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.

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|---|---|
| 1. a) Attempt **any THREE** of the following- | **12** |
| i. Define factor of safety. What factors affects its selection? | 4 |
| Answer:<br>**Factor of Safety:** Factor of safety is defined as the ratio of the maximum stress to the working stress or design stress.<br>Mathematically, Factor of Safety = Maximum Stress/ Working or design stress<br><br>In case of ductile materials-<br><br>$$\text{Factor of safety} = \frac{\text{Yield point stress}}{\text{Working or design stress}}$$<br><br>In case of brittle materials-<br><br>$$\text{Factor of safety} = \frac{\text{Ultimate stress}}{\text{Working or design stress}}$$ | 2 |
| **Selection of factor of safety (Any Four)**<br><br>1. The reliability of the properties of the material and change of these properties during service ;<br>2. The reliability of test results and accuracy of application of these results to actual machine parts<br>3. The reliability of applied load;<br>4. The certainty as to exact mode of failure;<br>5. The extent of simplifying assumptions;<br>6. The extent of localized stresses;<br>7. The extent of initial stresses set up during manufacture;<br>8. The extent of loss of life if failure occurs; | 2 |
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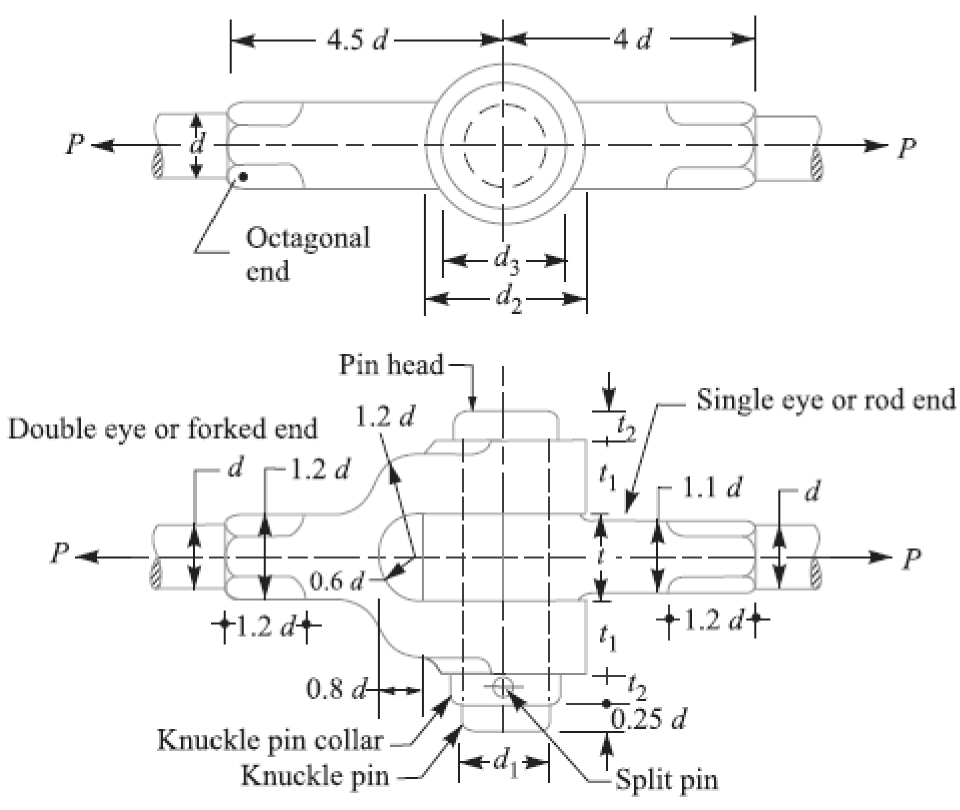
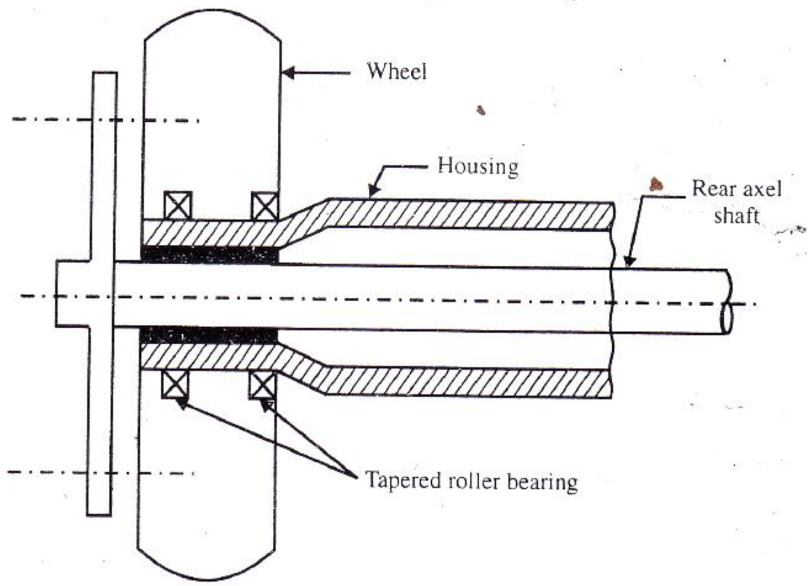
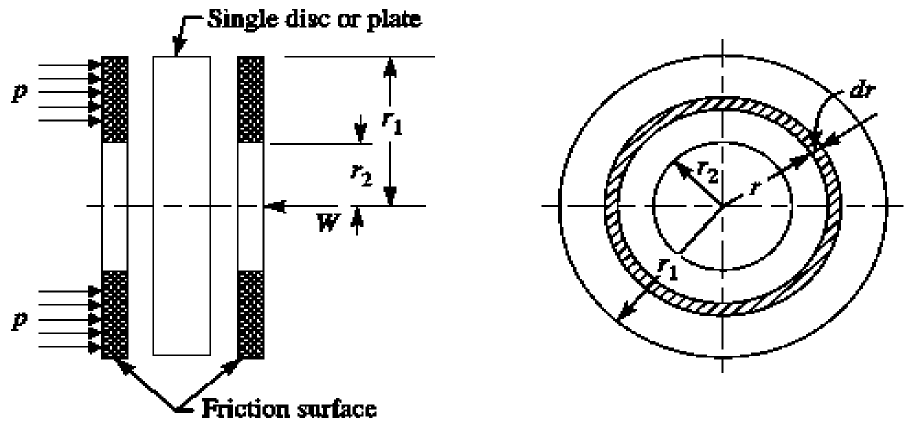
9. The extent of loss of property if failure occurs; and Factor of safety is selected to prevent the failure of material in service.	
ii. Define Stress concentration and state its causes.	4
Answer: **Stress Concentration:** Whenever a machine component changes the shape of its cross section, the simple stress distribution no longer holds good and neighborhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. **OR** Whenever there is a change in cross section of machine components, it causes high localized stresses. This effect is called as stress concentration.	2
Causes of Stress Concentration: (Any Four – ½ Marks Each) **i)** Variation in properties of material from point to point due to cavities, cracks or air pockets. **ii)** Abrupt changes of shape and cross section. **iii)** Concentrated loads applied at points or small areas of machine elements. **iv)** Force flow line is bent as it passes from the shank portion to threaded portion of component due to changes in cross section. This results in stress concentration in transition plane. **v)** Local Pressures	2
iii. Draw labeled sketch of a knuckle joint.	4
Answer: **(Simple sketch – 2Marks, Dimensions with notation-2 Marks)** 	4

iv. Describe the design procedure of a rear axle.	4
Answer: **Design procedure of a rear axle:** The rear axle is desinged on the basis of shaft design. By using the torsional equation, $T_{RA}/J_{RA} = \tau/r$ T_{RA} = Torque transmitted by rear axle shaft. τ = 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)	1
	1
$T_{RA} = T_e \times G_1 \times G_d$ $$P = \frac{2\Pi\ N\ T_e}{60}$$ T_e = Engine Torque. G_1 = Maximum gear Ratio in Gear Box G_d = Final gear reduction in differential J_{RA} = Polar moment of Inertia $= \pi/32 \times d^4$ (for Solid shaft) $= \pi/32 \times (d_o^4 - d_i^4)$ (for Hollow shaft)	1
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 equations, we can find out the diameter of rear axle of shaft.	1

1. b) Attempt **any One** of the following	**12**
i. Derive the relation for torque to be transmitted by single plate clutch considering uniform wear condition.	6

Answer:

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



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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,

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

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, 1

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

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,

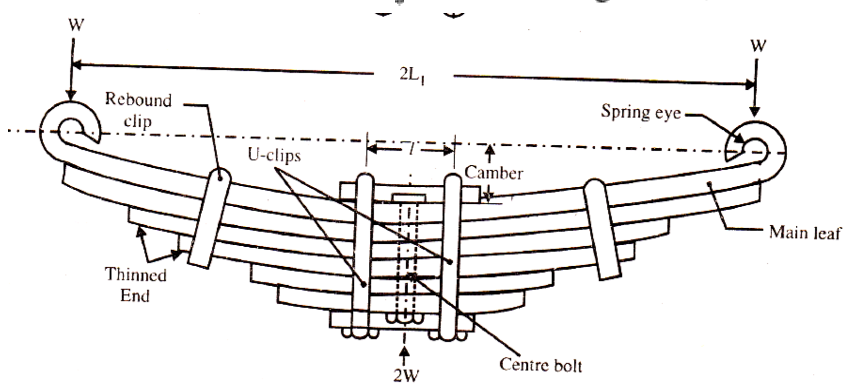
$$\delta W = p.2\pi r.dr = \frac{C}{r} \times 2\pi r.dr = 2\pi C.dr$$

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

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

or

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

We know that the frictional torque acting on the ring, $T_r = 2\pi\,\mu.p.r^2.dr = 2\pi\,\mu \times \frac{C}{r} \times r^2.dr = 2\pi\mu.C.r.dr$ ...($\because p = C/r$) $\therefore$ 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 $R = \frac{r_1 + r_2}{2}$ = Mean radius of the friction surface.	3
ii. Explain how a semi-elliptical leaf spring is designed.	6
Answer: Let, 2W = Central load 2L = Span of spring b = Width of leaves t = Thickness of leaves n = Total number of leaves l = Length of central band n_f = Number of full length leaves n_g = Number of graduated leaves  	2
Design steps for calculating thickness of Leaf Spring: (any two steps - 1 marks each) (1) Stress in leaf spring : $\sigma_b = \frac{6\,WL}{n\,b\,t^2}$ where, Effective length of spring $= 2L = 2L_1 - l$ (when central band is used) $= 2L = 2L_1 - \frac{2}{3} \cdot l$ (when U-bolt is used)	2

(2) Deflection in leaf spring :

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

(3) Stress in full length leaves :

$$\sigma_F = \frac{18\,WL}{bt^2\,(2n_g + 3n_f)}$$

(4) Stress in graduated leaves :

$$\sigma_G = \frac{12\,WL}{bt^2\,(2n_g + 3n_f)}$$

(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: (2 marks) 2

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 = $2L_1$ + (π (d+t) x 2

Where d = diameter of Eye.

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

2. Attempt **any FOUR** of the following	16
a) Define the term: i) Fatigue ii) Endurance limit with suitable example	4

Answer: (2 Marks Each)

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. 2

OR

ASTM Definition of fatigue

- The process of ***progressive localized permanent*** structural changes occurring in a material subjected to conditions that produce ***fluctuating stresses*** at some point or points and that may culminate in ***cracks*** or complete ***fracture*** after a sufficient number of fluctuations.

ii) Endurance limit with suitable example:

It is defined as maximum value of the completely reversed bending stress which a polished standard specimen can withstand without failure, for infinite number of cycles (usually 10^7 cycles). 2

OR

Endurance or fatigue limit can be defined as the magnitude of stress amplitude value at or below which no fatigue failure will occur, no matter how large the number of stress reversals are, in other words leading to an infinite life to the component or part being stressed. For most ferrous materials Endurance limit (Se) is set as the cyclic stress level that the material can sustain for 10 million cycles.

b) Mention the applications of cotter joint, knuckle joint and turn buckle.	4
Answer:(any Eight applications irrespective of type of joint- 1/2 mark each) i) Applications of Cotter joint: • Connecting a piston rod to cross head of steam engine • Joining a tail rod with piston rod of an air pump • Valve rod and its stem. ii) Applications of Knuckle joint: • Link of cycle chain • Tie rod joints for roof truss • Valve rod joint for eccentric rod pump rod joint • Tension link in bridge structure • Lever and rod connection of various types. • swing arm of two wheeler • Connection of link rod of leaf springs in multi axle vehicles • Piston ,Piston Pin ,Connecting Rod • Connections of leaf spring with chassis iii) Applications of Turn Buckle: • Tie rod of steering system • To connect compartments of locomotives • Tie strings of electric poles. • link rod of leaf springs in multi axle vehicles • linkages of gear shifter • Connection between brake pedal and master cylinder	4
c) State the different types of levers with applications	4
Answer: **(Types any four– 1 Marks Each)** **i) First Type Lever:** **Application:** Rocker arm in internal combustion engines, beam of a balance, foot lever etc. **ii) Second Type Lever:** **Application:** The application of such type of levers is found in levers of loaded safety valves, hand lever **iii) Third Type Lever:** **Application:** Pair of tongs, the treadle of a sewing machine etc. are examples of this type of lever, arm of JCB **iv) Also following are types of lever based on included angle between arms** Acute angle lever ,obtuse angle lever ,bell crank lever **v) When two first type of levers are connected to get amplified mechanical advantages its compound lever e.g nail cutter**	4

d) Give in brief design procedure of a bell crank lever.	4

Answer:

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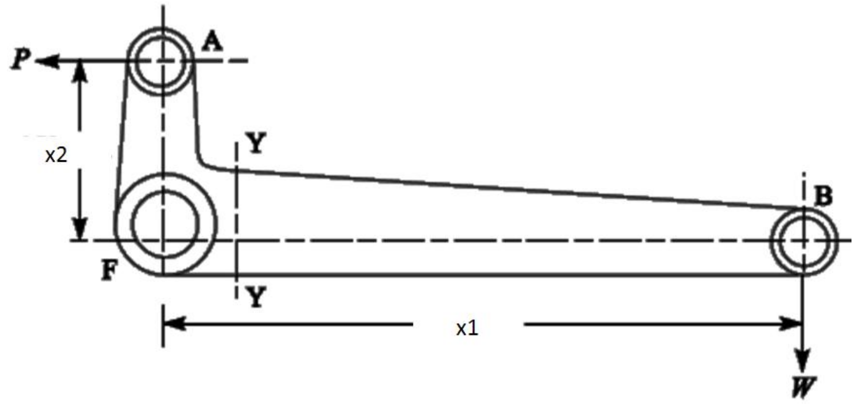


Fig.bell crank lever

Step I Calculate the reaction at fulcrum P_F

$W \times X_1 = P \times X_2$

obtaining value of P

determine R_F 1

$R_F = \sqrt{W^2 + P^2}$

Step II

Design of Fulcrum Pin

d= Dia of fulcrum pin

l= Length of fulcrum pin

we know that load on fulcrum pin

A) $\therefore \mathrm{Pb} = \dfrac{\text{Load}}{\text{Bearing Area}}$

$\mathrm{Pb} = \text{Bearing Pressure in N/mm}^2$

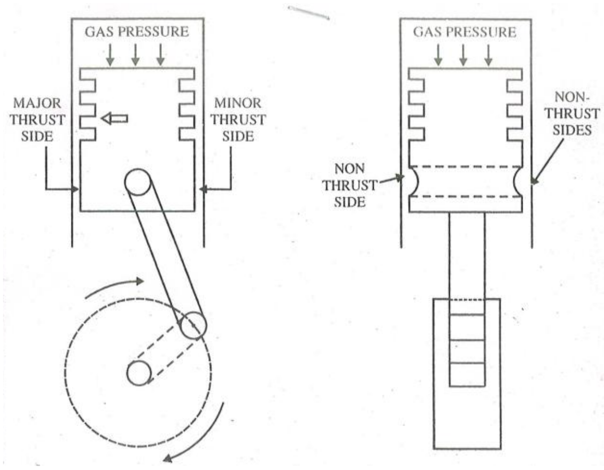
$\mathrm{Pb} = \dfrac{R_F}{l \times d}$

$R_F = \mathrm{Pb} \times l \times d$

Assuming l = 1.25d here l and d can be determined . **B) Checking induced shear stress in pin** Pin is in double shear $$\tau = \frac{R_F}{2 \times \left(\frac{\Pi}{4} \times d^2 \right)}$$	1
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 $$Pb = \frac{w}{d_2 \times l_2}$$ $l_2 = 1.25 d_2$ Here d_2 and l_2 can be determined checking the pin for shear stress	

$\tau = \dfrac{w}{2\times\left(\dfrac{\Pi}{4}\times d_2^2\right)}$	
thick of each eye = $t_1 = \dfrac{l_2}{2}$	
inner Dia of each eye = $d_2 + 2\times 3$	1
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	
$\dfrac{1}{6}\times t\times b^2$ Assume $(b = 3t)$	
$= \dfrac{1}{6}\times t\times(3t^2)$	
$\sigma_b = \dfrac{m}{z} = \dfrac{w(x_1 - a)}{\dfrac{1}{6}\times t\times(3t^2)}$	1
here t and b can be determined.	

e) Draw a neat sketch of piston showing thrust and non-thrust side.	4
Answer: (Each view 2 marks)	4



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

3. Attempt **any FOUR** of the following **:**.	16
a) A knuckle joint in required to with stand a tensile load of 25 kN. Design the joint if the permissible stresses are 56 N/mm^2, 40 N/mm^2 and 70 N/mm^2 in tension shear and crushing respectively (any two failure)	4

Answer:

$P=25000\ N = 25x10^3\ N$

$\sigma_t = 56 N/mm^2$

$\tau = 40\ N/mm^2$

$\sigma_c = 70 N/mm^2$

Failure of solid rod in tension:

let d = Diameter of rod,the rod

we know load transmitted (P)

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

$$25x10^3 = \frac{\pi}{4} x d^2 x\ 56$$

$$d^2 = \frac{25\ X10^3}{43.98}$$

d= 23.84 mm say **d=24mm** 1

From empirical relation

Dia. of Pin $d_1 = d = 24$ mm

Outer Dia. of single or double eye $d_2 = 2d = 48$ mm 1

Thickness of single eye $t = 1.25d = 1.25 \times 24 = 30$ mm

Thickness of double eye $t_1 = 0.75\,d = 0.75 \times 24 = 18$ mm

(Consider any two failures of either single, double eye or pin- 1 Mark for each failure**)**

1) **Failure of Single eye in tension**

$$P = (d_2 - d_1)\, t \times \sigma_t$$

$$25 \times 10^3 = (48 - 24)\, 30 \times \sigma_t$$

$$\sigma_t = 34.72\ \text{N/mm}^2$$

Induced stress is less than given stress 56 N/mm^2. Hence design is safe.

2) **Failure of single eye in crushing**

$$P = d_1 \times t \times \sigma_c$$

$$25 \times 10^3 = 24 \times 30 \times \sigma_c$$

$$\sigma_c = 34.72\ \text{N/mm}^2$$

Induced stress is less than given stress 70 N/mm^2. Hence design is safe.

3) **Failure of Single eye in shearing**

$$P = (d_2 - d_1)\, t \times \tau$$

$$25 \times 10^3 = (48 - 24)\, 30 \times \tau$$

$$\tau = 34.72\ \text{N/mm}^2$$

Induced stress is less than given stress 40 N/mm^2. Hence design is safe.

b) State various proportions of a rectangular sunk key with its neat sketch. 4

Answer: **Rectangular Key:** ***(Proportions- 2 Mark, Figure - 2 Mark)*** 4

(Parallel Sunk Key Should be equally considered)

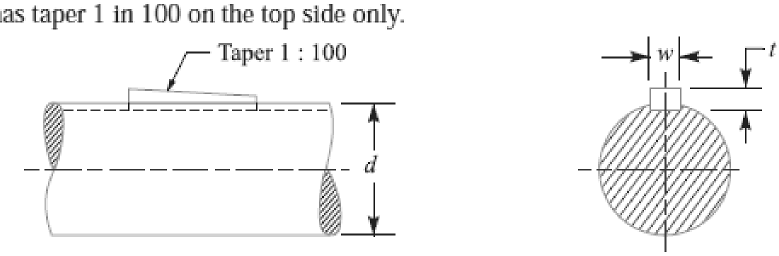
The usual proportions of this key are

Length of the key L = 1.5d

Width of key, $w = d/4$; and thickness of key, $t = 2w/3 = d/6$

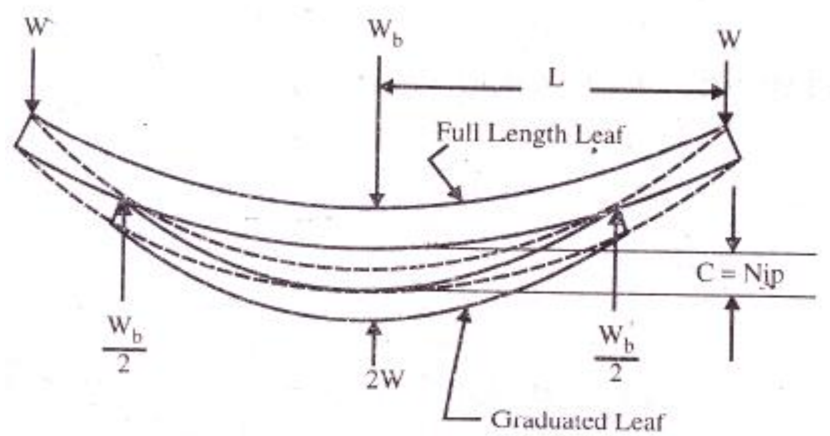
where d = Diameter of the shaft or diameter of the hole in the hub.

The key has taper 1 in 100 on the top side only.



c) Two mild steel rods of 40 diameter are to be connected by a cotter joint the thickness of cotter is 12 mm. Calculate the dimensions of the socket if the maximum allowable stresses are 46 N/mm^2 in tension 35 N/mm^2 in shear and 70 N/mm^2 in crushing.	4
Answer: $P = \Pi/4 \times d^2 \times \sigma_t$ $P = \Pi/4 \times 40^2 \times 46$ $P = 57.8 \times 10^3$ N	1
Inside Dia. of Cotter $P = d_2 \times t \times \sigma_c$ $57.8 \times 10^3 = d_2 \times 12 \times 70$ $d_2 = 68.8$ mm $d_2 = 70$ mm	1
Outside Dia. of Cotter $P = \left[\Pi/4 (d_1^2 - d_2^2) - [(d_1 - d_2) t] \right] X \ \sigma_t$ $57.8 \times 10^3 = \left[\Pi/4 (d_1^2 - 70^2) - [(d_1 - 70) 12] \right] X \ 46$ $57.8 \times 10^3 / 46 = 0.785\, d_1^2 - 0.785 \times 4900 - 12\, d_1 + 840$ $0 = 0.785\, d_1^2 - 12\, d_1 - 4263.021$ $0 = d_1^2 - 15.28\, d_1 - 5430.6$ By using quadratic equation we get, $d_1 = 81.72$ mm $d_1 = 82$ mm	2
d) Explain indicated power and brake power of an engine cylinder.	4
Answer: **Indicated Power** The power developed inside the cylinder is known as indicated power. It is called as indicated power because it is measured from indicator diagram. $ip = \frac{p_{im} \ LAnk}{60000} kW$ where	1

ip = indicated power (kW) p_{im} = indicated mean effective pressure (N/M^2) L= Length of the stroke (m) A= area of the piston (m^2) N= speed in revolutions per minute n= number of power strokes engine N/2 for a four – stroke engine N for a two –stroke engine K= number of cylinders **Brake Power:** This is the actual power delivered at the crankshaft. It is obtained by deducting various power losses in the engine from indicated power. Brake power is what would keep the vehicle running at any speed once you have accelerated B.P. (in kW) can be calculated with the formula B.P.= 2Π NT/ 60000 Where N= Engine speed in R.P.M. T= Torque in newton meters **OR** $bp = \frac{p_{im}\ LAnk}{60000} kW$ Where p_{im} = Brake mean effective pressure	1 1 1
e) Design the piston crown thickness from the following data-diameter of piston = 80mm.Maximum pressure on the piston = 4.5 N/mm^2 and allowable bending stress = 45 N/mm^2.	4
Answer: **Given** D = 80 mm, P = 4.5 N/mm^2 , σ_b = 45 N/mm^2 Let the thickness of piston head can be designed by assuming the head to be a flat plate uniform thickness and fixed at the edges and assuming the gas load to be uniform distributed $t_h = \sqrt{\frac{3PD^2}{16\sigma_b}}$ where t_h = thickness of piston crown P = P_{max}	1

$t_h = \sqrt{\dfrac{3\times 4.5\times 80^2}{16\times 45}}$ $t_h = 10.95$ mm $t_h = 11$ mm	3
4. a) Attempt **any THREE** of the following.	
i) Explain why nipping of leaf spring is necessary with neat sketch.	4
Answer: 	2
When the central bolt holding the leaves is tightened, the full length leaf bend back as shown by dotted line. And will have an initial stress in opposite direction. The graduated leaves will have an initial stress in the same direction as that of normal load. When the load is applied, the full length leaf gets relieved first, consequently the full length leaf will be stressed less than graduated leaf. The initial leaf between leaves may be so adjusted that under maximum load conditions, all the leaves are equally stressed.	2
ii) Explain design of piston pin on the basis of bearing pressure and shear strength.	4
Answer: **Design on the basis of bearing Pressure:** Let d_0 = Outside diameter of the piston pin in mm l_1 = Length of the piston pin in the bush of the small end of the connecting rod in mm. Its value is usually taken as 0.45 D. p_{b1} = Bearing pressure at the small end of the connecting rod bushing in N/mm². Its value for the bronze bushing may be taken as 25 N/mm². We know that load on the piston due to gas pressure or gas load $= \dfrac{\pi\ D^2}{4} \times p$ ..(*i*) and load on the piston pin due to bearing pressure or bearing load = Bearing pressure × Bearing area = $p_{b1} \times d_0 \times l_1$, ...(*ii*) From equations (*i*) and (*ii*), the outside diameter of the piston pin (d_0) may be obtained.	2
Design on the basis of Shear Strength: Gas load = 2 Π/4 $d_0^2 \tau$ ………………… ……...for solid pin Gas load = 2 Π/4 $(d_0^2 - d_i^2)\ \tau$ ………………… for hollow pin	2

iii) Write four methods of failure in terms buckle's design.	4
Answer: (Any four failure – 1 Mark each)	4

Let W = Design load = 1.3 x load carried by the rods (P).

τ = Permissible shear stress
σ_t = Permissible tensile stress
σ_c = Permissible crushing stress

Failure of rod in tension:
The rod may fail in tension due to load W,
We know that,

Area resisting tearing = $\pi / 4 \times (d_c)^2$

Tearing strength of rods = $W = \pi / 4 \times (d_c)^2 \times \sigma_t$

Hence the value of d_c can be determined.

From the standard table, value of nominal diameter d_0 & corresponding pitch can be determined or any other empirical formula to find out nominal diameter.

Shear failure of threads at their roots:

Area resisting shearing = $\pi \times d_c \times l$

$W = \pi \times d_c \times l \times \tau$

Hence the value of 'l' can be determined.

But in actual practice, length of coupler nut (l) can be taken as,

d_0 to $1.25d_0$ for steel & $1.5d_0$ to $2d_0$ for cast iron.

The tensile failure of coupler nut:

Outside diameter (D) of coupler nut is found by considering tensile failure,

$\sigma_t = W / (\pi/4 \times (D^2 - d_0^2))$

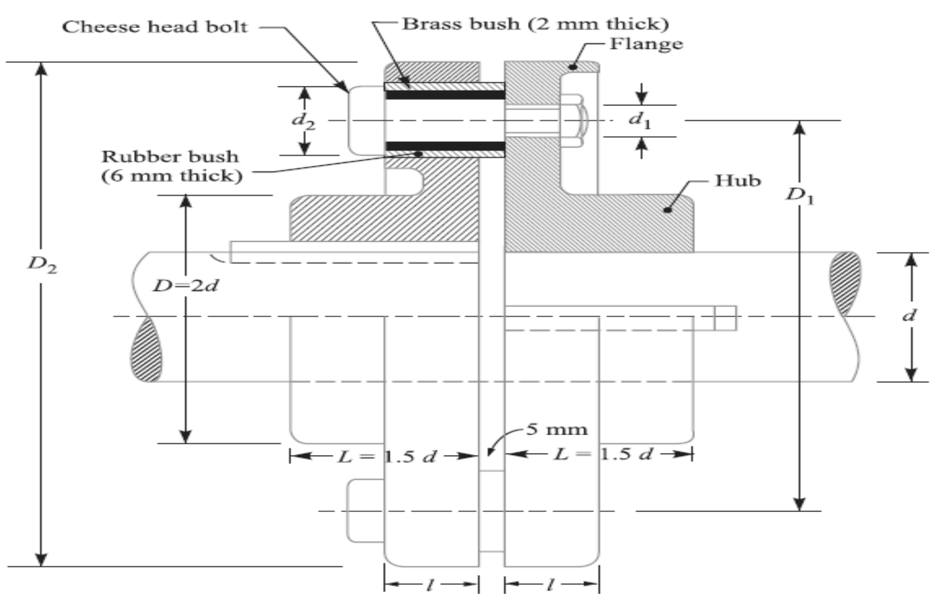
The tensile failure of coupler:

Outside diameter (D_2) of coupler nut is found by considering tensile failure,

$\sigma_t = W / (\pi/4 \times (D_2^2 - D_1^2))$

Where, inside diameter of coupler = $D_1 = d_0 + 6$

In practice, the outside diameter of coupler is taken as $1.5d_0$ to $1.7d_0$
Length of coupler = 6 d_0 & Thickness of coupler = t = $0.75d_0$

iv) List different type of coupling and explain any one.	
Answer: **Types of coupling: (Any four type – 02 Marks, Explanation of any one – 02 Marks)** 1.Rigid Coupling a. Flange coupling: i) Protected ii) Unprotected 2. Flexible coupling : a. Bushed pin type coupling	
b) Attempt **any** **ONE** of the following **:**	6
i) Explain ergonomic considerations in designing automobile components.	6
Answer: 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.	2
Physiology: With the help of physiology components are designed to be operated by hand or foot force. 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.	2
Psychology: Psychology affects mode of operation for e.g. size, color 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.	2
ii) Write stepwise design procedure for a bushed pin flexible coupling	6
Ans: (Neat fig. – 2 Marks, Any four steps – 1 Mark each) 	2

Design of Shaft: $P = 2\Pi NT/60$ $T = \Pi/16\, \tau\, d^3$	4
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. $= 3d$ 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.

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

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.

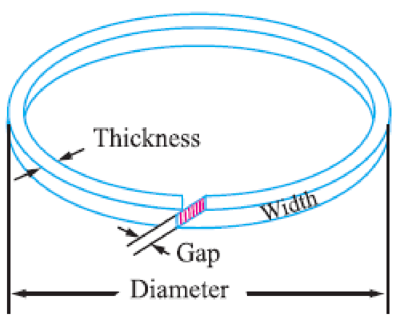
5. Attempt **any <u>TWO</u>** of the following **:**	16
a) Explain following theories of failure. (i) Maximum principal stress theory. (ii) Maximum shear stress theory	8

Answer: i)**Maximum Principal or Normal Stress Theory (Rankine's Theory)**

According to this theory, the failure or yielding occurs at a point in a member when the maximum principal or normal stress in a bi-axial stress system reaches the limiting strength of the material in a simple tension test.

$\sigma_{t1} = \sigma_{yt}/F.S$, for ductile materials

$= \sigma_u/F.S$, for brittle materials Where σ_{yt} = Yield point stress in tension as determined from simple tension test, and σ_u = Ultimate stress. Since the maximum principal or normal stress theory is based on failure in tension or compression and ignores the possibility of failure due to shearing stress, therefore it is not used for ductile materials. However, for brittle materials which are relatively strong in shear but weak in tension or compression,this theory is generally used. **ii) Maximum Shear Stress Theory (Guest's or Tresca's Theory)** According to this theory, the failure or yielding occurs at a point in a member when the maximum shear stress in a bi-axial stress system reaches a value equal to the shear stress at yield point in a simple tension test. Mathematically, $\tau_{max} = \tau_{yt} / F.S.$ ...(*i*) where τ_{max} = Maximum shear stress in a bi-axial stress system, τ_{yt} = Shear stress at yield point as determined from simple tension test, *F.S.* = Factor of safety. Since the shear stress at yield point in a simple tension test is equal to one-half the yield stress in tension, therefore the equation (*i*) may be written as $\tau_{max} = \sigma_{yt}/2 * F.S$ This theory is mostly used for designing members of ductile materials.	4 4
b) A hollow propeller shaft of a car with outside diameter of 75mm transmits 22.5kW at 1500 rpm to the wheels which are 900 mm in diameter. If the allowable shear stress is 60 N/mm^2, find out the cross-section of shaft. Take gear box reduction 5.	8
Answer: Given data $d_0 = 75mm$ $\tau = 60 N/mm^2$ P = 22.5kW $P = 22.5kW = 22.5 \times 10^3 W$ Gear reduction $G_1 = 5$ Now torque produced by the engine 'T_e' $P = \frac{2\Pi N T_e}{60}$ $22.5 \times 10^3 = \frac{2 \times 3.14 \times 1500 \times T_e}{60}$ $T_e = 143.24 \times 10^3$ N-mm Now torque transmitted by the propeller shaft 'T_p' we know that $T_p = T_e \times G_1$ $= 143.24 \times 10^3 * 5$ N-mm $T_p = 716.2 \times 10^3$ N-mm	3 2

For hollow shaft Let d_0 = outer diameter of shaft d_i = inner diameter of shaft $k = \frac{d_i}{d_0} = \frac{d_i}{75}$ We know that $T_P = \frac{\Pi}{16}\tau(d_0)^3(1-k^4)$ $716.2\times10^3 = \frac{3.14}{16}\times60\times(75)^3(1-k^4)$ $1-k^4=0.14$ $k^4=0.855$ $\frac{(d_i)^4}{(75)^4}=0.855$ $d_i = 72.1\text{mm}$	3
c) Write design calculation for piston rings.	8
Answer: Design of Piston Rings  **Fig.** Piston rings. The radial thickness (t_1) of the ring may be obtained by considering the radial pressure between the cylinder wall and the ring. From bending stress consideration in the ring, the radial thickness is given by $t_1 = D\sqrt{\frac{3p_w}{\sigma_t}}$ 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 2 2

$t_2 = \frac{D}{10\, n_R}$ where n_R = Number of rings. Width of top land, $b_1 = t_H$ to $1.2\, t_H$ 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 $4\, t_1$.	1 1 1
6. Attempt any <u>TWO</u> of the following.	16
<u>a</u>) A single plate dry clutch transmits 7.5 kW at 900 rpm, the axial pressure is limited to $0.7\ \text{N/mm}^2$. If the co-efficient of friction is 0.25, find (i) Mean radius and face width of friction lining assuming ratio of mean radius to face width as 4 (ii) Outer and inner radii of the clutch plate.	8
Answer :Given Data n= 2, Power $P = 7.5\text{kW} = 7.5\times10^3\,\text{W}$ N= 900 rmp Co-efficient of friction $\mu = 0.25$ Maximum intensity of pressure, $P_{max} = 0.7\text{N/mm}^2$ Let , r_1 and r_2 = outer and inner radius of frictional surfaces respectively r = mean radius of the friction lining in mm b= face width of friction lining Ratio of mean radius to the face width $\frac{r}{b} = 4$ We know that area of frication faces $= 2\,\Pi rb \times P$ Therefore normal or axial force acting on friction faces $W = A \times P$ $= 2\Pi rb \times P$ Torque transmitted, $T = n\,\mu\,W\,r(\text{ unifrom wear})$ $= n\,\mu\,(2\Pi\, r\, b \times P)\, r$ $= n\,\mu \left[2\,\Pi\, r \times (r/4) \times P\right] r$	 1

$= (\prod/2)\ n\ \mu\ P\ r^3$ $= (\prod/2) \times 2 \times 0.25 \times 0.07 \times r^3$ $= 0.5497\ r^3\ N\text{-}mm$ ……………………(1) Power transmitted $P = \frac{2\Pi\ N\ T}{60}$ $7.5 \times 10^3 = \frac{2 \times 3.14 \times 900 \times T}{60}$ $T = 79.56\ N\text{-}m$ $T = 79.56 \times 10^3\ N\text{-}mm$ ……………….(2) From equations (1) and (2), $r^3 = \frac{79.56 \times 10^3}{0.5497} = 144.733 \times 10^3$ $r = 52.49 = 53$ mm approx. Now face width of friction lining $b = \frac{r}{4}$ …………given $b = \frac{53}{4} = 13.25\ mm$ we know that , $b = r_1 - r_2 = 13.25\ mm$ ……………………….(3) $r = \frac{r_1 + r_2}{2}$ $r_1 + r_2 = 2r = 2 \times 53 = 106mm$ …………….(4) Equating equations (3) and (4) $r_1 = 79.5\ mm$ $r_2 = 26.5\ mm$	2 2 1 2
b) Explain design procedure of a connecting rod	8
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.

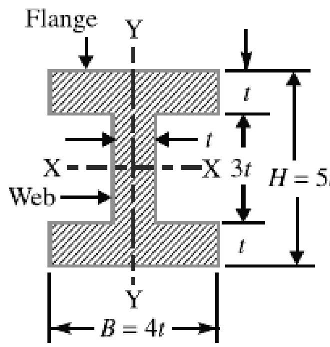


Fig a I-section of connecting rod.

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.

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 $1.25H$. 3

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

Maximum gas force,

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

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

$= d_c \,.\, l_c \,.\, pb_c$ (ii)

2

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

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

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

1

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,

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

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

2

From this expression, the value of t_c is obtained.

c) An automotive gear box gives three forward and a reverse speed top gear of unity and bottom and reverse gear ratio of 3.3:1 the center distance between shafts is 110 mm approximately. Gear teeth of module 3.25mm are to be employed. Determine different gear ratios. 8

Ans- Given data- Module=3.25 mm Centre distance between shafts =110 mm Since the pitch is same for all wheels and the centre distance same for all the pairs meeting the wheels, the total no. of teeth must be same for each pair Therefore $T_A + T_B = T_C + T_D = T_E + T_F = T$ $T = (2 \times C.D.) / M$ $T = (2 \times 110) / 3.25 = 67.8$ T = 68 teeth	1
In general practice for better results gear ratios are kept in geometric progression if G_1, G_2, G_3 are gear ratios in first, second and third gear ,then $\frac{G_1}{G_2} = \frac{G_2}{G_3}$ $G_2 = \sqrt{G_1 \times G_2}$ $= \sqrt{1 \times 3.3}$ = 1.817 In general practice, while designing a gear box it is desired that the gear ratio should be minimum possible in all cases, so that the sizes of the gear box can be kept minimum. To achieve this, the maximum reduction required in gear box (in first gear) is achieved in two equal steps. Now $G_1 = \frac{T_B}{T_A} \times \frac{T_D}{T_C}$ Let X be the reduction in one step therefore $G_1 = X \times X$ $3.3 = X^2$ X = 1.817 $\frac{T_B}{T_A} = \frac{T_D}{T_C} =$ 1.817 $T_B = 1.817\ T_A$ and $T_D = 1.817\ T_C$ $T_A + T_B = 68$, $T_A + 1.817\ T_A = 68$, $T_A = 68/2.817$ $T_A = 24$ teeth	3

T_B= 68-24 T_B=44 teeth Similarly T_C=24 teeth and T_D=44 teeth Actual gear ratio $G_1 = \frac{T_B}{T_A} \times \frac{T_D}{T_C}$ $G_1 = \frac{44}{24} \times \frac{44}{24}$ G_1=3.36:1 Second gear ratio $G_2 = \frac{T_B}{T_A} \times \frac{T_F}{T_E}$ $1.817 = \frac{44}{24} \times \frac{T_F}{T_E}$ $T_F = 0.991\, T_E$ $T_E + T_F = 68$ $T_E + 0.991\, T_E = 68$ T_E =34 teeth T_F=68 - 34 = 34 teeth	1
Actual gear ratio in second gear = $G_2 = \frac{44}{24} \times \frac{34}{34} = 1.833$ Top gear ratio G_3 =1:1 is obtained by directly coupling the main shaft to engine shaft. Now reverse gear and idler is fitted due to presence of which the direction is reversed also it is required that $T_D + T_I < 68$ to avoid interference of gear C and D Therefore $T_i < 68 - T_D$ $T_i < 68 - 44$ $T_i < 22$	2
Exact gear ratio $G_R = \frac{T_B}{T_A} \times \frac{T_D}{T_I}$ $G_R = \frac{44}{24} \times \frac{44}{20}$ G_R =4.03:1 Exact center distance = $C.D. = \frac{M(TA + TB)}{2}$ $= \frac{3.25(68)}{2}$ = 110.5 mm	1