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

……

Revised Answer for Q.No.2 (d)

	Marks
1. **A)** Attempt **any three** :	12
a) State stepwise procedure for component design.	04
Answer : **Following procedure is carried out for designing any machine element:** 1. **Need: -** First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed. 2. **Mechanisms: -** Select the possible mechanism or group of mechanisms which will give the desired motion. 3. **Analysis of forces: -** Find the forces acting on each member of the machine and the energy transmitted by each member. 4. **Material selection: -** Select the material best suited for each member of the machine. 5. **Design of elements (Size and Stresses):-** Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. 6. **Modification: -** Modify the size of the member to agree with the past experience and judgment to facilitate manufacture. The modification may also be necessary by consideration of manufacturing to reduce overall cost. 7. **Detailed drawing: -** Draw the detailed drawing of each component and the assembly of the machine with complete specification for the manufacturing processes suggested. 8. **Production: -** The component, as per the drawing, is manufactured in the workshop.	04

b) Define system design, product design. 04

Answer: (Credit should be given to equivalent definition)

System Design: - It is the design of any complex mechanical system which will satisfy the specified requirement of system. Systems design could be seen as the application of systems theory to product development. 02

Product Design: - It is the process of creating a new product to be sold by a business to customers. It is essentially the efficient and effective generation and development of ideas through a process that leads to new products. 02

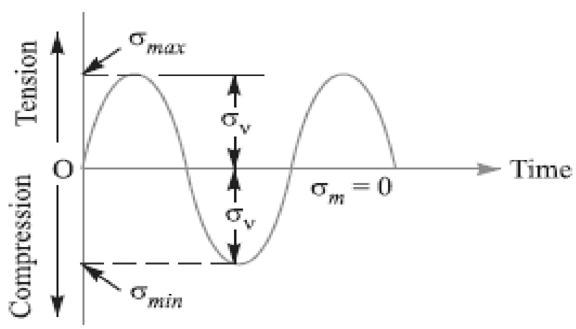
c) Define cyclic loading and describe any two types of cyclic stresses. 04

Answer: (Definition – 1 mark & 3 marks for types)

Cyclic Loading:- *Cyclic loading* is the application of repeated or fluctuating stresses, strains, or stress intensities to locations on structural components. 01

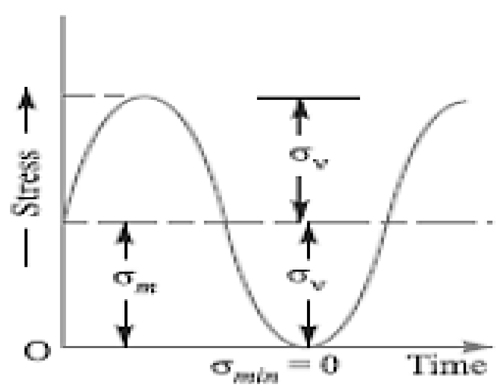
Cyclic stresses:- (Any Two)

1. **Completely Reversed stresses:-** The stresses which vary from a minimum value to a maximum value of the opposite nature and completely reversed cycle is known as completely Reversed stresses. 03

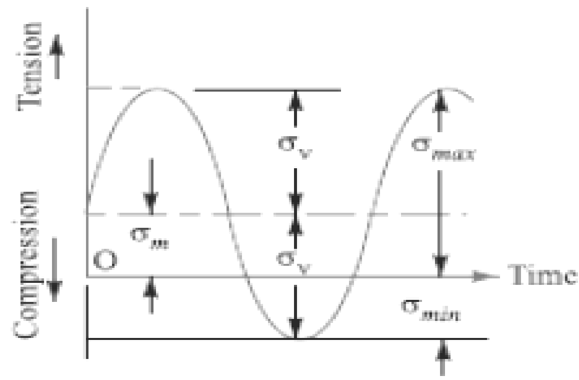


Completely reversed stress

2. **Repeated stresses:-** The stresses which vary from zero to a certain maximum value are called repeated stresses.



3. **Fluctuating Stresses:-** The stresses which vary from a minimum value to a maximum value of the same nature are called repeated stresses.



d) State all empirical relations required to design knuckle joint. — 04

Answer: Empirical relations required to design a Knuckle joint:- — 04

When, d= Diameter of rod

1. Diameter of knuckle pin d_1=d
2. Outer diameter of eye d_2=2d
3. Diameter of knuckle pin head or collar d_3=1.5d
4. Thickness of single eye rod t=1.25d
5. Thickness of double eye rod t_1=1.25d
6. Thickness of knuckle pin head or collar t_2=1.25d

B) Attempt **any one :** — 6

a) Define pitch circle diameter, diametral pitch, module and state relation between them. — 6

Answer:

Pitch circle diameter: -

It is the diameter of pitch circle. The size of gear is usually specified by the pitch circle diameter. It is denoted by letter '**D**'. — 1

Diametral Pitch:-

It is the ratio of number of teeth to the pitch circle diameter in millimeters. It is denoted by letter '$\mathbf{P_d}$'. — 1

Module:-

It is defined as length of pitch circle diameter per tooth. — 1

OR

It is the ratio of pitch circle diameter in millimeters to the number of teeth. It is denoted by letter '**m**'.

Relation between Pitch circle diameter, Diametral Pitch and Module:-

1. Module ,m = PCD (D)/ No. of teeths (T) 1

$$= \frac{D}{T}$$

2. Diametral Pitch, P_d = No. of Teeth (T) / PCD (D) 1

$$= T / D$$

3. $P_d = \frac{T}{D} = \frac{1}{m}$ 1

b) State relations between load, effort and reaction at fulcrum when 6
I) Load and efforts are parallel and acting opposite in direction.
II) Load and effort are inclined to each other.
III) Load and effort are right angled to each other and arms are inclined at an angle ‘θ’

Answer: (Three Cases – 2 marks each)

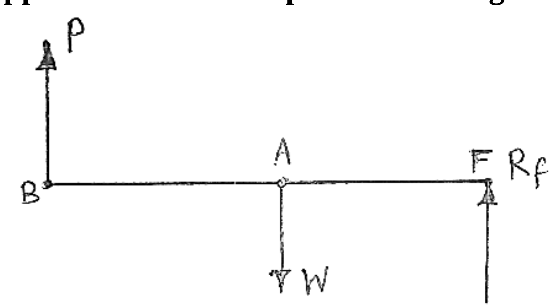
I) Load and effort are parallel and acting opposite in direction:
R_f is determine by using following relation :

$$R_f = W - P \; 3$$

Or

$$R_f = P - W$$

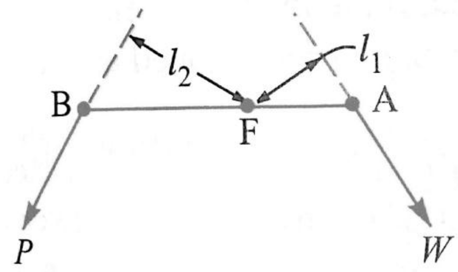
The direction of R_f will be opposite that of W or p whichever is greater.



02

II) Load and effort are inclined to each other:

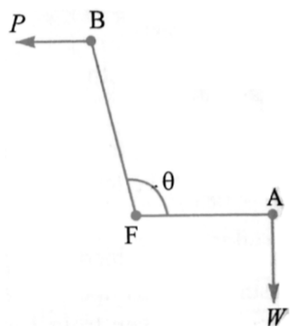
R_f Which is equal to the resultant of W and P, is determined by parallelogram law of forces. The line of action of R_f passes through intersection of W and P and also through F. The direction of R_f depends up on the direction of W and P 02



III) Load and effort are right angled to each other and arms are inclined at an angle ‘θ’:

R_f is determine by using following relation : 02

$R_f = \sqrt{P^2 + W^2 - 2\,PW\cos\theta}$



2. Attempt **any four** :	16
a) Write stepwise procedure to design of piston pin.	4

Answer :

Design of piston pin:-

The piston pin (also called gudgeon pin or wrist pin) is used to connect the piston and the connecting rod. It is usually made hollow and tapered on the inside, the smallest inside diameter being at the centre of the pin, as shown in Fig. 01

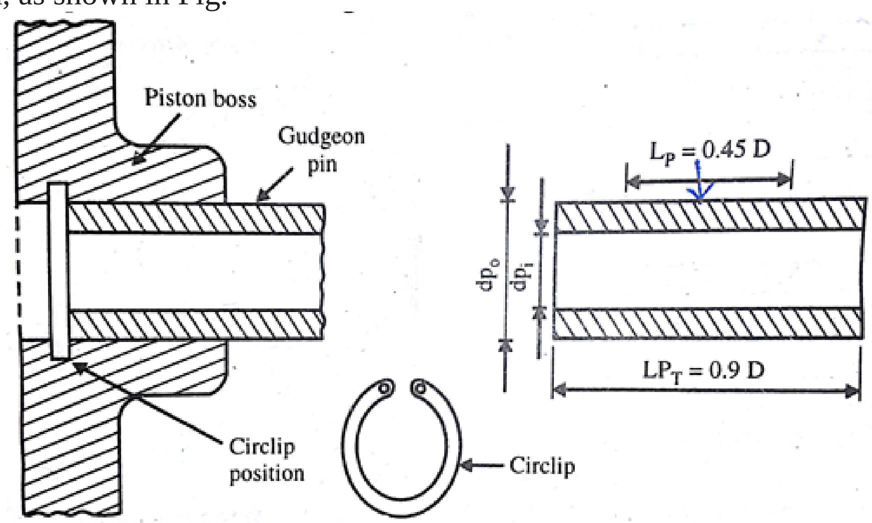


Fig. Piston Pin

The material used for the piston pin is usually case hardened steel alloy containing nickel, chromium, molybdenum or vanadium having tensile strength from 710 MPa to 910 MPa.

1. Design of piston pin on the basis of bearing strength:-

Maximum gas load on the piston (f),

$$f = P_{max} \frac{\pi}{4} D^2$$

Load on the piston pin due to bearing pressure is given as,

$$f = \text{Bearing pressure} \times \text{Bearing area}$$ 01

$$f = P_b \times dp_o \times l_p$$

Where,

dp_o = Outer diameter of piston pin,

l_p = Length of piston pin = 0.45 × D

dp_i = Inner diameter of piston pin = 0.6 × dp_o

2. Design piston pin on the basis of bending :-

The piston pin is checked for bending with gas load taken as U.D.L over entire length l_p and pin supported at the centre on bosses at two ends.

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

The diameter of piston pin is checked for above moment, 01

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

$$\frac{M}{\frac{\pi}{64}\left(dpo^2 - dpi^2\right)} = \frac{\sigma_b}{\frac{dpo}{2}}$$

If σ_b is greater than that of given, redesigning is necessary.

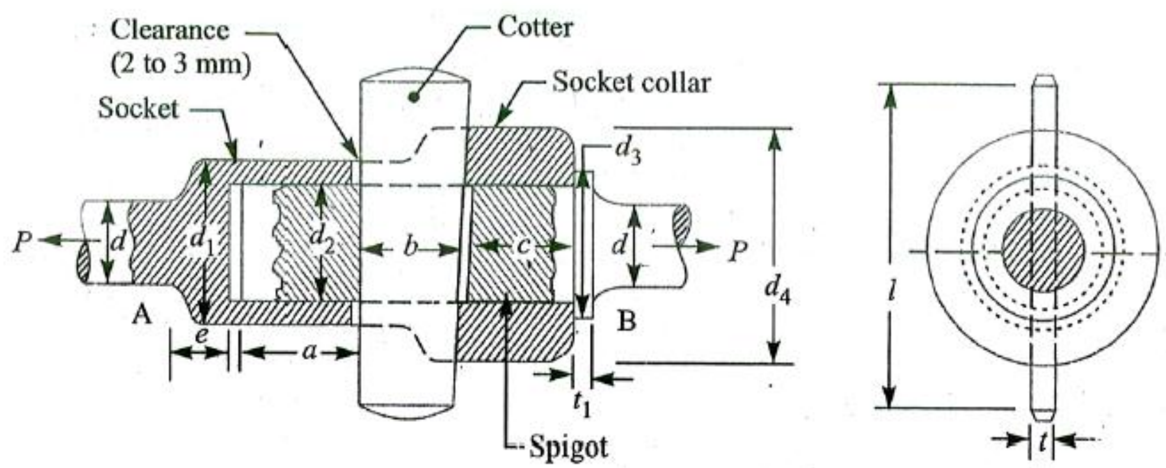
3. Design piston pin on the basis of shear stress :-

The pin is in double shear, so 01

$$\tau = \frac{f}{2A} = \frac{f}{2 \times \frac{\pi}{4} \times \left(dpo^2 - dpi^2\right)}$$

4. Total length of piston pin (lp_t)

$$lp_t = 0.9\,D$$

b) Suggest suitable materials for propeller shaft and leaf spring with suitable justifications.	4
Answer : (Material – 1mark & Justification – 1 mark) **Propeller shaft:-** **Material:-** Alloy steel such as nickel, nickel chromium, chrome vanadium steel. **Justification:** - Propeller shaft transmits power from gear box to differential. So it required high torsional strength and rigidity as well as material that can take a lot of fatigue. **Leaf spring:-** **Material:-** Plain carbon steel having 0.9 to 1.0% carbon. **Justification:** - The leaves are heat treated after forming processes. The heat treatment of spring steel produces greater strength and therefore greater load capacity, greater range of deflection and better fatigue properties.	2 2
c) Explain procedure for design of cotter joint.	4
Answer: (Any four steps – 1 mark each)  Fig. Socket and Spigot type cotter joint From fig. P = Load carried by the rods d = Diameter of the rods d1 = Outside diameter of socket d2= Diameter of Spigot or inside diameter of socket d3= Outside diameter of spigot collar t1= Thickness of spigot collar d4 = Diameter of socket collar c = Thickness of socket collar b = Mean width of cotter t = Thickness of cotter l = Length of cotter	04

1. Find the diameter of rod a considering failure of rod in tension.

$$P = \frac{\Pi}{4} d^2 \sigma_t$$

2. Find inside diameter of socket or outside diameter of spigot considering failure spigot in tension at the weakest section (at slot)

$$P = \left[\frac{\Pi}{4} d_2^2 - d_2 t\right] \sigma_t$$

$\therefore$ Assume $t = \frac{d_2}{4}$ ……. standard practice

3. Check the crushing stress induced in cotter or spigot by using above obtained values of 't' and 'd_2'

$$P = d_2 . t . \sigma_c$$

If the induced crushing stress is greater than the permissible design is unsafe, so redesign value of d_2 and t on basis of allowable σ_c.

4. Find the outside diameter of socket 'd_1 by considering failure socket in tension at weakest section (at slot)

$$P = \left\{\frac{\Pi}{4}\left(d_1^2 - d_2^2\right) - \left(d_1 - d_2\right)t\right\} \sigma_t$$

5. Find the diameter of spigot collar 'd_3' by using considering failure of spigot in crushing

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

6. Find the diameter of socket collar 'd_4' by considering failure of socket collar in crushing

$$P = \left(\boldsymbol{d}_4 - \boldsymbol{d}_2\right) t . \sigma_c$$

7. Find the width of cotter 'b' considering the failure of cotter in shearing since cotter is in double shear.

$$P = 2.b.t.\sigma_s$$

8. Find the thickness of spigot collar 't_1' by considering failure of collar in shear

$$P = \Pi . d_2 . t_1 \sigma_s$$

9. Find the thickness of socket collar 'c' by considering failure of socket collar in shear. $P = 2(\boldsymbol{d}_4 - \boldsymbol{d}_2)\boldsymbol{c}.\boldsymbol{\sigma}_s$ 10. Find distance from end of slot to end of rod 'a' by considering failure in shearing $P = 2a.d_2.\boldsymbol{\sigma}_s$ 11. Find length of cotter 'l' $l = 4d$ 12. Find thickness at neck of socket 'e' $e = 1.2d$ 13. Check the bending stress induced in cotter $\boldsymbol{\sigma_b} = \frac{\boldsymbol{P}(\boldsymbol{d}_4 + 0.5\boldsymbol{d}_2)}{2.\boldsymbol{t}.\boldsymbol{b}^2}$	
d) A bell crank lever is pivoted to a pin of diameter 20 mm, to raise a load of 5 kN short arm end. The lengths of short arm and long arm are 100 mm and 450 mm resp. Determine shear stress and bearing pressure induced in fulcrum pin.	4
Answer: **Given data:** Diameter of pin $d = 20mm$ Load $W = 5kN$ Length of short arm $L_1 = 100mm$ Length of long arm $L_2 = 450mm$ $\therefore$ **Bending movement of bell crank lever at fulcrum pin,** $(W \times L_1) - (P \times L_2) = 0$ $W \times L_1 = P \times L_2$ $5 \times 100 = P \times 450$ $\therefore P = 1.1111kN$ **Reaction at fulcrum pin,** $R_f = \sqrt{W^2 + P^2} = \sqrt{5^2 + 1.1111^2}\,kN$ $R_f = 5.122kN = 5.122 \times 10^3\,N$	1 1

i) **Shear stress induced in fulcrum Pin** : $R_f = 2\frac{\pi}{4}d^2 \times \sigma_s$ $5.122\times10^3 = 2\times\frac{\pi}{4}\times20^2\times\sigma_s$ $\therefore \sigma_s = 8.15\times10^{-3}\ N/mm^2$ Corrected Answer **ii)Bearing pressure induced in fulcrum pin:** $R_f = d\times l\times P_b$ Where, $l = 1.25d$ $l = 1.25\times20 = 25mm$ $\therefore 5.122\times10^3 = 20\times25\times P_b$ $\therefore\ P_b = 10.24\ \ N/mm^2$	1 1
e) Two arms of rocker arm are equal and make included angle of 160º . It is used to operate exhaust valve, for which maximum force required is 5kN. Determine length of fulcrum pin and induced shear stress in pin material if allowable bearing pressure in pin material is 7 N/mm^2 .	4
Answer: **Given data**: $\theta = 160^0$ $P = 5kN$ $P_b = 7N/mm^2$ **Two arms of rocker arm equal,** So, $L_1 = L_2$ $\therefore P = W = 5kN$ $\therefore$ **Reaction at fulcrum pin,** $R_f = \sqrt{P^2 + W^2 - 2PW\cos\theta}$ $= \sqrt{5^2 + 5^2 - (2\times5\times5\times\cos 160)}$ $R_f = 9.848kN = 9.848\times10^3 N$ $\therefore$ **Diameter of fulcrum pin,**	 1

$R_f = d \times l \times P_b$ Where, $l = 1.25d$ $\therefore 9.848\times10^3 = d\times1.25d\times7$ $\therefore d = 33.54mm \quad \therefore d \cong 34mm$ **i) Shear stress induced in pin,** $R_f = 2\frac{\Pi}{4}d^2\sigma_s$ $\therefore 9.848\times10^3 = 2\frac{\Pi}{4}(34)^2\times\sigma_s$ $\therefore \sigma_s = 5.423N/mm^2$ ii)**Length of fulcrum pin**, $l = 1.25d = 1.25\times34 = 42.5mm.$ $l \cong 43mm$	1 1 1
3. Attempt **any four** :	16
a) The tie rod is 40 mm in diameter. Determine the dimensions of coupler and coupler nut empirically.	04
Answer: Given, Diameter of tie rod, d = 40 mm 1. Diameter of coupler (D) D = 1.25 d = 1.25 $\times$ 40 = 50 mm 2. Inside diameter of coupler (D_1) D_1 = d + 6 = 40 + 6 = 46 mm 3. Outside diameter of coupler (D_2) D_2 = 1.5 d = 1.5 $\times$ 40 = 60 mm 4. Length of coupler nut (l) $l = d \text{ to } 1.5\ d = 40\ mm \text{ or } 60\ mm$ 5. Length of coupler (L) L = 6d = 6 $\times$ 40 = 240 mm	04
b) In a sliding mesh gear box with three forward and one reverse speeds, clutch shaft pinion has 14	04

teeth, low gear main shaft has 32 teeth, the corresponding lay shaft pinions have 36 and 18 teeth. Determine centre distance between shafts if gears of 3.25 mm module are to be employed also determine gear ratio for second forward speed.	
Answer: Given $T_A = 14$, $T_F = 32$, $T_E = 18$, $T_B = 36$, $M = 3.25$. 1. Centre distance = Centre distance = $\frac{M\,(T_A+T_B)}{2} = \frac{3.25\,(14+36)}{2} = 81.25\ mm$ 2. Gear ratio for second forward speed (G_2) = $\frac{T_B}{T_A} \times \frac{T_F}{T_E} = \frac{36}{14} \times \frac{32}{18} = 4.57$	02 02
c) If radial width of piston ring is 8 mm and if two compression rings and one oil ring are to be employed on piston having crown thickness of 10 mm, compute length of piston above skirt.	04
Answer: Given, Compression Rings = 2, Oil ring = 1, Width of piston ring b_r = 8 mm, Crown thickness t_H = 10 mm 1. **Top Land (b_1)** (b_1) $= t_H\ TO\ 1.2\ t_H = 10\ TO\ 10 \times 1.2 = 10\ to\ 12\ mm$ Consider, Top Land (b_1) = 11 mm 2. **Length of the Ring section** The axial thickness (t_2) of the rings = 0.7 b_r = 0.7×8 = 5.6 mm Length of other land (b_2) = 0.85 b_r = 0.85×8 = 6.8 mm Length of the Ring section = $3\ t_2 + 2\ b_2$ $= 3 \times 5.6 + 2 \times 6.8$ = 30.4 mm **Length of piston above skirt = Top Land (b_1) + Length of the Ring section** **Length of piston above skirt = 11 +30.4 = 41.4 mm**	01 02 01
d) For a cotter joint sustaining 50 kN load, spigot diameter is 46 mm. Determine outer diameter of	04

socket and diameter of socket cotter, if mean width of cotter is 40 mm. also determine induced bending stress in cotter.	
Answer: Given, P = 50 kN = 50 $\times 10^3$ N Spigot diameter d_2 = 46 mm, Mean width of cotter b = 40 mm Assume permissible tensile stress $\sigma_t = \frac{S_{yt}}{f_s} = \frac{400}{6} = 66.67\ N/mm^2$, t = 0.31d = 9.92 mm ≅ 10mm 1. **Outer diameter of socket d_1** $P = \left[\frac{\pi}{4}\left({d_1}^2 - {d_2}^2\right) - (d_1 - d_2)t\right] \sigma_t$ $50 \times 10^3 = \left[\frac{\pi}{4}\left({d_1}^2 - 40^2\right) - (d_1 - 40)10\right] 66.67$ ${d_1}^2 - 12.73 d_1 - 2045.59 = 0$ Solving the above quadratic equation, we have $d_1 = 52.04\,mm \cong 55\,mm$ 2. **Diameter of socket collar (d_4)** First find out the diameter of rod (d) $d = \sqrt{\frac{4P}{\pi\,\sigma_t}} = \sqrt{\frac{4 \times 50 \times 10^3}{\pi \times 66.67}} = 30.90 \cong 32\ mm$ Diameter of socket collar $d_4 = 2.4\ d = 2.4 \times 32 = 76.8 \cong 80\ mm$ 3. **Bending stress induced in cotter (σ_b)** (σ_b) = M_{max}/Z since M_{max} = Bending moment & Z is section modulus $\sigma_b = \frac{P\,(d_4 + 0.5\,d_2)}{2\,t \times b^2}$ $\sigma_b = \frac{50 \times 10^3 \times (80 + 0.5 \times 55)}{2 \times 10 \times 40^2} = 167.96\ N/mm^2$	01 01 01 01

e) State empirical relations used to design rectangular sunk key and prove that for a square key induced crushing stress is twice the shear stress.	04

Answer:

Assume, 'd' (mm) is diameter of shaft.

Empirical relations used to design rectangular sunk key are,

1. The length of the key(l) = $1.5\ d$ mm
2. Width of key (w) = $\frac{d}{4}$ mm
3. Thickness of key (t) = $\frac{d}{6}$ mm

01

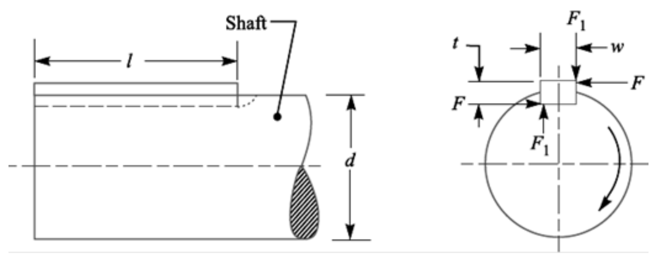


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Fig. shows forces acting on a key for clockwise torque transmitted from a shaft to hub.

We know that,

1. Strength equation considering shearing failure of key,

$$T = \tau \times l \times b \times \frac{d}{2} \quad \text{........ (a)}$$

01

2. Crushing equation considering crushing failure of key,

$$T = \sigma c \times l \times t \times \frac{d}{4} \quad \text{......... (b)}$$

And for square key, $t = b$

Substituting and equating the above equations (a) & (b),

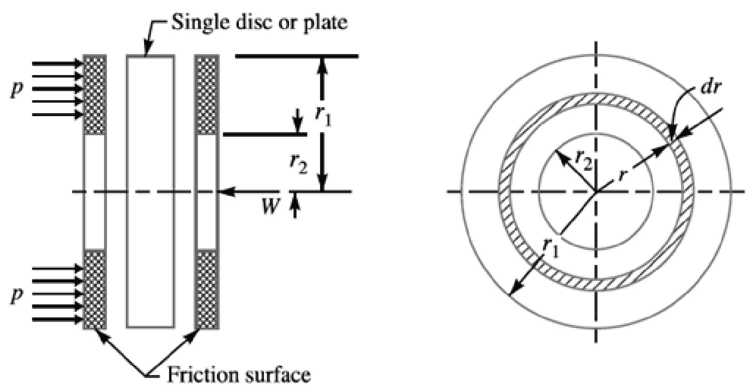
$$\tau \times l \times b \times \frac{d}{2} = \sigma c \times l \times b \times \frac{d}{4}$$

$$\tau = \sigma c \times \frac{1}{2}$$

01

$$\sigma c = 2\tau$$

So for a square key induced crushing stress is twice the shear stress

4. A) Attempt **any three** :	12
a) State stepwise procedure to obtain length of leaf spring leaves.	04
Answer: (Credit should be given to equivalent procedure & sketch) Let, $2L_1$ = Overall length of the spring, l = Width of band or distance between centers of U-bolts (Ineffective Length). n = total number of leaves. Active length of spring is calculated as, $2L = 2L_1 - l$ (When band is used) $2L = 2L_1 - \frac{2}{3}l$ (When U-Bolt is used) If a leaf spring has two full length leaves, then the length of leaves is obtained as follows: Length of smallest leaf = = $\frac{\text{Effective Length}}{n-1}$ + Ineffective length Length of next leaf = $\frac{\text{Effective Length}}{n-1} \times 2$ + Ineffective length Length of $(n-1)_{th}$ leaf = $\frac{\text{Effective Length}}{n-1} \times (n-1)$ + Ineffective length	04
b) State stepwise procedure to determine face width of frictional surface of a single plate clutch.	04
Answer: (Consider any one procedure out of below mentioned theories) **Design procedure to find out face width of single plate clutch:-**  Fig. Forces on a single plate clutch	04

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 the following two cases: **(consider any one case)**

1. **When there is a uniform pressure, and**
2. **When there is a uniform axial wear.**

1. **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,

Torque transmitted on ring section, $(T_r) = \mu \times P \times 2\pi r^2 d_r$

$$T = \mu W \times \frac{2}{3}\frac{(r_1^3 - r_2^3)}{(r_1^2 - r_2^2)}$$

Total Torque on frictional surface ,

$$T = \mu W R$$

Where, R = Mean radius of friction $= \frac{2}{3}\frac{(r_1^3 - r_2^3)}{(r_1^2 - r_2^2)}$

The face width of frictional surface (b) = $(r_1 - r_2)$

2. **Considering uniform axial wear.**

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

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

Torque transmitted on ring section, $T_r = \mu \times c \times 2\pi r d_r$

Total Torque on frictional surface ,

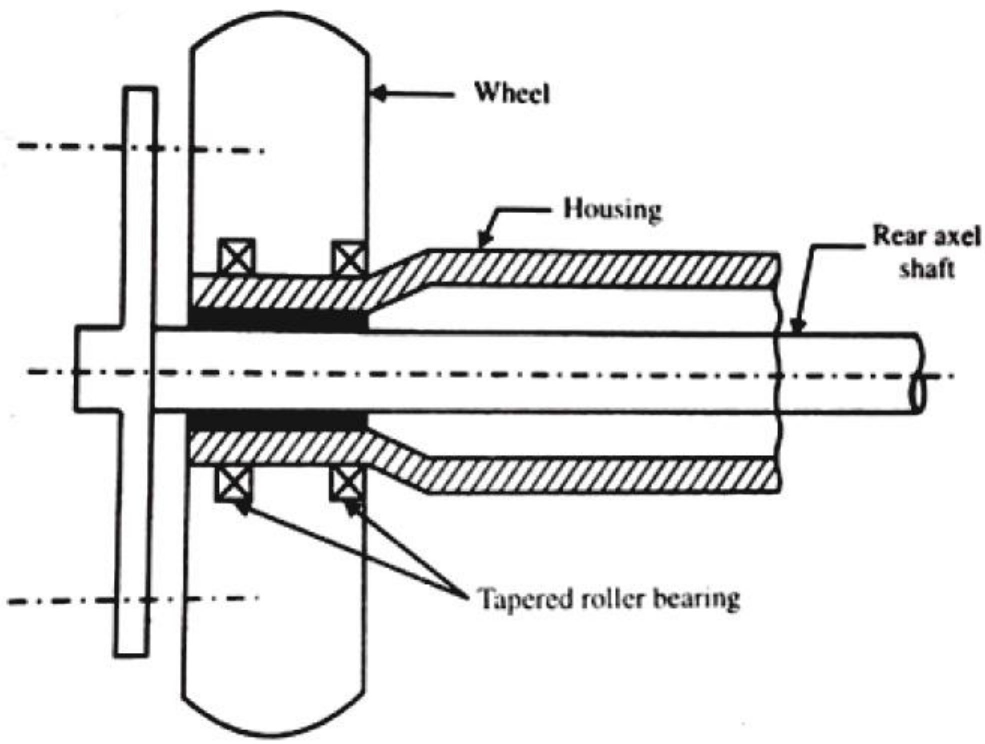
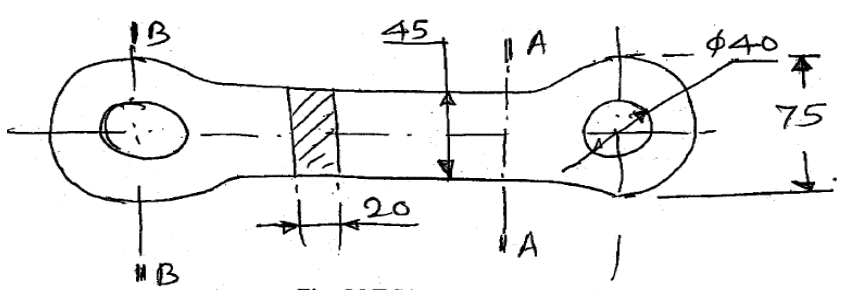
$$T = \mu W R$$

Where, R = Mean radius of friction $= \frac{(r_1 + r_2)}{2}$

The face width of frictional surface (b) = $(r_1 - r_2)$

c) State any two applications of each of knuckle joint and turn buckle in an automobile.	04
Answer: **Applications of Knuckle joint**: (Any two – 1 mark each) 1. Tie rod joints for roof truss 2. Valve rod joint for eccentric rod pump rod joint 3. Tension link in bridge structure 4. Lever and rod connection of various types. 5. swing arm of two wheeler 6. Connection of link rod of leaf springs in multi axle vehicles 7. Piston ,Piston Pin ,Connecting Rod 8. Connections of leaf spring with chassis **Applications of Turn Buckle:** (Any two – 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	02 02
d) Define whirling, critical speed and state effects of whirling on transmission shaft.	04
Answer: Definitions: **Whirling: -** 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: -** 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. **Effect of whirling on transmission shaft: -** 1. Propeller shafts of road vehicles are sufficiently long and operate at high speed. Consequently, whirling may occur at certain critical speed. 2. Whirling causes bending stresses in material that are higher than shearing stress caused by transmitted torque. 3. At critical speed shaft vibrates violently and this also sets up a sympathetic resonant vibrations in the vehicle body.	01 01 02

4 B) Attempt **any one** :	6
a) Describe importance of ergonomics and aesthetics in designing an automobile.	6
Answer: **Importance of ergonomics:** 1. The importance of ergonomics is to reduce the operational difficulties present in man – machine joint system & thereby reduce the resulting physical and mental stresses. 2. It gives exhaustive details of the dimensions and resisting forces of different control elements. 3. Standardization of automobile system controls as per the regional anthropometry.	03
Importance of aesthetic: - 1. It gives the functional requirements and appearance of the product, the functional requirements results in shapes which are aesthetically pleasing. 2. It gives the cumulative effect of a number of factors like form, color, rigidity, and tolerance, motion of individual components, manufacturing method and noise. 3. Better surface finish always attracts the observers which increases the customers satisfaction.	03
b) Design diameter of fully floating rear axle if engine power is 80 kW at 5000 rpm. Gearbox ratios are 4:1, 2.4:1, 1.5:1, 1:1. The differential reduction is 5:1. Allowable sheer stress for shaft material is 65 N/mm^2 sketch the arrangement of the axle.	06
Answer: Given:- $P = 80$ kW $= 80 \times 10^3$ W, $N = 5000$ rpm, Max. gear ratio, $G_1 = 4 : 1$, Differential reduction $G_d = 5:1$, Now the torque transmitted by the engine T_e :- $P = \frac{2\pi NT}{60}$ $80 \times 10^3 = \frac{2 \times 3.14 \times 5000 \times T_e}{60}$ $T_e = 152.79 \text{Nm} = 152.79 \times 10^3$ Nmm	01
Now torque transmitted by rear axle shaft T_{RA}, $T_{RA} = Te \times G_1 \times G_d$ $T_{RA} = 152.79 \times 10^3 \times 4 \times 5$ $T_{RA} = 3055.78 \times 10^3$	01

Let, d = diameter of rear axle, $T_{RA} = \frac{\pi}{16} f_s d^3$ $3055.78 \times 10^3 = \frac{\pi}{16} \times 65 \times d^3$ $d^3 = 239425.2$ $d = 62.095\, mm \cong 64\, mm$	02
Arrangement of fully floating rear axle:-  Fig. Fully floating rear axle	02
5. Attempt **any two:**	16
a) I) A C.I. link, as shown in fig. transmits a load of 45 kN. Find the tensile stress induced in link material at section A-A and B-B.  Fig. (N.T.S.)	4

Answer: Given $P = 45kN = 45 \times 10^3 N$ Tensile stress induced at section A-A We know that the cross sectional area of link at section A-A $A_1 = 45 \times 20 = 900\ mm^2$ $\therefore$ Tensile stress inducted at section A-A $\sigma = \frac{P}{A_1} = \frac{45 \times 10^3}{900} = 50\ N/mm^2$	2
Tensile stress induced at section B-B We know that the cross- sectional area of link at section B-B, $A_1 = 20(75 - 40) = 700\ mm^2$ $\therefore$ Tensile stress induced at section B-B $\sigma_2 = \frac{P}{A_1} = \frac{45 \times 10^3}{700} = 64.28 N/mm^2$	2
II) Calculate force required to punch a circular hole of Ø 60 mm in a plate of 5 mm thickness. If ultimate shear stress of plate material is 350 N/mm^2 and determine crushing stress experienced by punch.	4
Answer: Given: d = 60 mm ; t = 5 mm ; τ = 350 N/mm^2 **Area under shear,** $A = \pi\ d \times t = \pi \times 60 \times 5 = 942.6\ mm^2$ Force required to punch a hole, $P = A \times \tau_u = 942.6 \times 350 = 329\ 910\ N = 329.91\ kN$	2
Crushing Stress experienced by punch = $\sigma_c = P/A$ $= P/(d \times t)$ $= 329\ 910/(60 \times 5)$ $= 1099.7\ N/mm^2$	2

b) Design and draw flange coupling for a shaft transmitting 90 kW at 250 rpm. The allowable shear stress for shaft material is 40 MPa and the angle of twist is not to exceed 10 in a length of twenty diameters. The allowable shear stress in coupling bolts is 30 MPa and i.e. for C.I. is 14 MPa. Assume key and shaft materials are same.	8

Answer:

Solution : Given

$$P = 90 kW \times 10^3 W$$

$$N = 250 \times r.p.m$$

$$\tau_s = 40 MPa = 40 N/mm^2$$

$$\theta = 1^0 = \Pi/180 = 0.0175 rad$$

$$\tau_b = 30 MPa = 30 N/mm^2$$

Find the diameter of the shaft (d) the torque transmitted by the shaft.

$$T = \frac{P \times 60}{2\Pi N} = \frac{90 \times 10^3 \times 60}{2\Pi \times 250} = 3440\ N-m = 3440 \times 10^3\ N-mm$$

Considering strength of the shaft we know that

$$\frac{T}{J} = \frac{\tau_s}{d/2},$$

$$\frac{3440 \times 10^3}{\frac{\Pi}{32} \times d^4} = \frac{40}{d/2} \text{ or } \frac{35 \times 10^6}{d^4} = \frac{80}{d} \ldots\ldots\ldots \left(\therefore J = \frac{\Pi}{32} \times d^4 \right)$$

$$\therefore\ d^3 = 35 \times 10^6 / 80 = 0.438 \times 10^6 \text{ or } d = 76 mm$$

Considering rigidity of the shaft,

$$\frac{T}{J} = \frac{C \times \theta}{l}$$

$$\frac{3440 \times 10^3}{\frac{\Pi}{32} \times d^4} = \frac{84 \times 10^3 \times 0.0175}{20d} \text{ or } \frac{35 \times 10^6}{d^4} = \frac{73.5}{d} \text{ ....(Taking C=84kN/mm}^2\text{)}$$

$$\therefore\ d^3 = 35 \times 10^6 / 73.5 = 0.476 \times 10^6 \quad \text{or} \quad d = 78 mm$$ 2

Taking the larger of the two values,

$$d = 78 mm \text{ say } 80 mm \text{ Ans}.$$

1. Design for hub

Outer diameter of hub $D = 2d = 2 \times 80 = 160\,\text{mm Ans}$

And length of hub $L = 1.5d = 1.5 \times 80 = 120\,\text{mm Ans}$

Let us now check the induced shear stress in the hub by considering it as a hollow shaft. The shear stress for the hub material (which is cast iron) is usually 14 MPa. ,

Torque transmitted (T)=

$$3440 \times 10^3 = \frac{\Pi}{16} \times \tau_c \left[\frac{D^4 - d^4}{D}\right] = \frac{\Pi}{16} \times \tau_c \left[\frac{(160)^4 - (80)^4}{160}\right] = 754 \times 10^3 \tau_c$$ 1

$\therefore \tau_c = 3440 \times 10^3 / 754 \times 10^3 = 4.56\,\text{N/mm}^2 = 4.56\,\text{MPa}$

Since the induced shear stress for the hub material is less than 14MPa therefore the design for hub is safe.

2. Design for key

Find that the proportions of key for a 80 mm diameter shaft are Width of key, w = 25 mm Ans

And thickness of key t=14 mm Ans

The length of key (*l*)is taken equal to the length of hub(L).

$\therefore l = L = 120\,\text{mm Ans}$ 1

Assuming that the shaft and key are of the same material. Check the induced shear stress in key we know that the torque transmitted (*T*)

$$3440 \times 10^3 = l \times w \times \tau_k \times \frac{d}{2} = 120 \times \tau_k \frac{80}{2} = 120 \times 10^3 \tau_k$$

$\tau_k = 3440 \times 10^3 / 120 \times 10^3 = 28.7\,\text{N/mm}^2 = 28.7\,\text{MPa}$

Since the induced shear stress in the key is less than 40MPa, therefore the design for key is safe.

3. Design for flange:

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

$\therefore\ (t_f) = 0.5d = 0.5 \times 80 = 40\text{mm}$ Ans

Let us now check the inducted shear stress in the cast iron flange by considering the flange at the junction of the hub under shear. We know that the torque transmitted (*T*)

$$3440 \times 10^3 = \frac{\Pi D^2}{2} \times t_f \times \tau_c = \frac{\Pi (160)^2}{2} \times 40 \times \tau_c = 1608 \times 10^3 \tau_c$$

$$\therefore \quad \tau_c = 3440 \times 10^3 / 1608 \times 10^3 = 2.14 \ \text{N/mm}^2 = 2.14\text{MPa}$$ 1

Since the induced shear stress in the flange is less than 14 MPa therefore the design for flange is safe.

4. Design for bolts

Let d_1 Nominal diameter of bolts.

Since the diameter of the shaft is 80mm there for let us take number of bolts.

$n = 4$

And pitch circle diameter of bolts

$D_1 = 3d = 3 \times 80 = 240\text{mm}$ 1

The bolts are subjected to shear stress due to the torque transmitted, we know that torque transmitted (*T*).

$$3440 \times 10^3 = \frac{\Pi}{4}(d_1)^2 n \times \tau_b = \frac{D_1}{2} = \frac{\Pi}{4}(d_1)^2 \times 4 \times 30 \times \frac{240}{2} = 11311(d_1)^2$$

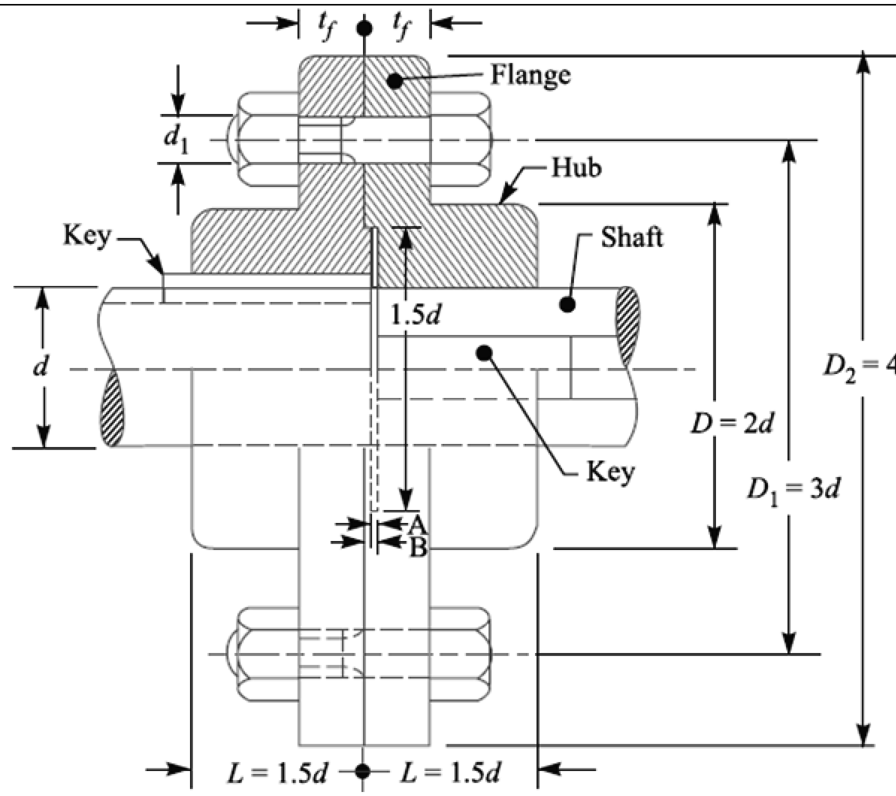
$$\therefore \ (d_1)^2 = 3440 \times 10^3 / 11311 = 304 \ \text{ or } d_1 = 17.4\,mm$$

Assuming coarse threads the standard nominal diameter of bolt is 18mm

The other proportions are taken as follows:

Outer diameter of the flange,

$D_2 = 4d = 4 \times 80 = 320\text{mm}$ Ans



Flange coupling

2

c) Design a piston pin for a piston having diameter 100 mm and sustaining max. gas pressure of 5 N/mm^2. The allowable stresses for pin material are 25 N/mm^2 in bearing, 70 N/mm^2 in shear and 140 N/mm^2 in bending. Draw a neat sketch of piston and locate piston pin centre on it. Assume max. bearing pressure on piston is limited to 0.45 N/mm^2. 8

Answer: Answer: **Given data,**

Dia. of piston = D = 100 mm.
Max. Gas pressure = P_{max} = 5 N/mm^2
Bearing pressure P_b = 25 N/mm^2
Bending stress = σ_b = 140N/mm^2
Shearing stress = τ = 70 N/mm^2

Let,

R = Normal side thrust acting on piston skirts

Maximum gas load $F = P_{max} \times \frac{\pi}{4} D^2$

$$F = 5 \times \frac{\pi}{4}(100)^2 = 39.2699 \times 10^3 \, N$$

$$R = 0.1 \times F = 0.08 \times 39.2699 \times 10^3$$ Assume $\because$ Side thrust=10% 1

$$\therefore \quad R = 3926.9 \, N$$

Let,

$$l_1 = \text{length of piston skirt}$$

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

$$\therefore\ l_1 = \frac{3926.9}{100 \times 0.45}$$

$$l_1 = 87.264\ \text{mm}$$

$$\therefore\ l_1 = 88\ \text{mm}$$

A) Design of piston spring on the basis of bearing pressure:

Maximum gas pressure = bearing load $F = d_{po} \times l_p \times p_b$

d_{po} = outer diameter of piston pin

l_p = length of piston pin small end of connecting rod = $0.45 \times D = 45\,\text{mm}$ 2

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

$$39.2699 \times 10^3 = d_{po} \times 45 \times 25$$

$$d_{po} = 34.90 = 35\,mm$$

B) Design of piston pin on the basis of bending

$$M = \frac{F \times D}{8} = \frac{39.2699 \times 10^3 \times 100}{8}$$

$$= 490.862 \times 10^3\ \text{N} - \text{mm}$$

$$M = \frac{\Pi}{32} \times \sigma_b \times d_{po}^{\ 3}$$

$$490.862 \times 10^3 = \frac{\Pi}{32} \times \sigma_b \times 35^3$$ 1

$$\sigma_b = 116.61\text{N/mm}^2$$

The induced bending stress are less than permissible stress 140N/mm^2 hence design is ok.

C) Design of piston pin on the basis of shear stress, due to double shear

$$F = 2\frac{\Pi}{4} \times d_{po}^{\ 2} \times \tau$$

$$39.2699 \times 10^3 = 2\frac{\Pi}{4} \times 35^2 \times \tau$$

$$\tau = 20.41\text{N/mm}^2$$ 1

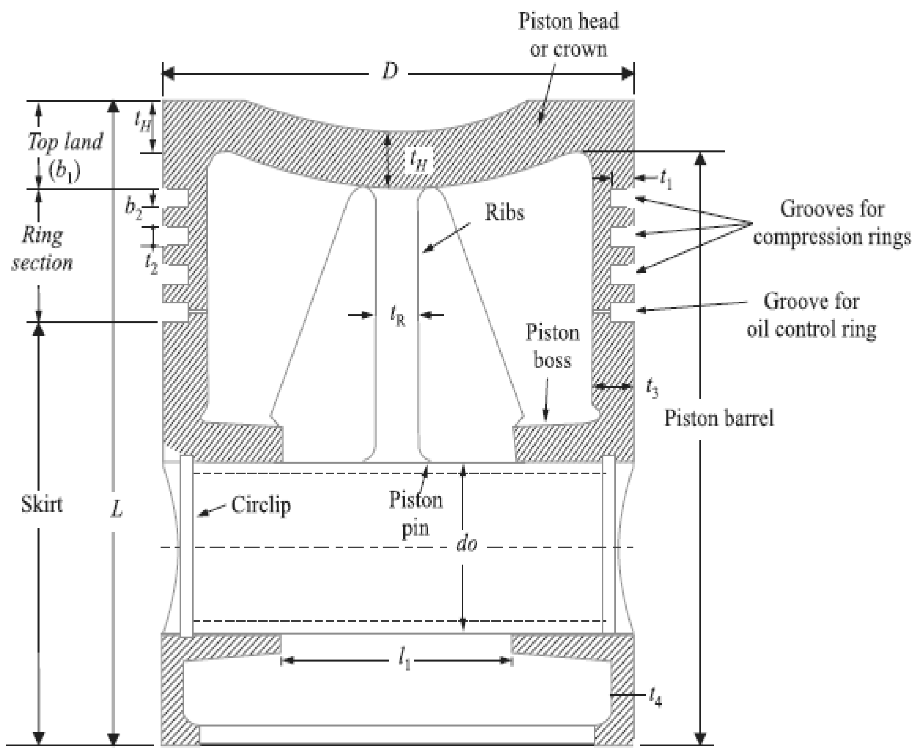
The induced shear stress are less than permissible stress 70N/mm^2 hence design is safe.

D) Total length of piston Pin $L_{p_T} = 0.9D = 0.9 \times 100 = 90\,mm$

E) Location of piston pin center $= \frac{l_1}{2} + 0.03D$ from bottom edge of piston

l_1 = length of piston skirt = 88mm

Location of piston pin center$= \frac{88}{2} + 0.03 \times 100 = 47$ from bottom edge of pistion — 1



2

6 Attempt **any two:**	16
a) Write stepwise design procedure for : I) Cylinder bore II) Cylinder head III) Cylinder head bolts or studs.	8

Answer:

I) Design procedure for Cylinder bore

P_m =Indicated mean effective pressure in N/mm^2

D= cylinder bore in mm

A= cross sectional area of the cylinder in mm^2

$= \Pi\, D^2/4$ — 3

l = length of stroke in meters
N = Speed of the engine in r.p.m.
n = number of working stroke per min
= N, for two stroke engine
= $N/2$ for four stroke engine

Power produced inside the engine cylinder i.e. indicated power

$$I.P. = \frac{P_m \times l \times A \times n}{60} \, watts$$

Form this expression the bore (D) and length of stroke (l) is determined. The length of stroke is generally taken as 1.25 D to 2 D

Since there is a clearance on both sides of the cylinder therefore length of the cylinders is taken length of the cylinder ,

$$\mathrm{L} = 1.15 \times \text{Length of stroke} = 1.15 l$$

II) Design of cylinder head:

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

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

t = thickness of cylinder head

D = diameter of cylinder

C = constant

= 0.1………for C.I.

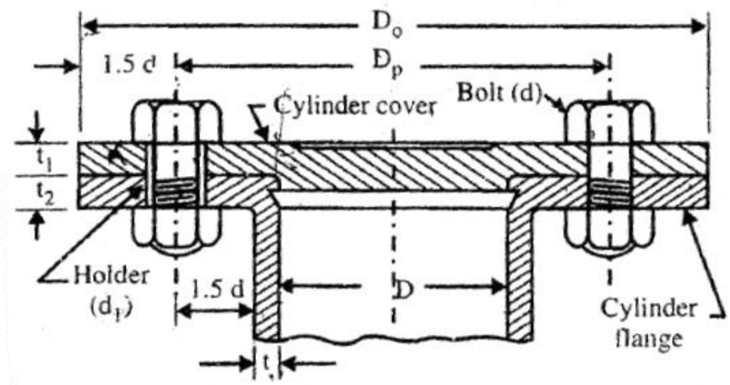
P_{max} = maximum gas pressure inside the cylinder

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.

2

III) Design of 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) 3

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

$$= D + 3d \text{ ……………………….(i)}$$

b) The gas pressure is assumed to be acting on P.C.D. of studs.

$$\therefore \text{ Gas load} = P_{max} \times \left(\frac{\Pi}{4} D_p^2\right)$$

$$P_{max} \times \frac{\Pi}{4}(D + 3d)^2 \text{ …………….(ii)}$$

c) This load is acting as tensile load on bolts or stud and this load is resisted by 'Z' numbers of bolts.

$$P_{max} \times \frac{\Pi}{4}(D + 3d)^2 = Z \times \frac{\Pi}{4} d_c^2 \times f \text{ ………….(iii)}$$

d) Numbers of bolts 'Z' is taken between

$$Z = \left(\frac{D}{100} + 4\right) \text{to} \left(\frac{D}{50} + 4\right) \text{ ………..(iv)}$$

Generally even value is selected for 'Z'

e) Value of 'd' is taken as

$d = \dfrac{d_c}{0.84}$ ………..(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 $\text{Pitch 'p'} = \dfrac{\Pi D_P}{Z}$ 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.	
b) A semi-elliptical leaf spring consists of two full length leaves and eight graduated leaves including master leaf. The effective length of the spring is 1 m and max. force acting on it is 10 kN width of each leaf is 50 mm. The spring is initially preloaded so that stresses induced in each leaf are 350 N/mm2. If modulus of elasticity of spring material is 207000 N/mm2, determine thickness of each leaf, deflection of spring and initial nip. Sketch proportionate fig. of semi-elliptical leaf spring.	8
Answer: **Given data** Max. force $2P = 10kN$, $P = 5000N$ Effective length of the spring $2L = 1m = 1000mm$, $L = 500mm$ $n_f = 2, n_g = 8$ $n = n_f + n_g = 2 + 8 = 10$ Width b = 50mm Modulus of elasticity $E = 207000 N/mm^2$ $\sigma_b = 350\ N/mm^2$ ***1 Thickness of leaves*** Since the stress are equal in all leaves $\sigma_b = \dfrac{6PL}{nbt^2}$ $350 = \dfrac{6 \times 5000 \times 500}{10 \times 50 \times t^2}$ Thickness $t = 9.26mm = 10\,mm$	2

2. Deflection of spring

$$\delta = \frac{12PL^3}{Ebt^3(3n_f + 2n_g)}$$ 2

$$\delta = \frac{12 \times 5000 \times (500)^3}{207000 \times 50 \times 10^3 (3 \times 2 + 2 \times 8)}$$

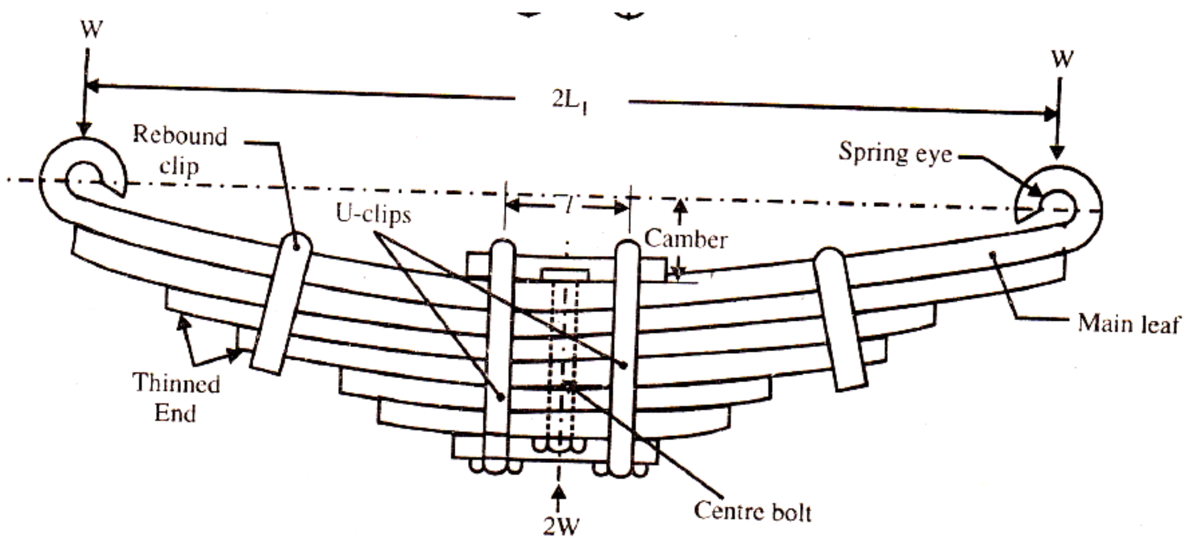
$$\delta = 32.93\ mm$$ 2

3. Initial nip

$$C = \frac{2PL^3}{Enbt^3} = \frac{2 \times 5000 \times 500^3}{207000 \times 10 \times 50 \times 10^3}$$

$$C = 12.07\,mm$$

2



c) Determine dimensions of the cross-section of a connecting rod of an I.C. engine for following data: Cylinder bore = 100 mm, Length of connecting rod= 350 mm
Max. gas pressure= 4 MPa, F.O.S. = 6,Rankies const.=1/7500 .Also find variations in height of cross-section and draw a neat proportionate sketch of connecting rod. 8

Answer:

D=100mm

Max. gas pressure $P_{max} = 4\ N/mm^2$

Length of connecting rod $L = 350mm$

$(fos) = 6$

Step I) Force acting on the connecting rod

$$P_c = \left(\frac{\Pi D^2}{4}\right) P_{max} = \left[\frac{\Pi (100)^2}{4}\right](4) = 31415.93N$$ 1

II) Critical bucking load

$$P_{cr} = P_c(fs) = 31415.96(6\ ) = 188495.58N$$

Step III Calculating of thickness t

Substituting, $A = 11t^2, k_{xx} = 1.78t$

$$a = \frac{1}{7500}, \quad \sigma_c = 330N/mm^2$$

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

$$188495.58 = \frac{(330)(11t^2)}{1 + \frac{1}{7500}\left(\frac{350}{1.78t}\right)^2}$$

$$\frac{188495.58}{(330)(11)} = \frac{t^2}{1 + \frac{5.16}{t^2}} \text{ or}$$

$$51.93 = \frac{t^4}{t^2 + 5.16}$$

$$t^4 - 51.93t^2 - 267.96 = 0$$

The above expression is a quadratic equitation in (t^2)

$$t^2 = \frac{51.93 \pm \sqrt{(51.93)^2 + 4(267396)}}{2}$$

$$= \frac{51.93 \pm 61.39}{2}$$

$$t^2 = 56.66$$

$$t = 7.53 = 8mm$$

3

IV) Dissension of cross section 1

$B = 4t = 4(8) = 32mm$

$H = 5t = 5(8) = 40mm$

Thickness of web = t=8mm

Thickness of flanges = t=8mm

The width (B=32mm) is kept constant throughout the length of connecting rod.

V) Variation of height

At the middle section $H = 5t = 5(8) = 40mm$ 1

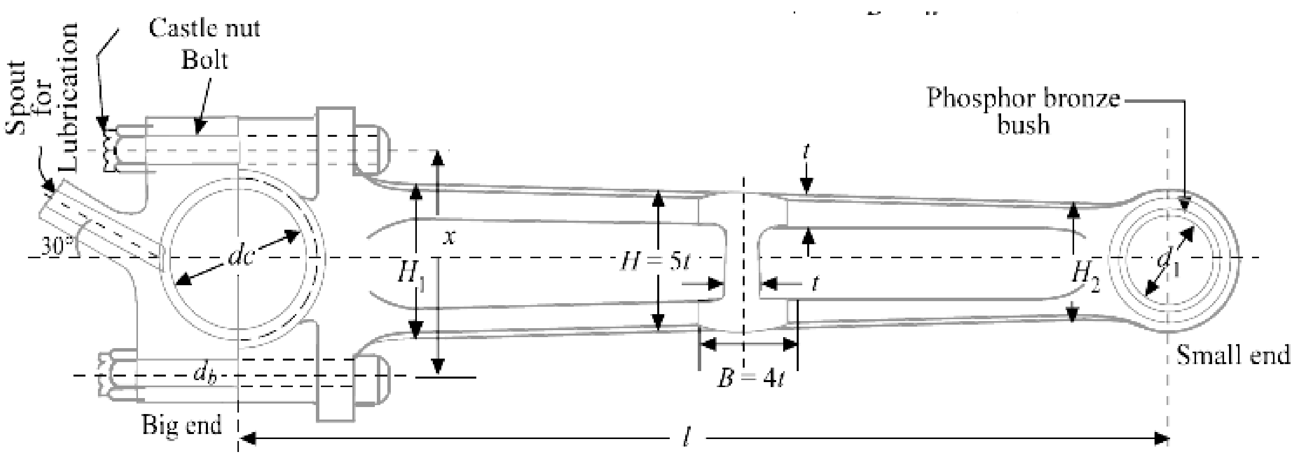
At the small end, $H_1 = 0.85H = 0.85(40)mm$ =34mm

At the big end $H_2 = 1.2H = 1.2(40) = 48mm$

Dimensions (B/H) of section at big end $= 32mm \times 48mm$

Dimensions (B/H) of section at middle $= 32mm \times 48mm$

Dimensions (B/H) of section at small end $= 32mm \times 48mm$



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