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.

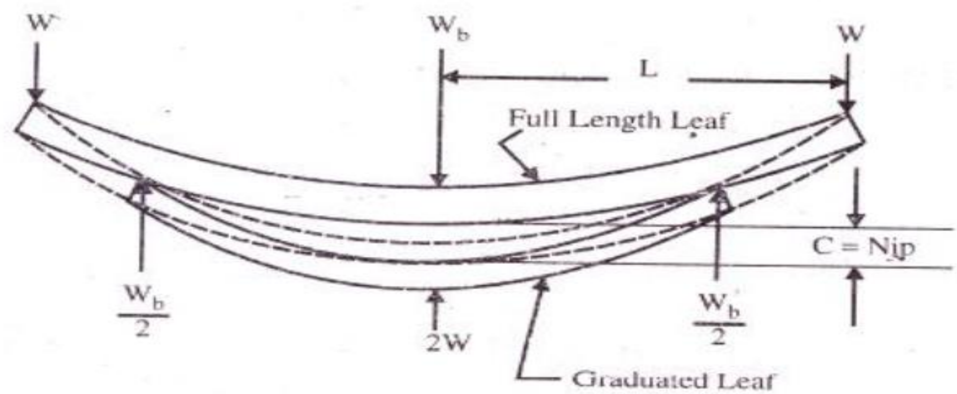
Q. No.	Sub Q. N.	Answer	Marking Scheme
1		**Attempt any FIVE of the following:**	**10**
	A	**State different modes of failure of automobile components.**	**02**
		Answer: 1. Failure of automobile components in Tension 2. Failure of automobile components in Shear 3. Failure of automobile components in Bending 4. Failure of automobile components in compression 5. Failure of automobile components (Gears) pitting 6. Failure of automobile components(Gears) Scoring 7. Failure of automobile components due to abrasive wear 8. Failure of automobile components due to corrosive wear 9. Failure of automobile components due to fatigue 10. Failure of automobile components due to crushing	**02 marks For Any four**
	B	**Define the terms.** **(i)Factor of safety (ii)working stress**	**02**
		Answer: **(i) Factor of Safety:** Factor of safety is defined as the ratio of the maximum stress to the working stress or design stress. Mathematically, $\text{Factor of Safety} = \dfrac{\text{Maximum Stress}}{\text{Working or design stress}}$ **In Case of Ductile Material,** $\text{Factor of Safety} = \dfrac{\text{Yield Point Stress}}{\text{Working or Design Stress}}$	**01**

		In case of Brittle Material , $\text{Factor of Safety} = \dfrac{\text{Ultimate Stress}}{\text{Working or Design Stress}}$ **(ii) Working Stress:** It is defined as the ratio of actual axial load and the original cross section of the specimen $\text{Working stress} = \dfrac{\text{Actual axial load}}{\text{Original cross section area}}$	01
	C	**State and justify material for leaf spring.**	02
	Ans	**Answer:** **Leaf spring:-** **Material:-** Plain carbon steel having 0.9 to 1.0% carbon. **Justification: -** The leaves are heat treated after forming processes. The heat treatment of spring steel produces greater strength and therefore greater load capacity, greater range of deflection and better fatigue properties.	01 01
	D	**List the functions of cylinder block**	02
		Answer: **The cylinder block has the following functions:** 1. The main functions of the cylinder block are maintaining the engine's stability and lubrication while withstanding a variety of temperatures and loads. 2. Transferring oil to all parts of the engine, lubricating all the critical components, via a number of oil galleries 3. Organizing and determining the position of various parts, 4. Ensuring the standard accuracy of the main components, 5. Accommodating cooling water, adjusting the temperature difference, 6. Placing lubricating oil channels and distributing lubricating oil, the sealing of cylinder block and cylinder head will make the pressure reach the ideal effect, and the power generated by airframe can be transmitted through the follower part.	02 (Any Two)
	E	**State and justify material for push rod.**	02
		Answer: **Material:-** 1018 steel tubing **Push rods** are typically made of 1018 steel tubing, which is not strong enough for springs that have more than 400 psi of open pressure, or **engines** that exceed 7,000 rpm. **Pushrods** made of 4130 or 4140 **Chromoly** which is a chrome-alloy steel with a medium carbon content and . 8% - 1.1% molybdenum for strength. **Pushrods** made of 4130 or 4140 **Chromoly** tubing are much stronger	02
	F	**Describe aesthetics in automobile component design .**	02
		Answer: **Aesthetics in automobile Design -** The appearance should contribute to the performance of the product, thought the extent of	

		contribution varies from product to product for example chromium plating of automobiles components improves the corrosion resistance along with the appearance. Similarly the aerodynamic shape of the car improves the performance the performance as well as gives the pleasing appearance lesser air resistance resulting in the lesser fuel consumption. The appearance should reflect the function of the product for example. The aerodynamic shape of the car increases the speed.	**02**
	G	**Sketch the method to reduced stress concentration in cylindrical members with holes.**	**02**
		Answer: **The methods of reducing stress concentration in cylindrical members with shoulders hole, subjected to tensile load.**  **Figure. Methods of reducing stress concentration in cylindrical members with holes.**	**02**
2		**Attempt any THREE of the following:**	**12**
	A	**List basic design requirement of connecting rod.**	**04**
		Answer: In designing a connecting rod for I.C. engine, the following points should be taken into consideration : • Strength to withstand the high inertia forces. • Minimum mass to minimise the inertia forces. • Provide sufficient bearing area to prevent undue wear. • High speed reciprocation without noise. • Rigid construction to withstand mechanical distortion.	**04 (Any Four)**
	B	**State any two uses of each of the following.** **(i) Stress-strain diagram** **(ii) S-N curve**	**04**
		Answer: **(i) Stress-Strain diagram:(Any Two)** 1. Stress-strain diagram is used to find out strength, elasticity, stiffness, ductility of material. 2. From stress-strain diagram, elongation of material is found out. 3. It is used to find out yield points for material in tension/compression test. 4. It is used to find out toughness. **(ii) S-N curve: (Any Two)** 1. It is used to study the effect of fatigue of material. 2. It is used to find the Effect of Loading on Endurance Limit—Load Factor. 3. It is used to find out Effect of Surface Finish on Endurance Limit-Surface Finish Factor.	**02** **02**

	C	**Why nipping is provided in leaf spring?**	**04**

Answer: (Sketch – 02 marks, Description – 02 marks) **02**



02

When the central bolt holding the leaves is tightened, the full length leaf bend back as shown by dotted line. And will have an initial stress in opposite direction. The graduated leaves will have an initial stress in the same direction as that of normal load. When the load is applied, the full length leaf gets relieved first, consequently the full length leaf will be stressed less than graduated leaf. The initial leaf between leaves may be so adjusted that under maximum load conditions, all the leaves are equally stressed. So for this reason nipping is provided in leaf spring.

	D	**Describe the design procedure for rocker arm.**	**04**

Ans

Answer:

Step I: Calculate reaction at the fulcrum pin

$$R_F = \sqrt{W^2 + P^2 - 2W \times P \times \cos\theta}$$ **01**

Step II: Design of fulcrum pin:

(a) Let d = Diameter of the fulcrum pin, and

l = Length of the fulcrum pin

= 1.25 d **01**

Considering the bearing of the fulcrum pin. We know that load on the fulcrum pin (R_F),

$$\therefore \text{ Bearing pressure } = \frac{\text{Load}}{\text{Bearing area}} = \frac{R_F}{l \times d} = \frac{R_F}{1.25d \times d}$$

From here, l and d can be determined.

(b) Checking shear stress induced in the fulcrum pin. As the pin is in double shear,

$$\tau = \frac{R_F}{2 \times \left(\frac{\pi}{4} \cdot d^2\right)}$$ **01**

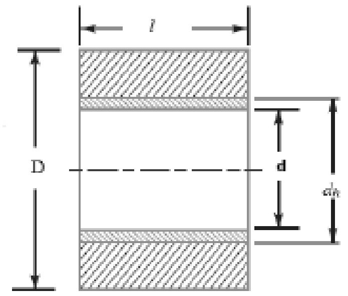
External diameter of the boss,

$D = 2\,d$

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



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

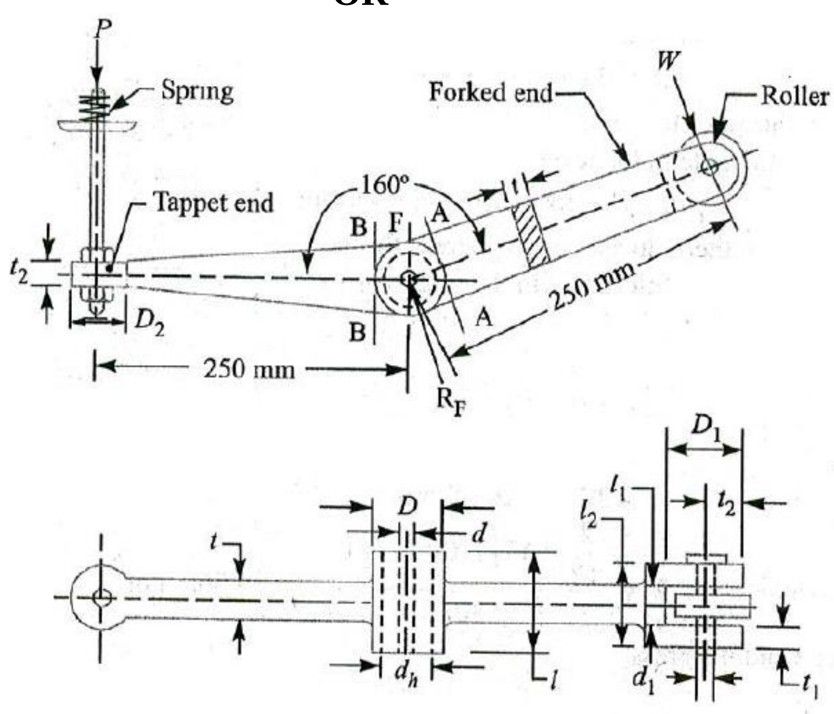


Figure. Rocker arm for operating exhaust valve

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.

3		**Attempt any THREE of the following:**	**12**
	A	**Describe the design procedure for front axle.**	**04**
	Ans	**Answer:**	**01**

Front axles are subjected to both bending and shear stresses.
Thus the maximum bending moment = W l, Nm
where, W = The load on one wheel, N
l = The distance between the centre of wheel and the spring pad, m

We know that, the portions projected after the spring pads are subjected to combined bending and torsion.

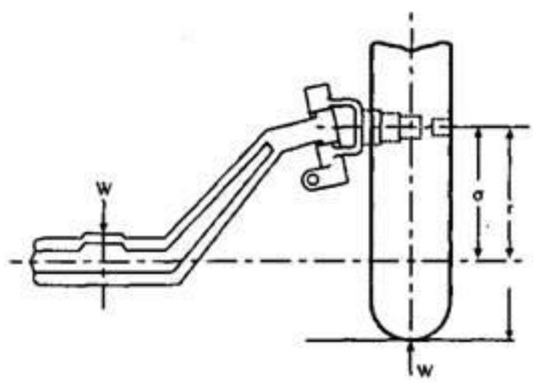


Fig. 27.2. Loads on front axle.

The magnitude of the torque = $R\,\delta$, Nm.

where, R = the resistance to motion, N

δ = The drop from the spindle axis to the centre of the section, m

$= \mu W r$, Nm.

r = The road wheel radius, m

μ = coefficient of adhesion between road and tyre

= 0.6 for dry,hard road surface (the maximum value of μ).

The braking torque is lower for the section lying between the spring pads and is given by $\mu W (r - \delta)$. In this portion the bending moment predominates whereas at the steering head, torsion predominates. Thus I-section is used for the portion where bending moment predominates and is gradually changed to circular, oval or rectangular section at the steering head. **01**

For I-section, the maximum bending moment is given by the relation,

$$\frac{M}{I} = \frac{f_b}{y}$$

where, M = the maximum bending moment, Nm

f_b = allowable bending stress for the material, N/m^2 or Pa

y = the maximum distance of the fibre from neutral axis (NA)

$= d/2$, m

I = The moment of inertia of the I–section about NA

$= \frac{bd^3 - ch^3}{12}$, m^4 for I-section shown in Fig. 27.3

where, d = the overall depth of I–section, m

b = the flange width, m

t = the flange thickness, m

w = the web thickness, m

$c = b - t$

$h = d - 2w$

Generally, $d = 6t$

$b = 4.25t$

$w = 2.5t$

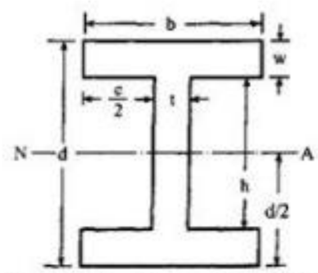


Fig. 27.3. I-section on the axle beam. **01**

For the circular or oval section, the maximum torsion is given by the relation,

$$\frac{T}{I_p} = \frac{f_s}{y}$$

where, T = the maximum torque in the plane of section, Nm

f_s = allowable shear stress in the material, N/m^2 or Pa

y = the distance from the neutral axis to the outermost fibre of the axle.

$= d/2$, m **01**

d = the diameter for the circular section

= the major axis for the oval section

I_p = Polar moment of inertia of the section

$= \frac{\pi}{32} d^4$ for circular section

$= \frac{\pi}{32} d^3 b$ for oval section with minor axis, b.

	B	**Explain design procedure for fully 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,

$$\frac{T_{RA}}{J_{RA}} = \frac{\sigma_s}{r}$$ **01**

Where, T_{RA} = Torque transmitted by rear axle shaft.

T_{RA} = Te x G1 x Gd **01**

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) **01**
$= \frac{\pi}{32}(d_o^4 - d_i^4)$ ....(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 **01**

$$k = \frac{d_i}{d_o}$$

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.

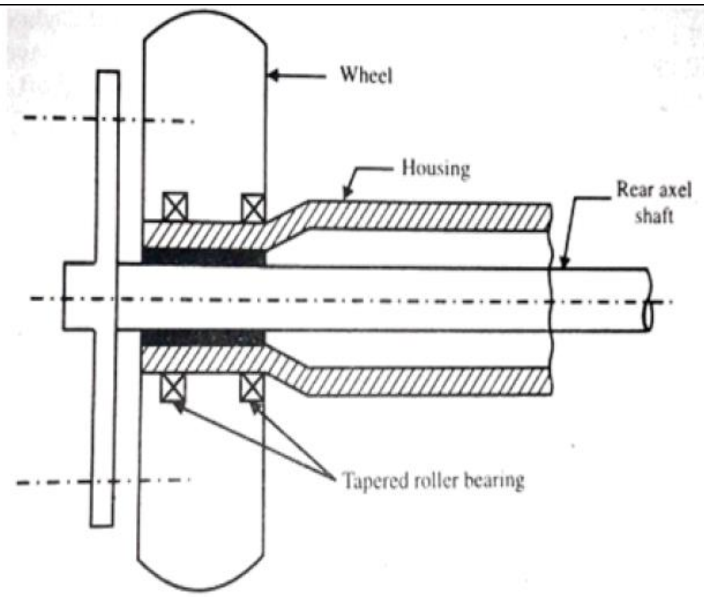


Figure. Fully floating rear axle

	C	**Describe design procedure for valve spring.**	**04**

Answer:

Helical compression spring is used as valve spring.

Let D = Mean diameter of the spring coil,

d = Diameter of the spring wire,

n = Number of active coils,

G = Modulus of rigidity for the spring material,

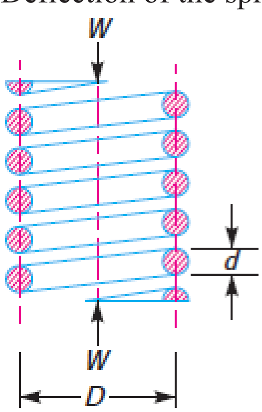
W = Axial load on the spring,

τ = Maximum shear stress induced in the wire,

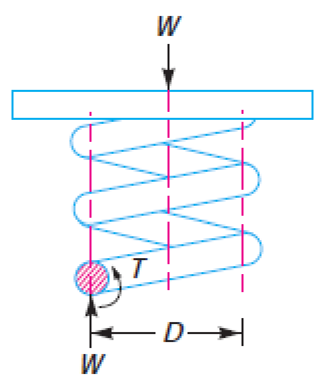
C = Spring index = D/d,

p = Pitch of the coils, and

δ = Deflection of the spring, as a result of an axial load W.



(a) Axially loaded helical spring.



(b) Free body diagram showing that wire is subjected to torsional shear and a direct shear.

1. **Mean diameter of spring coil**
   D = Inside diameter of spring + Diameter of spring wire **01**

Since the diameter of spring wire is obtained from maximum spring load , therefore maximum Twisting moment on the springs.

$T = W \times \frac{D}{2}$

And we also know that

Maximum twisting moment

$T = \frac{\pi}{16} \times \tau \times d^3$ ……………………………………………………(1)

From these equations Mean diameter and Diameter wire is calculated.

Now considering Wahl's stress factor (k),

We know that spring index

$C = \frac{D}{d}$

And Wahl's stress factor

$k = \frac{4C-1}{4C-4}$

We know that maximum shear stress (τ),

01

$\tau = K \times \frac{8WC}{\pi d^2}$ ……………………………………………………(2)

From this equation diameter of wire is calculated and Larger values of diameter of wire (d) is selected.

2. **Number of turns of the coil**
   We know that deflection of the spring(δ),

01

$\delta = \frac{8WD^3.n}{Gd^4}$

from this equation number of turns are calculated.

Taking the ends of the springs as squared and ground, the total number of turns,

n' = n +2

3. **Free length of spring**

$L_F = n' \times d + \delta_{max} + 0.15\,\delta_{max}$ **01**

4. Pitch of the coil

We know that pitch of the coil

$$= \frac{Free\ length}{n' - 1}$$

D **Explain the terms of preferred number and standardization.** **04**

Answer:

i) Preferred number: (02 marks)

Preferred numbers (also called preferred values) are standard guidelines for choosing exact product dimensions within a given set of constraints. The system is based on the use of **geometric progressive** to develop a set of numbers. Preferred numbers are used to specify the specification because a company may manufacture different models of same product. There are **four basic series**, denoted by **R5, R10, R20, R40** which increases in steps of 58%, 26%, 12%, 6% respectively. **02**

Each series has its own series factor given below,

Series	Series Factor
R5	$10^{1/5} = 1.58$
R10	$10^{1/10} = 1.26$
R20	$10^{1/20} = 1.12$
R40	$10^{1/40} = 1.06$

Standardization: (01 mark)

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. **01**

Advantages of Standardization:- (Any two- 01 mark)

1. Interchangeability of product or element is possible.
2. Mass production is easy.
3. Rate of production increases.
4. Reduction in labour cost.
5. Limits the variety of size and shape of product. **01**
6. Overall reduction in cost of production.
7. Improves overall performance, quality and efficiency of product.
8. Better utilization of labour, machine and time

E **Explain the maximum principal stress theory.** **04**

Answer:

Statement: According to this theory, the failure occurs at a point in a member when the maximum normal stress in a bi-axial stress system reaches the limiting strength of the material in

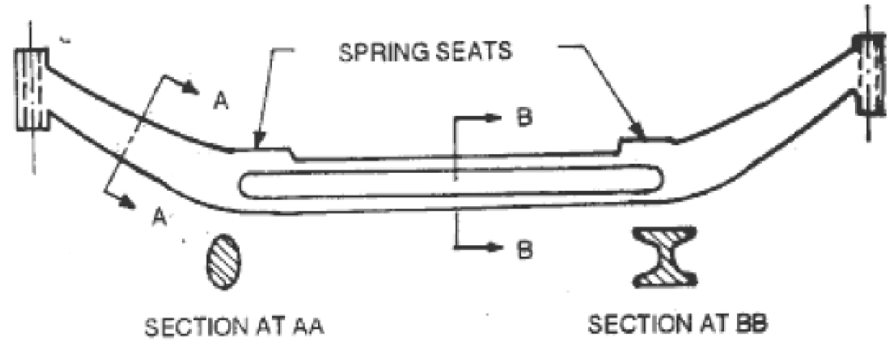
		a simple tension test. The maximum or normal stress in a bi-axial stress system is given by, $\sigma_{t1} = \frac{\sigma_{yt}}{F.S.}$, for ductile materials $= \frac{\sigma_u}{F.S.}$, for brittle materials σ_{yt} = Yield point stress in tension as determined from simple tension test, and σ_u = Ultimate stress. Brittle materials which are relatively strong in shear but weak in tension or compression, this theory are generally used.	02 02
4		**Attempt any TWO of the following:**	**12**
	A	**Describe the diameter of rear axle shaft for fully floating type with following data-** **Engine Power = 60 kW at 3000 rpm** **Gear Box Ratio = 4.5:1, 2.5:1, 1.6:1, 1:1 ;** **Differential reduction = 5:1** **f_s for shaft = 70 N/mm^2**	**06**
		Answer: **P = 60 kW = 60X 10^3 W** **N = 3000 rpm,** **Max gear ratio, G_1 = 4.5 : 1,** **Differential reduction, G_d = 5:1** **Now the Torque transmitted by the engine T_e :** $P = \frac{2\pi N T_e}{60}$ $\therefore T_e = \frac{P \, X \, 60}{2 \, X \, \pi \, X \, N}$ $= \frac{60 \, X \, 10^3 \, X \, 60}{2 \, X \, \pi \, X \, 3000}$ **= 190.98 Nm** **= 190.98 X 10^3 Nmm** **Now Torque transmitted by rear shaft T_{ra} :** $T_{ra} = T_e \, X \, G_1 \, X \, G_d$ $\therefore T_{ra}$ = 190.98 X 10^3 X 4.5 X 5 $\therefore T_{ra}$ = 4297.05 X 10^3 Nmm Let	 02 02

		d = diameter of rear axle Tra $= \frac{\pi}{16}$ X fs X d^3 ∴ 4297.05 X 10^3 = $\frac{\pi}{16}$ X 70 X d^3 ∴ d^3 = 312638.51 ∴ **d = 67.87 mm Say 68 mm**	02
	B	**A multi disc clutch has 5 plates having 4 pairs of active friction surface if the intensity of pressure is not to exceed 0.127N/mm^2. Find the power transmitted at 500 r.p.m. The outer and inner raddi of friction surfaces are 125 mm and 75 mm respectively. Assume uniform wear and take coefficient of friction =0.3.**	06
		Answer: **Given:** n1 + n2 = 5, n = 4, p = 0.127 N/mm^2 , N = 500 rpm, r1 = 125mm, r2 = 75mm , μ = 0.3. We know that for uniform wear p.r = C (a constant). Since the intensity of pressure is maximum at inner radius r2, therefore p. r_2 = C or C = 0.127 X 75 = 9.525 N/mm axial force required to engage the clutch , W = 2 π C ($r_1 - r_2$) ∴ W = 2 π X 9.525 (125 – 75) ∴ W =2992.36 N	01
		Mean radius of friction surface, R = $\frac{r_1 + r_2}{2}$, R = $\frac{125+75}{2}$ **R = 100 mm = 0.1 m**	01
		We know that torque transmitted , T = n.μ.W.R T = 4 X 0.3 X2992.36 X0.1 **T = 359.08 Nm**	02
		∴ Power Transmitted, P = $\frac{2\pi NT}{60}$ ∴ P= $\frac{2\ X\ \pi\ X\ 500\ X\ 359.08}{60}$ ∴ P = 18801.38 W **∴ P = 18.801 kW**	02

	C	**Design the piston pin with following data – Maximum pressure on piston = 4N/mm^2, Diameter of piston = 70mm,** **Allowable stresses due to bearing and bending and shearing are given 30 N/mm^2,80 N/mm^2 and 60 N/mm^2 respectively.**	06
	Ans	**Answer:** Dia. of piston = D = 70 mm. Max. pressure = Pmax = 4 N/mm^2 Bearing pressure Pb = 30 N/mm^2 Bending stress = σ b = 80 N/mm^2 Shearing stress = τ = 60 N/mm^2 **Maximum gas load,** $W = \frac{\pi}{4} D^2 \times P_{max}$ $W = \frac{\pi}{4} \times 70^2 \times 4$ $W = 15.39 \times 10^3\ N$ 1. Design the piston pin on the basis of bearing pressure Let, d_{po} = outer dia. of piston pin l_p = length of piston pin in small end of connecting rod $l_p = 0.45 x D = 0.45 x 70$ l_p = 31.5 mm $F = d_{po} \times l_p \times P_b$ $d_{po} = \frac{15.3938 \times 10^3}{31.5 \times 30}$ d_{po} = 16.29mm **d_{po} = 17mm** 2. Designing the piston pin on the basis of bending. 'Bending moment 'M' is calculated as $M = F \times \frac{D}{8}$ $M = \frac{15.3938 \times 10^3 \times 70}{8}$ $M = 134.69 \times 10^3$ N-mm	01 01 ½

		We know that, $M = \frac{\pi}{32} x\ \sigma_b\ x\ (d_{po})^3$ $134.69\ x\ 10^3 = \frac{\pi}{32} x\ \sigma_b\ x\ (17)^3$ $\boldsymbol{\sigma_b = 279.2589\ N/mm^2}$ The induced bending stresses are greater than permissible bending stress 80N/mm2 hence redesign is necessary. Now redesign value of d_{po} $M = \frac{\pi}{32} x\ \sigma_b\ x\ (d_{po})^3$ $134.69\ x\ 10^3 = \frac{\pi}{32} x\ 80\ x\ (d_{po})^3$ $d_{po} = 25.79$ mm **d_{po} = 26 mm** **c) Designing piston pin on the basis of shear stress, due to double shear.** $F = 2x\pi/4(Dpo)^2 x\ \tau$ $15.39\ x\ 10^3 = 2x\ \pi/4\ x\ 26^2 x\ \tau$ $T = 14.49\ N/mm^2$ The induced shear stresses are less than permissible shear stress. Hence design is safe. **d) The total length of piston pin is taken as** L_{pt} = 0.9D= 0.9x70 = 63mm 3. Designing piston pin on the basis of shear stress. $F = \frac{2\pi}{4} x\ (d_{po})^2\ x\ \tau$ $15.39\ x\ 10^3 = \frac{2\pi}{4} x\ (26)^2\ x\ \tau$ **$\tau = 14.49\ N/mm^2$** The induced shear stresses are less than permissible shear stress. Hence Design is safe 4. The total length of piston is taken as L_{pt} = 0.9 D = 0.9 x 70 = **63 mm**	½ 01 ½ ½ ½ ½
5		**Attempt any TWO of the following:**	**12**
	A	**State functions and name suitable materials for connecting rod. Select suitable cross-section for connecting rod with justification.**	**06**

	Ans	**Answer**: **Functions of connecting rod:** A connecting rod, also called a con rod, is the part of a piston engine which connects the piston to the crankshaft. Together with the crank, the connecting rod converts the reciprocating motion of the piston into the rotation of the crankshaft. The connecting rod is required to transmit the compressive and tensile forces from the piston, and rotate at both ends. **Material for connecting rod:** 1. There are some materials that are commonly used in connecting rods such as alloys of steel, aluminum and titanium. Mostly connecting rods are made by forged steel. It is widely use because it has high tensile and compressive strength. 2. Titanium is used in making connecting rods, since very long time but when it comes to lightweight of connecting rod, aluminum is preferred. Titanium is the most costly material among steel and aluminum. Another demerit of titanium is their fatigue life. Aluminum is also used for a long time. 3. Most of the connecting rods are made of steel but aluminum can also be used to manufacture connecting rods because of its light weight and it can also absorb high impact shock but its durability will be suffered. Titanium can also be used because it has good strength but it is expensive. **Cross – Section of connecting rod:** 1. The I-section of the connecting rod is used due to its lightness and to keep the inertia forces as low as possible. It can also withstand high gas pressure. 2. Sometimes a connecting rod may have rectangular section. For slow speed engines, circular sections may be used.	**02** **02** **02**
	B	**State functions and materials for front axle. Draw proportionate diagram of front axle showing cross-section at different positions.**	**06**
	Ans	**Answer:** **The functions of front axle : (*Any 02*)** 1) It supports the weight of front part of the vehicle. 2) It facilitates steering. 3) It absorbs shocks which are transmitted due to road surface irregularities. 4) It withstands cornering forces and braking torque etc. 5) If front axle is live, it transmits engine torque. 6) If front axle is live, it withstands torque reaction, driving thrust. **Material:** It is usually a steel drop forging having 0.4 % carbon steel or 1-3 % nickel steel.	**02** **02** **02**



Conventional front axle

	C	**Explain basic automobile component design procedure**	**06**
		Answer: **Following procedure is carried out for designing any machine element:** **1. 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. 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. **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.	**06**
6		**Attempt any TWO of the following:**	**12**
	A	**Define stress concentration. State its causes. Explain different methods to reduce stress concentration with suitable examples.**	**06**
		Answer: **Stress Concentration:** Whenever a machine component changes the shape of its cross section, the simple stress distribution no longer holds good and neighbourhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. **OR** Whenever there is a change in cross section of machine components, it causes high localized	**02**

stresses. This effect is called as stress concentration

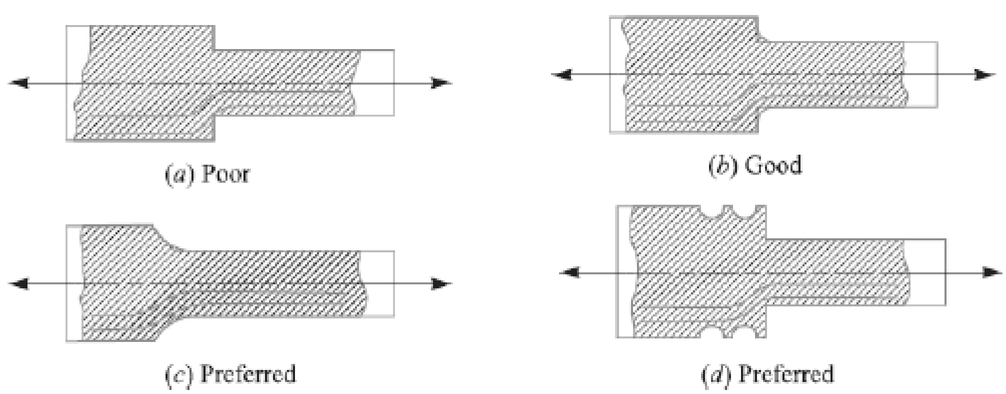
. **Causes of Stress Concentration: (Any Four – ½ Marks Each)**

i) Variation in properties of material from point to point due to cavities, cracks or air pockets.
ii) Abrupt changes of shape and cross section. **02**
iii) Concentrated loads applied at points or small areas of machine elements.
iv) Force flow line is bent as it passes from the shank portion to threaded portion of component due to changes in cross section. This results in stress concentration in transition plane.
v) Local Pressures

The methods of reducing stress concentration: (any Two Methods)

1. **The methods of reducing stress concentration in cylindrical members subjected to tensile load.**

02 (any two methods)



In Fig. (a) it is seen that stress lines tend to bunch up and cut very close to the sharp re-entrant corner. In order to improve the situation, fillets may be provided, as shown in Fig. (b) and (c) to give more equally spaced flow lines. It may be noted that it is not practicable to use large radius fillets as in case of ball and roller bearing mountings. In such cases, notches may be cut as shown in Fig.(d)

2. The methods of reducing stress concentration in cylindrical members with shoulders subjected to bending moment.

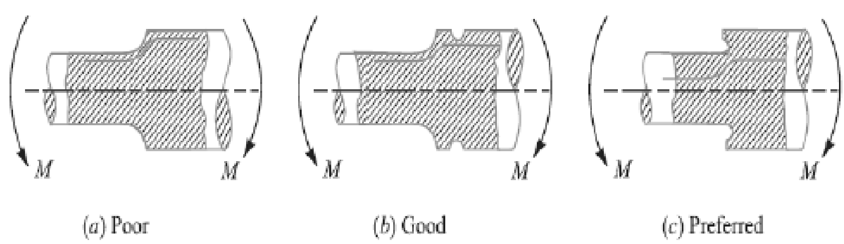


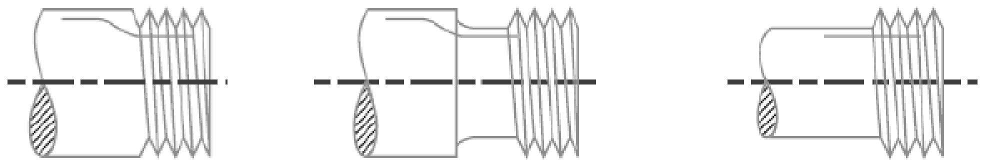
Figure. Methods of reducing stress concentration in cylindrical members with shoulders.

3. The methods of reducing stress concentration in cylindrical members with shoulders hole, subjected to tensile load.

(a) Poor (b) Preferred

Figure. Methods of reducing stress concentration in cylindrical members with holes.

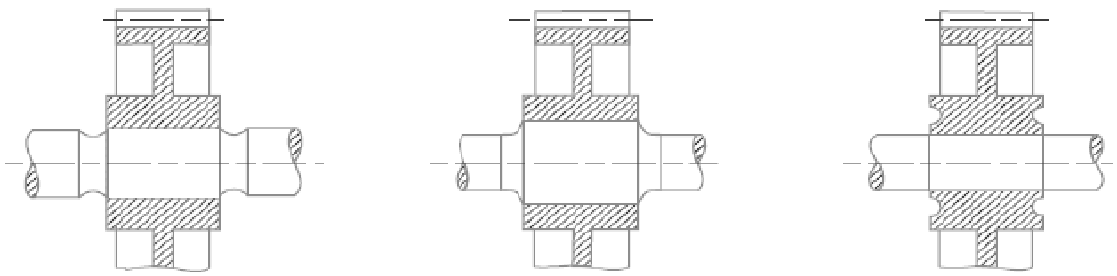
4. Methods of reducing stress concentration in cylindrical members with threads.



(a) Poor (b) Good (c) Preferred

Figure. Methods of reducing stress concentration in cylindrical members with threads.

5. Methods of reducing stress concentration of a press fit



(a) (b) (c)

Figure. Methods of reducing stress concentration of a press fit

	B	**Explain design procedure for propeller shaft.**	**06**

	Ans	**Answer:** **Design procedure of propeller shaft:** The propeller shaft is designed on the basis of torsional shear stress. By using the torsional equation, $$\frac{T_p}{J_p} = \frac{\sigma_s}{r}$$ *Where,* T_p = Torque transmitted by propeller shaft. Tp = Te x G1 Te = Engine Torque. G1 = Maximum gear Ratio in Gear Box Jp = 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 fibres. r = d/2 (for Solid shaft) r = d_o/2 (for Hollow shaft) After simplifying the equations, $$T_p = \frac{\pi}{16}\,\sigma_s\,d^3$$ …………………For solid shaft $$T_p = \frac{\pi}{16}\,\sigma_s\,d_o^3\,(1-k^4)$$……….. 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 propeller shaft.	01 01 01 01 01 01
	C	**A truck spring has 10 numbers of leaves. The spring supports are 1185 mm apart and central (support) is 85 mm wide. The load on the spring is 20 kN and take permissible**	06

		stress of 300 N/mm^2. Determine the thickness of the leaves If the width of spring is 85mm.	
	Ans	n = 10, $2L_1$ = 1185mm, l =85mm, 2W= 20 X 10^3 N, f_F = 300 N/mm^2, b= 85mm, It is usual to provide two full length leaves and the rest graduated leaves as let t = thickness of leaves n_f = 02. Now, W = 10 X 10^3 N we know that effective length of the spring, $2L = 2L_1 - l = 1185-85 = 1100$ mm $\therefore L = \frac{1100}{2} = 550$ mm And number of graduated leaves $n_g = n - n_F = 10-2 = 08$ assuming that the leaves are not initially stressed therefore maximum stress or bending stress for full length leaves (f_F) $300 = \frac{18\,W\,L}{b.t^2(2n_g+3n_f)}$ $= \frac{18\ X\ 10\ X\ 10^3\ X\ 550}{85\ X\ t^2\ (2X8+3X2)}$ $\therefore t^2 = 176.47$ $\therefore t = 13.28$ mm **t = 14 mm**	**01** **01** **01** **03**