MAHARASHTRA STATE BOARD OF TECHNICAL EDUCATION
(Autonomous)
(ISO/IEC - 27001 - 2013 Certified)

SUMMER – 19 EXAMINATION

Model Answer

Subject Name: Design of Automobile Components **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.

<table>
<tr><th>Q. No.</th><th>Sub Q. N.</th><th>Answer</th><th>Marking Scheme</th></tr>
<tr><td>1</td><td>(A)</td><td>Attempt any THREE:</td><td>12</td></tr>
<tr><td></td><td>(a)</td><td>What is factor of safety? How it is selected?</td><td>04</td></tr>
<tr><td></td><td>Ans.</td><td>(Definition 02 Marks and Selection Factor any four 02 Marks)
Factor of Safety: Factor of safety is defined as the ratio of the maximum stress to the working stressor design stress.
Mathematically,
$$\text{Factor of Safety} = \frac{\text{Maximum Stress}}{\text{Working or design stress}}$$
In case of ductile materials-
$$\text{Factor of Safety} = \frac{\text{Yield Point Stress}}{\text{Working or design stress}}$$
In case of brittle materials-
$$\text{Factor of Safety} = \frac{\text{Maximum Stress}}{\text{Working or design stress}}$$
Selection of factor of safety depends on (Any Four)
1. The reliability of the properties of the material and change of these properties during service.
2. The reliability of test results and accuracy of application of these results to actual machine parts.
3. The reliability of applied load.
4. The certainty as to exact mode of failure.
5. The extent of simplifying assumptions.
6. The extent of localized stresses.
7. The extent of initial stresses set up during manufacture.
8. The extent of loss of life if failure occurs.
9. The extent of loss of property if failure occurs.</td><td>Definition 02 Marks and Selection Factor any four 02 Marks</td></tr>
</table>

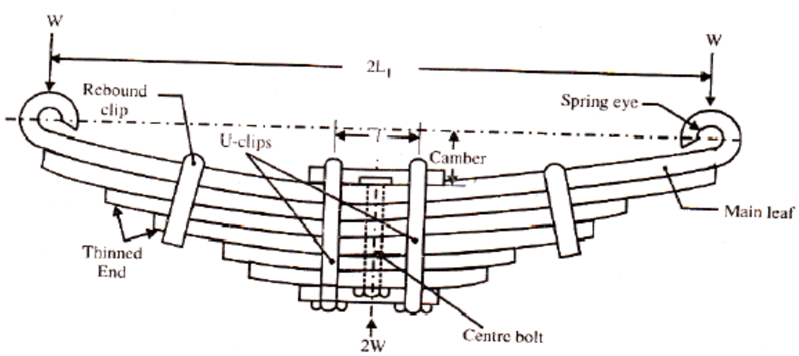
	(b)	**Explain role of ergonomics in automobile design.**	**04**
	Ans.	***(Correct Answer 04 Marks)*** **Role of Ergonomics In Automobile Design:** Anthropometry, Physiology and psychology are the components of ergonomics **Anthropometry:** With the help of anthropometry dimensions of the components are 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, color 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.	*Correct Answer 04 Marks*
	(c)	**State two applications of spigot type cotter joint and turn buckle.**	**04**
	Ans.	***(Any two Applications of Each 01 Mark each)*** **i) Applications of Cotter joint:** • Connecting a piston rod to cross head of steam engine • Joining a tail rod with piston rod of an air pump • Valve rod and its stem. **ii) Applications of Turn Buckle:** • Tie rod of steering system • To connect compartments of locomotives • Tie strings of electric poles. • Link rod of leaf springs in multi axle vehicles • Linkages of gear shifter • Connection between brake pedal and master cylinder	*Any two Applications of Each 01 Mark each*
	(d)	**State types of keys with their applications.**	**04**
	Ans.	***(Types of Keys 02 Marks and their Applications 02 Marks)*** **Types of Keys:** Common types of keys are: **1. Sunk keys** — A. Square sunk key; B. Parallel sunk key; C. Gib-head key; D. Feather key:; E. Woodruff key **2.Saddle keys** — a.Solid Key; b. Hollow Key **3.Tangent keys** **4.Round keys** **5. Splines**	*Types of Keys 02 Marks and their Applications 02 Marks*

Applications:

1. **Sunk Key:** It is used, where key is to be removed frequently
2. **Saddle Key:** They are suitable for light duty or low power transmission, as the power is transmitted due to friction. It is used as temporary fastening in fixing and setting eccentric parts, cams etc.
3. **Tangent Keys:** It is used to transmit the torque only in one direction.
4. **Round Keys:** These are used for low torque transmission.
5. **Splines:** These splines are used for power transmission of very high order and also provide axial movement between shaft and mounted member. Practical applications of splines may be seen in gear shifting mechanism used in automobile gear boxes.

	(B)	**Attempt any ONE:**	**06**
	(a)	**Explain design procedure for leaf spring.**	**06**

Ans. ***(Neat labelled Sketch 01 Mark and 01 Mark for each design step)***



Let,
2W = Central Load
2L = Span of Spring
b = Width of Leaves
t = Thickness of Leaves
n = Total numbers of Leaves
l = Length of Central Band
n_f = Nos. of full length leaves
n_g = Nos. of Graduated Leaves

Neat labelled Sketch 01 Mark and 01 Mark for each design step

(1) Stress in the Leaf Spring:

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

Where,
Effective Length of Spring = 2L = $2L_1 - l$……………….(**When Central band is used**)
= 2L = $2L_1 - \frac{2}{3}\,l$ …… … ….(**When U - Bolt is used**)

(2) Deflection of Leaf Spring:

$$\delta = \frac{6WL^3}{n\,E\,b\,t^2}$$

(3) Stress in full Length Leaves:

$$\sigma_f = \frac{18WL}{b\,t^2(2n_g + 3n_f)}$$

(4) Stress in Graduated Leaves:

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

(5) Deflection in Full Length and Graduated Leaves:

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

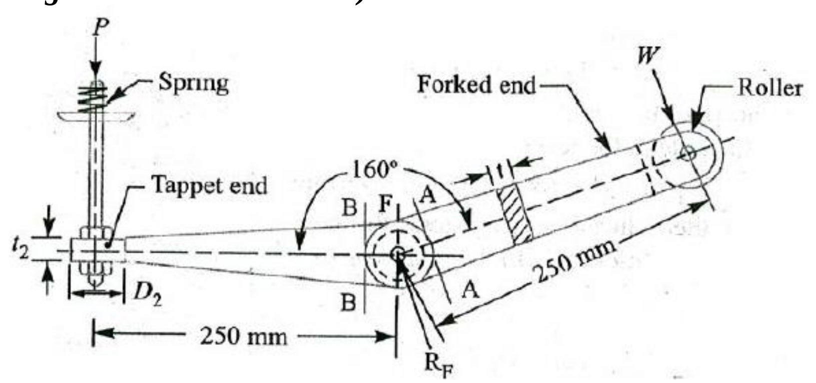
Design steps for calculating length of Leaf Spring:

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 = 2L1 + (π (d+t) x 2
Where d = diameter of Eye.

$$d = (32M/\pi\,\sigma_b)^{1/3}$$

	(b)	**Describe the procedure to design of fulcrum pin of Rocker Arm.**	**06**

Ans. *(Correct Design Procedure 06 Marks)*



The ratio of length to the diameter of the fulcrum pin and roller pin is taken as 1.25. The permissible bearing pressure on this pin is taken from 3.5 to 6 N/mm2.

The outside diameter of boss at fulcrum is usually taken twice the diameter of the pin at fulcrum. The boss is provided with a 3mm thick phosphor bronze bush to take up the wear.

Correct Design Procedure 06 Marks

Design Procedure:

Step I: Calculate Reaction at the Fulcrum Pin

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

Step II: Design of Fulcrum Pin:

(a) Let

d = Diameter of the fulcrum pin, and
l = Length of the fulcrum pin= 1.25 d
Considering the bearing of the fulcrum pin.
We know that load on the fulcrum pin (R_F),

$$Bearing\ Pressure\ = \frac{Load}{Bearing\ Area} = \frac{R_F}{l\ X\ d} = \frac{R_F}{1.25d\ X\ d}$$

From above equation l and d can be determined

Step III: Checking Shear stress Induced in the fulcrum Pin, as the pin is in double shear,

$$\tau = \frac{R_F}{2\ x\ (\frac{\pi}{4}\ .d^2)}$$

External diameter of the boss,

D = 2 d

Internal diameter of the hole in the lever,

dh= d + (2 × 3)

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

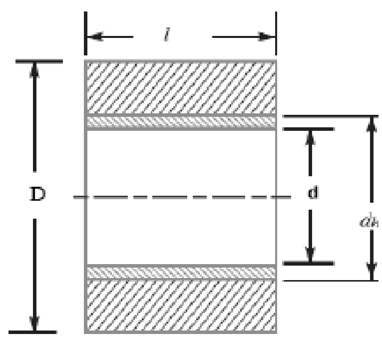


Figure: Boss of Fulcrum Pin

Bending moment at this section = W × L

Section Modulus $Z = \frac{\frac{1}{12} X\ l\ X\ (D_3 - d_{h3})}{\frac{D}{2}}$

Step IV: Induced bending stress,

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

2		**Attempt any FOUR:**	**16**
	(a)	**Describe design procedure for fully floating rear axle.**	**04**
	Ans.	***(Correct Design Procedure 04 Marks)***	

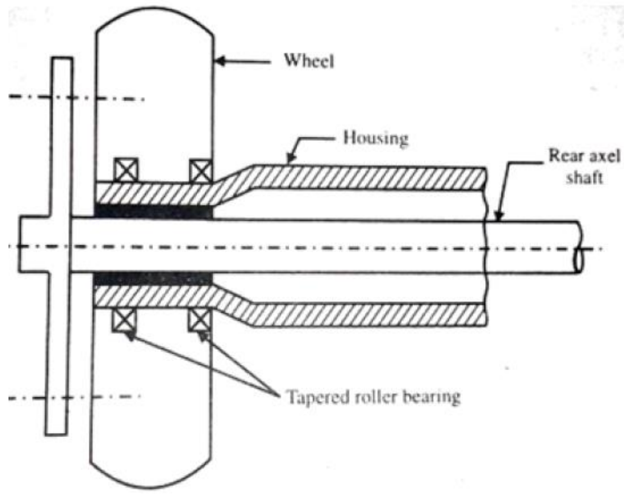


Figure: Fully Floating Rear Axle

Design procedure of a Fully Floating Rear Axle:

The rear axle is designed on the basis of shaft design.

By using the torsional equation,

$$\frac{T_{RA}}{J_{RA}} = \frac{\tau}{r}$$

TRA = Torque transmitted by rear axle shaft.

J_{RA} = Polar Moment of Inertia

$J_{RA} = \frac{\pi}{32} d^4$

$J_{RA} = \frac{\pi}{32}(d_o^4 - d_i^4)$

τ = Torsional shear stress r = distance from neutral axis to outer most fiber

r = $\frac{d}{2}$for Solid shaft)

r = $\frac{d_o}{2}$ (for Hollow shaft)

$$T_{RA} = T_e \text{ x } G_1 \text{ x } G_d$$

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

T_e = Engine Torque.

G_1 = Maximum gear Ratio in Gear Box

G_d = Final gear reduction in differential

After simplifying the equations,

$T_{RA} = \pi/16 \text{ x } \tau \text{ x } d^3$ (for Solid shaft)

$= \pi/16 \text{ x } \tau \text{ x } d_o^3 (1-k^4)$ (for Hollow shaft)

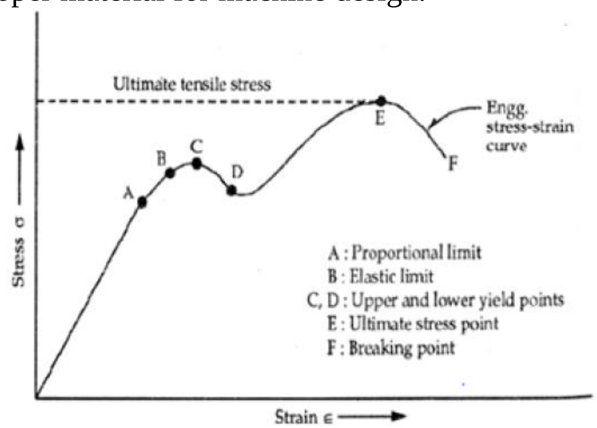
$k = d_i/d_o$

d_i = Inner diameter of shaft

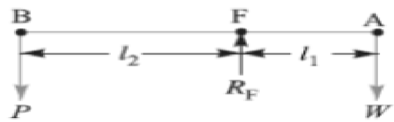
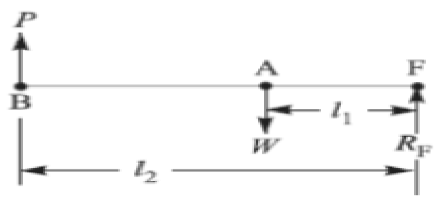
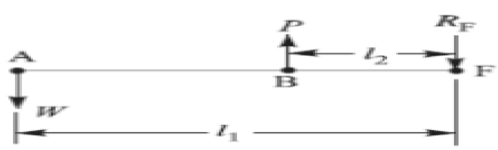
d_o = Outer diameter of shaft

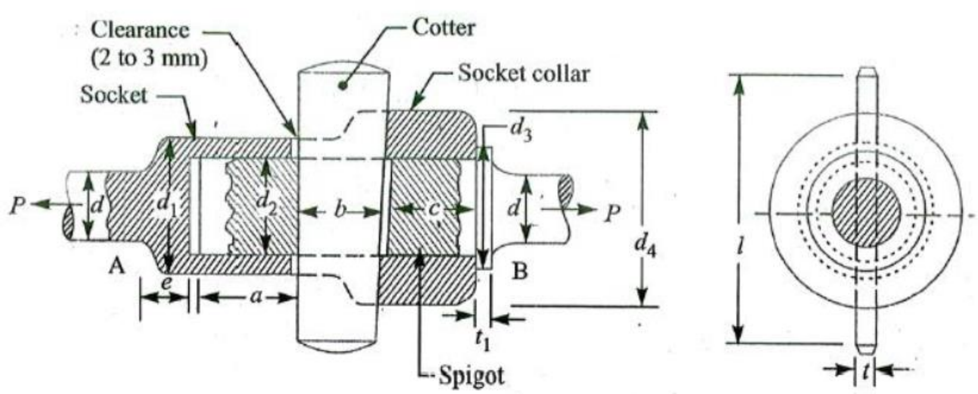
From this equation we can find diameter of fully floating rear axle

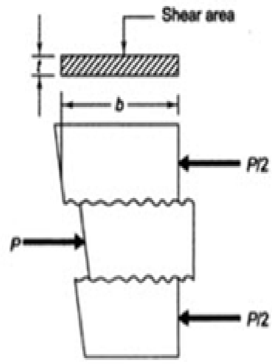
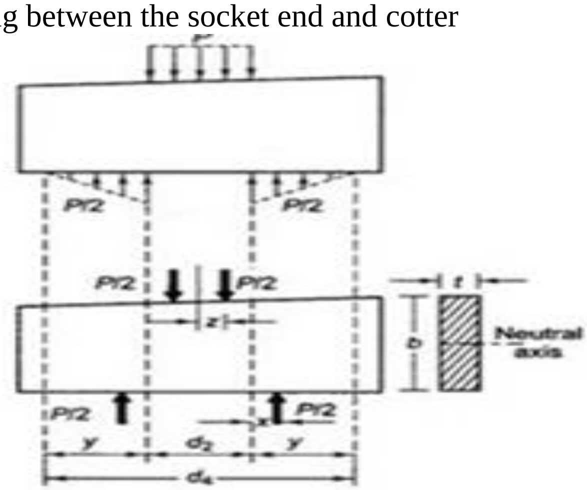
Correct Design Procedure 04 Marks

	(b)	**Draw stress, strain diagram for ductile material and state its importance.**	**04**
	Ans.	***(Importance 02 Marks, Neat Labelled Diagram 02 Marks)*** **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.  **Figure: Stress - Strain Diagram for Ductile Material**	*Importance 02 Marks, Neat Labelled Diagram 02 Marks*
	(c)	**Design the knuckle joint is required to withstand a tensile load of 20 kN if permissible stress are σ_t= 56 N/mm^2, τ = 40 N/mm^2, σ_c = 70 N/mm^2.**	**04**
	Ans.	**(Correct Ans.04 Marks)** **Given Data:** P = 20 × 103 N τ = 40 N/mm^2 σ_t= 56 N/mm^2 σ_c= 70 N/mm^2 **i. Find Diameter of rod:-** $$P= \frac{\pi}{4} d^2 . \sigma_t$$ $$20\ X10^3 = \frac{\pi}{4} d^2 . 56$$ d= 21.33mm say 22 mm **ii. Find dimensions of fork end, eye end and knuckle pin by empirical relations:-** 1. Diameter of knuckle pin d1=d = 22 mm 2. Outer diameter of eye end d2=2d = 44 mm 3. Diameter of knuckle pin head or collar d3=1.5d = 33 mm 4. Thickness of eye end t=1.25d = 27.5 mm 5. Thickness of forked end t1=0.75d = 16.5 mm 6. Thickness of collar or head t2=0.5d = 11 mm iii. **Induced stress in knuckle pin:-** $P = 2\ X\ \frac{\pi}{4} d_1^2 . \tau$ $20\ X10^3 = 2\ X \frac{\pi}{4} 22^2 . \tau$	**01** **01**

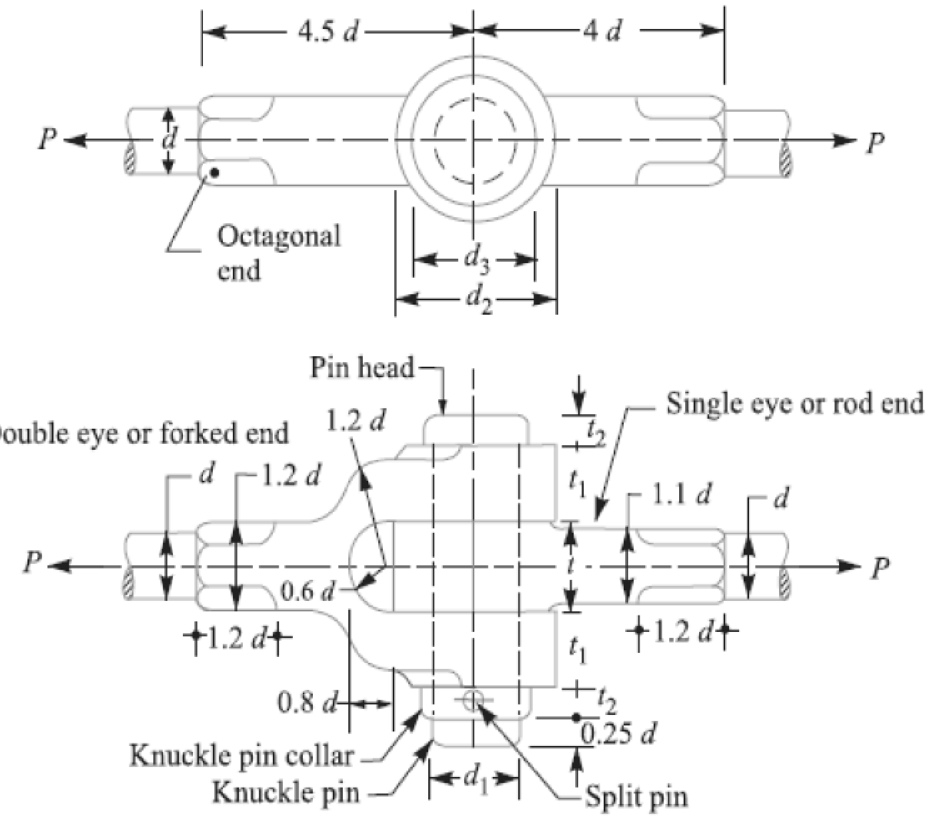
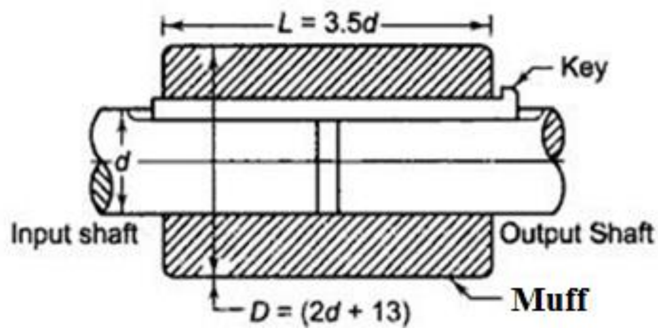
		$$\tau = 26.30 \frac{N}{mm^2} < 40\ N/mm^2$$ Therefore Design is safe. **iv. Induced stresses in eye end:-** **1. Failure in tension**: ∴ P = (d2 − d1)t × σt ∴ $20 \times 10^3 = (44 - 22)$ X 27.5 X σt $$\sigma t = 33.05 \frac{N}{mm^2} < 56\ N/mm^2$$ Therefore Design is safe. **2. Failure in shear:** ∴ P = (d2 − d1)t × τ ∴ $20 \times 10^3 = (44 - 22)$ X 27.5 X τ $$\tau = 33.05 \frac{N}{mm^2} < 40\ N/mm^2$$ Therefore Design is safe. **3. Failure in crushing**: ∴ P = d1 t × σc ∴ 20 × 103 = 22 × 27.5 × σc $$\sigma_c = 33.05 \frac{N}{mm^2} < 70\ N/mm^2$$ Therefore Design is safe. **Induced stresses in forked end:-** 1. Failure in tension: ∴ P = 2 x (d2 − d1)$t1$ × σt ∴ 20×10^3= 2 x (44 − 22) X 27.5 × σt $\sigma_t = 16.52\ N/mm^2 < 56\ N/mm^2$ Therefore Design is safe 2. Failure in shear: ∴ P = 2(d2 − d1)$t1$ × τ ∴ 20×10^3= 2(44 − 22) x 16.5 × τ $\tau = 27.54\ N/mm^2 < 40\ N/mm^2$ Therefore Design is safe 3. Failure in crushing: ∴ P = 2(d2 − d1)$t1$ × σc ∴ 20 × 103 = 2 × (44-22) x 16.5 × σc $\sigma_c = 27.54\ N/mm^2 < 70\ N/mm^2$ Therefore Design is safe	 **01** **01**
	(d)	**Define Lever. Describe three basic types of lever.**	**04**
	Ans.	***(Definition 01 Mark and Each Type of Lever 01 Mark)*** **Definition:-** A lever is a rigid rod or a bar capable of turning about a fixed point called fulcrum. **Types of lever**: **a) First Type Lever:**	*Definition 01 Mark and Each Type of Lever 01*

		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.  **Figure: First Type Lever** **b) 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.  **Figure: Second Type Lever** **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.  **Figure: Third Type Lever**	*Mark*
	(e)	**Design Propeller shaft of transmit 5kW at 5000 r.p.m. with a gear box reduction of 16;1. Assume permissible shear stress for shaft material as 45 N/mm^2**	**04**
	Ans.	***(Correct Answer 04 Marks)*** **Given Data:** **P= 5 ×10^{3}W,** **N=5000rpm** **G1=16:1,** **fs =45 N/mm^2** Now torque produced by the engine, $P = \frac{2\,\pi N T_e}{60}$ $5\ X\ 10^3 = \frac{2\pi\ x\ 5000\ x\ T_e}{60}$ *Te = 9.549 N-m* *Te = 9.549 X 10^{3}N-mm* Torque transmitted by the propeller shaft, Tp=Te× G1 Tp=9.549×103× 16	*Correct Answer 04 Marks*

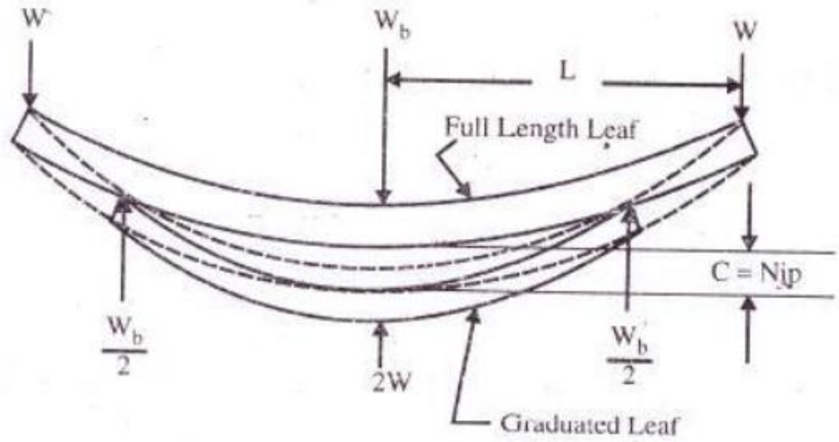
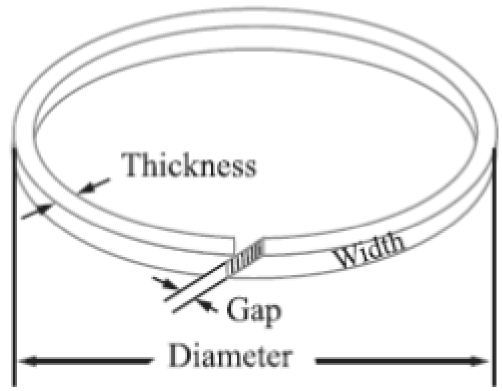
		Tp=152.78×103 N-mm Diameter of propeller shaft, $T_p = \frac{\pi}{16}\, \sigma_s\, d^3$ $152.78 \times 10^3 = \frac{\pi}{16}\, 45\, d^3$ **d=25.86mm** **d= 26 mm**	
3		**Attempt any FOUR:**	**16**
	(a)	**With neat sketch of socket and spigot cotter joint, write procedure to design of cotter only.**	**04**
	Ans.	***(Correct Answer 04 Marks)***  **Figure: Socket and Spigot Cotter Joint** The cotter is subjected to double shear. Total area of cotter that resists the shear failure = 2bt **Shear Stress in the Cotter Pin =** $\tau = \frac{P}{2bt} \leq [\tau]$	*Correct Answer 04 Marks*

		 **Bending Failure of Cotter :** When the cotter is tight in the socket and spigot, failure occurs due to shear, but when it is loose, it is subjected to bending moment as shown in figure It is assumed that: the force between cotter and spigot end is uniformly distributed and uniformly varying between the socket end and cotter  Maximum Bending Stress in the cotter = $$\sigma_b = \frac{My}{I} \leq [\sigma]$$	
	(b)	**What are the advantages of standardization?**	**04**
	Ans.	***(Any four 01 Mark Each)*** **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.	*Any four 01 Mark Each*

	(c)	**Calculate cylinder bore diameter and stroke length for four stroke six cylinder engine developing 90kW at 3000 rpm. The brake mean effective pressure is 11 X 10^5 N/$m^2$and L/D=1.2.**	**04**
	Ans.	***(Correct Answer 04 Marks)*** **Given:** Nos. of Cylinder, n = 6 Power Developed, P_{mb} = 90kW Engine Speed, N= 3000 r.p.m. Brake Mean Effective Pressure, Pm=11 N/m^2 Length of Stroke, L= 1.2D For Four Stroke Cycle, k =1/2 **Bore and length of cylinder:** Let D= Bore of the cylinder in mm A= across section area of the cylinder = $\frac{\pi}{4}D^2$ $$Brake\ Power\ Developed, BP = \frac{n\ x\ P_{mb}\ \ x\ L\ x\ A\ x\ Nk\ x\ 10}{6}$$ $$90 = \frac{6\ x\ 11\ x\ 1.2\ D\ x\ \pi D^2\ x\ 3000\ x\ \ 10}{6\ x\ 4\ x2}$$ $$D^3 = \frac{4320}{7464204}$$ *D^3= 0.0005787* *D = 0.083356m* *D = 83.35mm* $$\therefore Stroke\ Length, L = 1.2\ x\ D$$ $$\therefore Stroke\ Length, L = 1.2\ x\ 83.35$$ $$\therefore Stroke\ Length, L = 100.02\ mm$$	*Correct Answer 04 Marks*
	(d)	**Draw a labelled sketch of a knuckle joint.**	**04**
	Ans.	***(Neat Labelled Sketch 04 Marks)***	*Neat Labelled Sketch 04 Marks*

		 **Figure: Knuckle Joint**	
	(e)	**Explain the design procedure for muff coupling.**	**04**
	Ans.	***(Correct Design Procedure 04 Marks)***  **Figure: Muff Coupling** **Design of Shaft** Shafts are designed on the basis of torsional shear stress induced because of the torque to be transmitted. Shear stress induced in shaft for transmitting torque, T is given by, $$\tau = \frac{T\ r}{J} \leq [\tau]$$ Where, T= Twisting moment (or torque) acting upon the shaft,	*Correct Design Procedure 04 Marks*

		J = Polar moment of inertia of the shaft about the axis of rotation r = Distance from neutral axis to the outer most fibre = d/2 So dimensions of the shaft can be determined from above relation for a known value of allowable shear stress, [τ]. **Muff Design** Following relations are used to calculate the dimensions. D = 2d + 13 L = 3.5d Then the torsional shear stress in the sleeve is checked considering it as a hollow shaft. Shear stress, $\tau = \frac{T\,r}{J} \le [\tau]$ where, T = Twisting moment (or torque) to be transmitted J = Polar moment of inertia about the axis of rotation r = Distance from neutral axis to the outer most fibre = D/2 **Design of Key** Cross-section of the key is taken from the table corresponding to the shaft diameter or relations (square key) or and (for rectangular key) are used to find the cross-section, where w is width and h is the height of the key. Length of key in each shaft, The keys are then checked in shear and crushing Shear stress, $\tau = \frac{P}{wl} \le [\tau]$ and Crushing stress, $\sigma_{crushing} = \frac{P}{l\,h/2} \le [\sigma_c]$	
4	**(A)**	**Attempt Any Three:**	**12**
	(a)	**Explain concept of nipping.**	**04**
		Answer: 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**

		 Fig. Nipping in springs	02
	(b)	**State design procedure for piston ring and skirt length**	**04**
		Answer: Design of Piston Rings: (02 marks)  **Fig.** Piston rings. The radial thickness ($t1$) 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:	01

		$t_2 = \frac{D}{10\, n_R}$ 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 $4\ t_1$.	01
		2. 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% $\therefore \quad R = 0.1\,F$	01
		Let, l_1 = length of piston skirt 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	01

	(c)	**Design the turn buckle. Find the diameter of rod and coupler nut to withstand a load of 1600N. Given permissible stresses are 70N/mm^2 and 60 N/mm^2 in tension and shear respectively.**	**04**
		Solution: Given: P= 1600 N, σ_t = 70N/mm^2, τ= 60 N/mm^2 We know that the design load for the threaded section, **P_d = 1.3 P = 1.3 X 1600 = 2080 N ……………………………………..(i)** 1. Diameter of tie rod Let d = diameter of tie rod, and d_c = core diameter of thread on the tie rod considering tearing of the threads on the tie rod at their roots, We know that design load(P_d), $2080 = \frac{\pi}{4} (d_c)^2 \times \sigma_t = \frac{\pi}{4} (d_c)^2 \times 70 = 54.97 (d_c)^2$ $\therefore (d_c)^2 = 37.84$ $\therefore d_c = 6.15$ mm ……………………………………………………(ii) As $d_c = 0.84$ d $\therefore d = \frac{6.15}{0.84} = 7.32 = 8$ mm…………………………………………..(iii) 2. Outside diameter of Coupler Nut Let D = outer diameter of the coupler nut Considering tearing of the coupler nut. We know that axial load (P); $1600 = \frac{\pi}{4} (D^2 – d^2) \times \sigma_t$ $\therefore 1600 = \frac{\pi}{4} (D^2 – (8)^2) \times 70$ $\therefore (D^2 – (8)^2) = \frac{1600 \times 4}{\pi \times 70}$ $\therefore (D^2 – 64) = 29.10$ $\therefore D^2 = 29.10 + 64$ $\therefore D^2 = 93.10$ $\therefore$ D = 9.65 mm say 10mm ………………………………………(iv) Since the minimum outside diameter of coupler nut is taken as 1.25d (1.25 X 8 =10 mm) therefore the value of D is satisfactory.	 **01** **01** **01** **01**
	(d)	**A multiplate disc clutch has 6 active friction surface, power transmitted is 20kW at 400 rpm. Inner and outer radius of the friction surfaces are 90mm and 120mm respectively. Assuming uniform wear with a co-efficient of friction**	**04**

		0.25. Find the maximum axial intensity of pressure between the discs.	
		Answer: Given Data: n = 6, N = 400 rpm, r_1 = 120 mm, r_2 =90mm, μ = 0.25 Maximum axial intensity of pressure = P_{max} =? P = 20kW = 20 X10^3 W. **For uniform wear,** Mean radius R = $\frac{(r1+r2)}{2} = \frac{(90+120)}{2}$ = 105 mm …………………………..(i) We know that power transmitted is given by $P = \frac{2\pi NT}{60}$ $\therefore 20 X10^3 = \frac{2\pi\ X\ 400\ X\ T}{60}$ ∴ T = 477.46 N-m = 477.46 X 10^3 N-mm ………………………..(ii) **Torque transmitted is given as** T = n μ W R 477.46 X 10^3 = 6 X 0.25 X W X 105 ∴ W = 3031.49 N ……………………………………………………….(iii) **We know that** **Axial force required to engage the clutch,** W = 2πC (r1 – r2) ∴ 3031.49 = 2 X π X C X (120-90) ∴ 3031.49 = 2 X π X P_{max} X r_2 (30) ∴ 3031.49 = 2 X π X P_{max} X 90 X (30) ∴ P_{max} = 0.1786 N/mm^2 ……………………………………………………..(iv)	01 01 01 01
4	(B)	**Attempt any ONE.**	06
	(a)	**State stepwise procedure for component design.**	06
		Answer: (design procedure 06 marks) The general procedure to solve a design problem is as follows : 1. **Recognition of need.** First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed. 2. **Synthesis (Mechanisms).** Select the possible mechanism or group of mechanisms which will give the desired motion. 3. **Analysis of forces.** Find the forces acting on each member of the machine and the energy transmitted by each member. 4. **Material selection.** Select the material best suited for each member of the machine. 5. **Design of elements (Size and Stresses).** Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member	06

		should not deflect or deform than the permissible limit. 6. **Modification.** Modify the size of the member to agree with the past experience and judgment to facilitate manufacture. The modification may also be necessary by consideration of manufacturing to reduce overall cost. 7. **Detailed drawing.** Draw the detailed drawing of each component and the assembly of the machine with complete specification for the manufacturing processes suggested. 8. **Production.** The component, as per the drawing, is manufactured in the workshop.	
	(b)	**A single plate (clutch) with both side effective has outer and inner diameter 300mm and 200mm respectively. The maximum intensity of pressure at any point of contact is not to exceed 02 N/mm^2. If the co-efficient of friction is 0.3. Determine the power transmitted by clutch at shaft speed 3000 rpm.**	06
		Given Data: d1 = 300mm, r1 = d1/2 = 150mm d2 = 200mm, r2 = d2/2 = 100mm , P_{max} = 0.2 N/mm^2 , μ = 0.3 , N = 3000 rpm **Since the intensity of pressure is maximum at inner radius, therefore, for uniform wear,** Pmax × r2 = c c = 0.2 × 100 **c = 20 N/mm …………………………………………………………………..(i)** **We know that, axial thrust,** W = 2πc (r1 – r2) W = 2π × 20 × (150-100) **W = 6284 N ……………………………………………………………………(ii)** **And mean radius of friction,** R = (r1 + r2)/2 R = (150+100)/2 **R = 125 mm ……………………………………………………………… ….(iii)** **We know that, torque transmitted,** T = n. μ. W. R T = 2 × 0.3 × 6284 × 125 T = 471300 N-mm **T = 471.3 N-m ………………………………………………………...(iv)** **Power transmitted by clutch,** P = (2π N T)/60 P = (2× π × 3000 × 471.3)/60 P = 148063.26 W **P = 148.06 kW………………………………………………………………..(v)**	01 01 01 01 02
5		**Attempt any TWO:**	16
	(a)	**Describe the theories of failure of maximum principal stress theory and maximum shear stress theory.**	08
		Answer: i) **Maximum Principal or Normal Stress Theory (Rankine's Theory)** According to this theory, the failure or yielding occurs at a point in a member when	

		the maximum principal or normal stress in a bi-axial stress system reaches the limiting strength of the material in a simple tension test. $\sigma_{t1} = \frac{\sigma_{yt}}{F.S.}$ ……………………for ductile material $\sigma t1 = \frac{\sigma_{ut}}{F.S.}$ ……………………………for brittle material Where σ_{yt} = Yield point stress in tension as determined from simple tension test, and σ_{ut} = Ultimate stress. F.S. = factor of safety Since the maximum principal or normal stress theory is based on failure in tension or compression and ignores the possibility of failure due to shearing stress, therefore it is not used for ductile materials. However, for brittle materials which are relatively strong in shear but weak in tension or compression, this theory is generally used. **ii) Maximum Shear Stress Theory (Guest's or Tresca's Theory)** According to this theory, the failure or yielding occurs at a point in a member when the maximum shear stress in a bi-axial stress system reaches a value equal to the shear stress at yield point in a simple tension test. Mathematically, $\tau_{max} = \frac{\tau_{yt}}{F.S.}$. ……………………………….(***i***) where τ_{max} = Maximum shear stress in a bi-axial stress system, τ_{yt} = Shear stress at yield point as determined from simple tension test, *F.S.* = Factor of safety. Since the shear stress at yield point in a simple tension test is equal to one-half the yield stress in tension, therefore the equation **(*i*)** may be written as $\tau_{max} = \frac{\sigma_{yt}}{2 X F.S.}$ This theory is mostly used for designing members of ductile materials.	**04** **04**
	(b)	**An automobile gearbox gives three forward and a reverse speed with top gear of unity and bottom and reverse gear ratio 3.3:1, the center distance between the shaft is 110mm approximately. Gear teeth of module 3.25mm are to be employed. Determine different gear ratios of various gears and number of teeth.**	**08**
		Ans- Given data- Module=3.25 mm ,	

		Centre distance between shafts =110 mm , Since the pitch is same for all wheels and the centre distance same for all the pairs meeting the wheels, the total no. of teeth must be same for each pair Therefore TA + TB= TC+ TD= TE+ TF = T T = (2 × C.D.) / M T = (2 ×110) / 3.25=67.8 **T = 68 teeth ……………………………………………………..(i)**	**01**
		In general practice for better results gear ratios are kept in geometric progression if G1,G2,G3 are gear ratios in first, second and third gear ,then $\frac{G_1}{G_2} = \frac{G_2}{G_3}$ $\therefore$ **$G_2 = \sqrt{G_1 \, X \, G_3}$** $\therefore$ **$G_2 = \sqrt{1 \, X \, 3.3}$** $\therefore$ **G_2 = 1.817…………………………………………………….(ii)**	**01**
		In general practice, while designing a gear box it is desired that the gear ratio should be minimum possible in all cases, so that the sizes of the gear box can be kept minimum. To achieve this, the maximum reduction required in gear box (in first gear) is achieved in two equal steps. Now $G_1 = \frac{T_B}{T_A} \times \frac{T_D}{T_C}$ Let X be the reduction in one step therefore $G_1 = X \times X$ $3.3 = X^2$ $X = 1.817$ $\frac{T_B}{T_A} = \frac{T_D}{T_C} = 1.817$	

		$T_B = 1.817\ T_A$ and $T_D = 1.817\ T_C$ $T_A + T_B = 68$, $T_A + 1.817\ T_A = 68$, $T_A = 68/2.817$ $T_A = 24$ teeth ……(iii) $T_B = 68-24$ $T_B = 44$ teeth Similarly $T_C = 24$ teeth and $T_D = 44$ teeth Actual gear ratio $G_1 = \frac{T_B}{T_A} \times \frac{T_D}{T_C}$ $G_1 = \frac{44}{24} \times \frac{44}{24}$ $G_1 = 3.36:1$ …………………..(iv) Second gear ratio $G_2 = \frac{T_B}{T_A} \times \frac{T_F}{T_E}$ $1.817 = \frac{44}{24} \times \frac{T_F}{T_E}$ $T_F = 0.991\ T_E$ $T_E + T_F = 68$ $T_E + 0.991\ T_E = 68$ $T_E = 34$ teeth $T_F = 68 - 34 = 34$ teeth ……………………………………….(v) Actual gear ratio in second gear = $G_2 = \frac{44}{24} \times \frac{34}{34} = 1.833$ ….(vi)	01 01 01

		Top gear ratio G3 =1:1 is obtained by directly coupling the main shaft to engine shaft. Now reverse gear and idler is fitted due to presence of which the direction is reversed also it is required that $T_D + T_i < 68$ to avoid interference of gear C and D Therefore $T_i < 68 - T_D$ $T_i < 68 - 44$ $T_i < 22$ ……………………………………(vi) Exact gear ratio $G_R = \frac{T_B}{T_A} \times \frac{T_D}{T_I}$ $G_R = \frac{44}{24} \times \frac{44}{20}$ $G_R = 4.03:1$ ………………………………………(vii) Exact center distance = $C.D. = \frac{M(TA+TB)}{2}$ $= \frac{3.25(68)}{2}$ $= 110.5$ mm ……………………………………..(viii)	01 01 01
	(c)	Explain the design procedure for cylinder head thickness and bolts.	08
		Answer: **Design of cylinder head:** The cylinder head is designed by considering it a flat circular plate. The thickness is determined by following relation. $t = D\sqrt{\frac{C-P_{max}}{\sigma_c}}$ t = thickness of cylinder head D = diameter of cylinder	01

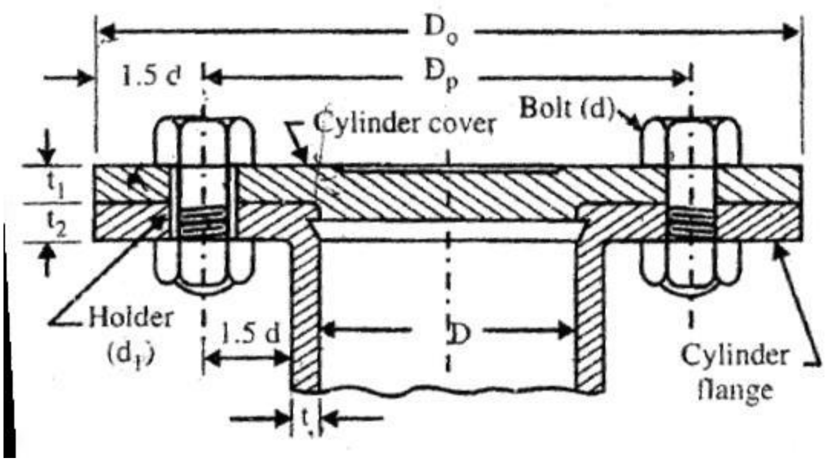
C = constant
= 0.1………for C.I.

P_{max} = maximum gas pressure inside the cylinder

σ_c = Allowable circumferential stress in MPa or N/mm^2. It may be taken as 30 to 50 MPa

The studs or bolts are screwed up tightly along with a metal gasket or asbestos packing to provide a leak proof joint between the cylinder and cylinder head. The tightness of the joint also depends upon the pitch of the bolts or studs which should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$ the pitch circle diameter (D_p) is usually taken as D+3d. **01**

Design of cylinder head bolts or studs :



01

a) The centre of stud is assumed at a distance of 1.25 to 1.5 d from inner wall of the cylinder where 'd' is diameter of bolt (let us assume 1.5d)

$D_p = D + 2 \times 1.5d$

$= D + 3d$ ……………………….(i)

b) The gas pressure is assumed to be acting on P.C.D. of studs.

$\therefore$ Gas load $= P_{max} \times \left(\frac{\Pi}{4} D_p^2\right)$ **01**

$P_{max} \times \frac{\Pi}{4}(D+3d)^2$ ……………..(ii)

		c) This load is acting as tensile load on bolts or stud and this load is resisted by 'Z' numbers of bolts. $P_{max} \times \frac{\Pi}{4}(D+3d)^2 = Z \times \frac{\Pi}{4} d_c^2 \times f$ ....(iii)	01
		d) Numbers of bolts 'Z' is taken between $Z = \left(\frac{D}{100}+4\right) \text{to} \left(\frac{D}{50}+4\right)$ ....(iv)	01
		Generally even value is selected for 'Z'	01
		e) Value of 'd' is taken as $d = \frac{d_c}{0.84}$ ....(v) f) Putting value from (iv) in equitation (iii) values of d, d_c and Z are calculated g) For a leak proof joint, value of 'd' greater than 16 should be used. h) The circular pitch of stud is calculated as $\text{Pitch 'p'} = \frac{\Pi D_P}{Z}$	
		For a leak proof joint minimum value of 'P' should be 3 d and maximum value of 'P' line between $19\sqrt{d}$ to $28\sqrt{d}$. If value of P is coming less decrease value of 'Z' and recalculate. If value of P is coming more increase value of 'Z' till condition is satisfied.	01
6		**Attempt any TWO:**	**16**
	(a)	**Design the piston pin with following data (i) P_{max} = 4.5 N/mm², (ii) Diameter of piston =70mm. Allowable stresses due to bearing , bending and shear are given 30N/mm², 80 N/mm² and 60 N/mm² respectively.**	
		Answer: Given data, Dia. of piston = D = 70 mm. Max. pressure = Pmax = 4.5 N/mm² Bearing pressure Pb = 30 N/mm² Bending stress = σ_b = 80N/mm² Shearing stress = τ = 60 N/mm² **Maximum Gas Load,** $F = \frac{\pi D^2}{4} \times P_{max}$	

		$\therefore F = \frac{\pi\,(70)^2}{4} X\ 4.5$ **∴ F = 17318.02 N……………………………………………………..(i)**	01
		(a) Design the piston pin on the basis of bearing pressure Let, dpo = outer dia. of piston pin lp = length of piston pin in small end of connecting rod lp = 0.45xD = 0.45x70 **lp = 31.5 mm……………………………………………………………..(ii)**	01
		Now $F = d_{po}\ X\ l_p\ X\ P_b$ $17318.02 = d_{po}\ X\ 31.5\ X\ 30$ $\therefore d_{po} = \frac{17318.02}{31.5\ X\ 30}$ **∴ d_{po} = 18.32 mm say 20 mm ……………………………………………(iii)**	01
		(b)Designing the piston pin on the basis of bending. 'Bending moment 'M' is calculated as $M = \frac{F\ x\ D}{8}$ $\therefore M = \frac{17318.02\ \ X\ 70}{8}$ **∴ M = 151.53 X 10^3 N-mm……………………………………….(iv)**	01
		We know that $M = \frac{\pi}{32}\ X\ \sigma_b\ X\ (d_{po})^3$ $\therefore \sigma_b = \frac{M\ X\ 32}{\pi\ X\ (d_{po})^3}$ **∴ σ_b = 192.93 N……………………………………………………(v)**	01
		The induced bending stresses are greater than permissible bending stress 80N/mm2 Hence redesign is necessary. Now redesign value of dpo $M = \frac{\pi}{32}$ X σb X (dpo)3 ∴ 151.53 X 103 = $\frac{\pi}{32}$ X 80 X (dpo)3 ∴ (dpo)3 = 19293.39 ∴ dpo = 26.82 **∴ dpo = 28 mm ………………………………………………….(vi)**	01
		c) Designing piston pin on the basis of shear stress, due to double shear	

		$F = 2 X \frac{\pi}{4} X (d_{po})^2 X \tau$ $\therefore 17318.02 = 2 X \frac{\pi}{4} X (28)^2 X \tau$ $\therefore \tau = \frac{17318.02 X 4}{2 X \pi X 28^2}$ **$\therefore \tau = 14.06$ N/mm^2……………………………………………..(vii)** The induced shear stress is less than the permissible shear stress. Therefore design is safe. **d) the total length of piston pin is taken as** **$L_{pt} = 0.9 D = 0.9 X 70 = 63$mm …………………………………..(viii)**	01 01
	(b)	**Design the connecting rod cross section with the following data $P_{max} = 5$ N/mm^2 , piston diameter = 70mm, stroke length =80 mm, effective length of connecting rod = 140mm, maximum allowable stress in the connecting rod in clipping is 110 N/mm^2. Take Rankine constant for steel $\frac{1}{6000}$.**	08
		Answer: Given Data Max. pressure inside $P_{max} = 5$ N/mm^2 Piston Dia D=70 mm Stroke length= l = 80mm Effective Length of connecting rod mm L= 140 mm Maximum allowable stress in the connecting rod in crippling is 110 N/mm^2 Rankine constant for steel is = $\frac{1}{6000}$ **Step I : Max gas load acting on the connecting rod** $W = P_{max} X \frac{\pi}{4} X D^2$ $\therefore W = 5 X \frac{\pi}{4} X (70)^2$ **$\therefore$ W = 19242.25 N……………………………………………..(i)** Area of cross section $A = 11 t^2$ Where t = thickness of flange a = Rankine constant = $\frac{1}{6000}$ $K_{xx} = \sqrt{3.18 t^2}$ **$\therefore K_{xx} = 1.78$ t………………………………………………(ii)**	01 01

		Step II: Critical Buckling load acting on connecting rod As factor of safety is not given, assuming factor of safety as 1, We know that Critical buckling load = W X FOS **∴ Critical buckling load = 19242.25 X 1= 19242.25 N……..(iii)**	01
		Assuming I section, Max. Crippling load is, $W_{cr} = \frac{\sigma_{crp}\ X\ A}{1+a[\frac{L}{K_{xx}}]^2}$ $\therefore 19242.25 = \frac{110\ X\ 11\ t^2}{1+\{\frac{1}{6000}\ X\ [\frac{140}{1.78\ t}]^2\}}$ $\therefore 19242.25 = \frac{110\ X\ 11\ t^2}{1+\frac{19600}{19010.4\ t^2}}$ $\therefore 19242.25 = \frac{110\ X\ 11\ t^2}{\frac{19010.4\ t^2+19600}{19010.4\ t^2}}$ $\therefore 19242.25 = \frac{110\ X\ 11\ t^2}{19010.4\ Xt^2+19600}$ X $19010.4\ t^2$ $\therefore 19242.25\ (19010.4\ t^2 + 19600) = 110 \times 11\ X\ 19010.4\ t^4$ $365802869.4\ t^2 + 377148100 = 23002584\ t^4$ $0= 23002584\ t^4 - 365802869.4\ t^2 - 377148100$ $0 = t^4 – 15.90\ t^2 -16.40$ Let $x = t^2$ $\therefore 0 = x^2 -15.90x – 16.40$ We know that $x = \frac{-b\pm\sqrt{b^2-4ac}}{2\ a}$ $\therefore x = \frac{-(-15.90)\pm\sqrt{(-15.90)^2-(4X\ 1\ X\ (-16.4))}}{2\ X\ 1}$ $\therefore x = \frac{15.90\pm\sqrt{318.41}}{2}$ $\therefore x = \frac{15.90\pm 17.84404}{2}$ $\therefore$ x =16.8720 $\therefore t^2 = 16.8720$……………………………………….($x= t^2$) **∴ t= 4.107556 mm …………………………………………..(iv)**	01

		Say t=5 mm **Dimension at the middle or center:** **(i) Depth or height of section:** H = 5 t =5 x 5 = 25 mm ……………………………………………(v) **(ii) width of cross section** B = 4 t = 4 x5 =20mm ……………………………………………..(vi) **Dimension at the big end (crank end):** **(i) Depth or height of section:** At the big end H2= 1.2H =1.2(25)=30*mm ……………………………(vii)* **(ii) width of cross section** B2=B=20 mm ……………………………………………………(viii)	01 01 01 01
	(c)	**Design flange coupling to transmit 15 kW at 900 rpm. The service factor may be used 1.3. following permissible stress may be assumed, shear stress for for shaft, bolt and key material is 40MPa, crushing stress for bolt and material is 80MPa and shear stress for cast iron is 8 MPa.**	08
		Given : P = 15 kW = 15 X 10^3 W, N = 900 rpm, Service factor = 1.3, $\tau_s = \tau_k = 40$ N/mm^2 , $\sigma_{cb} = \sigma_{ck} = 80$ N/mm^2 , $\tau_c = 8$ N/mm^2 The flange coupling is designed as follows: **1. Design for hub:** First of all let us find the diameter of the shaft (d). $$T = \frac{P \times 60}{2\Pi N} = \frac{15 \times 10^3 \times 60}{2\Pi \times 900} = 159.13 N - m$$ …(i) Since the service factor is 1.3 therefore the maximum torque transmitted by the shaft, T_{max} = 1.3 X 159.13 = 206.869 N-m **T_{max} = 206.869 X 10^3 N-mm ……………………………………………………(ii)** **Torque transmitted by shaft (T_{max}),** 206.869 X 10^3 = $\frac{\pi}{16}$ X τ_s X d^3	 01 01

		$\therefore 206.869 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$ $\therefore d^3 = \frac{206.869 \times 10^3 \times 16}{\pi \times 40} = 26328.78$ **$\therefore$ d = 29.74 say 30 mm …………………………………………………….(iii)** Outer diameter of the hub D= 2d = 2 X 30 = 60mm Length of hub L= 1.5d = 1.5 X 30 =45mm Let us now check the induced shear stress for the hub materials which is cast iron. Considering hub as a hollow shaft. We know that the maximum torque transmitted (T_{max}), $206.869 \times 10^3 = \frac{\pi}{16} \times \tau_c \times (\frac{D^4 - d^4}{D})$ $\therefore 206.869 \times 10^3 = \frac{\pi}{16} \times \tau_c \times (\frac{60^4 - 30^4}{60})$ $\therefore 206.869 \times 10^3 = 39760.78 \times \tau_c$ **$\therefore \tau_c$ = 5.20 MPa…………………………………………………………(iv)** Since the induced shear stress for the hub material (i.e. cast iron) is less than the permissible value of 8 MPa therefore the design of hub is safe. **2. Design for key:** Since the crushing stress for the key material is twice its shear stress therefore a square key may be used. Width of key w= 12 mm Thickness of key t=w=12mm The length of key(l) is taken equal to the length of hub l =1.5 d = 1.5 X 30 = 45mm Let us now check the induced stresses in the key by considering it in shearing and crushing considering the key in shearing we know that the maximum torque transmitted (T_{max}). $206.869 \times 10^3 = l \times w \times \tau_k \times \frac{d}{2}$ $\therefore 206.869 \times 10^3 = 45 \times 12 \times \tau_k \times \frac{30}{2}$ $\therefore \tau_k = \frac{206.869 \times 10^3 \times 2}{45 \times 12 \times 30}$ **$\therefore \tau_k$ = 25.53 N/mm² ………………………………………………..(v)** Considering the key in crushing. We know that the maximum torque transmitted (T_{max}). $206.869 \times 10^3 = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$	**01** **01** **01**

		$\therefore$ 206.869 X 10^3 = 45 X $\frac{12}{2}$ X σ_{ck} X $\frac{30}{2}$ $\therefore$ $\sigma_{ck} = \frac{206.869 X 10^3 X 2 X 2}{45 X 12 X 30}$ **$\therefore$ σ_{ck} = 51.07 N/mm^2 ……………………………………………..(vi)** Since the induced shear and crushing stresses in the key are less than the permissible stresses therefore the design for key is safe. **3. Design for flange:** The thickness of flange (tf) is taken as 0.5 d t_f = 0.5 d = 0.5 X 30 =15 mm Let us now check the induced shearing stress in the flange by considering the flange at the junction of the hub in shear. We know that the maximum torque transmitted (T_{max}) 206.869 X10^3 = $\frac{\pi D^2}{2}$ X τ_c X t_f $\therefore$ 206.869 X10^3 = $\frac{\pi X 60^2}{2}$ X τ_c X 15 $\therefore$ $\tau_c = \frac{206.869 X 10^3 X 2}{\pi X 60^2 X 15}$ **$\therefore$ τ_c = 2.43 N/mm^2 ……………………………………………………….(vii)** Since the induced shear stress in the flange is less than 8 MPa therefore the design is safe. **4. Design for bolts:** Let d1= nominal diameter of bolts. Since the diameter of the shaft is 30 mm therefore let us take number of bolts N=3 And pitch circle diameter of bolts D_1 = 3d = 3X 30 = 90mm The bolts are subjected to shear stress due to the torque transmitted. We know that maximum torque transmitted (T_{max}) *206.869 X 10^3= $\frac{\pi}{4}$ $(d_1)^2$X τ_b X n X $\frac{D_1}{2}$* *$\therefore$206.869 X 10^3= $\frac{\pi}{4}$ $(d_1)^2$X 40 X 3 X $\frac{90}{2}$* *$\therefore$ $(d1)^2 = \frac{206.869 X 10^3 X 4X 2}{\pi X 40 X 3X 90}$* *$\therefore$ $(d1)^2$ = 48.77* ***$\therefore$ d1 = 6.98 mm ……………………………………………………..(viii)*** Assuming coarse threads the nearest standard size of bolt is M 8 Other proportion of the flange are taken as follows: Outer diameter of the flange. *D_2 = 4d =4X 30 = 120 mm* Thickness of the protective circumferential flange. t_p = 0.25d =0.25 X 30 = 7.5 mm say 8mm	01 01 01