Model Answer

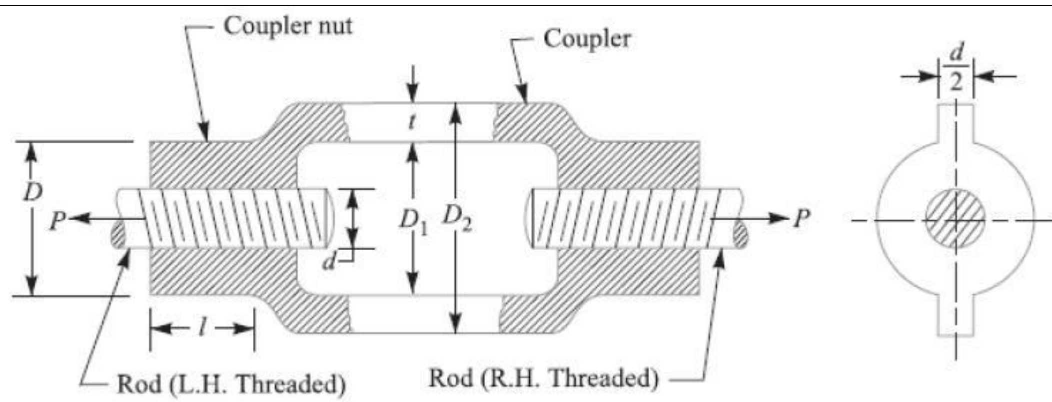
Subject Name: Design and Drawing of Auto Components **Subject Code:** 17525

Important Instructions to examiners:

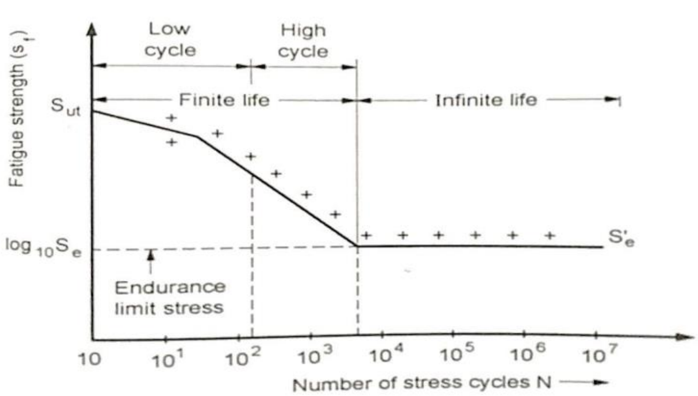
1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling 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.

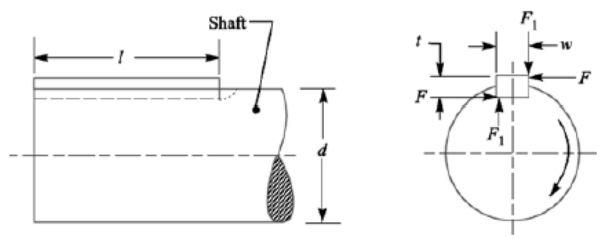
Q. No	Sub Q.N.	Answer	Marking Scheme
1	A)	**Attempt any THREE**	12
1A	i)	**State eight considerations in machine design**	04
		Answer: Design considerations in machine design: (Any eight) 1. Types of loads and stresses caused by the load. 2. Motion of parts and kinetics of machine. 3. Material selection 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
1A	ii)	**Define standardization and state the four advantages of it.**	04
		Answer: *(Definition- 2mark, Advantages-2 mark)* **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. Mass production is easy. 2. Rate of production increases.	02 02

		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.	
1A	**iii)**	**Define (a) shaft (b) axle (c) spindle (d) key.**	**04**
		(a) Shaft : shaft is a rotating element which is used to transmit power from one place to another. (b) Axle: An axle is a stationary machine element and is used for the transmission of bending moment only. It simply act as a support for some rotating body such as hoisting drum, a car wheel or a rope sheave. (c) Spindle : spindle is a short shaft that imparts motion either to a cutting tool or to a work piece. (d) Key: Key is a piece of mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them.	**04**
1A	**iv)**	**List the important factors that influence the magnitude of F.O.S.**	**04**
		Answer : ***(Any Four – 1 Marks Each)*** **The factors that influence the magnitude of factor of safety:** 1. The reliability of applied load and nature of load, 2. The reliability of the properties of material and change of these properties during service, 3. The reliability of test results & accuracy of application of these results to actual machine parts, 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 setup during manufacture, 8. The extent of loss of property if failure occurs, 9. The extent of loss of life if failure occurs.	**04**
1	**B)**	Attempt any **ONE:**	**06**
1B	**i)**	**The rear axle shaft connecting differential to side wheel is required to transmit 40 kW at 1600 rpm. If Maximum torque is two times average torque and allowable shear stress is 80 N/mm^2 for axle shaft material, find out diameter of axle shaft if (a) shaft is solid (b) shaft is hollow with outside diameter 1.6 times inside diameter.**	**06**
		Given Data: **$P = 40$ kW $= 40 \times 10^3$ W** **$N = 1600$ rpm,** **$T_{max} = 2 T_{avg}$,** **$\tau = 80$ N/mm^2 ,**	

		(a) **Shaft is solid** **Let d = diameter of shaft** **We know that torque transmitted by shaft,** $Tavg = \frac{P \times 60}{2\pi N} = \frac{40000 \times 60}{2\pi \times 1600} = 238.73$ N-m $= 238.73 \times 10^3$ N-mm **Max torque transmitted by shaft** $Tmax = 2 \times Tavg = 2 \times 238.73 \times 10^3 = 477.46 \times 10^3$ N-mm **We also know that max. torque transmitted by shaft,** $Tmax = \frac{\pi}{16} \times d^3 \times \tau$ $477.46 \times 10^3 = \frac{\pi}{16} \times d^3 \times 80$ $\therefore \boxed{d = 31.20 \text{ mm say } 32 \text{ mm}}$ (b) **Shaft is hollow with outside diameter 1.6 times inside diameter.** **Let di = inside diameter of hollow shaft** **do = outside diameter of shaft = 1.6 di** **k = di/ do = di/ 1.6 di = 0.625** **we know that for hollow shaft max. torque transmitted by shaft,** $Tmax = \frac{\pi}{16} (do)^3 \times \tau \times (1 - k^4)$ $477.46 \times 10^3 = \frac{\pi}{16} (do)^3 \times 80 \times (1 - 0.625^4)$ $\therefore \boxed{do = 32.97 \text{ mm say } 34 \text{ mm}}$ And $\boxed{di = 34 \times 0.625 = 21.25 \text{ mm}}$	01 01 01 01 01 01
1B	**ii)**	**Draw neat sketch of turn buckle joint. Also write design procedure for it.**	**06**
		 **Fig. Turn Buckle Joint** **Design procedure for Turn Buckle:**	02

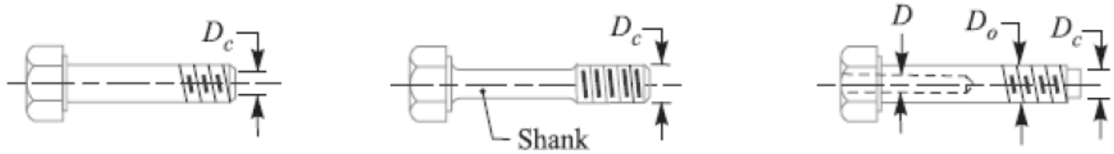
		1. To design diameter of rod:- $$P_d = \frac{\pi}{4} d_c^2 \sigma_t$$ $$d = \frac{d_c}{0.84}$$ *Where,* P_d = Design Load d = diameter of rod d_c = Core diameter of the rod σ_t = Allowable tensile stress	01
		2. To design diameter of Coupler Nut:- $$\therefore P_d = \frac{\pi}{4} (D^2 - d^2)\sigma_t$$ *Where,* D = Diameter of the Coupler nut	01
		3. To design diameter of Coupler :- $$\therefore D_1 = d + 6$$ $$\therefore P = \frac{\pi}{4} (D_2^2 - D_1^2)\sigma_t$$ *Where,* D_1 = Inside Diameter of the Coupler D_2 =Outside Diameter of the Coupler P = Load on turn buckle	01

		4. To design length of Coupler Nut :- i. Failure in shear: $\therefore P_d = \pi\, d_c \times l \times \sigma_s$ ii. Failure in crushing: $\therefore P_d = \frac{\pi}{4}\ (d^2 - d_c^2) \times n \times l \times \sigma_c$ *Where,* l = length of the threaded portion of Coupler nut σ_s = Allowable shear stress σ_c = Allowable crushing stress	01
2.		**Attempt any FOUR:**	**16**
2	**i)**	Define fatigue and endurance limit. Draw the S-N curve for cyclic loading.	**04**
		i) Fatigue: When the system or element is subjected to fluctuating (repeated) loads, the material of system or element tends to fails below yield stresses by the formation of progressive crack this failure is called as fatigue. The failure may occur without prior indication. The fatigue of material is affected by the size of component, relative magnitude of static and fluctuating load and number of load reversals. **ii) Endurance limit:** It is defined as maximum value of the completely reversed bending stress which a polished standard specimen can withstand without failure, for infinite number of cycles (usually 10^7 cycles). The term endurance limit is used for reversed bending cycle only. The endurance limit of ma depends on: Type of load, Surface finish, Size of object, Working temperature.  Fig. S-N curve	**01** **01** **02**

2	**ii)**	**For a square key equally strong in shearing and crushing . Show that the permissible crushing stress is twice the shear stress.**	**04**
		Answer: Assume, 'd' (mm) is diameter of shaft. Empirical relations used to design rectangular sunk key are, 1. The length of the key(l) = $1.5\ d$ mm 2. Width of key (w) = $\frac{d}{4}$ mm 3. Thickness of key (t) = $\frac{d}{6}$ mm  Fig. shows forces acting on a key for clockwise torque transmitted from a shaft to hub.	**01**
		We know that, 1. Strength equation considering shearing failure of key, $T = \tau \times l \times b \times \frac{d}{2}$ .... (a) 2. Crushing equation considering crushing failure of key, $T = \sigma c \times l \times t \times \frac{d}{4}$ .... (b) And for square key, $t = b$ Substituting and equating the above equations (a) & (b), $\tau \times l \times b \times \frac{d}{2} = \sigma c \times l \times b \times \frac{d}{4}$ $\tau = \sigma c \times \frac{1}{2}$ $\sigma c = 2\tau$ So for a square key induced crushing stress is twice the shear stress	**01** **02**

2	iii)	**Explain the two methods to make bolt of uniform strength.**	04

Answer: Bolts of uniform strength:



(a) (b) (c)

(02)

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. (01)
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.

$$\therefore \quad \frac{\pi}{4}D^2 = \frac{\pi}{4}\left[(D_o)^2 - (D_c)^2\right]$$ (01)

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

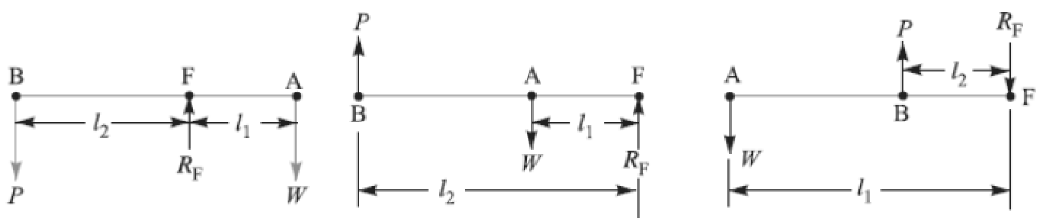
$$\therefore \quad D = \sqrt{(D_o)^2 - (D_c)^2}$$

2	iv)	**A knuckle joint is required to withstand a tensile load of 30 kN. Design the joint if the permissible stresses are 56 N/mm² in tension, 40N/mm² in shear and 70 N/mm² in crushing respectively.**	04

Given Data:

$$P = 30 \times 10^3 \text{ N}$$
$$\sigma_s = 40 \text{ N/mm}^2$$
$$\sigma_t = 56 \text{ N/mm}^2$$

		σ_c = 70 N/mm^2 **i. Find Diameter of rod:-** $P=\frac{\pi}{4}d^2\sigma_t$ $30\times10^3=\frac{\pi}{4}d^2\times56$ d = 26.11 mm $\boxed{\textbf{d = 28 mm}}$ **ii. Find dimensions of fork end, eye end and knuckle pin by empirical relations:-** 1. Diameter of knuckle pin d_1=d = 28 mm 2. Outer diameter of eye end d_2=2d = 56 mm 3. Diameter of knuckle pin head or collar d_3=1.5d = 42 mm 4. Thickness of eye end t=1.25d = 35 mm 5. Thickness of forked end t_1=0.75d = 21 mm 6. Thickness of collar or head t_2=0.5d = 14 mm **iii. Induced stress in knuckle pin:-** $\therefore P = 2\times\frac{\pi}{4}d_1^2\times\sigma_s$ $\therefore 30\times10^3 = 2\times\frac{\pi}{4}28^2\times\sigma_s$ $\therefore \boxed{\sigma_c = 24.36\ N/mm^2} <$ permissible shear stress Therefore Design is safe. **iv. Induced stresses in eye end:-** 1. Failure in tension: $\therefore P = (d_2 - d_1)t\times\sigma_t$ $\therefore 30\times10^3 = (56-28)35\times\sigma_t$ $\therefore \boxed{\sigma_t = 30.61\ N/mm^2} <$ permissible tensile stress Therefore Design is safe. 2. Failure in shear: $\therefore P = (d_2 - d_1)t\times\sigma_s$ $\therefore 30\times10^3 = (56-28)35\times\sigma_s$ $\therefore \boxed{\sigma_s = 30.61\ N/mm^2} <$ permissible shear stress Therefore Design is safe. 3. Failure in crushing: $\therefore P = d_1\,t\times\sigma_c$ $\therefore 30\times10^3 = 28\times35\times\sigma_c$ $\therefore \boxed{\sigma_c = 30.61\ N/mm^2} <$ permissible crushing stress Therefore Design is safe. **Induced stresses in forked end:-**	½ ½ ½ ½

		1. Failure in tension: $\therefore P = 2(d_2 - d_1)t_1 \times \sigma_t$ $\therefore 30 \times 10^3 = 2(56 - 28)21 \times \sigma_t$ $\therefore \boxed{\sigma_t = 25.51\ N/mm^2}$ < permissible tensile stress Therefore Design is safe. 2. Failure in shear: $\therefore P = 2(d_2 - d_1)t_1 \times \sigma_s$ $\therefore 30 \times 10^3 = 2(56 - 28)21 \times \sigma_s$ $\therefore \boxed{\sigma_s = 25.51\ N/mm^2}$ < permissible shear stress Therefore Design is safe. 3. Failure in crushing: $\therefore P = 2(d_2 - d_1)t_1 \times \sigma_c$ $\therefore 30 \times 10^3 = 2 \times 28 \times 21 \times \sigma_c$ $\therefore \boxed{\sigma_c = 24.36\ N/mm^2}$ < permissible crushing stres Therefore Design is safe.	½ ½ ½ ½
3		**Attempt any FOUR:**	**16**
3	i)	**Define lever. Describe three basic types of lever.**	**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	**04**

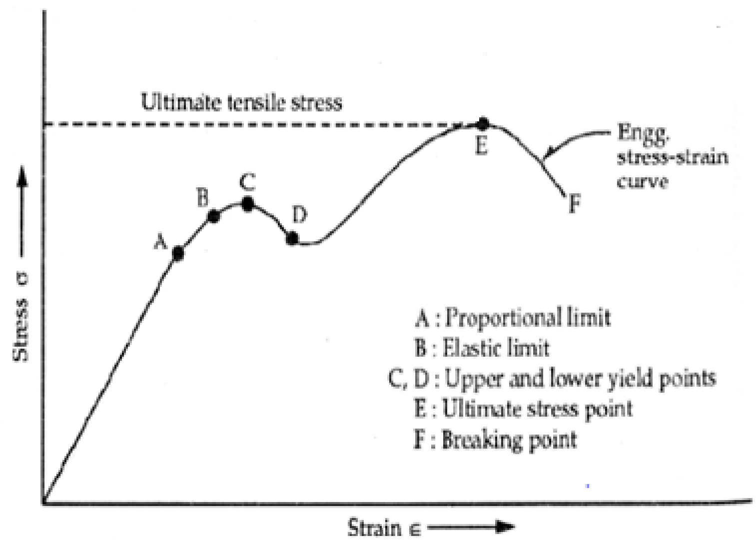
		signaling arrangement, rocker arm in internal combustion engines, handle of a hand pump, hand wheel of a punching press, beam of a balance, foot lever etc. b) **Second type lever:** In the second type of levers, the load is in between the fulcrum and effort. In this case, the effort arm is more than 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.	
3	**ii)**	**Design a propeller shaft to transmit 8 kW at 6500 rpm with gear box reduction of 16:1. Assume shear stress fs = 52 N/mm^2.**	**04**
		Answer: Given Data: **$P = 8\times10^3$W,** **N=6500rpm** **G_1=16:1,** **$f_s = \sigma_s$=52 N/mm^2** Now torque produced by the engine, $$P = \frac{2\,\pi N T_e}{60}$$ $$8\times10^3 = \frac{2\,\pi\times6500\times T_e}{60}$$ $$\boldsymbol{T_e = 11.75\ \text{Nm} = 11.75\ \times10^3\ \text{Nmm}}$$ Torque transmitted by the propeller shaft, $T_p = T_e\times G_1$ $T_p = 11.75\times10^3\times 16$ **$T_p = 188.04 \times 10^3$ Nmm** Diameter of propeller shaft, $$T_p = \frac{\pi}{16}\ \sigma_s\ d^3$$	**01** **01**

		$188.04 \times 10^3 = \frac{\pi}{16}\ 52\ d^3$ **d = 26.40 mm** $\boxed{\textbf{d=28 mm}}$	**02**
3	**iii)**	**Design the piston crown thickness from the following data- diameter of piston = 90 mm, max. pressure on the piston = 4.6 N/mm^2 and allowable bending stress = 45 N/mm^2.**	**04**
		Given : **D= 90mm, P=4.6 N/mm^2 , σ_b = 45 N/mm^2** **Let the thickness of the piston head can be designed by assuming the head to be a flat plate uniform thickness and fixed at the edges and assuming the gas load to be uniform distributed .** $t_h = \sqrt{\frac{3\ P\ D^2}{16\ \sigma_b}}$ **where** **t_h = thickness of piston crown** **P =Pmax**	**02**
		$\therefore t_h = \sqrt{\frac{3\ X\ 4.6\ X 90^2}{16\ \ X\ 45}}$ $\therefore t_h = 12.32$ mm Say $\boxed{t_h = 13 \text{ mm}}$	**02**
3	**iv)**	**Compare hand lever and foot lever on the basis of (a) Effort required to operate (b) Cross section used (c) Application (d) material**	**04**
		Answer*(1 mark for each parameter)*	**04**

Parameter	Hand lever	Foot lever
1. Load carrying capacity	400N	800N
2. Cross section used	Circular, Rectangular or cross shaped.	Circular, Rectangular or cross shaped.
3. Operational method	Hand operated.	Foot operated.
4. Applications	Hand pump, Clutch lever and brake lever of motorcycle, hand brake.	Rear brake lever of motorcycle, Four wheeler clutch, brake, accelerator lever.

3	**v)**	**Explain aesthetic consideration in designing automobile components.**	**04**
		Answer***: (Any two – 2 marks each)*** **Aesthetic consideration in designing automobile components:** 1. **Shape:** The external appearance is an important feature, which gives grace & luster to the product. This is true for automobile, household appliances. The role of designer is to create the new shapes of machines which have aesthetic look. E.g. Aerodynamic shape of aero plane for functional requirements to resist minimum air resistance. 2. **Colour:** Selection of proper colour is an impotent consideration in product design. Many colors are associated with different conditions. Morgan has suggested the meaning of colors in the following table. **Colour** — **Meaning** Red — Danger-Hazard- Hot Orange — Possible danger Yellow — Caution Green — Safety Blue — Caution-Cold Grey — Dull Product 3. **Surface finish:** For greater strength, bearing loads, good fatigue life & wear qualities of product, and the good surface finish is required. Better surface finish always attracts the observers.	**04**
4	**A)**	**Attempt ant THREE:**	**12**
4A	**i)**	**Write applications of cotter joint, knuckle joint and turn buckle.**	**04**
		Answer:(any Eight applications irrespective of type of joint- 1/2 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 Knuckle joint: □ Link of cycle chain □ Tie rod joints for roof truss □ Valve rod joint for eccentric rod pump rod joint	

		□ Tension link in bridge structure □ Lever and rod connection of various types. □ swing arm of two wheeler □ Connection of link rod of leaf springs in multi axle vehicles □ Piston ,Piston Pin ,Connecting Rod □ Connections of leaf spring with chassis iii) 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	**04**
4A	**ii)**	**A multi disc clutch has 5 plates having 4 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 radi of friction surfaces are 130mm and 80 mm respectively. Assume uniform wear and take co-efficient of friction = 0.35.**	**04**
		Given: $n_1 + n_2 = 5$, $n = 4$, $p = 0.127$ N/mm^2 , $N = 500$ rpm, $r_1 = 130$mm, $r_2 = 80$mm , $\mu = 0.35$. **we know that for uniform wear p.r = C (a constant). Since the intensity of pressure is maximum at inner radius r_2,** **therefore** $p. r_2 = C$ or $C = 0.127 X 80 = 10.16$ N/mm **axial force required to engage the clutch ,** $W = 2 \pi C (r_1 – r_2)$ $\therefore W = 2 \pi X 10.16 (130 – 80)$ $\therefore W = 3191.85$ N **Mean radius2 of friction surfaces,** $R = \frac{r_1+r_2}{2} = \frac{130+80}{2} = 105$ mm $= 0.105$ m **We know that torque transmitted ,** $T = n. \mu.W.R$ $= 4 X 0.35 X 3191.85 X 0.105$ $= 469.2$ Nm $\therefore$ **Power transmitted,** $P = \frac{2 \pi N T}{60}$ $= \frac{2 X \pi X 500 X 469.2}{60}$ $= 24567.35$ W $\boxed{P= 24.567 kW}$	**01** **01** **01** **01**

4A	**iii)**	**Draw stress diagram for ductile material and state its importance.**	**04**
		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**	**02** **02**
4A	**iv)**	**Define indicated power and brake power of an engine cylinder.**	**04**
		Answer: **Indicated Power** The power developed inside the cylinder is known as indicated power. It is called as indicated power because it is measured from indicator diagram. $$ip = \frac{p_{im} \; LAnk}{60000} kW$$ **Where**	**02**

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ip = indicated power (kW)
p_{im} = indicated mean effective pressure (N/M^2)
L= Length of the stroke (m)
A= area of the piston (m^2)
N= speed in revolutions per minute
n= number of power strokes engine
N/2 for a four – stroke engine
N for a two –stroke engine
K= number of cylinders

Brake Power:
This is the actual power delivered at the crankshaft. It is obtained by deducting various power losses in the engine from indicated power. Brake power is what would keep the vehicle running at any speed once you have accelerated
B.P. (in kW) can be calculated with the formula
B.P.= 2Π NT/ 60000
Where N= Engine speed in R.P.M.
T= Torque in newton meters
OR
$bp = \frac{p_{im} \; LAnk}{60000} kW$
Where
p_{im} = Brake mean effective pressure

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<tr><td>4</td><td>B)</td><td>**Attempt any ONE:**</td><td>06</td></tr>
<tr><td></td><td>i)</td><td>**Design a rigid flange coupling to transmit a torque of 250 N-m. The shaft, key, bolt are made of alloy steel and flange are made of C.I. The allowable stresses for shaft material in shear – 100 MPa, in crushing 250 MPa and allowable stresses for C.I. in Shear – 20MPa.**</td><td>06</td></tr>
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Given : T = 250 N-m = 250 × 10^3 N-mm ; n = 4; τ_s = 100 MPa = 100 N/mm^2 ; σ_{cs} = 250 MPa = 250 N/mm^2 ; τ_k = 100 MPa = 100 N/mm^2 ; σ_{ck} = 250 MPa = 250 N/mm^2 ; τ_c = 20 MPa = 20 N/mm^2 ; τ_b = 100 MPa = 100 N/mm^2
The cast iron flange coupling of the protective type is designed as discussed below :
1. *Design for hub*
First of all, let us find the diameter of the shaft (d). We know that the torque transmitted by the shaft (T),
$$250 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 100 \times d^3 = 19.64\, d^3$$
∴ d^3 = 250 × 10^3 / 19.64 = 12 729 or d = 23.35 say 25 mm
We know that the outer diameter of the hub,
D = 2 d = 2 × 25 = 50 mm
and length of hub, L = 1.5 d = 1.5 × 25 = 37.5 mm

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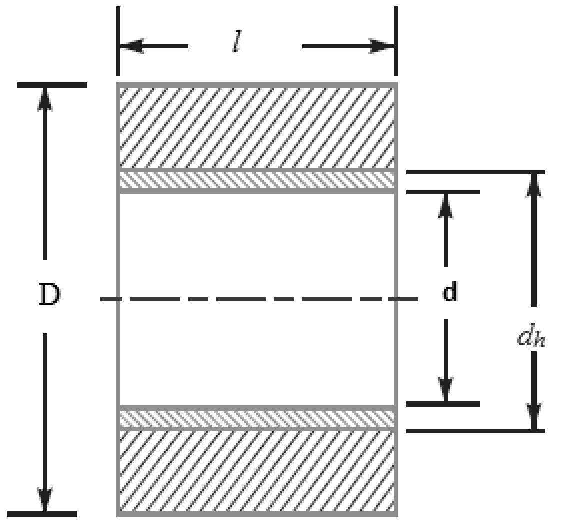
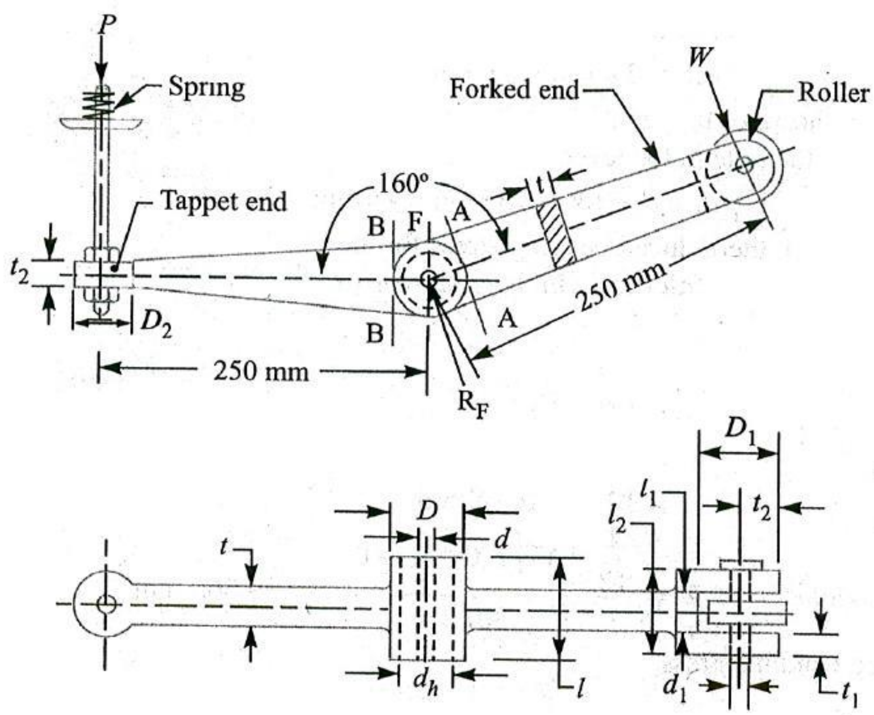
		Let us now check the induced shear stress in the hub by considering it as a hollow shaft. We know that the torque transmitted (T), $$250 \times 10^3 = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D}\right) = \frac{\pi}{16} \times \tau_c \left[\frac{(50)^4 - (25)^4}{50}\right] = 23\,013\,\tau_c$$ $\therefore$ $\tau_c = 250 \times 10^3/23\,013 = 10.86\ \text{N/mm}^2 = 10.86\ \text{MPa}$ Since the induced shear stress for the hub material (*i.e.* cast iron) is less than 20 MPa, therefore the design for hub is safe.	01
		2.Design for Key From key properties, we find that the proportions of key for 25 mm shaft are, Width of key, $w = 10$ mm Ans. and thickness of key, $t = 8$ mm Ans. The length of key (l) is taken equal to the length of hub, $\therefore$ $l = L = 37.5$ mm Ans. Let us now check the induced shear and crushing stresses in the key. Considering the key in shearing. We know that the torque transmitted (T), $$250 \times 10^3 = l \times w \times \tau_k \times \frac{d}{2} = 37.5 \times 10 \times \tau_k \times \frac{25}{2} = 4688\,\tau_k$$ $\therefore$ $\tau_k = 250 \times 10^3 / 4688 = 53.3\ \text{N/mm}^2 = 53.3\ \text{MPa}$ Considering the key in crushing. We know that the torque transmitted (T), $$250 \times 10^3 = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2} = 37.5 \times \frac{8}{2} \times \sigma_{ck} \times \frac{25}{2} = 1875\,\sigma_{ck}$$ $\therefore$ $\sigma_{ck} = 250 \times 10^3 / 1875 = 133.3\ \text{N/mm}^2 = 133.3\ \text{MPa}$ Since the induced shear and crushing stresses in the key are less than the given stresses, therefore the design of key is safe.	01
		3. *Design for flange* The thickness of the flange (t_f) is taken as 0.5 d. $\therefore$ $t_f = 0.5\,d = 0.5 \times 25 = 12.5$ mm Ans. Let us now check the induced shear stress in the flange by considering the flange at the junction of the hub in shear. We know that the torque transmitted (T), $$250 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi (50)^2}{2} \times \tau_c \times 12.5 = 49\,094\,\tau_c$$ $\therefore$ $\tau_c = 250 \times 10^3 / 49\,094 = 5.1\ \text{N/mm}^2 = 5.1\ \text{MPa}$ Since the induced shear stress in the flange of cast iron is less than 20 MPa, therefore design of flange is safe.	01
		4.Design for bolts Assuming no of bolts n = 4	

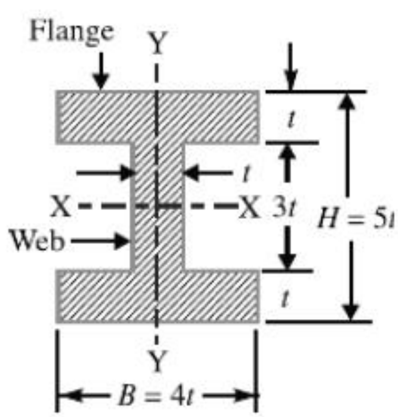
		d_1 = Nominal diameter of bolts. We know that the pitch circle diameter of bolts, $\therefore \quad D_1 = 3\,d = 3 \times 25 = 75$ mm Ans. The bolts are subjected to shear stress due to the torque transmitted. We know that torque transmitted (T), $250 \times 10^3 = \frac{\pi}{4}(d_1)^2 \tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4}(d_1)^2\, 100 \times 4 \times \frac{75}{2} = 11\,780\,(d_1)^2$ $\therefore \quad (d_1)^2 = 250 \times 10^3 / 11\,780 = 21.22$ or $d_1 = 4.6$ mm Assuming coarse threads, the nearest standard size of the bolt is M 6. Ans. Other proportions of the flange are taken as follows : Outer diameter of the flange, $D_2 = 4\,d = 4 \times 25 = 100$ mm Ans. Thickness of the protective circumferential flange, $t_p = 0.25\,d = 0.25 \times 25 = 6.25$ mm Ans.	01
	ii)	**Explain design procedure of a rocker arm for operating exhaust valve.**	**06**
		Answer: **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), $\therefore$ Bearing pressure $= \frac{\text{Load}}{\text{Bearing area}} = \frac{R_F}{l \times d} = \frac{R_F}{1.25d \times d}$ From here, l and d can be determined. **(b) Checking shear stress induced in the fulcrum pin. As the pin is in double shear,** $\tau = \frac{R_F}{2 \times \left(\frac{\pi}{4} \cdot d^2\right)}$ External diameter of the boss, $D = 2\,d$ Internal diameter of the hole in the lever, $d_h = d + (2 \times 3)$ check the induced bending stress for the section of the boss at the fulcrum	01 01 01 01 01

		 Bending moment at this section = $W \times L$ Section Modulus $Z = 1/12 \times l \times (D^3 - d_h^3) / D/2$ Induced bending stress, $$\sigma_b = \frac{M}{Z}$$ ***OR*** 	**01**

		In designing a rocker arm the following procedure may be followed : 1. Rocker arm is usually I-Section it is subjected to bending moment. To find bending moment it is assumed that the arm of the lever extends from point of application of load to center of pivot. 2. 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. 3. 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. 4. One end of rocker arm has a forked end to receive roller. 5. The outside diameter of the eye at the forked end is also taken as twice the diameter of pin. The diameter of roller is slightly larger (at least 3mm more) than the diameter of eye at the forked end. The radial thickness of each eye of the forked end is taken half the diameter of pin. Some clearance about 1.5mm must be provided between the roller and the eye at the forked end so that roller can move freely. The pin should, therefore be checked for bending. 6. The other end of rocker arm (i.e. tappet end) is made circular to receive the tappet which is a stud with a lock nut. The outside diameter of the circular arm is taken as twice the diameter of the stud. The depth of section is also taken twice the diameter of stud.	
5		**Attempt any TWO:**	**16**
5	**i)**	**A 4-stroke diesel engine has the following specifications:** **Brake power =6 kW, speed = 1200 rpm, Indicated mean effective pressure = 0.35 N/mm^2, Mechanical efficiency = 80 %. Determine,** **(a) Bore and length of cylinder** **(b) Thickness of cylinder head**	**08**
		(Note: Assume l = 1.5 D OR l = 1.08 D , Constant C=0.1 , Tensile stress for cylinder cover=52 N/mm^2) **Given:** **B.P. = 6 kW = 6000 W** **N = 1200 rpm** **n = 1200/2 = 600 rpm …………………..…… for four stroke engine** **P_m = 0.35 N/mm^2** **η_m = 80 % = 0.8** **for cylinder cover σ_t = 52N/mm^2……………………………...Assumed** **Length of stroke L = 1.5 D= 1.5 D/ 1000 m……………………Assumed** **1. Bore and Length of cylinder** **Let D = bore of cylinder in mm** **A = cross sectional Area of cylinder**	

		$= \frac{\pi}{4} D^2$ **We know that Indicated Power** $I.P. = \frac{B.P.}{\eta_m} = \frac{6000}{0.8} = 7500\ W$ **We also know that** $I.P. = \frac{P_m\,L\,A\,n}{60}$ $7500 = \frac{0.35\ X\ 1.5\ D\ X\ \pi D^2\ X\ 600}{60\ X\ 1000\ X\ 4}$ $7500 = 4.12 \times 10^{-3}\ D^3$ $D^3 = \frac{7500}{4.12\ X\ 10^{-3}}$ $D^3 = 1818.91\ X\ 10^3$ $D = 122.06$ mm **Say** $\boxed{D = 124\ mm}$ $L = 1.5\ D = 1.5\ X\ 124 = 186$ mm **Taking a clearance on both sides of the cylinder equal to 15% of the stroke, therefore length of the cylinder,** $\boxed{\text{Length of cylinder} = 1.15\ X\ L = 1.15\ X\ 186 = 213.9 = 214\ mm}$ **2. Thickness of the cvlinder head :** Since the maximum pressure (P) in the engine cylinder is taken as 9 to 10 times means effective pressure (Pm) therefore let us take $P = 9P_m = 9 \times 0.35 = 3.15 N/mm^2$ We know that thickness of the cylinder head $t_h = D\sqrt{\frac{C\ X\ P}{\sigma_t}}$ $t_h = 124\ X\ \sqrt{\frac{(0.1\ X\ 3.15)}{52}}$ $t_h = 9.65$ mm say $\boxed{t_h = 10\ mm}$	01 02 01 01 01 02
5	ii)	**A truck spring has 12 numbers of leaves , two of which are full length leaves. The spring supports are 1.05 m apart and central band is 85 mm wide. The central load is to be 5.4 kN with a permissible stress of 280 N/mm^2. Determine the thickness of and width of steel spring leaves. The ratio of total depth to width of the spring is 3. Also determine the deflection of the spring.**	08

		Answer : Given : $n=12$, $n_F=2$, $2L_1=1.05m=1050mm$, $l=85mm$, $2W=5.4\,kN=5400N$ or $W=2700\,N$, $f_F=280\,N/mm^2$. Thickness and width of the spring leaves Let, $t=$ Thickness of the leaves, and $b=$ Width of the leaves Since it is given that the ratio of the total depth of the spring $(n\times t)$ width of the spring (b) is 3 , therefore $\frac{n\times t}{b}=3$ $\frac{12\times t}{b}=3$ or $b=12\,t/3=4t$	01
		We know that the effective lenght of the spring, $2L=2L_1-l=1050-85=965mm$ $L=\frac{965}{2}=482.5\,mm$ and number of graduated leaves , $n_G=n-n_F=12-2=10$ Assuming that the leave are not initially stressed therefore maximum stress or bending stress for full length leaves (f_F)	01
		$280=\frac{18\,W.L}{b.t^2(2n_g+3n_F)}=\frac{18\times 2700\times 482.5}{4\,t\times t^2(2\times 10+3\times 2)}=\frac{225.476}{t^3}$ $t^3=225.476/280=805.3$	02
		or $t=9.3$ say 10 mm and $b=4\,t=4\times 10=40mm$	01
		Deflection of the spring : We know that deflection the spring $\delta=\frac{12\,W.L^3}{E.b.t^3(2n_g+3n_F)}=\frac{12\times 2700\times(482.5)^3}{0.21\times 10^6\times 40\times 10(2\times 10+3\times 2)}\,mm$ $=16.7\,mm$ …………….. (Taking $E=0.21\times 10^6\,N/mm^2$)	01

			01 **01**
5	**iii)**	**Explain the design procedure of connecting rod.**	**08**
		Answer: Design of Connecting Rod 1. ***Dimensions of cross-section of the connecting rod*** According to Rankine's formula, $W_B = \dfrac{\sigma_c . A}{1 + a\left(\dfrac{L}{k_{xx}}\right)^2}$ Let A = Cross-sectional area of the connecting rod=11 t^2 L =Effective length of the connecting rod, σ_c = Crippling or Buckling stress, W_B = Buckling load, a = Rankine's constant $k_{xx}^2 = 3.18\ t^2$ from this relation t (thickness of the flange and web of the section) can be determined.  *Fig a I*-section of connecting rod. Width of the section, $B = 4\ t$ and depth or height of the section, $H = 5t$ The dimensions $B = 4\ t$ and $H = 5\ t$, as obtained above by applying the Rankine's formula, are at the middle of the connecting rod. The width of the section (B) is kept constant throughout the length of the connecting rod, but the depth or height varies. The depth near the small end (or piston end) is taken as $H_1 = 0.75\ H$ to $0.9H$ The depth near the big end (or crank end) is taken $H_2 = 1.1H$ to $1.25H$.	**01** **01** **01**

		2. Dimensions of the at the big end and small end of connecting rod Maximum gas force, $F_L = \frac{\pi D^2}{4} \times p$ **(i)** where D = Cylinder bore or piston diameter in mm, and p = Maximum gas pressure in N/mm^2 Let d_c = Diameter of the crank pin in mm, l_c = Length of the crank pin in mm, pb_c = Allowable bearing pressure in N/mm^2, and d_p, l_p and pb_p = Corresponding values for the piston pin, load on the crank pin = Projected area × Bearing pressure $= d_c \cdot l_c \cdot pb_c$ (ii) Similarly, load on the piston pin $= d_p \cdot l_p \cdot pb_p$ *(iii)* Equating equation ***(i)*** and ***(ii)***, we have $F_L = d_c \cdot l_c \cdot pb_c$ Taking $l_c = 1.25\, dc$ to $1.5\, d_c$, the value of d_c and l_c are determined from the above expression. Again, equating equations ***(i)*** and ***(iii)***, we have $F_L = d_p \cdot l_p \cdot pb_p$ Taking $l_p = 1.5\, d_p$ to $2\, d_p$, the value of d_p and l_p are determined from the above expression. **3. *Size of bolts for securing the big end cap*** F_I = Inertia load acting on bolts Let dc_b = Core diameter of the bolt in mm, σ_t = Allowable tensile stress for the material of the bolts in MPa, and n_b = Number of bolts. Generally two bolts are used. Force on the bolts	**01** **01** **01**

		$F_I = \frac{\pi}{4}(d_{cb})^2 \sigma_t \times n_b$ From this expression, *dcb* is obtained. The nominal or major diameter (d_b) of the bolt is given by $d_b = \frac{d_{cb}}{0.84}$ **4. *Thickness of the big end cap*** The thickness of the big end cap (t_c) may be determined as below, Maximum bending moment acting on the cap will be taken as $M_C = * \frac{F_I \times x}{6}$ where , x = Distance between the bolt centres. = Dia. of crankpin or big end bearing (d_c) + 2 × Thickness of bearing liner (3 mm) + Clearance(3mm) Let b_c = Width of the cap in mm. It is equal to the length of the crankpin or big end bearing (l_c), and σ_b = Allowable bending stress for the material of the cap in MPa. Section modulus for the cap, $Z_C = \frac{b_c (t_c)^2}{6}$ ∴ Bending stress, $\sigma_b = \frac{M_C}{Z_C} = \frac{F_I \times x}{6} \times \frac{6}{b_c (t_c)^2} = \frac{F_I \times x}{b_c (t_c)^2}$ From this expression, the value of t_c is obtained.	01 01
6		**Attempt any TWO**	16
6	i)	**A four speed gear box is to constructed for providing the ratio 1.0, 1.46, 2.28 and 3.93 to 1as nearly as possible. The diametral pitch of gear is 3.25 mm and the smallest pinion is to have at least 15 teeth.** **Determine the suitable number of teeth of the different gear. Also calculate the distance between main and layout shaft.**	08
		Answer: ***(Assume module of 3.25mm instead of diametral pitch)***	01 01 01

		$$G_1 = \frac{T_B}{T_A} \times \frac{T_D}{T_C} = 3.93$$ We have $\frac{T_B}{T_A} \times \frac{T_D}{T_C} = \sqrt{3.93} = 1.98$ Adopting $T_A = T_C = 15$ the lowest value given We get $T_B = T_D = 1.98 \times 15 = 29.7 = 30$ Thus actual ratio $= \frac{30}{15} \times \frac{30}{15} = 4:1$ $$T_A + T_B = T_C + T_D = T_E + T_F = T_G + T_H = 45$$ Second gear ratio $$G_2 = \frac{T_B}{T_A} \times \frac{T_F}{T_E} = 2.28$$ Or $\frac{T_F}{T_E} = 2.28 \times \frac{T_A}{T_B} = 2.28 \times \frac{15}{30} = 1.14$ Hence, $T_E + T_F = 2.14 \times T_E = 45$ Or $T_E = \frac{45}{2.14} = 21$	01 01 01

		and $T_F = 45 - 21 = 24$ The actual ratio $= \frac{30}{15} \times \frac{24}{21} = 2.286:1$ Third gear ratio, $G_3 = \frac{T_B}{T_A} \times \frac{T_H}{T_G} = 1.46$ Or $\frac{T_H}{T_G} = \frac{1.46}{2} = 0.73$ But $T_H + T_G = 45$ Or $T_G = \frac{45}{1.73} = 26$ Hence, $T_H = 45 - 26 = 19$ Actual ratio $= \frac{30}{15} \times \frac{19}{26} = 1.461 : 1$ Top gear ratio $G_4 = 1:1$ **The centre distance between the shaft** $= \frac{3.25 \times 45}{2}$ $= 73.125\,\text{mm}$	02
6	ii)	**Determine the thickness of plain cylinder head for 0.4 m cylinder diameter. The maximum gas pressure is 3.2 N/mm^2. Design the studs and cylinder cover. Take allowable tensile stress for cylinder cover and bolt equal to 42 N/mm^2 and 63 N/mm^2 repectively.**	08
		Given : **D = 0.4 m = 400 mm,**	

		$P = 3.2\ N/mm^2$ **$C = 0.1$ ……………………………Assumed.** **$\sigma_{t(cylinder)} = 42\ N/mm^2$** **$\sigma_{t(bolt)} = 63\ N/mm^2$**	
		1. Thickness of Plain Cylinder $t_h = D\sqrt{\frac{C\ X\ P}{\sigma_{t(cylinder)}}}$ $t_h = 400\ x\ \sqrt{\frac{0.1\ X\ 3.2}{42}}$ **$t_h = 34.91$ mm** **say $\boxed{t_h = 35\ mm}$**	01 02
		2. Design of studs and cylinder cover **Let d = nominal dia od stud** **dc = core dia of stud (0.84 d)** **$\sigma_{t(bolt)} = 63\ N/mm^2$** **$n_s$ = no. of studs** **we know that the force acting on the cylinder head (or on the studs)** $= \frac{\pi}{4}\ X\ D^2\ X\ P$ $= \frac{\pi}{4}\ X\ 400^2\ X\ 3.2$ **= 402123.8 N**	 01
		The number of studs usually taken between **$n_s = 0.01$ D(i.e (0.01 X 400) + 4 = 8) and 0.02 D + 4 (i.e. (0.02 X 400) +4 = 12)** **taking $n_s = 12$** **we know that resisting force offered by all the studs** $402123.8 = n_s\ X\ \frac{\pi}{4}(dc)^2\ X\ \sigma_{t(bolt)} = 12\ X\ \frac{\pi}{4}(0.84\ d)^2\ X\ 63$ $\therefore d = 28.39$ mm $\therefore \boxed{d = 30\ mm}$	 02
		The pitch circle diameter of the stud (D_p) is taken as D + 3d **$D_p = 400 + (3\ X\ 30)$** **= 490 mm** **We know that pitch of the studs** $= \frac{\pi\ D_p}{n_s} = \frac{\pi\ X\ 490}{12} = 128.28$ mm **For leak proof joint , the pitch of the stud should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$** **Where d is nominal diameter of the stud.**	 01

		∴ minimum pitch of the stud $= 19\sqrt{d} = 19 \times \sqrt{30} = 104.06$ mm And maximum pitch of the stud $= 28.5\sqrt{d} = 28.5 \times \sqrt{30} = 156.10$ mm Since the pitch of the stud obtained above (i. e. 128.28 mm) lies between 104.06 mm and 156.10 mm, therefore size of the stud (d) calculated above is satisfactory. $\therefore \boxed{d = 30mm}$	01
6	iii)	A single plate dry clutch transmit 8kW at 940 rpm, the axial pressure is limited to 0.7 N/mm^2. If coefficient of friction is 0.25, find; a) Mean radius and face width of friction lining assuming ratio of mean radius to face width as 4 and b) Outer and inner radii of clutch plate.	08
		Given Data: n = 2, Power P = 8kW = 8000 W, N = 940 rpm Co-efficient of friction $\mu = 0.25$ Maximum intensity of pressure, $P_{max} = 0.7 N/mm^2$ Let , r_1 and r_2 = outer and inner radius of frictional surfaces respectively r = mean radius of the friction lining in mm b= face width of friction lining Ratio of mean radius to the face width $\frac{r}{b} = 4$ We know that area of frication faces $= 2\Pi rb \times P$ Therefore normal or axial force acting on friction faces $W = A \times P$ $= 2\Pi rb \times P$ Torque transmitted, $T = n\,\mu\,W\,r$(unifrom wear) $= n\,\mu\,(2\Pi\,r\,b \times P)\,r$ $= n\,\mu\,[2\,\Pi\,r \times (r/4) \times P]\,r$ $= (\pi/2)\,n\,\mu\,P\,r^3$ $= (\pi/2) \times 2 \times 0.25 \times 0.7 \times r^3$	01

		$= 0.5497\ r^3$ N-mm……………………………………….(1)	
		Power Transmitted	
		$P = \frac{2\pi NT}{60}$	01
		$8000 = \frac{2\pi \times 940 \times T}{60}$	
		$\therefore T = 81.27$ N-m	
		$\therefore T = 81.27 \times 10^3$ N-mm…………………………………………….(2)	
		From equation (1) and (2)	
		$r^3 = \frac{81.27 \times 10^3}{0.5497}$	01
		$\therefore r^3 = 147.845 \times 10^3$	
		$\therefore r = 52.877$ mm = 53 mm Approximately.	
		Now face width of the friction lining	
		$b = \frac{r}{4} = \frac{53}{4} = 13.25$	01
		we know that	
		$b = r_1 - r_2 = 13.25$ mm……………………………………..(3)	
		$r = \frac{r_1+r_2}{2}$	
		$\therefore r1 + r2 = 2\,r = 2 \times 53 = 106$mm ………………………(4)	01
		Equating equations (3) and (4)	
		$\boxed{r_1 = 59.625 \text{ mm}}$ and	01
		$\boxed{r_2 = 46.375 \text{ mm}}$	
			01
			01