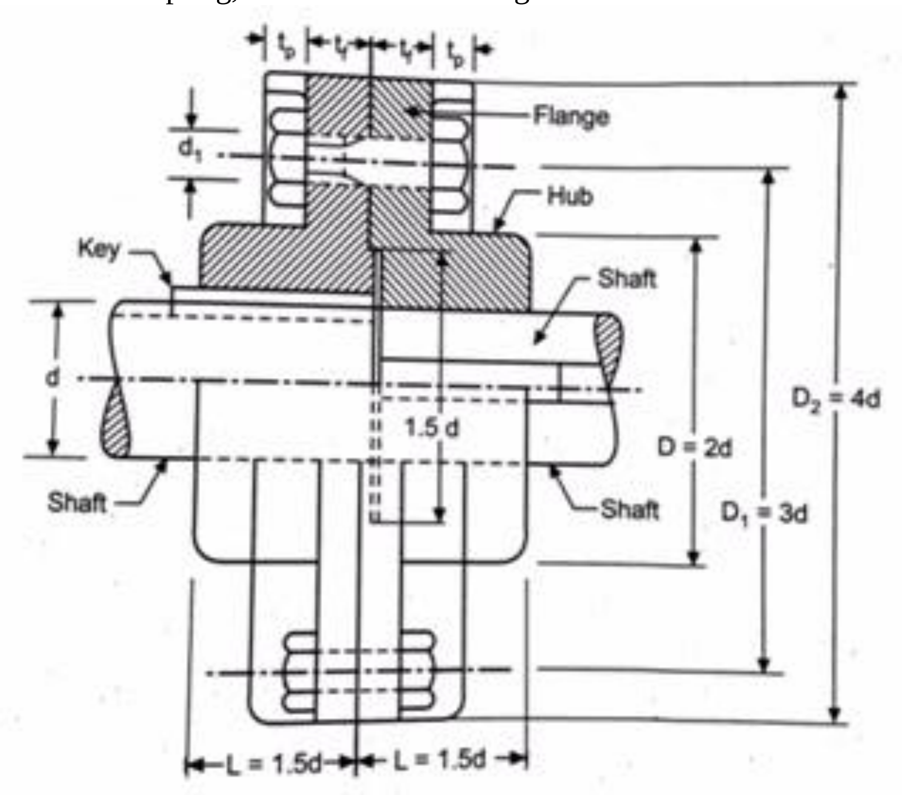
$$\therefore \quad \frac{\pi}{4}D^2 = \frac{\pi}{4}\left[(D_o)^2 - (D_c)^2\right]$$

$$D^2 = (D_o)^2 - (D_c)^2$$

$$\therefore \quad D = \sqrt{(D_o)^2 - (D_c)^2}$$

</td><td>**04**</td></tr>
</table>

	d)	**Design the rectangular key for a shaft of 50 mm diameter. The available shear and crushing stresses for key material are 42 MPa and 70 MPa respectively.**	**04**
	Ans.	$d = 50mm$ $\sigma_{sk} = 42\ N/mm^2$ $\sigma_{ck} = 70\ N/mm^2$ i) Width and thickness for rectangular key: $\therefore w = \frac{d}{4} = \frac{50}{4} = 12.5 \cong 13mm$ $\therefore t = \frac{d}{6} = \frac{50}{6} = 8.33 \cong 9mm$ ii) Torque transmitted by key: $T = \frac{\pi}{16}\ \sigma_s\ d^3$ Assume, σ_s for shaft $= 42\ N/mm^2$ $T = \frac{\pi}{16} \times 42 \times 50^3$ $\therefore T = 10.308 \times 10^5\ Nmm$ iii) Length of key by considering failure in shear: $T = l \times w \times \sigma_{sk} \times \frac{d}{2}$ $10.308 \times 10^5 = l \times 13 \times 42 \times \frac{50}{2}$ $\therefore l = 75.51mm$ $\therefore l \cong 76mm$ iv) Length of key by considering failure in crushing: $T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ $10.308 \times 10^5 = l \times \frac{9}{2} \times 70 \times \frac{50}{2}$ $\therefore l = 130.79mm$ $\therefore l \cong 131mm$ Take maximum value of length of key for safe design $\therefore l \cong 131mm$	01 01 02
1	**B**	**Attempt any one**	**06**
	a)	**Draw a neat sketch of turn buckle joint. Design the turn buckle tie rod diameter only to withstand a load of 2000N. Permissible stresses are $\varepsilon_t = 70\ N/mm^2$, $\tau_s = 60\ N/mm^2$.**	**06**
	Ans.	**Figure. Turn buckle joint**	03

		Let D = diameter of rod Dc =core diameter of rod D =diameter of coupler nut D1 = inside diameter of coupler at centre D2 = outer diameter of coupler at centre l = Length of screw L = length of coupler L1 = length of threaded portion in each rod σ_t , σ_C & τ are permissible tensile, crushing and shear stresses. P=2000 N Permissible tensile Stress= 70 N/mm^2 Permissible Shear Stress= 60 N/mm^2 **Design Load** $P_d = 1.3\,P$ **= 1.3 X 2000 = 2600 N** **Now Design of Tie Rod:** $P_d = (\frac{\pi}{4} dc^2 \times \sigma_t)$ $2600 = (\frac{\pi}{4} dc^2 \times 70)$ d_c=6.87 mm AS d_c=0.84 d_o d=6.87/0.84= 8.77 = 9 mm	01 01 01
	b)	**Write the stepwise design procedure for the design of protective type flange coupling.**	**06**
	Ans.	**The stepwise design procedure for the design of protective type flange coupling:** In a protected type flange coupling, the protruding bolts and nuts are protected by flanges on the two halves of the coupling, in order to avoid danger to the workman.  **Figure - Protective type flange coupling**	01

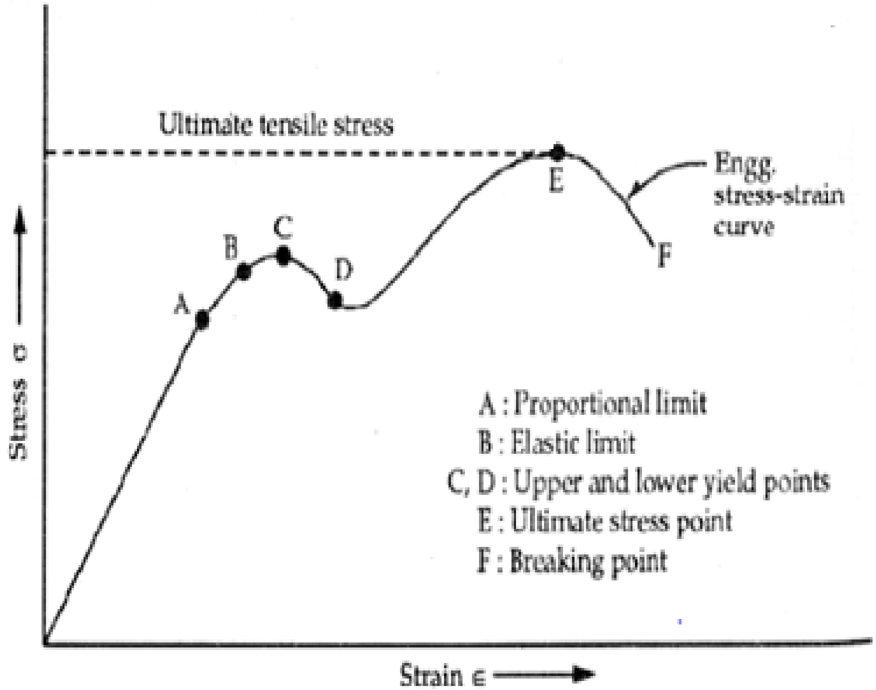
		Design Procedure of flange coupling Consider a flange coupling as shown in figure. Let, d = Diameter of shaft or inner diameter of hub D = Outer diameter of hub D= 2d d_1 = Nominal or outside diameter of bolt D_1 = Pitch circle diameter of bolts, $D_1 = 3d$ Length of the hub, L= 1.5d Outside diameter of flange, $D_2 = D_1 + (D_1 - D)$ $D_2 = 2D_1 - D = 4d$ Thickness of flange, t_f=0.5d Number of bolts = 3 for d up to 40 mm = 4, for d up to 100 mm = 6, for d up to 180 mm t_p = Thickness of protective circumferential flange = 0.25d N= Number of bolts t_f = Thickness of flange τ_s= Allowable shear stress for shaft τ_b = Allowable shear stress for bolt τ_k = Allowable shear stress for key τ_c = Allowable shear stress for flange material i.e. cast iron σ_{cb} = Allowable crushing stress for bolt σ_{ck} = Allowable crushing stress for key	**01** **01** **01** **01**
		1 Design of Shaft The Shaft is designed on the basis of torque equation $T = (\frac{\pi}{16}\tau_s \times d^3)$ $P = 2\pi NT/60$ **2 Design of Hub** The hub is designed by considering it as a hollow shaft, transmitting the same torque (T) as that of solid shaft. $T = \frac{\pi}{16} X \tau_c [\frac{D^4-d^4}{D}]$ The outer diameter of hub is usually taken as twice the diameter of shaft.D=2d From above relation, the induced shearing stress in the hub may be checked. The length of hub (L) = 1.5 d **3 Design of Key** Design of key with usual proportions of crushing and shearing stresses induced. Length of key = Length of hub **4 Design of Flange** Considering shear failure of flange, we get Torque transmitted, T= Circumference of hub X Thickness of flange X Shear stress of flange X Radius of Hub $T = \pi D X t_f X \tau_c X \frac{D}{2}$ Where, t_f = Thickness of flange = d/2 From above equation, shear stress induced in the flange is checked. **5 Design for bolts** a) **Considering, shearing failure of bolts** Load on each bolt = $(\Pi/4)X(d_1)^2 X \tau_b$ Total load on all (i.e. n) bolts $= \Pi/4 (d_1)^2 X \tau_b X n$	**01**

		Torque transmitted $T = \frac{\pi}{4}(d_1)^2 X \tau_b X\ n\ X \frac{D_1}{2}$ From above equation, diameter of bolt (d1) may be obtained. b) **Considering , crushing failure of bolts** The area resisting crushing of all the bolts = n X d_1 X t_f And crushing strength of all the bolts = (nXd_1 X t_f X σ_{cb}) $\frac{D_1}{2}$ From above equation, the induced crushing stress in the bolts may be checked.	
2		**Attempt any four**	**16**
	a)	**Explain any eight design considerations in machine design.**	**04**
	Ans.	**Design considerations in automobile design: (Any eight)** 1. Types of loads and stresses caused by the load. 2. Motion of parts and kinetics of machine. 3. Material selection criteria based on cost, properties etc. 4. Shape and size of parts. 5. Frictional resistance and lubrication. 6. Use of standard parts. 7. Safety operations. 8. Work shop facilities available. 9. Manufacturing cost. 10. Convenient of assembly and transportation.	**04**
	b)	**Design a knuckle joint for a tensile force of 40 KN. The safe stresses in the parts are 60 N/mm^2 in shear, 80 N/mm^2 in tensile and 50 N/mm^2 in crushing.**	**04**
	Ans.	Given Data: P = 40 × 103 N σ_s = 60 N/mm2 σ_t = 80 N/mm2 σ_c = 50 N/mm2 **i. Find Diameter of rod:-** **P = $\frac{\pi}{4} d^2 \sigma_t$** **40 X 103 = $\frac{\pi}{4} d^2 X\ 80$** **d = 25.23mm** **d = 26 mm** **ii. Find dimensions of fork end, eye end and knuckle pin by empirical relations:-** 1. Diameter of knuckle pin — d_1=d = 26 mm 2. Outer diameter of eye end — d_2=2d = 52 mm 3. Diameter of knuckle pin head or collar — d_3=1.5d = 39 mm 4. Thickness of eye end — t=1.25d = 32.5 mm 5. Thickness of forked end — t_1=0.75d = 19.5 mm 6. Thickness of collar or head — t_2=0.5d = 13 mm	**01**

		iii. Induced stress in knuckle pin:- $\therefore P = 2 \times \frac{\pi}{4} d_1^2 \times \sigma_s$ $\therefore 40 \times 10^3 = 2 \times \frac{\pi}{4} 26^2 \times \sigma_s$ $\therefore \sigma_s = 37.68 \frac{N}{mm^2} < \textit{Permissible shear stress}$ Therefore Design is safe. **iv. Induced stresses in eye end:-** 1. Failure in tension: $\therefore P = (d_2 - d_1)t \times \sigma_t$ $\therefore 40 \times 10^3 = (52 - 26)32.5 \times \sigma_t$ $\therefore \sigma_t = 47.33 \frac{N}{mm^2} < \textit{Permissible tensile stress}$ Therefore Design is safe. 2. Failure in shear: $\therefore P = (d_2 - d_1)t \times \sigma_s$ $\therefore 40 \times 10^3 = (52 - 26)32.5 \times \sigma_s$ $\therefore \sigma_s = 47.33 \frac{N}{mm^2} < \textit{Permissible shear stress}$ Therefore Design is safe. 3 Failure in Crushing P = $d_1\ t\ X\ \sigma_c$ 40X 10^3 = 26 X 32.5 X σ_c $\sigma_c = 47.33 \frac{N}{mm^2}$ < Permissible crushing stress Therefore design is safe. **v. Induced stresses in forked end:-** 1. Failure in tension: $\therefore P = 2(d_2 - d_1)t_1 \times \sigma_t$ $\therefore 40 \times 10^3 = 2(52 - 26)19.5 \times \sigma_t$ $\therefore \sigma_t = 39.44 \frac{N}{mm^2} < \textit{Permissible tensile stress}$ Therefore Design is safe. 2. Failure in shear: $\therefore P = 2(d_2 - d_1)t_1 \times \sigma_s$ $\therefore 40 \times 10^3 = 2(52 - 26)19.5 \times \sigma_s$ $\therefore \sigma_s = 39.44 \frac{N}{mm^2} < \textit{Permissible shear stress}$ Therefore Design is safe. **3 Failure in crushing** P = 2 $d_1 t_1 X\ \sigma_c$ 40X10^3 = 2 X 26 X 19.5 X σ_c σ_c = 39.44$\frac{N}{mm^2}$ < Permissible crushing stress Therefore Design is safe.	01 01 01

c) **Draw stress strain diagram for ductile material and state its importance.** **04**

Ans. **Stress strain diagram for ductile material:** **03**



Importance of Stress-Strain diagram for ductile material:
The most important properties of materials are strength, elasticity, stiffness, ductility etc. From stress-strain diagram, material properties like ultimate strength, elastic limit, ductility etc. can be found out. Hence, these values can be used for designing and selection of proper material for machine design. **01**

d) **Design a propeller shaft to transmit 5 KW at 5000 rpm. With gear box reduction 16:1. Assume permissible shear stress for shaft material is 45 N/mm2.** **04**

Ans. **03**

$P = 5\times10^3$ W, $N = 5000$ rpm

$G_1 = 16:1$, $\sigma_s = 45\ N/mm^2$

Now torque produced by the engine,

$$P = \frac{2\pi N T_e}{60}$$

$$5\times10^3 = \frac{2\pi\times5000\times T_e}{60}$$

$$T_e = 9.549\,Nm = 9.549\times10^3\ Nmm$$

Torque transmitted by the propeller shaft,

$$T_p = T_e\times G_1$$

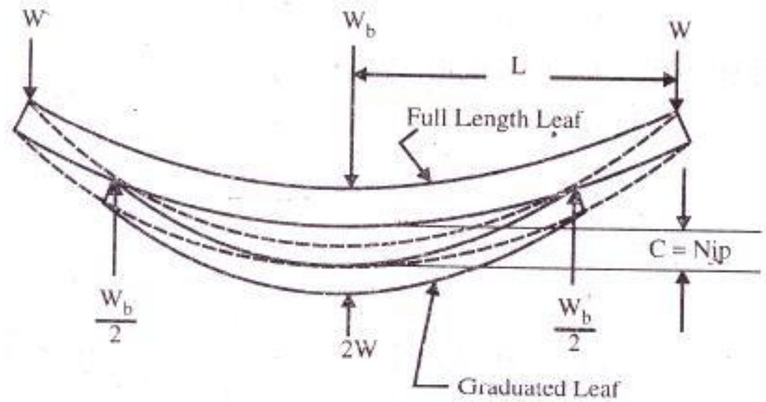
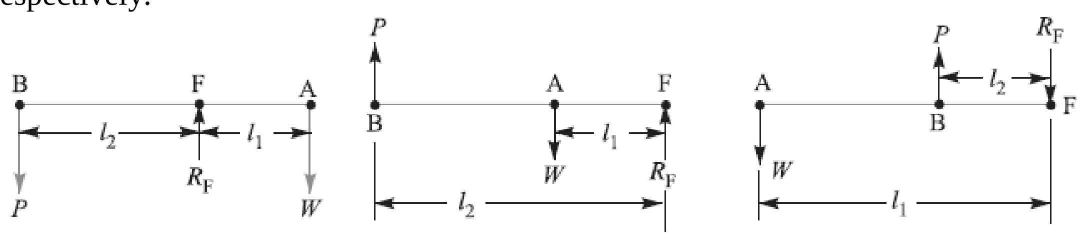
$$T_p = 9.549\times10^3\times16$$

$$T_p = 152.78\times10^3\ Nmm$$

Diameter of propeller shaft,

$$T_p = \frac{\pi}{16}\sigma_s d^3$$

		$152.78 \times 10^3 = \frac{\pi}{16}\, 45\, d^3$ **d=25.86mm** **d= 26 mm**	**01**
	e)	**Explain the term standardization. State any four advantages of it.**	**04**
	Ans.	**Standardization:** - 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:(Any Four)** 1. Interchangeability of product or element is possible. 2. Mass production is easy. 3. Rate of production increases. 4. Reduction in labour cost. 5. Limits the variety of size and shape of product. 6. Overall reduction in cost of production. 7. Improves overall performance, quality and efficiency of product. 8. Better utilization of labour, machine and time.	**02** **02**
3		**Attempt any four.**	**16**
	a)	**Find the diameter of solid shaft to transmit 20kW at 200rpm. The ultimate shear stress for the shaft may be taken as 360N/mm^2 and the F.O.S. as 8.**	**04**
		Answer: Given Data: P= 20kW= 20×10^3W N= 200rpm σ_s=360/8 =45 N/mm^2 Now the torque transmitted by the engine T:- $P = \frac{2\,\pi N T}{60}$ $20 \times 10^3 = \frac{2 \times 3.14 \times 200 \times T}{60}$ $T = 955.41\text{Nm} = 955.41 \times 10^3 \text{ Nmm}$ Let, d = diameter of rear axle, $T = \frac{\pi}{16} f_s\, d^3$ $955.41 \times 10^3 = \frac{\pi}{16} \times 45 \times d^3$ $d^3 = 108130.74$ $d = 47.64\, mm \cong 48\, mm$	**02** **02**

	b)	**Describe nipping of leaf spring with neat sketch.**	**04**
		(Sketch – 2 marks & explanation – 2 marks) **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'. When the central bolt holding the leaves is tightened, the full length leaf bend back as shown by dotted line. And will have an initial stress in opposite direction. The graduated leaves will have an initial stress in the same direction as that of normal load. When the load is applied, the full length leaf gets relieved first; consequently the full length leaf will be stressed less than graduated leaf. The initial leaf between leaves may be so adjusted that under maximum load conditions, all the leaves are equally stressed. So for this reason nipping is provided in leaf spring.	**02** **02**
	c)	**Define Lever. Describe three basic types of leaver.**	**04**
		Answer: ***(Defination-1 mark, Types of lever with description -1 mark each)*** **Definition:-** A lever is a rigid rod or a bar capable of turning about a fixed point called fulcrum. The load W and the effort P may be applied to the lever in three different ways as shown in Figure. **Types of leaver:** First type, second type and third type levers shown in figure at (a), (b) and (c) respectively.  (*a*) First type of lever. (*b*) Second type of lever. (*c*) Third type of lever. **Figure: Types of lever** a) **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. b) **Second type lever:** In the second type of levers, the load is in between the fulcrum	**04**

		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. c) **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.	
	d)	**Explain- Max. principal stress theory.**	**04**
	Ans	**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.	**04**
	e)	**Design the diameter of rear axle shaft for fully floating type with the following data:** **Engine power = 10 kW at 300 rpm.** **Gear Box ratio= 4:1, 2.4:1, 1.5:1, 1:1.** **Differential reduction = 6:1.** **shear stress for shaft material = 70 N/mm^2.**	**04**
		Given:- P = 10 kW = 10×10^3 N = 300 rpm, Max. gear ratio, G_1 = 4 : 1, Differential reduction G_d = 6:1, Shear stress = 70 N/mm^2 **Now the torque transmitted by the engine T_e :-** $P = \frac{2\pi NT}{60}$ $10 \times 10^3 = \frac{2 \times 3.14 \times 300 \times T_e}{60}$ $T_e = 318.47\text{Nm} = 318.47 \times 10^3$ Nmm **Now torque transmitted by rear axle shaft T_{RA},**	**01**

		$T_{RA} = Te \times G_1 \times G_d$ $T_{RA} = 318.47 \times 10^3 \times 4 \times 6$ $T_{RA} = 7643.28 \times 10^3$ Let, d = diameter of rear axle, $T_{RA} = \frac{\pi}{16} f_s d^3$ $7643.28 \times 10^3 = \frac{\pi}{16} \times 70 \times d^3$ $d^3 = 556098.64$ $d = 82.233\, mm \cong 83\, mm$	01 02
4	**(A)**	**Attempt any three.**	**12**
	a)	**Define factor of safety. State the factors affecting its selection.**	**04**
		Answer: ***(Def^n- 2 marks, List of factors- 2 marks.)*** **Factor of Safety:** Factor of safety is defined as the ratio of the maximum stress to the working stress or design stress. Mathematically, $Factor\ of\ Safety = \frac{Maximum\ stress}{Working\ stress}$ In case of ductile material, $Factor\ of\ Safety = \frac{Yeild\ point\ stress}{Working\ stress}$ In case of brittle material, $Factor\ of\ Safety = \frac{Ultimate\ stress}{Working\ stress}$ **The factors that influence the magnitude of factor of safety:*(any two)*** 1. Degree of Economy desired. 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.	**02** **02**
	b)	**Give the application of following joints:** **i) Knuckle Joint ii) Turn Buckle**	**04**
		i) Application of Knuckle joint: ***(Any two – 1 mark each)*** 1. Tie rod joints for roof truss 2. Valve rod joint for eccentric rod pump rod joint 3. Tension link in bridge structure 4. Lever and rod connections	**02**

<table>
<tr><td></td><td></td><td>5. Swing arm of two wheeler
6. Connection of link rod of leaf springs in multi axle vehicles
7. Piston ,Piston Pin ,Connecting Rod
8. Connections of leaf spring with chassis
ii) Application of Turn Buckle: (Any two – 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.</td><td>02</td></tr>
<tr><td></td><td>c)</td><td>Define :
i) Indicated power
ii) Brake power.
iii) Frictional power and state relation between them.</td><td>04</td></tr>
<tr><td></td><td></td><td>Answer: (Each correct definition- 1 mark, Relation between power -1 mark)
i) Indicated power: The power developed inside the cylinder is known as indicated power.
ii) Brake power: This is the actual power delivered at the crankshaft.
iii) Frictional power: Power lost in frictional losses at the working surfaces like bearing, piston rings, valves etc. is known as frictional power.
Relation between Indicated power, Brake power and Frictional power:
(Frictional power = Indicated power - Brake power)</td><td>04</td></tr>
<tr><td></td><td>d)</td><td>Differentiate between hand lever and foot lever.</td><td>04</td></tr>
<tr><td></td><td></td><td>Answer(any four point: 1 mark each)
<table>
<tr><th>Parameter</th><th>Hand lever</th><th>Foot lever</th></tr>
<tr><td>1. Load carrying capacity</td><td>400N</td><td>800N</td></tr>
<tr><td>2. Cross section used</td><td>Circular, Rectangular or cross shaped.</td><td>Circular, Rectangular or cross shaped.</td></tr>
<tr><td>3. Operational method</td><td>Hand operated.</td><td>Foot operated.</td></tr>
<tr><td>4. Applications</td><td>Hand pump, Clutch lever and brake lever of motorcycle, hand brake.</td><td>Rear brake lever of motorcycle, Four wheeler clutch, brake, accelerator lever.</td></tr>
</table></td><td>04</td></tr>
<tr><td></td><td>(B)</td><td>Attempt any one.</td><td>06</td></tr>
<tr><td></td><td>a)</td><td>Design bushed pins only for a flexible coupling to transmit 18 kW at 900 rpm. Diameter of shaft for coupling is 60 mm. Allowing shear and bending stresses in pin are 25 N/mm^2 and 50 N/mm^2 respectively. The allowable bearing pressure in rubber bush in 0.3 N/mm^2.</td><td>06</td></tr>
<tr><td></td><td></td><td>Answer:
Given $P = 18kw = 18\times10^3 watts$
N= 900 r.p.m.</td><td></td></tr>
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		D=60mm $d_1 = 55\text{ mm}$ $f_{sp} = 25\text{N/mm}^2$ $f_{sk} = 40\text{N/mm}^2$ $P_b = 0.3\text{ N/mm}^2$ $f_{bp} = 50\text{N/mm}^2$ **We know that torque transmitted,** $T = \frac{P \times 60}{2\prod N} = \frac{18\times10^3\times60}{2\times3.14\times900}$ $=191\text{N}-\text{m}$ $T = 191\times10^3\text{N}-\text{mm}$ $n = \text{no. of pins}$ $d_1 = \text{diameter of pin at neck}$ $d_3 = \text{diameter of pin in the bush}$ $t_1 = \text{thickness of brass bush}$ $t_2 = \text{thickness of rubber bush}$ $D_1 = \text{diameter of pitch circle of pins}$ =3D =3 x 60= 180 $t_3 = \text{thickness of pin head}$ $d_4 = \text{diameter of pin head}$ We know that $n = \frac{d}{25} + 3$ $\therefore \quad n = \frac{60}{25} + 3$ $\therefore \quad n = 5.4$ **Taking next higher even number** $\therefore \quad n = 6$ Now diameter of pin, $d_1 = \frac{0.5d}{\sqrt{n}} = \frac{0.5\times60}{\sqrt{6}}$ $d_1 = 12.24\text{ mm}$ $d_1 = 13\text{ mm}$ Now $d_3 = 1.5d_1$ $=1.5\times13$ $\mathbf{d_3} = 19.5\text{ mm}$ A brass sleeve of thickness' t_1 ' and a rubber bush of thickness ' t_2 ' is fitted on this pin diameter d_3 $t_1 = 2\text{ mm}$ And $t_2 = 6\text{ mm}$ **Now outer diameter of rubber bush,** $d_2 = d_3 + 2\times t_1 + 2\times t_2$ $=19.5 + 2\times2 + 2\times6$	**01** **01**

		$d_2 = 35.5\,\text{mm}$ **Now pitch circle diameter of pins** $D_1 = 3d$ $= 3\times 60$ $D_1 = 180\ \text{mm}$ Let us assume thickness of pin head. $t_3 = 3\ \text{mm}$ Now diameter of pin head $d_4 = d_2 - t_3$ $= 35.5 - 3$ $d_4 = 32.5\ \text{mm}$ Let **W= load on each pin** L= Length of bush in left hand flange. Now torque $T = W.n.\left(\frac{D_1}{2}\right)$ $\therefore\ W = \frac{T\times 2}{D_1 \times n}$ $\therefore\ W = \frac{191\times 10^3 \times 2}{180.0\times 6}$ $\therefore\ W = 353.7N$ **Bearing pressure on rubber bush,** $P_b = 0.3\ N/mm^2$ given $w = d_2 \times i \times P_b$ $353.7 = 35.5 \times i \times 0.3$ $l = 33.21\,\text{mm}$ $l = 34\,\text{mm}$ Clearance between flanges, C = 0.1 d $= 0.1\times 60$ $= 6mm$ **Stresses in pin** $f_{sp} = \text{direct shear stress in pin}$ $= \frac{W}{\frac{\Pi}{4} d_1^2}$ $= \frac{353.7}{\frac{\Pi}{4}(13)^2}$ $= 2.66 N/mm^2$ **Bending moment in pin** $M = w\left[\frac{l}{2} + C\right]$ $\therefore\ M = 353.7\left[\frac{34}{2} + 6\right]$ $\therefore\ M = 353.7[23]$	01 01

		$\therefore \quad M = 8.135 \times 10^3\, N - mm$ $\therefore$ Bending stress, $F = \frac{m}{z} = \frac{m}{\frac{\Pi}{32}(d_1)^3}$ $\therefore \quad F = \frac{8.135\times10^3}{\frac{\Pi}{32}(13)^3}$ $\therefore \quad F = 37.73 N/mm^2$ Now maximum principal stress induced in pin (Maximum bending stress) $f = \frac{1}{2}\left[F\sqrt{(F)^2 + 4(f_{sp})^2}\right]$ $= \frac{1}{2}\left[37.73 + \sqrt{(37.73)^2 + 4(2.66)^2}\right]$ $= \frac{1}{2}[37.73 + 38.103]$ $= 37.91\, N/mm^2$ This value is less than allowable bending stress in pin (50 N/mm^2), hence design is safe. Now maximum shear stress induced in pin. $f_{s\,max} = \frac{1}{2}\sqrt{(F)^2 + 4(f_s)^2}$ $= \frac{1}{2}\times\sqrt{(37.73)^2 + 4(2.66)^2}$ $= \frac{1}{2}\times 38.10$ $\therefore \quad f_{s\,max} = 19.05\, N/mm^2$ This value is less than allowable shear stress in pin (25 N/mm^2), hence the design is safe.	01 01
	b)	**A four stroke diesel engine has following specifications.** **B.P.-5kW at 1200 rpm** **Indicated mean effective pressure 0.35 N/mm^2** **Mechanical efficiency 80%** **Determine :** **i) Bore and length of cylinder** ii) **Thickness of cylinder head**	06
		Answer: ***(Note: Assume l = 1.5 D OR l = 1.08 D)*** **Given:** B.P. = 5kW= 5000 W N= 1200 r.p.m. or n= N/12 = 600 P_m=0.35N/mm ^{2} $\eta_m = 80\% = 0.8$ **1. Bore and length of cylinder** Let D= Bore of the cylinder in mm A= across section area of the cylinder $= \frac{\Pi}{4}\times D^2 mm^2$ l= length of the stroke in m. = 1.5 D mm = 1.5 D/ 1000 m	

		We know that the indicated power $I.B. = B.P./\eta_m = 5000/0.8 = 6250w$ We also know that the indicated power (I.P.) $6250 = \frac{P_{m.}l.A.n}{60} = \frac{0.35 \times 1.5D \times \Pi D^2 \times 600}{60 \times 1000 \times 4} = 4.12 \times 10^{-3} D^3$ $\therefore \quad D^3 = 6250/4.12 \times 10^{-3} = 1517 \times 10^{-3}$ or $D = 115$ mm $l = 1.5D = 1.5 \times 115 = 172.5$mm Taking a clearance on both sides of the cylinder equal to 15 % of the stroke therefor length of the cylinder. $L = 1.15l = 1.15 \times 172.5 = 198$ say 200 mm	03
		2. **Thickness of the cylinder head** Since the maximum pressure (P) in the engine cylinder is taken as 9 to 10 times means effective pressure (P_m) therefore let us take $P = 9P_m = 9 \times 0.35 = 3.15 N/mm^2$ We know that thickness of the cylinder head, (Taking C= 0.1 and σ_t = 42 MPa = 42 N/mm^2) $t_h = D\sqrt{\frac{C.p}{\sigma_t}} = 115\sqrt{\frac{0.1 \times 3.15}{42}} = 9.96$ say 10 mm	03
5		**Attempt any TWO of the following.**	**12**
	a)	**A truck spring has 12 number of leaves, two of which are full length leaves.the spring supports are 1.05 m aparts and central band is 85 mm wide. The central load is 5.4 kN with a permissible stress of 280 N/mm^2. Determine thickness and width of the steel spring leaves. The ratio of total depth to the depth to the width of the spring is 3. Also determine the deflection of the spring.**	
		Answer: Given data Total number of leaves n=12 Number of full length leaves n_F =2 Number of graduate leaves $n_G = n - n_f = 12-2=10$ Distance between spring supports $2L_1$ = 1.05 m =1050 mm Width of central band l=85 mm Effective length of spring $2L = 2L_1 - l$ =1050 – 85 =965 mm L = 482.5 mm Central Load 2W =5.4 KN =5400 N W=2700 N	

		Permissible stress =280 N/mm^2 Depth /width =3 = (n x t) / b $12\frac{t}{b} = 3$ $b = 4t$ Assuming that leaves are not initially stressed $\sigma_b = \frac{18WL}{bt^2(2n_G + 3n_F)}$ $280 = \frac{18WL}{bt^2(2n_G + 3n_F)}$ $280 = \frac{18 \times 2700 \times 482.5}{4t \times t^2(2 \times 10 + 3 \times 2)}$ $t^3 = 805.27$ t=9.31 mm **Thickness t=10 mm and Width of spring b=4 t =4 x 10 = 40 mm** **Deflection of the spring:** $\delta = \frac{12WL^3}{Ebt^3(2n_G + 3n_F)}$ $280 = \frac{12 \times 2700 \times 482.5^3}{0.21 \times 10^6 \times 40 \times 10(2 \times 10 + 3 \times 2)}$ (Taking E = 0.21 x 10^6 N/mm^2) **Deflection of the spring δ =16.7 mm**	4 4
	(b)	**Design piston pin with following data:** **Max. gas pressure = 4 N/mm^2** **Diameter of piston =70 mm** **Allowable stresses due to bearing, bending and shear are given 30 N/mm^2,80 N/mm^2, 60 N/mm^2 respectively.**	
		Dia. of piston = D = 70 mm. Max. pressure = P_{max} = 4 N/mm2 Bearing pressure P_b = 30 N/mm2 Bending stress = σ_b = 80N/mm2 Shearing stress = τ = 60 N/mm2 **Maximum gas load,** $W = \frac{\pi}{4}D^2 \times P_{max}$	

		$W = \frac{\pi}{4} \times 70^2 \times 4$ $W = 15.39 \times 10^3\ N$ **1.** Design the piston pin on the basis of bearing pressure Let, d_{po} = 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 $F = d_{po} \times l_p \times P_b$ $d_{po} = \frac{15.3938 \times 10^3}{31.5 \times 30}$ $d_{po} = 16.29$mm **$d_{po} = 17$mm** **2.** Designing the piston pin on the basis of bending. 'Bending moment 'M' is calculated as $M = F \times \frac{D}{8}$ $M = \frac{15.3938 \times 10^3 \times 70}{8}$ $M = 134.69 \times 10^3$ N-mm We know that, $M = \frac{\pi}{32} \times \sigma_b \times (d_{po})^3$ $134.69 \times 10^3 = \frac{\pi}{32} \times \sigma_b \times (17)^3$ **$\sigma_b = 279.2589$ N/mm^2** The induced bending stresses are greater than permissible bending stress 80N/mm2 hence redesign is necessary. Now redesign value of d_{po} $M = \frac{\pi}{32} \times \sigma_b \times (d_{po})^3$ $134.69 \times 10^3 = \frac{\pi}{32} \times 80 \times (d_{po})^3$ $d_{po} = 25.79$ mm **d_{po} = 26 mm** **c) Designing piston pin on the basis of shear stress, due to double shear.** $F = 2x\pi/4(Dpo)^2 x\ \tau$ $15.39 \times 10^3 = 2x\ \pi/4 \times 26^2 x\ \tau$ $T = 14.49$ N/mm^2 The induced shear stresses are less than permissible shear stress. Hence design is safe. **d) The total length of piston pin is taken as** $L_{pt} = 0.9D = 0.9x70 = 63$mm	**2 Marks for each point**

3. Designing piston pin on the basis of shear stress.

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

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

$$\boldsymbol{\tau = 14.49\ N/mm^2}$$

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

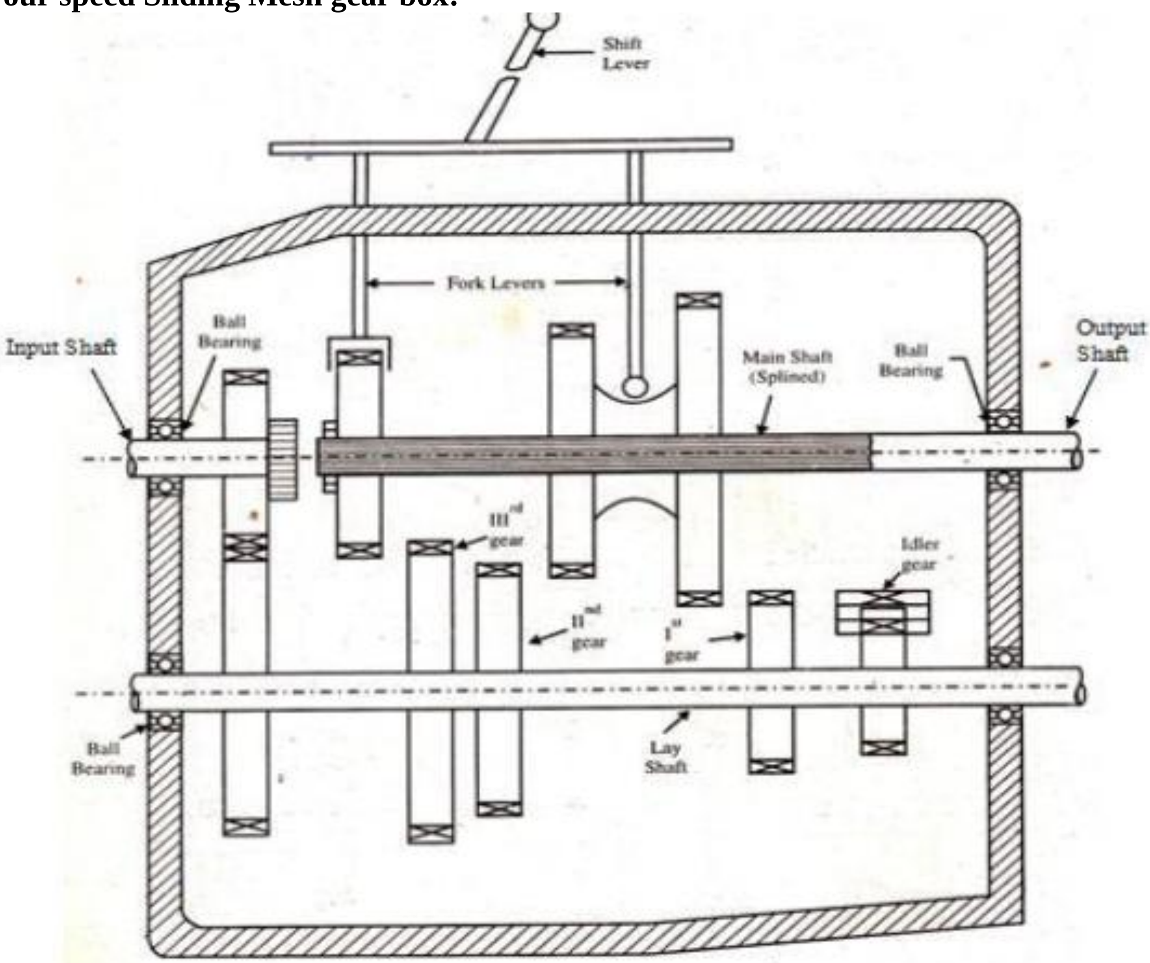
4. The total length of piston is taken as

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

(C) **Draw a neat sketch of sliding mesh gear box and write the design procedure for teeth calculation.**

Answer: (Sketch – 3 marks, Correct Labeling – 2 Mark, design procedure for teeth calculation-3 marks)

Four speed Sliding Mesh gear box:



5

Design procedure for teeth calculation. /TC

First gear ratio: $G1 = \frac{Tb}{TA} X \frac{TD}{TC}$

Second gear ratio: $G2 = \frac{TB}{TA}\ X \frac{TF}{Te}$

		Third gear ratio: G3=1:1 Reverse gear ratio: $\frac{TA}{TB} X \frac{TI}{TG} X \frac{TR}{TL}$	**3**
6		**Attempt any TWO of the following**	
	a)	**A multiple disc clutch has five plates having four pairs of active friction surfaces. If the intensity of pressure is not to exceed 0.127 N/mm^2. Find power transmitted at 500 rpm. The outer and inner radii of friction surfaces are 125 mm and 76 mm respectively. Assume uniform wear and take coefficient of friction =0.3**	
		Answer: Given Data: $n_1 + n_2 = 5$; $n = 4$ P max= 0.127 N/mm^2 Speed of clutch N = 500 rpm Outer radius r_1 = 125 mm Inner radius r_2 = 76 mm Coefficient of friction μ = 0.3 Power transmitted by the clutch P For uniform wear, p.r = C (a constant). **The intensity of pressure is maximum at the inner radius (r_2),** therefore, Pmax x .r_2 = C C = 0.127 × 76 **C = 9.652 N/mm** **Axial force required to engage the clutch,** $W = 2\pi C (r_1 - r_2)$ $W = 2\pi \times 9.652 (125 - 76)$ **W = 2970 N** **Mean radius of the friction surfaces, R = (76+ 125)/2 = 100.5 mm** **The torque transmitted,** T = n.μ.W.R T= 4 × 0.3 × 2970 × 100.5 **T= 358.18 x 10^3 N-mm** **Power transmitted,** $P = (2 \pi n T) / 60$ $P = (2\pi \times 500 \times 358.8) / 60$ **P= 18.74 KW**	 **2** **2** **2** **2**

b) **Explain the design procedure used to design the piston rings and piston skirts.**

(Piston rings 4Marks and piston skirts 4Marks)

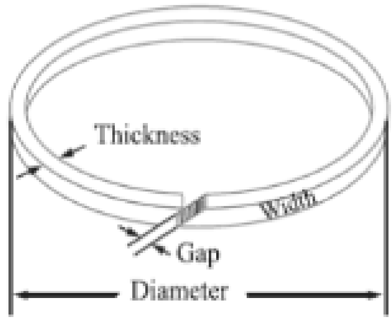


Fig. Piston rings.

The radial thickness (t_1) of the ring may be obtained by considering the radial pressure between the cylinder wall and the ring. From bending stress consideration in the ring, the radial thickness is given by

$$t_1 = D\sqrt{\frac{3p_w}{\sigma_t}}$$

Where, D = Cylinder bore in mm,

p_w = Pressure of gas on the cylinder wall in N/mm^2.

σ_t = Allowable bending (tensile) stress in MPa.

The axial thickness (t_2) of the rings may be taken as 0.7 t_1 to t_1.

The minimum axial thickness (t_2) may also be obtained from the following empirical relation:

$$t_2 = \frac{D}{10\, n_R}$$

Where, n_R = Number of rings.

Width of top land,

$b_1 = t_H$ to $1.2\, t_H$

Width of other ring lands,

$b_2 = 0.75\, t_2$ to t_2

The gap between the free ends of the ring is given by 3.5 t_1 to 4 t_1. **4**

Design of Skirt Length:

R = Normal side thrust acting on piston skirts

Maximum gas load $F = P_{max} \times \frac{\pi}{4} D^2$

R = Normal side thrust acting on piston skirts

$\because$ Side thrust = 10%

$\therefore \quad R = 0.1\,F$

Let,

l_1 = length of piston skirt **4**

The piston skirt act as a bearing inside the liner

We have , $R = l_1 \times D \times P_b$

Where P_b = allowable bearing pressure on the piston skirt

	c)	**Design the connecting rod cross-section with the following data of petrol engine: Max. Pressure inside the cylinder = 4.5N/mm^2, piston diameter =70 mm, Stroke Length= 80 mm , effective length of connecting rod =140 mm and maximum allowable stress in the connecting rod in crippling is 100 N/mm^2 .Take Rankine constant for steel is 1/1600.**	

Answer: Given Data

Max. pressure inside Pmax =4.5 N/mm^2
Piston Dia D=70 mm
Stroke length= l
Effective Length of connecting rod mm L= 140 mm
Maximum allowable stress in the connecting rod in crippling is 100 N/mm^2
Rankine constant for steel is = 1/1600. 1

Step I Max gas load acting on the connecting rod

$$W = P_{max} \times \frac{\pi}{4} \times D^2$$

W = 3846.5 x 4.5 = 17310 N

Area of cross section A = 11 t^2 1

where t = thickness of Flange
a = Rankine constant = 1/ 1600
$K_{xx}^2 = 3.18\, t^2$
$K_{xx} = 1.78\, t$

Step II Critical bucking load acting on the connecting rod

As Factor of safety is not given in example statement, so consider factor of safety as 1)

Critical bucking load = W x FOS Consider FOS =1

Critical bucking load =17310 x 1 = 17310 N

Assuming I section, Max. crippling load is, 4

$$W_{cr} = \frac{\sigma_c \times A}{1 + a[\frac{L}{K_{xx}}]^2}$$

$$17310 = \frac{100 \times 11t^2}{1 + \frac{1}{1600}[\frac{140}{1.78t}]^2}$$

2

t=4.5 mm
Consider thickness t=5 mm
Dimension at the middle or center:
(i) Depth or height of section:
H = 5 t =5 x 5 = 25 mm
(ii) width of cross section
B = 4 t = 4 x5 =20mm
Dimension at the big end (crank end):
(i) Depth or height of section:
At the big end H_2= 1.2H =1.2(25)=30*mm*
(ii) width of cross section
B2=B=20 mm