Important Instructions to examiners:

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

Q.1- A) Attempt any FIVE **(5 x 4) (20)**

a) State the steps involved in general design procedure. **(4marks)**

1. Recognition of 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. Synthesis (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.

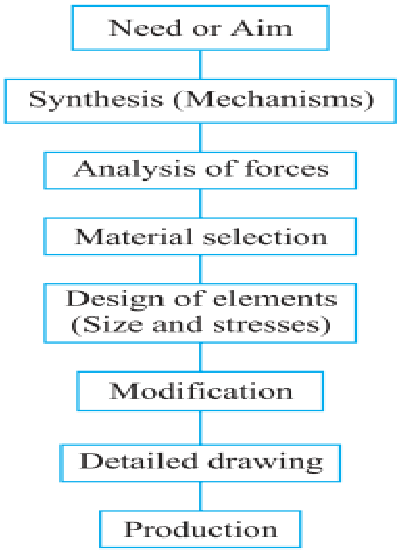
5. Design of elements (Size and Stresses). Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform than the permissible limit.

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. Prepare assembly drawing giving part numbers, overall dimensions and part list. The component drawing is supplied to the shop flow for manufacturing purpose, while assembly drawing is supplied to the assembly shop

8. Production. The component, as per the drawing, is manufactured in the workshop.

OR Each Step : ½ Mark)



b) Factor of safety depends on Following are the factors

i) The extent of loss of life if failure occurs.
ii) The extent of loss of property if failure occurs.
iii) The reliability of properties of material.
iv) The extent of assumption made in design process.
v) The reliability of applied load.
vi) The extent of stress concentration.
vii) Types of loading i.e whether static,fatigue,Impact etc.
viii) service conditions
ix) quality of manufacturing **(any Four points : 4 M)**

C)Write down the names of any four theories of elastic failure

1) Maximum principal stress theory (Rankine's theory)
2) Maximum shear stress theory (Tresca & Guest theory)
3) Maximum strain energy theory (Haigh's theory)
4) Distortion energy theory (Von Mises &Hency theory)
5) Maximum principal strain theory (Saint venant's theory)

……(any 4)………2 mark each

Maximum principal **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.

$\sigma_1 = \frac{\sigma_{yt}}{F.S.}$ for ductile material $\sigma_1 = \frac{\sigma_{ut}}{F.S.}$ for Brittle material**………2 mark**

d) Effect of Keyways

The keyway is a slot machined either on the shaft or in hub to accommodate the key. It is cut by vertical or horizontal milling cutter.
A little consideration will show that 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.

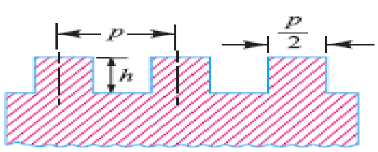
$$e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

where *e* = Shaft strength factor.
w = Width of keyway,
d = Diameter of shaft, and
h = Depth of keyway =Thickness of key (t)/2
It is usually assumed that the strength of the keyed shaft is 75% of the solid shaft, which is somewhat higher than the value obtained by the above relation.
In case the keyway is too long and the key is of sliding type, then the angle of twist is increased in the ratio *k*□□as given by the following relation

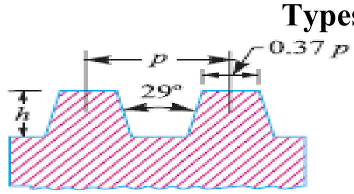
$$K\theta = 1 + 0.4\left(\frac{w}{d}\right) - 0.7\left(\frac{h}{d}\right)$$

where *k*ɵ= Reduction factor for angular twist. ………………..**4 marks**

e) various types of screw threads :
Ans: Following are the three types of screw threads mostly used for power screws:
1. *Square thread.*
2. *Acme thread*
3.trapezoidal thread.
4. *Buttress thread*………… **Types . 2 marks**



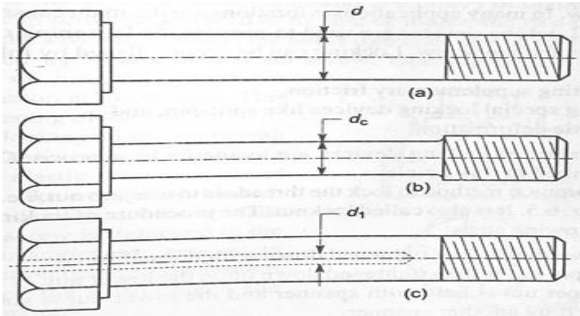
h = 0.5 p
(a) Square thread.



h = 0.5 p + 0.25 mm
(b) Acme thread.

Any two Sketches ..2 marks

f) Bolts of uniform strength.
(Explanation including two methods with figure. 2 marks each)



In an ordinary bolt shown in **Fig. (a)**, the effect of the impulsive loads applied axially is concentrated on the weakest part of the bolt i.e. the cross-sectional area at the root of the threads. In other words, the stress in the threaded part of the bolt will be higher than that in the shank. Hence a great portion of the energy will be absorbed at the region of the threaded part which may fracture the threaded portion because of its small length.

If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (d) as shown in **Fig. (b)**, then shank of the bolt will undergo a higher stress. This means that a shank will absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us bolts of uniform strength. The resilience of a bolt may also be increased by increasing its length.

A second alternative method of obtaining the bolts of uniform strength is shown in **Fig. (c).** An axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread

g) Advantages of V-threads over square threads: Any 4… 1 mark each advantage

1. These are used to tighten the parts together in bolts, studs, nuts, tap bolts etc.

2. V- threads offer greater frictional resistance of motion than square thread hence better suitable for fastening purpose

3. These are stronger than square thread.

4. These are cheaper than Square threads.

5. These thread are easy to cut by die or on machine than square thread.

h) Contribution of Ergonomics and Aesthetic Consideration:

Ergonomics is concerned with optimizing the overall and details design of controls, operations& maintenance interfaces between machines & men in both ,normal and emergency working.

The purpose of applying ergonomics information to design situation is to ensure that the environments provided and the design prepared offer the man the greatest comfort, advantages and safety.

Ergonomics design provides health, happiness and effectiveness

It decreases physical and mental stresses

It helps to study man machine relationship

Increase human efficiency
It improves reliability ,safety.
It provides low cost . ………………….. **2 marks**

Aesthetics:
Aesthetics means a set of principles of appreciation of beauty. It deals with the appearance. An engineering design with good aesthetics will thus be pleasing o the eye and give a visual impression of functioning efficiency.
In presence day of buyers' market, a number of products available in the market are
having most of the parameters identical ,the appearance of the products is often a major factor in affecting the customer.
This is particularly true for customers durables like automobiles ,domestics refrigerators,
television sets ,music systems
At any stage in the product life ,the asthetic quality cannot be separated from the product quality.
Aesthetic consideration regarding shape increases the aesthetic value of a product.
a proper shape of a product to make th e product more attractive.
The color of the product should be pleasing to the eye.
Regarding surface finish: product with better surface finish are always aesthetic pleasing …………………**2 Marks**

Qu.2 Attempt any FOUR 4X4M

a) Fatigue failure & Creep failure

Fatigue failure: When a material is subjected to repeated stresses, it fails at stresses below yield point stresses such type of failure is called as fatigue failure.
This failure is caused by means of progressive crack formation .the failure may occur even without any prior intimation.
The fatigue failure depend on number of cycles, mean stress, stress amplitude, stress concentration.
Creep Failure: When a component is subjected to constant stress at high temperature over a long period of time ,it will undergo a slow and progressive deformation called creep. it is a function of stress and temperature .it occurs in a component operating at elevated temperature. it occurs in three stages .Third stage rate is high due to necking
And finally results into fracture . **OR**

Sr.No	**Fatigue Failure**	**Creep Failure**
1	Subjected to repeated stress	Subjected to constant stress
2	Not subjected to Temperature	Subjected to Temperature
3	Slow & progressive deformation	Progressive crack formation
4	Failure may occur with prior intimation (Necking)	Failure may occur without any prior intimation.
5	Results in compression and tensile stresses	Result in increasing strain

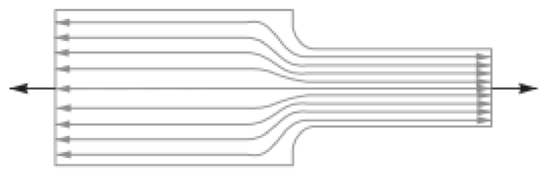
6	Occurs in a rotating components generally.	Occurs in a component operating at elevated temperature.

Any 4 points……1 mark each

b) Stress Concentration:

Whenever a machine component changes the shape of its cross-section, the simple stress distribution no longer holds good and the neighbourhood of the discontinuity is different.

This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. It occurs for all kinds of stresses in the presence of fillets, notches, holes, keyways, splines, surface roughness or scratches etc.



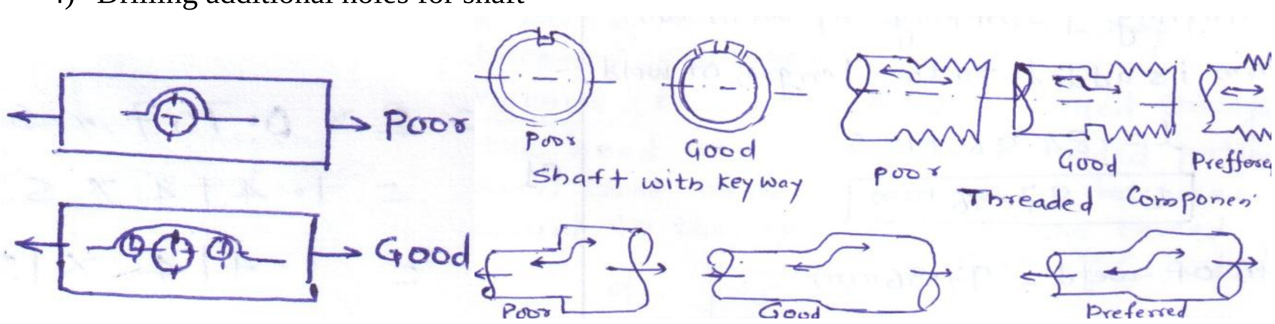
Causes of stress concentration are as under.

i) Abrupt changes in cross-section like in keyway, steps, grooves, threaded holes results in stress concentration. ii) Poor surface finish – The surface irregularities is also one of the reason for stress

concentration. iii) Localized loading – Due to heavy load on small area the stress concentration occurs in the vicinity of loaded area. iv) Variation in material properties – Particularly defects like internal flaws, voids, cracks, air holes, cavities also results in stress concentration.**…Correct explanation………2 marks)**

Two methods of reducing stress concentration :(Any Two Methods with sketch 2 Marks)

1) Introducing additional notches and holes in tension member
2) Fillet radius ,undercutting & notches for member I bending
3) Reduction of stress concentration in threaded portion
4) Drilling additional holes for shaft



C) Strength equation of Knuckle Joint:

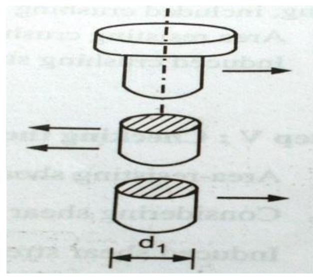
(Any 4 equations with skectch.1mark each equation)

1. ***Failure of the solid rod in tension***

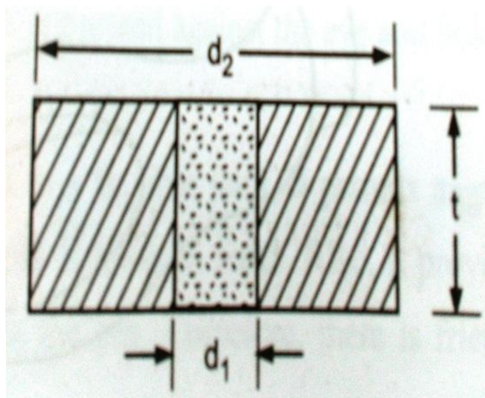
$$P = \frac{\pi}{4} \times d^2 \times \sigma_t$$

2. *Failure of the knuckle pin in shear*

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

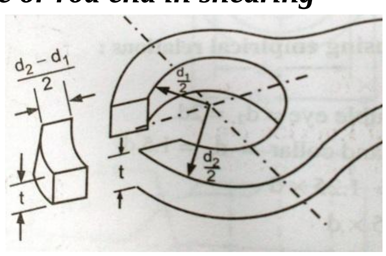


3Failure of the single eye or rod end in tension



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

4. *Failure of the single eye or rod end in shearing*

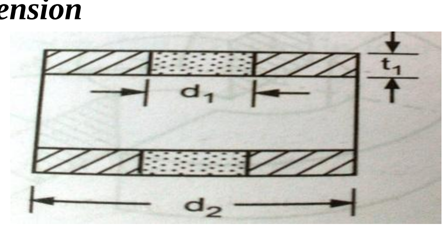


$$P = (d_2 - d_1)\, t \times \tau$$

5. *Failure of the single eye or rod end in crushing*

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

6. *Failure of the forked end in tension*



$$P = (d_2 - d_1) \times 2t_1 \times \sigma_t$$

7. *Failure of the forked end in shear*

$$P = (d_2 - d_1) \times 2t_1 \times \tau$$

Any 4 eq. ……… 1 Mark for each equation

d)

Given :

Outside diameter of hollow shaft (do) = Diameter of solid shaft (d)

For same material: Density of solid = density of hollow shaft

$L_S = L_H$, di = inside diameter of hollow shaft = 0.75 do , $k = \frac{di}{do}$ =0.75

I) Comparison of weight:

We know that weight of a hollow shaft

W_H = Cross sectional area x Length x Density

$= \pi/4[\,(do)^2 - (di)^2\,] X$ Length x Density............I

and Weight of the solid shaft

W_S = Cross sectional area x Length x Density

$= \pi/4 \quad (d)^2\ X$ Length x Density.................. ..II

Since both the shafts have the same material and length, therefore by dividing equation ***(i)*** by equation ***(ii)***, we get

$$\frac{WH}{WS} = \frac{= [\,(do)^2 - (di)^2\,]}{(d)^2}$$

As K= di/do

$\frac{WH}{WS} = 1 - K^2$ =1- $(0.75)^2$ =0.44

W_H=0.44 W_S .... Ans**........2 Marks**

II) Comparison of Strength:

Strength of hollow shaft $T_H = \frac{\pi}{16} \tau\, do^3\ x\ (1 - k^4)$

Strength of Solid shaft $Ts = \frac{\pi}{16} \tau\, do^3$

$$\frac{TH}{TS} = \frac{[\,(do)^3\,(1 - K^4)\,]}{do^3}$$

$=(1 - k^4) = (1 - 0.75^4)$

T_H=0.68 T_S ………………Ans**………..2 Marks**

e))Self locking of screw:

Torque required to lower the load

$$T = P \times \frac{d}{2} = W \tan(\phi - \alpha)\frac{d}{2}$$

If however , $\phi > \alpha$, the torque required to lower the load will be positive,indicating that an effort is applied to lower the load,such a screw is known as self locking screw.

A screw will be self locking

1) if the friction angle is greater than helix angle or coefficient of friction is greater than tangent of helix angle

   $i.e.$ μ or $\tan\phi > \tan\alpha$.

2) if the frequency is less than 50 % i.e $\eta < 50\%$ **(Correct Ans: 03 M)**

a screw will be self locking if the friction angle is greater thanhelix angle or coefficientof friction is greater than tangent of helix angle .μ or tanΦ>or = tan α .

We know that the efficiency of screw,

$$\eta = \frac{\tan\phi}{\tan(\alpha + \phi)}$$

Therefore, Efficiency for self locking screws,

$$\eta \le \frac{\tan\phi}{\tan(\phi+\phi)} \le \frac{\tan\phi}{\tan 2\phi} \le \frac{\tan\phi\,(1-\tan^2\phi)}{2\tan\phi} \le \frac{1}{2} - \frac{\tan^2\phi}{2}$$

From this expression we see that efficiency of self locking screws is less than ½ or 50%. If the efficiency is more than 50%, then the screw is said to be overhauling.

------------------- 1 Mark

f) Procedure for selection of bearing from manufacturer's catalogue.

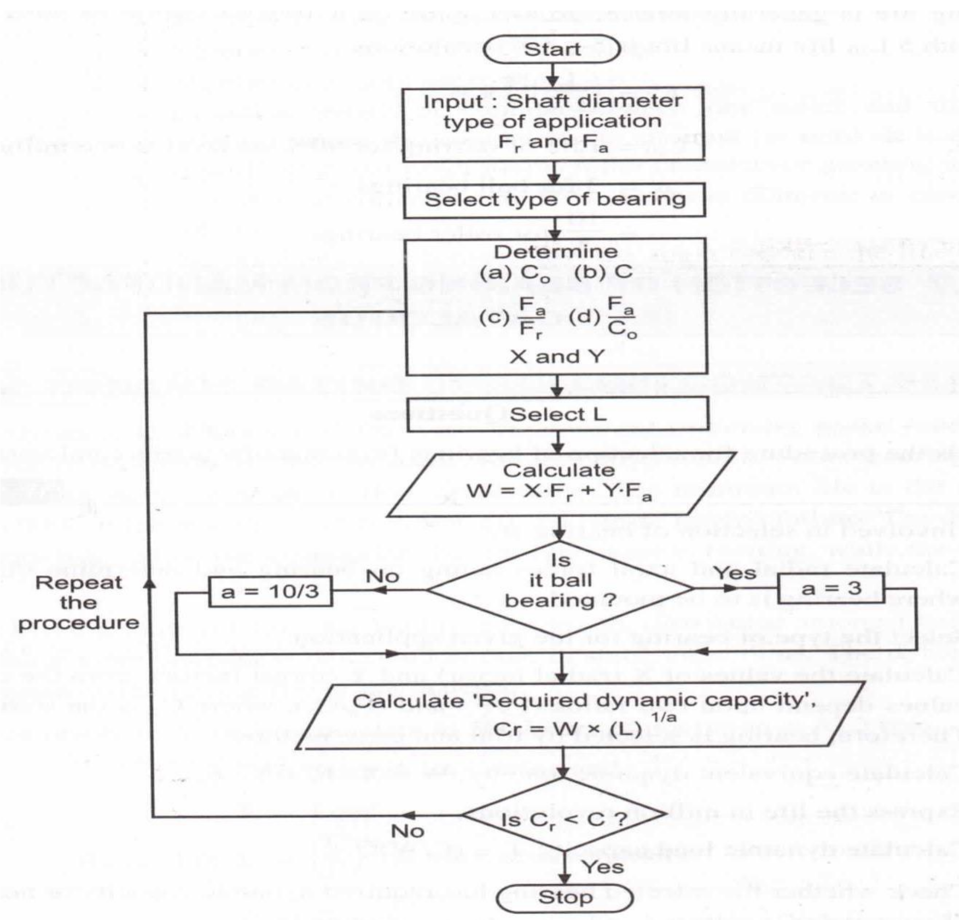
(Correct Procedure OR Flow chart - 4 Marks)

1) Calculate radial and axial forces and determine dia. of shaft.
2) Select proper type of bearing.
3) Start with extra light series for given diagram go by trial of error method.
4) Find value of basic static capacity (co) of selected bearing from catalogue.
5) Calculate ratios Fa/VFr and Fa/Co.
6) Calculate values of radial and thrust factors.(X & Y) from catalogue.
7) For given application find value of load factor Ka from catalogue.
8) Calculate equivalent dynamic load using relation. Pe = (XVFr + YFA) Ka.
9) Decide expected life of bearing considering application. Express life in million revolutions L10.

10) Calculate required basic dynamic capacity for bearing by relation.

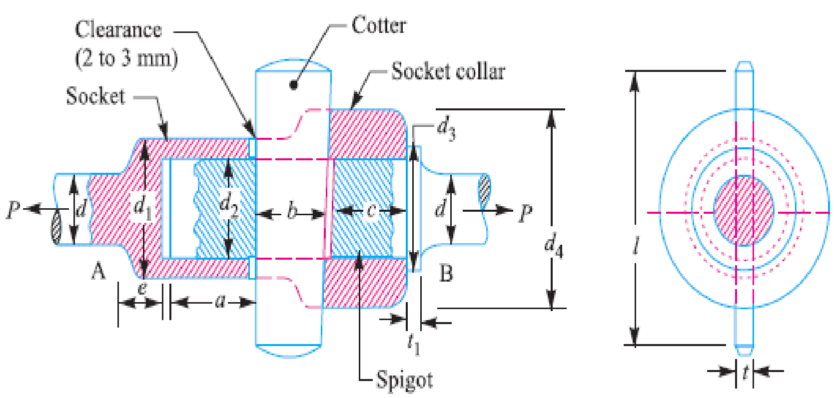
11) Check whether selected bearing has req. dynamic capacity, IF it not select the bearing of next series and repeat procedure from step-4.

OR



Qu.3 . Attempt any TWO :

a) State the design procedure of a cotter joint with neat diagram.



Let 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,
a = Distance from the end of the slot to the end of rod,
σt= Permissible tensile stress for the rods material,
τ□= Permissible shear stress for the cotter material, and
σc= Permissible crushing stress for the cotter material.

Dia.1 mark & procedure 7 marks

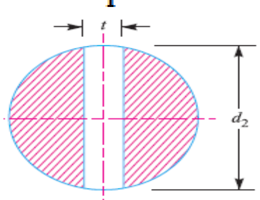
1. Failure of the rods in tension
The rods may fail in tension due to the tensile load P. We know that

$$P = \frac{\pi}{4} X\, d^2 X\, \sigma t$$

From this equation, diameter of the rods (d) may be determined.

2. Failure of spigot in tension across the weakest section (or slot)

$$P = [\frac{\pi}{4} X\, (\mathrm{d2})^2 - \mathrm{d2}\, X\, t]\sigma\mathrm{t}$$

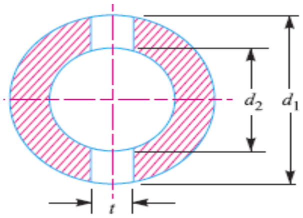


From this equation, the diameter of spigot or inside diameter of socket (d2) may be determined
3. Failure of the rod or cotter in crushing

$P = d_2 \times t \times \sigma_c$ this equation, the induced crushing stress may be checked

4. Failure of the socket in tension across the slot

$$P = [\frac{\pi}{4} X\, (\mathrm{d1})^2 - (\mathrm{d2})^2 - (\mathrm{d1} - \mathrm{d2})X\, t]\sigma\mathrm{t}$$



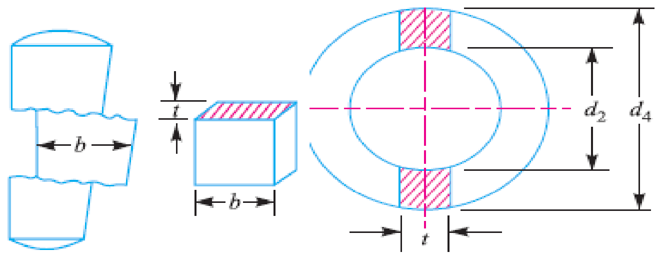
From this equation, outside diameter of socket (d1) may be determined

5. Failure of cotter in shear

Considering the failure of cotter in shear as shown in Fig, Since the cotter is in double shear,

$P = 2\, b \times t \times \tau$

From this equation, width of cotter (b) is determined.



6. Failure of the socket collar in crushing

Considering the failure of socket collar in crushing.

$P = (d4 - d2)\, t \times \sigma c$

From this equation, the diameter of socket collar (d4) may be obtained.

7. Failure of socket end in shearing

Since the socket end is in double shear,

$P = 2\,(d4 - d2)\, c \times \tau$□

From this equation, the thickness of socket collar (c) may be obtained

8. Failure of rod end in shear

Since the rod end is in double shear, $P = 2\, a \times d2 \times \tau$

From this equation, the distance from the end of the slot to the end of the rod (a) may be obtained.

9. Failure of spigot collar in crushing

Considering the failure of the spigot collar in crushing,

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

From this equation, the diameter of the spigot collar (d3) may be obtained.

10. Failure of the spigot collar in shearing

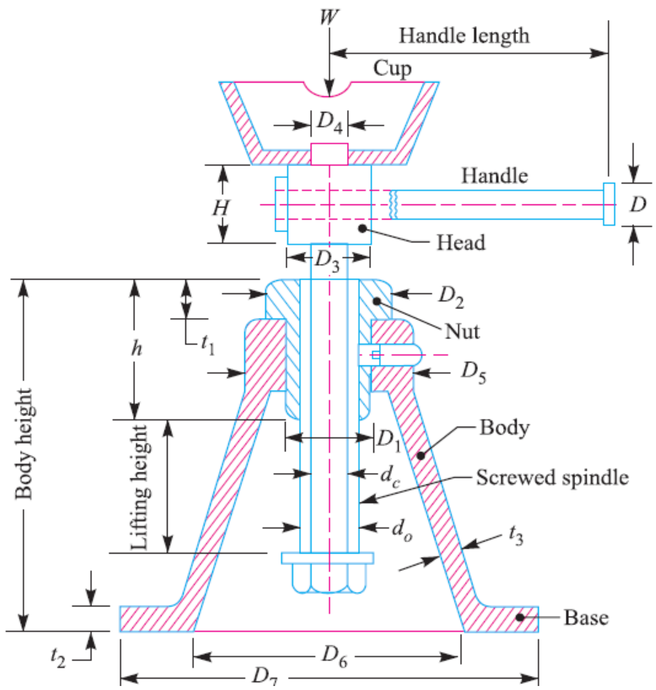
Considering the failure of the spigot collar in shearing

$P = \pi$□□$d2 \times t1 \times \tau$

From this equation, the thickness of spigot collar (t1) may be obtained.

11. The length of cotter (l) is taken as 4 d.

b) Design of screw spindle and Nut of Screw Jack:



1. First of all, find the core diameter (*dc*) by considering that the screw is under pure compression,

$$W = \sigma_c \times A_c = \sigma_c \times \frac{\pi}{4}(d_c)^2$$

2. Find the torque (*T*1) required to rotate the screw and find the shear stress (τ) due to this torque.

We know that the torque required to lift the load,

$$T_1 = P \times \frac{d}{2} = W \tan(\alpha + \phi) \frac{d}{2}$$

P = Effort required at the circumference of the screw, and

d = Mean diameter of the screw.

∴ Shear stress due to torque *T*1,

$$\tau = \frac{16\,T_1}{\pi (d_c)^3}$$

Also find direct compressive stress (σ*c*) due to axial load, *i.e.*

$$\sigma_c = \frac{W}{\frac{\pi}{4}(d_c)^2}$$

3. Find the principal stresses as follows:

Maximum principal stress (tensile or compressive),

$$\sigma_{c(max)} = \frac{1}{2}\left[\sigma_c + \sqrt{(\sigma_c)^2 + 4\tau^2}\right]$$

and maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2}$$

These stresses should be less than the permissible stresses.

4. Find the height of nut (h), considering the bearing pressure on the nut. We know that the bearing pressure on the nut,

$$p_b = \frac{W}{\frac{\pi}{4}\left[(d_o)^2 - (d_c)^2\right] n}$$

where n = Number of threads in contact with screwed spindle.

∴ Height of nut, $h = n \times p$

where p = Pitch of threads.

5. Check the stresses in the screw and nut as follows :

$$\tau_{(screw)} = \frac{W}{\pi\, n.d_c.t}$$

$$\tau_{(nut)} = \frac{W}{\pi\, n.d_o.t}$$

6. Find inner diameter ($D1$), outer diameter ($D2$) and thickness ($t1$) of the nut collar.

The inner diameter ($D1$) is found by considering the tearing strength of the nut. We know That

$$W = \frac{\pi}{4}\left[(D_1)^2 - (d_o)^2\right]\sigma_t$$

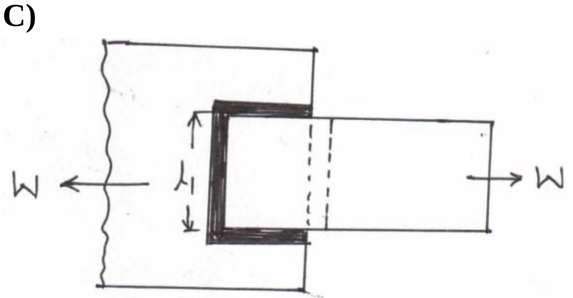
The outer diameter ($D2$) is found by considering the crushing strength of the nut collar. We know that

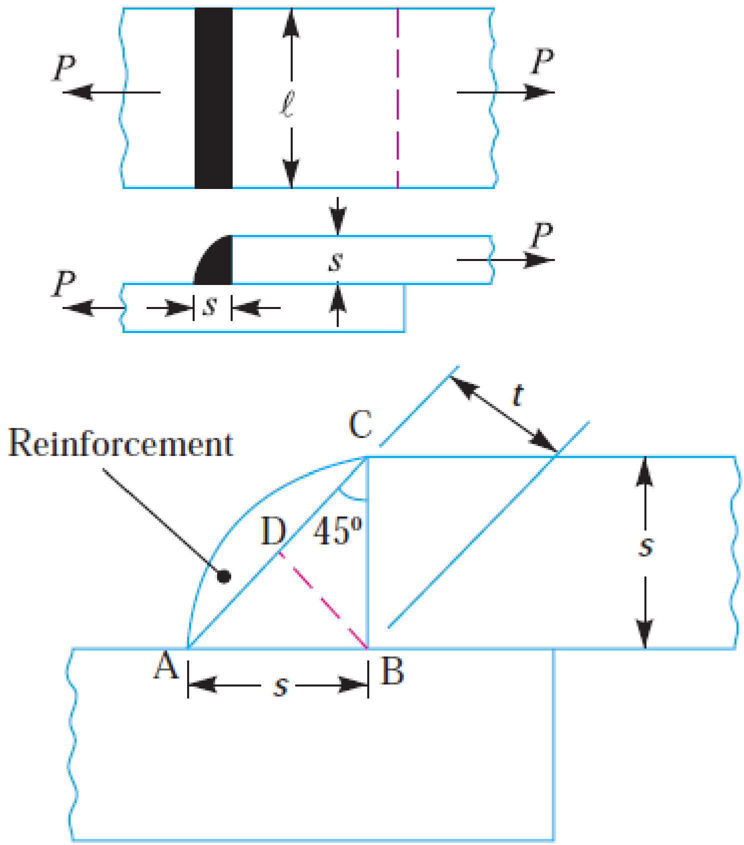
$$W = \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right]\sigma_c$$

The thickness ($t1$) of the nut collar is found by considering the shearing strength of the nut collar. We know that $W = \pi\, D1.t1.\tau$

(Sketch 02 M & Design 6x1 Marks)

C)





Let t = Throat thickness (BD), ………**Sketch 2 Marks**

s = Leg or size of weld,

= Thickness of plate, and

l = Length of weld,

From Fig. 10.7, we find that the throat thickness,

$t = s \times \sin 45° = 0.707\ s$

∴*Minimum area of the weld or throat area,

A = Throat thickness ×

Length of weld

$= t \times l = 0.707\ s \times l$ ***…………..1 marks***

If σt is the allowable tensile stress for the weld metal, then the tensile strength of the joint for single fillet weld,

P = Throat area × Allowable tensile stress = $0.707\ s \times l \times \sigma t$

and tensile strength of the joint for double fillet weld,

$P = 2 \times 0.707\ s \times l \times \sigma t = 1.414\ s \times l \times \sigma t$ ***………………..2marks***

If τ is the allowable shear stress for the weld metal, then the shear strength of the joint for singleparallel fillet weld,

P = Throat area × Allowable shear stress = $0.707\ s \times l \times \tau$

and shear strength of the joint for double parallel fillet weld,

$P = 2 \times 0.707 \times s \times l \times \tau = 1.414\, s \times l \times \tau$**………………..*2marks***

The strength of the joint is given by the sum of strengths of single transverse and double parallel fillet welds.

Mathematically,

$P = 0.707 s \times l1 \times \sigma t + 1.414\, s \times l2 \times \tau$ …………1mark

Qu.4 Attempt any FOUR **4 X4 M**

a) Lewis equation for strength of gear tooth

$W_T = \sigma w .b. Pc .y = \sigma w .b.\pi m. y$ ………..**(Pc =πm)**

WT = Tangential load acting at the term **(Equation :2 marks)**

σw = Beam strength of the tooth

b = Width of the gear face

Pc = Circular pitch

m = Module

Y is Lewis form factor or tooth form factor…….**(Meaning of terms 2 marks)**

b) Given: $W = 75$ kN $= 75 \times 10^3$ N ; $v = 300$ mm/min ; $p = 6$ mm ;
$do = 40$ mm ; $\mu = \tan\alpha = 0.15$

We know that mean diameter of the screw,
$d = do - p / 2 = 40 - 6 / 2 = 37$ mm

$$\tan \propto = \frac{P}{\pi d} == \frac{6}{\pi x\, 37} = 0.0516$$

We know that tangential force required at the circumference of the screw,

$$P = W \tan(\propto + \emptyset) = W[\frac{\tan \propto + \tan\emptyset)}{1 - \tan \propto \tan\emptyset)}]$$

$$P = 80 X\, 10^3\, [\frac{(0.0516 + 0.15)}{1 - 0.0516\, X\, 0.15)}]$$

$$P = 16.25\, X\, 10^3 \text{ N}$$

and torque required to operate the screw

$$T = PX\frac{d}{2} = 16.25\, X\, 10^3\, [\frac{37]}{2} = 300.63 X\, 10^3 \text{ N} - \text{mm}$$

Since the screw moves in a nut at a speed of 300 mm / min and the pitch of the screw is 6 mm, therefore speed of the screw in revolutions per minute (r.p.m.),

$$N = \frac{Speed\ in\ mm/min}{Pitch\ in\ mm} == \frac{300}{6} = 50\ r.p.m$$

and angular speed, $\omega = 2\pi N / 60 = 2\pi \times 50 / 60 = 5.24$ rad /s

Power of the motor = $T.\omega$= 300.63 × 5.24 = 1575.30 W = 1.575 kW.

C) Wahl's correction factor or stress factor is modification of factor Ks in following equation

$$\tau = \frac{8\,W.D}{\pi\,d^3}\left(1 + \frac{1}{2C}\right) = K_S \times \frac{8\,W.D}{\pi\,d^3}$$

Where, Ks= 1+1/2c

In the above equation, we have considered only torsion shear stress & direct shear stress due to direct load ……….**01 mark**

Here, We have neglected the stress due to curvature of wire.

When the wire bend, the stress concentration occurred at the inner fiber of coil and this curvature effect is considerably larger for high value of spring index. Therefore , Wahl's correction factor in introduced. **01 mark**

By considering, torsion shear, direct shear and curvature shear stress

∴ Maximum shear stress induced in the wire,

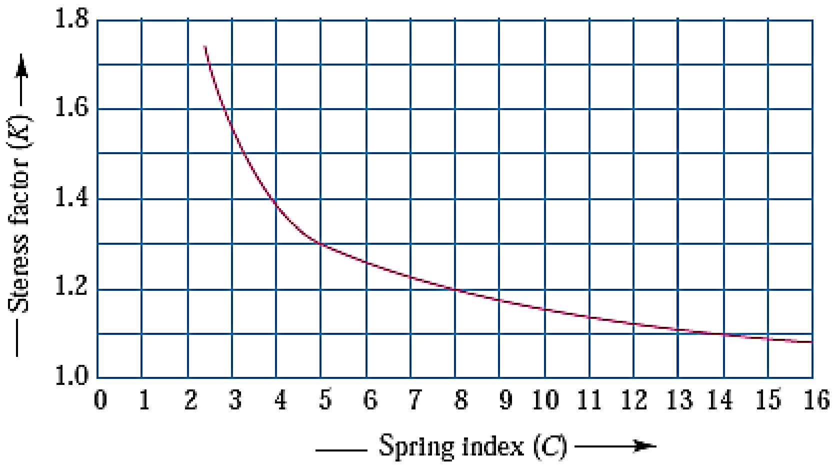
$$\tau = K \times \frac{8\,W.D}{\pi\,d^3} = K \times \frac{8\,W.C}{\pi\,d^2}$$

where

$$K = \frac{4C - 1}{4C - 4} + \frac{0.615}{C}$$

01 mark

Prof.A.M.Wahl's determined this factor hence it is known as wahl's factor.



01 mark

Wahl's stress factor increases very rapidly as the spring index decreases.

d)Example of Screwed Joint:

1) Fasteners
2) Automobile components
3) Crank shaft assembly.
4) Temporary Joint
5) Man whole in vessel or tank. **Any 4 Examples …………….1 mark each**

e) Classification of bearing

1. ***Depending upon the direction of load to be supported.*** The bearings under this group are classified as:

(a) Radial bearings, and **(*b*)** Thrust bearings.

2. ***Depending upon the nature of contact.*** The bearings under this group are classified as :**(*a*)** Sliding contact bearings, and **(*b*)** Rolling contact bearings.
Classification …… 2 marks

Life of bearing is expressed by Rating Life
Rating life denoted by $L_{10.}$

$$L10 = \frac{Lh10\,x\,n\,x\,60}{10^6}$$

Life of bearing depends on the magnitude of the repetitive contact stress at the most heavily stressed contact which in turn depends upon the dynamic load.
The life of individual bearing means the total number of revolution which bearing can complete before the first evidence of fatigue failure develops on the balls or races.
Since the life of an individual bearing cannot be predicted, it is necessary to define the life of a group of apparently identical bearings. The bearing can be expressed in by rating life …………. **2 Marks**

F)

Given Data :—

$P = 50\,kW = 50\times10^3\,N$

$N = 500\,rpm$

$\tau_s = \tau_{key} = 40\,N/mm^2$

$\sigma_c\,shaft, key = 80\,N/mm^2$

Torque transmitted by shaft

$$P = \frac{2\pi N T}{60}$$

$$50\times10^3 = \frac{2\times\pi\times500\times T}{60}$$

$$\therefore\ T = 954.93\ N\cdot m$$

$$T = 954.93\times10^3\ N\cdot mm \ \ldots\ldots \tfrac{1}{2}M$$

Diameter of shaft :—

$$T = \frac{\pi}{16} d^3 \times \tau_{shaft}$$

$$954.93\times10^3 = \frac{\pi}{16}\times d^3\times 40$$

$$\therefore\ d^3 = 121585.46$$

$$\therefore\ \boxed{d = 49.54\ mm} \cong 50\,mm \ \ldots\ldots 1M$$

Dia. of Sleeve :—

outside dia. of sleeve

$$D = 2d + 13 \Rightarrow 113\ mm$$

Length of muff $L = 3.5\,d$

$$= 175\ mm \ \ldots\ldots 1M$$

Design of Key :-

Width of key = $W_K = \frac{d}{4} = \frac{50}{4} = 12.5 \cong 13mm$

$t_K = \frac{d}{4} = 13\ mm$

$l_K = \frac{L}{2} = \frac{175}{2} = 88mm$

Shearing strength of key :-

$T = w_K \times l_K \times \tau_{key} \times d/2$

$954.93 \times 10^3 = 13 \times 88 \times \tau_{key} \times 25$

$\boxed{\therefore \tau_{key} = 33.39\ N/mm^2 < 40\ N/mm^2}$

Thus, the induced shear stress in key is less than permissible stress ∴ Design of key is safe.

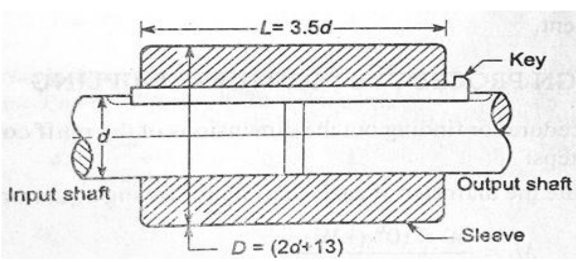
Crushing strength of key:-

$T = l_K \times \frac{t_K}{2} \times \sigma_{c\,key} \times d/2$

$954.93 \times 10^3 = 88 \times \frac{13}{2} \times \sigma_{c\,key} \times 25$

$\boxed{\sigma_{c\,key} = 66.78\ N/mm^2 < 80\ N/mm^2}$

∴ Design of key is Safe .. Design of key … $1\frac{1}{2}$ M



Q.5 Attempt any Two of the following (16 marks)

a) Give the design procedure of hand lever with neat sketch.

(Dig-1 mark and for each step 1 marks)

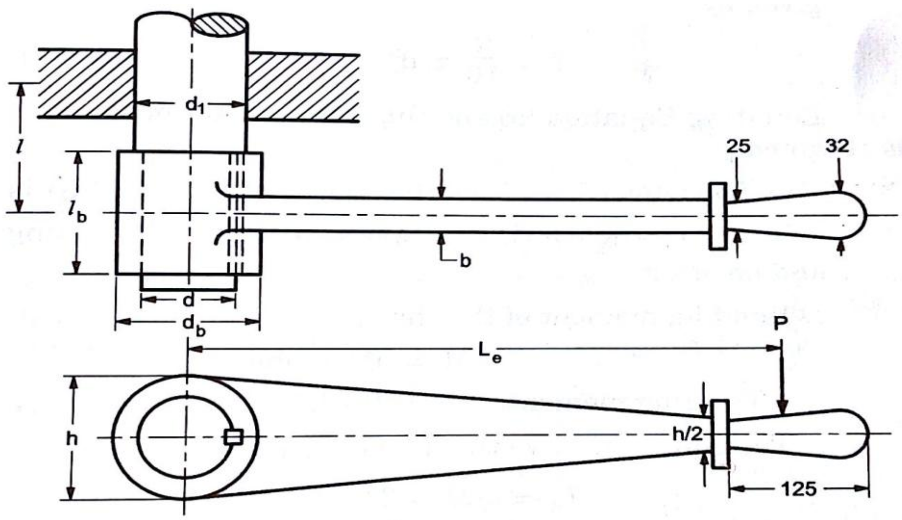


Fig. 2.4.2 : Hand lever

Let, P = Force of effort applied at the handle N.
L_e = Effective length of the lever arm mm
l = Overhead length of the shaft in mm
σ_t = Permissible tensile stress of the lever, N/mm^2
τ_s = Permissible shear stress of the lever in N/mm^2

Ans: Step No 1: Assume maximum effort applied by man with hand is 300N-400N

Step No 2: Torque =Effort ×Effective Length of lever

T= P × Le …………….(a) Then,

Diameter of shaft can be determine by considering shaft is under pure tension and resisting torque is given by

$$T=\frac{\pi}{16}\times\tau s\times d^3 \ldots\ldots\ldots\ldots (b)$$

Step No 3: To find the diameter of the shaft at the centre of bearing **(d_1)**

Considering it is under combine twisting and bending. So bending moment of shaft is given by,

M=P× *l*………….(c)

& twisting moment is given by, **T=P × Le………….(d)**

Combine twisting moment is given by,

$$Te=\sqrt{M^2+T^2}\ldots\ldots\ldots\ldots\ldots (e)$$

$$=\sqrt{(Pl)^2+(PLe)^2}$$

And by torsion equation, **$T_e = \frac{\pi}{16} \times \tau s \times d_1^3$ …………..(f)**

From equation **e**& **f** value of d_1 is obtain.

Step No 4: Diameter of the boss of the lever (d_b) is taken by empirical relation ,

d_p=1.6d ……………………(g)

Step No 5: Length of boss of the lever is taken by empirical relation **,**

***l_b*= d or 1.5d………………….(h)**

Step No 6: Fix the dimension of key from the standard shaft diameter, and as per the requirement square or rectangular key can be selected,

w= width of key

t = thickness of key

l_k=$l_b$length of key is same as that of length of boss

1. **For Square key** , w=t=d/4**………………………..(i)**
2. **For rectangular key,** w=d/4 & t=2/3w**…………….(j)**
3. & length of key can be obtained **by shearing and crushing failure of key.**
   Considering shearing of key,
   $T=\tau_k \times w \times l_k \times d/2$ …………………..(k)
   Considering crushing failure of key,
   $T=\sigma_c \times l_k \times d/2 \times t/2$…………………..(l)

Step No 7:

(7) Dimension of the lever cross section :

Consider the rectangular cross-section of lever.

Let, b = width of the lever, mm

t = depth or thickness of the lever mm

The width of the lever is taken as 2 to 3 times the thickness (t)

$$\boxed{b = 2\,t \text{ or } 3\,t}$$

The lever is subjected to bending moment. The maximum bending moment on the lever is taken near the boss.

$$M = P \times \left(L_e - \frac{d_b}{2}\right)$$

$$\therefore \text{Maximum bending stress} = \frac{M}{Z} = \frac{M}{I/y}$$

$$= \frac{P \times \left(L_e - \frac{d_b}{2}\right)}{\frac{b\,h^3}{12}\left(\frac{h}{2}\right)}$$

$$\sigma_b = \frac{P\left(L - \frac{d_b}{2}\right) \times \frac{h}{2}}{\frac{b\,h^3}{12}} = \frac{6\,P\left(L - \frac{d_b}{2}\right)}{b\,h^2}$$

Q.5 (b) Explain with neat sketch Design procedure of bush -pin type flexible coupling.**(Dig-1 mark,step 1,2,3,4-1mark each, and for pin design 3marks)**

Ans: It consist of two shafts, two key, flanges, key, pin, rubber bush, brass bush, & pin ,following are the designations of various coupling dimensions which are use for the design procedure.

d = diameter of shaft , **d_1**= diameter of enlarge portion of pin ,

d_2 =diameter of rubber bush , **d_b**= nominal diameter of pin,

D = outer diameter of hub , **D_1**= diameter of bolt circle

D_2= outer diameter of flange **,** ***l***= length of hub

l_b= length of bush in hub , **t_f**= thickness of flange **t_p**=thickness of protection

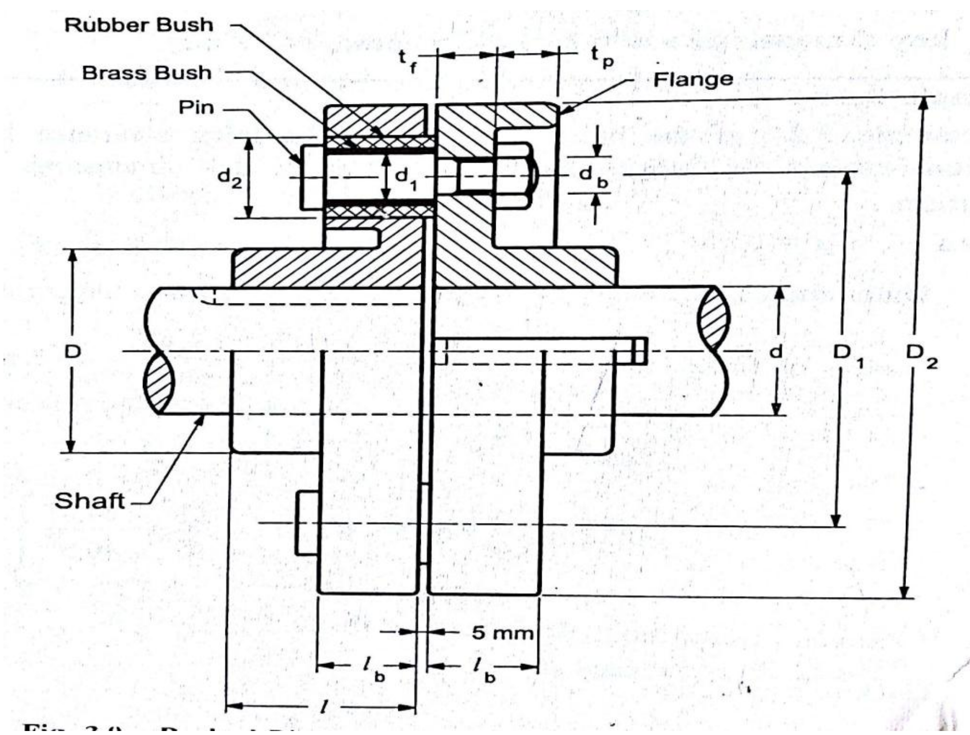
n = no of pins, and no of pins are selected from diameter of shaft

=3 for d upto 30mm

=4 for d upto 75 mm

=6 for d upto 110 mm

=8 for d upto 150 mm



Step no 1: Design of shaft (d)

Considering maximum shear stress,

$$T=\frac{\pi}{16} \times \tau_{max} \times d^3 \text{..........................(a)}$$

Step no 2: Design of key

Fix the dimension of key from the standard shaft diameter, and as per the requirement square or rectangular key can be selected,

w= width of key

t = thickness of key

l_k=length of key =**1.5d**by proportion…………..(1)

1. **For Square key** , w=t=d/4**………………………..(b)**
2. **For rectangular key,** w=d/4 & t=2/3w**…………….(c)**
3. & length of key can be obtained **by shearing and crushing failure of key.**
   Considering shearing of key,
   $T=\tau_k \times w \times l_k \times d/2$ …………………..(2)
   Considering crushing failure of key,
   $T=\sigma_c \times l_k \times d/2 \times t/2$…………………..(3)
   - Consider maximum value of length of key from equation 1,2,3

Step no 3: Design of hub

1. D = outer diameter of hub **D=2d**
2. *l*= length of hub ***l* =1.5d**
3. Tortional shear stress in the hub can be calculated from below equation,

$$T=\frac{\pi}{16} \times \tau_h \times (1-k^4)$$

Where, k=d/D

Step no 4: Design of flange

1. t_f= thickness of flange **t_f=0.5d**
2. t_p =thickness of protection **t_p =0.25d**
3. **Torque transmitted by flange**
   T=shear area × direct shear stress× outside radius of hub

$$T=\pi D t_f \times \tau_f \times D/2$$

From above equation value of τ_f can be calculated and checked

Step no 5: Design of pin

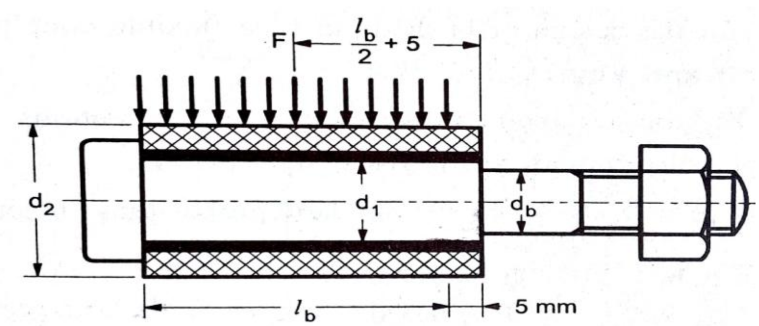


Fig .shows the pin with the rubber bush .The design of pin is as follows

- Nominal Diameter of pin (d_b)

$$d_b = \frac{0.5d}{\sqrt{n}} \longrightarrow ①$$

- Diameter of enlarge portion of pin (d_1)

$$d_1 = d_b + 4 \longrightarrow ②$$

- Outer Diameter of Rubber bush (d_2)

Assume that brass bush 2 mm thickness & rubber bush 6mm thickness are fitted

$$d_2 = d_1 + (2\times2) + (2\times6) \longrightarrow ③$$

- Dia. of bolt circle (D_1)

$$D_1 = D + d_2 + (2\times8) \longrightarrow ④$$

- Length of bush in the flange (l_b)

considering bearing pressure on pin

$$F = P_b \times d_2 \times l_b$$ &

Torque Transmitted,

$$T = n \times F \times D_1/2,$$

$$T = n \times P_b \times d_2 \times l_b \times D_1/2 \longrightarrow ⑤$$

Q.5 (c)A rail wagon of mass 20tones is moving with a velocity 10kmph .It is brought to arrest by using two buffer springs of 300mm dia. The maximum deflection of spring is 250 mm. An allowable shear stress for spring material is 800 mpa. Design the spring for buffers.

Ans: Given data: m=20tones=20×10^3 kg, v= 10km/hr =2.77 m/sec,

δ_{max}=250 mm, τ=800 N/mm^2 ,D=300mm, no. of buffer spring=2

Design of spring-Step no 1

Diameter of spring wire ,

Kinetic energy of railway wagon=Energy stored by the spring

K.E = U

$\frac{1}{2}\times m\times v^2$ = no of buffers$\times 1/2\times W\times \delta$

$$\tfrac{1}{2}\times20\times10^3\times(2.77)^3\times10^3 = \tfrac{1}{2}\times2\times W\times250$$

$$72.90\times10^6 = 250W$$

$W=291.60\times10^3$ N………………………………1mark

As we know that torque, T=W× (D/2)

$$=291.60\times10^3\times(300/2)$$

$T=43.74\times10^6$ N-mm……………………………..1mark

Twisting moment is given by,

$$T=\frac{\pi}{16}\times\tau\times d^3$$

$$43.74\times10^6=\frac{\pi}{16}\times800\times d^3$$

Diameter of spring wire, d =65.31mm……………….1mark

Step no 2-Numbers of turns **(n)**

As value of G is not given assume it (84×10^{3})

$$\delta=\frac{8\times W\times D3\times n}{G\times d4}$$

n=6.06…. ..Say 7 numbers of turns………………1mark

Assuming square and grounded ends, total numbers of turns is given by**,**

n'=n+2=7+2=9 numbers of turns………………….1mark

Step no 3-Solid length (Ls)

Ls=n'×d= 9×65.31 =587.79mm………………….1mark

Step no 3-Free length (Lf)

$Lf=n'\times d\times\delta_{max}\times0.15\times\delta_{max}$

Lf=875.29mm……………………………………….1 mark

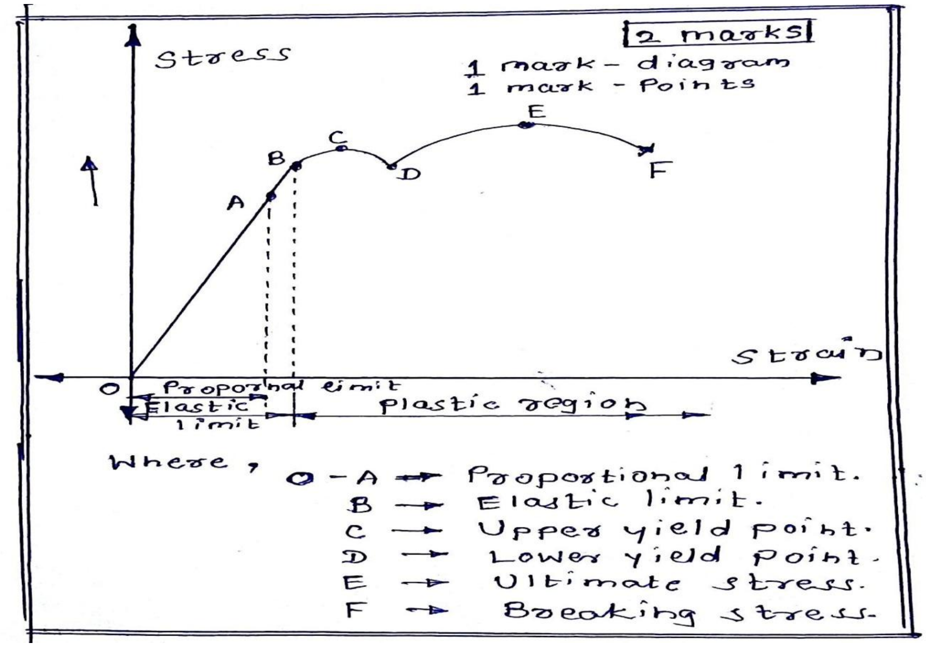
Step no 3-Pitch of the coil (p)

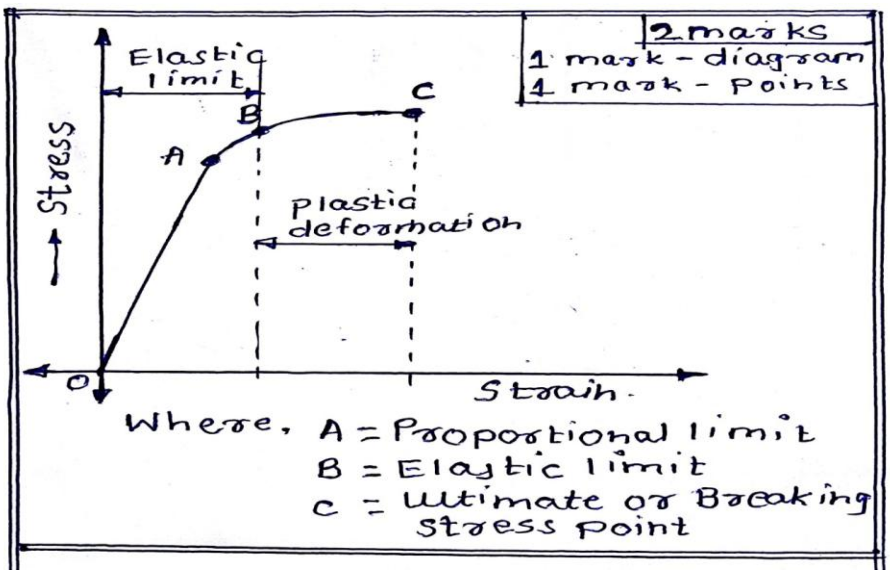
p = (Free length)/ (n'-1)

=109.41mm……………………………………….1 mark

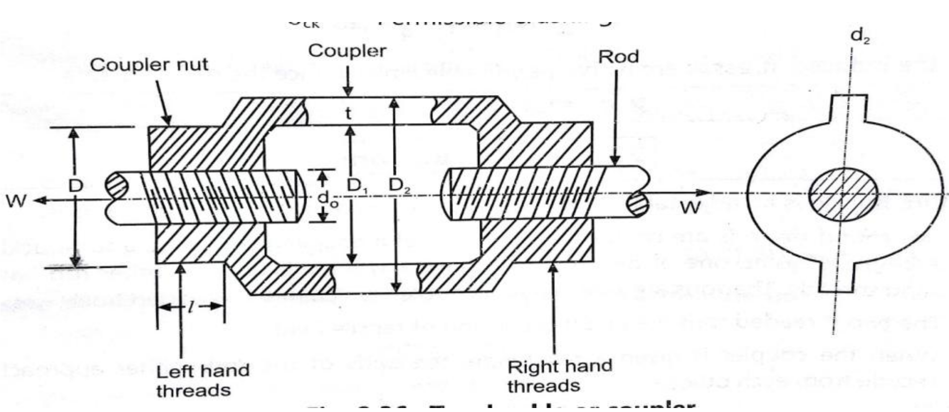
Q.6 Attempt any FOUR of the following (16marks)

a) Draw stress strain diagram for ductile and brittle material state all points on it**.**





Q.6 (b) Write any four equation s in the design of turn buckle with relevant sketches **(1 mark for each step or equation)**



Where,

W=design load =1.3 or1.4 times load carried by rods

τ=permissible shear stress in N/mm^2 **σ_t**=permissible tensile stress in N/mm^2

σ_{ck}=permissible crushing stress in N/mm^2 **dc**=core diameter of rod in mm, **do**=nominal diameter of rod in mm **p=**pitch of the thread in mm ,**n=**no threads,

***l*=**length of coupler nut in mm **D**=diameter of coupler nut,

D_1=inside diameter of coupler **D_2**=outside diameter of coupler,

t=thickness of coupler

Step 1: Design of rod (d_c)

Considering tensile failure,

$$\sigma_t = \frac{W}{\frac{\pi}{4} \times d_c^2} \quad \text{———— 1 mark}$$

after calculating dc, do & pitch can be determine from std. table.

Step 2: Design of coupler nut (l)

Considering shear failure,

$$\tau = \frac{W}{\pi d_c l}$$

Step 3: Checking of crushing stress induced in thread (σ_c)

$$\sigma_c = \frac{W}{\frac{\pi}{4}[d_o^2 - d_c^2] \times n \times l}$$

Wher, $n = \frac{1}{\text{Pitch}}$

Step 4: Design of coupler nut (D)

Considering tensile failure,

$$\sigma_t = \frac{W}{\frac{\pi}{4} \times [D^2 - d_o^2]}$$

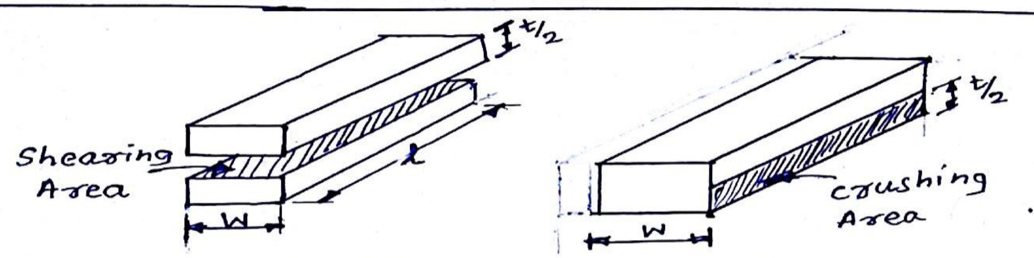
Step 5: Design of outer dia. of coupler [D_2]

$$\sigma_t = \frac{W}{\frac{\pi}{4}[D_2^2 - D_1^2]}$$

Where, $D_1 = d_o + (6 \text{ to } 8 \text{ mm})$

$t = 0.75\, d_o$

Q.6 (c) For a square key equally strong in shearing and crushing, show that crushing stress is twice the shearing shear stress.



1 mark

(a) Shearing failure (b) Crushing failure

- Considering shearing failure of key,

from fig (a),

Torque Transmitted by key $= T = F \times \frac{d}{2}$ — (1)

Where, F = Shearing force (tangential)

F = Shearing Area × Shearing stress

$= l \times w \times \tau$, put in eq^n 1

$T = l \times w \times \tau \times \frac{d}{2}$ ——— (2) 1 mark

- Considering crushing failure of key

from fig (b),

F = tangential crushing force

= crushing area × crushing stress

$= l \times \frac{t}{2} \times \sigma_{ck}$, put in eq^n 1

Torque transmitted by key $= T = F \times \frac{d}{2}$

$T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ ——— (3) 1 mark

- If key is equally strong in shearing & crushing,

(2) = (3)

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

as, for square key $w = t$ 1 mark

$$\boxed{\frac{\tau}{\sigma_{ck}} = \frac{1}{2} \text{ or } \sigma_{ck} = 2\tau}$$

Q.6 (d) A closed coil helical spring of 12 active coils has spring stiffness of K N/m^2 it is cut into two springs having 4 and 8 turns .Determine the spring stiffness of resultant spring.

Given data:

$n_1 = 4$, $n_2 = 8$, $n = 12$, K = spring Stifness

Stifness $(S) = K = \frac{W}{\delta}$

$$\delta = \frac{8 W D^3 n}{G \times d^4}$$ 1 mark

$$\therefore \quad K = \frac{G \times d^4}{8 D^3 n} = A \times \frac{1}{n}$$

$$= A \times \frac{1}{12} \qquad \left[A = \frac{G \times d^4}{8 D^3}\right]$$

Thus, $A = 12K$ 1 mark

Similarly, $K_1 = A \times \frac{1}{4} = 12K \times \frac{1}{4}$

$$\boxed{K_1 = 3K}$$ 1 mark

&

$K_2 = A \times \frac{1}{8} = 12K \times \frac{1}{8}$

$$\boxed{K_2 = 1.5K}$$ 1 mark

K_1 & K_2 are the stiffness of resultant spring

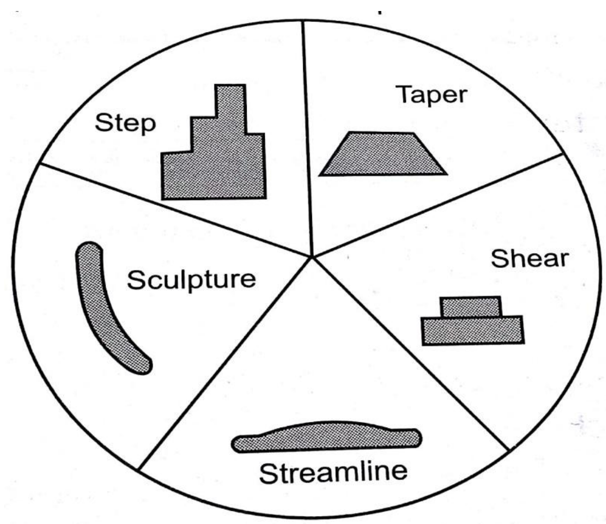
Q.6 (e) State any four name of rolling contact bearing. State the application of each**.**
(1 mark for each name and its application)

Sr.no	**Name of bearing**	**Application**
1	Deep grove ball bearing	Electric motor
2	Needle roller bearing	Differential of an automobile
3	Taper roller bearing	Axle housing of an automobile
4	Cylindrical roller bearing	High speed applications(automobile etc)
5	Single row deep grove ball bearing	Automobile wheels

Q.6 (f) State the importance of shape and colour in the design of aesthetics.

(2 marks for shape and 2marks for colour)

1. SHAPE: There are five basic shape of the products, such as step, taper, shear, streamline and sculpture as shown in fig.



- **Step form:**The step form is a stepped structure having vertical accent. It is similar to the shape of multistorage building.
- **Shear form:** It has square outlook**.**
- **Streamline form:** It has streamline shape having a smooth flow as a seen in automobile and aeroplane structure.
- **Sculpture form:** It consist of hyperboloids ,paraboloids and ellipsoids.
- **Taper form:** It consists of taper blocks or tapered cylinders.

2. COLOUR:

Colour is the major contributors to the aesthetic appeal of the product, many colours are linked with different moods and conditions.

Sr.no	**Colour**	**Meaning**
1	Red	Danger, hot
2	Orange	Possible danger
3	Yellow	Caution
4	Green	Safe
5	Blue	Cold
6	Gray	Dull