Model Answer

Subject: Design of Automobile Components **Subject Code: 17525**

Important Instructions to examiners:

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

Q. No.	Sub Q. N.	Answer	Marking Scheme
1.	**(A)**	Attempt **any three** of the following:	**12**
	a)	Define factor of safety. State the factors affecting its selection.	**04**
		Answer: ***(Def^n- 2 marks, List of factors- 2 marks.)*** **Factor of Safety:** Factor of safety is defined as the ratio of the maximum stress to the working stress or design stress. Mathematically, $$Factor\ of\ Safety = \frac{Maximum\ stress}{Working\ stress}$$ In case of ductile material, $$Factor\ of\ Safety = \frac{Yeild\ point\ stress}{Working\ stress}$$ In case of brittle material, $$Factor\ of\ Safety = \frac{Ultimate\ stress}{Working\ stress}$$ **The factors that influence the magnitude of factor of safety:*(any two)*** 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	**04**

		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.	
	b)	Define the term: i) Fatigue and ii) 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. 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 with suitable example:** 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 material depends on: • Type of load • Surface finish • Size of object • Working temperature	**02** **02**
	c)	State the effect of keyways on the strength of shaft.	**04**
		Answer: **Effect of key way cut into the shaft:** The keyway cut into the shaft reduces the load carrying capacity of the shaft. This is due to the stress concentration near the corners of the keyway and reduction in the cross-sectional area of the shaft. It other words, the torsional strength of the shaft is reduced. The following relation for the weakening effect of the keyway is based on the experimental results by H.F. Moore. $e = 1- 0.2 (w/d) – 1.1 (h/d)$ where, e = Shaft strength factor, w = width of key way, d = diameter of shaft, and h = depth of keyway	**04**

		It is usually assumed that the strength of the keyed shaft is 75% of the solid shaft.	
	d)	Determine the bore and length of cylinder of 4-stroke diesel engine for following specification- Brake Power – 5KW, speed – 1200RPM, P_m -0.35 N/mm^2, Mechanical efficiency - 80%, L/D =1.08.	04
		Answer: **Given:** B.P. = 5kW= 5000 W N= 1200 r.p.m. or n= N/2 = 600 P_m=0.35N/mm^2=0.35× 10^6 N/m^2 $\eta_m = 80\% = 0.8$ L= 1.08D **Bore and length of cylinder:** Let D= Bore of the cylinder in mm A= across section area of the cylinder $= \frac{\Pi}{4} \times D^2 mm^2$ We know that the indicated power, $$I.P. = \frac{B.P.}{\eta_m} = \frac{5000}{0.8} = 6250\ watt$$	01
		We also know that the indicated power (I.P.), $$6250 = \frac{P_m.l.A.n}{60} = \frac{0.35\times10^6 \times 1.08D \times \Pi D^2 \times 600}{60\times4} = 2.96\times10^6 D^3$$ $$\therefore \quad D^3 = 6250/2.96\times10^6 = 2.11\times10^{-3} \text{ or } D = 0.128\text{m} = 128\text{mm}$$	01
		$$L = 1.08\times128 = 138.24mm \cong 139mm$$	01
		Taking a clearance on both sides of the cylinder equal to 15 % of the stroke therefore length of the cylinder. Length of cylinder= 1.15L= 1.15× 139**=160mm**	01
	B)	**Attempt any one of the following:**	06
	a)	Explain the design procedure of 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*	06

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,

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

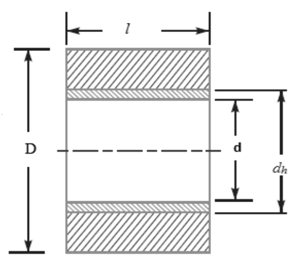
External diameter of the boss,

$$D = 2\,d$$

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

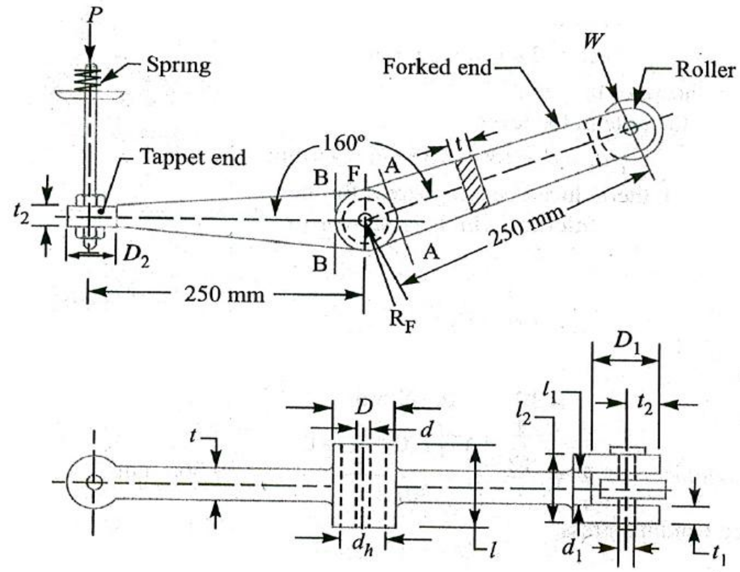


Bending moment at this section = $W \times$ L

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

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

OR



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.

b) A single plate dry clutch transmits 7.5KW at 900rpm. The axial pressure is 0.7N/mm^2. Determine the outer and inner diameters of frictional surfaces if μ=0.25. Take ratio of diameter as 1.25. Assume uniform wear theory. **06**

Answer:

$P = 7.5\times1000$ watt

$N = 900$rpm

$P_{max} = 0.1$ N/mm^2

$\mu = 0.25$

		$d_1/d_2 = r1/r2 = 1.25$ $n=2$ We know that power transmitted , $P = \frac{2\pi NT}{60}$ $\therefore 7.5 \times 10^3 = \frac{2\pi \times 900 \times T}{60}$ $\therefore T = 79.57\ Nm = 79.57 \times 10^3 Nmm$	01
		Since the intensity of pressure is considered as maximum, $\mathbf{C = P_{max} \times r_2 = 0.7\ r_2}$	01
		Axial thrust transmitted by the frictional surface, $\mathbf{W = 2\pi c\ (r_1 - r_2)}$ $W = 2\pi \times 0.7\ r_2 \times (1.25r_2 - r_2)$ $\mathbf{W = 1.099\ r_2^2\ N}$	01
		And mean radius of friction, $\mathbf{R = (r_1 + r_2)/2}$ $R = (1.25r_2 + r_2)/2$ $\mathbf{R = 1.125\ r_2\ mm}$	01
		We know that, torque transmitted, $\mathbf{T = n.\ \mu.\ W.\ R}$ $79.57 \times 10^3 = 2 \times 0.25 \times 1.099\ r_2^2 \times 1.125\ r_2$ $r_2^3 = 128714.99$ $r_2 = 50.49\ mm \cong 50mm$	01
		Now, $r_1 = 1.25\ r_2 = 1.25 \times 50 = 63.50mm =$ **64mm** Therefore diameters of frictional surfaces, $\mathbf{d_2 = 2r_2 = 100mm}$ and $\mathbf{d_1 = 2r_1 = 128mm}$	01
2.		Attempt **any four** of the following**:**	**16**
	a)	Explain maximum shear stress theory of failure.	**04**
		Answer: According to this theory, the failure or yielding occurs at a point in a member when the maximum shear stress in a bi-axial stress system reaches a value equal to the shear stress at yield point in a simple tension test.	**04**

Mathematically,

$$\tau_{max} = \tau_{yt} / F.S.$$

τ_{max} = Maximum shear stress in a bi-axial stress system,

τ_{yt} = Shear stress at yield point as determined from simple tension test, and

$F.S.$ = Factor of safety.

Since the shear stress at yield point in a simple tension test is equal to one-half the yield stress in tension, therefore the equation (i) may be written as

$$\tau_{max} = \frac{\sigma_{yt}}{2 \times F.S.}$$

This theory is mostly used for designing members of **ductile materials**.

b) Enlist any two applications of cotter and knuckle joint. **04**

Answer: **04**

Applications of cotter joint: ***(Any two – 1 mark each)***

1. It is used in connecting a piston rod to cross head of steam engine
2. It is used in joining a tail rod with piston rod of an air pump
3. It is used in valve rod and its stem
4. Foundation bolt

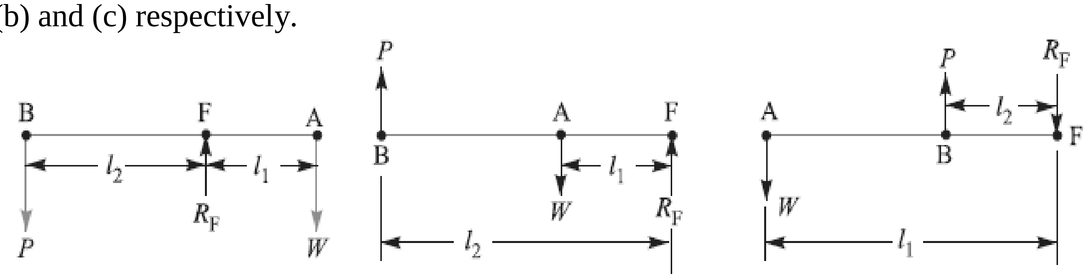
Applications of Knuckle joint: ***(Any two – 1 mark each)***

1. Tie rod joints for roof truss
2. Valve rod joint for eccentric rod pump rod joint
3. Tension link in bridge structure
4. Lever and rod connections
5. Swing arm of two wheeler
6. Connection of link rod of leaf springs in multi axle vehicles
7. Piston ,Piston Pin ,Connecting Rod
8. Connections of leaf spring with chassis

c) State different types of levers with suitable applications (any two). **04**

Answer: ***(any two- 2 marks each)*** **04**

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 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.

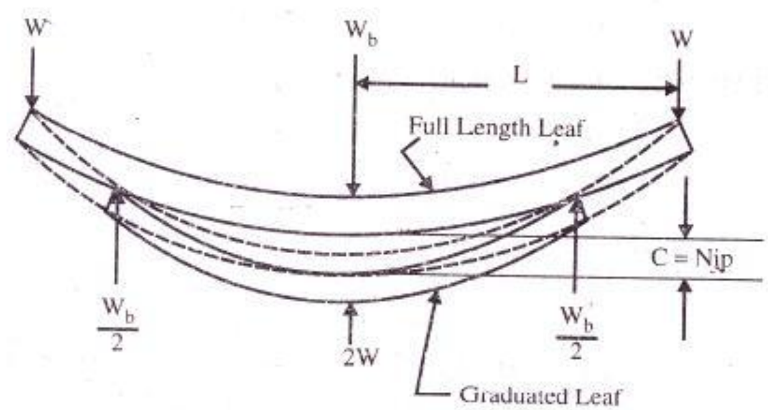
b) **Second type lever:** In the second type of levers, the load is in between the fulcrum and effort. In this case, the effort arm is more than load arm; therefore the mechanical advantage is more than one.
Examples: It is found in levers of loaded safety valves.

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.

	d)	Explain why nipping of leaf spring is necessary with neat sketch.	**04**

Answer: ***(Sketch – 2 marks & explanation – 2 marks)*** **04**

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'.

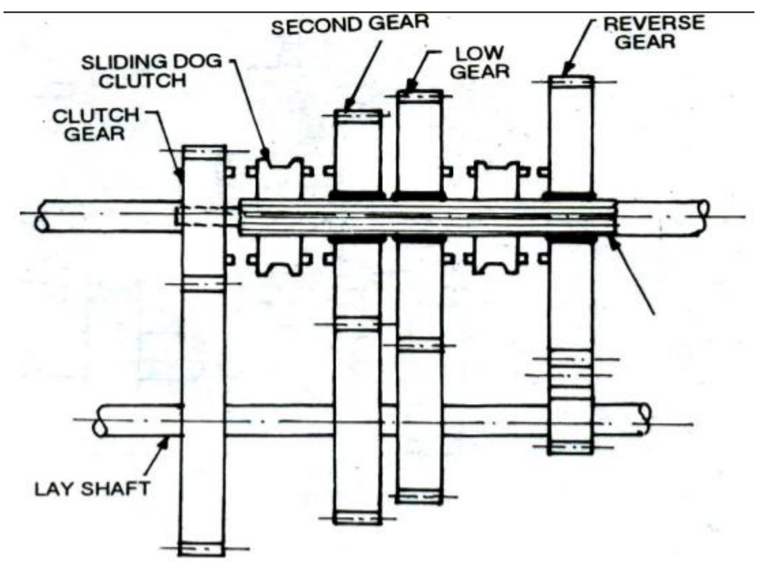
When the central bolt holding the leaves is tightened, the full length leaf bend back as shown by dotted line. And will have an initial stress in opposite direction. The graduated leaves will have an initial stress in the same direction as that of normal load. When the load is applied, the full length 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.	
	e)	Determine the thickness of plain cylinder head for 0.3 m cylinder. The maximum gas pressure is 3.2 N/mm^2. Take C=0.1, σ_t= Tensile stress= 42 N/mm^2.	04
		Answer: **Given Data:** D= 0.3m= 300mm P = 3.2 N/mm^2 C=0.1 σ_t= 42 N/mm^2 **Thickness of plain cylinder:-** $$t_h = D\sqrt{\frac{C.p}{\sigma_t}} = 300\sqrt{\frac{0.1\times 3.2}{42}} = 26.18 \text{ say } 27\,\text{mm}$$	04
3		Attempt **any four** of the following**:**	16
	a)	Explain aesthetic considerations in designing of automobile components.	04
		Answer: ***(Any two – 2 marks each)*** **Aesthetic consideration in designing of 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
	b)	Explain the term: i)Standardization and ii) Interchangeability in design.	04
		Answer: **i) Standardization:** It is defined as obligatory norms, to which various characteristics of a product should conform. The characteristics include	02

		materials, dimensions and shape of the component, method of testing and method of marking, packing and storing of the product. **Advantages of Standardization:-** 1. Mass production is easy. 2. Rate of production increases. 3. Reduction in labour cost. 4. Limits the variety of size and shape of product. 5. Overall reduction in cost of production. 6. Improves overall performance, quality and efficiency of product. 7. Better utilization of labour, machine and time. **ii) Interchangeability in design:** Interchangeable parts are parts (components) that are, for practical purposes, identical. They are made to specifications that ensure that they are so nearly identical that they will fit into any assembly of the same type. One such part can freely replace another, without any custom fitting (such as filing).	**02**
	c)	Write the type of Keys with their applications.	**04**
		Answer: **The following types of keys are *:(Any four types- 2marks, and one application each- 2 marks)*** 1. Sunk keys: (i) Rectangular Sunk Key (ii) Square sunk key (iii) Parallel sunk key (iv) Gib-headed key (v) Feather key (vi) Woodruff key 2. Saddle keys: (i) Flat saddle key (ii) Hollow saddle key 3. Tangent keys, 4. Round keys, 5. Splines **Their applications** 1 Sunk Keys- It is used, where key is to be removed frequently. 2 Saddle keys- They are suitable for light duty or low power transmission, as the power is transmitted due to friction. It is used as temporary fastening in fixing and setting eccentric parts, cams etc. 3 Splines- These splines are used for power transmission of very high order and also provide axial movement between shaft and mounted member. Practical applications of splines may be seen in gear shifting mechanism used in automobile gear boxes.	**04**
	d)	State the design considerations in semi-elliptical leaf spring.	**04**
		Answer: **Design considerations in semi-elliptical leaf spring:*(Any Eight)*** 1. Material available 2. Tensile strength 3. Yield strength	**04**

		4. Young Modulus 5. Design stress 6. Total length 7. Spring Weight 8. Arc length between axle seat and front eye 9. Arc height at axle seat 10. Spring rate 11. Normal static loading 12. Available space for spring width	
	e)	Write the design procedure for designing of piston head or crown by strength consideration.	**04**
		Answer: **Design of Piston Head or Crown Strength basis:** The piston head or crown is designed keeping in view the following two main considerations, i.e. 1. It should have adequate strength to withstand the straining action due to pressure of explosion inside the engine cylinder, and 2. It should dissipate the heat of combustion to the cylinder walls as quickly as possible. **I) On the basis of Strength:-** The thickness of the piston head (t_H), according to Grashoff's formula is given by $t_H = \sqrt{\frac{3p.D^2}{16\sigma_t}}$ (in mm) p = Maximum gas pressure or explosion pressure in N/mm^2, D = Cylinder bore or outside diameter of the piston in mm, and σ_t = Permissible bending (tensile) stress for the material of the piston in MPa or N/mm^2. It may be taken as 35 to 40 MPa for grey cast iron, 50 to 90 MPa for nickel cast iron and aluminium alloy and 60 to 100 MPa for forged steel.	**04**
4	**(A)**	Attempt **any three** of the following**:**	**12**
	a)	State the general design considerations.	**04**
		Answer: **Design considerations in automobile 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.	**04**

		8. Work shop facilities available. 9. Manufacturing cost. 10. Convenient of assembly and transportation.	
	b)	Design turn buckle rod diameter to withstand a load of 1600 N, if permissible stresses are 70 N/mm^2 and 60 N/mm^2 in tension and shear respectively.	**04**
		Answer: Given, P=1600 N, ft =70 N/mm^2, fs = 60 N/mm^2 Design load, $\mathbf{P_d = 1.3\ P = 1.3 \times 1600 = 2080\ N}$ Let , Core diameter of rod = dc Now, $Pd = \Pi/4.\ dc^2.\ ft$ $2080 = \Pi/4.\ dc^2.\ 70$ **dc = 6.15 mm** Rod diameter, d = 6.15 /0.84 **= 7.32 mm ≈ 8 mm**	**01** **02** **01**
	c)	Design a rear axle for engine power- 40 KW at a speed of 2000rpm. Lower gear box ratio -3:1 and differential reduction as 5. Take allowable shear stresses 56MPa.	**04**
		Answer: Given:- P = 40 kW N = 2000 rpm, Lower gear ratio, G_1 = 3 : 1, Differential reduction G_d = 5:1, Now the torque transmitted by the engine Te :- $P = (2 \times \pi \times N \times Te) /60$ $40\times10^3 = (2\times\pi \times2000\times Te) /60$ $\mathbf{Te = 190.98\ Nm = 190.98 \times 10^3\ Nmm}$ Now torque transmitted by rear axle shaft TRA, $TRA = Te \times G_1 \times G_d$ $TRA = 190.98 \times 10^3 \times 3 \times 5$	**01**

		TRA = 2864.78 × 10^3 Nmm	**01**
		Let, d = diameter of rear axle, $$T_{RA} = \frac{\pi}{16} f_s d^3$$ $2864.78 \times 10^3 = (\pi/16) \times 56 \times d^3$ $d^3 = 260539.37$ **d = 63.86 mm = 64mm**	**02**
	d)	Draw a neat sketch of constant mesh gear box.	**04**
		Answer:*(Neat sketch- 4marks)*  **Fig. Constant mesh gear box**	**04**

	(B)	Attempt **any one** of the following:	**06**
	a)	Define stress concentration. State it causes. Explain the different methods to reduce stress concentration with suitable example.	**06**

Answer:

Stress Concentration: Whenever a machine component changes the shape of its cross section, the simple stress distribution no longer holds good and neighborhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. **01**

OR

Whenever there is a change in cross section of machine components, it causes high localized stresses. This effect is called as stress concentration.

Causes of Stress Concentration: ***(Any Four – ½ Marks Each)*** **02**

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.

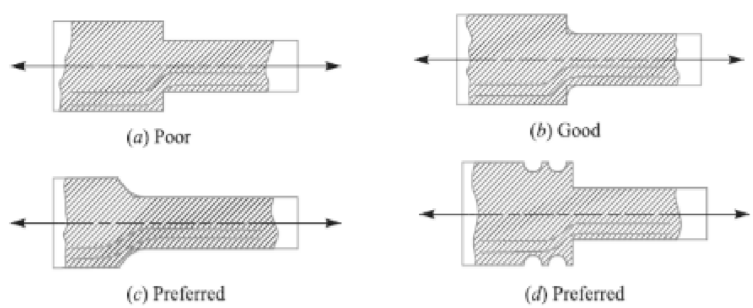
iii. Concentrated loads applied at points or small areas of machine elements.

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

Methods of reducing stress concentration: ***(Any three)*** **03**

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



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

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.

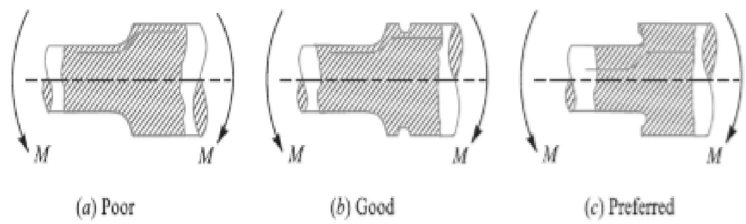
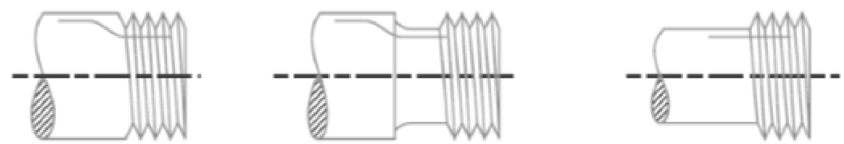


Figure. Methods of reducing stress concentration in cylindrical members with shoulders.

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

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

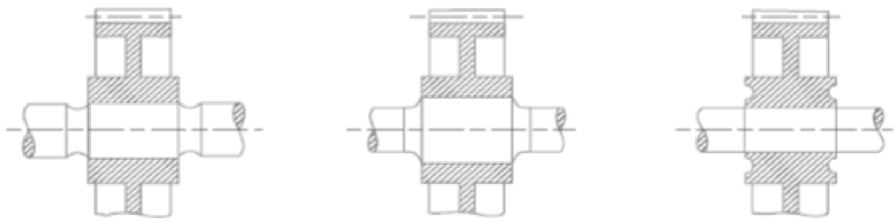
4. Methods of reducing stress concentration in cylindrical members with threads.



(*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



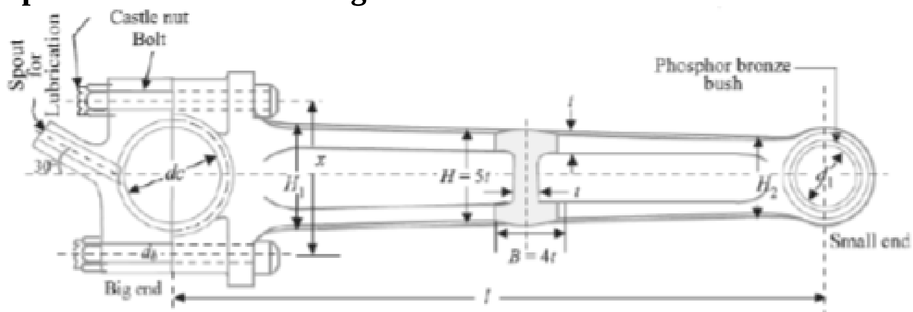
(*a*) (*b*) (*c*)

Figure. Methods of reducing stress concentration of a press fit

The stress concentration effects of a press fit may be reduced by making more gradual transition from the rigid to the more flexible shaft. The various ways of reducing stress concentration for such cases are shown in (a), (b) and (c).

	b)	Write the design procedure for connecting rod.	06
		Answer: ***(Figure-2mark, Design procedure-4 marks)***	06

Design procedure for connecting rod:



1. **Dimensions of cross-section of the connecting rod:**

According to Rankine's formula:

$$W_B = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{xx}}\right)^2}$$

Let-

WB = Buckling load (Maximum gas load)

σc = Crippling or Buckling stress or crushing stress.

A = Cross-sectional area of the connecting rod=11 t^2

a = Rankine's constant

L =Effective length of the connecting rod

(i.e distance between centres of crank pin and piston pin)

$k_{xx}{}^2 = 3.18\ t^2$ from this relation t (thickness of the flange and web of the section) can be determined.

Width of the section, *B* = 4 *t* and

Depth or height of the section, *H* = 5*t*

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 0.9*H***

The depth near the big end (or crank end) is taken ***H*2 = 1.1*H* to 1.25*H***.

2. Dimensions of the at the big end and small end of connecting rod:

Maximum gas force:

$$F_L = \frac{\pi D^2}{4} \times p$$

Where-

D = Cylinder bore or piston diameter in mm, and

p = Maximum gas pressure in N/mm2

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/mm2, and

d_p, l_p and pb_p = Corresponding values for the piston pin

load on the crank pin = Projected area × Bearing pressure

= dc . lc . pb_c (ii)

Similarly, load on the piston pin = dp . lp . pbp (iii)

Equating equation (i) and (ii),

we have,

FL = dc · lc · pbc

Taking lc = 1.25 dc to 1.5 dc,

the value of dc and lc are determined from the above expression.

Again, equating equations **(i)** and **(iii)**,

we have,

FL = dp · lp · pbp

Taking lp = 1.5 dp to 2 dp,

the value of dp and lp are determined from the above expression

3. **Size of bolts for securing the big end cap**:

FI = Inertia load acting on bolts

Let *dcb* = Core diameter of the bolt in mm,

σt = Allowable tensile stress for the material of the bolts in MPa,

and *nb* = Number of bolts. Generally two bolts are used.

Force on the bolts:

$$F_I = \frac{\pi}{4}(d_{cb})^2 \sigma_t \times n_b$$

From this expression, *dcb* is obtained. The nominal or major diameter (*db*) 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 (*tc*) may be determined as below,

Maximum bending moment acting on the cap will be taken as

		$M_C = \frac{F_1 \times x}{6}$ Where, x = Distance between the bolt centers. = Dia. of crankpin or big end bearing (dc) + 2 × Thickness of bearing liner(3 mm) + Clearance(3mm) Let, bc = Width of the cap in mm. It is equal to the length of the crankpin or big end bearing (lc), and σb = 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, $\sigma_b = \frac{M_C}{Z_C} = \frac{F_1 \times x}{6} \times \frac{6}{b_c (t_c)^2} = \frac{F_1 \times x}{b_c (t_c)^2}$ From this expression, the value of tc is obtain.	
5.		Attempt **any two** of the following:	**16**
	a)	Design socket & spigot type cotter joint with the following data; Load = 30kN, Allowable tensile stresses =50Mpa, Allowable crushing stresses = 90 Mpa & Allowable shear stresses =35 Mpa.	**08**
		Answer: **Given:** $P = 30 \times 10^3$ N $\sigma_t = 50 N/mm^2$ $\tau = 35 N/mm^2$ $\sigma c = 90 N/mm^2$ Let- d= diameter of rod d1 = outer diameter of socket d2= outer diameter of spigot d3= diameter of spigot collar d4= diameter of socket collar a= distance between end of slot and end of spigot b=width of cotter c=width of socket collar e=width of socket neck t= thickness of cotter t1= thickness of spigot collar l= length of cotter	

		1.<u>Find diameter of rod "d" considering failure in tension</u>: $$P = \frac{\pi}{4}(d)^2 \times \sigma t$$ $30 \times 10^3 = \frac{\pi}{4}(d)^2 \times 50$ **<u>d=27.64 mm say 28 mm</u>**	
		2. <u>Find outside diameter of spigot " d2" considering failure in tension :</u>	**01**
		$$P = [\frac{\pi}{4}(d2)^2 - d2 \times t] \times \sigma t$$ $30 \times 10^3 = [\frac{\pi}{4}(d2)^2 - d2 \times \frac{d2}{4}] \times 50$ $(assuming\ \ t = \frac{d2}{4})$ $30 \times 10^3 = \frac{(d2)^2}{4}[\pi - 1] \times 50$ **<u>d_2=33.46 mm say 34 mm</u>** $t = \frac{d2}{4} = \frac{34}{4} = 8.5 = 9\ mm$ **<u>t=9 mm</u>**	**01**
		3.<u>Check the crushing stress considering failure at cotter in crushing :</u> $$P = (d_2 \times t)\sigma_c$$ $$30 \times 10^3 = (34 \times 9)\ \sigma_c$$ $$\sigma_c = 98.03 \frac{N}{mm^2} > Permissible\ crushing\ stress\ 90\ \frac{N}{mm^2}$$ So design is not safe.. Hence redesign the values of d_2 & t $$P = (d_2 \times t)\sigma_c$$ $$30 \times 10^3 = (d_2 \times \frac{d_2}{4})\ \times 90$$ $$d_2 = 36.51\ mm = 37\ mm$$ $t = \frac{d2}{4} = \frac{37}{4} = 9.25 = 10\ mm$ **<u>d_2= 37 mm and t=10 mm</u>**	**01**
		4.<u>Find outside diameter of socket "d1 " considering failure socket in tension</u> $$P = [\frac{\pi}{4}({d_1}^2 - {d_2}^2) - (d_1 - d_2)t] \times \sigma t$$ $$30 \times 10^3 = [\frac{\pi}{4}(d_1)^2 - \frac{\pi}{4}(37)^2 - (d_1 \times 10) + (37 \times 10)] \times 50$$ $$\frac{30 \times 10^3}{50} = 0.785(d_1)^2 - 1075.21 - 10d_1 + 370$$ $$0.785(d_1)^2 - 10d_1 - 1305.21 = 0$$ $$(d_1)^2 - 12.74d_1 - 1662.69 = 0$$ For this quadratic equation- a=1,b=-12.74 & c=-1662.69 $$d_1 = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$ $d_1 = 47.64\ mm\ say\ 48\ mm$ **<u>d_1= 48 mm</u>**	**01**

5. Find the diameter of spigot collar considering failure in crushing :

$$P = \frac{\pi}{4}(d_3{}^2 - d_2{}^2) \times \sigma_c$$
$$30 \times 10^3 = \frac{\pi}{4}(d_3{}^2 - 37^2) \times 90$$
$$\frac{30 \times 10^3}{90} = \frac{\pi}{4}(d_3)^2 - \frac{\pi}{4}(37)^2$$
$$333.33 = 0.785(d_3)^2 - 1075.21$$
$d_3 = 42.35\ mm\ say\ 43\ mm$ **d_3= 43 mm**

6. Find diameter of socket collar considering failure in crushing : 01

$$P = (d_4 - d_2) \times t \times \sigma_c$$
$$30 \times 10^3 = (d_4 - 37) \times 10 \times 90$$
$d_4 = 70.33\ mm = 71\ mm$ **d_4= 71 mm**

7. Find the width of cotter "b" considering failure in shear :

$$P = 2 \times b \times t \times \tau$$
$$30 \times 10^3 = 2 \times b \times 10 \times 35$$
$b = 42.85\ mm = 43\ mm$ **b= 43 mm**

8. Find the thickness of spigot collar " t1" by considering failure in shear: 01

$$P = \pi \times d_2 \times t_1 \times \tau$$
$$30 \times 10^3 = \pi \times 37 \times t_1 \times 35$$
$t_1 = 7.37\ mm = 8\ mm$ **t_1= 08 mm**

9. Find the thickness of socket collar "c" by considering failure in shear :

$$P = 2(d_4 - d_2) \times c \times \tau$$
$$30 \times 10^3 = 2(71 - 37) \times c \times 35$$
$c = 12.60\ mm = 13\ mm$ **c= 13 mm**

01

10. Find the distance from cotter slot to end of spigot rod "a"

By Considering Failure of shear

$$P = 2 \times d_2 \times a \times \tau$$
$$30 \times 10^3 = 2 \times 37 \times a \times 35$$
$a = 11.58\ mm = 12\ mm$ **a= 12 mm**

11. Find the length of cotter :

$$L = 4 \times d$$
$$L = 4 \times 28$$
$L = 112\ mm$ **L= 112 mm**

12. Find the thickness of socket of neck"e" :

$$e = 1.2 \times d$$
$$e = 1.2 \times 28$$
$e = 33.6\ mm = 34\ mm$ **e= 34 mm** 01

	b)	A hollow propeller shaft of a car with outside diameter of 75mm transmits 22.5kW at 1500 rpm to the wheels which are 900 mm in diameter. If the allowable shear stress is 60 N/mm^2, find the cross-section of shaft. Take gear box reduction 5.	**08**
		Answer: Given data: $d_0 = 75mm$ $\tau = 60N/mm^2$ $P = 22.5kW$ $P = 22.5kW = 22.5\times10^3 W$ Gear reduction $G_1 = 5$ Now torque produced by the engine 'T_e' $P = \frac{2\Pi N T_e}{60}$ $22.5\times10^3 = \frac{2\times3.14\times1500\times T_e}{60}$ $T_e = 143.24\times10^3$ N-mm	**02**
		Now torque transmitted by the propeller shaft 'T_p' we know that $T_p = T_e \times G_1$ $= 143.24\times10^3 * 5$ N-mm $T_p = 716.2\times10^3$ N-mm	**02**
		For hollow shaft Let d_0 = outer diameter of shaft d_i = inner diameter of shaft $k = \frac{d_i}{d_0} = \frac{d_i}{75}$ We know that $T_P = \frac{\Pi}{16}\tau(d_0)^3(1-k^4)$	**01**
		$716.2\times10^3 = \frac{3.14}{16}\times60\times(75)^3(1-k^4)$ $1-k^4=0.14$ $k^4=0.855$	

		$\frac{(d_i)^4}{(75)^4} = 0.855$ **d_i= 72.1mm= 73mm**	**03**
	c)	Design piston pin for following data: Max. gas pressure = 4 N/mm^2. Diameter of piston =70mm, Allowable stresses due to bearing, bending and shear are 30 N/mm^2,80 N/mm^2,60 N/mm^2 respectively.	**08**
		Answer: Given data, Dia. of piston = D = 70 mm. Max. pressure = P_{max} = 4 N/mm^2 Bearing pressure P_b = 30 N/mm^2 Bending stress = σ_b = 80N/mm^2 Shearing stress = τ = 60 N/mm^2 **Maximum gas load,** $= \frac{\pi D^2}{4} \times p_{\mathbf{max}}$ $F = \frac{\pi}{4}(70)^2 \times 4 = 15.3938 \times 10^3$ N	**02**
		(a) 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 = 0.45xD = 0.45x70 l_p = 31.5 mm $F = dp_o \times l_p \times P_b$ $dp_o = 15.3938 \times 10^3 / 31.5 \times 30$ dp_o = 16.29 mm $dp_o \approx$ 17 mm	**02**
		(b)Designing the piston pin on the basis of bending. 'Bending moment 'M' is calculated as M = F x D /8 $= \frac{15.3938 \times 10^3 \times 70}{8}$ N-mm	

		M = 134.69 x 10^3 N-mm M = π / 32xσ $_b$ x $(d_{po})^3$ σ_b = 279.2589N/mm^2 The induced bending stresses are greater than permissible bending stress 80N/mm^2 Hence redesign is necessary. Now redesign value of d_{po} M = π / 32x σ $_b$ 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 τ 15.39 x 10^3 =2x $\pi/4$ x 26^2x τ 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	 **02** **02**
6.		Attempt **any two** of the following:	**16**
	a)	Design a flange coupling to transmit 15kW at 900 rpm, for the following data. Service factor =1.35, Shear stress = 40MPa; Shear stress for C.I. = 8 MPa, Crushing stress = 80 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 Outer diameter of the hub, $D=2d=2\times35=70mm$ Length of hub, $L=1.5d=1.5\times35=52mm$ 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 $(T_{\max})$ $2.15\times10^3=\frac{\Pi}{16}\times\tau_c\left[\frac{D^4-d^4}{D}\right]=\frac{\Pi}{16}\times\tau_c\left[\frac{(70)^4-(35)^4}{70}\right]=63147\tau_c$ $\tau_c=215\times10^3/63147=3.4N/mm^2=3.4MPa$ 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 $\therefore\ l=L=52.5mm$ 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 $(T_{\max})$. $2.15\times10^3=l\times w\times\tau_k\times\frac{d}{2}=52.5\times12\times\tau_k\times\frac{35}{2}=11025\tau_k$ $\tau_k=215\times10^3/11025=19.5N/mm^2=19.5MPa$ Considering the key in crushing. We know that the maximum torque transmitted $(T_{\max})$. $2.15\times10^3=l\times\frac{t}{2}\times\sigma_{ck}\times\frac{d}{2}=52.5\times\frac{12}{2}\times\sigma_{ck}\times\frac{35}{2}=5512.5\sigma_{ck}$ $\sigma_{ck}=215\times10^3/5512.5=39N/mm^2=39MPa$ Science the induced shear and crushing stresses in the key are less than the permissible stresses therefore the **design for key is safe.** 3. **Design for flange**: The thickness of flange (t_f) is taken as 0.5 d $t_f=0.5d=0.5\times35=17.5mm$ Let us now check the induced shearing stress in the flange by considering the flange at	**02** **02**

		the junction of the hub in shear. We know that the maximum torque transmitted $(T_{\max})$ $215\times10^3=\frac{\Pi D^2}{2}\times\tau_c\times t_f=\frac{\Pi(70)^2}{2}\times\tau_c\times17.5=13473\tau_c$ $\therefore \quad \tau_c=215\times10^3/13473=1.6N/mm^2=1.6\text{MPa}$ 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 $D_1=3d=3\times35=105mm$ The bolts are subjected to shear stress due to the torque transmitted. We know that maximum torque transmitted $(T_{\max})$, $215\times10^3=\frac{\Pi}{4}\times(d_1)^2\tau_b\times n\times\frac{D_1}{2}=\frac{\Pi}{4}(d_1)^2 40\times3\times\frac{105}{2}=4950(d_1)^2$ $\therefore \quad (d_1)^2=215\times10^3/4950=43.43 \text{ or } d_1=6.6mm$ 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. $D_2=4d=4\times35=140mm$ Thickness of the protective circumferential flange. $t_p=0.25d=0.25\times35=8.75 say 10\,\text{mm}$	**02** **02**
	b)	A four speed gear box is to be constructed for providing the ratios of 1.0, 1.46, 2.28 and 3.93 to 1 as nearly as possible. The module of each 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 shafts.	**08**
		Answer: First Gear Ratio: $G_1=\frac{T_B}{T_A}\times\frac{T_D}{T_C}=3.93$ We have $\frac{T_B}{T_A}\times\frac{T_D}{T_C}=\sqrt{3.93}=1.98$ Adopting $T_A=T_C=15 \text{ the lowest value given}$ We get $T_B=T_D=1.98\times15=29.7=30$	**02**

		Thus actual ratio $= \frac{30}{15}\times\frac{30}{15} = 4:1$ $T_A + T_B = T_C + T_D = T_E + T_F = T_G + T_H = 45$	01
		Second gear ratio: $G_2 = \frac{T_B}{T_A}\times\frac{T_F}{T_E} = 2.28$ Or $\frac{T_F}{T_E} = 2.28\times\frac{T_A}{T_B} = 2.28\times\frac{15}{30} = 1.14$	01
		Hence, $T_E + T_F = 2.14\times T_E = 45$ Or $T_E = \frac{45}{2.14} = 21$	01
		and $T_F = 45-21 = 24$ The actual ratio $= \frac{30}{15}\times\frac{24}{21} = 2.286:1$ Third gear ratio: $G_3 = \frac{T_B}{T_A}\times\frac{T_H}{T_G} = 1.46$	01
		Or $\frac{T_H}{T_G} = \frac{1.46}{2} = 0.73$ But $T_H + T_G = 45$	01
		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 $G_4 = 1:1$	01
		The center distance between the shaft $= \frac{3.25\times 45}{2}$ $= 73.125\text{mm}$	

	c)	i) Design piston ring for following: Number of ring =5 , wall pressure P_w= 0.035 N/mm^2, Bending Stress for ring =85 N/mm^2, Diameter of cylinder bore =240 mm.	04
		Answer : Given data: Number of ring n_R=5, D =240 mm Pw=0.035 N/mm^2 σ_t =85 N/mm^2 **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 $t_1 = D\sqrt{\frac{3p_w}{\sigma_t}}$ $t_1 = 240\sqrt{(3x0.035)/85}$ t_1=8.43 mm = say 9mm 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. =0.7x9=6.3 mm **The minimum axial thickness** (t_2) may also be obtained from the following empirical relation: $t_2 = \frac{D}{10\, n_R}$ where n_R = Number of rings. t_2 =240/(10*5) t_2=4.8 mm Selecting maximum value of above t_2=6.3mm **Width of other ring lands**, b_2 = 0.75 t_2 to t_2 =0.85 t_2=0.85x6.3=5.35 mm . **The gap between the free ends of the ring** is given by 3.5 t_1 to 4 t_1. =3.75t1=3.75x9=33.75mm **Length of the Ring section** Length of the Ring section = 5 t_2 + 4b_2 =(5x6.3) +4 x5.35) **= 52.90 mm**	01 01 01 01
	c)	ii) Design the skirt length of the piston with the following data of petrol engine. Maximum pressure inside the cylinder =6.5 N/mm^2. Piston diameter =100mm, side	04

		thrust is limited to 10% of maximum load on the piston. Allowable bearing pressure = 0.3 N/mm^2.	
		Answer: Given data, Pmax =6.5 N/mm^2 Piston diameter =D= 100mm Side thrust = 10%=0.10 $P_b = 0.3 N/mm^2$ Let, R = Normal side thrust acting on piston skirts F = Total force produced due to combustion P_{max} = max. gas pressure inside the engine D = Dia. Of piston $F = P_{max} \times \frac{\pi}{4} D^2$ F= 6.5 x0.786x100^2 F=51050.8N R=0.1F Side thrust = 10%=0.10 $\therefore$ R 5105.08N 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 l_1=5105.08/(100x0.3) **l_1 =170.16 mm**	 01 01 01 01