Important Instructions to examiners:

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

Q. No.	Sub Q. N.	Answer	Marking Scheme
1	A	**Attempt any THREE of the following:**	12
	(i)	**Define factor of safety. What factors affect its selection ?**	04
		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, Factor of Safety = $\frac{\text{Maximum Stress}}{\text{Working or design stress}}$ **In Case of Ductile Material,** Factor of Safety = $\frac{\text{Yield Point Stress}}{\text{Working or Design Stress}}$ **In case of Brittle Material ,** Factor of Safety = $\frac{\text{Ultimate Stress}}{\text{Working or Design Stress}}$	02
		Selection of factor of safety (Any Four) 1. The reliability of the properties of the material and change of these properties during service ; 2. The reliability of test results and accuracy of application of these results to actual machine parts 3. The reliability of applied load; 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 set up during manufacture; 8. The extent of loss of life if failure occurs; 9. The extent of loss of property if failure occurs;	02

	(ii)	**Draw and explain the stress-strain diagram for ductile material.**	**04**

(Dia: 02+ Explanation: 02 Marks)

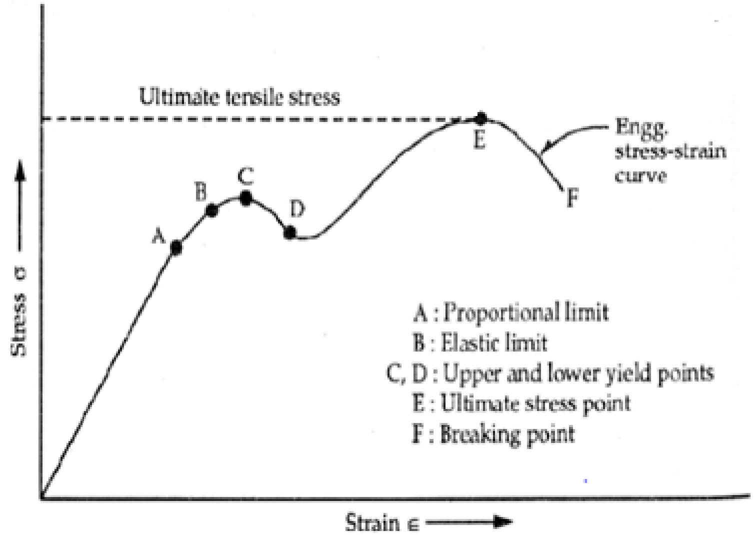


Fig. Stress-strain diagram for ductile material

02

1. Proportional limit. We see from the diagram that from point O to A is a straight line, which represents
that the stress is proportional to strain. Beyond point A, the curve slightly deviates from the straight line. It is thus obvious, that Hooke's law holds good up to point A and it is known as **proportional limit**. It is defined as that stress at which the stress-strain curve begins to deviate from the straight line.

2. Elastic limit. It may be noted that even if the load is increased beyond point A up to the point B, the material will regain its shape and size when the load is removed. This means that the material has elastic properties up to the point B. This point is known as **elastic limit.** It is defined as the stress developed in the material without any permanent set.

02

3. Yield point. If the material is stressed beyond point B, the plastic stage will reach i.e. on the the load, the material will not be able to recover its original size and shape. A little consideration will show that beyond point B, the strain increases at a faster rate with any increase in the stress until the point C is reached. At this point, the material yields before the load and there is an appreciable strain without any increase in stress. In case of mild steel, it will be seen that a small load drops to D, immediately after yielding commences. Hence there are two yield points C and D. The points C and
D are called the **upper** and **lower yield points** respectively. The stress corresponding to yield point is known as **yield point stress.**

4. Ultimate stress. At D, the specimen regains some strength and higher values of stresses are required for higher strains, than those between A and D. The stress (or load) goes on increasing till the point E is reached. The gradual increase in the strain (or length) of the specimen is followed with the uniform reduction of its cross-sectional area. The work done, during stretching the specimen, is transformed largely into heat and the specimen becomes hot. At E, the stress, which attains its maximum value is known as **ultimate stress.** Itis defined as the largest stress obtained by dividing the largest value

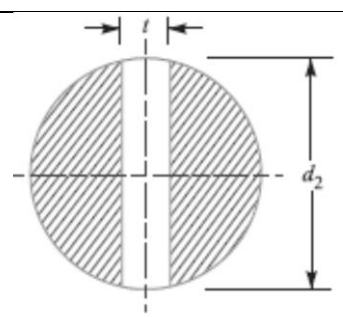
		of the load reached in a test to the original cross-sectional area of the test piece. **5. Breaking stress.** After the specimen has reached the ultimate stress, a neck is formed, which decreases the cross-sectional area ofthe specimen, as shown in Fig. The stress is, therefore, reduced until the specimen breaks away at point F. The stress corresponding to point F is known a**s breaking stress.**	
	(iii)	**State four applications of knuckle joint.**	**04**
		Answer: (Any four applications- 04 marks) Applications of Knuckle joint: **1.** Link of cycle chain **2.** Tie rod joints for roof truss **3.** Valve rod joint for eccentric rod pump rod joint **4.** Tension link in bridge structure **5.** Lever and rod connection of various types. **6.** swing arm of two wheeler **7.** Connection of link rod of leaf springs in multi axle vehicles **8.** Piston ,Piston Pin ,Connecting Rod **9.** Connections of leaf spring with chassis	**04**
	(iv)	**State the types of keys and their applications.**	**04**
		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.**	**02**
		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.	**02**

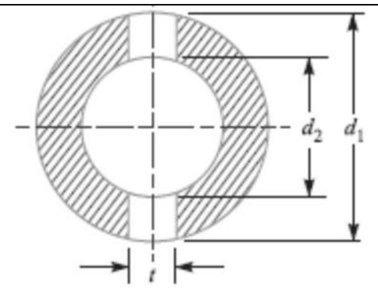
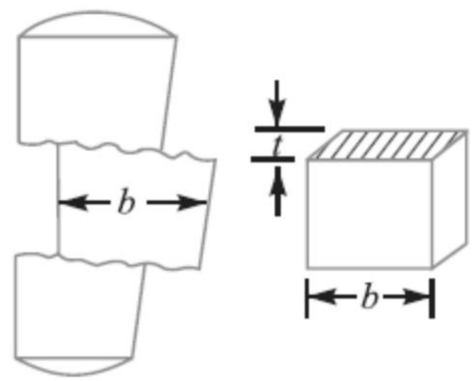
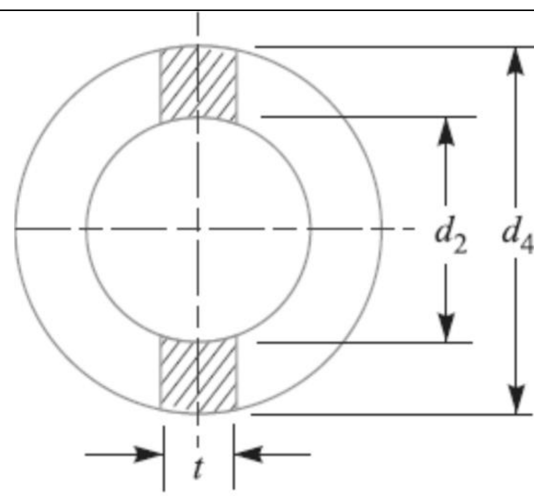
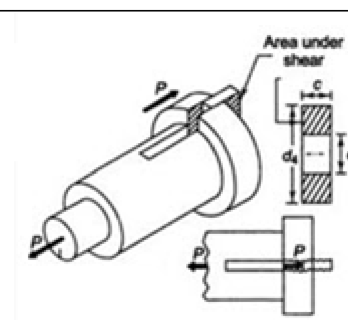
1	B	**Attempt any ONE of the following :**	6
	(i)	**Explain Ergonomic aspects and Aesthetics in designing automobile with their applications.**	
		Ergonomic aspects of machine design: The word 'ergonomics' is coined from two Greek words ergon = work and nomos = natural laws. Ergonomics means the natural laws of work. Anthropometry, Physiology and psychology are the components of ergonomics. **Anthropometry:** With the help of anthropometry dimensions of the components are finalized so that they can be easily handled by operator without fatigue and with consistence efficiency for e.g. diameter of steering wheel, distance from chair to pedals. **Physiology:** With the help of physiology components are designed to be operated by hand or foot force. For e.g. Gear shifting, Steering wheel are designed to be operated by hand because they require speed and accuracy which is imparted by hand and brake pedal clutch pedal etc. are designed to be operated by foot force because they require great amount of force is require than accuracy. **Psychology:** Psychology affects mode of operation for e.g. size, colour and push operation of emergency stops button of any machine. The size of emergency control is made large and painted in red so that they can be easily identified and always they are push operated. All these components make design of automobile components user friendly.	03
		Aesthetics in designing automobile: ***"Aesthetics is the branch of science which deals with nature of art and beauty. It is related to the appearance of the product ."*** When there are number of products in the market, having the same qualities of efficiency, durability and cost , the customer is naturally attracted towards the most appealing product. The external appearance is an important feature, which gives grace to the product and dominates the markets. Selection of proper colour is an important consideration in product aesthetics. The choice of colour should be compatible with the conventional ideas of the operator. Following table gives the meaning of the common colour.	03

Colour	Meaning
Red	Danger-Hazard-hot
Orange	Possible Danger
Yellow	Caution
Green	Safety
Blue	Caution
Grey	Dull

	(ii)	**Explain Maximum principal stress theory of failure.**	06
		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,	02

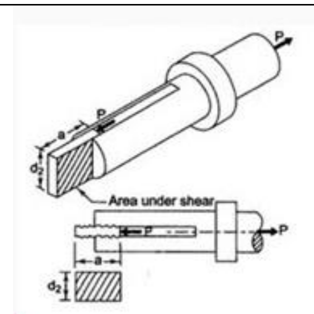
		$\sigma_{t1} = \frac{\sigma_{yt}}{F.S.}$, for ductile materials $= \frac{\sigma_u}{F.S.}$, for brittle materials σ_{yt} = Yield point stress in tension as determined from simple tension test, and σ_u = Ultimate stress. Brittle materials which are relatively strong in shear but weak in tension or compression, this theory are generally used.	02 2
2		**Attempt any FOUR of the following:**	**16**
	(a)	Explain design consideration in Automobile design.	**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. 8. Work shop facilities available. 9. Manufacturing cost. 10. Convenient of assembly and transportation	**04 (any 08)**
	(b)	Write any four strength equations in design of socket and spigot cotter joint with relevant sketches.	**04**
		Answer:(any four with sketches – 01 mark each) **Following are the strength equations in design of socket and spigot cotter joint:**	**04**

1. Failure of rod in Tension	
$P = \frac{\pi}{4} \times d^2 \times \sigma_t$	
2. Failure of Spigot in Tension across the weakest section	
$P = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t$	
3. Failute of rod or cotter in crushing	
$P = d_2 \times t \times \sigma_c$	

4. Failure of socket in tension across slot	
$P = \left\{\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\,t\right\}\sigma_t$	
5. Failure of cotter in shear	
$P = 2\,b \times t \times \tau$	
6. failure of socket collar in crushing	
$P = (d_4 - d_2)\,t \times \sigma_c$	
7.Failure of socket end in shear	
$P = 2\,(d_4 - d_2)\,c \times \tau$	

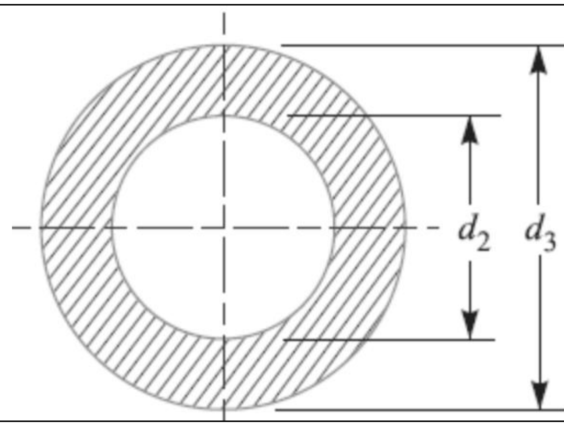
8. Failure of rod end in shear

$$P = 2\,a \times d_2 \times \tau$$



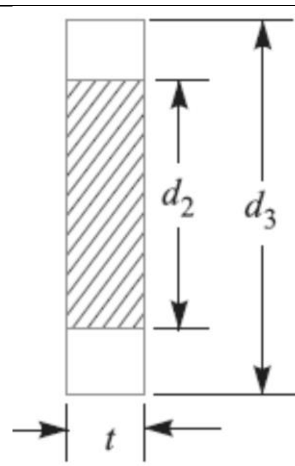
9. Failure of spigot collar in crushing

$$P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$$



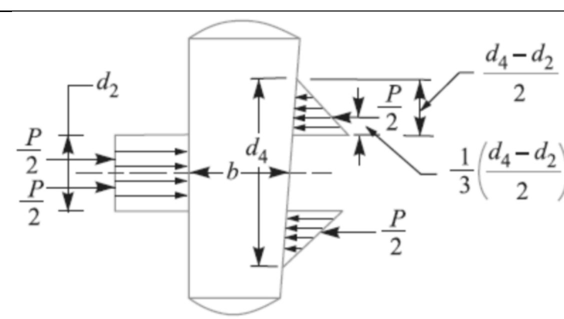
10. Failure of the spigot colllar in shearing

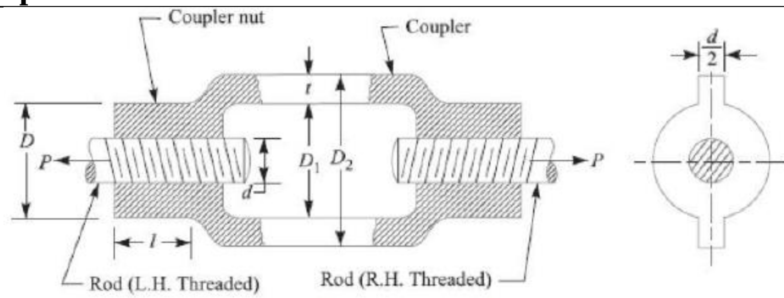
$$P = \pi\, d_2 \times t_1 \times \tau$$



11. Failure of cotter in bending

$$\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)}{t \times b^2/6} = \frac{P\,(d_4 + 0.5\,d_2)}{2\,t \times b^2}$$



	(c)	Explain design procedure of turn buckle.	04
		 **Fig. Turn Buckle Joint**	01
		Design procedure for Turn Buckle: 1. To design diameter of rod:- $$P_d = \frac{\pi}{4} d_c^2 \sigma_t$$ $$d = \frac{d_c}{0.84}$$	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:- $$\therefore P_d = \frac{\pi}{4} (D^2 - d^2)\sigma_t$$	01
		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	½ ½
	(d)	A knuckle joint is to withstand a load of 30 kN. Design the joint, if permissible stresses are, $\boldsymbol{\sigma_t}$ = 56 N/mm^2, $\boldsymbol{\tau}$= 35 N/mm^2 and $\boldsymbol{\sigma_c}$= 70 N/mm^2, assume suitable data.	**04**
		Answer: **Given Data:** P = 30 × 103 N, τ = 35 N/mm^2,σ_t= 56 N/mm^2,σ_c= 70 N/mm^2 **i. Find Diameter of rod:-** $$P = \frac{\pi}{4} d^2 . \sigma_t$$ $$30\ X 10^3 = \frac{\pi}{4} d^2 . 56$$ d= 26.11mm say 28 mm **ii. Find dimensions of fork end, eye end and knuckle pin by empirical relations:-** 1. Diameter of knuckle pin d1=d = 28 mm 2. Outer diameter of eye end d2=2d = 56 mm 3. Diameter of knuckle pin head or collar d3=1.5d = 42 mm 4. Thickness of eye end t=1.25d = 35 mm 5. Thickness of forked end t1=0.75d = 21 mm	 **01** **01**

		6. Thickness of collar or head t2=0.5d = 14 mm iii. **Induced stress in knuckle pin:-** $P = 2 X \frac{\pi}{4} d_1^2 . \tau$ $30 X10^3 = 2 X \frac{\pi}{4} 28^2 . \tau$ $\tau = 24.36 \frac{N}{mm^2} < 35 N/mm^2$ Therefore Design is safe. **iv. Induced stresses in eye end:-** **1. Failure in tension**: $\therefore P = (d2 - d1)t \times \sigma t$ $\therefore 30 \times 10^3 = (56 - 28) X 35 X \sigma t$ $\sigma t = 30.61 \frac{N}{mm^2} < 56 N/mm^2$ Therefore Design is safe. **2. Failure in shear:** $\therefore P = (d2 - d1)t \times \tau$ $\therefore 30 \times 10^3 = (56 - 28) X 35 X \tau$ $\tau = 30.61 \frac{N}{mm^2} < 35 N/mm^2$ Therefore Design is safe. **3. Failure in crushing**: $\therefore P = d1\ t \times \sigma c$ $\therefore 30 \times 10^3 = 28 \times 35 \times \sigma c$ $\sigma_c = 30.61 \frac{N}{mm^2} < 70 N/mm^2$ Therefore Design is safe. **Induced stresses in forked end:-** 1. Failure in tension: $\therefore P = 2 x (d2 - d1)t1 \times \sigma t$ $\therefore 30 \times 10^3 = 2 x (56 - 28) X 21 \times \sigma t$ $\sigma_t = 25.51 N/mm^2 < 56 N/mm^2$ Therefore Design is safe 2. Failure in shear: $\therefore P = 2(d2 - d1)t1 \times \tau$ $\therefore 30 \times 10^3 = 2(56 - 28) x 21 \times \tau$ $\tau = 25.51 N/mm^2 < 35 N/mm^2$ Therefore Design is safe 3. Failure in crushing: $\therefore P = 2(d2 - d1)t1 \times \sigma c$ $\therefore 30 \times 10^3 = 2 \times (56\text{-}28) x 21 \times \sigma c$ $\sigma_c = 25.51 N/mm^2 < 70 N/mm^2$ Therefore Design is safe	**01** **01**

	(e)	Describe the design procedure of a rear axle.	**04**

Answer: (Design Procedure 04 Marks)

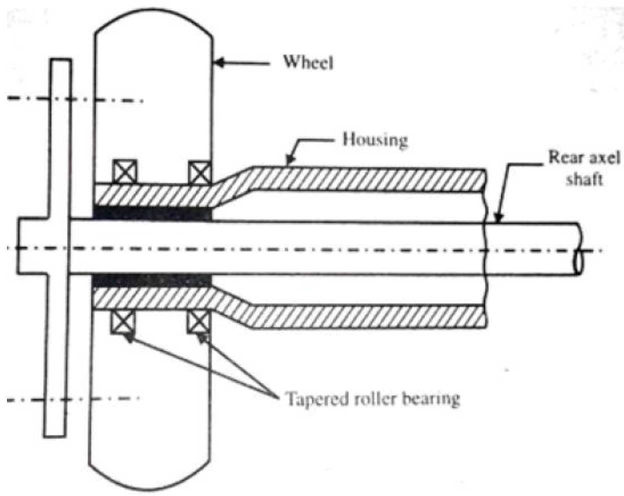


Figure: Fully Floating Rear Axle

Design procedure of a Fully Floating Rear Axle:

The rear axle is designed on the basis of shaft design.

By using the torsional equation,

$$\frac{T_{RA}}{J_{RA}} = \frac{\tau}{r}$$

TRA = Torque transmitted by rear axle shaft.

J_{RA} = Polar Moment of Inertia

$J_{RA} = \frac{\pi}{32} d^4$

$J_{RA} = \frac{\pi}{32}(d_o^4 - d_i^4)$

τ = Torsional shear stress r = distance from neutral axis to outer most fiber

r = $\frac{d}{2}$for Solid shaft)

r = $\frac{d_o}{2}$ (for Hollow shaft)

$$T_{RA} = T_e \times G_1 \times G_d$$

$$P = \frac{2\Pi N T_e}{60}$$

T_e = Engine Torque.

G_1 = Maximum gear Ratio in Gear Box

G_d = Final gear reduction in differential

After simplifying the equations,

$T_{RA} = \pi/16 \times \tau \times d^3$ (for Solid shaft)

$= \pi/16 \times \tau \times d_o^3 (1 - k^4)$ (for Hollow shaft)

$k = d_i/d_o$

d_i = Inner diameter of shaft

d_o = Outer diameter of shaft

From this equation we can find diameter of fully floating rear axle

04 Marks

3		**Attempt any FOUR of the following :**	**16**
	(a)	Write stepwise design procedure for a bushed pin flexible coupling.	**04**

Answer: *(Neat fig. – 1 Marks, Any three steps – 1 Mark each)*

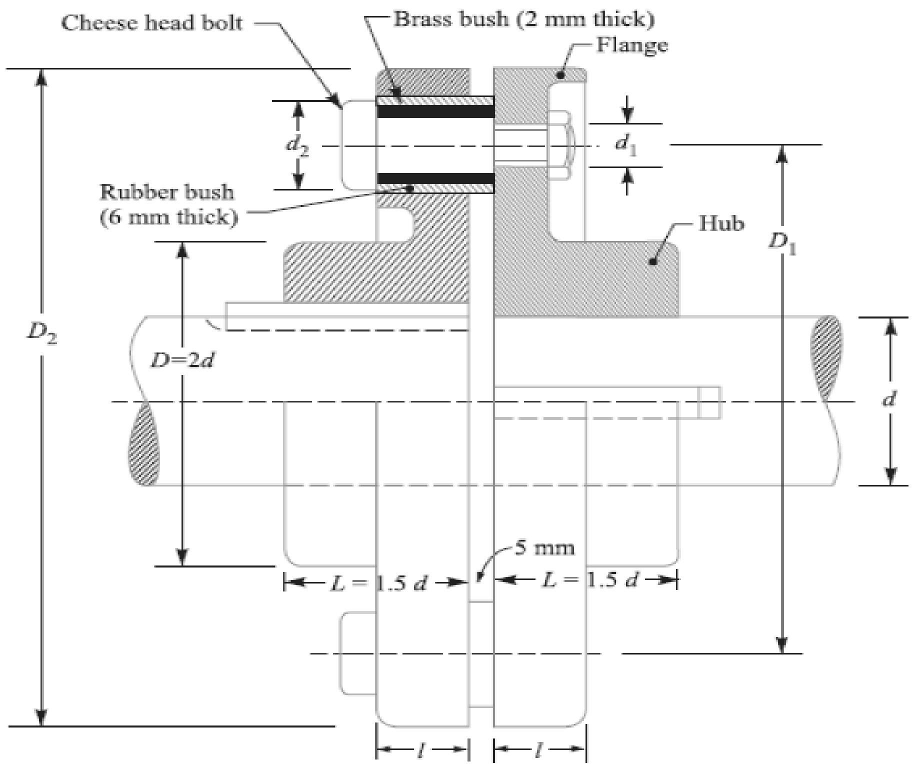


Figure. Bushed Pin Flexible Coupling

Design Procedure:

Design of Shaft:

$$P = 2\Pi NT/60$$

$$T = \Pi/16\ \tau\ d^3$$

Design of Pin:

Let l = Length of bush in the flange,

d_2 = Diameter of bush,

p_b = Bearing pressure on the bush or pin,

n = Number of pins, and

$n = d/25 + 3$

Diameter of pin $d_1 = 0.5d/\sqrt{n}$

Dia. of pin in rubber bush $d_3 = 1.5d_1$

$d_2 = d_1 + 6$ mm

D_1 = Diameter of pitch circle of the pins.

$= 3d$

04

We know that bearing load acting on each pin,

$$W = p_b \times d_2 \times l$$

∴ Total bearing load on the bush or pins

$$= W \times n = p_b \times d_2 \times l \times n$$

and the torque transmitted by the coupling,

$$T = W \times n\left(\frac{D_1}{2}\right) = p_b \times d_2 \times l \times n\left(\frac{D_1}{2}\right)$$

Direct shear stress due to pure torsion in the coupling halves,

$$\tau = \frac{W}{\frac{\pi}{4}(d_1)^2}$$

maximum bending moment on the pin,

$$M = W\left(\frac{l}{2} + 5 \text{ mm}\right)$$

We know that bending stress,

$$\sigma = \frac{M}{Z} = \frac{W\left(\frac{l}{2} + 5 \text{ mm}\right)}{\frac{\pi}{32}(d_1)^3}$$

Since the pin is subjected to bending and shear stresses, therefore the design must be checked either for the maximum principal stress or maximum shear stress by the following relations :

Maximum principal stress

$$= \frac{1}{2}\left[\sigma + \sqrt{\sigma^2 + 4\tau^2}\right]$$

and the maximum shear stress on the pin

$$= \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2}$$

Design of Hub:

The hub is designed by considering it as a hollow shaft, transmitting the same torque (T) as that of a solid shaft.

$$\therefore \quad T = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D}\right)$$

The outer diameter of hub is usually taken as twice the diameter of shaft. Therefore from the above relation, the induced shearing stress in the hub may be checked.

The length of hub (L) is taken as 1.5 d.

Design of Key:

For rectangular Key, w = d/4, t = d/6

For square key, w = d/4, t = d/4

$$T = l \times w \times \tau \times \frac{d}{2} \quad \text{... (Considering shearing of the key)}$$

$$= l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2} \quad \text{... (Considering crushing of the key)}$$

Design of flange:

The flange at the junction of the hub is under shear while transmitting the torque. Therefore, the troque transmitted,

T = Circumference of hub × Thickness of flange × Shear stress of flange × Radius of hub

$$= \pi D \times t_f \times \tau_c \times \frac{D}{2} = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

The thickness of flange is usually taken as half the diameter of shaft. Therefore from the above relation, the induced shearing stress in the flange may be checked.

	(b)	Design a propeller shaft to transmit 5 kW at 5000 rpm with gear box reduction 16 : 1. Assume permissible shear stress for shaft material is 45 N/mm^2.	**04**

Answer:

Given Data:

P= 5 ×10^{3}W,

N=5000rpm

G1=16:1,

fs =45 N/mm^2

Now torque produced by the engine,

$$P = \frac{2\,\pi N T_e}{60}$$

$$5\,X\,10^3 = \frac{2\pi\,x\,5000\,x\,T_e}{60}$$ **01**

Te = 9.549 N-m

Te = 9.549 X 10^{3}N-mm

Torque transmitted by the propeller shaft,

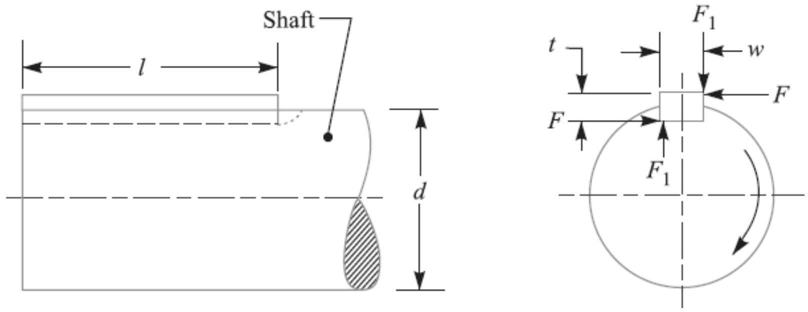
Tp=Te× G1

Tp=9.549×103× 16

		Tp=152.78×103 N-mm Diameter of propeller shaft, $$T_p = \frac{\pi}{16}\sigma_s d^3$$ $$152.78 \times 10^3 = \frac{\pi}{16} 45\, d^3$$ **d=25.86mm** **d= 26 mm**	**01** **02**
	(c)	State and explain the effect of keyways on 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 It is usually assumed that the strength of the keyed shaft is 75% of the solid shaft. In case the keyway is too long and the key is of sliding type, then the angle of twist is increased in the ratio K_θ as given by the following relation: $$k_\theta = 1 + 0.4\left(\frac{w}{d}\right) + 0.7\left(\frac{h}{d}\right)$$ k_θ = Reduction factor for angular twist.	**02** **01** **01**
	(d)	State the concept of whirling and critical speed of the shaft.	**04**
		Answer: **Whirling speed of shaft: -** The speed, at which the shaft rotates so that the deflection of the shaft from the axis of rotation becomes infinite, is known as whirling speed. **Critical speed of shaft : -** The speed at which the shaft tends to vibrate violently in transverse direction. **OR** The speed at which the shaft runs so that the additional deflection of shaft from the axis of rotation becomes infinite.	**02** **02**

	(e)	Write stepwise design procedure for sunk key.	04

Answer: (sketch-01 mark, any three points-01 mark each)



Let T = Torque transmitted by the shaft,
F = Tangential force acting at the circumference of the shaft,
d = Diameter of shaft,
l = Length of key,
w = Width of key.
t = Thickness of key, and
τ and σc = Shear and crushing stresses for the material of key.

Considering shearing of the key, the tangential shearing force acting at the circumference of the
shaft,
F = Area resisting shearing × Shear stress = $l \times w \times \tau$
∴ Torque transmitted by the shaft,

$$T = F \times \frac{d}{2} = l \times w \times \tau \times \frac{d}{2}$$

Considering crushing of the key, the tangential crushing force acting at the circumference of the
shaft,
F = Area resisting crushing × Crushing stress

$$= l \times \frac{t}{2} \times \sigma_c$$

04

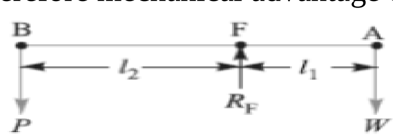
∴ Torque transmitted by the shaft,

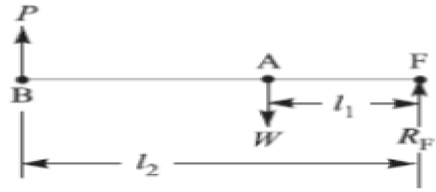
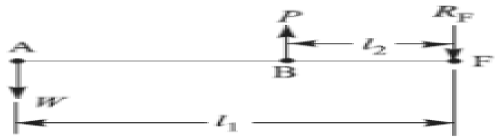
$$T = F \times \frac{d}{2} = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

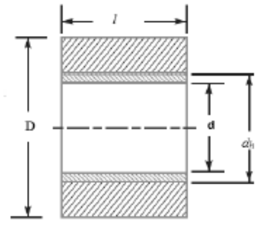
The key is equally strong in shearing and crushing, if

$$l \times w \times \tau \times \frac{d}{2} = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\frac{w}{t} = \frac{\sigma_c}{2\tau}$$

		In order to find the length of the key to transmit full power of the shaft, the shearing strength of the key is equal to the torsional shear strength of the shaft. We know that the shearing strength of key, $T = l \times w \times \tau \times \frac{d}{2}$ and torsional shear strength of the shaft, $T = \frac{\pi}{16} \times \tau_1 \times d^3$ **From above equations** $l \times w \times \tau \times \frac{d}{2} = \frac{\pi}{16} \times \tau_1 \times d^3$ $\therefore \quad l = \frac{\pi}{8} \times \frac{\tau_1 d^2}{w \times \tau} = \frac{\pi d}{2} \times \frac{\tau_1}{\tau} = 1.571\, d \times \frac{\tau_1}{\tau}$ ... (Taking $w = d/4$) When the key material is same as that of the shaft, then $\tau = \tau_1$. $\therefore \quad l = 1.571\, d$	
4	A	**(A) Attempt any THREE of the following :**	**12**
	(i)	Define a lever. Describe three basic types of lever.	**04**
		(Definition 01 Mark and Each Type of Lever 01 Mark) **Definition:-** A lever is a rigid rod or a bar capable of turning about a fixed point called fulcrum. **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.  **Figure: First Type Lever** **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.	**04 *(Definition 01 Mark and Each Type of***

		 **Figure: Second Type Lever** **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.  **Figure: Third Type Lever**	***Lever 01 Mark)***
	(ii)	Explain design procedure of rocker arm.	**04**
		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), $$\therefore \text{ Bearing pressure } = \frac{\text{Load}}{\text{Bearing area}} = \frac{R_F}{l \times d} = \frac{R_F}{1.25d \times d}$$ 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	**01** **01** **01**



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

Section Modulus $Z = 1/12 \times l \times (D^3 - d_h^3) / D/2$ **01**

Induced bending stress,

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

OR

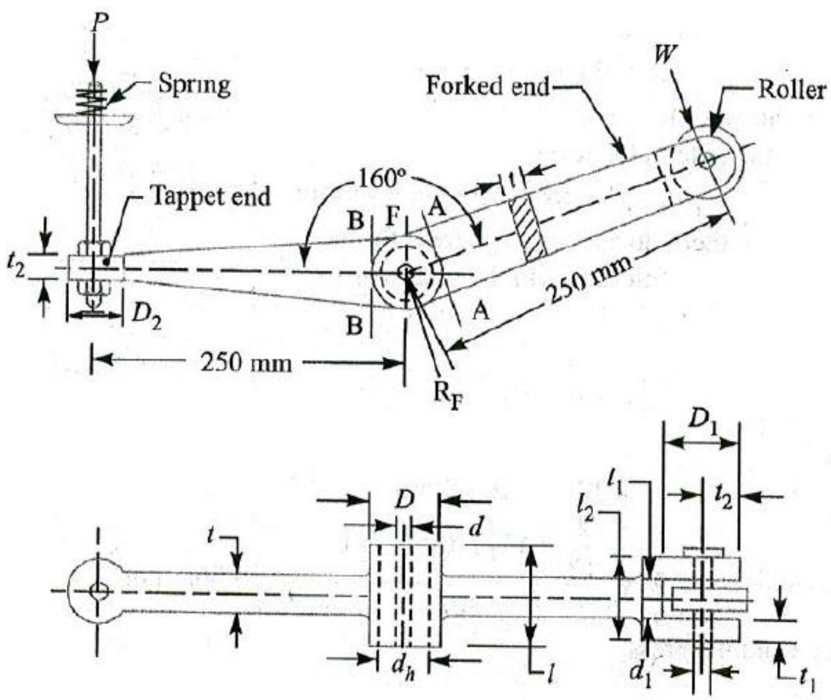
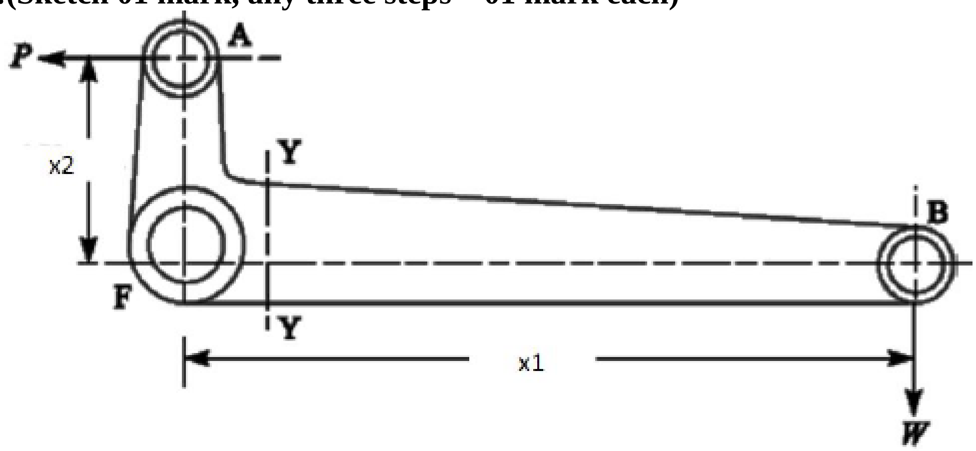


Figure. Rocker arm for operating exhaust valve

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.	
	(iii)	**Explain design procedure of bell crank lever.**	**04**
		Answer:(Sketch 01 mark, any three steps – 01 mark each)  Fig.bell crank lever	

		Step I Calculate the reaction at fulcrum P_F $W \times X_1 = P \times X_2$ obtaining value of P determine R_F $R_F = \sqrt{W^2 + P^2}$ **Step II** Design of Fulcrum Pin d= Dia of fulcrum pin l= Length of fulcrum pin we know that load on fulcrum pin A) $\therefore Pb = \frac{\text{Load}}{\text{Bearing Area}}$ $Pb = \text{Bearing Pressure in N/mm}^2$ $Pb = \frac{R_F}{l \times d}$ $R_F = Pb \times l \times d$	**04(Sketch 01 mark, any three steps – 01 mark each)**

Assuming $l = 1.25d$

here l and d can be determined .

B) Checking induced shear stress in pin

Pin is in double shear

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

c) brass bush in 3mm thickness is pressed in the

Dai. of hole in lever = $d + 2 \times 3$

Dai. of boss at fulcrum = 2d

Step III

Design of pin at A

Checking the effort at a the value of R_F

If it is same, take same dimension As fulcrum pin

d_1 = Dai of pin At 'A'

l_1 = Length of pin at A = $1.25d_1$

d_2 = Dai of pin At B

l_2 = length of pin At B

we know that

a) The load on pin At B

$$Pb = \frac{w}{d_2 \times l_2}$$

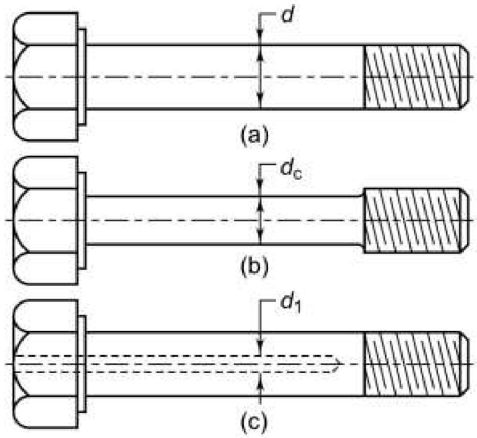
$$l_2 = 1.25d_2$$

Here d_2 and l_2 can be determined

Checking the pin for shearing

		$\tau = \frac{w}{2\times\left(\frac{\Pi}{4}\times d_2^2\right)}$ thick of each eye $= t_1 = \frac{l_2}{2}$ inner Dia of each eye = $d_2 + 2\times 3$ outer Dia of each eye = $D = 2d_2$ **Step IV** Design of lever t= thick of lever At section Y-Y b= width of lever At section Y-Y take distance a form centre of fulcrum Y-Y max. bending moment $y.y = w(x_1 - a)$ section modulus $\frac{1}{6}\times t\times b^2$ Assume $(b = 3t)$ $= \frac{1}{6}\times t\times(3t^2)$ $\sigma_b = \frac{m}{z} = \frac{w(x_1 - a)}{\frac{1}{6}\times t\times(3t^2)}$ here t and b can be determined.	
	(iv)	**Write factors to be consider while selecting material for piston or connecting rod or cylinder head.**	**04**
		Answer:(Credits to be given for appropriate answer) **Factors to be considered for selection of material for design of machine elements** a) Availability: Material should be available easily in the market. b) Cost: the material should be available at cheaper rate. c) Manufacturing Consideration: the manufacturing play a vital role in selection of	

		material and the material should suitable for required manufacturing process. d) Physical properties: like colour, density etc. f) Mechanical properties: such as strength, ductility, Malleability etc. g) Corrosion resistance: it should be corrosion resistant. **OR** 1. **Availability:** The material should be readily available in the market, in large enough quantities to meet the requirements. Cast iron &aluminum alloys are easily available in market. **2. Material Cost:** For every application there is a limiting cost beyond which designer can't afford. When this limit exceeded, the designer consider other alternative material. 3. **Mechanical properties:** It is a technical factor governing the selection of material. They include strength under fluctuating, static load, elasticity, stiffness, toughness, hardness. Depending upon the working conditions & requirements, the properties are considered and material is selected. Eg. Material for connecting rod should be capable to withstand fluctuating stress induced so here endurance limit becomes the selection criteria. 4. **Manufacturing considerations:** Machinability of material is an important considered in selection. When material is complex shaped, casting property is important. The manufacturing processes such as forging, casting, rolling, machining, extrusion etc. governs the selection of material. 5. **Manufacturing Cost:** It includes cost of processing the material into finished goods.	**04**
4	**B**	**Attempt any ONE of the following :**	**06**
	(i)	**Explain Bolts of uniform strength with sketch.**	
		Answer: In an ordinary bolt shown in **Fig. (a)**, the effect of the impulsive loads applied axially is concentrated on the weakest part of the bolt i.e. the cross-sectional area at the root of the threads. In other words, the stress in the threaded part of the bolt will be higher than that in the shank. Hence a great portion of the energy will be absorbed at the region of the threaded part which may fracture the threaded portion because of its small length. If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (d) 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. A second alternative method of obtaining the bolts of uniform strength is shown in **Fig. (c).** 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.	**03**

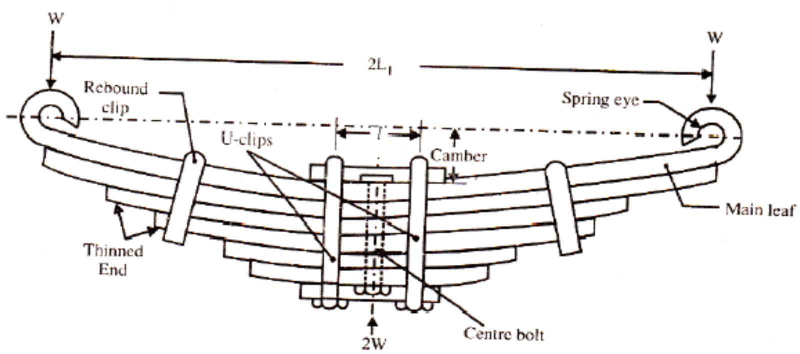


Bolts of Uniform Strength

03

(ii) Explain design procedure of semi-elliptical leaf spring. **06**

(Neat labelled Sketch 01 Mark and 01 Mark for each design step)



Neat labelled Sketch 01 Mark and 01 Mark for each design step

Let,
2W = Central Load
2L = Span of Spring
b = Width of Leaves
t = Thickness of Leaves
n = Total numbers of Leaves
l = Length of Central Band
n_f = Nos. of full length leaves
n_g = Nos. of Graduated Leaves

(1) Stress in the Leaf Spring:

$$\sigma_b = \frac{6WL}{n\, b\, t^2}$$

Where,
Effective Length of Spring = 2L = $2L_1 - l$……………….(**When Central band is used**)
= 2L = $2L_1 - \frac{2}{3}\, l$ …. . … ….(**When U - Bolt is used**)

(2) Deflection of Leaf Spring:

$$\delta = \frac{6WL^3}{n\, E\, b\, t^2}$$

(3) Stress in full Length Leaves:

$$\sigma_f = \frac{18WL}{b\,t^2(2n_g + 3n_f)}$$

(4) Stress in Graduated Leaves:

$$\sigma_g = \frac{12WL}{b\,t^2(2n_g + 3n_f)}$$

(5) Deflection in Full Length and Graduated Leaves:

$$\delta = \frac{12WL^3}{E\,b\,t^3(2n_g + 3n_f)}$$

Design steps for calculating length of Leaf Spring:

Length of Smallest Leaf = (L x 1)/ (n-1) + l
Length of second smallest leaf = (L x 2)/ (n-1) + l
Length of $(n-1)^{th}$ leaf = (L x (n-1))/ (n-1) + l
Length of master leaf = 2L1 + (π (d+t) x 2
Where d = diameter of Eye.

$$d = (32M/\pi\,\sigma_b)^{1/3}$$

5		**Attempt any TWO of the following :**	**16**
	(a)	**Draw the neat sketch of sliding mesh gear box and write the design procedure for teeth calculation.**	**08**

Answer: (Sketch – 2 marks, design procedure for teeth calculation-6 marks)

Fig: Four speed Sliding Mesh gear box:

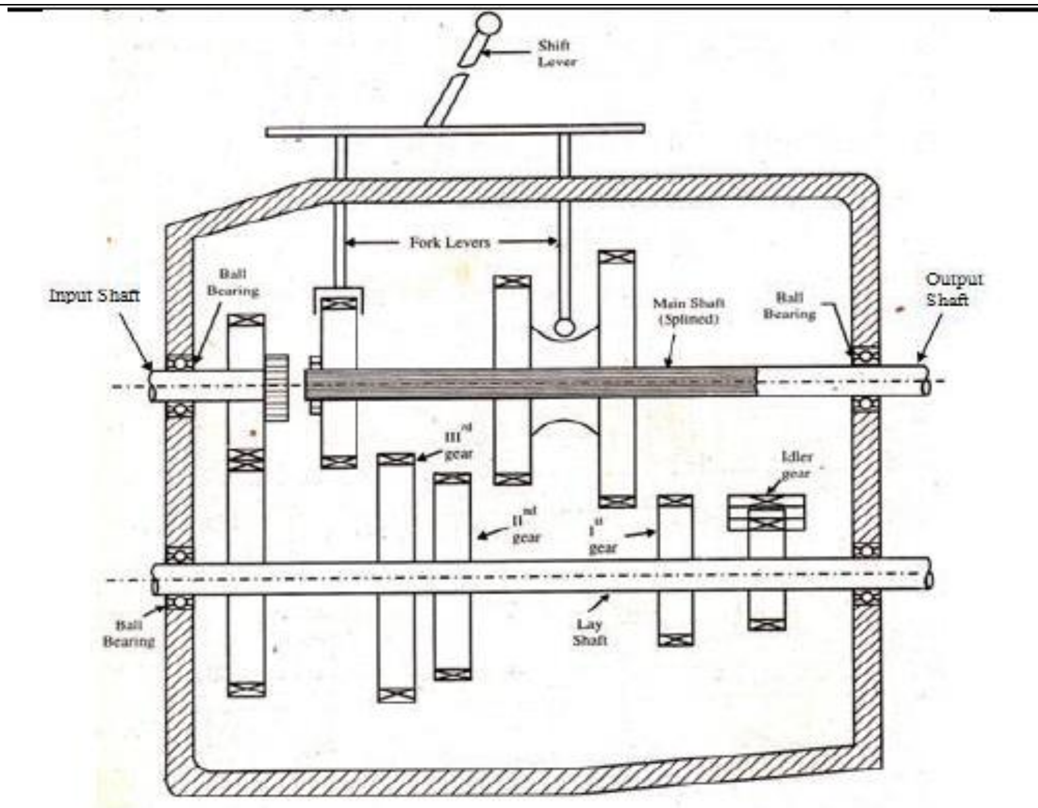
Design procedure for teeth calculation.

First gear ratio:

$$\therefore G_1 = \frac{T_b}{T_a} \times \frac{T_d}{T_c}$$ **01**

Second gear ratio:

$$\therefore G_2 = \frac{T_b}{T_a} \times \frac{T_f}{T_e}$$ **01**



02

01

Third Gear Ratio:

$$\therefore G_3 = \frac{T_b}{T_a} X \frac{T_h}{T_g}$$

And Fourth gear ratio: 01

$$\therefore G_4 = 1:1$$

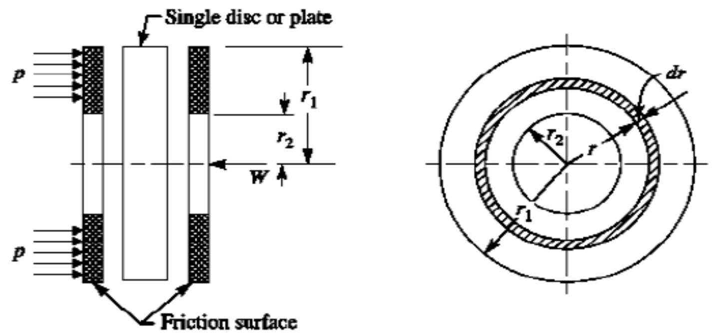
Reverse gear ratio: 02

$$\therefore G_r = \frac{T_b}{T_a} \times \frac{T_i}{T_g} \times \frac{T_r}{T_i}$$

	(b)	**Derive the relation for torque to be transmitted by single plate clutch considering uniform wear condition and uniform pressure condition.**	08

Answer:

Consider two friction surfaces maintained in contact by an axial thrust (W) as shown in Fig.



01

Let,

T = Torque transmitted by the clutch,

p = Intensity of axial pressure with which the contact surfaces are held together,

$r1$ and $r2$ = External and internal radii of friction faces,

r = Mean radius of the friction face, and

μ = Coefficient of friction.

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, 01

$$\delta W = \text{Pressure} \times \text{Area} = p \times 2\pi.\ r.dr$$

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

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

Therefore Frictional torque acting on the ring,

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

We shall now consider the following case :

Uniform Wear :

Let p be the normal intensity of pressure at a distance r from the axis of the clutch. Since the intensity of pressure varies inversely with the distance, therefore

$$p.r = C \text{ (a constant) or } p = C / r$$

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, 01

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

or

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

We know that the frictional torque acting on the ring,

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

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

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

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

$$= \pi\mu \times \frac{W}{2\pi\,(r_1 - r_2)}[(r_1)^2 - (r_2)^2] = \frac{1}{2} \times \mu.W\,(r_1 + r_2) = \mu.W.R \qquad \textbf{01}$$

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

Design procedure of single plate clutch using uniform pressure condition:-

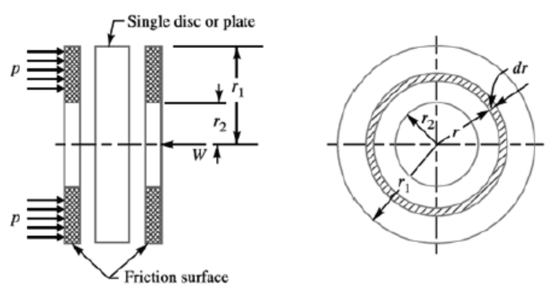


Fig. Forces on a single plate clutch **01**

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.

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 = ***2π.r.dr***

Therefore Normal or axial force on the ring,

$$\delta W = \text{Pressure} \times \text{Area} = p \times 2\pi.r.dr$$

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

$$Fr = \mu.\,\delta W = \mu.p \times 2\pi.r.dr \qquad \textbf{01}$$

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 pressure: When the pressure is uniformly distributed over the entire area of the friction face as shown in Fig. , then the intensity of pressure, $p = \frac{W}{\pi\left[(r_1)^2 - (r_2)^2\right]}$ Where, W = Axial thrust with which the friction surfaces are held together. We have discussed above that the frictional torque on the elementary ring of radius r and thickness dr is $T_r = 2\pi\,\mu.p.r^2.dr$ Integrating this equation within the limits from r_2 to r_1 for the total friction torque. Total frictional torque acting on the friction surface or on the clutch, $T = \int_{r_2}^{r_1} 2\pi\,\mu.p.r^2.dr = 2\pi\mu.p\left[\frac{r^3}{3}\right]_{r_2}^{r_1}$ $= 2\pi\,\mu.p\left[\frac{(r_1)^3 - (r_2)^3}{3}\right] = 2\pi\,\mu \times \frac{W}{\pi\left[(r_1)^2 - (r_2)^2\right]}\left[\frac{(r_1)^3 - (r_2)^3}{3}\right]$ ... (Substituting the value of p) $= \frac{2}{3}\mu.W\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right] = \mu.W.R$ where $R = \frac{2}{3}\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right]$ = Mean radius of the friction surface.	01 01
	(c)	**Write design procedure for connecting rod.**	08
		Answer: Design of 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 A = Cross-sectional area of the connecting rod=11 t^2 L =Effective length of the connecting rod, σ_c = Crippling or Buckling stress, W_B = Buckling load, a = Rankine's constant $k_{xx}^2 = 3.18\ t^2$ from this relation t (thickness of the flange and web of the section) can be determined.	01 01

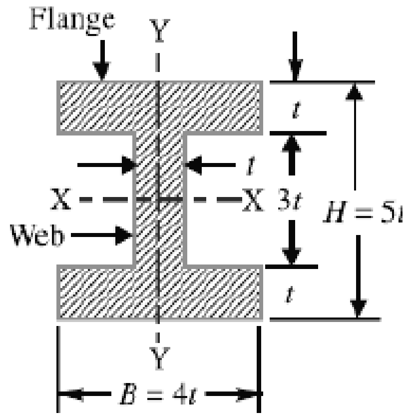


Fig a *I*-section of connecting rod.

01

Width of the section, $B = 4\,t$

and depth or height of the section, $H = 5t$

The dimensions $B = 4\,t$ and $H = 5\,t$, as obtained above by applying the Rankine's formula, are at the middle of the connecting rod.

The width of the section (B) is kept constant throughout the length of the connecting rod, but the depth or height varies. 01

The depth near the small end (or piston end) is taken as $H_1 = 0.75\,H$ to $0.9H$

The depth near the big end (or crank end) is taken $H_2 = 1.1H$ to $1.25H$.

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

Maximum gas force,

$$F_L = \frac{\pi D^2}{4} \times p \qquad \text{(i)}$$

01

where D = Cylinder bore or piston diameter in mm, and

p = Maximum gas pressure in N/mm^2

Let d_c = Diameter of the crank pin in mm,

l_c = Length of the crank pin in mm,

pb_c = Allowable bearing pressure in N/mm^2, and

d_p, l_p and pb_p = Corresponding values for the piston pin, load on the crank pin = Projected area × Bearing pressure

$= d_c \cdot l_c \cdot pb_c$ (ii)

Similarly, load on the piston pin $= d_p \cdot l_p \cdot pb_p$ *(iii)* 01

Equating equation **(*i*)** and **(*ii*)**, we have

$F_L = d_c \cdot l_c \cdot pb_c$

Taking $l_c = 1.25\ dc$ to $1.5\ d_c$, the value of d_c and l_c are determined from the above expression.

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

$F_L = d_p \cdot l_p \cdot pb_p$

Taking $l_p = 1.5\ d_p$ to $2\ d_p$, the value of d_p and l_p are determined from the above expression.

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

F_I = Inertia load acting on bolts

Let dc_b = Core diameter of the bolt in mm,

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

n_b = Number of bolts. Generally two bolts are used.

Force on the bolts

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

From this expression, *dcb* is obtained. The nominal or major diameter (d_b) of the bolt is given by 01

$$d_b = \frac{d_{cb}}{0.84}$$

4. *Thickness of the big end cap*

The thickness of the big end cap (t_c) may be determined as below,

Maximum bending moment acting on the cap will be taken as

$$M_C = \frac{F_I \times x}{6}$$

where ,

x = Distance between the bolt centres.

= Dia. of crankpin or big end bearing (d_c) + 2 × Thickness of bearing liner (3 mm) + Clearance(3mm)

Let b_c = Width of the cap in mm. It is equal to the length of the crankpin or big end bearing (l_c), and

σ_b = Allowable bending stress for the material of the cap in MPa.

Section modulus for the cap,

$$Z_C = \frac{b_c (t_c)^2}{6}$$

$\therefore$ Bending stress, $\sigma_b = \dfrac{M_C}{Z_C} = \dfrac{F_I \times x}{6} \times \dfrac{6}{b_c (t_c)^2} = \dfrac{F_I \times x}{b_c (t_c)^2}$ 01

From this expression, the value of t_c is obtained.

6		Attempt any TWO of the following :	16
	(a)	**(a) Describe in detail design procedure to design** (i) **Thickness of cylinder head.** (ii) **Cylinder head bolts or studs.**	08

Answer:

i)Design of cylinder head Thickness:

The cylinder head is designed by considering it a flat circular plate. The thickness is determined by following relation.

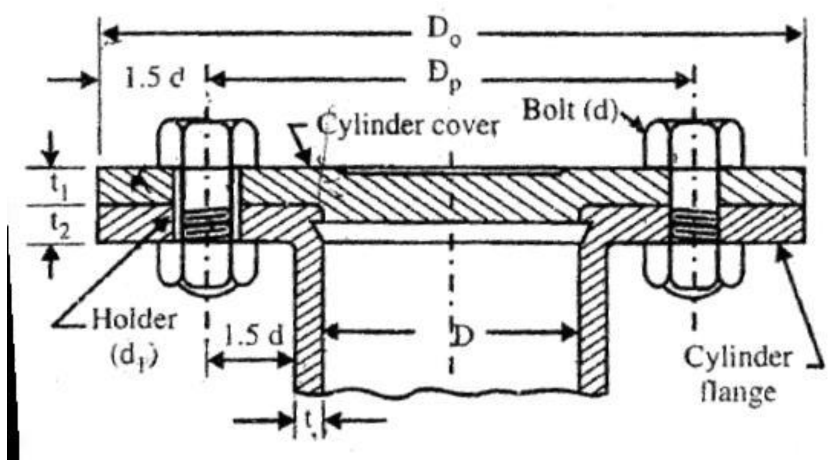
$$t = D\sqrt{\frac{C - P_{max}}{\sigma_c}}$$ **02**

t = thickness of cylinder head
D = diameter of cylinder
C = constant
= 0.1………for C.I.
P_{max} = maximum gas pressure inside the cylinder

σ_c = Allowable circumferential stress in MPa or N/mm^2. It may be taken as 30 to 50 MPa

The studs or bolts are screwed up tightly along with a metal gasket or asbestos packing to provide a leak proof joint between the cylinder and cylinder head. The tightness of the joint also depends upon the pitch of the bolts or studs which should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$ the pitch circle diameter (D_p) is usually taken as D+3d. **02**

ii)Design of cylinder head bolts or studs :



01

		a) The centre of stud is assumed at a distance of 1.25 to 1.5 d from inner wall of the cylinder where 'd' is diameter of bolt (let us assume 1.5d) $D_p = D + 2\times1.5d$ $= D + 3d$ ....(i)	½
		b) The gas pressure is assumed to be acting on P.C.D. of studs. $\therefore$ Gas load $= P_{max}\times\left(\frac{\Pi}{4}D_p^2\right)$ $P_{max}\times\frac{\Pi}{4}(D+3d)^2$ ....(ii)	½
		c) This load is acting as tensile load on bolts or stud and this load is resisted by 'Z' numbers of bolts. $P_{max}\times\frac{\Pi}{4}(D+3d)^2 = Z\times\frac{\Pi}{4}d_c^2\times f$ ....(iii)	½
		d) Numbers of bolts 'Z' is taken between $Z = \left(\frac{D}{100}+4\right)$ to $\left(\frac{D}{50}+4\right)$ ....(iv) Generally even value is selected for 'Z' e) Value of 'd' is taken as $d = \frac{d_c}{0.84}$ ....(v) f) Putting value from (iv) in equitation (iii) values of d, d_c and Z are calculated g) For a leak proof joint, value of 'd' greater than 16 should be used. h) The circular pitch of stud is calculated as Pitch 'p' $= \frac{\Pi D_p}{Z}$ For a leak proof joint m inimum value of 'P' should be 3 d and maximum value of 'P' line between $19\sqrt{d}$ to $28\sqrt{d}$. If value of P is coming less decrease value of 'Z' and recalculate. If value of P is coming more increase value of 'Z' till condition is satisfied.	½ 01

	(b)	**Design the piston pin with following data:** **Maximum pressure on the piston is 4 N/mm^2, diameter of piston 70 mm, Allowable stresses, due to bearing is 30 N/mm^2, bending 80 N/mm^2, and shear stress 60 N/mm2. Assume suitable data.**	08

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 = 80N/mm^2
Shearing stress = τ = 60 N/mm^2

Maximum gas load,

$$W = \frac{\pi}{4} D^2 \times P_{max}$$ 01

$$W = \frac{\pi}{4} \times 70^2 \times 4$$

$$W = 15.39 \times 10^3 \, N$$

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

$l_p = 0.45 \times D = 0.45 \times 70$ 01
$l_p = 31.5$ mm

$F = d_{po} \times l_p \times P_b$

$$d_{po} = \frac{15.3938 \times 10^3}{31.5 \times 30}$$ 01

$d_{po} = 16.29$mm
$\mathbf{d_{po} = 17mm}$

2. Designing the piston pin on the basis of bending.

'Bending moment 'M' is calculated as

$$M = F \times \frac{D}{8}$$

$$M = \frac{15.3938 \times 10^3 \times 70}{8}$$ 01

$M = 134.69 \times 10^3$ N-mm

We know that,

$$M = \frac{\pi}{32} \times \sigma_b \times (d_{po})^3$$

$$134.69 \times 10^3 = \frac{\pi}{32} \times \sigma_b \times (17)^3$$

$$\boldsymbol{\sigma_b = 279.2589\ N/mm^2}$$

The induced bending stresses are greater than permissible bending stress 80N/mm2 hence redesign is necessary. Now redesign value of d_{po} 01

$$M = \frac{\pi}{32} \times \sigma_b \times (d_{po})^3$$

$$134.69 \times 10^3 = \frac{\pi}{32} \times 80 \times (d_{po})^3$$

$$d_{po} = 25.79\ mm$$

$$\boldsymbol{d_{po} = 26\ mm}$$ 01

c) Designing piston pin on the basis of shear stress, due to double shear.

$F = 2x\pi/4(Dpo)^2 x\ \tau$

$15.39 \times 10^3 = 2x\ \pi/4 \times 26^2 x\ \tau$ 01

$T = 14.49\ N/mm^2$

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.

$$F = \frac{2\pi}{4} \times (d_{po})^2 \times \tau$$

$$15.39 \times 10^3 = \frac{2\pi}{4} \times (26)^2 \times \tau$$ ½

$$\boldsymbol{\tau = 14.49\ N/mm^2}$$

The induced shear stresses are less than permissible shear stress. Hence Design is safe ½

4. The total length of piston is taken as

$L_{pt} = 0.9\ D = 0.9 \times 70 =$ **63 mm**

	(c)	**(i) Estimate length of piston & (ii) Draw thrust and non-thrust sides of I.C. Engine piston.**	**08**

Answer:

(i) Estimate length of piston

The length of piston can be estimated by using following equations:

We know that maximum gas load on the piston,

$$P = p \times \frac{\pi D^2}{4}$$ **01**

$\therefore$ Maximum side thrust on the cylinder,

$$R = P/10 = 0.1\, p \times \frac{\pi D^2}{4} \qquad ...(i)$$ **01**

where p = Maximum gas pressure in N/mm^2, and

D = Cylinder bore in mm.

The side thrust (R) is also given by

R = Bearing pressure × Projected bearing area of the piston skirt

$= p_b \times D \times l$

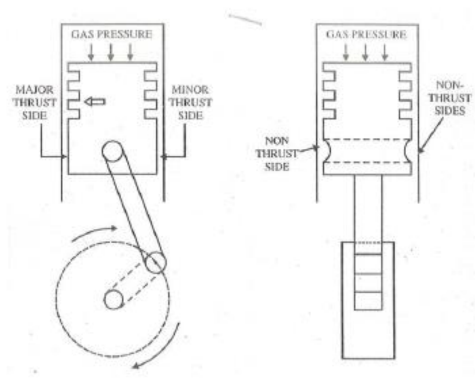
where l = Length of the piston skirt in mm. ...(ii) **01**

From equations (i) and (ii), the length of the piston skirt (l) is determined. In actual practice, the length of the piston skirt is taken as 0.65 to 0.8 times the cylinder bore. Now the total length of the piston (L) is given by

L = Length of skirt + Length of ring section + Top land **01**

The length of the piston usually varies between D and 1.5 D. It may be noted that a longer piston provides better bearing surface for quiet running of the engine, but it should not be made unnecessarily long as it will increase its own mass and thus the inertia forces.

(ii) thrust and non-thrust sides of I.C. Engine piston. **04**



Thrust side of I.C. Engine piston Non-thrust side of I.C. Engine piston