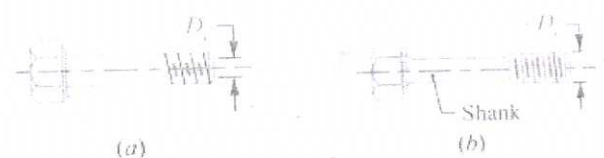
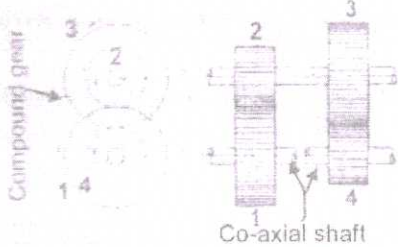


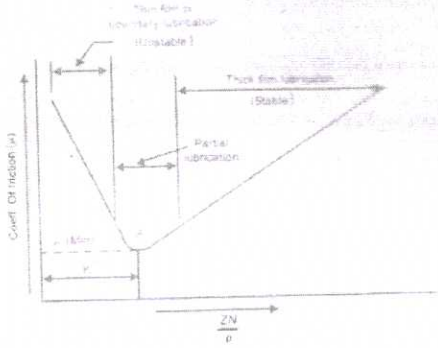
SCHEME OF EVALUATION

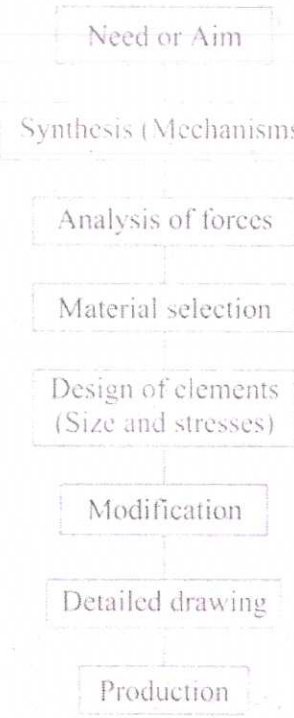
(Scoring Indicators)

Revision : 2015		Course Code : 5021		
Course Title : DESIGN OF MACHINE ELEMENTS				
Qn NO	Scoring indicator	Split up score	Sub Total	Total
PART A				
1	Load, Mechanism, Selection of materials, Manufacturing processes, Shape, Size, Quantity, Effect of environment, Durability, Reliability, Cost, Life, Safety	0.5x4	2	10
2	$\frac{T}{J} = \frac{\tau}{R} = \frac{C\theta}{l}$ T= torque, J = polar modulus, C= Modulus of rigidity, l= length of shaft, θ = Angle of twist in radian, τ = Shear stress, R = radius of shaft	Equ 1 + Ter 1	2	
3	Sensitiveness of a governor is defined as the ratio of the range of speed to the mean speed of governor. OR Sensitiveness of a governor is defined as the ratio of difference between maximum speed and minimum speed to the mean speed of governor.	1x2	2	
4	It is the ratio of number of teeth of a gear to the pitch circle diameter. $P_d = T/d$	1x2	2	
5	Sunk keys, Saddle keys, Tangent keys, Round keys, Splines	0.5x4	2	
PART B				
1	 <p>When a bolt is subjected to shock loading, as in case of a cylinder head bolt of an internal combustion engine, the resilience of the bolt should be considered in order to prevent breakage at the thread. In an ordinary bolt shown in Fig. (a), the effect of the impulsive loads applied axially is concentrated on the weakest part of the bolt i.e. the cross-sectional area at the root of the threads. In other words, the stress in the threaded part of the bolt will be higher than that in the shank. Hence a great portion of the energy will be absorbed at the region of the threaded part which may fracture the threaded portion because of its small length. If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (D_c) 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.</p>	Fig 3 + Marking 3	6	

Qn NO	Scoring indicator	Split up score	Sub Total	Total				
2	<p>Given : Power, $P = 560 \text{ kW}$ $= 560 \times 10^3 \text{ W}$</p> <p>Speed, $N = 300 \text{ rpm}$</p> <p>Max. shear stress, $\tau = 60 \text{ N/mm}^2$</p> $P = \frac{2\pi NT}{60} \quad \text{or} \quad T = \frac{60 \cdot P}{2\pi N}$ $T = \frac{60 \times 560 \times 10^3}{2 \times \pi \times 300} = 17834.39 \text{ Nm}$ $= 17834.39 \times 10^3 \text{ N-mm}$ <p>Torque, $T = \tau \cdot \frac{\pi D^3}{16} \quad \text{or} \quad D = \sqrt[3]{\frac{16T}{\pi \cdot \tau}}$</p> $D = \sqrt[3]{\frac{16 \times 17834.39 \times 10^3}{\pi \times 60}}$ $= 114.8 \text{ mm}$ <p>Diameter of shafts $D = 114.8 \text{ mm Ans.}$</p>	Equ 3 + Ans 3	6					
3	<table border="1"> <thead> <tr> <th>Flywheel</th> <th>Governor</th> </tr> </thead> <tbody> <tr> <td> <ol style="list-style-type: none"> 1. Flywheel stores and redistributes energy within a cycle to control speed. 2. A flywheel takes care of fluctuations of speed during a cycle. 3. A flywheel works continuously from cycle to cycle 4. A flywheel has no control over the quality and quantity of working agent. 5. A flywheel is not essential element of every prime mover. 6. It is used only in case when there is undesirable cyclic fluctuation of energy output or input. </td> <td> <ol style="list-style-type: none"> 1. Governor controls the amount of fuel to an engine to match the load requirements to maintain a specified speed. 2. A governor takes care of speed due to variation of load. 3. A governor works intermittently only when there is change in load. 4. A governor takes care of change of quality and quantity of the working agent. 5. A governor is an essential element of every prime mover. 6. It is an adjuster of supply of fuel with demand. </td> </tr> </tbody> </table>	Flywheel	Governor	<ol style="list-style-type: none"> 1. Flywheel stores and redistributes energy within a cycle to control speed. 2. A flywheel takes care of fluctuations of speed during a cycle. 3. A flywheel works continuously from cycle to cycle 4. A flywheel has no control over the quality and quantity of working agent. 5. A flywheel is not essential element of every prime mover. 6. It is used only in case when there is undesirable cyclic fluctuation of energy output or input. 	<ol style="list-style-type: none"> 1. Governor controls the amount of fuel to an engine to match the load requirements to maintain a specified speed. 2. A governor takes care of speed due to variation of load. 3. A governor works intermittently only when there is change in load. 4. A governor takes care of change of quality and quantity of the working agent. 5. A governor is an essential element of every prime mover. 6. It is an adjuster of supply of fuel with demand. 	1X3+1x 3	6	
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4		Fig 4 + marking 2	6					

Qn NO	Scoring indicator	Split up score	Sub Total	Total
5	<p>Reverted gear train is a special type of compound gear train in which the driver and driven gears are mounted on co-axial shafts and they are rotated on same direction. Main advantage of this gear train is reduced the space occupied.</p>  <p>Fig. 10.18 Reverted gear train</p> <p>Let N_1, N_2, N_3 and N_4 be the speeds of gear 1, 2, 3 and 4 respectively, and T_1, T_2, T_3 and T_4 be the teeth of gear 1, 2, 3 and 4 respectively.</p> <p>Consider a pair of gears 1 and 2, the train value is</p> $\frac{N_2}{N_1} = \frac{T_1}{T_2} \quad \dots (i)$ <p>Similarly, pair of gears 3 and 4, the train value is</p> $\frac{N_4}{N_3} = \frac{T_3}{T_4} \quad \dots (ii)$ <p>Multiplying equations (ii) by (i), the train value of reverted gear train is,</p> $\frac{N_4}{N_3} \times \frac{N_2}{N_1} = \frac{T_3}{T_4} \times \frac{T_1}{T_2} \quad \text{(But } N_3 = N_2, \text{ Since compound gear.)}$ $\frac{N_4}{N_1} = \frac{T_1 T_3}{T_2 T_4} \quad \dots (10.11)$ <p>i.e., $\text{Train value} = \frac{\text{Speed of follower}}{\text{Speed of driver}} = \frac{\text{Product of the number of teeth on the drivers}}{\text{Product of the number of teeth on the followers}}$</p>	Fig 4 + exp 2	6	30

Qn NO	Scoring indicator	Split up score	Sub Total	Total
6	 <p data-bbox="518 616 957 660">Fig. 6.12 Variation of coefficient of friction with $\frac{ZN}{p}$</p> <p data-bbox="319 694 1101 1008">The factor $\frac{ZN}{p}$ is a dimensionless number and is termed as <i>bearing characteristic number</i>. The curve obtained by plotting the relationship between the coefficient of friction and the bearing characteristic number is shown in Fig. 6.12. From the figure we see that the minimum amount of friction occurs at point A. The value of $\frac{ZN}{p}$ for the minimum value of coefficient of friction is known as <i>bearing modulus</i> and is denoted K. If $\frac{ZN}{p} > K$, the bearing may operate with thick film lubrication. On the other hand, if $\frac{ZN}{p} < K$, the oil film has ruptured and there is a metal to metal contact. An empirical data on small journal bearings to</p>	Exp 3 + Sig 3	1X6	

Qn NO	Scoring indicator	Split up score	Sub Total	Total
	<p style="text-align: center;">GENERAL PROCEDURE IN MACHINE DESIGN</p> <p>In designing a machine component, there is no rigid rule. The problem may be attempted in several ways. However, the general procedure to solve a design problem is as follows:</p> <ol style="list-style-type: none"> 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 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. 			
7	<p style="text-align: center;">GENERAL PROCEDURE IN MACHINE DESIGN</p>  <pre> graph TD A[Need or Aim] --> B[Synthesis (Mechanisms)] B --> C[Analysis of forces] C --> D[Material selection] D --> E[Design of elements (Size and stresses)] E --> F[Modification] F --> G[Detailed drawing] G --> H[Production] </pre>	Exp 6 or Diagram 6	1X6	

Qn NO	Scoring indicator	Split up score	Sub Total	Total
	PART C			
III a.	<p>Given :</p> <p>Diameter of cylinder, $D = 300$ mm</p> <p>Maximum steam pressure, $p = 1.2$ MPa = 1.2 N/mm²</p> <p>Number of studs, $n = 12$</p> <p>Safe tensile stress, $\sigma_t = 28$ MPa = 28 N/mm²</p> <p>Analysis :</p> <p>The force acting on the cylinder head of engine, $P = \frac{\pi}{4} D^2 p = \frac{\pi}{4} \cdot 300^2 \cdot 1.2$</p> <p>Resistance offered by the studs, $P = \frac{\pi}{4} d_c^2 \sigma_t n = \frac{\pi}{4} \times d_c^2 \times 28 \times 12$</p> <p>Equating the above two equations $\frac{\pi}{4} \times d_c^2 \times 28 \times 12 = \frac{\pi}{4} \cdot 300^2 \cdot 1.2$</p> $d_c^2 \times 28 \times 12 = 300^2 \times 1.2$ <p>\therefore Core diameter of studs, $d_c = \sqrt{\frac{300^2 \times 1.2}{28 \times 12}} = 17.93$ mm</p> <p>\therefore Nominal diameter, $d = \frac{d_c}{0.84} = \frac{17.93}{0.84} = 21.35$ say 22 mm</p> <p>Result :</p> <p>Size of the studs are M 22.</p>	Data 1+ Equ 3 + Ans 5	9	15

6

Qn NO	Scoring indicator	Split up score	Sub Total	Total
III b.	<p>Given :</p> <p>Tensile load , $P = 20 \text{ kN} = 20 \times 10^3 \text{ N}$</p> <p>Safe tensile stress, $\sigma_t = 100 \text{ MPa} = 100 \text{ N/mm}^2$</p> <p>Analysis :</p> <p>Using the relation for external tensile load, $P = \frac{\pi}{4} d_c^2 \sigma_t$</p>			
	<p>\therefore Core diameter of bolt, $d_c = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4 \times 20 \times 10^3}{\pi \times 100}} = 15.96 \text{ mm}$</p> <p>Using the relation core diameter, $d_c = 0.84d$</p> <p>\therefore Nominal diameter, $d = \frac{d_c}{0.84} = \frac{15.96}{0.84} = 19 \text{ say } 20 \text{ mm}$</p> <p>Result :</p> <p>Size of the bolt is M20.</p>			

7

Qn NO	Scoring indicator	Split up score	Sub Total	Total
IV a.	<p>Given:</p> <p>Maximum steam pressure, $p_1 = 1 \text{ MPa} = 1 \text{ N/mm}^2$</p> <p>Back pressure, $p_b = 0.015 \text{ MPa} = 0.015 \text{ N/mm}^2$</p> <p>Diameter of cylinder, $D = 300 \text{ mm}$</p> <p>Allowable stress, $\sigma_t = 45 \text{ MPa} = 45 \text{ N/mm}^2$</p> <p>Analysis :</p> <p>Intensity of effective steam pressure is obtained by using the relation</p> <p>Effective steam pressure, $p = p_1 - p_b = 1 - 0.015 = 0.985 \text{ N/mm}^2$</p> <p>Force acting on the piston rod of a steam engine, $P = \frac{\pi}{4} D^2 p = \frac{\pi}{4} \times 300^2 \times 0.985$</p> <p>Resistance offered by the screwed end of piston rod, $P = \frac{\pi}{4} d_c^2 \sigma_t = \frac{\pi}{4} \times d_c^2 \times 45$</p> <p>Equating the above two equations, $\frac{\pi}{4} \times d_c^2 \times 45 = \frac{\pi}{4} \times 300^2 \times 0.985$</p> $d_c^2 \times 45 = 300^2 \times 0.985$ <p>\therefore Core diameter of screwed end, $d_c = \sqrt{\frac{300^2 \times 0.985}{45}} = 44.38 \text{ mm}$</p> <p>$\therefore$ Nominal diameter, $d = \frac{d_c}{0.84} = \frac{44.38}{0.84} = 52.83 \text{ say } 56 \text{ mm}$</p> <p>Result :</p> <p>Nominal diameter of the screwed end of the piston rod is 56 mm.</p>	Data 1+ Equ 3 + Ans 4	9	

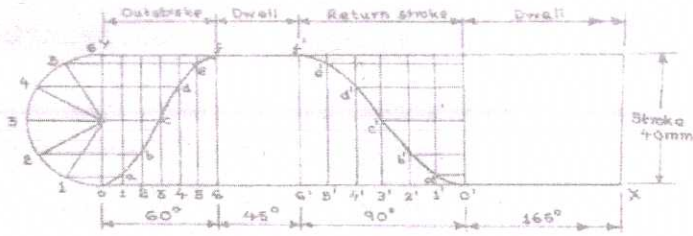
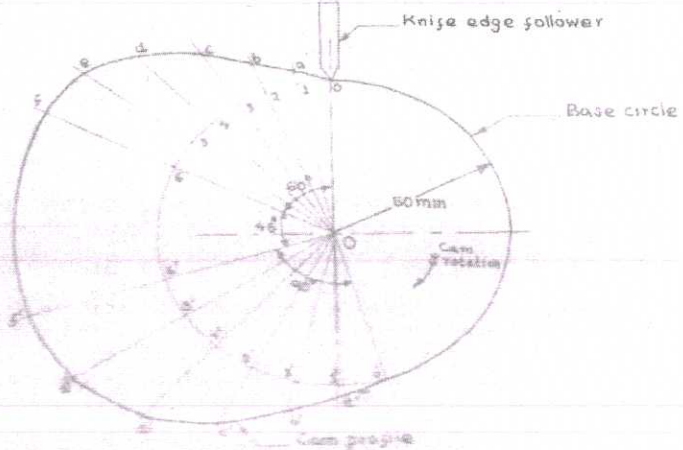
Qn NO	Scoring indicator	Split up score	Sub Total	Total
IV b.	<p>Given :</p> <p>Diameter of shaft, $d = 40$ mm</p> <p>Tangential force, $F = 20$ kN $= 20 \times 10^3$ N</p> <p>Allowable stress in key, $\tau = 60$ MPa $= 60$ N/mm²</p> <p>Analysis :</p> <p>Using the empirical relations</p> <p>Width of the key, $w = \frac{d}{4} = \frac{40}{4} = 10$ mm</p> <p>Thickness of the key, $t = \frac{d}{6} = \frac{40}{6} = 6.67$ say 8 mm</p> <p>Use a standard rectangular key 10 mm \times 8 mm</p> <p>Now consider the expression for tangential force $F = lwr$</p> <p>\therefore Length of the key, $l = \frac{F}{wr} = \frac{20 \times 10^3}{10 \times 60} = 33.33$ say 34 mm</p>	Data 1+ Equ 2+ Ans 3	6	15

Qn NO	Scoring indicator	Split up score	Sub Total	Total
V a.	<p>Given: $P = 40 \text{ kW}$ $N = 356 \text{ rpm}$</p> <p>$\tau_s = 30 \text{ N/mm}^2$ $\tau_m = 15 \text{ N/mm}^2$</p> <p>$T_{\text{max}} = 25\% \times T_{\text{mean}} = 1.25 T_{\text{mean}}$</p> <p><u>Design of shaft</u> $T_{\text{mean}} = \frac{60 \times 40 \times 10^3}{2 \times \pi \times 350} = 1091.35 \text{ Nm}$</p> <p>$T_{\text{max}} = 1.25 T_{\text{mean}} = 1.25 \times 1091.35 = 1364.19 \times 10^3 \text{ Nm}$</p> <p>$d = 70 \text{ mm}$</p> <p><u>Design of sleeve</u></p> <p>$\tau_m = \frac{16 T_{\text{max}}}{\pi D^3 (1 - K^4)} = 2.03 \text{ MPa}$</p> <p><u>Design of key</u></p> <p>width of key, $w = \frac{d}{4} = \frac{70}{4} = 17.5 \approx 18 \text{ mm}$</p> <p>Thickness of key, $t = \frac{d}{6} = \frac{70}{6} = 11.6 \approx 12 \text{ mm}$</p> <p>Length of key in each shaft, $l = \frac{L}{2} = \frac{245}{2} = 122.5 \text{ mm}$</p>			
			<p>Der 1 + Eg. 3 + Ans 5</p>	9

Qn NO	Scoring indicator	Split up score	Sub Total	Total
V b.	<p>Given : Torque, $T = 9.6 \text{ Nm} = 9.6 \times 10^3 \text{ Nmm}$ Length of shaft $L = 2 \text{ m} = 2000 \text{ mm}$ Angle of twist, $\theta = 2^\circ = \left(\frac{2 \times \pi}{180}\right)$ $= 0.0349 \text{ rad}$ Modulus of rigidity, $G = 0.8 \times 10^5 \text{ N/mm}^2$ Considering torsional rigidity: $\frac{T}{J} = \frac{G \theta}{L}$ or $J = \frac{T L}{G \theta} = \frac{9.6 \times 10^3 \times 2000}{0.8 \times 10^5 \times 0.0349}$ $= 6876.79 \text{ mm}^4$ For solid circular shafts $J = \frac{\pi D^4}{32} = 6876.79$ or $D = \sqrt[4]{\frac{6876.79 \times 32}{\pi}} = 16.26 \text{ mm Ans.}$ Considering shear strength: Torque, $T = \tau \cdot \frac{\pi D^3}{16}$ Shear stress induced, $\tau = \frac{16 \times T}{\pi D^3}$ $= \frac{16 \times 9.6 \times 10^3}{\pi \times 16.26^3}$ $= 11.38 \text{ N/mm}^2 \text{ Ans.}$</p>	Data 1+ Equ 2+ Ans 3	6	
VI a.	<p>$d = 100 \text{ mm}$, $n = 250 \text{ rpm}$, $T_{\text{max}} = 5 \text{ kNm} = 5 \times 10^6 \text{ Nmm}$ $\tau_{\text{cs}} = \tau_{\text{cb}} = \tau_{\text{ck}} = 50 \text{ N/mm}^2$ $\sigma_{\text{cb}} = \sigma_{\text{ck}} = 100 \text{ N/mm}^2$ <u>Design of shaft</u> $\rightarrow d = 100 \text{ mm}$ <u>Design of hub</u> $\rightarrow \text{OD} = 2d = 200 \text{ mm}$ $\text{ID} = d = 100 \text{ mm}$ Length of hub $= 1.5d = 150 \text{ mm}$ <u>Design of Plate</u> $\rightarrow \text{OD } D_0 = 4d = 400 \text{ mm}$ Thickness, $t = 0.5d = 50 \text{ mm}$ <u>Design of key</u> \rightarrow width, $w = d/4 = 25 \text{ mm}$ Thickness, $t = w = 25 \text{ mm}$ length of key in each shaft, $L = 150 \text{ mm}$ <u>Design of bolts</u> number of bolts, $n = \frac{4}{150} d + 3 = 6$ pitch diameter, $D_p = 3d = 300 \text{ mm}$ Bolt diameter, $d_1 = \sqrt{\frac{8 T_{\text{max}}}{\pi n \tau_{\text{cb}} D_p}} = 11.89 \approx 12 \text{ mm}$</p>	Data 1+ Equ 3+ Ans 5	9	

Qn NO	Scoring indicator	Split up score	Sub Total	Total
VI b.	<p>Given .</p> <p>Power transmitted, $P = 1 \text{ MW} = 1 \times 10^6 \text{ W}$</p> <p>Speed of the shaft, $N = 240 \text{ rpm}$</p> <p>Maximum torque, $T_{\text{max}} = 20\% \text{ more than mean torque}$ $= 120\% \text{ of mean torque} = 1.2 \bar{T}_{\text{mean}}$</p> <p>Maximum allowable shear stress, $\tau = 60 \text{ MPa} = 60 \text{ N/mm}^2$</p> <p>Analysis:</p> <p>Using the power equation.</p> <p>Power transmitted by the shaft, $P = \frac{2\pi N T_{\text{mean}}}{60}$</p> <p>Mean torque, $T_{\text{mean}} = \frac{60P}{2\pi N} = \frac{60 \times 1 \times 10^6}{2\pi \times 240} = 39788.74 \text{ N-m} = 39788.74 \times 10^3 \text{ N-mm}$</p> <p>Maximum torque, $T_{\text{max}} = 1.2 \bar{T}_{\text{mean}} = 1.2 \times 39788.74 \times 10^3 = 47746488 \text{ N-mm}$</p> <p>Using strength equation, $T_{\text{max}} = \frac{\pi}{16} \tau d^3$</p> <p>Diameter of solid shaft, $d = \sqrt[3]{\frac{16 T_{\text{max}}}{\pi \tau}} = \sqrt[3]{\frac{16 \times 47746488}{\pi \times 60}} = 159.44 \text{ say } 160 \text{ mm}$</p> <p>Result:</p> <p>Suitable diameter of the shaft have 160 mm</p>	Data 1+ Equ 2 + Ans 3	6	15

12

Qn NO	Scoring indicator	Split up score	Sub Total	Total
VII a.	 <p>(a) Displacement diagram</p>  <p>(b) Cam profile</p>	Fig 2x4.5	9	15

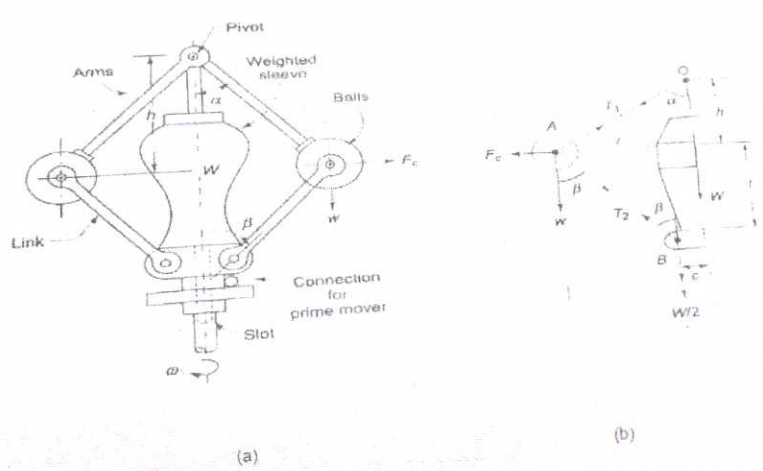
~~9~~ 13

Qn NO	Scoring indicator	Split up score	Sub Total	Total
VII b.	<p>Given :</p> <p>Diameter of journal, $d = 65 \text{ mm} = 65 \times 10^{-3} \text{ m}$</p> <p>Load on bearing, $W = 5 \text{ kN} = 5 \times 10^3 \text{ N}$</p> <p>Speed of journal, $N = 200 \text{ rpm}$</p> <p>Ratio of length to diameter, $\frac{l}{d} = 3$</p> <p>Coefficient of friction, $\mu = 0.02$</p> <p>Analysis :</p> <p>Given the ratio of length to diameter, from this find out the length of the bearing</p> <p>$\therefore l = 3d = 3 \times 65 = 195 \text{ mm}$</p> <hr/> <p>Projecting bearing area, $A = ld = 195 \times 65 = 12675 \text{ mm}^2$</p> <p>Bearing pressure, $p_b = \frac{W}{A} = \frac{5 \times 10^3}{12675} = 0.39 \text{ N/mm}^2 = 0.39 \text{ MPa}$</p> <p>Rubbing velocity, $v = \frac{\pi d N}{60} = \frac{\pi \times 65 \times 10^{-3} \times 200}{60} = 0.68 \text{ m/s}$</p> <p>$\therefore$ Heat generated by friction, $Q_g = \mu W v = 0.02 \times 5 \times 10^3 \times 0.68 = 68 \text{ W or J/s}$</p> <p>Result :</p> <p>Bearing pressure is 0.39 MPa and the heat generated by friction is 68 J/s.</p>	Data 1+ Equ 2 + Ans 3	6	

14

Qn NO	Scoring indicator	Split up score	Sub Total	Total
VIII a.	<p>Given :</p> <p>Diameter of shaft, $d = 180$ mm</p> <p>Load on bearing, $W = 30$ kN $= 30 \times 10^3$ N</p> <p>Coefficient of friction, $\mu = 0.04$</p> <p>Speed of shaft, $N = 120$ rpm.</p> <p>Analysis :</p> <p>Using the equation for mean radius of shaft, i.e.,</p> $R = \frac{2}{3}r = \frac{2}{3} \times 90 = 60 \text{ mm}$ <p>\therefore Frictional torque, $T = \mu WR = 0.04 \times 30 \times 10^3 \times 60 = 72000 \text{ N-mm} = 72000 \times 10^{-3} \text{ N-m}$</p> <p>$\therefore$ Power lost in friction, $P = \frac{2\pi NT}{60} = \frac{2 \times \pi \times 120 \times 72000 \times 10^{-3}}{60} = 904.78 \text{ W}$</p> <p>Result :</p> <p>Power lost in over coming friction 904.78 W</p>	Data 1+ Equ 3 + Ans 5	9	

15

Qn NO	Scoring indicator	Split up score	Sub Total	Total
VIII b.	 <p style="text-align: center;">(a) (b)</p> <p style="text-align: center;">Fig 8.3 Porter governor</p> <p>Simple watt governor is unsuitable for high speeds. However this drawback has been overcome by loading the governor with a dead weight or by means of a spring. If the sleeve of a watt governor is loaded with a heavy weight, it becomes a porter governor. Fig 8.3 shows a porter governor which is dead weight type of centrifugal governor. It consists of two masses called the governor balls attached to the spindle with the help of four links or arms. The lower arms are attached to the sleeve which acts as a central weight. Governor balls rotate at different speeds depends upon the load on the engine about the axis of the governor shaft, which is driven through suitable gearing from the engine crankshaft. The increase of speed, the governor balls fly outwards and the sleeve moves upwards thus closing the working fluid passage till the engine speed comes back to its designed speed. On the other hand the speed of rotation of balls decreases as the load on the engine increases and the governor balls move near to the governor axis due to reduction in the centrifugal force on the fly balls. At that time the sleeve moves downwards thus opening the working fluid passage more till the engine speed comes back to its designed speed. The speed of rotation of balls due to the centrifugal force is just balanced by the inward controlling force provided by dead weight. Fig 8.3 (b) shows the forces acting on the governor.</p>	Fig 4 + Exp 2	6	15

16

Qn NO	Scoring indicator	Split up score	Sub Total	Total
IX a.	<p>Given :</p> <p>Diameter of larger pulley, $d_2 = 600$ mm</p> <p>Diameter of smaller pulley, $d_1 = 400$ mm</p> <p>Central distance between the pulleys, $C = 6$ m $= 6 \times 10^3$ mm</p> <p>Analysis :</p> <p>Case(a) : Open belt drive.</p> <p>Using the relation for length of belt in open belt drive</p> $L_o = \frac{\pi}{2}(d_2 + d_1) + 2C + \frac{(d_2 - d_1)^2}{4C}$ $= \frac{\pi}{2}(600 + 400) + 2 \times 6 \times 10^3 + \frac{(600 - 400)^2}{4 \times 6 \times 10^3} = 13572.46 \text{ mm} = 13.57 \text{ m}$ <p>Case(b): Crossed belt drive.</p> <p>Using the relation for length of crossed belt</p> $L_c = \frac{\pi}{2}(d_2 + d_1) + 2C + \frac{(d_2 + d_1)^2}{4C} = \frac{\pi}{2}(600 + 400) + 2 \times 6 \times 10^3 + \frac{(600 + 400)^2}{4 \times 6 \times 10^3}$ $= 13612.46 \text{ mm} = 13.61 \text{ m}$	Data 1+ Equ 3 + Ans 5	9	

Qn NO	Scoring indicator	Split up score	Sub Total	Total
IX b.	<p>Given :</p> <p>Diameter of pulley, $d = 400 \text{ mm} = 400 \times 10^{-3} \text{ m}$</p> <p>Speed of pulley, $N = 750 \text{ rpm}$</p> <p>Tension on tight side, $T_1 = 300 \text{ N}$</p> <p>Tension on slack side, $T_2 = 45.35 \text{ N}$</p> <p>Analysis :</p> <p>Using the expression for power transmitted ,i.e.,</p> $P = (T_1 - T_2) \frac{\pi d N}{60} = (300 - 45.35) \frac{\pi \times 400 \times 10^{-3} \times 750}{60}$ $= 4000.03 \text{ W} = 4 \text{ kW}$ <p>Result :</p> <p>Power transmitted by flat belt drive is 4 kW</p>	Data 1+ Equ 2 + Ans 3	6	15
X a.	<p>Diameter of pulley , $d = 600 \text{ mm} = 600 \times 10^{-3} \text{ m}$</p> <p>Speed of pulley, $N = 200 \text{ rpm}$</p> <p>Coefficient of friction between belt and pulley , $\mu = 0.25$</p> <p>Angle of lap, $\theta = 160^\circ = 160 \times \frac{\pi}{180} = 2.79 \text{ rad}$</p> <p>Maximum tension, $T_1 = 2.5 \text{ kN} = 2.5 \times 10^3 \text{ N}$</p> <p>Analysis :</p> <p>Using the power equation for belt drive</p> $P = T_1 \left(1 - \frac{1}{e^{\mu \theta}} \right) \frac{\pi d N}{60} = 2.5 \times 10^3 \left(1 - \frac{1}{e^{0.25 \times 2.79}} \right) \times \frac{\pi \times 600 \times 10^{-3} \times 200}{60}$ $= 7888.09 \text{ W} = 7.89 \text{ kW}$ <p>Result :</p> <p>Power transmitted by belt is 7.89 kW.</p>	Data 1+ Equ 3 + Ans 5	9	

Qn NO	Scoring indicator	Split up score	Sub Total	Total
X b.	<p>Advantages</p> <ol style="list-style-type: none"> 1. They give positive drives and constant speed ratio without any slippage. 2. The drive is more compact due to shorter center distance used in such drives. 3. Can be operated at higher speeds. 4. High efficiency, reliable in service and simple operation. 5. Lighter loads on the shafts and bearings. 6. Used where precise timing is desired. 7. Wide range of power transmitted. 8. Maintenance is unexpensive. 9. Can be used for non intersecting and non parallel shafts <p>Disadvantages</p> <ol style="list-style-type: none"> 1. Manufacture of gears is complex. Special tools and equipments are needed to manufacturing. 2. Not suitable for large centre distance shafts. 3. Errors and inaccuracy in manufacturing gears causes vibrations and noises during operation. 4. Requires perfect alignment of shafts. 5. Requires more attention to lubrication. 	Adv 3+ Dis 3	6	15