Important Instructions to examiners:

1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling errors should not be given more importance. (Not applicable for subject English and Communication Skills).
4) While assessing figures, examiner may give credit for principal components indicated in the figure. The figures drawn by candidate and model answer may vary. The examiner may give credit for any equivalent figure drawn.
5) Credits may be given step wise for numerical problems. In some cases, the assumed constant values may vary and there may be some difference in the candidate's answers and model answer.
6) In case of some questions credit may be given by judgment on part of examiner of relevant answer based on candidate's understanding.
7) For programming language papers, credit may be given to any other program based on equivalent concept.

……

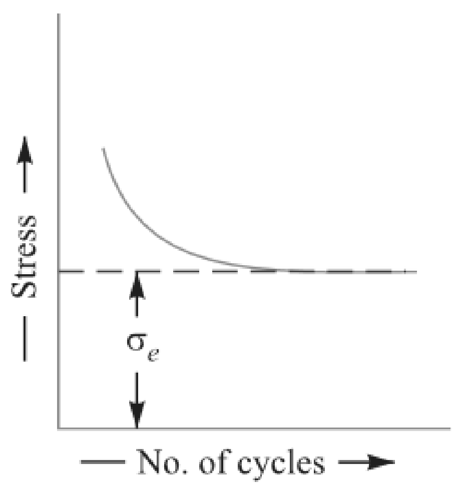
	Marks
1. A) **Attempt any five:**	20
a) Describe fatigue and endurance limit.	04

Answer:

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. — 02

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). — 02

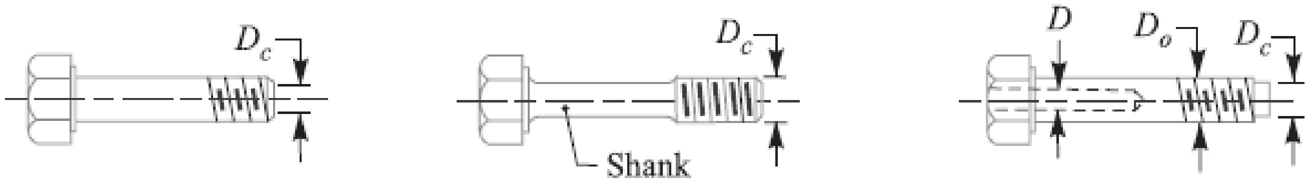


The term endurance limit is used for reversed bending cycle only. The endurance limit of material depends on:

- Type of load, Surface finish, Size of object, Working temperature.

b) Describe the concepts of " Bolts 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. — 02
2. A second alternative method of obtaining the bolts of uniform strength is shown in Fig. (c). In this method, an axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread.

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

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

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

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

c) State strength equations to design spigot, spigot end, spigot collar.	04

Answer:

1. Strength equation to design spigot end:

$$P = \frac{\Pi}{4} d^2 \boldsymbol{\sigma}_t$$ — 01

Where,
P = Load carried by the rods
d = Diameter of the rods
σ_t = Allowable tensile stress

2. Strength equation to design spigot (at slot):
   i) failure in tension:

$$P = \left[\frac{\Pi}{4} d_2^2 - d_2 t\right] \boldsymbol{\sigma}_t$$ — 02

$\therefore$ Assume $t = \frac{d_2}{4}$ ……. standard practice *Where,* d_2= Diameter of Spigot t = thickness of cotter ii) failure in crushing: $P = d_2.t.\sigma_c$ *Where,* σ_c= Allowable crushing stress 3. Strength equation to design spigot collar: i) failure in crushing: $P = \frac{\Pi}{4}\left(d_3^2 - d_2^2\right)\sigma_c$ *Where,* d_3= Outside diameter of spigot collar ii) failure in shear: $P = \Pi.d_2.t_1\sigma_s$ *Where,* t_1= Thickness of spigot collar	01
d) Design rectangular key for a shaft of 50mm diameter. The allowable shear and crushing stresses for key material are 42 MPa and 70 MPa respectively.	04
Answer: **Given data:** **d = 50mm** **σ_{sk}= 42 N/mm^2** **σ_{ck}= 70 N/mm^2** i) Width and thickness for rectangular key: $\therefore w = \frac{d}{4} = \frac{50}{4} = 12.5 \cong 13mm$ $\therefore t = \frac{d}{6} = \frac{50}{6} = 8.33 \cong 9mm$ ii) Torque transmitted by key: $T = \frac{\pi}{16}\,\sigma_s\,d^3$ Assume, **σ_s for shaft= 42 N/mm^2** $T = \frac{\pi}{16} \times 42 \times 50^3$ $\therefore T = 10.308 \times 10^5\ Nmm$	01 01

iii) Length of key by considering failure in shear: 01

$$T = l \times w \times \sigma_{sk} \times \frac{d}{2}$$

$$10.308 \times 10^5 = l \times 13 \times 42 \times \frac{50}{2}$$

$$\therefore l = 75.51mm$$

$$\therefore l \cong 76mm$$

iv) Length of key by considering failure in crushing: 01

$$T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$$

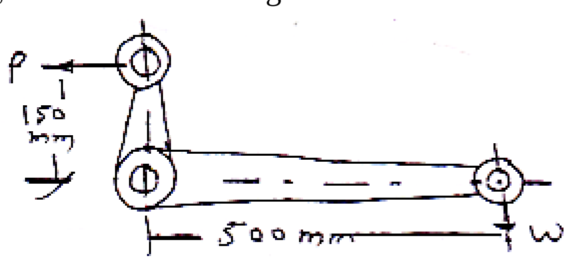
$$10.308 \times 10^5 = l \times \frac{9}{2} \times 70 \times \frac{50}{2}$$

$$\therefore l = 130.79mm$$

$$\therefore l \cong 131mm$$

Take maximum value of length of key for safe design $\therefore l \cong 131mm$

e) For a right angled bell crank lever, horizontal arm is 500 mm long and a load of 4.5 kN acts on its at its end. At the end of vertical arm, 150mm in length a force ‘P’ acts as shown. Calculate force ‘p’, mechanical advantage and reaction at fulcrum. 04



Answer:

Given data:

Load $W = 4500$ N

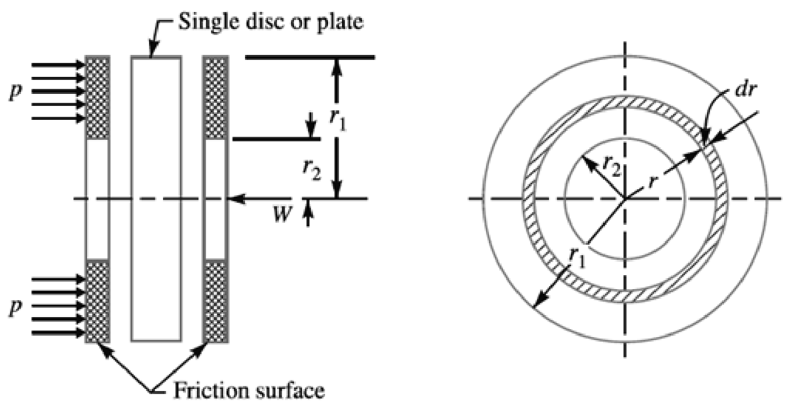
Length of load arm $L_1 = 500$ mm

Length of effort arm $L_2 = 150$ mm

i. Bending movement of right angled bell crank lever at fulcrum pin:

$$(W \times L_1) - (P \times L_2) = 0$$

$$W \times L_1 = P \times L_2$$

$4500 \times 500 = P \times 150$ $\therefore \mathbf{P = 15\ kN}$	02
ii. Mechanical advantage: $M.A. = \dfrac{\text{effort arm distance}}{\text{load arm distance}} = \dfrac{150}{500}$ $\therefore \mathbf{M.A. = 0.3}$	01
iii. Reaction at fulcrum pin: $R_f = \sqrt{W^2 + P^2} = \sqrt{4.5^2 + 15^2}$ $\therefore \mathbf{R_f = 15.66\ kN}$	01
f) State the design procedure of single plate clutch using wear condition.	04
Answer: **Design procedure of single plate clutch using wear condition:-**  Fig. Forces on a single plate clutch Consider two friction surfaces maintained in contact by an axial thrust (*W*) as shown in Fig. Let, W = Axial force/thrust T = Torque transmitted by the clutch, p = Intensity of axial pressure r_1 and r_2 = External and internal radii of friction faces, r = Mean radius of the friction face, and μ = Coefficient of friction. b=face width of frictional surface.	01

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

We know that area of the contact surface or friction surface = $2\pi.r.dr$

Therefore Normal or axial force on the ring,

δW = Pressure × Area = $p \times 2\pi.r.dr$

and the frictional force on the ring acting tangentially at radius *r*,

$Fr = \mu.\ \delta W = \mu.p \times 2\pi.r.dr$ 01

Therefore Frictional torque acting on the ring,

$Tr = Fr \times r = \mu.p \times 2\pi.r.dr \times r = 2\pi\mu.p.\ r^2.dr$

Considering uniform axial wear:

Let, P be the normal intensity of pressure at a distance r from the axis of clutch, so

$$Pr = c \qquad P = \frac{c}{r}$$

and the normal force on the ring,

$$\delta W = p.2\pi r.dr = \frac{C}{r} \times 2\pi r.dr = 2\pi C.dr$$

$\therefore$ Total force acing on the friction surface,

$$W = \int_{r_2}^{r_1} 2\pi\ C\ dr = 2\pi\ C\ [r]_{r_2}^{r_1} = 2\pi\ C\ (r_1 - r_2)$$

or

$$C = \frac{W}{2\pi\ (r_1 - r_2)}$$

We know that the frictional torque acting on the ring,

$$T_r = 2\pi\ \mu.p.r^2.dr = 2\pi\ \mu \times \frac{C}{r} \times r^2\ .dr = 2\pi\mu.C.r.dr \qquad ...(\because p = C/r)$$ 02

$\therefore$ Total frictional torque acting on the friction surface (or on the clutch),

$$T = \int_{r_2}^{r_1} 2\pi\ \mu\ C.r.dr = 2\pi\ \mu C\left[\frac{r^2}{2}\right]_{r_2}^{r_1}$$

$$= 2\pi\mu\ .C\left[\frac{(r_1)^2 - (r_2)^2}{2}\right] = \pi\ \mu.C\ [(r_1)^2 - (r_2)^2]$$

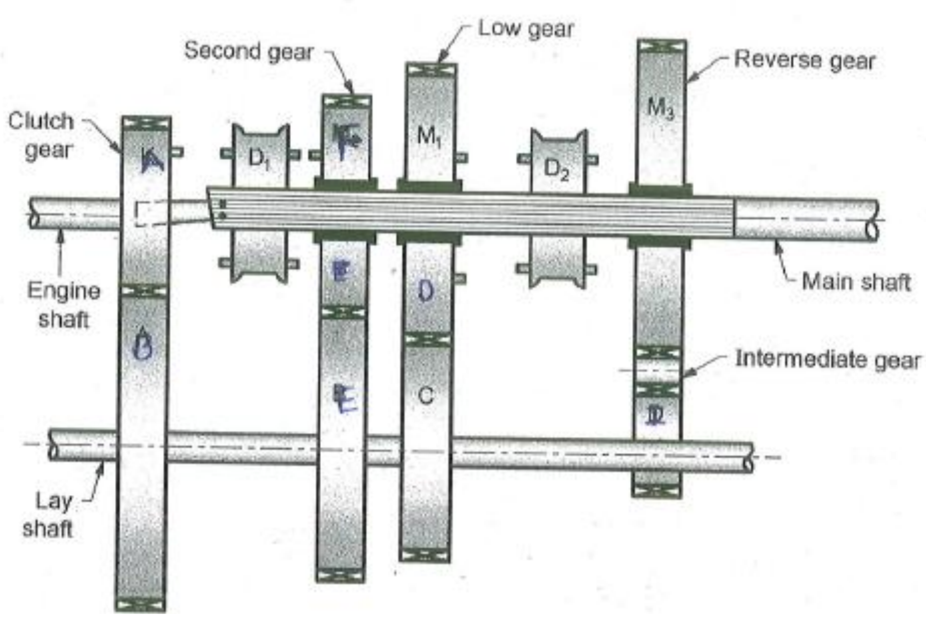
$$= \pi\mu \times \frac{W}{2\pi\ (r_1 - r_2)}[(r_1)^2 - (r_2)^2] = \frac{1}{2} \times \mu.W\ (r_1 + r_2) = \mu.W.R$$

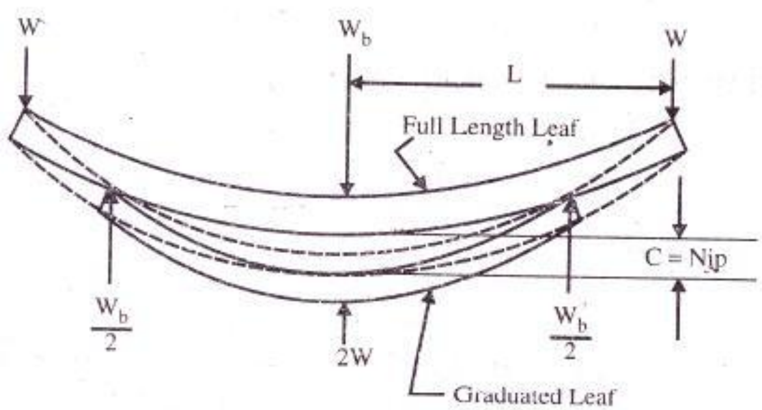
where $R = \dfrac{r_1 + r_2}{2}$ = Mean radius of the friction surface.

g) State any four criteria's for selection of "factor of safety".	04
Answer: **Criteria's for selection of factor of safety: *(Any four)*** 1. Degree of economy desired. 2. The reliability of applied load and nature of load, 3. The reliability of the properties of material and change of these properties during service, 4. The reliability of test results & accuracy of application of these results to actual machine parts, 5. The certainty as to exact mode of failure, 6. The extent of simplifying assumptions, 7. The extent of localized stresses, 8. The extent of initial stresses setup during manufacture, 9. The extent of loss of property if failure occurs, 10. The extent of loss of life if failure occurs.	04
2. Attempt any four :	16
a) Define : 1. Strength 2. Stiffness 3. Creep 4. Resilience	04
Answer : *(Note: one mark each)* **1. Strength:** It is ability of material to resist the externally applied force without breaking or yielding. **2. Stiffness:** It is the rigidity of an object - the extent to which it resists deformation in response to an applied force. **3. Creep:** If the metal is subjected to a constant load (below elastic limit) at high temperature for a long period of time, then it will undergo slow and permanent deformation called as creep. **4. Resilience:** It is property of material to absorb the energy and to resist shock and impact load.	04
b) Two rods of 52mm diameter are joined by knuckle joint to transmit a load of 150kN. Determine induced stresses in single eye and knuckle pin.	04
Answer : **Given Data:** **$d = 52mm$** **$P = 150 \times 10^5$ N** **i. Find dimensions of single eye and knuckle pin by empirical relations:-** When, d= Diameter of rod 1. Diameter of knuckle pin $d_1 = d = 52$ mm 2. Outer diameter of eye $d_2 = 2d = 104$ mm 3. Diameter of knuckle pin head or collar $d_3 = 1.5d = 78$ mm	01

4. Thickness of single eye rod t=1.25d = 65 mm	
ii. Induced stress in knuckle pin:- $$\therefore P = 2 \times \frac{\pi}{4} d_1^2 \times \sigma_s$$ $$\therefore 150 \times 10^3 = 2 \times \frac{\pi}{4} 52^2 \times \sigma_s$$ $$\therefore \sigma_s = 35.31 \frac{N}{mm^2}$$	01
iii. Induced stresses in single eye:- (*any two*) 1. Failure in tension: $$\therefore P = (d_2 - d_1)t \times \sigma_t$$ $$\therefore 150 \times 10^3 = (104 - 52)65 \times \sigma_t$$ $$\therefore \sigma_t = 44.37 \frac{N}{mm^2}$$ 2. Failure in shear: $$\therefore P = (d_2 - d_1)t \times \sigma_s$$ $$\therefore 150 \times 10^3 = (104 - 52)65 \times \sigma_s$$ $$\therefore \sigma_s = 44.37 \frac{N}{mm^2}$$ 3. Failure in crushing: $$\therefore P = d_1 t \times \sigma_c$$ $$\therefore 150 \times 10^3 = 52 \times 65 \times \sigma_c$$ $$\therefore \sigma_c = 44.37 \frac{N}{mm^2}$$	02
c) State strength equation only, required to design turn buckle.	04
Answer: : **Strength equations of turn buckle:** 1. To design diameter of rod:- $$P_d = \frac{\pi}{4} d_c^2 \sigma_t$$ $$\therefore d = \frac{d_c}{0.84}$$	01

Where, P_d = Design Load d = diameter of rod d_c = Core diameter of the rod σ_t = Allowable tensile stress 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 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 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 01 01
d) Describe design procedure for full floating rear axle.	04
Answer: *Note: Credit should be given to sketch if drawn* **Design procedure of a fully floating rear axle:** The rear axle is designed on the basis of shaft design. By using the torsional equation, $$\frac{T_{RA}}{J_{RA}} = \frac{\sigma_s}{r}$$ *Where,* T_{RA} = Torque transmitted by rear axle shaft. $T_{RA} = T_e \times G_1 \times G_d$	04

T_e = Engine Torque. G_1 = Maximum gear Ratio in Gear Box G_d = Final gear reduction in differential J_{RA} = Polar moment of inertia. $= \pi/32 \times d^4$ ……….. …… (for Solid shaft) $= \frac{\pi}{32}(d_o^4 - d_i^4)$ ………..(for Hollow shaft) σ_s = Torsional shear stress. r = distance from neutral axis to outer most fiber. r = d/2 (for Solid shaft) r = d_o/2 (for Hollow shaft) After simplifying the equations, $T_{RA} = \frac{\pi}{16}\sigma_s d^3$ ………………….For solid shaft $T_{RA} = \frac{\pi}{16}\sigma_s d_o^3 (1 - k^4)$ ………… For hollow shaft $k = \frac{d_i}{d_o}$ d_i = Inner diameter of shaft d_o = Outer diameter of shaft From these equations, we can find out the diameter of rear axle of shaft.	
e) An automotive gear box given three forward and a reverse speed. Sketch neat layout and state all gear ratio in terms of number of teeth on each gear pair.	04
Answer : **Three forward and a reverse speed gear box:** 	02

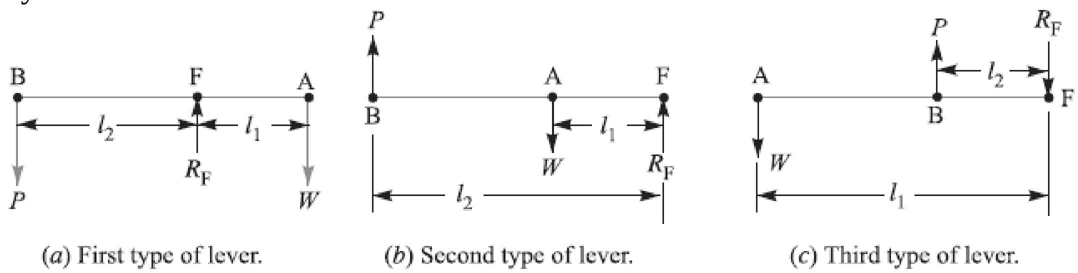
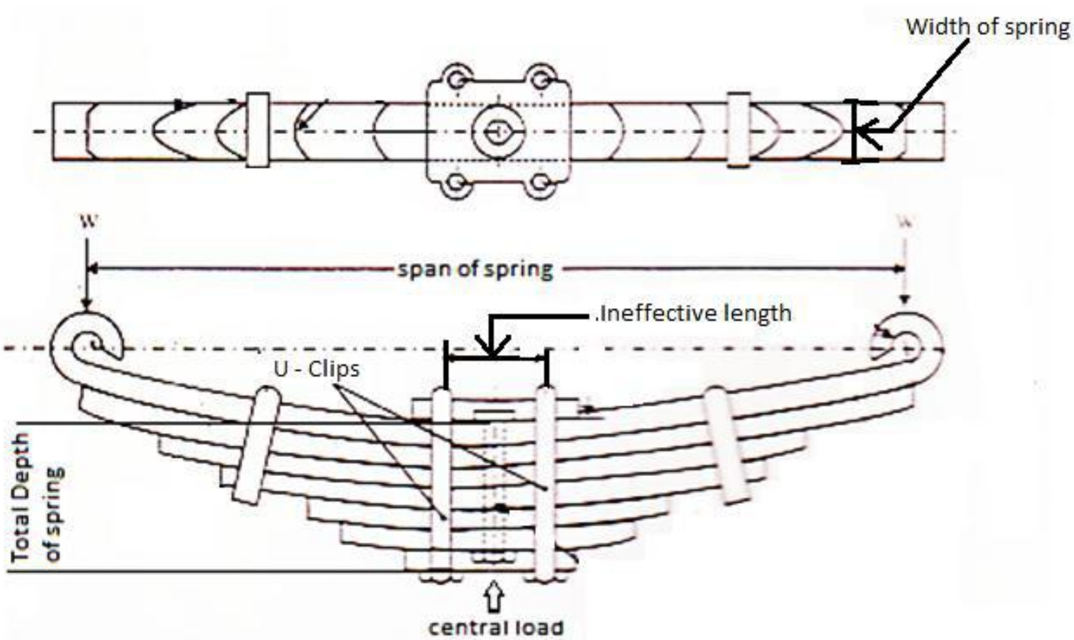
First gear ratio: $$\therefore G_1 = \frac{T_b}{T_a} \times \frac{T_d}{T_c}$$	½
Second gear ratio: $$\therefore G_2 = \frac{T_b}{T_a} \times \frac{T_f}{T_e}$$	½
Third gear ratio: $$\therefore G_3 = 1 : 1$$	½
Reverse gear ratio: $$\therefore G_r = \frac{T_b}{T_a} \times \frac{T_i}{T_g} \times \frac{T_r}{T_i}$$	½
f) 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'.  	02
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

3. Attempt any four:	16
a) Describe spindle, axle and types of shaft with their appropriate example.	04
Answer: Spindle, axle and types of shaft with their appropriate example.	
Spindle: A spindle is a short rotating shaft. Spindles are used in all machine tools the small drive shaft of lathe or spindle of the milling machine.	01
Axle: The term axle is used for a shaft that supports a rotating element like wheel, hoisting drum or rope sheave and which is fitted to the hosing by means of bearings. An axle is subjected to bending moments due to transverse loads like bearing reactions and does not transmit any useful torque. Occasionally, the axle also transmits torque. An axle may rotate with the wheel or simply supports a rotating wheel, e.g. automobile rear axle.	01
Types of shaft:	
Countershaft: It is a secondary shaft, which is driven by the mainshaft and from which the power is supplied to a machine component. Countershafts are used in multi-stage gear boxes.	01
Lineshaft: A Lineshaft consists of number of number of shafts which are connected in axial direction by means of couplings. They are used in group drive. In group drive construction, single electric motors drive the lineshaft. Number of pulleys is mounted on the lineshaft and power is transmitted to the individual machines by different belts.	01
b) Enlist types of failure and describe any three of them.	04
Answer: Types of failure: (Any four-1 mark) 1. Force-and /or temperature induced elastic deformation 2.Yielding 3. Brinnelling 4. Ductile rupture 1. Brittle fracture 6. Fatigue 7. Corrosion 8. Wear 9. Impact 10. Fretting 11. Creep 12. Buckling 13. Combined creep and fatigue 14. Radiation damage	01
(Any three- 1 mark each) 1. **Force-and /or temperature induced elastic deformation:** Failure occurs when the elastic deformation in a machine member, brought about by the imposed operational loads or temperatures, becomes great enough to interfere with the ability of the machine to satisfactorily perform its intended function. 2. **Yielding:** Failure occurs when the plastic deformation in a ductile material brought about by the imposed operational loads or motions, becomes great enough to interfere with the ability of the machine to satisfactorily perform its intended function. 3. **Brinnelling:** Failure occurs when the static forces between two curved surfaces in contact result in local yielding of one or both mating members to produce a permanent surface discontinuity of significant size. 4. **Ductile rupture**: Failure occurs when the plastic deformation in machine part that exhibits ductile behavior is carried out the extreme so that the member separates into two pieces. 5. **Brittle fracture:** Failure occurs when the elastic deformation, in a machine part that exhibits brittle behavior, is carried to the extreme so that the primary inter atomic bonds are broken and the member separates into two or more pieces.	03

6. **Fatigue:** Fatigue failure is a general term given to the sudden and catastrophic separation of a machine part into two or more pieces as a result of the application of fluctuating loads or deformations over a period of time. Failure takes place by the imitation and propagation of a crack until it becomes unstable and propagates suddenly to failure. The loads and deformations that are cause failure by fatigue are typically far below the static failure levels.
7. **Corrosion:** Failure implies that a machine part is rendered incapable of performing its intended function because of the undesirable deterioration of the material as a result of chemical or electrochemical interaction with the environment. Corrosion often interacts with other failure modes such as wear or fatigue.
8. **Wear:** It is undesired cumulative change in dimensions brought about by the gradual removal of discrete particles from contacting surfaces in motion, usually sliding, predominantly as a result of mechanical action.
9. **Impact:** Failures result when a machine member is subjected to monostatic loads that produce in the part stresses or deformations of such magnitude that the member no longer is capable of performing its function. The failure is brought about by the interaction of stress or strain waves generated by the dynamic or suddenly applied loads, which may induce local stresses and strains many times greater than would be induced by static application of the same loads.
10. **Fretting:** Fretting action may occur at the interface between any two solid bodies whenever they are pressed together by a normal force and subjected to small-amplitude cyclic relative motion with respect to each other. Fretting usually takes place in joints that are not intended to move but, because of vibrational loads or deformations, experience minute cyclic relative motions.
11. **Creep:** Failure results when the plastic deformation in a machine member accrues over a period of time under the influence of stress and temperature until the accumulated dimensional changes interfere with the ability of the machine part to satisfactorily perform its intended function.
12. **Buckling:** Failure occurs when, because of critical combination of magnitude and/or point of load application, together with the geometrical configuration of a machine member, the deflection of the member suddenly increases greatly with only a slight increase in load. This nonlinear response results in buckling failure.
13. **Combined creep and fatigue:** It is a combination failure mode in which all of the conditions for both creep failure and fatigue failure exist simultaneously, each process influencing the other to produce accelerated failure.
14. **Radiation damage:** Failure occurs when the change in material properties induced by exposure to a nuclear radiation field are of such a type and magnitude that the machine part is no longer able to perform its intended function.

c) Describe types of lever with their appropriate example.	04

Answer: (Description -3 mark & sketch -1 mark)
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 lever: 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	01
First type lever: In the first type of levers, the fulcrum is in between the load and effort. In this case, the effort arm is greater than load arm; therefore mechanical advantage obtained is more than one. **Examples:** Such type of levers are commonly found in bell cranked levers used in railway signaling arrangement, rocker arm in internal combustion engines, handle of a hand pump, hand wheel of a punching press, beam of a balance, foot lever etc. **Second type lever:** In the second type of levers, the load is in between the fulcrum and effort. In this case, the effort arm is more than load arm; therefore the mechanical advantage is more than one. **Examples:** It is found in levers of loaded safety valves. **Third type lever:** In the third type of levers, the effort is in between the fulcrum and load. Since the effort arm, in this case, is less than the load arm, therefore the mechanical advantage is less than one. **Examples:** The use of such type of levers is not recommended in engineering practice. However a pair of tongs, the treadle of a sewing machine etc. are examples of this type of lever.	03
d) Draw neat sketch of leaf spring and show span of spring, ineffective length, central load, width of spring, total depth of spring.	04
Answer: 	04

e) Define : 1) Indicated power 2) Brake power. 3) Frictional power and state relation between them	04
Answer: *(Each correct definition- 1 mark, Relation between power -1 mark)* **1) Indicated power:** The power developed inside the cylinder is known as indicated power. **2) Brake power:** This is the actual power delivered at the crankshaft. **3) Frictional power:** Power lost in frictional losses at the working surfaces like bearing, piston rings, valves etc. is known as frictional power. **Relation between Indicated power, Brake power and Frictional power** Frictional power = Indicated power - Brake power	01 01 01 01
f) State any four requirements of piston material.	04
Answer: **Requirements of piston material** *(Any four-1 mark each)* 1. High thermal conductivity 2. Low density for light weight construction and less inertia forces. 3. Good tensile strength 4. More Wear strength 5. Corrosion resistance	04
4. Attempt any two :	16
a) Design flange coupling to transmit 15kW at 900 rpm. The service factor may be used as 1.35. Following permissible stresses may be assumed. Shear stress for shaft, bolt and key material is 40MPa. Crushing stress for bolt and key material is 80 MPa and shear stress for cast iron is 8 MPa.	08
Answer: Given: $P = 15kW = 15\times10^3 W$, N= 900 r.p.m. Service factor $=1.35$; $\tau_s = \tau_k = 40N/mm^2$ $\sigma_{cb} = \sigma_{ck} = 80MPa = 80N/mm^2; \tau_c = 8MPa = 8N/mm^2$ The protective type flange coupling is designed: **1. Design for hub:** First of all let us find the diameter of the shaft (d). $$T = \frac{P\times60}{2\Pi N} = \frac{15\times10^3\times60}{2\Pi\times900} = 159.13N-m$$ Since, the service factor is 1.35 therefore the maximum torque transmitted by the shaft, $$T_{\max} = 1.35\times159.13 = 215N-m = 215\times10^3 N-mm$$ Torque transmitted by the shaft (t), $215\times10^3 = \frac{\Pi}{16}\times\tau_s\times d^3 = \frac{\Pi}{16}\times40\times d^3 = 7.86d^3$ $d^3 = 215\times10^3/7.86 = 27.4\times10^3$ or $d = 30.1$ saY 35 mm	02

Outer diameter of the hub,

$$D = 2d = 2\times 35 = 70mm$$

Length of hub, $L = 1.5d = 1.5\times 35 = 52mm$

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 $(T_{\max})$

$$2.15\times 10^3 = \frac{\Pi}{16}\times\tau_c\left[\frac{D^4-d^4}{D}\right] = \frac{\Pi}{16}\times\tau_c\left[\frac{(70)^4-(35)^4}{70}\right] = 63147\,\tau_c$$

$$\tau_c = 215\times 10^3/63147 = 3.4N/mm^2 = 3.4MPa$$

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 02

$\therefore\quad l = L = 52.5mm$

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 $(T_{\max})$.

$$2.15\times 10^3 = l\times w\times\tau_k\times\frac{d}{2} = 52.5\times 12\times\tau_k\times\frac{35}{2} = 11025\,\tau_k$$

$$\tau_k = 215\times 10^3/11025 = 19.5N/mm^2 = 19.5MPa$$

Considering the key in crushing. We know that the maximum torque transmitted $(T_{\max})$.

$$2.15\times 10^3 = l\times\frac{t}{2}\times\sigma_{ck}\times\frac{d}{2} = 52.5\times\frac{12}{2}\times\sigma_{ck}\times\frac{35}{2} = 5512.5\sigma_{ck}$$

$$\sigma_{ck} = 215\times 10^3/5512.5 = 39N/mm^2 = 39MPa$$

Science the induced shear and crushing stresses in the key are less than the permissible stresses therefore the **design for key is safe.** 02

3. **Design for flange**:

The thickness of flange (t_f) is taken as 0.5 d

$$t_f = 0.5d = 0.5\times 35 = 17.5mm$$

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 $(T_{\max})$

$$215\times 10^3 = \frac{\Pi D^2}{2}\times\tau_c\times t_f = \frac{\Pi(70)^2}{2}\times\tau_c\times 17.5 = 13473\tau_c$$

$$\therefore\quad \tau_c = 215\times 10^3/13473 = 1.6N/mm^2 = 1.6\,\text{MPa}$$

Since the induced shear stress in the flange is less than 8 MPa therefore the **design of safe.**

4. Design for bolts: Let d_1= nominal diameter of bolts. Since the diameter of the shaft is 35 mm therefore let us take number of bolts N=3 And pitch circle diameter of bolts $D_1 = 3d = 3\times 35 = 105mm$ The bolts are subjected to shear stress due to the torque transmitted. We know that maximum torque transmitted $(T_{\max})$, $215\times 10^3 = \frac{\Pi}{4}\times (d_1)^2 \tau_b \times n \times \frac{D_1}{2} = \frac{\Pi}{4}(d_1)^2 40\times 3\times \frac{105}{2} = 4950(d_1)^2$ $\therefore \quad (d_1)^2 = 215\times 10^3 / 4950 = 43.43 \text{ or } d_1 = 6.6mm$ 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. $D_2 = 4d = 4\times 35 = 140mm$ Thickness of the protective circumferential flange. $t_p = 0.25d = 0.25\times 35 = 8.75\, say\, 10 \text{ mm}$	02
b) A multiplate clutch is to transmit 4.5 kW at 750 rpm. The inner and outer radii of contact surfaces are 40 and 70 mm respectively. The co-efficient of friction is 0.1. The average intensity of pressure is 0.35 N/nm^2. Find total no. of clutch plates, actual axial force required, actual average pressure and actual maximum pressure.	08
Answer: Given Data: Power transmitted by clutch P = 4.5 KW Speed of clutch N = 750 rpm Inner radius r_2 = 40 mm Outer radius r_1 = 70 mm Coefficient of friction μ = 0.1 Average intensity of pressure P_{avg} = 0.35 N/mm^2 Power transmitted by the clutch, $P = \frac{2\Pi NT}{60}$ $4.5\times 10^3 = \frac{2\Pi \times 750}{60}\times T$ T = 57.29 Nm = 57.29 X 10^3N mm For uniform wear, mean radius of friction surface,	01

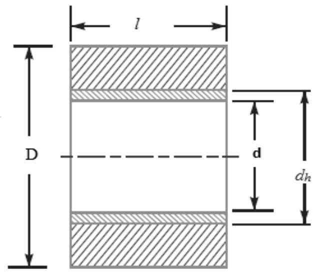
$r = \frac{r_1 + r_2}{2} = \frac{70+40}{2} = 55mm$ Axial force, $W = P_{avg} \times \Pi \times (r_1^2 - r_2^2)$ $W = 0.35 \times \Pi \times (70^2 - 40^2)$ W= 3628.539 N **1. Total number of clutch plates:** Torque transmitted by the clutch, $T = n\,\mu\,W\,r$ $57.29 \times 10^3 = n \times 0.1 \times 3628.539 \times 55$ n = 2.83 $n \approx 3$ Since the number of pairs of contact surfaces must be even, therefore we use four pairs of contact surfaces. **n = 4**	01
2. Actual axial force required: W_1 = Actual axial force required Torque, $T = n\,\mu\,W\,r$ $57.29 \times 10^3 = 4 \times 0.1 \times W_1 \times 55$ **W_1 = 2604.09 N**	02
3. Actual average pressure, $P_{avg} = \frac{W_1}{\Pi\,(r_1^2 - r_2^2)} = \frac{2604.09}{\Pi(70^2 - 40^2)}$ **P_{avg} = 0.251 N/mm^2**	02
4. Actual maximum pressure, $P_{max} = \frac{C}{r_2}$ $C = \frac{W_1}{\Pi(r_1 - r_2)} = \frac{2604.09}{\Pi(70-40)} = 27.63 \quad N/mm$ $P_{max} = \frac{27.63}{40} = 0.690$ N/mm^2	02
c) A four stroke diesel engine has following specifications. 1) B.P.- 5kW at 1200 rpm 2) Indicated mean effective pressure 0.35 N/mm^2 3) Mechanical efficiency 80% Determine :	08

1) Bore and length of cylinder 2) Thickness of cylinder head 3) Size of studs for cylinder head if allowable tensile strength for stud materials is 65MPa.	
Answer: ***(Note: Assume l = 1.5 D OR l = 1.08 D)*** **Given:** B.P. = 5kW= 5000 W N= 1200 r.p.m. or n= N/12 = 600 P_m=0.35N/mm ^{2} $\eta_m = 80\% = 0.8$ **1. Bore and length of cylinder** Let D= Bore of the cylinder in mm A= across section area of the cylinder $= \frac{\Pi}{4} \times D^2 mm^2$ l= length of the stroke in m. = 1.5 D mm = 1.5 D/ 1000 m	01
We know that the indicated power $I.P. = B.P./\eta_m = 5000/0.8 = 6250w$	01
We also know that the indicated power (I.P.) $6250 = \frac{P_{m.}l.A.n}{60} = \frac{0.35 \times 1.5D \times \Pi D^2 \times 600}{60 \times 1000 \times 4} = 4.12 \times 10^{-3} D^3$ $\therefore \quad D^3 = 6250/4.12 \times 10^{-3} = 1517 \times 10^{-3}$ or $D = 115$ mm $l = 1.5D = 1.5 \times 115 = 172.5$mm	01
Taking a clearance on both sides of the cylinder equal to 15 % of the stroke therefore length of the cylinder. $L = 1.15l = 1.15 \times 172.5 = 198$ say 200 mm	01
2. Thickness of the cylinder head Since the maximum pressure (P) in the engine cylinder is taken as 9 to 10 times means effective pressure (P_m) therefore let us take $P = 9P_m = 9 \times 0.35 = 3.15 N/mm^2$ We know that thickness of the cylinder head $t_h = D\sqrt{\frac{C.p}{\sigma_t}} = 115\sqrt{\frac{0.1 \times 3.15}{42}} = 9.96$ say 10 mm (Taking C= 0.1 and σ_t = 42 MPa = 42 N/mm^2)	02
3. Size of studs for the cylinder head Let d= Nominal diameter of the stud in mm d_c = core diameter of the stud in mm. it is usually taken 0.84d. σ_t = Tensile stress for the material of the stud which is usually nickel steel n_s = Number of stud We know that the force acting on the cylinder head (or on the studs)	

$=\frac{\Pi}{4}D^2\times p=\frac{\Pi}{4}(115)^2 3.15=32702\,N$ …….(i) The number of studs n_s are usually taken between $0.01\,D+4$ (i.e. $0.01\times115+4=5.15$) and $0.02\,D+4$ (i.e. $0.02\times115+4=6.3$). Let us take $n_s=6$ $=n_s\times\frac{\Pi}{4}(d_c)^2\,\sigma_t=6\times\frac{\Pi}{4}(0.84d)^2 65=216d^2 N$ …….(ii) Form equations (i) and (ii) $d^2=32702/216=151$ or $d=12.3$ say 14 mm The pitch circle diameter of the stud (D_p) is taken D+ 3d. $\therefore\quad D_p=115+3\times14=157\,mm$ We know that pitch of the studs $=\frac{\Pi\times D_p}{n_s}=\frac{\Pi\times157}{6}=82.2\,mm$ We know that for a leak proof joint the pitch of the studs should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$ Where d is the nominal diameter of the stud. Minimum pitch of the studs $==19\sqrt{d}=19\sqrt{14}=71.1$ mm And maximum pitch of the studs $=28.5\sqrt{d}=28.5\sqrt{14}=106.6$ mm Since the pitch of the studes obtained above (i.e. 82.2 mm) lies within 71.1 mm and 106. 6 mm therefore size of the stud (d) calculated above is satisfactory $\therefore\quad d=14$ mm	02
5. Attempt any two:	16
a) Design piston pin for following data piston diameter 70mm, max. gas pressure inside cylinder 4.5N/mm^2. Allowable stresses in bending, shear and bearing are 100 MPa, 70 MPa and 25 MPa respectively.	08
Answer: Given data, Dia. of piston = D = 70 mm. Max. Gas pressure = P_{max} = 4.5 N/mm^2 Bending stress = σ_b = 100N/mm^2 Shearing stress = τ = 70 N/mm^2 Bearing pressure P_b = 25 N/mm^2 Let, R = Normal side thrust acting on piston skirts **Maximum gas load** $F=P_{max}\times\frac{\pi}{4}D^2$ $F=4.5\times\frac{\pi}{4}(70)^2=17.318\times10^3\,N$	01

A)Design of piston pin on the basis of bearing pressure: Maximum gas pressure = bearing load $F = d_{po} \times l_p \times p_b$ d_{po} = outer diameter of piston pin l_p = length of piston pin small end of connecting rod $= 0.45 \times \mathrm{D} = 31.5\ \mathrm{mm}$ $F = d_{po} \times l_p \times p_b$ $17.318 \times 10^3 = d_{po} \times 31.5 \times 25$ $d_{po} = 21.99 = 22mm$	02
B)Design of piston pin on the basis of bending: $\mathrm{M} = \frac{\mathrm{F} \times \mathrm{D}}{8} = \frac{17.318 \times 10^3 \times 70}{8}$ $= 151.53 \times 10^3\ \mathrm{N-mm}$ $\mathrm{M} = \frac{\Pi}{32} \times \sigma_b \times {d_{po}}^3$ $151.53 \times 10^3 = \frac{\Pi}{32} \times \sigma_b \times 22^3$ $\sigma_b = 144.9\ \mathrm{N/mm}^2$ The induced bending stress are **greater than** permissible stress 100N/mm^2 **hence redesign is** necessary. $\mathrm{M} = \frac{\Pi}{32} \times \sigma_b \times {d_{po}}^3$ $151.53 \times 10^3 = \frac{\Pi}{32} \times 100 \times {d_{po}}^3$ $d_{po} = 24.89 = 25mm$	02
C)Design of piston pin on the basis of shear stress, due to double shear: $\mathrm{F} = 2\frac{\Pi}{4} \times {\mathrm{d}_{po}}^2 \times \tau$ $151.53 \times 10^3 = 2\frac{\Pi}{4} \times 25^2 \times \tau$ $\tau = 17.63\mathrm{N/mm}^2$ The induced shear stress is **less than** permissible stress 70N/mm^2. **Hence design is safe.**	02
D)Total length of piston Pin: $L_{p_T} = 0.9D = 0.9 \times 70 = 63mm$	01

b) I) Enlist applications of cotter joint and knuckle joint. II) State the design procedure of rocker arm.	08
Answer: **I) Applications of -** **Cotter joint:** ***(Any Two)*** i. Joint between a piston rod to cross head of steam engine ii. Joint between the slide spindle and the fork of the mechanism iii. Joint between the piston rod and the tail or pump rod **iv.** Foundation bolt **Knuckle joint:*(any two)*** i. It is used in link of cycle chain, ii. It is used in tie rod joints for roof truss, iii. It is used in valve rod joint for electric rod, iv. It is used in pump rod joint, v. It is used in tension link in bridge structure, vi. It is used in lever and rod connection of various types.	02 02
b. II) State the design procedure of rocker arm.	4
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

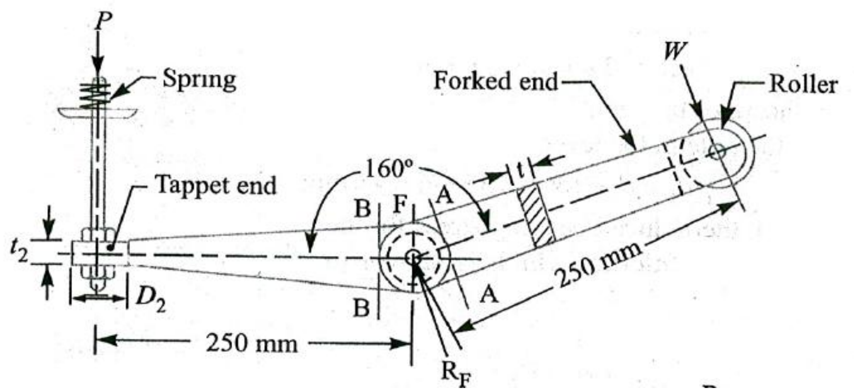


Bending moment at this section = $W \times L$

Section Modulus $Z = 1/12 \times l \times (D^3 - d_h^3) / D/2$
Induced bending stress,

$$\sigma_b = \frac{M}{Z}$$ 01

OR



04

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 centre 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.

c) Design bushed pins only for a flexible coupling to transmit 18 kW at 900 rpm. Diameter of shaft for coupling is 60 mm. Allowing shear and bending stresses in pin are 25 N/mm^2 and 50 N/mm^2 respectively. The allowable bearing pressure in rubber bush in 0.3 N/mm^2.	08
Answer: Given $P = 18kw = 18\times10^3$ watts N= 900 r.p.m. D=60mm $d_1 = 55$ mm $f_{sp} = 25N/mm^2$ $f_{sk} = 40N/mm^2$ $P_b = 0.3\ N/mm^2$ $f_{bp} = 50N/mm^2$ We know that torque transmitted, $T = \frac{P\times60}{2\prod N} = \frac{18\times10^3\times60}{2\times3.14\times900}$ $= 191N-m$ $T = 191\times10^3 N-mm$	01
$n =$ no. of pins $d_1 =$ diameter of pin at neck $d_3 =$ diameter of pin in the bush $t_1 =$ thickness of brass bush $t_2 =$ thickness of rubber bush $D_1 =$ diameter of pitch circle of pins $t_3 =$ thickness of pin head $d_4 =$ diameter of pin head We know that $n = \frac{d}{25} + 3$ $\therefore \quad n = \frac{60}{25} + 3$ $\therefore \quad n = 5.4$ Taking next higher even number $\therefore \quad n = 6$	01
Now diameter of pin, $d_1 = \frac{0.5d}{\sqrt{n}} = \frac{0.5\times60}{\sqrt{6}}$ $d_1 = 12.24$ mm $d_1 = 13$ mm	01

Now $d_3 = 1.5d_1$ $= 1.5 \times 13$ $= 19.5\,\text{mm}$	
A brass sleeve of thickness' t_1 ' and a rubber bush of thickness ' t_2 ' is fitted on this pin diameter d_3 $t_1 = 2\,\text{mm}$ And $t_2 = 6\,\text{mm}$ Now outer diameter of rubber bush, $d_2 = d_3 + 2 \times t_1 + 2 \times t_2$ $= 19.5 + 2 \times 2 + 2 \times 6$ $d_2 = 35.5\,\text{mm}$	01
Now pitch circle diameter of pins $D_1 = 3d$ $= 3 \times 60$ $D_1 = 180$ mm Let us assume thickness of pin head. $t_3 = 3$ mm Now diameter of pin head $d_4 = d_2 - t_3$ $= 35.5 - 3$ $d_4 = 32.5$ mm	01
Let W= load on each pin L= Length of bush in left hand flange. Now torque $T = W.n.\left(\frac{D_1}{2}\right)$ $\therefore \quad W = \frac{T \times 2}{D_1 \times n}$ $\therefore \quad W = \frac{191 \times 10^3 \times 2}{180.0 \times 6}$ $\therefore \quad W = 353.7N$ Bearing pressure on rubber bush, $P_b = 0.3\,\text{N/mm}^2$ given $\therefore \quad W = P_b.d_2 \times l 7N$ $353.7 = 0.3 \times 35.5 \times l$ $l = 33.21\,\text{mm}$ $l = 34\,\text{mm}$ Clearance between flanges, C = 0.1 d $= 0.1 \times 60$ $= 6mm$ Stresses in pin $f_{sp} = \text{direct shear stress in pin}$	

$= \dfrac{W}{\frac{\Pi}{4} d_1^2}$	
$= \dfrac{353.7}{\frac{\Pi}{4}(13)^2}$	
$= 2.66 N/mm^2$	
Bending moment in pin $M = w\left[\dfrac{l}{2} + C\right]$	01
$\therefore \quad M = 353.7\left[\dfrac{34}{2} + 6\right]$	
$\therefore \quad M = 353.7[23]$	
$\therefore \quad M = 8.135 \times 10^3 N - mm$	
$\therefore$ Bending stress, $F = \dfrac{m}{z} = \dfrac{m}{\frac{\Pi}{32}(d_1)^3}$	
$\therefore \quad F = \dfrac{8.135 \times 10^3}{\frac{\Pi}{32}(13)^3}$	
$\therefore \quad F = 37.73 N/mm^2$	
Now maximum principal stress induced in pin (Maximum bending stress)	
$f = \dfrac{1}{2}\left[F\sqrt{(F)^2 + 4(f_{sp})^2}\right]$	
$= \dfrac{1}{2}\left[37.73 + \sqrt{(37.73)^2 + 4(2.66)^2}\right]$	
$= \dfrac{1}{2}[37.73 + 38.103]$	01
$= 37.91 N/mm^2$	
This value is less than allowable bending stress in pin (50 N/mm^2), hence design is safe.	
Now maximum shear stress induced in pin.	
$f_{s\max} = \dfrac{1}{2}\sqrt{(F)^2 + 4(f_s)^2}$	
$= \dfrac{1}{2} \times \sqrt{(37.73)^2 + 4(2.66)^2}$	
$= \dfrac{1}{2} \times 38.10$	
$\therefore \quad f_{s\max} = 19.05 N/mm^2$	01
This value is less than allowable shear stress in pin (20 N/mm^2), hence the design is safe.	

6. Attempt any two:	16
a) Design connecting rod cross-section with following data 1. Max. pressure inside cylinder 6.5 N/mm^2 2. Piston diameter = 100 mm. 3. Stroke length =110mm 4. Effective length of connecting rod =220 mm 5. Max. allowable stress in crippling = 120 MPa 6. Rankin's constant 1/6000 Also calculate height of cross- section at both the ends.	08

Answer:

Given Data- P_{max} =6.5 N/mm^2 , D =100 mm, l=110 mm,L= 220 mm, σ_c= 120 N/ mm^2,

A = 11 t^2 where t = thickness of rod

a = Rankine contant = 1/ 6000

$K_{xx} = \sqrt{3.18t^2}$

W = maximum gas load= P_{max} x $\frac{\pi}{4}D^2 = 6.5\times\frac{\pi}{4}100^2$.

W= 51.05 x 10^3 N 01

Assuming I-section

$$W = \frac{\sigma_c \times A}{1+a[\frac{L^2}{k^2_{xx}}]}$$

$$51.05 \times 10^3 \text{ N} = \frac{120\times 11t^2}{1+1/6000[\frac{220^2}{3.18t^2}]}$$

$$\therefore t^4 - 38.674t^2 - 98.07 = 0$$

By using quadratic equation,

t^2= 41.19

t = 6.418 mm say **t = 6.5 mm** 04

Other dimensions of I-section at middle, small end and big end-

a) Dimension at the middle or center

(i) depth or height of section

H = 5 t =5 x 6.5

H = 32.5 mm

(ii) width of cross section B 01

B = 4 t = 4 x 6.5 =26 mm

b) Dimension at small end (i) depth or height of section H_1=0.82H=0.82x32.5=26.65mm (ii) width of cross section B B= B_1=26mm **c)Dimension at big end** **(i) depth or height of section** H_2=1.182H=1.182x32.5=38.35mm **(ii) width of cross section B** B= B_2=26mm	01 01
b) State the design procedure for piston rings and skirt length.	8

Answer: Design of Piston Rings:

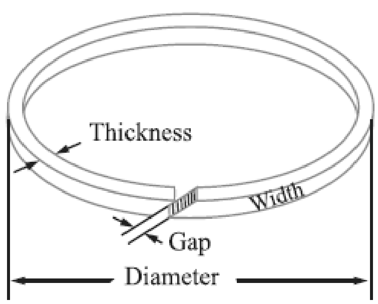


Fig. Piston rings. 01

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

$$t_1 = D\sqrt{\frac{3p_w}{\sigma_t}}$$ 01

Where, D = Cylinder bore in mm,

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

σ_t = Allowable bending (tensile) stress in MPa.

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

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

$$t_2 = \frac{D}{10\,n_R}$$ 01

Where, n_R = Number of rings.

Width of top land,

$b_1 = t_H$ to 1.2 t_H 01

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. **Design of Skirt Length:** R = Normal side thrust acting on piston skirts **Maximum gas load** $F = P_{max} \times \frac{\pi}{4} D^2$ R = Normal side thrust acting on piston skirts $\because$ Side thrust $= 10\%$ $\therefore \quad R = 0.1 F$ Let, $l_1 =$ length of piston skirt 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 3
c) Describe service factor, overload factor, velocity factor and factor of safety.	8
Answer: **Service factor (SF):** Service factor is described the service limit of the component for definite period of cycle. Overload capacity considered while designing component, device, engine, motor, etc., as a safety factor. It is expressed usually a number greater than one: a SF of 1.15 means the item can take 15 percent more load than its rated capacity without breakdown. This means that a 10-hp motor with a 1.15 SF could provide 11.5 hp when required for short-term use. Some fractional horsepower motors have higher service factors, such as 1.25, 1.35, and even 1.50. In general, it's not a good practice to size motors to operate continuously above rated load in the service factor area. **Overload factor:** The overload factor makes allowance for the externally applied loads which are in excess of the nominal tangential load. In determining the overload factor, consideration should be given to the fact that many prime movers and driven equipment, individually or in combination, develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment. There are many possible sources of overload which should be considered. Some of these are: system vibrations, acceleration torques, over speeds, variations in system operation, split path load sharing among multiple prime movers, and changes in process load conditions.	02 02

Velocity Factor: The velocity factor accounts for - 1. The severity of impact as successive pairs of teeth comes into engagement. 2. Factors such as pitch line velocity, manufacturing and assembly accuracies. 3. Polar mass moments of inertia of pinion and gear mesh shaft and bearing stiffness. 4. The velocity factor is used to take care of possibilities of fatigue failure.	02
Factor of Safety: Factor of safety is defined as the ratio of the maximum stress (yield point stress for ductile material) to the working stress or design stress. In case of ductile materials- $\text{Factor of safety} = \dfrac{\text{Yield point stress}}{\text{Working or design stress}}$	02