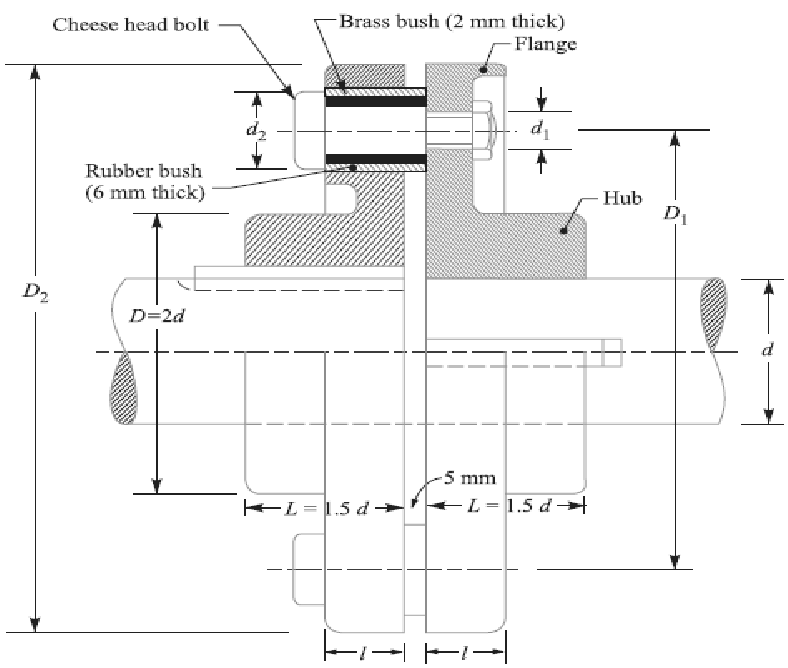
Important Instructions to examiners:

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

Q. No.	Sub Q. N.	Answer	Marking Scheme
1	(A)	Attempt **any three:**	12
	a)	Explain design considerations 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
	b)	Explain Ergonomic aspects of machine design.	04
		Answer: **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	01 01

		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.	**01** **01**
	c)	Write two application of each of the following: i) Turn buckle ii) Knuckle joint	**04**
		Answer: **i)** **Turn Buckle:** (*Any two applications – 1 mark each*) 1. Tie rod of steering system 2. To connect compartments of locomotives 3. Tie strings of electric poles 4. link rod of leaf springs in multi axle vehicles 5. linkages of gear shifter 6. Connection between brake pedal and master cylinder **ii)** **Knuckle joint:** (*Any two applications – 1 mark each*) 1. It is used in link of cycle chain 2. It is used in tie rod joints for roof truss 3. It is used in valve rod joint for electric rod 4. It is used in pump rod joint 5. It is used in tension link in bridge structure 6. It is used in lever and rod connection of various types	**02** **02**
	d)	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.	**02**

		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
	(B)	Attempt **any one:**	06
	a)	Write stepwise design procedure for a bushed pin flexible coupling.	06
		Answer: *(Neat fig. – 2 Marks, Any four steps – 1 Mark each)*  **Figure. Bushed Pin Flexible Coupling**	02
		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,

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

$\therefore$ 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

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

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

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 knuckle joint (i) fork end (ii) Eye end (iii) Knuckle pin where the tensile force 40×10^3N is acting. The safe stress in the parts are shear stress= 60 N/mm^2, Tensile stress= 80N/mm^2, and crushing stress= 40 N/mm^2. **06**

Answer: Given Data:

$$\mathbf{P = 40 \times 10^3\ N}$$
$$\boldsymbol{\sigma_s = 60\ N/mm^2}$$
$$\boldsymbol{\sigma_t = 80\ N/mm^2}$$
$$\boldsymbol{\sigma_c = 40\ N/mm^2}$$

01

i. Find Diameter of rod:-

$$P = \frac{\pi}{4} d^2 \sigma_t$$
$$40 \times 10^3 = \frac{\pi}{4} d^2 \times 80$$
$$d = 25.23 \text{ mm}$$
$$\mathbf{d \approx 26\ mm}$$

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ii. Find dimensions of fork end, eye end and knuckle pin by empirical relations:-

1. Diameter of knuckle pin — d_1=d = 26 mm
2. Outer diameter of eye end — d_2=2d = 52 mm
3. Diameter of knuckle pin head or collar — d_3=1.5d = 39 mm
4. Thickness of eye end — t=1.25d = 32.5 mm
5. Thickness of forked end — t_1=0.75d = 19.5 mm
6. Thickness of collar or head — t_2=0.5d = 13 mm

iii. Induced stress in knuckle pin:-

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$$\therefore P = 2 \times \frac{\pi}{4} d_1^2 \times \sigma_s$$
$$\therefore 40 \times 10^3 = 2 \times \frac{\pi}{4} 26^2 \times \sigma_s$$
$$\therefore \sigma_s = 37.68 \frac{N}{mm^2} < \mathit{Permissible\ shear\ stress}$$

Therefore Design is safe.

iv. Induced stresses in eye end:- ½

1. Failure in tension:

$$\therefore P = (d_2 - d_1) t \times \sigma_t$$
$$\therefore 40 \times 10^3 = (52 - 26) 32.5 \times \sigma_t$$
$$\therefore \sigma_t = 47.33 \frac{N}{mm^2} < \mathit{Permissible\ tensile\ stress}$$

		Therefore Design is safe. 2. Failure in shear: $\therefore P = (d_2 - d_1)t \times \sigma_s$ $\therefore 40 \times 10^3 = (52 - 26)32.5 \times \sigma_s$ $\therefore \sigma_s = 47.33 \frac{N}{mm^2} < Permissible\ shear\ stress$ Therefore Design is safe. 3. Failure in crushing: $\therefore P = d_1\, t \times \sigma_c$ $\therefore 40 \times 10^3 = 26 \times 32.5 \times \sigma_c$ $\therefore \sigma_c = 47.33 \frac{N}{mm^2} > Permissible\ crushing\ stress$ Therefore Design is unsafe. Redesign it, $\therefore 40 \times 10^3 = 26 \times t \times 40$ **t= 39mm** **v. Induced stresses in forked end:-** 1. Failure in tension: $\therefore P = 2(d_2 - d_1)t_1 \times \sigma_t$ $\therefore 40 \times 10^3 = 2(52 - 26)19.5 \times \sigma_t$ $\therefore \sigma_t = 39.44 \frac{N}{mm^2} < Permissible\ tensile\ stress$ Therefore Design is safe. 2. Failure in shear: $\therefore P = 2(d_2 - d_1)t_1 \times \sigma_s$ $\therefore 40 \times 10^3 = 2(52 - 26)19.5 \times \sigma_s$ $\therefore \sigma_s = 39.44 \frac{N}{mm^2} < Permissible\ shear\ stress$ Therefore Design is safe. 3. Failure in crushing: $\therefore P = 2(d_2 - d_1)t_1 \times \sigma_c$ $\therefore 40 \times 10^3 = 2 \times 26 \times 19.5 \times \sigma_c$ $\therefore \sigma_c = 39.44 \frac{N}{mm^2} < Permissible\ crushing\ stress$ Therefore Design is safe.	½ ½ ½ ½ ½
2		Attempt **any four** of the following**:**	**16**
	a)	Derive the relation for torque to be transmitted by single plate clutch considering uniform wear conditions.	**04**
		Answer: **Design procedure of single plate clutch using wear condition:-**	**04**

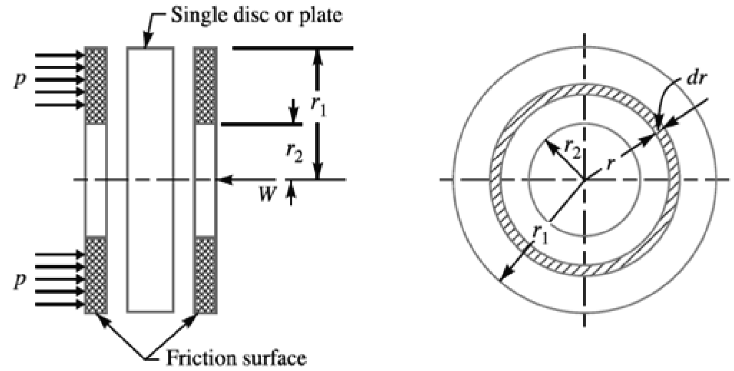


Fig. Forces on a single plate clutch

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

Let,

W = Axial force/thrust
T = Torque transmitted by the clutch,
p = Intensity of axial pressure
r_1 and r_2 = External and internal radii of friction faces,
r = Mean radius of the friction face, and
μ = Coefficient of friction.
b=face width of frictional surface.

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,

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

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

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

Therefore Frictional torque acting on the ring,

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

Considering uniform axial wear:

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

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

and the normal force on the ring,

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

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

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

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

We know that the frictional torque acting on the ring,

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

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

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

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

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

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

	b)	Define the terms: i) Fatigue and ii) Endurance limit with suitable example.	04

Answer:

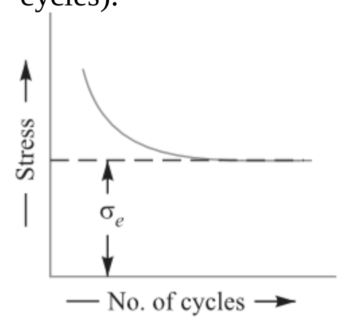
i) Fatigue:

When the system or element is subjected to fluctuating (repeated) loads, the material of system or element tends to fails below yield stresses by the formation of progressive crack this failure is called as fatigue. **02**

The failure may occur without prior indication. The fatigue of material is affected by the size of component, relative magnitude of static and fluctuating load and number of load reversals.

ii) Endurance limit:

It is defined as maximum value of the completely reversed bending stress which a polished standard specimen can withstand without failure, for infinite number of cycles (usually 10^7 cycles). **02**



Example. For most ferrous materials Endurance limit (Se) is set as the cyclic stress level that the material can sustain for 10 million cycles.

	c)	List No. of different factors to be consider to find design load and explain any two of them.	04

Answer:

List of Factors to be consider finding design load :*(List 2 marks)* **02**

1. **Service factor (SF)**
2. **Overload factor**
3. **Velocity Factor**
4. **Factor of Safety**

Explanation: (*Any two- 2marks*) **02**

1. Service factor (SF):

Service factor is described the service limit of the component for definite period of cycle. It is expressed usually a number greater than one: a SF of 1.15 means the item can take 15 percent more load than its rated capacity without breakdown.

T_{max}= Service factor × $T_{average}$

2. Overload factor:

The overload factor makes allowance for the externally applied loads which are in excess of the nominal tangential load. In determining the overload factor, consideration should be given to the fact that many prime movers and driven equipment, individually or in combination, develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment. There are many possible sources of overload which should be considered. Some of these are: system vibrations, acceleration torques, over speeds, variations in system operation, split path load sharing among multiple prime movers, and changes in process load conditions.

3. Velocity Factor:

This factor is considered in design of gears. It depends on velocity of operating gears. It is represented by $C_{v.}$

Permissible working stress is obtained as follows:

$\sigma_{wp} = \sigma_p \times C_v$

Where,

σ_{wp}= Permissible working stress
σ_p= Permissible stress
C_v= Velocity factor

4. Factor of Safety:

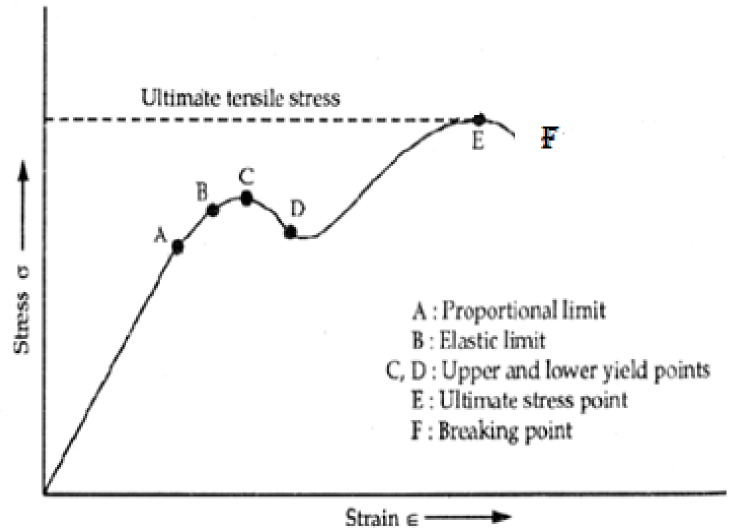
Factor of safety is defined as the ratio of the maximum stress (yield point stress for ductile material) to the working stress or design stress. In case of ductile materials-

$$\text{Factor of safety} = \frac{\text{Yield point stress}}{\text{Working or design stress}}$$

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

		Answer: Given Data: **$P = 5\times10^3$W, N=5000rpm** **G_1=16:1, σ_s=45 N/mm^2** Now torque produced by the engine, $P = \frac{2\pi N T_e}{60}$ $5\times10^3 = \frac{2\pi\times5000\times T_e}{60}$ $\boldsymbol{T_e = 9.549\text{Nm} = 9.549\times10^3\ \text{Nmm}}$ Torque transmitted by the propeller shaft, $T_p = T_e\times G_1$ $T_p = 9.549\times10^3\times16$ $\boldsymbol{T_p = 152.78\times10^3}$ **Nmm** Diameter of propeller shaft, $T_p = \frac{\pi}{16}\sigma_s d^3$ $152.78\times10^3 = \frac{\pi}{16}45\,d^3$ **d=25.86mm** **d= 26 mm**	**01** **01** **02**
	e)	Explain maximum principal stress theory of failure.	**04**
		Answer: **Statement:** According to this theory, the failure occurs at a point in a member when the maximum normal stress in a bi-axial stress system reaches the limiting strength of the material in a simple tension test. The maximum or normal stress in a bi-axial stress system is given by, $\sigma_{t1} = \frac{\sigma_{yt}}{F.S.}$, for ductile materials $= \frac{\sigma_u}{F.S.}$, for brittle materials σ_{yt} = Yield point stress in tension as determined from simple tension test, and σ_u = Ultimate stress. **Brittle material** which are relatively strong in shear but weak in tension or	**02** **01**

		compression, this theory are generally used.	01
3		Attempt **any four** of the following**:**	16
	a)	Describe the design procedure of a rear axle.	04
		Ans: **Design procedure of a fully floating rear axle:** The rear axle is designed on the basis of shaft design. By using the torsional equation, $$\frac{T_{RA}}{J_{RA}} = \frac{\sigma_s}{r}$$ *Where,* T_{RA} = Torque transmitted by rear axle shaft. $T_{RA} = T_e \times G_1 \times G_d$ T_e = Engine Torque. G_1 = Maximum gear Ratio in Gear Box G_d = Final gear reduction in differential J_{RA} = Polar moment of inertia. $= \pi/32 \times d^4$ ……….. …… (for Solid shaft) $= \frac{\pi}{32}(d_o^4 - d_i^4)$………..(for Hollow shaft) σ_s = Torsional shear stress. r = distance from neutral axis to outer most fiber. r = d/2 (for Solid shaft) r = d_o/2 (for Hollow shaft) After simplifying the equations, $$T_{RA} = \frac{\pi}{16}\,\sigma_s\,d^3$$ …………………For solid shaft $$T_{RA} = \frac{\pi}{16}\,\sigma_s\,d_o^3\,(1-k^4)$$……….. For hollow shaft $$k = \frac{d_i}{d_o}$$ d_i = Inner diameter of shaft d_o = Outer diameter of shaft From these equations, we can find out the diameter of rear axle of shaft.	04
	b)	Draw and explain the stress strain diagram for ductile material.	04
		Answer: *(Figure- 2mark and explanation-2 mark)* Stress strain curve for ductile material has different regions and points. i. Proportional limit ii. Elastic limit iii. Yield point iv. Ultimate stress point v. Fracture or breaking point.	



Stress –Strain Diagram

02

i. **Proportional Limit(A):** It is the region in the strain curve which obeys Hooke's law i.e. within elastic limit the stress is directly proportion to the strain produced in the material. In this limit the ratio of stress with strain gives us proportionality constant known as young's modulus. The point O**A** in the graph is called the proportional limit.

ii. **Elastic Limit (B):** It is the point in the graph up to which the material returns to its original position when the load acting on it is completely removed. Beyond this limit the material cannot return to its original position and a plastic deformation starts to appear in it. The point **B** is the Elastic limit in the graph.

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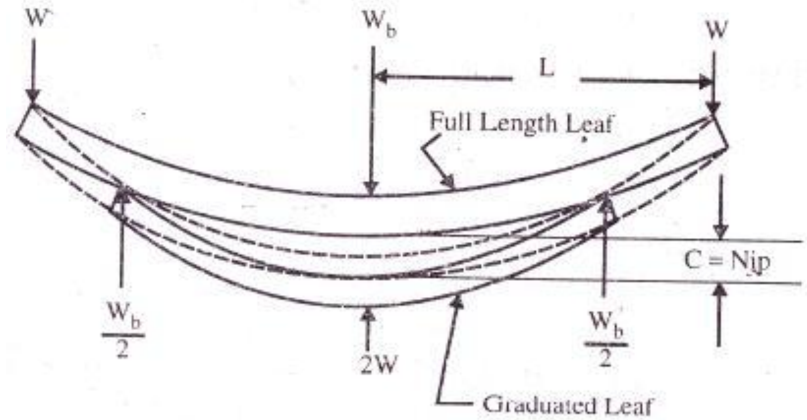
iii. **Yield Point or Yield Stress Point (C, D):** Yield point in a stress strain diagram is defined as the point at which the material starts to deform plastically. After the yield point is passed there is permanent deformation develops in the material and which is not reversible. There are two yield points and it is upper yield point and lower yield point. The stress corresponding to the yield point is called yield point stress. The point **C** is the upper yield stress point and **D** is the lower yield stress point.

iv. **Ultimate Stress Point (E):** It is the point corresponding to the maximum stress that a material can handle before failure. It is the maximum strength point of the material that can handle the maximum load. Beyond this point the failure takes place. Point **E** in the graph is the ultimate stress point.

v. **Fracture or Breaking Point (F):** It is the point in the stress strain curve at which the failure of the material takes place. The fracture or breaking of material takes place at this point. The point **F** is the breaking point in the graph.

	c)	Describe the Nipping of leaf springs with neat sketch.	**04**
		Answer: *(Figure- 2mark and explanation-2 mark)* **Nipping:**	



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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’.

It is seen that, stress in full length leaves is 50% greater than the stress in graduated leaves. In order to make best use of material; it is necessary that all the leaves must be equally stressed.

This can be achieved by in following two ways:

i) By making full length leaves of smaller thickness than graduated leaves. In this way the full length leaves will induce a smaller bending stress due to small distance from neutral axis to edge of the leaf.

ii) By giving a greater radius of curvature to the full length leaves than graduated leaves before leaves are assembled to form a spring.

By doing so, gap or clearance will be left between the leaves.

02

d) Design rectangular key for a shaft of 50mm diameter. The allowable shearing and crushing stresses for key material are 42 N/mm^2 and 70 N/mm^2 respectively. For shaft to resist torque 5000Nm. **04**

Answer:

Given data:

$d = 50mm$

$\sigma_{sk} = 42\ N/mm^2$

$\sigma_{ck} = 70\ N/mm^2$

$T = 5000\ Nm = 5 \times 10^6\ Nmm$

i) Length of key:

$l = 1.57\ d$

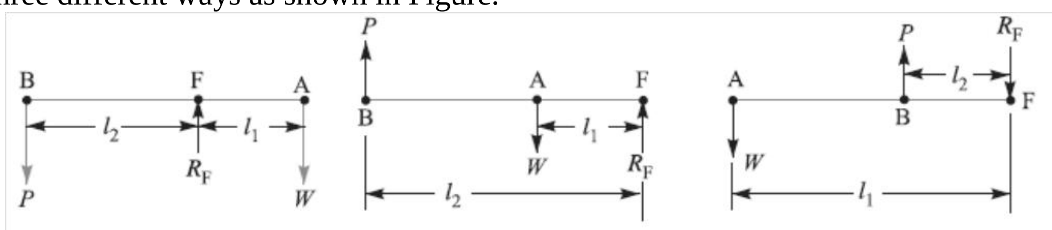
$l = 1.57 \times 50$

l = 78.5 =□ 79mm **02**

ii) Width of key by considering failure in shear:

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

		$5 \times 10^6 = 79 \times w \times 42 \times \frac{50}{2}$ $\therefore w = 60.27\ mm$ $\therefore w \cong 61\ mm$ iii) Thickness of key by considering failure in crushing: $T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ $5 \times 10^6 = 79 \times \frac{t}{2} \times 70 \times \frac{50}{2}$ $\therefore t = 72.33\ mm$ $\therefore t \cong 73\ mm$	01 01
	e)	Find the diameter of a solid shaft to transmit 20 kW at 200rpm. The ultimate shear stress for the shaft may be taken as 360 N/mm^2 and the factor of safety as 8.	04
		Answer: Given Data: **P= 20kW= 20×10^3W** **N= 200rpm** **σ_s=360/8 =45 N/mm^2** Now the torque transmitted by the engine T:- $P = \frac{2\ \pi NT}{60}$ $20\ \times 10^3 = \frac{2\ \times 3.14\ \times 200 \times T}{60}$ $T = 955.41\text{Nm} = 955.41\ \times 10^3\ \text{Nmm}$ Let, d = diameter of rear axle, $T = \frac{\pi}{16}\ f_s\ d^3$ $955.41 \times 10^3 = \frac{\pi}{16} \times 45\ \times\ d^3$ $d^3 = 108130.74$ $d = 47.64\ mm\ \cong 48\ mm$	 02 02
4	(A)	Attempt **any four** of the following**:**	12
	a)	Define standardization and state the four advantages of it.	04
		Answer: *(Defination- 2mark, Advantages-2 mark)* **Standardization: -** It is defined as obligatory norms to which various characteristics	

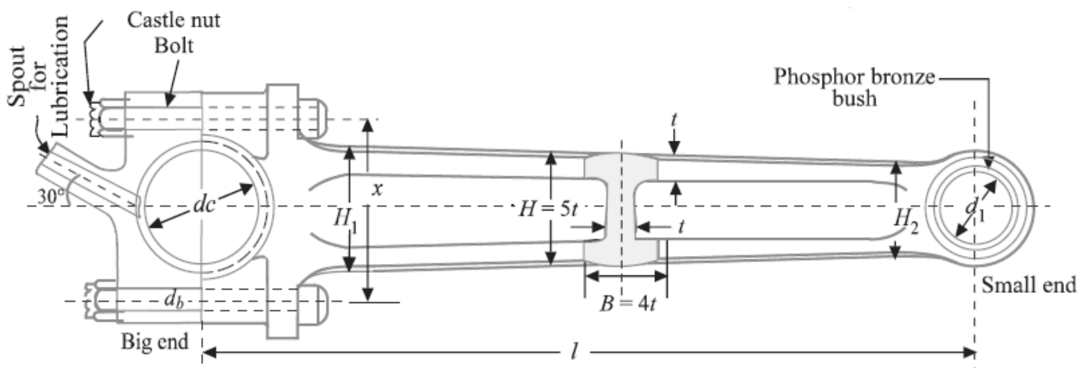
		of a product should conform. The characteristics include materials, dimensions and shape of the component, method of testing and method of marking, packing and storing of the product.	**02**
		Advantages of Standardization:- *(Any four)* 1. Mass production is easy. 2. Rate of production increases. 3. 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.	**02**
	b)	Define a lever. Describe three basic types of lever.	**04**
		Answer: *(Defination-1 mark, Figure-1 mark, explanation with example-2 mark)* **Def^n:-** A lever is a rigid rod or a bar capable of turning about a fixed point called fulcrum.	**01**
		Types of leaver: First type, second type and third type levers shown in figure at (a), (b) and (c) respectively. The load W and the effort P may be applied to the lever in three different ways as shown in Figure.  (*a*) First type of lever. (*b*) Second type of lever. (*c*) Third type of lever.	**01**
		First type lever: In the first type of levers, the fulcrum is in between the load and effort. In this case, the effort arm is greater than load arm; therefore mechanical advantage obtained is more than one. **Examples:** Such type of levers are commonly found in bell cranked levers used in railway signaling arrangement, rocker arm in internal combustion engines, handle of a hand pump, hand wheel of a punching press, beam of a balance, foot lever etc. **Second type lever:** In the second type of levers, the load is in between the fulcrum and effort. In this case, the effort arm is more than load arm; therefore the mechanical advantage is more than one. **Examples:** It is found in levers of loaded safety valves. **Third type lever:** In the third type of levers, the effort is in between the fulcrum and load. Since the effort arm, in this case, is less than the load arm, therefore the mechanical advantage is less than one. **Examples:** The use of such type of levers is not recommended in engineering practice. However a pair of tongs, the treadle of a sewing machine etc. are examples of this type of lever.	**02**
	c)	A single plate clutch with both sides effective has outer and inner diameter 300mm and 200mm respectively. The maximum intensity of pressure at any point in the contact surface is not exceed 0.1 N/mm^2. If the coefficient of friction is 0.3, determine the power transmitted by clutch at a speed of 2500 r.p.m.	**04**

		Answer: Given Data: $d_1 = 300mm$, $r_1 = d1/2 = 150mm$ $d_2 = 200mm$, $r_2 = d2/2 = 100mm$ $P_{max} = 0.1\ N/mm^2$ $\mu = 0.3$ N = 2500 rpm Since the intensity of pressure is maximum at inner radius, therefore, for uniform wear, $\mathbf{P_{max} \times r_2 = c}$ $c = 0.1 \times 100$ **c = 10 N/mm** We know that, axial thrust, $\mathbf{W = 2\pi c\ (r_1 - r_2)}$ $W = 2\pi \times 10 \times (150\text{-}100)$ **W = 3142 N** And mean radius of friction, $\mathbf{R = (r_1 + r_2)/2}$ R = (150+100)/2 **R = 125 mm** We know that, torque transmitted, $\mathbf{T = n.\ \mu.\ W.\ R}$ $T = 2 \times 0.3 \times 3142 \times 125$ T = 235650 N-mm **T = 235.65 N-m** Power transmitted by clutch, $\mathbf{P = (2\pi\ N\ T)/60}$ $P = (2\times \pi \times 2500 \times 235.65)/60$ **P = 61693W** **P = 61.693kW**	 **01** **01** **01** **01**
	d)	Describe stepwise procedure for designing the piston crown of an engine for bending strength and thermal considerations.	**04**
		Answer: **Design of Piston Head or Crown:** 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 $$t_H = \sqrt{\frac{3p.D^2}{16\sigma_t}} \text{ (in mm)}$$ 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.	02
		II) On the basis of Heat Dissipation:- The thickness of the piston head should be such that the heat absorbed by the piston due combustion of fuel is quickly transferred to the cylinder walls. Treating the piston head as a flat circular plate, its thickness is given, $$t_H = \frac{H}{12.56k(T_C - T_E)} \text{ (in mm)}$$ where H = Heat flowing through the piston head in kJ/s or watts, k = Heat conductivity factor in W/m/°C. Its value is 46.6 W/m/°C for grey cast iron, 51.25 W/m/°C for steel and 174.75 W/m/°C for aluminium alloys. T_C = Temperture at the centre of the piston head in °C, and T_E = Temperature at the edges of the piston head in °C. The temperature difference ($T_C - T_E$) may be taken as 220°C for cast iron and 75°C for aluminium. The heat flowing through the positon head (H) may be determined by the following expression, *i.e.,* $$H = C \times HCV \times m \times B.P. \text{ (in kW)}$$ where C = Constant representing that portion of the heat supplied to the engine which is absorbed by the piston. Its value is usually taken as 0.05. HCV = Higher calorific value of the fuel in kJ/kg. It may be taken as 45×10^3 kJ/kg for diesel and 47×10^3 kJ/ kg for petrol, m = Mass of the fuel used in kg per brake power per second, and $B.P.$ = Brake power of the engine per cylinder	02
	(B)	Attempt **any one** of the following:	**06**
	a)	Explain design procedure of a connecting rod.	**06**

Answer: *(Figure-2mark and any two points of procedure-4 marks)*

Design of Connecting Rod:



02

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.

04

Width of the section, $B = 4t$ and
Depth or height of the section, $H = 5t$

The dimensions $B = 4t$ and $H = 5t$, 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 $H_1 = 0.75H$ 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,

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

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/mm2, and
d_p, l_p and pb_p = Corresponding values for the piston pin

load on the crank pin = Projected area × Bearing pressure
$= dc \cdot lc \cdot pbc$ (ii)
Similarly, load on the piston pin = $dp \cdot lp \cdot pbp$ *(iii)*
Equating equation **(*i*)** and **(*ii*)**,
we have,
$FL = dc \cdot lc \cdot pbc$
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,
$FL = dp \cdot lp \cdot pbp$
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

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

where , *x* = Distance between the bolt centres.
= 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,

		$Z_C = \frac{b_c (t_c)^2}{6}$ $\therefore$ Bending stress, $\sigma_b = \frac{M_C}{Z_C} = \frac{F_I \times x}{6} \times \frac{6}{b_c (t_c)^2} = \frac{F_I \times x}{b_c (t_c)^2}$ From this expression, the value of *tc* is obtained.	
	b)	Design fulcrum pin of rocker arm which carries load of 5000N and has equal lengths of load arm. The lengths of arms are 250mm. The angle between the arms is 160^0. The allowable bearing pressure is $7 N/mm^2$.	**04**
		Answer: **Given data**: $\theta = 160^0$ $P = 5000N = 5kN$ $L_1 = L_2 = 250mm$ $P_b = 7 N/mm^2$ **Two arms of rocker arm equal,** So, $L_1 = L_2$ $\therefore P = W = 5kN$ $\therefore$ **Reaction at fulcrum pin,** $R_f = \sqrt{P^2 + W^2 - 2PW\cos\theta}$ $= \sqrt{5^2 + 5^2 - (2 \times 5 \times 5 \times \cos 160)}$ $R_f = 9.848kN = 9.848 \times 10^3 N$ $\therefore$ **Diameter of fulcrum pin,** $R_f = d \times l \times P_b$ Where, $l = 1.25d$ $\therefore 9.848 \times 10^3 = d \times 1.25d \times 7$ $\therefore d = 33.54mm$ $\therefore d \cong 34mm$ $\therefore$ **Length of fulcrum pin**, $l = 1.25d = 1.25 \times 34 = 42.5mm.$ $\boldsymbol{l} \cong 43mm$	 **01** **01** **01** **01**

Q.5		**Attempt any two of the following:**	**16**
Q.5	a)	Design a socket and spigot type cotter joint which has to withstand a load of 20 X10^3N. Take safe tensile stress 56 N/mm^2, shear stress 40N/mm^2 and crushing stress 40 N/mm^2.	**8**
		Given Data: $P=20x10^3$ KN\` f_t=56N/mm^2 f_s=40N/mm^2 f_c=40N/mm^2 Let, d= diameter of rod d_1 = outer diameter of socket d_2= outer diameter of spigot d_3= diameter of spigot collar d_4= 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 t_1= thickness of spigot collar l= length of cotter 1. Find dia. Of rod "d" considering failure in tension of rod We know that, $P = \frac{\pi}{4}(d^2)\, f_t$ $\therefore \quad d^2 = \frac{4}{\pi} x P x \frac{1}{f_t}$ $\therefore \quad d^2 = \frac{4}{\pi}\ x\ \frac{20 x 10^3}{56}$	

		$\therefore \quad d = 21.32$ mm **d =22mm** 2. Find outside diameter of spigot "d_2" considering failure in tension We know that $\quad P=[\frac{\pi}{4}(d_2^2)- d_{2\,x\,t}]\ f_t$ $P=[\frac{\pi}{4}(d_2^2)- \frac{d_2^2}{4}]\ f_t$……………….$[t=\frac{d_2^2}{4}]$ $\therefore 20 \times 10^3 =[\frac{\pi}{4}(d_2^2)- \frac{d_2^2}{4}] \times 56$ $\frac{20 \times 10^3 \times 4}{56} = \pi \times d_2^2 - d_2^2$ $\frac{20 \times 10^3 \times 4}{56} = d_2^2[\pi -1]$ $d_2 = 25.82$mm **d_2= 26mm** 3. Check the rushing stress considering failure at cotter in crushing We Know that $P= [d_2 \times t]\ f_c$ $\therefore f_c = \frac{P}{d_2 \times t} \quad \therefore f_c=\frac{20 \times 10^3}{26 \times [\frac{26}{4}]}$ $\therefore f_c = 118.34$ N/mm^2 Permissible stress is less than induced stress , so Design is unsafe for safety redesign "d_2"and "t" We know that, $P = d_2 \times \frac{d_2}{4} \times f_c$ $20 \times 10^3 = \frac{d_2^2}{4} \times 40$ $\therefore d_2 = 44.72$ mm **d_2= 45mm**	1 1 1

		$t = \frac{d_2}{4}$ $\therefore t = 11.25mm$ **t= 11.50mm** 4. Find outside diameter of socket "d1" considering failure socket in tension We know that, $P = [\frac{\pi}{4}(d_1^2 - d_2^2) - [d_1\text{-}d_2] \times t] f_t$ $20 \times 10^3 = [\frac{\pi}{4}(d_1^2 - 45^2) - [d_1\text{-}45] \times 11.5] \times 56$ $20 \times 10^3 = [\frac{\pi}{4}d_1^2 - 1590.43 - 11.5d_1 + 517.5] \times 56$ $357.14 = 0.785d_1^2 - 11.5d_1 - 1072.93$ $0.785d_1^2 - 11.5d_1 - 1430.07 = 0$ $d_1^2 - 14.65d_1 - 1821.75 = 0$ In above equation $d_1 = \frac{-b \pm \sqrt{b^2-4ac}}{2a}$, (a= 1, b=-14.65, c= -1821.75) $d_1 = \frac{14.65 \pm \sqrt{(-14.65)^2 - 4(-1821.75)}}{2}$ $d_1 = 50.63mm$ **d_1 = 51 mm**	
			1
		5. Find the diameter of spigot collar considering failure in crushing We know that, $P = \frac{\pi}{4}[d_3^2 - d_2^2] f_c$ $\frac{20 \times 10^3}{40} = \frac{\pi}{4}[d_3^2 - 45^2]$ $d_3 = 51.59$ mm **d_3 = 52mm**	1
		6. Find diameter of socket collar considering failure in crushing We know that, $P = (d_4\text{-}d_2) \times t \times f_c$ $d_4 = 88.47mm$ **d_4 = 89mm**	1
		7. Find the width of cotter "b" considering failure in shear We know that, $P = 2 \times b \times t \times f_s$ $b = 21.74mm$	

		b= 22mm 8. Find the thickness of spigot collar “t_1” by considering failure in shear We know that, $P = \pi \times d_2 \times t_1 f_s$ $t_1 = 3.53mm$ **t_1=4mm** 9. Find the thickness of socket collar “c” by considering failure in shear We know that, $P = 2 \times (d_4 - d_2) \times c \times f_s$ c = 5.68mm **c= 6mm** 10. Find the distance from cotter slot to end of spigot rod “a” by considering failure in shear We know that, $P = 2 \times d_2 \times a \times f_s$ a = 5.55mm **a= 6mm** 11. Find the length of cotter, We know that, L= 4d **L=88mm** 12. Find the thickness of socket of neck”e” e = 1.2d e = 26.4mm **e = 27mm**	**1** **1**
Q.5	b)	Draw the neat sketch of sliding mesh gear box and write the design procedure for teeth calculation.	**8**
		Answer: (Sketch – 3 marks, Correct Labeling – 1 Mark, design procedure for teeth calculation-4 marks) Fig: Four speed Sliding Mesh gear box:	

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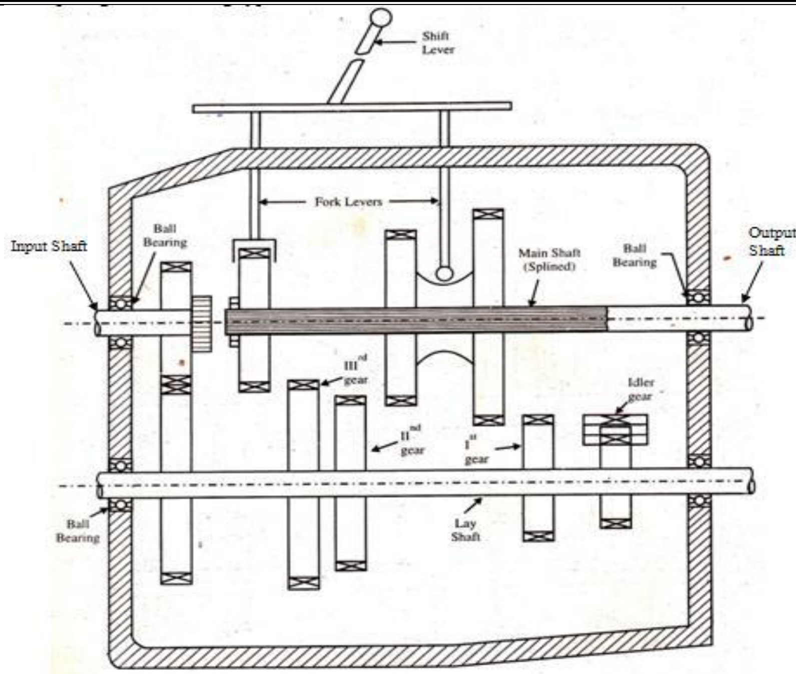

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Design procedure for teeth calculation.

First gear ratio:

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

Second gear ratio:

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

Third gear ratio:

$$\therefore G_3 = 1 : 1$$

Reverse gear ratio:

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

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<tr><td>Q.5</td><td>c)</td><td>Design the piston pin with following data: Maximum pressure on the piston is 4 N/mm^2; diagram of piston 70 mm, Allowable stresses due to bearing is 30 N/mm^2, bending 80 N/mm^2 and shear stress 60 N/mm^2.</td><td>8</td></tr>
</table>

		Answer: Given data, Dia. of piston = D = 70 mm. Max. pressure = P_{max} = 4 N/mm^2 Bearing pressure P_b = 30 N/mm2 Bending stress = σ b = 80N/mm^2 Shearing stress = τ = 60 N/mm^2 Maximum gas load, $=\frac{\pi D^2}{4} \times P_{max}$ $F = \frac{\pi(70)^2}{4} \times 4$ $\mathbf{F = 15.3938 \times 10^3\ N}$	1
		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 $l_p = 0.45 \times D = 0.45 \times 70$ $l_p = 31.5$ mm $F = d_{po} \times l_p \times P_b$ $d_{po} = \frac{15.3938 \times 10^3}{31.5 \times 30}$ $d_{po} = 16.29$mm $\mathbf{d_{po} = 17mm}$	2
		2. Designing the piston pin on the basis of bending. 'Bending moment 'M' is calculated as $M = F \times \frac{D}{8}$ $M = \frac{15.3938 \times 10^3 \times 70}{8}$ $M = 134.69 \times 10^3$ N-mm We know that, $M = \frac{\pi}{32} \times \sigma_b \times (d_{po})^3$ $134.69 \times 10^3 = \frac{\pi}{32} \times \sigma_b \times (17)^3$ $\mathbf{\sigma_b = 279.2589\ N/mm^2}$ The induced bending stresses are greater than permissible bending stress 80N/mm2 hence redesign is necessary. Now redesign value of d_{po} $M = \frac{\pi}{32} \times \sigma_b \times (d_{po})^3$ $134.69 \times 10^3 = \frac{\pi}{32} \times 80 \times (d_{po})^3$ d_{po} =25.79 mm $\mathbf{d_{po} = 26\ mm}$	2

		3. Designing piston pin on the basis of shear stress. $F = \frac{2\,\pi}{4} \times (d_{po})^2 \times \tau$ $15.39 \times 10^3 = \frac{2\,\pi}{4} \times (26)^2 \times \tau$ $\boldsymbol{\tau = 14.49\ N/mm^2}$ The induced shear stresses are less than permissible shear stress. Hence Design is safe **4.** The total length of piston is taken as L_{pt} = 0.9 D = 0.9 x 70 = **63 mm**	**2** **1**
Q.6		**Attempt any two of the following:**	**16**
Q.6	a)	A semi-elliptical spring has an overall length of 1m and sustain a load of 70 KN at its centre. The spring has 3 full length leaves and 15 graduated leaves with a central band of 100 mm width. All the leaves are to be stressed to 400 N/mm^2 when fully loaded. The ratio of total spring depth to that of the width is 2. Young modulus $E = 0.2 \times 10^6\ N/mm^2$.Determine: 1) Thickness and width of the leaves. 2) Initial gap that should be provided between full length and graduated leaves before the band load is applied. 3) The load exerted on the band after the spring is assembled.	**8**
		Given Data: 2W= Central load = 70KN $W = 35\times10^3 N$ Overall Length of spring 24= 1M L= 500mm , ℓ = 100mm $n_f = 03$, $n_g = 15$ $n = n_f + n_g = 18$ Ratio of total spring depth to width = 2 $\sigma_b = 400 N/mm^2$ Modulus of Elasticity $E = 02\times10^6\ N/mm^2$ 1) Thickness and width of leaves:- t = thickness of leaves, b= width of leaves $n = n_f + n_g = 15+3=18$	**1**

		As ratio of spring depth (n×t) to width of leaves is 2 $\therefore \frac{n\times t}{b} = 2 \quad \therefore \frac{18\times t}{b} = 2$ **b= 9t** Effective length of leaves $2L = 2L_1 - \ell = 1000-100 = 900mm$ $\therefore$ **L = 450mm** As all leaves are equally stressed $\frac{6WL}{nbt^2} \quad \therefore \; 400 = \frac{6\times35\times10^3\times450}{18\times9t\times t^2} = \frac{583\times10^3}{t^3}$ $t^3 = 1458 \quad \therefore t = 11.34mm = 12mm$ **b = 9t = 108mm** 2) Initial Gap: $C = \frac{2WL^3}{nEbt^3} = \frac{2\times35\times10^3\times450^3}{18 \times 0.2 \times 10^6 \times 108 \times 12^3}$ **C= 9.5mm** 3) The load exerted on the band after the spring is assembled. $W_b = \frac{2\,nf\times ng\;\times W}{n(2\,ng+3nf)} = \frac{2\times3\times15\times35\times10^3}{18(2\times15+3\times3)}$ $\mathbf{W_b = 4487N}$	1 1 1 2 2
Q.6	b)	Describe in detail the design procedure used to design the piston rings and piston skirts.	8

Answer: Design of Piston Rings:

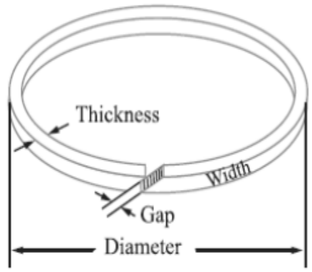


Fig. Piston rings. 1

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 1

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

Where, D = Cylinder bore in mm,

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

σ_t = Allowable bending (tensile) stress in MPa.

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

The minimum axial thickness (t_2) may also be obtained from the following empirical relation: 1

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

Where, n_R = Number of rings.

Width of top land, 1

$b_1 = t_H$ to 1.2 t_H

1

		Width of other ring lands, $b_2 = 0.75\, t_2$ to t_2 The gap between the free ends of the ring is given by $3.5\, t_1$ to $4\, t_1$. **Design of Skirt Length:** R = Normal side thrust acting on piston skirts **Maximum gas load** $F = P_{max} \times \frac{\pi}{4} D^2$ R = Normal side thrust acting on piston skirts $\because$ Side thrust = 10% $\therefore \quad R = 0.1\,F$ Let, l_1 = length of piston skirt The piston skirt act as a bearing inside the liner We have , $R = l_1 \times D \times P_b$ Where P_b = allowable bearing pressure on the piston skirt	1 1 1
Q.6	c)	Describe in detail the design procedure used to design: i) Thickness of cylinder head. ii) Cylinder head bolts or studs.	8
		i) Design Procedure to design Thickness of Cylinder Head:	

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

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

4

t = thickness of cylinder head

D = diameter of cylinder

C = constant

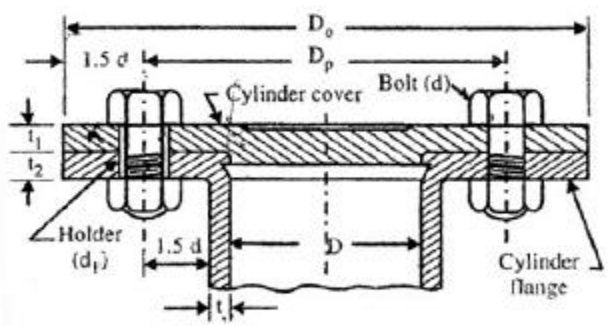
= 0.1………for C.I.

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.

ii) Design Procedure to design Cylinder Head bolts or studs:



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)

$$D_p = D + 2 \times 1.5d$$

$$= D + 3d \text{ ……………………(i)}$$

		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) 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) 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.	4