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) Write the design procedure of designing any machine element | 4 |
| **Answer: (1/2 Marks each)**<br>**Following procedure is carried out for designing any machine element:**<br>i. State the purpose for which the machine element is to be designed.<br>ii. Make the sketch of the machine element showing its use.<br>iii. Find the forces acting upon machine element.<br>iv. Select the suitable material for the machine element.<br>v. Find the factor of safety and allowable stresses of the material.<br>vi. Consider the various manufacturing process available for the manufacturing.<br>vii. Calculate the size of each member of the machine element by finding out the forces acting upon them during the working of machine element.<br>viii. Make an assembly drawing & make a detail drawing of each component of the job showing dimensions, manufacturing accuracy, surface finish and other data related to its manufacturing. | 4 |
| ii) Define standardization and state the four advantages of it. | 4 |
| Answer:<br>**Standardization: ( 2 marks)**<br>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. | 2 |
| **Following are the four advantages of Standardization: (Any four – ½ marks each)**<br>i. Interchangeability of product or element is possible.<br>ii. Mass production is easy.<br>iii. Rate of production increases.<br>iv. Reduction in labour cost. | 2 |
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v. Limits the variety of size and shape of product.
vi. Overall reduction in cost of production.
vii. Improves overall performance, quality and efficiency of product.
viii. Better utilization of labour, machine and time.

iii) Define factor of safety for ductile material and state the four factor consider while selecting factor of safety. 4

Answer:

Factor of Safety: (2 Marks)

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- 2

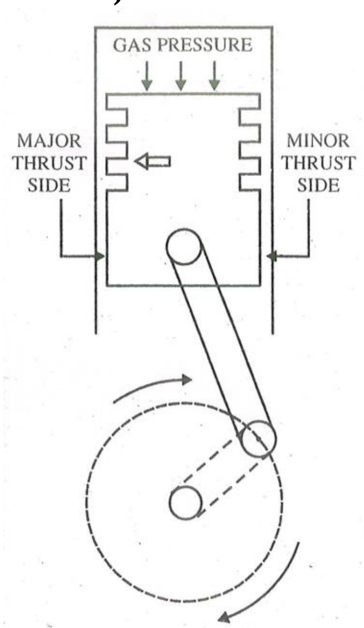
$$\text{Factor of safety} = \frac{\text{Yield point stress}}{\text{Working or design stress}}$$

Following factors affects selection of factor of safety: (Any Four- ½ marks each)

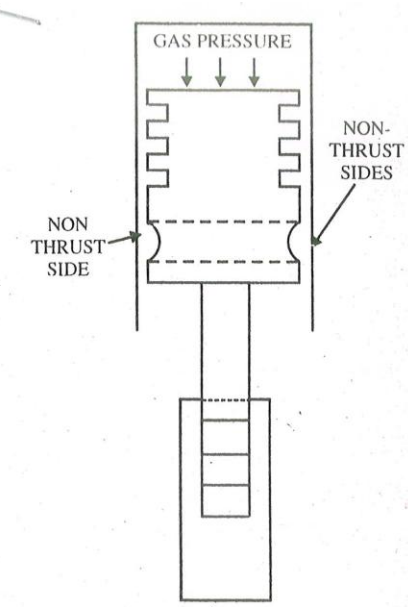
i. The reliability of applied load and nature of load;
ii. The reliability of the properties of material and change of these properties during service;
iii. The reliability of test results & accuracy of application of these results to actual machine parts;
iv. The certainty as to exact mode of failure; 2
v. The extent of simplifying assumptions;
vi. The extent of localized stresses;
vii. The extent of initial stresses setup during manufacture;
viii. The extent of loss of property if failure occurs;
ix. The extent of loss of life if failure occurs.

iv) Show the thrust side and non thrust side of I.C. engine piston with neat sketch. 4

Answer: **(2 Marks each sketch)** 4



Thrust side of I.C. Engine piston



Non-thrust side of I.C. Engine piston

b) Attempt **ANY ONE** of the following	06
i) What is stress concentration? What are the different causes of stress concentration and explain the remedies of it with neat sketches.	06

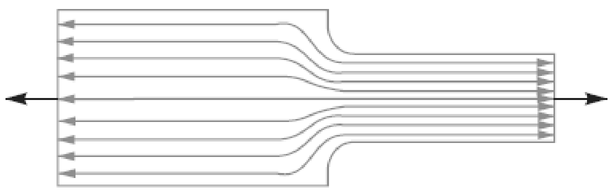
Answer:

Stress Concentration: (2 Marks)

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. 2
- **v)** Local Pressures

Reduction of stress concentration- (Any two with Figure – 1 Marks Each)

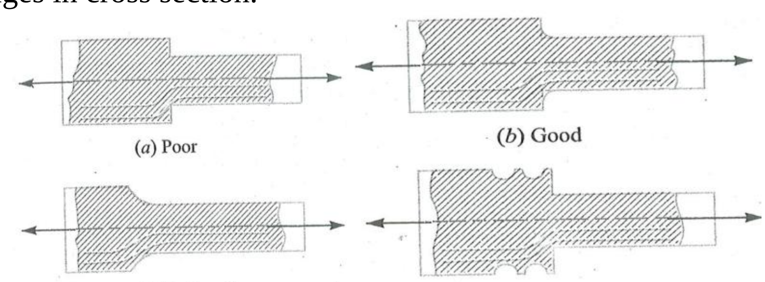
i) Stepped Shaft:

Remedies:

Avoid abrupt changes in cross section

Provide fillet when change in cross section if necessary. 2

Make gradual changes in cross section.



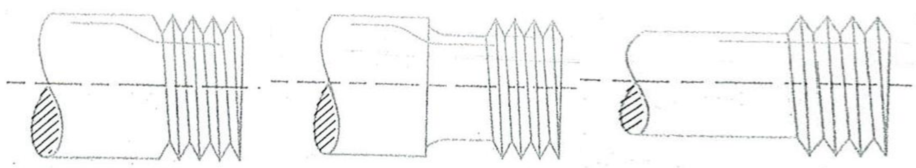
(a) Poor (b) Good (c) Preferred (d) Preferred

ii) Cylindrical members with threads

Remedies:

Small under cut is taken between shank and the threaded portion of the component and a fillet radius is provided for this under cut.

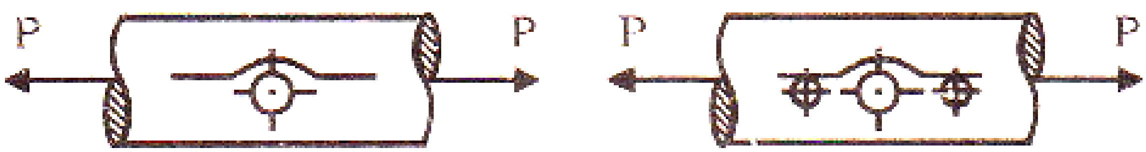
Shank diameter is reduced and made equal to the core diameter of the thread.



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

iii) Cylindrical member with hole

Remedies:
Member should with be provided with small hole near large hole

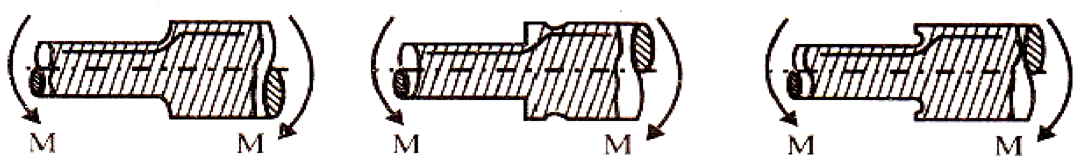


(a) Poor (b) Preferred

iv) Cylindrical member with shoulders subjected to bending moment:

Remedies:

Member should be provided with notches near the corners



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

ii) A single side plate 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 to exceed 0.1 N/nm^2. If the co-efficient of friction is 0.3 determine the power transmitted by a clutch at a speed of 2500 rpm. — 6

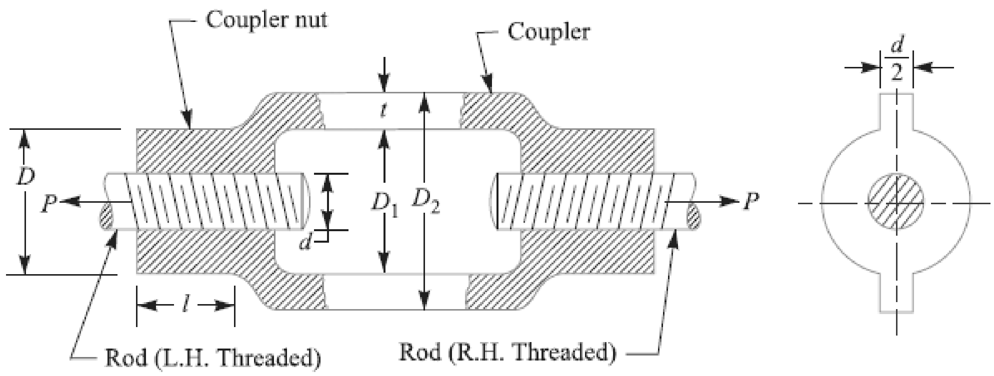
Answer:

$d_1 = 300mm$, $r_1 = d_1/2 = 150mm$
$d_2 = 200mm$, $r_2 = d_2/2 = 100mm$
$P_{max} = 0.1\ N/mm^2$ — 1
$\mu = 0.3$
$N = 2500$ rpm

Since the intensity of pressure is maximum at inner radius, therefore, for uniform wear,

$$P \times r_2 = c$$

$c = 0.1 \times 100$ **$c = 10$ N/mm** We know that, axial thrust, $W = 2\pi c\,(r_1 - r_2)$ $W = 2\pi \times 10 \times (150-100)$ **$W = 3142$ N** and mean radius of friction, $r = (r_1 + r_2)/2$ $r = (150+100)/2$ **$r = 125$ mm** we know that, torque transmitted, $T = n.\ \mu.\ W.\ r$ $T = 2 \times 0.3 \times 3142 \times 125$ $T = 235650$ N-mm **$T = 235.65$ N-m** Power transmitted by clutch, $P = (2\pi\ N\ T)/60$ $P = (2 \times \pi \times 2500 \times 235.65)/60$ **$P = 61693$W** **$P = 61.693$kW**	1 1 1 1 1
2. Attempt **any FOUR** of the following :	16
a) Write the design procedure of turn buckle.	4
Answer: **Following is the design procedure of turn buckle:** 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	1/2



1

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

½

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.

2. **Shear failure of threads at their roots:**

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

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

½

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.

3. **Checking the crushing stress induced in threads:**

Checking the safe crushing, induced crushing stress should not exceed permissible crushing stress.

Area resisting crushing at fork end = $\pi/4 \times (d_0^2 - d_c^2) \times n \times l$

½

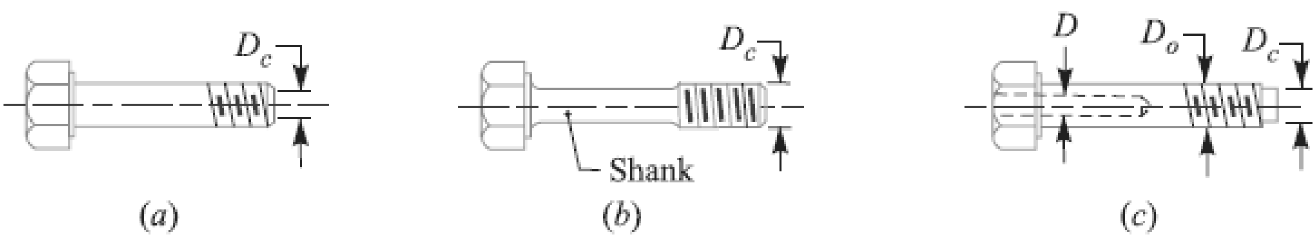
Crushing strength = $W = \pi/4 \times (d_0^2 - d_c^2) \times n \times l \times \sigma_c$

Where, n = number of threads per mm of length = 1/ pitch.

$\sigma_c = W / (\pi/4 \times (d_0^2 - d_c^2) \times n \times l)$

4. The tensile failure of coupler nut: Outside diameter (D) of coupler nut is found by considering tensile failure, $\sigma_t = W / (\pi/4 \times (D^2 - d_0^2))$ **5. 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 = $0.75d_0$	½ ½
b) Explain the two methods to make bolt of uniform strength	4

Answer:



In order to make the bolt of uniform strength, the shank of the bolt is reduced in diameter. the shank diameter can be reduced in following two manners: **2**

1. If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (Dc) as shown in Fig. (b), then shank of the bolt will undergo a higher stress. This means that a shank will absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us bolts of uniform strength. The resilience of a bolt may also be increased by increasing its length.
2. A second alternative method of obtaining the bolts of uniform strength is shown in Fig. (c). In this method, an axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread.

Let D = Diameter of the hole.
D_o = Outer diameter of the thread, and
D_c = Root or core diameter of the thread. **2**

$$\therefore \quad \frac{\pi}{4} D^2 = \frac{\pi}{4}\left[(D_o)^2 - (D_c)^2\right]$$

$$D^2 = (D_o)^2 - (D_c)^2$$

$$\therefore \quad D = \sqrt{(D_o)^2 - (D_c)^2}$$

c) Draw a neat sketch of knuckle joint and insert important dimensions in terms of 'd' diameter of knuckle pin. Calculate the diameter of rod to withstand a load of 400kN permissible stresses are σ_t = 70 N/mm^2, τ = 60 N/mm^2.	4

Answer:

P = 400 kN = 400 x 10^3 N.
σ_t = 70 N/mm^2,
τ = 60 N/mm^2.

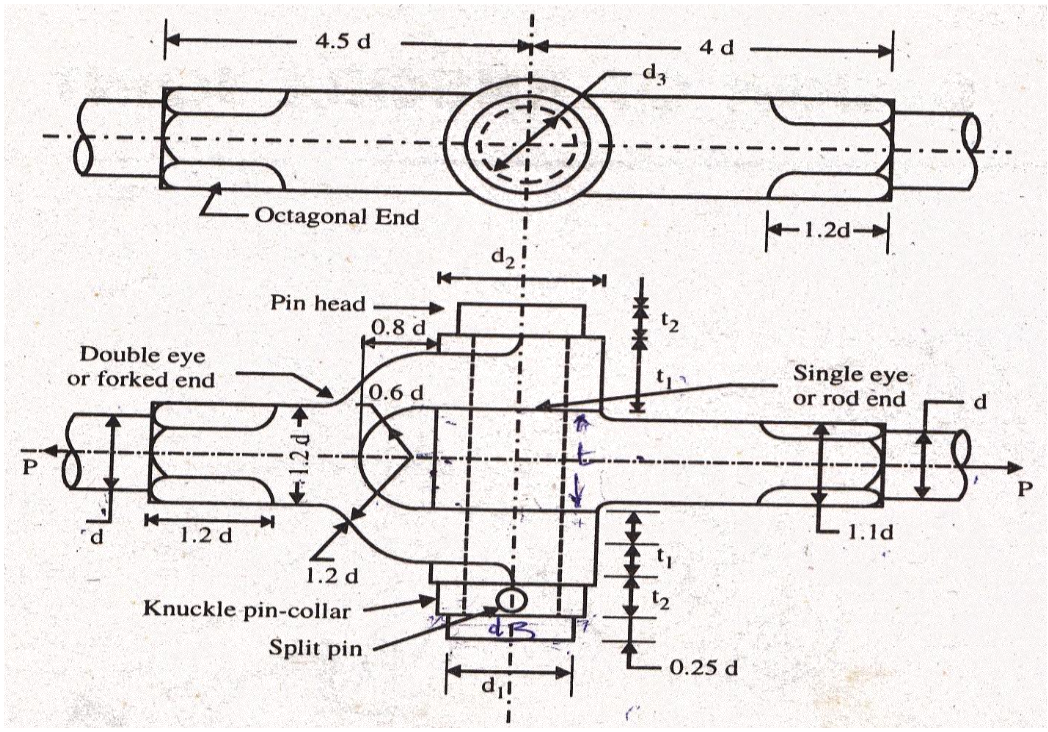
To calculate the diameter of rod, We know that,

$P = \pi/4 \times d^2 \times \sigma_t$

$d^2 = 400 \times 10^3 / (\pi/4 \times 70)$
d^2 =7275.65
d = 85.29mm — 1

d= 86 mm

Finding all other dimensions,
d_1 = d = 86mm
d_2 = 2d = 172mm
d_3 = 1.5d = 129 mm — 1
t = 1.25d = 107.5mm
t_1 = 0.75d = 64.5mm
t_2 = 0.5d = 43mm



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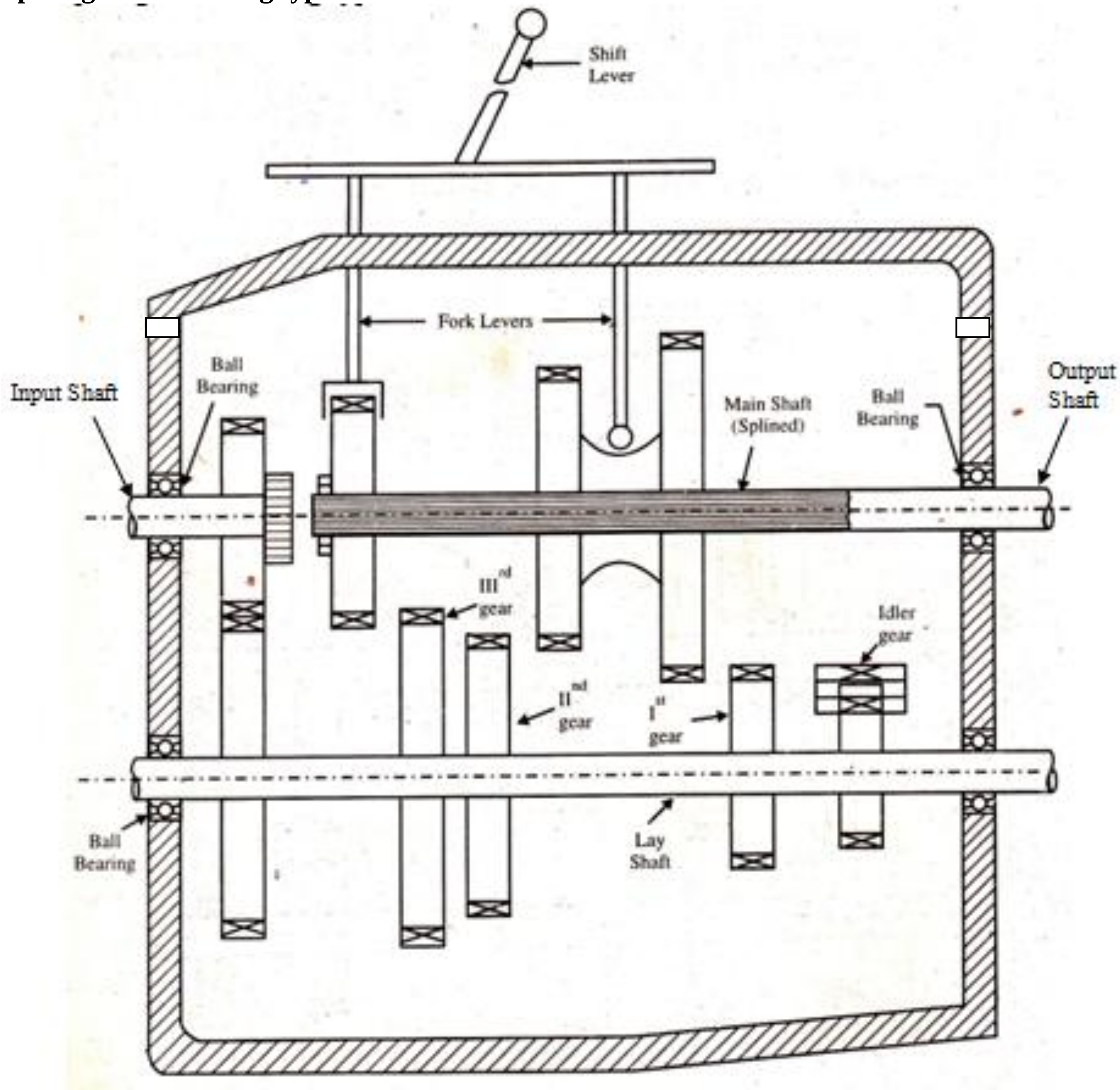
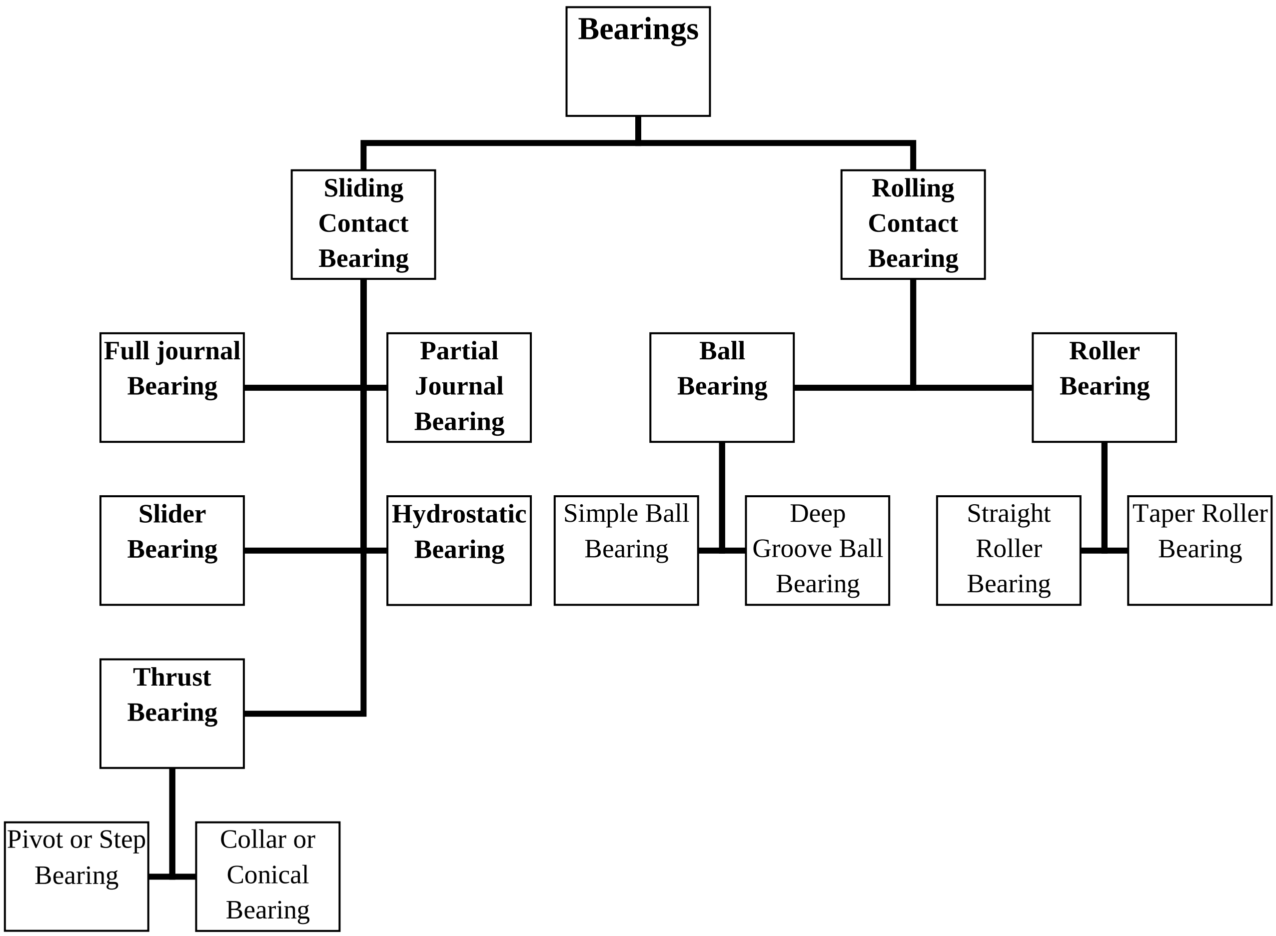
d) Draw the four speed gear box sliding type showing clearly input shaft, output shaft, lay shaft and various bearing locations.	4
Answer: (Sketch – 3 marks, Correct Labeling – 1 Mark) **Four speed gear box sliding type:**	4

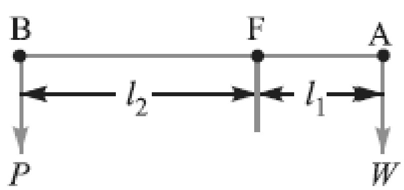


e) Write the detail classification of bearing	4
Answer: Bearings are classified mainly by following two ways: **1. Depending upon the direction of load to be supported:** a) Radial Bearing b) Thrust Bearing – Foot step or Pivot bearing & Collar bearing	4

2. Depending upon the nature of contact:

a) Sliding contact Bearing
b) Rolling contact bearing.



3. Attempt **any** **FOUR** of the following :	16
a) Define the following : i) Shaft ii) Axle iii) Spindle iv) Key	4
Answer: **Shaft:** A shaft is a rotating machine element which is used to transmit the power from one place to another. A shaft is used for the transmission of torque and bending moment.	1
Axle: An axle is a stationary or non-rotating machine element and is used for the transmission of bending moment.	1
Spindle: A spindle is a short rotating shaft, which forms the integral part of machine.	1
Key: A key is a piece of mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them.	1
b) What is lever? Explain the principle on which it works.	4
Answer: **Lever:** A lever is a rigid rod or bar capable of turning about a fixed point (fulcrum).	1
The principle on which the lever works is same as that of moments.	1
Consider a straight lever with parallel forces acting in the same plane as shown in Fig. The points *A* and *B* through which the load and effort is applied are known as load and effort points respectively. *F* is the fulcrum about which the lever is capable of turning. The perpendicular distance between the load point and fulcrum ($l1$) is known as ***load arm*** and the perpendicular distance between the effort point and fulcrum ($l2$) is called ***effort arm***. According to the principle of moments, $W \times l_1 = P \times l_2$ or $\frac{W}{P} = \frac{l_2}{l_1}$  *i.e.* Mechanical advantage, $M.A. = \frac{W}{P} = \frac{l_2}{l_1}$ The ratio of the effort arm to the load arm *i.e.* $l2$ / $l1$ is called ***leverage***.	2

c) Write the design procedure for hand lever.	4
Ans: (1 Mark- Sketch, 3 Marks – Any three design steps)	4

Let P = Force applied at the handle,

L = Effective length of the lever,

σ_t = Permissible tensile stress, and

τ = Permissible shear stress.

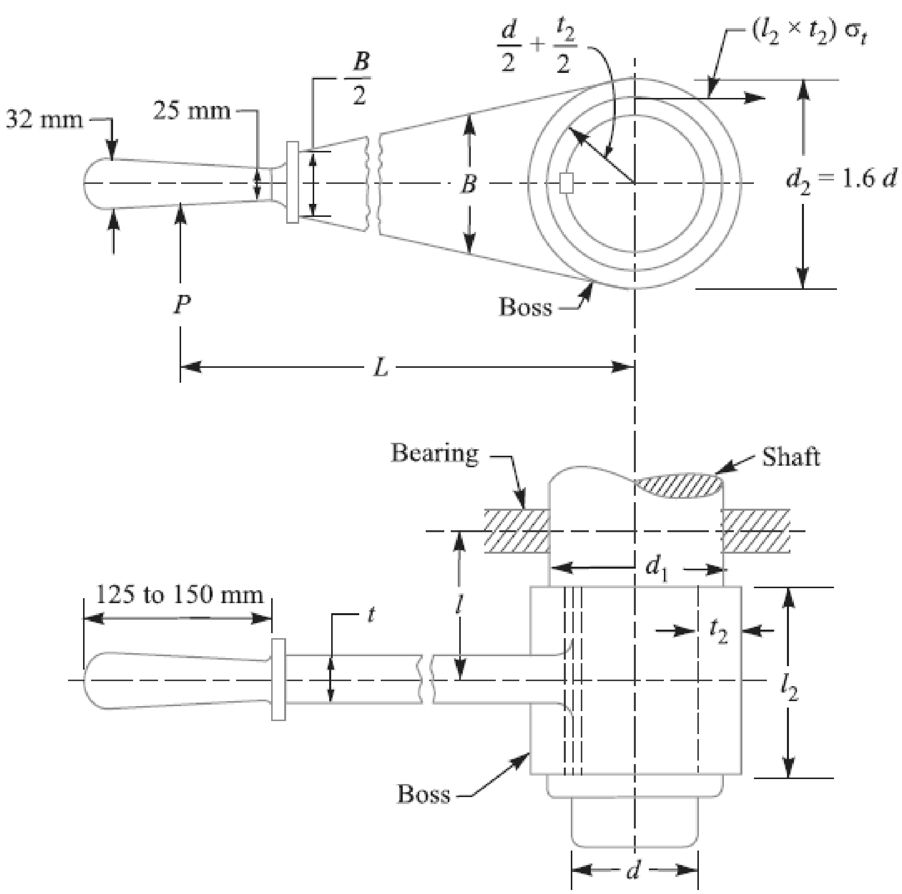
In designing hand levers, the following procedure may be followed :

1. The diameter of the shaft (d) is obtained by considering the shaft under pure torsion know that twisting moment on the shaft,

$$T = P \times L$$

and resisting torque,
$$T = \frac{\pi}{16} \times \tau \times d^3$$

From this relation, the diameter of the shaft (d) may be obtained.



2. The diameter of the boss (d_2) is taken as 1.6 d and thickness of the boss (t_2) as 0.3 d.

3. The length of the boss (l_2) may be taken from d to 1.25 d. It may be checked for a trial thickness t_2 by taking moments about the axis. Equating the twisting moment ($P \times L$) to the moment

of resistance to tearing parallel to the axis, we get

$$P \times L = l_2 t_2 \sigma_t \left(\frac{d+t_2}{2}\right) \quad \text{or} \quad l_2 = \frac{2P \times L}{t_2 \sigma_t (d+t_2)}$$

4. The diameter of the shaft at the centre of the bearing (d_1) is obtained by considering the shaft in combined bending and twisting.

We know that bending moment on the shaft,

$$M = P \times l$$

and twisting moment, $T = P \times L$

∴ Equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(P \times l)^2 + (P \times L)^2} = P\sqrt{l^2 + L^2}$$

We also know that equivalent twisting moment,

$$T_e = \frac{\pi}{16} \times \tau (d_1)^3 \quad \text{or} \quad P\sqrt{l^2 + L^2} = \frac{\pi}{16} \times \tau (d_1)^3$$

The length l may be taken as $2\, l_2$.

From the above expression, the value of d_1 may be determined.

5. The key for the shaft is designed as usual for transmitting a torque of $P \times L$.

6. The cross-section of the lever near the boss may be determined by considering the lever in bending. It is assumed that the lever extends to the centre of the shaft which results in a stronger section of the lever.

Let t = Thickness of lever near the boss, and
B = Width or height of lever near the boss.

We know that the bending moment on the lever,

$$M = P \times L$$

Section modulus, $Z = \frac{1}{6} \times t \times B^2$

We know that the bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{P \times L}{\frac{1}{6} \times t \times B^2} = \frac{6P \times L}{t \times B^2}$$

The width of the lever near the boss may be taken from 4 to 5 times the thickness of lever, *i.e.* $B = 4\,t$ to $5\,t$. The width of the lever is tapered but the thickness (t) is kept constant. The width of the lever near the handle is $B/2$.

d) A truck spring has 10 numbers of leaves. The supports are 1185mm apart and the central (support) is 85mm wide. The load on the spring is 20 kN and takes permissible stress of 300 N/nm^2. Determine the thickness of the leaves if the width of spring is 85mm 4

Ans:

Solution : Given : $n = 10$. Assuming $\eta_f = 2$

$2L_1 = 1185$ mm

$l = 85$ mm

$2W = 20$ kN $= 200000$ M

$W = 10000$ N

$f_f = 300$ N/mm^2

Let, t = thickness of the leaves

and b = Width of the leaves.

We know that the effective length of the spring

$$2L = 2L_1 - l$$
$$= 1185 - 85 = 1100 \text{ mm}$$
$$\boxed{\therefore L = 550 \text{ mm}}$$ 2

and number of graduated leaves

$$\eta_a = n - n_f = 10 - 2 = 8$$

Assuming that the leaves are not initially stressed.

Therefore maximum bending stress for full length leaves **(f_f)**

$$300 = \frac{18\ W \cdot L}{bt^2\,(2n_g + 3\ n_f)}$$

$$300 = \frac{18 \times 10000 \times 550}{85 \times t^2\,(2 \times 8 + 3 \times 2)}$$

$$300 = \frac{18 \times 10000 \times 550}{85 \times t^2\,(22)}$$

$$330 = \frac{52941.176}{t^2}$$

$$t = 13.28\ \text{mm}$$

$$\boxed{\therefore t \approx 14\ \text{mm}}$$

2

OR

$f = 6WL/ nbt^2$

$300 = 6 \times 10000 \times 550/ 10 \times 85 \times t^2$

$t = 11.37$ mm

$t = 12$ mm (approx..)

e) A multi disc clutch has 5 plate having 4 pairs of active friction surface if the intensity of pressure is not to exceed $0.127 N/nm^2$ find the power transmitted at 500 rpm. The outer and the inner radii of friction surface are 125 mm and 75 mm respectively. Assume uniform wear and take co-efficient of friction 4

Solution : Given : $n_1 + n_2 = 5$, and $n = 4$.

$N = 500$ rpm, $\mu = 0.3$, $P_{max} = 0.127\ N/mm^2$, $r_1 = 125$ mm and $r_2 = 75$ mm.

Since the intensity of pressure is maximum at the inner radius r_2, therefore $P_{max} \times r_2 = c$

or
$$c = 0.127 \times 75$$
$$c = 9.525\ N/mm$$

1

We know that the axial force required to engage the clutch,

$$W = 2\pi c\,(r_1 - r_2)$$
$$= 2\pi \times 9.525 \times (125 - 75)$$
$$= 2990\ N$$

1

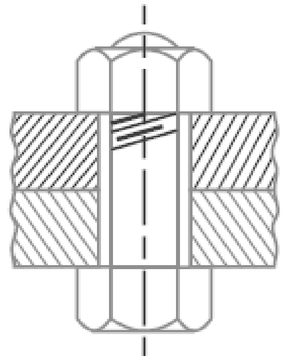
Mean radius of frictional surface,

$$r = \frac{r_1 + r_2}{2}$$
$$= \frac{125 + 75}{2}$$
$$= 100\ \text{mm}$$

We know that torque transmitted, $T = n \mu W r$ $= 4 \times 0.3 \times 2990 \times 100$ $= 358.8 \times 10^3$ N-mm $= 358.8$ N-m	1
Now power transmitted, $P = \frac{2\pi n T}{60}$ $= \frac{2 \times 3.14 \times 500 \times 358.8}{60}$ $= 18800$ W $= 18.8$ kW	1
4 a) Attempt **any THREE** of the following :	12
i) Draw the neat sketch of through bolt and tap bolt and give their application.	

Answer:

Sketch of Through bolt:



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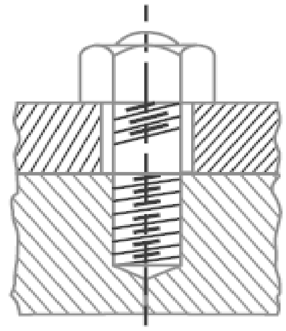
Application: machine bolts, carriage bolts, automobile bolts, eye bolts etc. 1

Sketch of Tap bolt:



1

Application: Cylinder head, machine foundation bolt, lifting bolt, Gear box housing etc. 1

ii) Write two application of each of the following 1) Socket and Spigot cotter joint 2) Turn buckle	4
Answer: **Following are the applications of Socket & Spigot type joint: (Any Two)** i. Valve rod and its stem ii. Connecting a piston rod to cross head of steam engine iii. Joining a tail rod with piston rod of an air pump etc. **Following are the applications of Turn Buckle type joint: (Any Two)** i. Aeroplanes ii. Tie bar of jib crane iii. To connect compartments of locomotives iv. To tie string of electric poles. v. Tie rod of steering system vi. Length adjuster of various linkages etc.	2 2
iii) Design the turn buckle tie rod diameter to with stand a load of 2500N, permissible stresses are ft= 70N/nm^2 and ts=60 N/nm$^{2.}$	4
Answer: Given P=2500 N, ft =70 N/mm2, ts =60 N/mm2 1. Design load P_d = 1.3 P = 1.3 x 2500 = 3250 N 2. Let Core diameter of rod = d_c Now, $P_d = \pi/4\ d_c^2$ d_c^2= 59.11 mm d_c = 7.68 mm 3. Rod diameter $d = d_c /0.84$ = 7.68/0.84 d= 9.15 mm ≈ 10 mm	1 1 2
iv) Explain the effect of keyway on the shaft	4
Ans: 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. $e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$ where e = Shaft strength factor. It is the ratio of the strength of the shaft with keyway to the strength of the same shaft without keyway, w = Width of keyway, d = Diameter of shaft, and h = Depth of keyway = $\frac{\text{Thickness of key } (t)}{2}$ It is usually assumed that the strength of the keyed shaft is 75% of the solid shaft.	2 1 1

b) Solve **any one** of the following	06
i) Design a muff coupling for a shaft which transmits 37.5kW at 240rpm .The allowable shear for shaft is 60N/nm^2 and for cast iron muff is 10N/nm^2. The stresses for Ray are 60N/nm^2 and 126N/nm^2 in shear and bearing respectively.	06

Ans:

Given data:

$$P = 37.5\text{ kW} = 37 \times 10^3\text{ W}; \quad N = 240\text{ r.p.m.}$$
$$F_s = 60\text{ N/mm}^2, \quad F_{sc} = 10\text{ N/mm}^2$$
$$F_{sk} = 60\text{ N/mm}^2 \text{ and } F_{ck} = 126\text{ N/mm}^2$$

L = length of the sleeve
l = length of key in each shaft
T = torque transmitted by the shaft
t = thickness of key
w = width of key

1. Torque transmitted by shaft ,T :
power transmitted

$$P = \frac{2\pi N T}{60}$$
$$T = \frac{P \times 60}{2\pi N}$$
$$T = \frac{37.5 \times 10^3 \times 60}{2 \times 3.14 \times 240}$$
$$T = 1.492 \times 10^3\text{ N-m}$$
$$T = 1.492 \times 10^6\text{ N-mm}$$ 1

2. Diameter of shaft ,d'

$$T = \frac{\pi}{16} F_s d^3$$
$$d^3 = \frac{16\,T}{\pi \times F_s}$$
$$d^3 = \frac{16 \times 1.492 \times 10^6}{3.14 \times 60}$$
$$d^3 = 0.126 \times 10^6$$
$$d = 51\text{ mm}$$ 1

3 Outer diameter of muff 'D' :
D = 2d + 13 mm
= 2 x 51+13mm

= 115 mm
Length of muff, L = 3.5 d

L= 3.5 x51 = 178.5 mm **Torque transmitted by a hollow section(muff)** $T = \frac{\pi}{16} F_{sc} \left(\frac{D^4 - d^4}{D}\right)$ $1.492 \times 10^6 = \frac{\pi}{16} \times F_{sc} \left(\frac{(115)^4 - (51)^4}{115}\right)$ $F_{sc} = 5.21$ N/mm^2 < allowable shear stress **Hence design is safe.** **4. Dimensions of key** l = L/2 =178.5/2 =89.25 mm **Considering shear failure of key,** $T = l \cdot w \cdot F_s \times \frac{d}{2}$ $1.492 \times 10^6 = 89.25 \times w \times 60 \times \frac{51}{2}$ **w = 10.92 =11 mm** **Considering crushing of key,** $T = l \times \frac{t}{2} \times F_{ck} \times \frac{d}{2}$ $1.492 \times 10^6 = 89.25 \times \frac{t}{2} \times 126 \times \frac{51}{2}$ $t = 10.4$ mm $t \approx 10.5$ mm **Taper in key 1:100.** **(Consideration of square is also allowed)**	1 1 1 1
ii) Describe the procedure to design fulcrum pin of rocker arm.	6
Answer: **Step I: Calculate reaction at the fulcrum pin** $R_F = \sqrt{W^2 + P^2 - 2W \times P \times \cos\theta}$ **Step II: Design of fulcrum pin:** (a) Let d = Diameter of the fulcrum pin, and l = Length of the fulcrum pin = 1.25 d Considering the bearing of the fulcrum pin. We know that load on the fulcrum pin (R_F),	1 1

$$\therefore \text{ Bearing pressure} = \frac{\text{Load}}{\text{Bearing area}} = \frac{R_F}{l \times d} = \frac{R_F}{1.25d \times d}$$ 1

From here, l and d can be determined.

(b) Checking shear stress induced in the fulcrum pin. As the pin is in double shear,

$$\tau = \frac{R_F}{2 \times \left(\frac{\pi}{4} \cdot d^2\right)}$$ 1

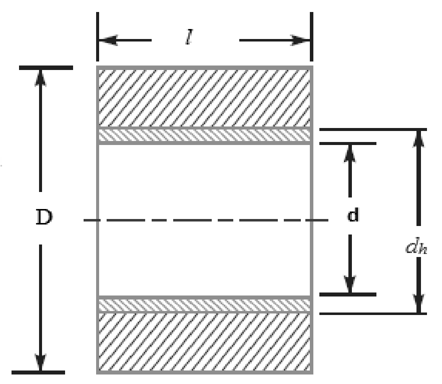
External diameter of the boss,

$D = 2\,d$

Internal diameter of the hole in the lever,

$d_h = d + 2 \times 3$

check the induced bending stress for the section of the boss at the fulcrum



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

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

Induced bending stress,

$$\sigma_b = \frac{M}{Z}$$ 1

5 .Attempt **any TWO** of the following : 16

a) Draw the neat sketch of the fully floating axle. And design the diameter of rear axle shaft for fully floating type with the following data **8**
Engine Power = 10 kW at 300 rpm.
Gear box ratio = 4:1,2.4:1,1.5:1 And 1:1
Differential Reduction =6:1
τ for the shaft = 70 N/mm^2

Answer: **Answer:**

Given:

$P = 10$ kW $= 10 \times 10^3$ W
$N = 300$ rpm
Maximum gear ratio, $G_1 = 4:1$
Differential reduction, $G_d = 5:1$
Shear Stress $\tau = 70$ N/mm^2
Now torque produced by the engine T_e :

We know that,

$$P = \frac{2.\pi.N.Te}{60}$$

$$10 \times 10^3 = \frac{2\pi \times 300 \times t_e}{60}$$

T_e =318.309 Nm

T_e = 318.309$10^3$ Nmm 2

Now torque transmitted by rear axle shaft, T_{RA},

$T_{RA} = T_e \text{ x } G_1 \text{ x } G_d$

$T_{RA} = 318.309 \text{ x } 10^3 \text{ x } 4 \text{ x } 6$ 2

$T_{RA} =$ 7639.43x 10^3 N-mm

Let us 'd' is diameter of rear axle shaft,

$T_{RA} = \Pi/16 \text{ x } \tau \text{ x } d^3$

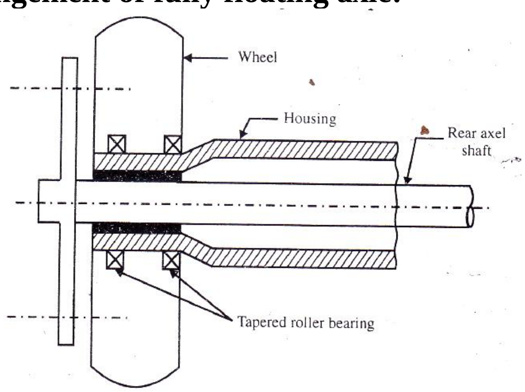
$7639.43\text{x } 10^3 = \Pi/16 \text{ x } 70 \text{ x } d^3$

$d^3 = 555819.06$

d = 82.22 mm 2

d=83 mm

Sketch for arrangement of fully floating axle:



2

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b) A four speed gear box is to be constructed for providing the ratios of 1.0, 1.46, 2.28 and 4 to 1 as nearly as possible. The dimensional pitch (module) of each gear is 3.25 mm and the smallest pinion is to have at least 15 teeth. Determine the suitable number of teeth of any gear ratio. What is the distance between the main layout shafts?	8

Answer: **Answer:**

First Gear Ratio:

$$G_1 = \frac{T_B}{T_A}\frac{T_D}{T_C} = 3.93$$

We have, $$\frac{T_B}{T_A} = \frac{T_D}{T_C} = \sqrt{3.93} = 1.98$$

Adopting $T_A = T_C = 15$ the lowest value given,

we get $T_B = T_D = 1.98 \times 15 = 29.7 = 30$ — 2

Thus actual ratio $= \frac{30}{15} \times \frac{30}{15} = 4:1$

and $T_A + T_B = T_C + T_D = T_E + T_F = T_G + T_H = 45.$

Second gear ratio,

$$G_2 = \frac{T_B}{T_A}\frac{T_F}{T_E} = 2.28$$

or $$\frac{T_F}{T_E} = 2.28\frac{T_A}{T_B} = 2.28 \times \frac{15}{30} = 1.14$$

Hence, $T_E + T_F = 2.14\, T_E = 45$ — 2

The actual ratio $= \frac{30}{15} \times \frac{24}{21} = 2.286:1$

Third gear ratio,

$$G_3 = \frac{T_B}{T_A}\frac{T_H}{T_G} = 1.46$$

or $$\frac{T_H}{T_G} = \frac{1.46}{2} = 0.73$$

But $T_H + T_G = 45.$

or $$T_G = \frac{45}{1.73} = 26$$

Hence, $T_H = 45 - 26 = 19$ — 2

Actual ratio $= \frac{30}{15} \times \frac{19}{26} = 1.461:1.$

Top gear ratio, $G_4 = 1:1$

The center distance between the shafts =

= (3.25 x 45)/2 — 2

= 73.125 mm

c) Describe the procedure to design valve seat and valve lift — 8

Answer: In designing a valve, it is required to determine the following dimensions:

(a) Size of the valve port

Let a_p = Area of the port,

v_p = Mean velocity of gas flowing through the port,

a = Area of the piston, and

v = Mean velocity of the piston. — 3

$$a_p \cdot v_p = a.v$$

$$a_p = \frac{a.v}{v_p}$$

(b) Thickness of the valve disc

The thickness of the valve disc (t), as shown in Fig. may be determined empirically from the following relation, *i.e.*

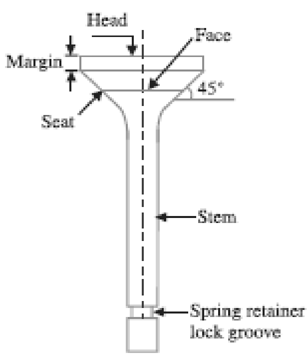
$$t = k.d_p \sqrt{\frac{p}{\sigma_b}}$$

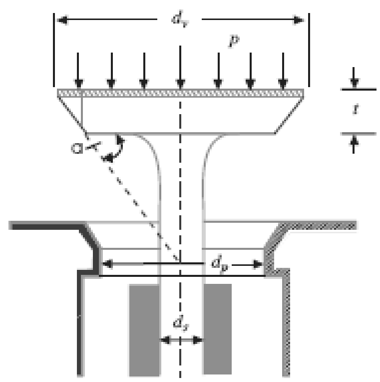
where k = Constant = 0.42 for steel and 0.54 for cast iron, — 3

d_p = Diameter of the port in mm,

p = Maximum gas pressure in N/mm2, and

σ_b = Permissible bending stress in MPa or N/mm2





(c) Maximum lift of the valve

h = Lift of the valve.

The lift of the valve may be obtained by equating the area across the valve seat to the area of the port. For a conical valve, as shown in Fig. — 2

$$\pi\, d_p \cdot h \cos\alpha = \frac{\pi}{4}(d_p)^2 \quad \text{or} \quad h = \frac{d_p}{4\cos\alpha}$$

where α = Angle at which the valve seat is tapered = 30° to 45°.

In case of flat headed valve, the lift of valve is given by

$$h = \frac{d_p}{4}$$

...(In this case, $\langle \alpha = 0° \rangle$)

6. Attempt **any TWO** of the following	16
a) Design the piston pin with following data : i) Maximum pressure on the piston =4 N/nm^2 ii) Diagram of piston = 70 mm Allowable stresses due to bearing bending and shear are given 30 N/mm^2 ,80 N/mm^2 and 60 N/mm^2 respectively.	8

Answer: **Given data,**

Dia. of piston = D = 70 mm.

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

Bearing pressure P_b = 30 N/mm^2

Bending stress = σ_b = 80N/mm^2

Shearing stress = τ = 60 N/mm^2

Maximum gas load,

$$= \frac{\pi D^2}{4} \times p_{\mathbf{max}}$$ **1**

$$F = \frac{\pi}{4}(70)^2 \times 4 = 15.3938 \times 10^3 \text{ N}$$

(a) Design the piston pin on the basis of bearing pressure

Let, d_{po} = outer dia. of piston pin

l_p = length of piston pin in small end of connecting rod

l_p = 0.45xD = 0.45x70

l_p = 31.5 mm **2**

$$F = dp_o \times l_p \times P_b$$

$$dp_o = 15.3938 \times 10^3 / 31.5 \times 30$$

$$dp_o = 16.29 \text{ mm}$$

$$dp_o \approx 17 \text{ mm}$$

(b)Designing the piston pin on the basis of bending.

'Bending moment 'M' is calculated as

M = F x D /8

$$= \frac{15.3938 \times 10^3 \times 70}{8} \text{ N-mm}$$

M = 134.69 x 10^3 N-mm

$M = \pi / 32 x \sigma_b \times (d_{po})^3$ **2**

$\sigma_b = 279.2589 N/mm^2$

The induced bending stresses are greater than permissible bending stress $80 N/mm^2$ hence redesign is necessary. Now redesign value of d_{po}

$M = \pi / 32 x\ \sigma_b \times (d_{po})3$

$d_{po} = 25.79$ mm

$d_{po} = 26$ mm

c) Designing piston pin on the basis of shear stress, due to double shear. **2**

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

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

$T = 14.49\ N/mm^2$

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

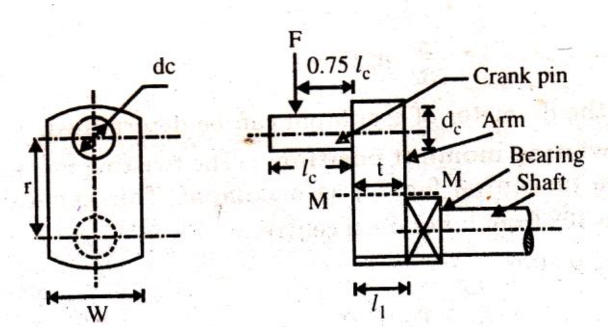
d) The total length of piston pin is taken as **1**

$L_{pt} = 0.9D = 0.9x70 = 63mm$

b) Describe the procedure to design an overhang crank shaft of an I.C. engine. 8

Answer:

Overhang crank shaft (2-Marks Fig.)



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

D = diameter of cylinder bore

P_{max} = maximum of crank pressure

d_c = diameter of crank pin

t= thickness of crank web

W = Width of crank web

r = crank radius or crack throw = $\frac{1}{2}\times$ piston Stroke

d= diameter of shaft at main bearing

Some proportionate dimensions: 8

Thickness of crack web, t = 0.5 to $0.6\,d_c \approx 0.6\,d_c$

W = $1.14\,d_c$ + 12.5 mm

$\frac{\text{Lenth of crank pin}}{\text{dia of crank pin}} = \frac{l_c}{d_c}$ = 0.6 to 1.5 $\approx$ 1.1

(Any two design considerations of the following -3 Marks Each)

1. Design of crank pin : Every crank shaft is designed or checked for atleast two position one when bending moment is maximum and other when twisting moment is maximum

a) Maximum bending moment position: When the crank is on inner dead centre maximum bending moment will act in crank shaft. The thrust in connecting rod 'f' will be equal to piston gas load

$$\therefore \qquad F = \frac{\pi}{4}D^2 \times P_{\max}$$

Due to this gas load bending moment is induced in pin which can be given by

$M = 0.5 l_c \times f$

but as the pin is cantilever and the gas load 'f' may not be evenly distributed this moment is taken as

M = $0.75\,l_c \times F$

As pin is circular section modulus $Z = \frac{\pi}{32} d_c^3$

Now bending moment $\sigma_b = \frac{M}{Z}$

$$\sigma_b = \frac{0.75 l_c \times F}{\frac{\pi}{32}(d_c)^3}$$

Using above equation the diameter of crank pin can be determined

b) Maximum twisting moment position: The twisting moment on the crank shaft will be maximum when the tangential force f_t is maximum. This generally occurs when crank is at angle between 25^0 to 30^0 from inner dead centre.

The gas load along piston

$$F = \frac{\pi}{4} D^2 \times P$$

The thrust along connecting rod

$$Q = \frac{F}{\cos\phi} .... \text{ where } \sin\theta = \frac{\sin\theta}{L/r}$$

Tangential component of thrust

$$F_t = Q\cos(\theta + \phi)$$

Radial component of thrust

$$F_r = Q\cos(\theta + \phi)$$

Due to tangential component F_t twisting moment is induced in pin and it is given by

$$M_t = F_t \times r$$

due to radial component F_r bending moment is induced in pin and it is given by

$$M_b = F_r \times 0.75 l_c$$

Equivalent twisting moment due to M_t and M_b is given by

$$M_{te} = \sqrt{(M_t)^2 + (M_b)^2}$$

The diameter of crank pin can be calculated as

$$M_{te} = \frac{\pi}{16} f_s d_c^3$$

The larger value of d_c out of equation (i) and (ii) is used

The length of crank pin can be determined by using ratio

$$\frac{l_c}{d_c} = 1.1$$

This dimension of crank pin is checked for bearing pressure (or crushing)

Bearing load at crank pin = Max.gas load

Bearing load = Projected area of bearing surface X Permissible bearing pressure

$$F = l_c \times d_c \times P_b$$

P_b = permissible bearing pressure of crank pin or big end of connecting rod

≈ 7 to 15 N/mm^2

2.Dismesions of crank web : The dimension width 'W' and thickness 'T' if crank web are assumed from empirical relation and checked for induced stresses or only thickness 't' is assumed using empirical relation and other dimension width 'W' is calculated on basis of allowable stress.

Empirical relations are

a)Thickness, t = (0.5 to 0.7) d_c

$\approx$ 0.6 dc

b) Width W = 1.14 d_c

Stress in crank web:

a) Direct compressive stress in crack web

$$F_c = \frac{F}{W \times t}$$

b) Bending stress in crank web,

$$F_b = \frac{M_b}{z}$$

= F(0.75 l_c + 0.5 t)

c) Total stress in crank web

=direct stress + bending stress

$= F_c + F_b$

$= F_b = \frac{F}{W \times t} + F(0.75 l_c + 0.5t)$

The total stress should not exceed allowable stress in crank web

3. Shaft at junction of crank web: The diameter of shaft'd' at junction of crank web is design on the basis of

a) Bending stress induced at maximum bending moment position (i.e. when crank is at dead centre)

b) Shear stress induced at maximum twisting moment position

I) Maximum bending moment position

Bending moment about bearing is

M_b = F (0.75 l_c + t + 0.5l_1)

l_1 = Length of bearing

$\approx 1.5\,d_c$

And shaft diameter'd' is obtained by relation

$$M_b = \frac{\pi}{32} fb\ d^3$$

ii) Maximum twisting moment position

Bending moment

M= Q(0.75l_c + t)

and Torque T = F_t x r

$\therefore$ Equivalent bending moment

$$T_e = \sqrt{m^2 + T^2}$$

The shaft diameter'd' is obtained by relation

$$T_e = \frac{\pi}{16} f_s\ d^3$$

The larger value of'd' out of equations (iii) and (iv) is selected

The dimensions of journal are to be checked tor total bearing pressure.

4. Checking dimensions of web for stresses induced at maximum twisting moment position:

Following stresses are induced in crank web when crank is at maximum twisting moment position

i) Bending and direct compressive stress due to f_r

ii) Bending and torsional stress due to f_t

a) Bending stress due to f_r

$$f_{br} = \frac{M_{br}}{Z} \frac{F_r(0.75l_c + \frac{t}{2})}{\frac{Wt^2}{6}}$$

Where M_{br} = Bending moment due to F_r

$\pm\ F_r(0.7l_c + t/2)$

b) Direct compressive stress due to F_r

$$F_c = \frac{F_r}{W.t}$$

c) Bending stress due to F_t

$$F_{bt} = \frac{M_{bt}}{Z} = \frac{F_t(r - d/2)}{\frac{tW^2}{6}}$$

Where M_{bt} = bending moment due to F_t

$$= F_t(r - \frac{d}{2})$$

d) Maximum compressive stress :

$$= F_d = f_{br} + f_c + f_{bt}$$

e) Torsional stress in arm due to f_t

$$F_{st} = \frac{T}{J} = \frac{f_t(0.75l_c + 0.5t)}{\frac{Wt^2}{4.5}}$$

f) Total combined stress (maximum principal stress)

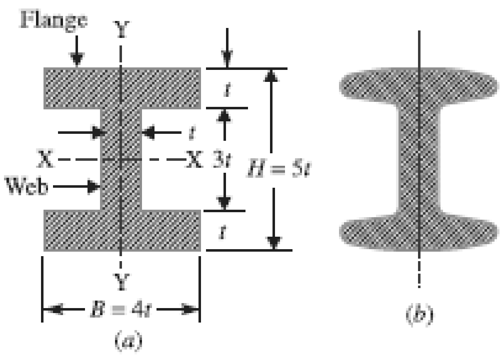
$$F_{\max} = \frac{1}{2}(f_d + \sqrt{(f_d)^2 + 4(f_{st})^2})$$

This $F_{\max}$ should not exceed allowable stress in crack web

c) Design the connection road cross section with the following data of petrol engine Maximum pressure inside the cylinder =4.5 N/nm^2 , Piston diameter = 70mm , Stroke length = 80 mm, Effective length of connecting rod =140mm. Maximum allowable stress in the connection road in clopping is 100 N/nm^2. Take Rankine constant for steel 1/6000 .Explain why I section are used for connecting rod	8

Answer: I-sections are usually found to be most suitable for high speed engine connecting rod lightness is essential in order to keep inertia forces as small as possible .I-section also provides sufficient strength required to with stand momentary high gas pressure in the cylinder .I- section is four times stronger for buckling about X-X axis than Y-Y axis. Thus I-section fulfills most desirable conditions for connecting rod i.e. adequate strength and stiffness and minimum weight.

2



The most suitable section for the connecting rod is *I*-section with the proportions as shown in Fig.

Let thickness of the flange and web of the section = *t*

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

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

From Fig. find that area of the section,

$A = 2\,(4\,t \times t) + 3\,t \times t = 11\,t^2$

Given Data- P_{max} =4.5 N/mm^2 , D =70 mm, l=80 mm,L= 180 mm, σ_{cu} = 330 N/ mm^2,

σ_c = 100 N/ mm^2,

A = 11 t^2 where t = thickness of rod

a = Rankine contant = 1/ 1600

K = $\sqrt{3.18t^2}$

w = maximum gas load= P_{max} x $\frac{\pi}{4}D^2 = 4.5 \times \frac{\pi}{4} 70^2.$

w = 17.32 x 10^3 N

Assuming I-section

$w = \dfrac{\sigma_c \times A}{1 + a[\dfrac{L^2}{k^2_{xx}}]}$	4
$17.32 \times 10^3 \text{ N} = \dfrac{100 \times 11t^2}{1 + 1/1600[\dfrac{140^2}{3.18t^2}]}$	
t = 4.08mm say t = 4.5 mm	
Other dimensions of I-section at middle , small end and big end	
a) at the middle or centre dimension	
(i) depth or height of section	
H= 5 t =5 x 4.5	
H= 22.5 mm	
(ii) width of cross section B	
B= 4 t = 4 x 4.5 =18 mm	
b)Dimension at small end	
(i) depth or height of section	2
H_1=0.82H=0.82x22.5=18.45mm	
(ii) width of cross section B	
B= B_1=18mm	
c)Dimension at big end	
i)depth or height of section	
H_2=1.18H=26.55 mm	
ii) width of cross section B	
B_2=B=18mm	