TH-2 DESIGN OF MACHINE ELEMENTS

1.0 Introductions

1.1 Introduction to Machine Design & classify it.
1.2 Different mechanical engineering material used in design with their uses & their mechanical & physical properties.
1.3 Define working stress, yield stress, ultimate stress & Factor of Safety & Stress - strain curve for M.S & C.I
1.4 Modes of Failure (By elastic deflection, general yielding & fracture)
1.5 State the factor governing the design of machine elements.
1.6 Describe design produce.

2.0 DESIGN OF FASTENING ELEMENTS:-

2.1 Joints and their classification
2.2 State types of welded joints.
2.3 State advantages of welded joints over other joints.
2.4 Design of welded joints for eccentric loads.
2.5 State types of riveted joints & types of rivets
2.6 Describe failure of riveted joints.
2.7 Determine strength & efficiency of riverted joints.
2.8 Design riverted joints for pressure vessel
2.9 Solve numerical on welded joint and riverted joints.

3.0 DESIGN OF SHAFTS & KEYS:

3.1 State function of shafts.
3.2 State materials for shafts.
3.3 Design solid & hollow shaft to transmit a given power at given rpm based on.
 a) strength: (i) shear stress., (ii) Combined bending tension;
 (b) Rigidity: (i) Angle of twist, (ii) Deflection, (iii) Modulus of rigidity.
3.4 State standard size of shafts as per I.S.
3.5 State function of keys, types of keys & material of keys.

3.6 Describe failure of key, effect of key way

3.7 Design rectangular sunk ~~by~~ key considering its failure against shear & crushing

3.8 Design rectangular sunk key be using empirical relation for given diameter of shaft

3.9 State ~~application~~ specification of parallel key, gib-head key, taper key as per I.S.

3.10 ~~3.10~~ Solve numerical on Design of shaft & keys.

4.0 DESIGN OF COUPLING

4.1 Design of shaft coupling

4.2 Requirements of a good shaft coupling

4.3 Types of coupling

4.4 Design of sleeve or Muff-coupling

4.5 Solve simple numerical on above.

5.0 DESIGN A CLOSED COIL HELICAL SPRING

5.1 Material used for helical spring

5.2 Standard size spring wire (SWG)

5.3 Terms used in compression spring

5.4 Stress in helical spring of a circular wire

5.5 Deflection of helical spring of circular wire

5.6 Surge in spring

5.7 Solve ~~8~~ numerical on design of closed coil helical compression spring.

Introduction to machine Design :-

It is the creation of new & better machines & improving the existing one so that cost of production is operation will be economical.

Imp classification of ~~Deg~~ Design

1) Adaptive Design → Here the existing component is modified
→ so that its operation becomes smoother for this design no technical knowledge is required.
→ Here generally the change of design takes place by the experience of the operator or user.

2) Development Design →
→ Here some components are replaced by existing components or some new technology is added to increase its performance to increase its wide range of application.
→ Here technical knowledge scientific training & design ability is required for modification.

3) New Design :-
→ It is done for a complete new component. It needs lot of ~~nee~~ research technical ability & creative thinking.

General considerations in m/c design

1) Type of load & stresses caused by the load
2) Kinematic of the m/c (motion of the parts)
3) Selection of material
4) Form & size of parts
5) Convenient & economical features
6) Use of standard parts
7) Safety of operation
8) No. of m/cs to be manufactured
9) workshop facilities
10) Frictional resistance & lubrication
11) Cost of construction
12) Assembling.

- Types of Load! -

Load → Any external force acting upon a m/c parts.

i) Dead or steady load → A load is said to be steady when it does not change in ~~many~~. magnitude or dirn.

ii) Live or variable load → when it changes continously

iii) Suddenly applied or shock load → when load is applied or removed suddenly

iv) Impact load → when the load is applied with some initial velocity

Imp - Working stress

During design of m/c parts, the stress is always taken lower than the maxm or ultimate stress at which failure of the material takes place. This ~~process~~ stress is called working stress / design / safe / allowable stress

Imp Yield stress

The stress corresponding to yield point called yield point stress. After elastic limit, the stress increases at a faster rate with any increase in the stress before, it reaches the upper yield point. Here, the material yields before the load & there is an appreciable strain without any increase in stress.

Imp - Ultimate stress

The maxm load carrying capacity of a material to its original cross-sectional area is called ultimate stress

2 marks Factor of Safety

- It is defined as the ratio betn maxm stress to the work stress

For ductile material, $FS = \frac{\text{Yield point stress}}{\text{working stress}}$

" Brittle material, $FS = \frac{\text{ultimate stress}}{\text{working stress}}$

* FS is selected by the following considerations.
- → Type of material
- → mode of manufacture
- → Type of stress
- → General service condn
- → shape of the parts.

Mechanical properties of materials:-

1) Strength → It is the ability of a material to resist the external load without failure
2) Stiffness → It is the ability of a material to resist deformation under stress & measures stiffness
3) Elasticity → Property to regain its original shape after deformation when external load is removed
4) Plasticity → property to not to regain its original shape after deformation when external load is removed.
5) Ductility → property of the material to be ~~draun~~ drawn into wire with application of tensile load.
6) Brittleness → It is the property of breaking of the material with little permanent distortion
7) Malleability → ~~It is the~~ property ~~of bro~~ to deform the material into plate / flat shape.
8) Toughness → Ability to resist fracture due to high impact loads. It ↓ with heat.
9) Machinability → Ability to be machined
10) Resilience → Ability to absorb energy & to resist shock of impact loads. The amount of energy absorbed per unit volume within elastic limit
11) Creep → when a component is subjected to constant stress at high temp for a long period of time, it will undergo slow & permanent deformation is called creep.

12) **Fatigue** → When material is subjected to repeated stresses, it fails at stresses below the yield point stresses, called fatigue.

13) **Hardness** – It is the resistance to wear, scratching deformation & machinability etc.

← **Factors governing the design of m/c element!** –

5 mark Imp L10

Design procedure! →

$\boxed{\text{Need Analysis}}$ Complete statement of the problem indicating need, purpose

↓

$\boxed{\text{Synthesis of mechanisms}}$

↓

$\boxed{\text{Analysis of forces}}$

↓

$\boxed{\text{material selection}}$

↓

$\boxed{\text{Design of elements (size \& stresses)}}$

↓

$\boxed{\text{Modification}}$

↓

$\boxed{\text{Detailed drawing}}$

↓

$\boxed{\text{production}}$

max^m stress

$$F_S = \frac{\sigma_{max^m}}{\sigma_{allowable}}$$

σ_{yield} – (ductile)

$\sigma_{ultimate}$ – (brittle)

NEED ANALYSIS

→ Firstly a complete statement of the problem has to be ~~malt~~ melt by considering its need, aim on purpose for which the machine is required to be ~~desired~~. design

Synthesis of mechanism

↳ Here all possible mechanism on group of mechanism are selected which will give the desired motion

Analysis of forces

↳ Here the forces acting on each member of the machine and the energy ~~transfer~~ transmitted by each member is calculated.

Selection of Material

↳ Here the material which is based suited for each member of the machine is selected.

Design of element (Size & stress)

↳ Here size of each member of the machine is determined by considering the forces acting on the member and the permissible stresses for the material used. Always it has to be kept in mind that each member should not deflect on deform than the permissible limit.

Modification

↳ Here the size of member on part may modified with respect to past experience and suitability to manufacture.

→ Modification is done by consideration of reduction in overall cost

Detailed Drawing

↳ Here detailed drawing of each component is done and assembly of the machine with complete specification for the manufacturing process is ~~sug~~ suggested.

Production

↳ Here the component as per the drawing is manufacture

STRESS - STRAIN DIAGRAM :-

→ While designing different mechanical components, it is ~~nece~~ necessary to know how the material will function in required place of application or service.

→ Most of the mechanical properties of the material are found from standard tensile test.

→ Tensile test consist of gradually loading a standard specimen of a material & noting the corresponding values of load & elongation until the specimen fractures. The load is applied & measured by a testing machine called Universal testing Machine (UTM)

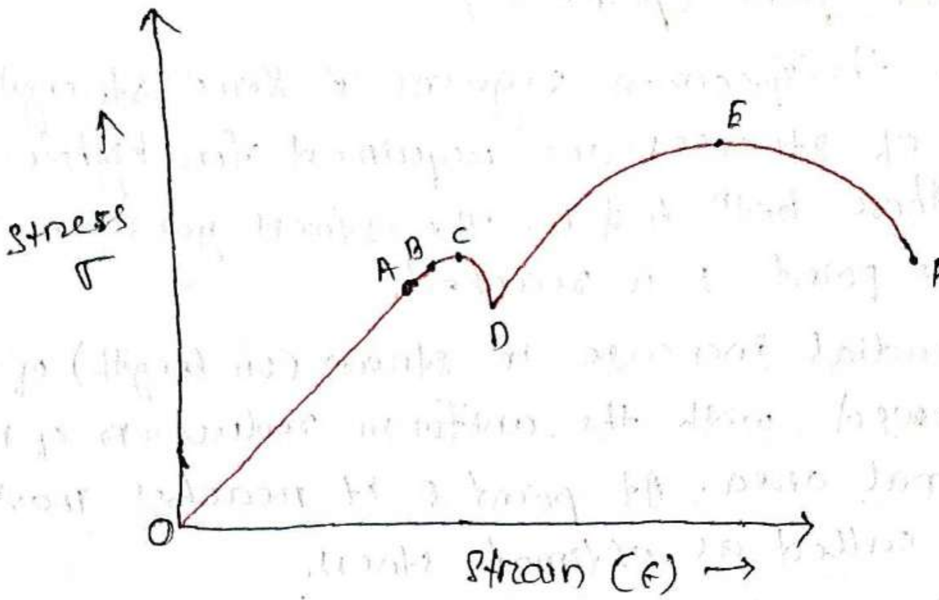
→ The stress is determined by dividing load values by the original cross-sectional area of the specimen.

→ Elongation is measured by determining the amount that two reference points on the specimen are moved apart by the action of the m/c

Gauge length → Original distance betn two reference points.

→ Strain is determined by dividing the elongation values by gauge length.

→ The values of stress & corresponding strain are used to draw the $\sigma - \epsilon$ diagram of the material tested.



(σ - ϵ curve for mild steel)

1) Proportional limit (point A)

From point O to A, $\sigma \propto \epsilon$

Hooke's law holds upto point A i.e proportional limit, defined as the stress at which the σ - ϵ curve begins to deviate from straight line.

2) Elastic limit (Point B)

If load is increased beyond point A upto point B, the material will regain its shape & size, when the load is removed i.e material has elastic property upto point B called as elastic limit (No permanent deformation)

3) Yield point (point C & D)

If material is stressed beyond point B, the plastic stage will reach i.e on removal of load also the material will not recover to its original shape & size.

Beyond point B, strain increases at a faster rate with any increase in stress until point 'C' is reached At point C, material yields before the load & there is an appreciable strain without any increase in stress

C → Upper yield point

D → Lower " "

4) Ultimate point (point E)

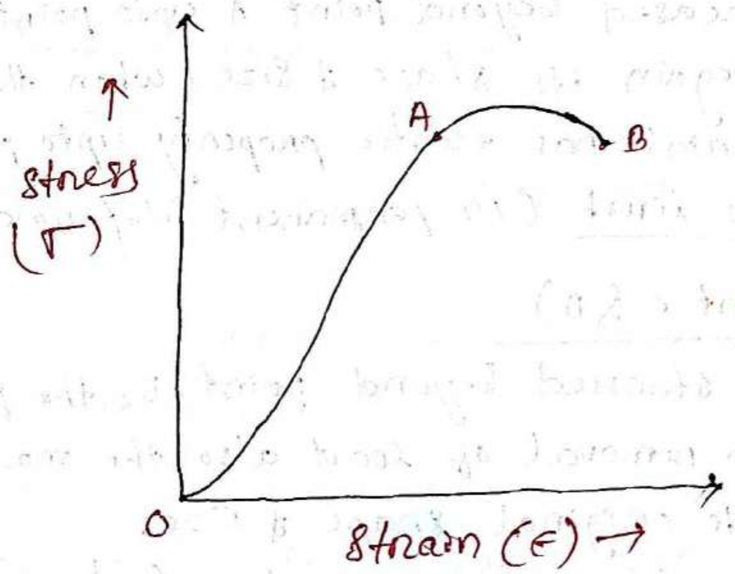
At D, the specimen regains some strength & higher values of stresses are required for higher strains than those betn A & D, The stress goes on increasing till the point E is reached.

the gradual increase in strain (or length) of the specimen is followed with the uniform reduction of its cross-sectional area. At point E it reaches maxm value of stress called as ultimate stress.

5) Breaking point (point F)

When load reaches ultimate point, neck is formed, which decrease the cross-sectional area of the specimen The stress required to break the specimen is less then the maxm stress, so stress is reduced until the specimen breaks at point F.

STRESS STRAIN CURVE FOR CAST IRON :-



From O to A : It is approximately propentional limit

From A to B : stress & strain both increases

point B → breaking or fracture point

Brittle material develop internal cracks first & then the cracks reach upto the surface. The internal cracks will propogat

eg (concrease, glass, cast iron)

On further increase of σ ultimately the cracks propagates inner to outer surface & finally the material fails. Here there is little or no plastic deformation

Q Modes of failure :→

A mechanical member may fail i.e it can not function satisfactorily by the following 3 modes of failure.

1) Failure by elastic deflection
2) " by yielding (general yielding)
3) " by fracture.

1) Failure by elastic deflection

→ In applications like transmission shaft supporting the gears, the maxm force acting on the shaft without affecting its performance is limited by the permissible elastic deflection.

→ Sometimes the elastic deflection results in unstable conditions such as buckling of columns or vibrations. The design of mechanical component in all these cases is based on the permissible lateral or torsional deflection.

→ The stresses induced in the component are not significant & properties of material are not of primary importance The E & G are the important properties & dimension of components are determined by the load deflection equation

→ In other words, in a components like columns, beams shafts, etc, the torsional deflection in an elastic region is termed as failure of the component.

2) Failure by general yielding

→ For ductile material deformation occures after the yield point, resulting in permanent deformation of the machine element which ultimately breaks at breaking point

→ So for ductile material, failure is usually considered to have occured when yielding i.e plastic deformation reach a limit, when engineering ~~usefull~~ usefulness of the part is destroyed, even through there is no rupture or fracture of the m/c part

→ So yield point is criteria of failure of ductile material subjected to static loading. This type of failure is called elastic failure.

3) Failure ~~of~~ by fracture :

→ In case of brittle materials the yield point of ultimate strain is very nearly equal to unity. So brittle material are considered to have failed by fracture with little or no permanent deformation.

→ Sudden separation or a breakage of a material along the cross-section normal to the direction of stress is called fracture

→ Fracture is sudden failure without plastic deformation Brittle different conditions.

a) Deflection :

↳ Under cond's of stable equ^n, such as the ~~stretch~~ stretch of a tension member; the angle of twist of shaft of the deflection of an end-loaded cantilever beam

b) Buckling :-

↳ Sudden deflection associated with unstable equⁿ of resulting in total applied gradually to a slender column ~~exceed~~ exceeds the ~~prat~~ Pcritical or Euler load

c) Elastic deflections

↳ These are the amplitudes of vibrations of a member sometimes associated with failure of the member resulting from objectionable noise, shaking forces, collision of moving parts with stationary parts etc, which result from the vibrations

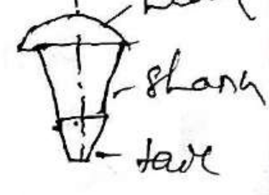
RIVETED JOINTS:-

Fastening are of 2 types :-

1) Permanent Fastening → which can not be disassembled without destroying the connecting parts.
Ex- welding, riveting, soldering, brazing etc

2) Temporary Fastening → which can be disassembled without destroying the connecting components.
ex- Screwed joints, keys, cotters, pin, knuckle & cotter joint etc.

→ Rivet is a short cylindrical shank with a head integral to
→ Rivet is Specified by shank dia. standard size are, 12, 14, 16, 18, 20, 22, 24, 27, 30, 33, 36, 39, 42, & 48 mm



→ Riveting < hot (end of shank is heated to 1000° to 1100°c they Blows are applied by hammer)
Cold (not like this)

Riveting < Hand riveting
m/c "

methods of riveting :-

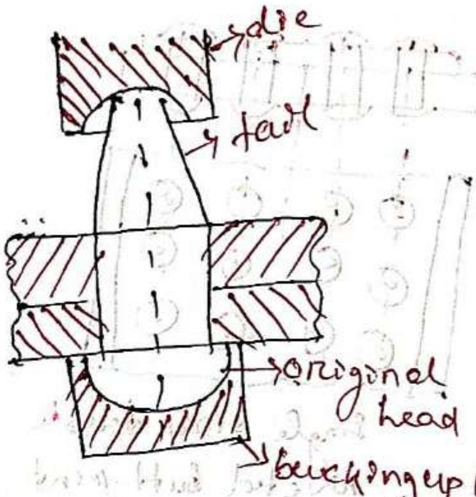
→ Function of riveted joint is to ensure strength & tightness
strength is required to prevent failure of the joint
tightness " " leakage in boiler or ship hul

→ The two plates which are required to be joined by rivet are first holed either by punching then reaming or by drilling.

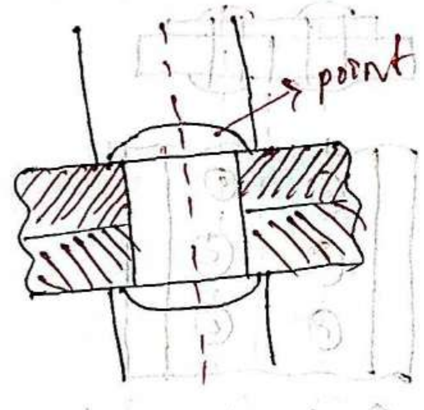
→ Now a cold rivet or a hot rivet is introduced in the hole of the plates. used in structural joint; used in leak proof joint

→ Then riveting is done by hand or m/c

→ In hand riveting, the original rivet head is backed up by a hammer or die bar then the die set is placed against the end to be headed then blows are applied by hammer. This cause shank to expand, thus filling the hole of tail is ~~Conov~~ converted into point



(Initial position)



(Final position)

Material of Rivets

It should be tough & ductile

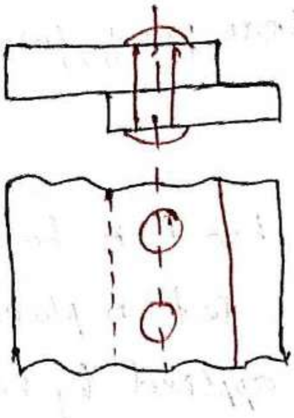
Stell (low c steel, Ni steel), brass Al, Co.

Types of riveted joints : -

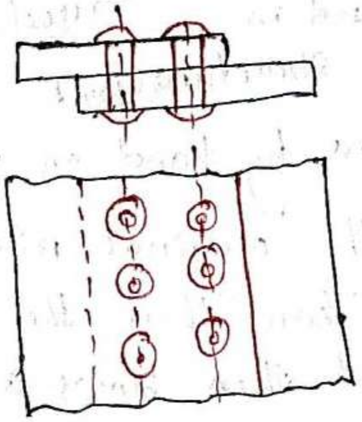
It is of 2 types
i) Lap joint → one plate overlap, on other
ii) Butt joint → Two plate are placed side by side

Single strap (one cover plate)

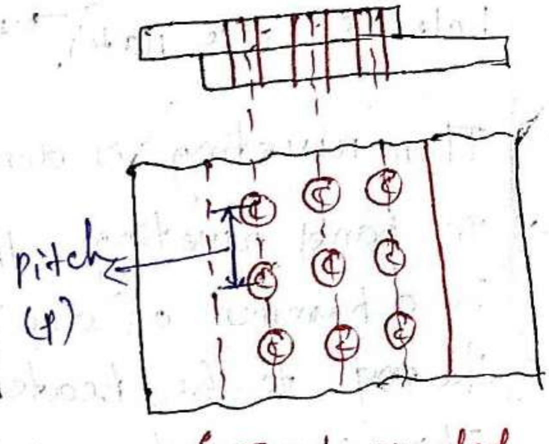
Double strap (two cover plate)



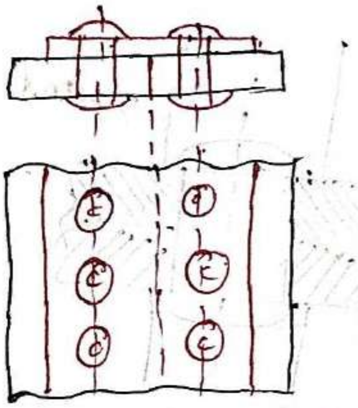
(Single riveted lap joint)



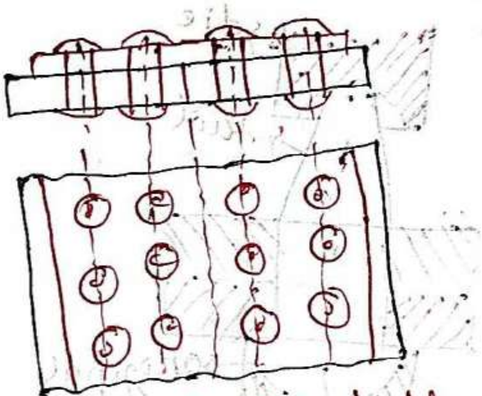
(Double riveted lap joint)



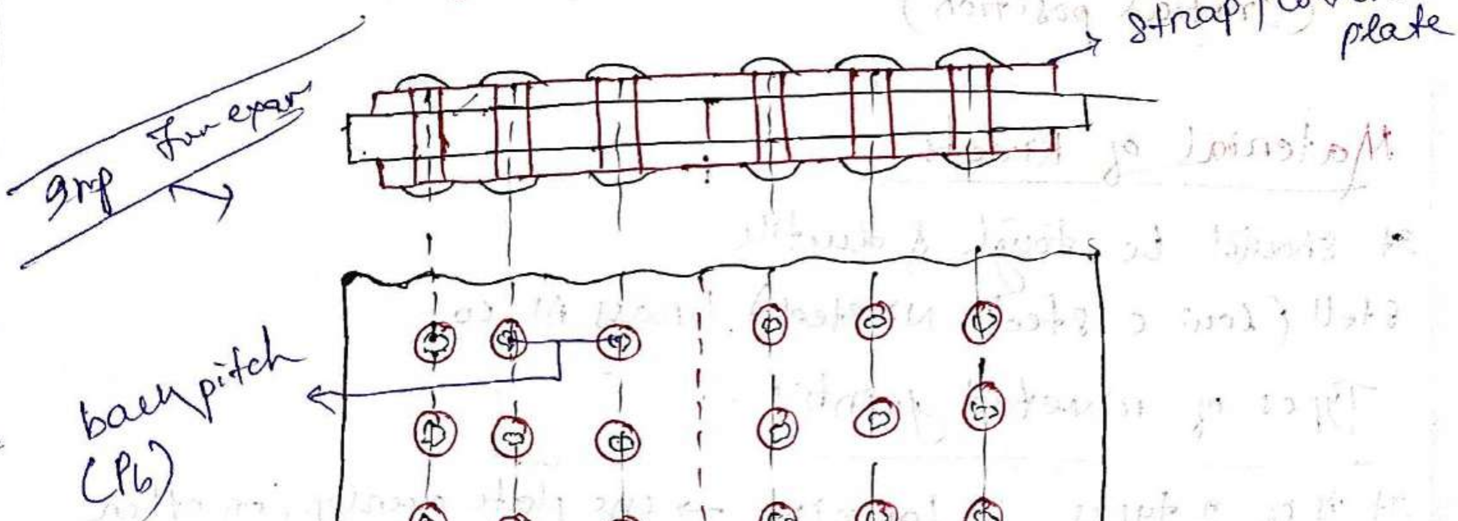
(Triple riveted lap joint)



Single strap single riveted butt joint



Single strap double riveted butt joint



(Double strap triple riveted zigzag butt joint)

Imp

Terms used in riveted joint:-

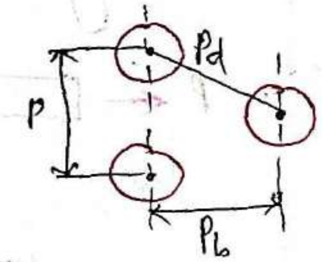
1) Pitch (P) → It distance from centre of ~~cent~~ one rivet to the centre of next rivet

2) back pitch (P_b) → It distance between the centre lines of the successive rows on two adjecent rows of rivet

3) Diagonal pitch (P_d) → Distance between centre of two adjacent rows of rivet in zig zag ~~re~~ riveting.

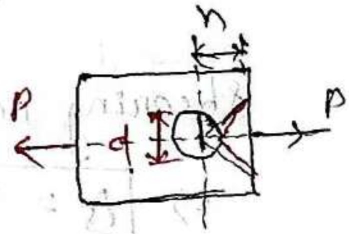
4) margin (m) → distance between centre of rivet to the nearest edge of the plate.

usually $P = 3d$
$m = 1.5d$



failure of riveted joint:-

1) Tearing of plate about its edge :-
Joint may fail due to tearing of plate at an edge To avoid this $m = 1.5d$ is taken



2) Tearing failure of plate across a row of rivets :-

let t = thickness of the plate
d = diameter of the rivet
P = pitch of the rivet joint
σ_t = permissible tensile stress for plate material
σ_c = " crushing " " rivet "
τ_s = " shear " " " "
P_T = Tensile load / strength act on the plate
P_c = crushing " " " rivet
P_s = shear " " " "

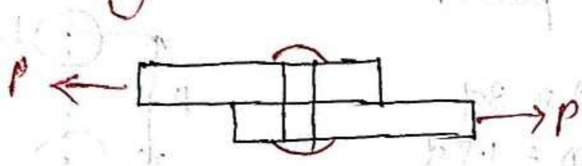
Tearing resistance of the plate across the rivet hole per pitch length = permissible tensile strength of plate × area of cross-section of plate

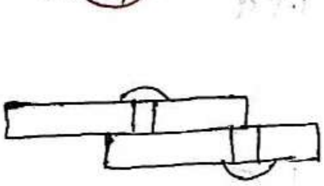
$P_t = \sigma_t A_t$

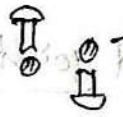
$$\boxed{P_t = \sigma_t (p-d) t}$$

∴ When tearing resistance (P_t) is greater than the applied load (P) per pitch length, then this failure will not occur.

3) Shearing failure of rivet :→





⇒  → failure area of rivet

Shearing resistance of rivet = $\tau \times A_s$

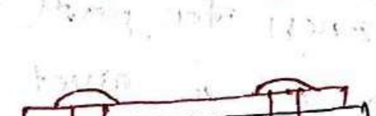
⇒ $\boxed{P_s = n \times \tau \times \frac{\pi}{4} d^2}$ (for n = 1) $A_s = \frac{\pi}{4} d^2$ for single shear

where n → No. of rows of rivet

$= 2 \times \frac{\pi}{4} d^2$ " double " theoretically

$= 1.875 \times \frac{\pi}{4} d^2$ / Acc^n to IBR for double shear)

→ For single strap butt joint :→



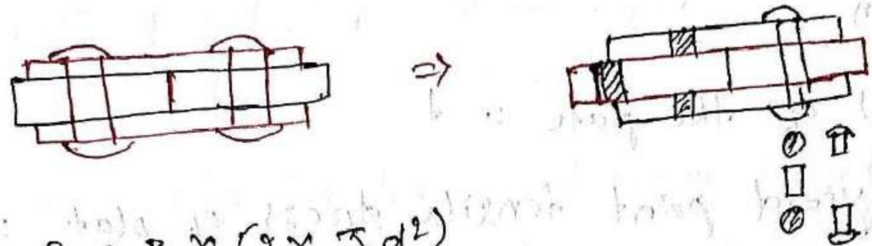
☐ deformat^n area

Here $\boxed{P_s = n \times \tau \times \frac{\pi}{4} d^2}$ (n=1)

Cover plate = t_1 = 1.125 t (for single cover)

" " = t_2 = '

→ For double strap butt joint



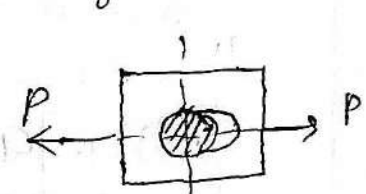
Here $P_s = 2 \times (2 \times \frac{\pi}{4} d^2)$

Acc^n to IBR, $\boxed{P_s = n\tau (1.875 \times \frac{\pi}{4} d^2)}$ (n = 2)

4) Crushing failure of plate on rivet :→

Due to crushing the rivet hole becomes an oval shape and hence the joint becomes loose such type of failure of rivet is also called bearing failure. Here bearing area is taken as the projected area of hole on rivet on diamentional plane.

crushing strength = σ_c × projected area of rivet
$= \sigma_c \times (d \times t)$



Generally $\boxed{P_c = n \sigma_c (d \times t)}$

n ⇒ no of rows of rivet

when crushing resistance (P_c) is > applied load (P) per pitch length then this type of failure will not occurs.

Efficiency of the riveted joint :→

Imp

It is the ratio between least of resistance of different modes to the strength of similar unriveted plate.

$$\eta = \frac{\text{least of } (P_t \text{ or } P_s \text{ or } P_c)}{\text{strength of unriveted plate}}$$

$$= \frac{\text{least of } (P_t / P_s / P_c)}{(p \times t) \sigma_t}$$

Steps for design of a riveted joint :→

1) Data given :-

thickness of the plate = t

max^m on yield point tensile stress of plate = σ_t'

" " " " shear " " rivet = τ'

" " " " crushing " " " = σ_c'

2) Calculation of permissible stresses

Assume factor of safety if not given

for ductile material FS = 4

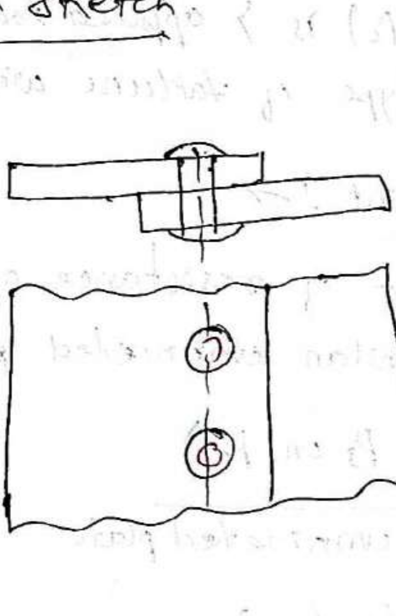
" brittle " FS = 6

permissible tensile stress of plate = $\sigma_t = \dfrac{\sigma_t'}{FS}$

" shear " " rivet = $\tau = \dfrac{\tau'}{FS}$

" crushing " " " = $\sigma_c = \dfrac{\sigma_c'}{FS}$

3) Rough sketch



if t < 8 mm

then calculate d by

$P_S = P_c$

$\Rightarrow d = ?$

d should > t

4) calculation of diameter of rivet (d)

$\boxed{d = 6\sqrt{t}}$ when $t > 8mm$ Unwins formula (Reference)

while using databook write page no, table no of formula in reference standardise the dia of rivet by refering data book.

Always take higher standard dia with reference to calculated dia (not possible then at least round off to next whole no)

5) calculation of pitch (p)

It can be found by tearing resistance of plate = shearing resistance of rivet

$\Rightarrow P_t = P_s$ (Rivet) (against the strength)

(plate) $\Rightarrow \sigma_t \cdot (p-d)t = n \times \tau \times \frac{\pi}{4} d^2$

$\Rightarrow \boxed{p = ?}$ This pitch is against strength

Acc. to IBR, $P_{max} = (Ct + 41.28)$ (Against lean proof)

value of C can be found from databook

calculate the two pitches and takes lower one for design

6) calculation of back pitch (P_b)

There is no P_b for single riveted joint

$P_b = 0.33p + 0.67d$ (for zig-zag riveting)

$P_b = 2d$ (for chain riveting)

7) calculation of margin (m) $\rightarrow \boxed{m = 1.5d}$

8) calculation of diagonal pitch (P_d) $\rightarrow P_d = \sqrt{\left(\frac{p}{2}\right)^2 + P_b^2}$

9) calculation of thickness of butt strap

here $t_1 > 10mm$, $t_1 = 1.125t$ for single strap butt joint

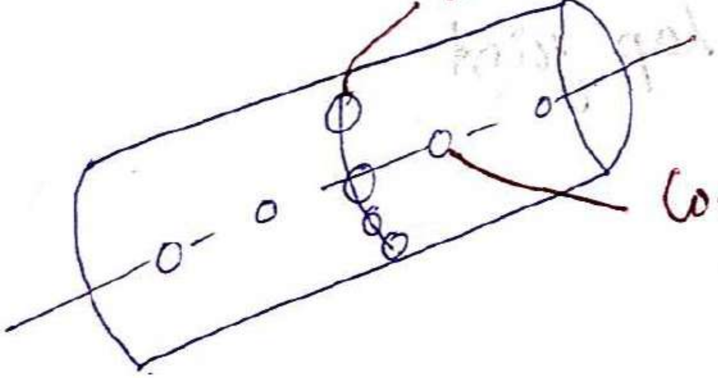
$t_1 = 0.725t$
$t_2 = 0.625t$] for double strap unequal width butt joint

$t_2 = t_1 = 0.725t$ (for double strap equal width butt joint)

Design of boiler joints :-

Imp A boiler shell consist of two types of riveted joints

i) longitudinal joint ii) circumferential joint



Longitudinal joint

→ It is carried out in the boiler shell to increase the diameter of the shell by joining ends of the plate

→ This joint is found along the length of the boiler.

Circumferential joint

→ Riveted joint carried out in the circumference side or in the periphery of the boiler shell is called circumferential joint

For ~~all~~ this a lap joint with one ring overlapping the other alternately is used.

→ It is found along the periphery purpose is to increase the length of the boiler shell

* In case of cylindrical shell two types of stresses are induced.
i) Hoop stress
ii) Longitudinal stress

→ Out of these two stresses the magnitude of hoop stress is twice that the longitudinal stress

$\sigma_h = \frac{pd}{2t}$ $\sigma_l = \frac{pd}{4t}$

→ Therefore the strength of the longitudinal joint should be more than that of the circumferential joint to withstand the hoop stress.

→ So in boiler ~~sh~~ shell butt joint is carried out in the longitudinal side & lap joint is carried out in circumference side

* Strength of butt joint is more than lap joint & more expensive,

So in boiler shell the joints are
i) Longitudinal butt joint
ii) Circumferential lap joint

Design →

let P = Internal pressure inside the boiler

t = thickness of boiler shell

D = Internal dia of boiler

σ_t = permissible tensile stress of boiler shell material

η_l = efficiency of longitudinal joint

Lap joint	max^m η	Butt joint (double strap)	max^m η
Single rivetal	→ 63.3	Single riveted	→ 63.3
double "	→ 77.5	Double "	→ 86.6
Triple "	→ 86.6	Triple "	→ 95
		Quadraple "	→ 98.1

$$\sigma = \frac{PD}{2t}$$

$$\Rightarrow t = \frac{PD}{2\sigma} = \frac{PD}{2\sigma_t \eta_l} + CA$$ ← corrosion allowance of 1mm

Step-1 Dia of rivet

It is same for both circumferential & longitudinal joint as t is same

$d = 6\sqrt{t}$ $\quad t > 8mm$ $\quad$ ~~Unm~~ Unwin's emperical relatⁿ

If $t < 8$ mm $\quad P_s = P_c \quad (\because d \not\approx t)$

Step-2 No of Rivets

lap joint so single shear

It can be obtained by equating the force due to pressure with equivalent shear strength of rivet

pressure load = shear strength of all rivet

$P \times$ area $= \tau \times A_s \times N$ ← No of rivet

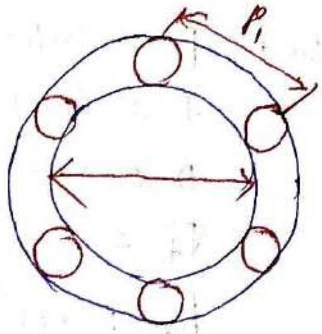
$$\Rightarrow P \times \frac{\pi}{4} D^2 = \tau \times \frac{\pi}{4} d^2 \times N$$

$$\Rightarrow \boxed{N = \frac{PD^2}{\tau d^2}}$$

Step-4 pitch of rivets

$$P_1 = \frac{\pi(D+t)}{\text{no. of rivet in each row}}$$

$$= \frac{\pi(D+t)}{(N/n)}$$



$P_{max} = (t + 41.28)$

If $P_{max} < P_1$ take $P_1 = P_{max}$

Step-3 : No. of rows of rivet

$$n = \frac{\text{Total no. of rivets}}{\text{no. of rivets in one row}} = \frac{N}{\left(\frac{\pi(D+t)}{P_1}\right)}$$

Assume no. of rows by thumb rule

$N < 10$, assume $n = 1$

$N > 10$, assume $n = 2$

Step-5 : back pitch

$P_{b_1} = 2d$ (for design)

$P_{b_1} = 0.33 P_1 + 0.67d$ (for zig-zag)

Step-6 : margin → $m = 1.5d$

Step-7 : overlap

overlap = (No of rows of rivets − 1) P_b + m

$= (n-1) P_{b_1} + m$ (margin)

Step-8 : efficiency

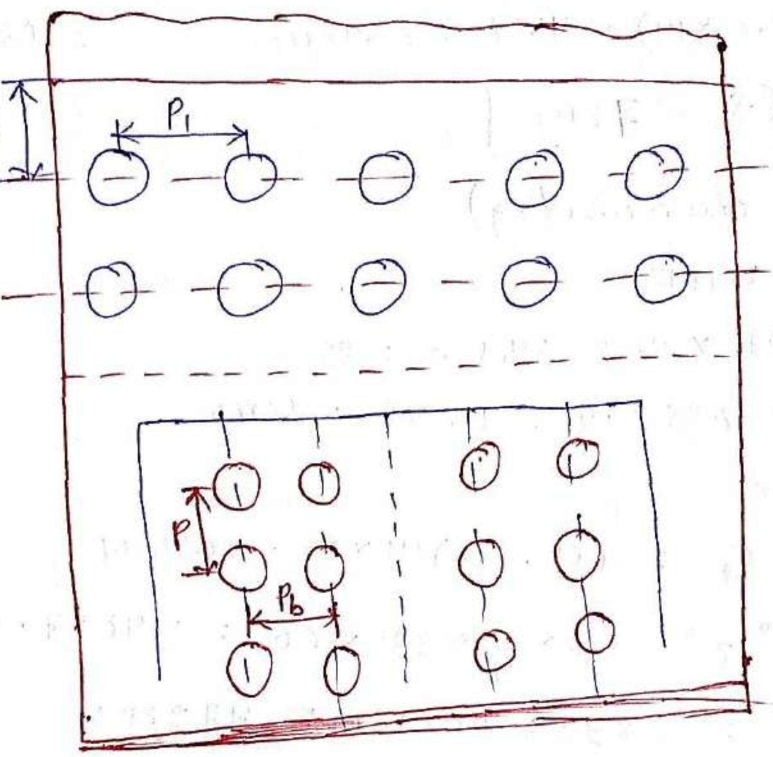
$$\eta = \frac{\text{tensile strength of riveted plate}}{\text{" " " unriveted plate}}$$

$$\Rightarrow \eta = \frac{(P_1 - d) \times \sigma_t}{P_1 \times \sigma_t}$$

$$\Rightarrow \boxed{\eta = \frac{P_1 - d}{P_1}}$$

imp

Design for longitudinal butt joint :-
- Same as done in riveted joint :-



Q A cylindrical pressure vessel with 0.8m internal dia subjected to internal p of 2MPa. The permissible stresses in tension, shear & compression are 80, 60, 120 MPa η of longitudinal joint is 80%. For the purpose of calculating plate thickness Design the complete boiler joint.

Soln Longitudinal butt joint

Given $D = 0.8m = 800\ mm$, $\eta_L = 80\%$.

$P = 2MPa$; $\sigma_t = 80MPa$,

$\tau = 60MPa$, $\sigma_c = 120MPa$,

i) calculation of t

$$t = \frac{PD}{2\sigma_t \eta_L} + \underset{\text{corrosion allowance}}{CA} = \frac{2\times 800}{2\times 80\times 0.8} + 1 = 14mm$$

ii) Calculation of d of rivet

$d = 6\sqrt{t} = 6\sqrt{14} = 22.44 \approx$ ~~23mm~~ 24 mm

iii) Calculation of p of rivet

$P_s = P_t$

$\Rightarrow 2\times \frac{\pi}{4}d^2\times n = \cancel{(p-d)\times F_t}$

$= (p-d)t\sigma_t$

(assuming double riveted chain butt joint) with single strap)

$\Rightarrow 60 \times \frac{\pi}{4} \times \cancel{23}^2 \times 2 = (P - \cancel{23}) 14 \times 80$

$\Rightarrow P = \cancel{6\ldots mm}\ 76.59 mm$ Ref $c = 2.8$ for single.

$P_{max} = (ct + 41.28)$ $\longrightarrow$ $P_{max} = ct + 41$

$= (2.8 \times 14) + 41.28 = 80.48 mm$ $= (3.06 \times 14) + 41$

$= 83.84$

design $\boxed{P = \cancel{76.5} \approx \cancel{7}8 mm}$

iv) $P_b = 2d$ (let chain riveting)

$= 2 \times 23 = 46 mm$

v) $m = 1.5d = 1.5 \times 23 = 34.5 \approx = 35$

vi) $t_1 = 1.125 t = 1.25 \times 14 = 15.75 = 16 mm$

vii) η calculation

$P_t = (P - d) t \sigma_t = (68 - 23) 14 \times 80 = 50400 N$

$P_s = n \times \frac{\pi}{4} d^2 \cdot \tau^2 = 2 \times \frac{\pi}{4} \times 23^2 \times 60 = 49857.07 N$

$P_c = n d t \sigma_c = 2 \times 23 \times 14 \times 120 = 77280 N$

$P = P t \sigma_t = 78 + 14 \times 80 = \cancel{76160 N}\ 87360 N$

$\eta = \frac{49857.07}{76160} = 65.46\%$

circumferential lap joint

i) $t = 14 mm$ iii) $N = \frac{PD^2}{2d^2} = \frac{2 \times 800^2}{60 \times 23} = 40.32 \approx 44$

ii) $d = 23 mm$

iv) n (no. of rivets in each row)

let us assume double riveted lap joint

$n = \frac{44}{2} = 22$ nos.

v) Pitch $(P_1) = \frac{\pi(D+t)}{22} = \frac{\pi(800+14)}{22} = 116.23$

$P_{max} = (ct + 41.28 = (3.5 \times 14) + 41.28 =$

vi) back pitch $(P_b) = 2d = 2 \times 23 = 46 mm$

vii) margin (m) $= 1.5d = 1.5 \times 23 = 34.5 = 35 mm$

viii) overlap $= (n-1) P_{b_1} + m = (2-1) 46 + 35 =$

ix) $\eta = \frac{P_1 - d}{P_1} =$

Design $P_1 =$ — (assuming chain riveting)

18/10/22

WELDED JOINT

9/4

Advantages of welded joint over riveted joint

① → Str. are lighter than riveted joint

⑨ → It's provides maxm efficiency (may be 100%) which is impossible in riveted joint

⑤ → Welded assemblies are tide and leak proof than riveted assemblies.

⑦ → Due to ~~but~~ drilling of holes in riveted joint crossectional area decrease, which results in stress concentration.

② → The str. has smooth and ~~elegant~~ plesant appearance

③ → The project^n of riveted adversly affect the appearance.

④ → Design of welded assemblies can be easily and economically modified to need the requirement.

⑥ → Cost of welded assemblies is less due to eliminat^n of rivets, cover plates etc,

⑧ → It's takes less time than riveting

Dis Advantages of welded joint over riveting

→ capacity of welded str. to damped vibration of is less.

→ It's results thermal distribution of parts, so stress reliving heat treatment is required.

→ Quality of joint depends on skill of welder.

→ Inspect^n of welded joint is more ~~Specilized~~ Specialised & more costly.

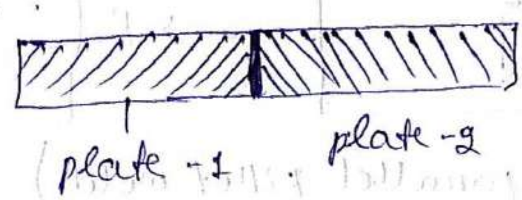
Types of welded Joints

→ It is of 2 types — ① Butt joint
② ~~Fillet~~ / lap Joint / Fillet Joint

18/10/22

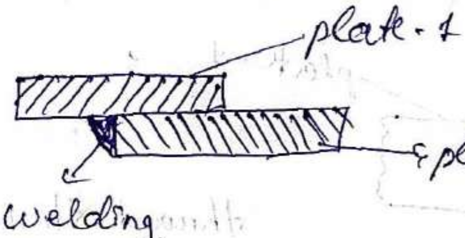
BUTT JOINT

→ Here 2 metal plate are placed side by side.
Types of butt joint depends on its thickness of reliability.



LAP JOINT / Fillet Joint

→ Here 2 metal plates are placed one over the another in over-lapping manner/ways.



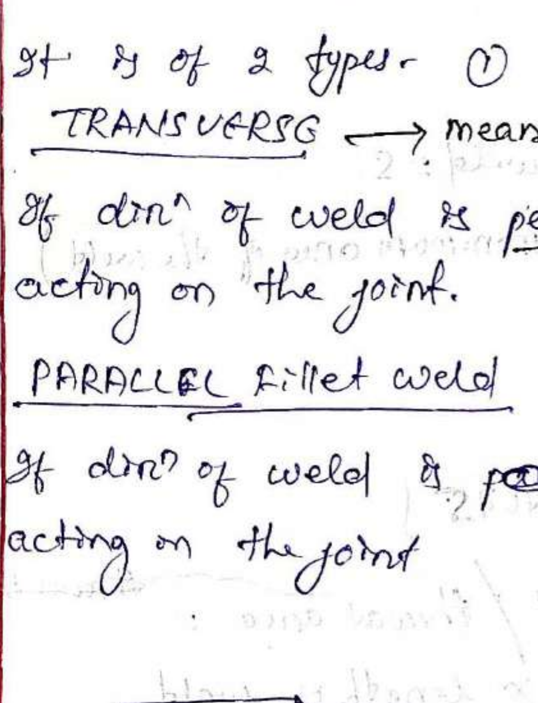
→ It is of 2 types - (1) Transverse Fillet weld

(1) TRANSVERSE → means 90°

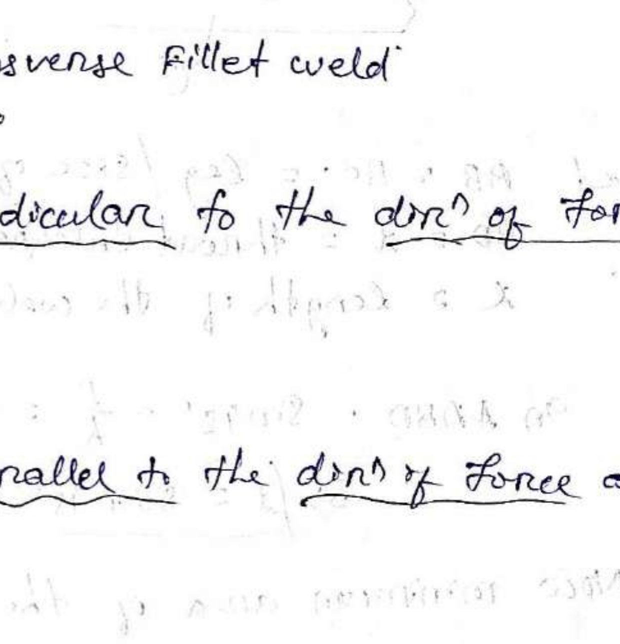
→ If dirn of weld is perpendicular to the dirn of force acting on the joint.

(2) PARALLEL Fillet weld

→ If dirn of weld is ~~pos~~ parallel to the dirn of force ~~and~~ acting on the joint



(Single transverse Fillet weld)

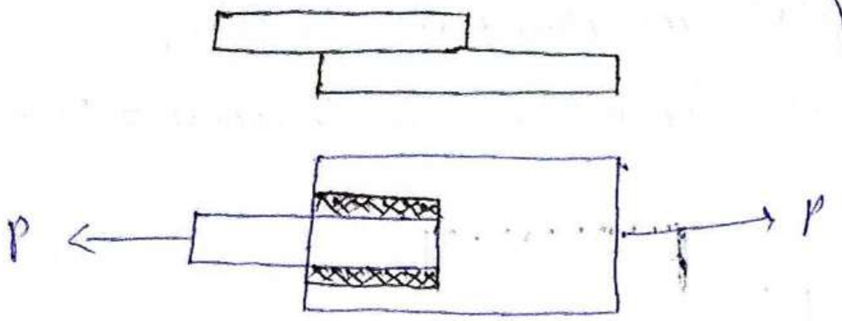


(Double transverse Fillet joint)

18/10/22

transverse - tensile

transverse weld is tensile failure
parallel weld - shear failure

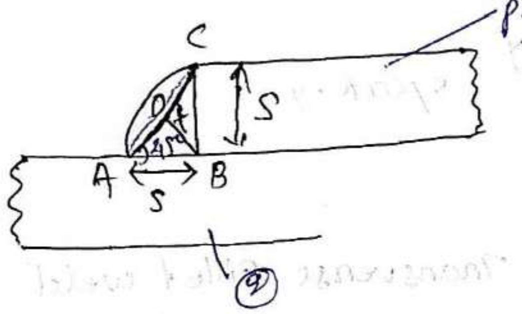


(Double parallel fillet weld)

20/10/22

Strength of transverse fillet welded joint

To find the strength of fillet joint. The section of folled is assumed that right angle triangle



throat thickness is less

Let $AB = BC = $ leg / size of the weld $= S$

$BD = t =$ throat thickness (minimum area of the weld)

$l =$ length of the weld

In $\triangle ABD$ · $\sin 45^\circ = \frac{P}{h} = \frac{t}{S}$

$\Rightarrow \boxed{t = S\sin 45^\circ = 0.707S}$

Now minimum area of the weld / throat area = ~~throat th~~

= throat thickness × length of weld

$= t \times l$

$A = 0.707S \times l$

If σ_t = allowable ~~st~~ tensile stress for weld metal

then tensile strength of the joint in single fillet weld

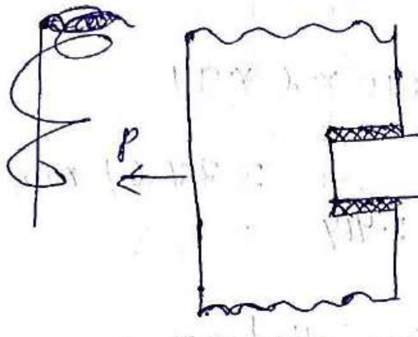
$\boxed{P = \sigma_t \times 0.707 S l}$

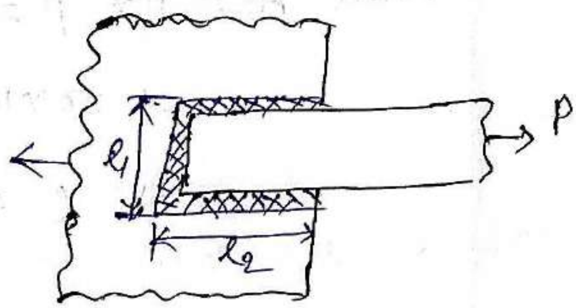
20/10/22 Tensile strength of the joint on double fillet weld = 2P

$= ~~2\times~~ = 2 \times \sigma_t \times 0.707 s l$

$$\boxed{P = 1.414\, \sigma_t\, s l}$$

Strength of parallel fillet welded joints





Double ∥ fillet weld (Combination of parallel & transverse fillet weld)

→ Shear strength of the joint for single parallel fillet weld

P = throat Area × allowable shear stress

$P = A_t \times \tau$

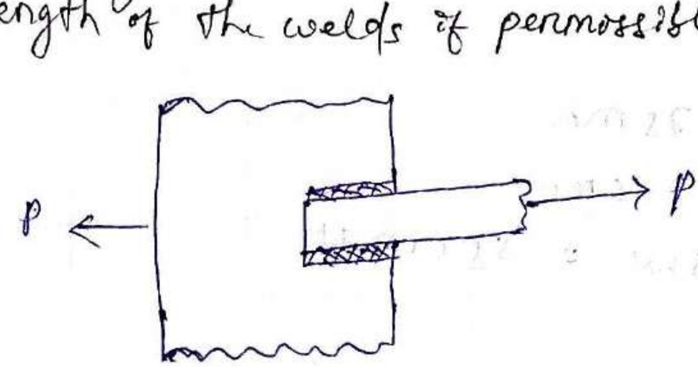
$$\boxed{P = 0.707 s l \tau}$$

→ Now for double parallel fillet weld, strength of the joint $= P = 2 \times 0.707 s l \tau$

$$\boxed{P = 1.414 s l \tau}$$

✠ To find required length of the weld, 15mm has to be added for starting & stoping of weld run

Q (Smarus quest^n clear) A steel plate, 100mm wide & 10mm thick, each welded to another steel plate by double parallel fillet welds. The plates are subjected to tensile force of 50KN. Determine required length of the welds if permissible shear stress is 94 N/mm²



20/10/22

Given $\quad \omega = 100mm \quad T \to$ plate thickness

$T = 10mm = S \quad t \to$ Throat thickness

$P = 50kN$

$l = ?$

$\tau = 94 N/mm^2$ (permissible shear stress)

Area ~~A = p x~~ $P = 1.414 S l \tau$

$\Rightarrow 50 \times 10^3 = 1.414 \times 10 \times l \times 94$

$\Rightarrow l = \dfrac{50 \times 10^3}{94 \times 10 \times 1.414} = 37.61 mm$

Now adding starting & stoping distance of 15mm, required design length of the weld =

$l = 37.61 + 15$

$l = 52.6 \approx 54 mm$

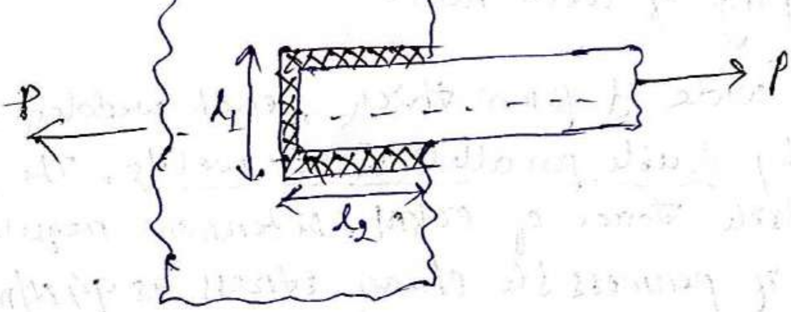
E/O Q — A plate, 75mm wide and 10mm thick, is joint with another steel plate by single transverse and double parallel fillet welds. The joint is subjected to max^m tensile force of 55kN, permissible tensile & shear stress in weld material are 70 & 50 N/mm^2 respectively. Find / required length of each parallel fillet weld.

5 marks

Imp

clear

28/10/22



Sol^n ~~$P = 55 \times 10^3 N$~~

$\omega = 75mm$

$S = t = 10mm$

$\sigma_t = 70 N/mm^2$

$P = 55kN = 55000 N$

$\tau = 50 N/mm^2$

28/10/22 → Strength of transverse fillet weld (P_1)

$P_1 = 0.707 S l \sigma_t$

$= 0.707 \times 10 \times 75 \times 70 = 37117.5 N$

Strength of double parallel fillet weld (P_2)

$P_2 = 1.414 S l \tau$

$= 1.414 \times 10 \times l \times 50 = 707 l$

Total strength of the joint should be 55 KN

$55 KN = P_1 + P_2 \Rightarrow 55000 = 37117.5 + 707 l \Rightarrow l = 25.29 mm$

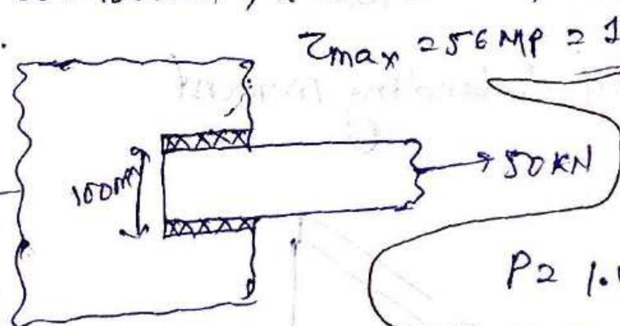
Now
Adding for starting & stopping distance, $l = 15 + 25.29 = 40.29 \simeq 45 mm$

★ Q. Imp Q. for Sem.
A plate 100mm wide & 12.5mm thick is to be welded to another plate by means of double parallel fillet welds. The plates are subjected to A load of 50KN. Find the length of the weld so that the max^m stress does not ~~exist~~ exceed 56MPa. Considered the joint 1st under static loading & then under ~~fatigue loading~~ Fatigue loading

Static loading

$w = 100mm$, $t = S = 12.5mm$, $P = 50KN$

$\tau_{max} = 56 MP = 1N/mm^2$



$P = 1.414 S l \tau$

$50/ = 1.414 \times 12.5 \times l \times 56$

$\Rightarrow l = \frac{50000}{1.414 \times 12.5 \times 56} = 50.5 mm$

Now designing l for adding starting & stopping distance

$= 50.5 + 15 = 65.5 \approx 66 mm$

(2.7) Fatigue loading

Stress concentrat^n factor for fatigue loading can be refered from data book (Page no - 11.14, Table no. 1.3 NO calculation and addit^n)

$\tau_{max} = \frac{56}{2.7} = 20.74 MPa$

$P = 1.414 S l \tau$

$\Rightarrow 50 \times 1000 = 1.414 \times 12.5 \times l \times 20.74$

$\Rightarrow l = 136.39$

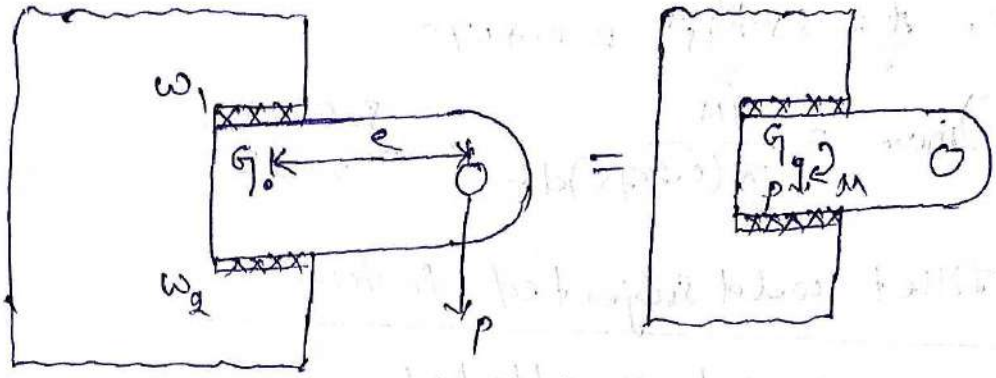
Now Adding stopping & starting distance $136.39 + 15 = 151.39$

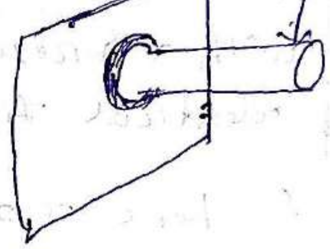
$= 152 mm$

18/10/22

Welded joints subjected Accentric loading

Whenever the line of action of the external load does not coincide the centroid of the welded joint then it is called Accentric loading.





Let G = centre of gravity of the welded joint

e = essentricity betn C.G & line of actn of load P.

According to mechanics P can be replaced by a coupled (P×e) having moment & the same load

- The force P acting through C.G causes direct shear stress on the weld called as primary shear stress. It is assume to be uniformly distributed over throat area.

primary s.s = $\tau_1 = P/A$

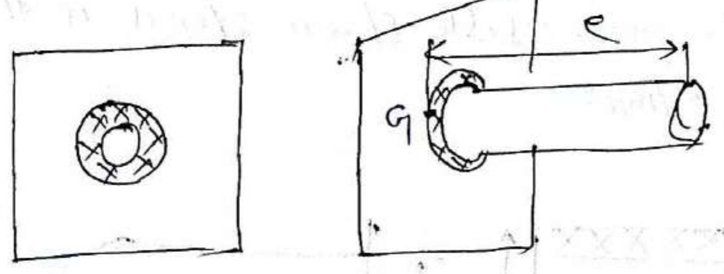
- The moment of the couple cause torsional shear stress on throat area of the weld called as secondary shear stress.

$$\left(\frac{T}{J} = \frac{\tau}{R} \Rightarrow \tau = \frac{TR}{J} \right) = \frac{M R_{max}}{J} = \frac{(P \times e) R}{J}$$

τ

31/10/22

other type of Accentric loading



→ Here the Following 2 type stress

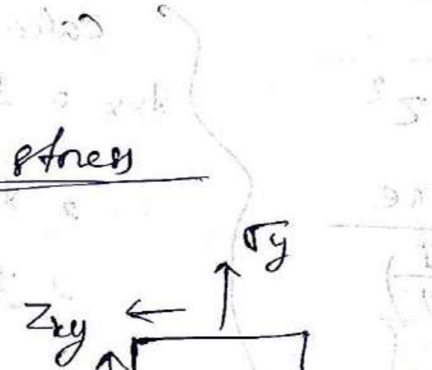
(1) Direct stress due to shear load $\tau_1 = P/A$

(2) Bending stress due to Bending load $\frac{M}{I} = \frac{\sigma_b}{y}$

$$\Rightarrow \sigma_b = \frac{My}{I} = \frac{M}{\boxed{Z}} \leftarrow \text{from databook}$$

$$= \frac{P \times e}{Z}$$

From principle stress



$$\sigma_{1,2} = \left(\frac{\sigma_x + \sigma_y}{2}\right) \pm \sqrt{\left(\frac{\sigma_y - \sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

$$= \frac{\sigma_x + \sigma_y}{2} \pm \frac{1}{2}\sqrt{(\sigma_y - \sigma_x)^2 + 4\tau_{xy}^2}$$

$$\tau_{max} = \frac{\sigma_b}{2} \pm \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2}$$

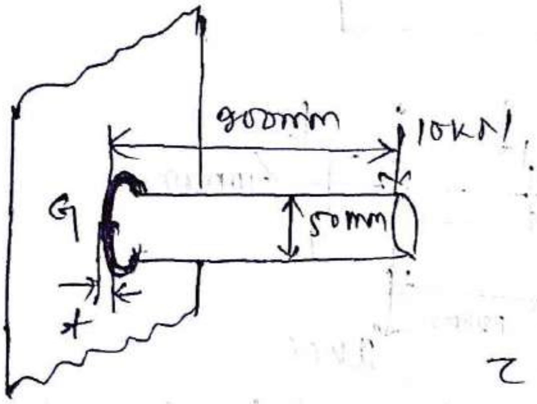
$$= \frac{\sigma_b}{2} \pm \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

$$\tau_{max} = \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

Q 31/03/22

A 50mm dia solid shaft is welded to a flat plate as shown in the fig. If the size of the weld 15mm find the max^m normal & shear stress in the weld.

πr^2



Given

$s = 15mm$

$e = 200mm$

$P = 10kN = 10{,}000N$

$$\tau = \frac{P}{A} = \frac{10\times10^3}{\pi d t}$$

$$= \frac{10\times10^3}{\pi\times50t} = \frac{63.66}{t} = 6$$

$$\boxed{t = 0.707s = 0.707\times15 = 10.605}$$

$$\sigma_b = \frac{M}{Z} = \frac{P\times e}{Z}$$

$$= \frac{10\times10^3\times200}{\frac{\pi d^2 t}{4}} = \frac{200\times10\times10^3\times4}{\pi d^2 t} = \frac{1018.6}{t}$$

$$= \frac{1018.6}{10.605} = 96.05$$

max^m normal (σ_{max})
$$= \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

$$= \frac{96.05}{2} + \sqrt{\left(\frac{96.05}{2}\right)^2 + 6^2}$$

$$= 96.41\ N/mm^2$$

$$\tau_{max} = \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

$$= 48.39\ N/mm^2$$

31/10/22

8 marks

Imp

Riveted joint

caulking & Fullering operⁿ

Caulking

↳ They are used to obtained leak proof and fluid tight joints used in pressure vessel and boilers.

→ Caulking process is applied to the edges of the plate in lap joint & the edges of strap / cover plate in butt joint.

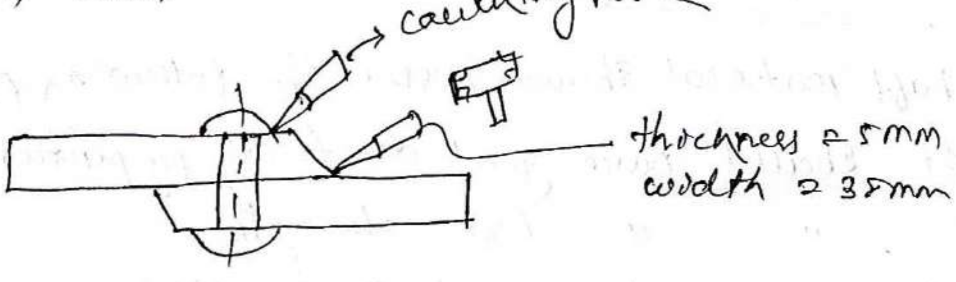
→ These edges are ~~bevelt~~ ~~vbelt to~~ bevelt to 70 - 75°

& caulking tool is hammered on the edges. This is done by hand hammer or by using ~~nmetic~~ pneumatic or hydraulic hammer

→ The head of the rivet is also hammered down with the caulking tool

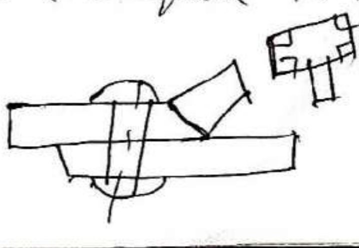
→ The blows on caulking tool closes surface respectively cracks on the contacting surface betn 2 plate & also betn the rivet and plate resulting in leak proof joints.

→ This tool cannot be used for plates having thickness less than 6mm.



Fullering

→ It is similar to caulking process except the shape of the tool. The width of the fullering tool is equal to the thickness of the plate being hammered



08/11/22 # CHAPTER - 3 DESIGN of SHAFT

2marks

SHAFT

0 ↳ Shaft is a rotating m/c element which is used to transmit power from oneplace to another

→ Generally it is cylindrical in shape. To transmit power from one to another shaft the varies members such as pullies, gears, etc are mounted on it.

AXLE

0 ↳ It does not transmit power but only transmit motion with same speed. It supports rotating element like wheel, hoisting drums or rope sheaves which is fitted to the Housing by means of bearing

→ Axle is subjected to bending moment due to transverse load like bearing reaction, & does not transmit any useful torque/

eg Rear axle, of railway wagon.

SPINDLE

→ It is a short rotating shaft

eg used in all m/c tools such as drive shaft of lathe, spindle of drilling m/c etc

Material used in shaft

The shaft material should posses the following properties

1. It should have good ductility properties
2. " " " high strength
3. " " " good machinability
4. " " " good Flowbelity
5. " " " " wear resistance

- Low or medium carbon steel
- Alloy steel

07/11/22

Manufacturing of shaft

↳ It is generally manufacture by rolling, cold rolled shafts, are stronger than hot rolled shafts For better accuracy shaft is melt by forzing & extruction which is more time taking.

Types of shaft

(1) Transmission shaft → They transmit power bet^n 2 element

ex: Counter shaft, line shaft, Factory shaft, overhead shaft etc

→ These shaft carry m/c part like pulley, gear, etc so it is Subjected to bending in addition to twisting.

MACHINE SHAFT

→ They form an integral part of the m/c itself

Stresses in shaft

(1) shear stress due to transmission of torque

(2) Bending " " " gear & pulley mounted on it

on due to self weight of the shaft

(3) stress. due to combined bending & torsion

05/11/22 Calculation of permissible stresses on shaft

Acc^n to ASME (American Society of mechanical engineer) code for transmission shaft

(1) permissible tensile stress (σ_t) is 60% of elastic limit (σ_{et}) in tension but not more than 36% of ultimate tensile stress.

$\boxed{\sigma_t = 0.6\, \sigma_{elastic}}$

or $= 0.36\, \sigma_{ultimate}$ } take smaller one

$\sigma_t = $ ______

05/11/22 ③ permissible shear stress (τ) is 30% of elastic limit of less than 18% of $\sigma_{ultimate}$

$$\left.\begin{array}{l}\boxed{\tau = 0.3\ \sigma_{elastic}} \\ \text{or}\ \boxed{\tau = 0.18\ \sigma_{ut}}\end{array}\right\}\ \text{take smaller one}$$

08/11/22 DESIGN OF SHAFT

(1) Based on strength

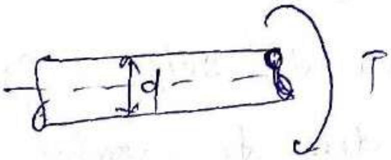
a) Shaft subjected to twisting moment only
b) Shaft " " bending moment only
c) " " " combined twisting & bending moment
Not in syllabus → d) " " " Axial load with twisting & B.M

(A) SHAFT SUBJECTED TO TWISTING MOMENT ONLY

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\theta}{L} \quad (\text{Torsion equat}^n)$$

$$I = \frac{\pi}{64} d^4$$

$$J = \frac{\pi}{32} d^4$$



r_{max} = radius of the shaft

$$\Rightarrow \frac{T}{\frac{\pi}{32} d^4} = \frac{\tau}{d/2}$$

$$\Rightarrow \frac{16 \cdot 32\,T}{\pi d^4} = \frac{2\tau}{d}$$

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

$$\Rightarrow \boxed{d = \sqrt[3]{\frac{16T}{\pi \tau}}} \quad (\text{For Solid Shaft})$$

05/11/22

Hollow Shaft

$$\frac{T}{J} = \frac{\tau}{R}$$

(d_o = outer dia)
(d_i = inner dia)

$$\Rightarrow \frac{T}{\frac{\pi}{32}(d_o^4 - d_i^4)} = \frac{\tau}{d_o/2}$$

$$\Rightarrow \boxed{d_o = d_i}$$

$T = P \times S$

Q2 V.Imp Clear 04/01/23

A solid shaft is transmitting ~~one~~ 1 MW at 240 rpm. Determine the dia of the shaft if the max torque transmitted ~~ext~~ exceeds the mean torque by 20%. Take max allowable shear stress as 60 MPa

Given

$T = F \times D$

(power) $= P = 1\ MW = 10^6 W$

(Speed) $= N = 240\ rpm$

$d = ?$ (dia of shaft)

$\boxed{\text{(Torque) } T_{max} = 1.2\ T_{mean}}$

(Allowable shear stress) $= \tau = 60\ MPa$

$P = T\omega$

$P = T_{mean} \frac{2\pi N}{60}$

$\Rightarrow 10^6 = T_{mean} \times \frac{2\pi \times 240}{60}$

$\Rightarrow T_{mean} = \frac{10^6 \times 60}{2\pi \times 240}$

$\Rightarrow T_{mean} = 39788.73\ Nm$

$= 39788.7 \times 10^3\ N\text{-}mm$

$T_{max} = 1.2 \times 39788.7 \times 10^3$

$T_{max} = 47746440\ N\text{-}mm$

$$\tau = \frac{16\ T_{max}}{\pi d^3}$$

$$\Rightarrow 60 = \frac{16 \times 47746440}{\pi \times d^3}$$

$$\Rightarrow d = \sqrt[3]{\frac{16 \times 47746440}{60\pi}}$$

$\Rightarrow d = 159.4 \simeq 160\ mm$ (standardised dia)

$$d = \sqrt[3]{\frac{16\ T_{max}}{\pi \tau}}$$

03/01/23

T_{max} = max^m torque

τ = allowable shear stress

12/11/22

Q. Find the dia of a solid shaft to transmit 90 kW at 900 rpm. The ultimate shear stress for steel is 360 MP. & Factor of safety is 8. If a hollow shaft is to be used in place of solid shaft, find the inside & outside dia if $\frac{d_i}{d_o} = 0.5$

Given

$P = 90\ kW$ | $d_i = ?$

$N = 900\ rpm$ | $d_o = ?$

$\tau_{max} = 360\ mp$ | $d = ?$

$F.S = 8$

$\frac{d_i}{d_o} = \frac{1}{2} = 0.5$

~~954.93~~ $\times 10^3$ N-mm

mm
cem
dm
10^3 mm

$\tau = \frac{\tau_{ultimate}}{FS} = \frac{360}{8} = 45\ MPa$

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

$\Rightarrow 90 \times 10^3 = \frac{2\pi \times 900 \times T}{60}$

$\Rightarrow \cancel{1200}^{600} \times 10^3 = 2\pi \times \cancel{900} \times T$

$\Rightarrow T = \frac{600{,}000}{2\pi} = 95.493\ N\text{-}mm =$

$= 954.93 \times 10^3\ N\text{-}mm$

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

~~$\Rightarrow d = \left(\frac{16T}{\pi\tau}\right)^{1/3} = \frac{\pi \times 45}{16 \times 954.93 \times 10^3} = 47.63$~~

~~$\Rightarrow d = 47.63$~~

$d^3 = \frac{16T}{\pi\tau}$ ~~$= \frac{16 \times 954.93 \times 10^3}{16T} = 2$~~

$d = \frac{16 \times 954.93 \times 10^3}{\pi \times 45} = 47.63$

$= 50mm$ (standard size)

12/11/22

$$\frac{T}{J} = \frac{\tau}{R}$$

$$\Rightarrow \frac{T}{\pi/32\,(d_0^4 - d_i^4)} = \frac{\tau}{\left(\frac{d_0}{2}\right)}$$

$$\Rightarrow \frac{T}{\frac{\pi}{32}\left[d_0^4 - (0.5d_0)^4\right]} = \frac{\tau}{\left(\frac{d_0}{2}\right)}$$

$\Rightarrow d_0 = 48.67 \approx 50$ mm

$\Rightarrow d_i = \cancel{48.67}\ 50 \times 0.5 = 25$ mm

(b) case II

Shaft subjected bending moment only

(Bending equn $\frac{M}{I} = \frac{\sigma_b}{y}$, $y = d/2$)

$$\frac{M}{\frac{\pi}{64}d^4} = \frac{\sigma_b}{d/2}$$

$$\Rightarrow \frac{32M}{\pi d^4} = \frac{\sigma_b}{d}$$

$$= \boxed{\sigma_b = \frac{32M}{\pi d^3}}$$

$$\Rightarrow \sigma_b \pi d^3 = 32M$$

$$\Rightarrow \boxed{d = \sqrt[3]{\frac{32M}{\sigma_b \pi}}} \rightarrow \text{for solid shaft}$$

For HOLLOW SHAFT

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

$$\Rightarrow \frac{M}{\frac{\pi}{64}(d_0^4 - d_i^4)} = \frac{\sigma}{\left(\frac{d_0}{2}\right)} \quad \text{(Hollow shaft)}$$

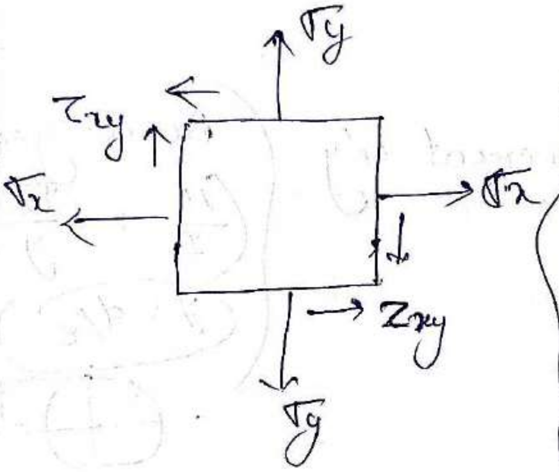
(c). Case III

12/11/22 shaft subjected to combined ~~sta~~ twisting & bending moment

For elastic failure of materials subjected to various types of combined stresses, 2 theories of failures are used

① Maxm shear stress theory (used for ductile materials)

② " Normal " " (used for Brittle ")



$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_y - \sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

$$\tau_{max} = \sqrt{\left(\frac{\sigma_y - \sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_x = 0, \quad \sigma_y = \sigma_b$$

$$\sigma_{1,2} = \frac{\sigma_b}{2} \pm \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

$$\tau_{max} = \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

$$= \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2}$$

$$= \frac{1}{2}\sqrt{\left(\frac{32M}{\pi d^3}\right)^2 + 4\left(\frac{16T}{\pi d^3}\right)^2}$$

$$\boxed{\tau_{max} = \frac{16}{\pi d^3}\underbrace{\sqrt{M^2 + T^2}}_{\text{equivalent twisting moment}}}$$

T_e is defined as that twisting moment produces the the same shear stress as the actual twisting moment

By limiting max^m shear stress = the ~~alable~~ allowable shear stress for the material

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \tau d^3$$

$d = ?$

For max^m Normal stress theory, ~~max^m~~

Max^m normal stress on the shaft

$$\sigma_{max} = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

$$= \frac{32M}{\pi d^3 \times 2} + \sqrt{\left(\frac{32M}{\pi d^3 \times 2}\right)^2 + \left(\frac{16T}{\pi d^3}\right)^2}$$

$$= \frac{32}{\pi d^3}\Big[\underbrace{\tfrac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)}_{M_e \,=\, \text{equivalent bending moment}}\Big] = \frac{32}{\pi d^3}\left[\tfrac{1}{2}(M + T_e)\right]$$

M_e is defined as that moment which when acting alone produces the same tensile or compressive stress as the actual bending moment

$$M_e = \frac{32}{\pi d^3}\left[\tfrac{1}{2}(M + T_e)\right] = \frac{\pi}{32} \sigma_b d^3$$

$\Rightarrow \boxed{d = ?}$

For HOLLOW SHAFT

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \tau (d_o)^3 (1 - K^4) \quad \text{where } K = \frac{d_i}{d_o}$$

$$M_e = \tfrac{1}{2}\left(M + \sqrt{M^2 + T^2}\right) = \frac{\pi}{32} \sigma_b (d_o)^3 (1 - K^4)$$

Q
14/11/22
Imp
9/5

A Solid circular shaft is subjected to a bending moment of 3000 NM & A torque of 10,000 N-m. The shaft is made of 45C8 steel having ultimate stress = 700 MPa. & ultimate shear stress = 500 MPa, Assume FS = 6. Determine dia of the shaft.?

Given

$B.M = 3000\,Nm$

$T = 10,000\,N\text{-}m$

$FS = 6$

$\sigma_{ultimate} = 700\,MPa$

$\tau_{ulti} = 500\,MPa$

$d = ?$

$(a^2 + b^2)$

10,000,0 + 900

$\sigma_t = \frac{700}{6} = 116.67\,MPa$

$\tau = \frac{500}{6} = 83.33\,MPa.$

10,440

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

$= \sqrt{(10,000)^2 + (3000)^2}$

$= \underline{10,440.3\,N\text{-}m}$

$\boxed{T_e = 10,440.3 \times 10^3\,N\text{-}mm}$

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

$\Rightarrow d = \left(\frac{16\,T_e}{\tau\pi}\right)^{1/3} = \sqrt[3]{\frac{16 \times 10440.3 \times 10^3}{83.33\,\pi}} = 86.10\,mm$

$M_e = \frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)$

$= \frac{1}{2}\left(\overset{1500}{\cancel{3000}} + \underline{10,440.3}\right)$

$M_e = 6720.15 \times 10^3 =$

$$\sigma_b = \frac{32M}{\pi d^3}$$

$$\Rightarrow d = \sqrt[3]{\frac{32 M_e}{\sigma_b \pi}} = \sqrt{\frac{32 \times 6720.15 \times 10^3}{116.67\pi}}$$

$$= 76.$$

$$d =$$

18/11/22 A solid circular shaft ~~is~~ subjected a bending moment of 3500 N-m & torque is 10500 Nm. The shaft is made of steel having ultimate tensile stress 450 MPa ultimate shear stress 550 MPa. Determine the dia of the shaft by using two theories of failure. Assume factor of safety = 6

<u>Given</u>

$M = 3500$ N-m

$T = 10,500$ N-m

$\sigma_{ultimate} = 450$ MPa

$\tau_{ultimate} = 550$ MPa

$FS = 6$

$$\sigma_t = \frac{450}{6} = \cancel{73.33}\ 75 \qquad \tau = \frac{550}{6} = 91.66$$

$$\cancel{\tau_{max}\ \frac{550}{6} = 88.33 = \tau}$$

$$\cancel{T_e = \sqrt{T^2 + M^2} = \sqrt{(75)^2 + (88.33)^2} = \sqrt{56.25 +}}$$

$$T_e = \sqrt{(10,500)^2 + (3500)^2}$$

$$= 11067.97 \text{ N-m}$$

$$= 11067.97 \times 10^3 \text{ N-mm}$$

$$\tau = \frac{16 T_e}{\pi d^3} \Rightarrow d = \left(\frac{16 T_e}{\tau \pi}\right)^{1/3} = \sqrt[3]{\frac{16 \times 11067.97 \times 10^3}{\cancel{88.33} \times \pi \times 91.66}}$$

$$\cancel{d = 86.09} \qquad d = 85.03$$

15/11/24

$M_e = \frac{1}{2}\left(M + \sqrt{M^2 + T_e^2}\right)$

$= \cancel{\frac{1}{2}(3500 + 86.09 \quad 11067.97)}$ 1500

$= \frac{1}{2}\left(M + 11067.97\right)$

$= 7283.98 \ N\text{-}m = 7283.98 \times 10^3 \ N\text{-}mm$

$\sigma_b = \frac{32M}{\pi d^3}$

$d = \sqrt[3]{\frac{32 M_e}{\pi \sigma_t}} = \sqrt[3]{\frac{32 \times 7283.98 \times 10^3}{91.66 \times \pi}}$

$\boxed{d = 93.19}$

So $d = 85.03 + 15 = 100.03 \simeq 101$ mm

Shaft Subjected Fluctuating load

- In actual practice shafts are subjected fluctuating torque & bending moment. So Accⁿ to ASME code, the bending & torsional moment are to be multiply by factor K_b & K_t respectively to account for shock & fatigue in operating condition where K_b = Combined shock & fatigue factor for bending

  K_t = Combined shock & fatigue factor for torsion

- Now equivalent torsional moment $T_e = \sqrt{(T \times K_t)^2 + (M \times K_b)^2}$

- Now equivalent bending moment $M_e = \frac{1}{2}\left[(M \times K_b) + \sqrt{(T \times K_t)^2 + (M \times K_b)^2}\right]$

25/11/22

The factor K_b & K_t can be found from data book

Q A mild steel shaft transmit 20kW at 200 rpm. It carries a central load of 900 N & is simply supported betn the bearing which are 2.5 m apart. Determine the size of the shaft if the allowable shear stress is 42 MPa, & the maxm tensile or compressive stress is not to exceed 56 MPa. What size of the shaft will be require if it is subjected to gradually applyed load

10%

Given

$P = 20\,kW$

$N = 200\,rpm$

$\frac{2.50 \times 1000}{2500}$ (0.25)

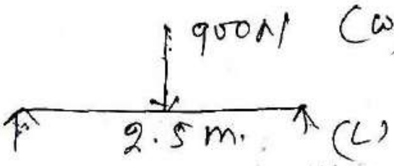
$P = T\omega$

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

$T = P \times \frac{60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2\pi \times 200}$

$T = 954.92\ Nm$

$= 954.9 \times 10^3\ N\text{-}mm.$



$M = \frac{wl}{4}$

$= \frac{900 \times 2.5}{4}$

$= 562.5 \times 10^3\ N\text{-}mm$

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

$= \sqrt{(562.5)^2 + (954.9)^2}$

$= 1108.26\ N\text{-}m$

$T_e = 1108.26 \times 10^3\ N\text{-}mm$

$\tau = \frac{16\,T_e}{\pi d^3} \Rightarrow d = \left(\frac{16\,T_e}{\tau \times \pi}\right)^{1/3} = \sqrt[3]{\frac{16 \times 1108.26 \times 10^3}{42 \times \pi}}$

$\boxed{d = 51.22\ mm}$

$$M_e = \frac{1}{2}(M + T_e) = \cancel{\frac{1}{2}(562.5\times10^3 + 1107.\times10^3)}$$

$$= 835380\ \text{N-mm}$$

$$= \frac{1}{2}(562.5\times10^3 + 1108.26\times10^3)$$

$$= 835380$$

$$\Rightarrow \sigma_b = \frac{32 M_e}{\pi d^3}$$

$$\Rightarrow d^3 = \frac{32 M_e}{\pi \sigma_t}$$

$$= \cancel{\frac{32\times 562.5\times10^3}{\pi\times 56}}$$

$$= \frac{32\times 835380}{\pi \sigma_t}$$

$$= \sqrt[3]{\frac{32\times 835380}{\pi\times 56}}$$

$$\boxed{d = 53.36}$$

$$d = 56$$

$K_b = 1.5$, $K_t = 2$ } from Page No - 5.14 S.Md (databook)

Now equivalent torsional moment

$$T_e = \sqrt{(T\times K_t)^2 + (M\times K_b)^2}$$

$$= \sqrt{(984.9\times10^3\times1)^2 + (562.5\times10^3\times1.5)^2}$$

$$= \cancel{1497.54 - 1341.13} = 1274263.737$$

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

$$\Rightarrow d = \left(\frac{16 T_e}{\pi \tau}\right)^{1/3} = \sqrt[3]{\frac{16\times 1341.13}{\pi\times 42}}$$

$$\Rightarrow d = 53.6$$

$$M_e = \frac{1}{2}\left[(M \times K_b) + \sqrt{(T \times K_t)^2 + (M \times K_b)^2}\right]$$

$=$

$d = 48.76$

$d = 53.6 \approx 56 \text{ mm}$

22/11/22

DESIGN OF KEY

→ The key may failed either in shear or in crushing

→ Let T = Torque transmitted by the shaft

F = Tangential force acting at the circumference of the shaft

d = Dia of the shaft

l = length of the key

w = width " " "

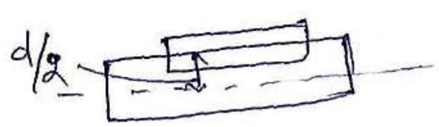
t = thickness " " "

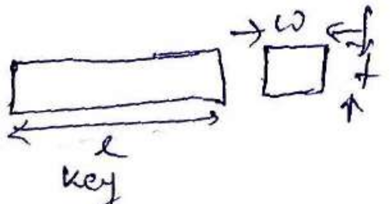
σ_c = permissible crushing stress for key

τ = " shear " " key

→ Due to power transmission the key may brake from middle as half portion inside the shaft & other half portion inside the hub so shaft presses a force against the key & from hub side in opposite direction another force will press it

SHEAR FAILURE





→ Shear Area = $l \times w$

→ Shear force $(F) = \tau \times (l \times w)$

→ $\boxed{\text{Torque } (T) = (\tau l w)\frac{d}{2}}$

★ To find length of key, equate shear strength of key & torque torsional shear stress of shaft

CRUSHING FAILURE

→ crushing Area $= l \times \frac{t}{2}$

→ " force $(F) = \sigma_c \times l \times \frac{t}{2}$

→ $\boxed{\text{Torque } (T) = \sigma_c \, l \cdot \frac{t}{2} \times \frac{d}{2}}$

Design of shafts on the basis of rigidity :-

→ A transmission shafts is said to be rigid on the basis of torsional rigidity. if it does not twist too much under the action of an external torque

→ Similarly a transmission shaft is said to be rigid on the basis of lateral rigidity. if it does not deflect too much under the action of external forces & bending moment.

→ In application like machine spindles, it is ~~neccess~~ neccessary to design the shaft on torsional rigidity basis i.e permissible angle of twist per meter length of shaft.

we know $\frac{\tau}{r} = \frac{T}{J} = \frac{G\theta}{L}$

$$\Rightarrow \theta = \frac{TL}{GJ}$$

Where

θ = angle of twist in radian

T = Twisting moment or torque on the shaft

J = Polar moment of inertia of the cross-sectional area about the axis of rotations.

$= \frac{\pi}{32} d^4$ (for solid shaft)

$= \frac{\pi}{32} (d_o^4 - d_i^4)$ (for hollow shaft)

G = Modulus of rigidity of shaft material

L = Length of the shaft

~~Torq~~

Torsional rigidity

→ The permissible amount of twist should not exceed 0.25° per meter length of the length for line or transmission shafts, deflections 2.5 to 3° per meter length is called as limiting value.

Generally 1° deflection is allowed in a length equal to 20 times the dia of the shaft.

Lateral rigidity

It is important for maintaining proper bearing characters & for correct gear teeth allignment, otherwise it will create unbalance forces.

If shaft is uniform, then lateral deflection of shaft can be obtained by using deflection formula as in SOM but when the shaft is of variable cross-section then it is obtained from elastic curve of beam.

Q. Compare the weight, strength, & stiffness of a hollow shaft of the same external dia as that of solid shaft. The inside dia of hollow shaft = ½ outside dia. Both the shafts have the same material & length.

Sol^n $d_o = d$, $d_i = \frac{d_o}{2}$ $\quad k = \frac{d_i}{d_o} = 0.5$

Comparission of weight

weight of hollow shaft = W_H = cross-sectional area × length × dia

$$= \frac{\pi}{4}(d_o^2 - d_i^2) \times L \times \rho$$

" " Solid " $= W_S = \frac{\pi}{4} d^2 \times L \times \rho$

$$\frac{W_H}{W_S} = \frac{d_o^2 - d_i^2}{d^2} = \frac{d_o^2 - \left(\frac{d_o}{2}\right)^2}{d_o} = 0.75$$

Comparission of strength

Torque for hollow shaft $= T_H = \frac{\pi}{16} \tau \times d_o^3 (1-k^4)$

" " solid " $= T_S = \frac{\pi}{16} \tau d^3$

$$\frac{T_H}{T_S} = \frac{d_o^3(1-k^4)}{d^3} = 0.9375$$

Comparission of stiffness

$$\frac{T}{\theta} = \frac{GJ}{L}$$

Stiffness of hollow shaft $= S_H = \frac{G}{L}\frac{\pi}{32}(d_o^4 - d_i^4)$

" " Solid " $= S_s = \frac{G}{L}\frac{\pi}{32}d^4$

$$\frac{S_H}{S_s} = \frac{d_o^4 - d_i^4}{d^4} = 0.9375$$

Standard size of shaft as per IS:-

→ Standard size's of transmission shafts are

25mm to 60mm with 5mm steps

60mm to ~~110~~ 110mm " 10mm "

110mm to 140mm " 15mm "

140mm to 500mm " 20mm "

→ Standard length of shafts are 5m, 6m, 7m

→ " dia " " " 6, 8, 10, 12, 14, 16, 18, 20, 22, 25, 28, 32, 36, 40, 45, 50, 56, 63, 71, 80, 90, 100

Design of shaft

→ Key ~~~~ is a machine element which is used to connect the transmission shaft to hub of rotating element like pulleys gears spockers on Flywheels.

Function of Key :-

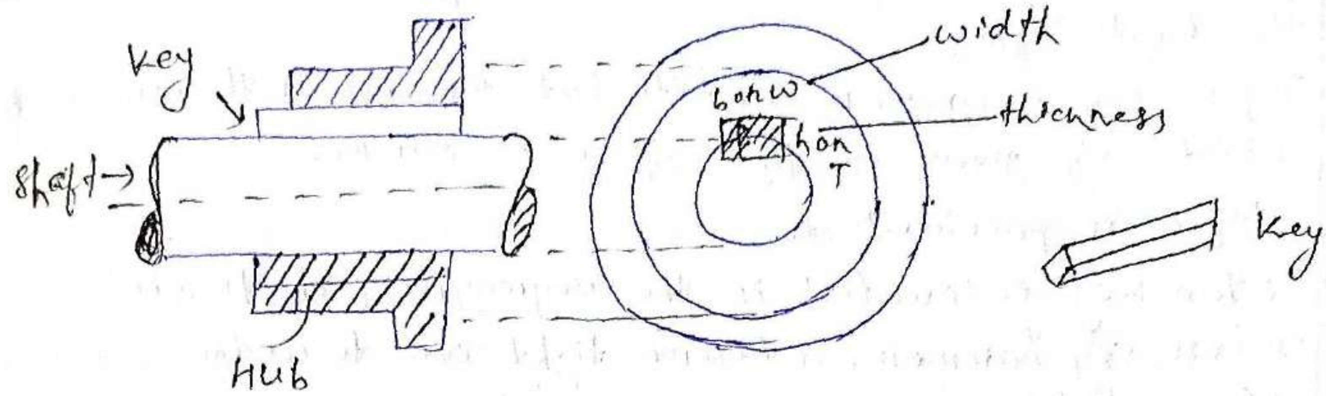
→ Used to transmit the torque from the shaft to hub of the mating element or vice versa.

→ Used to prevent relative rotational motion between the shaft & the joint m/c element like gear on pulley.

It also prevents axial motion between two elements except in case of feather key on splined connection.

- It is a piece of mild steel inserted between the shaft & hub on boss of pulley to connect these together in order to prevent relative motion between them.
- It is always inserted parallel to the axis of the shaft.

- They are used as temporary fastening & they are subjected to crushing & shearing stresses
- A key-way is a slot or recess in a shaft & hub of pulley to accomodate a key. It is ~~used~~ cut by milling cutter, It results in stress concentration in the shaft & the part become weak.



Types of key :-

1) Sunk key
2) Saddle "
3) Tangent "
4) Round key
5) Splines

1) Sunk key → This is standard form of key & here power is transmitted due to shear resistance of key. Suitable for duty application, +ve drive as no slip.

→ They are provided half on the keyway of the shaft & half on the keyway of the hub or boss of the pulley it is of following types

(a) Rectangular sunk key, thumb rule without stress analysis.

Here width (w) = $\frac{d}{4}$

& thickness (t) = $\frac{d}{6}$ (where d = dia of the shaft)

Key has taper 1 in 100 on the top side only

(b) Square sunk key :-

Here $w = t = \frac{d}{4}$

* Sunk keys with rectangular cross section are called flat key. Square keys are used in general industrial machinary & Flat keys are used in m/c tool applications where additional stability of connection is required.

(c) parallel & taper sunk key :-

→ parallel key is uniform in width & height / thickness throughout the length of key

→ Taper key is uniform in width but tapered in thickness Bottom is Flat only given on top surface in 1 in 100.

taper is provided to

(i) when key is inserted in the keyways of shafts & hub and process by hammer. It become tight due to wedge action - to ensure tightness of joint & prevents loosening of parts.

(ii) Due to taper, easy to remove key & dismantle the joints

Gib head is the projection helps in ~~wid~~ withdrawl of key

(d) Feather key :-

It is a parallel key which is fixed either to the shaft or to the hub. & it permits relative axial movement betn them. It is a sunk key with uniform width & thickness.

There is a clearance fit betn the key & key-way in the hub. So hub is free to slide over the key. But donot have, relative to rotational movement betn shaft & hub. So it transmit torque as well as axial movement ~~to~~ of hub.

Ex In clutch, gear shifting devices (alternative of ~~stop~~ splined connection)

Woodruff Key →

It is a sunk key with almost semi-circular disk of uniform thickness. The keyway on the shaft. is of semi-circular recess with ~~the~~ same curvature as that of key. The bottom of the key fits onto the circular keyway on the shaft. The projecting part of woodruff key fits on the keyway in the hub once placed in position. It tilts & align ~~it~~ self on shaft

~~Adv~~

Advantage

→ can be used on tapered shafts as it can align by slight rotation
→ Extra depth of key prevents slipping of it from shaft

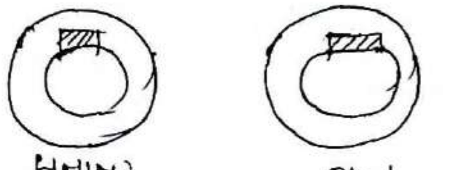
Dis Advantage

→ Extra depth of key in shaft ↑ stress conc^n & reduces stres
→ Key does not permit axial movement betn shaft & hub.

use - On tapered shafts of m/c tool & automobiles.

② Saddle Key :-

→ Saddle key is a key which fits in the keyway of the hub only so no keyway on shaft. It is of two types.
i) Flat ~~(has~~ Flat surface at bottom)
ii) Hollow (has concave surface at the bottom to match circular surface of shaft)



Hollow Flat

Flat is better than hollow in power transmission capacity

→ Friction betn shaft key & hub prevents relative motion betn shaft Suitable for light duty appl^n used for light duty Appl^n
→ cost is less

③ Tangent key :-

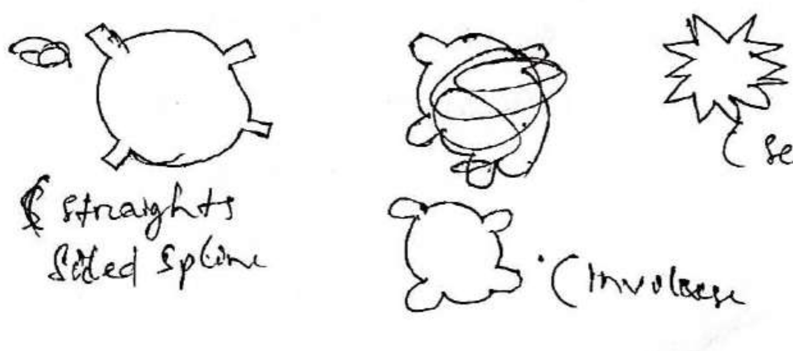
They are fitted in pairs at right angles Each key has to withstand torsion in one dir^n only.
- used for large heavy duty shafts.

④ Round key →

keys are circular in C/A & fit into holes drilled partly in shaft & partly in hub.
- used for low power drives.

⑤ Splines

They are made integral with shaft, used when there is a relative axial movement betn shaft & hub, Splines are cut on shaft ~~key~~ by milling & on hub ~~key~~ by broaching



Straight Sided Spline

(Serration).

(Involute

CHAPTER-4 DESIGN OF COUPLING

4.1 Design of Shaft coupling ?

Shaft are usually available upto 7m length due to inconvenience in transportation. So to get greater length, it is required to join two or more shafts by means of coupling.

i) To connect shafts
ii) To provide for misalignment of the shafts or to introduce mechanical flexibility
iii) To reduce the transmission of shock loads from one shaft to another
iv) To introduce protection against overloads.

4.2 Requirements of a good shaft coupling :

i) It should be easy to connect or disconnect
ii) It should transmit the full power from one shaft to the other shaft without losses.
iii) It should hold the shaft in perfect alignment
iv) It should reduce the transmission of shock loads from one shaft to another shaft
v) It should have no projecting parts.

4.3 Types of shaft couplings :-

Mainly it is of 2 types

a) Rigid coupling → used to connect two shafts which are perfectly alligned. It is of 3 types

i) Sleeve or muff coupling
ii) Clamp or split muff or compression coupling
iii) Flange coupling

(b) Flexible coupling → used to connect two shafts have both lateral & angular misalignment. It is of 3 type

i) Bushed pin type coupling
ii) Universal coupling
iii) Oldhan

4.4 Design of sleeve or Muff coupling →

→ It is the simplest type of rigid coupling, made of cast iron

→ It consists of a hollow cylinder whose inner dia is same as the shaft dia.

→ It is fitted over the ends of the two shafts by means of a gib head key.

→ Power is transmitted from one shaft to other by means of a key & sleeve. So all the elements must be strong to transmit the torque.

→ Usual proportions of cast iron sleeve coupling are

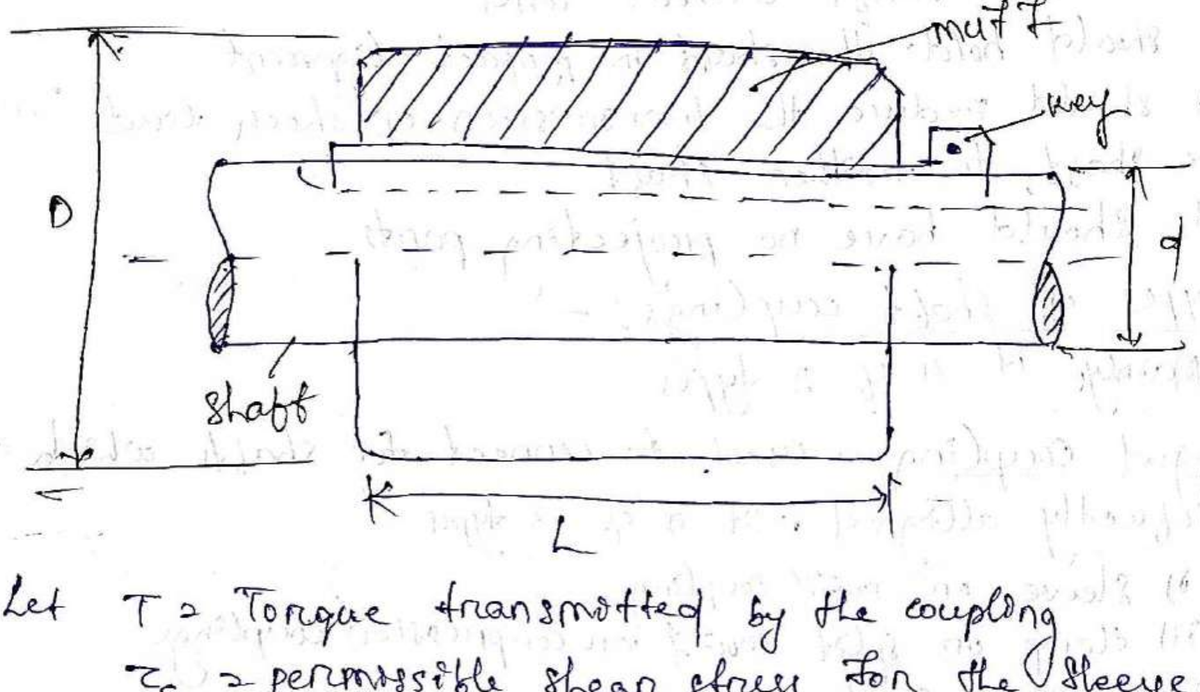
outer dia of sleeve — $D = 2d + 13mm$

length of " " — $L = 3.5d$

d = dia of shaft

Design for sleeve

Sleeve is designed by considering it as a hollow shaft



Let T = Torque transmitted by the coupling

τ_c = permissible shear stress for the sleeve material (cast iron)

Torque transmitted by a hollow shaft

$$T = \frac{\pi}{16}\tau_c\left(\frac{D^4 - d^4}{D}\right) = \frac{\pi}{16}\tau D^3(1-K^4) \quad \left(\because K = \frac{d}{D}\right)$$

From this expression, induced shear stress in the sleeve can checked

9) Design for key

Length of coupling key is equal to the length of sleeve.

i.e $l = L = 3.5d$

The coupling key is usually made into two parts so that the length of the key in each shaft

$$l = \frac{L}{2} = \frac{3.5d}{2}$$

After fixing length of key in each shaft, induced shearing & crushing stresses can be checked.

we know $T = l w \tau \frac{d}{2}$ (considering shearing of the key)

$T = l \frac{t}{2} \sigma_c \frac{d}{2}$ (" crushing of the key)

Q Design a muff coupling which is used to connect two steel shafts transmitting 40KN at 350 rpm. The material for the shafts & key is plain carbon steel for which allowable shear & crushing stresses is 40MPa & 80MPa respectively. The material for the muff is cast iron for which the allowable shear stress is 15MPa.

Soln Given $P = 40KN = 40 \times 10^3 N$

$N = 350$ rpm

$\tau = 40$ MPa

$\sigma_c = 80$ MPa

$\tau_c = 15$ MPa.

1) Design for shaft

$d \rightarrow$ dia of shaft

Torque transmitted $= T = \frac{P \times 60}{2\pi N} = \frac{40 \times 10^3 \times 60}{2\pi \times 350} = 1100 N\text{-}m$

$= 1100 \times 10^3 N\text{-}mm$

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

$\Rightarrow 1100 \times 10^3 = \frac{\pi}{16} \times 40 d^3$

$\Rightarrow d = \sqrt[3]{140 \times 10^3}$

$d = 52$ (standardise) Ans

$d = 55$ mm

Design for sleeve

Outer dia of muff $= D = 2d + 13$ mm

$= (2 \times 55) + 13 = 123 \approx 125$ mm (Ans)

length of muff $= L = 3.5d = 3.5 \times 55 = 192.5 \approx 195$ mm (Ans)

checking

Let $\tau_c \rightarrow$ Induced shear stress in muff made of cast iron muff is a hollow shaft . so

$$\therefore 1100 \times 10^3 = \frac{\pi}{16} \tau_c \left(\frac{D^4 - d^4}{D} \right)$$

$$= \frac{\pi}{16} \tau_c \left[\frac{125^4 - 55^4}{125} \right]$$

$$= 370 \times 10^3 \tau_c$$

$$\Rightarrow \tau_c = \frac{1100 \times 10^3}{370 \times 10^3} = 2.97 \text{ MPa}$$

As $\tau_c < \tau$ (permissible shear stress $= 15$ MPa given)

So design of muff is safe

Design for key

From data book for shaft of 55mm dia

~~width~~ width of key $= w = 18$mm (Ans)

As crushing stress for key material is twice the shearing stress . so a ~~ke~~ ~~key~~ square key can be used.

$\therefore$ Thickness of key, $t = w = 18$ ~~..~~ mm (Ans)

Now, length of key in each shaft, $l = \frac{L}{2} = \frac{195}{2} = 97.5$mm

checking

τ_c & σ_c on key can be checked

For shearing of key $1100 \times 10^3 = l w \tau \frac{d}{2}$

$$\Rightarrow 1100 \times 10^3 = 97.5 \times 18 \times \tau \times \frac{55}{2}$$

$$\Rightarrow \tau = 22.8 \text{ MPa}$$

For crushing of key, $1100 \times 10^3 = l \frac{d_1}{2} \sigma_c \frac{d}{2}$

$\Rightarrow 1100 \times 10^3 = 97.05 \times \frac{18}{2} \times \sigma_c \times \frac{55}{2}$

$\Rightarrow \sigma_c = 45.6$ MPa

As induced shear & crushing ~~stresses~~ are less than the permissible stresses, so design of key is safe

4.5 (II) Design of clamp or compression coupling →

→ It is also called as split muff coupling

→ Here sleeve or muff is made into two parts or halves and are bolted together.

→ Both halves of the muff are made of cast iron.

→ The shaft ends are made to abutt each other & a single key is fitted directly in the keyways of both the shafts.

→ One half of the muff is fixed from below & the other half is placed from above Both halves are held together by means of mild steel studs or ~~set~~ bolts & nuts.

Numbers of bolts are 2, 4 or 6.

The nuts are ~~rece~~ recessed into the bodies of the muff castings.

→ This coupling is used for heavy duty & moderate speeds.

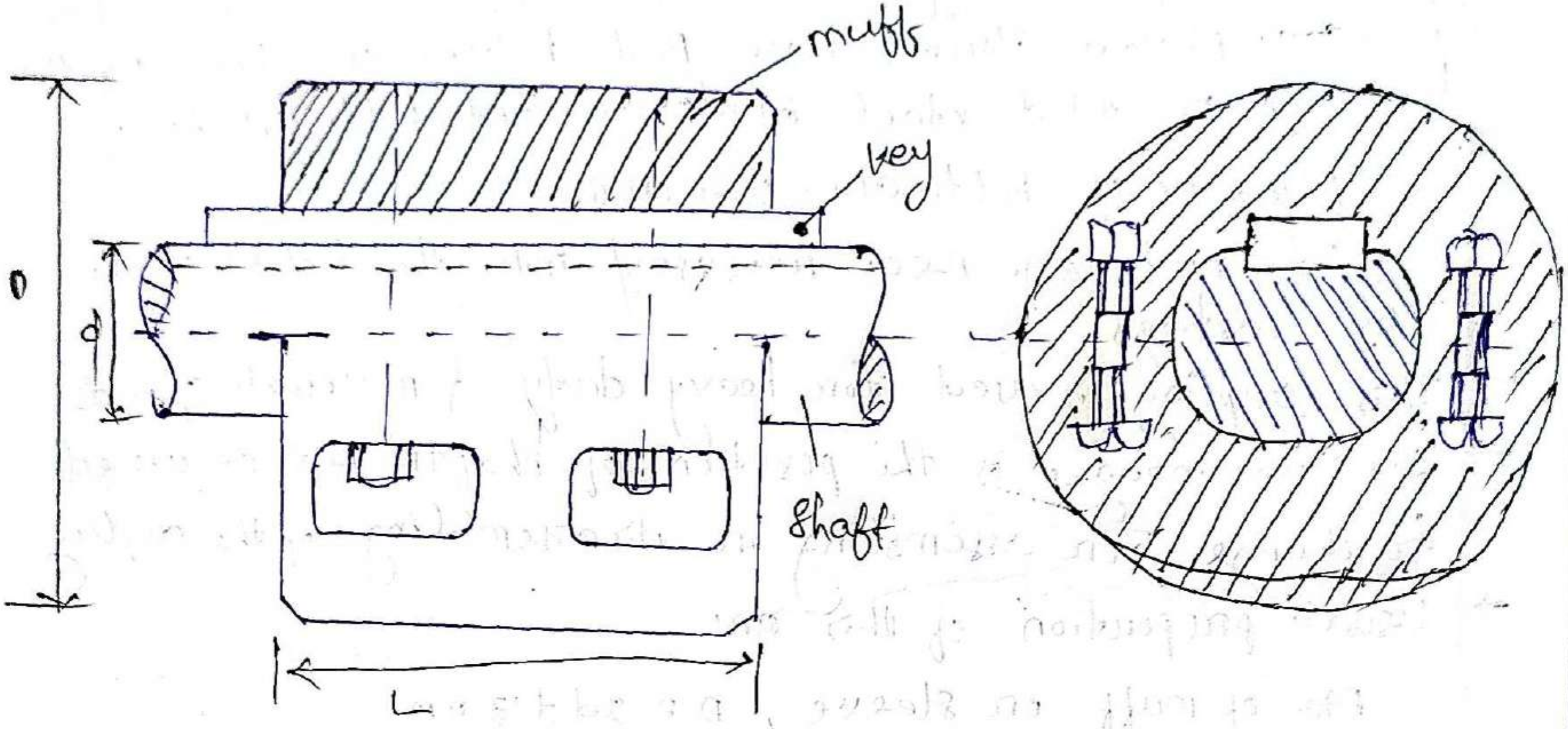
→ Its advantages is the position of shafts not required to change for assembling or disassembling of the coupling.

→ Usual proportion of this are

Dia of muff or sleeve, $D = 2d + 13$ mm

length " " , $L = 3.5d$

where d = dia of shaft



<compression coupling>

Here power is transmitted from one shaft to another by :- of key & the friction between the muff & shaft.

Design of muff & key

It is same as muff coupling

Design of clamping bolts

Let T = Torque transmitted by the shaft

d = dia of shaft

d_b = root or effective dia of bolt

n = no. of bolts

σ_t = permissible tensile stress for bolts material

μ = Coeff. of friction betn the muff & shaft

L = Length of muff

Force exerted by each bolt $= \frac{\pi}{4}(d_b)^2 \sigma_t$

Force exerted by bolts on each side of the shaft

$$= \frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2}$$

Let p = pressure on the shaft & muff surface due to the force

From uniform pressure theory

$$p = \frac{\text{Force}}{\text{Projected area}} = \frac{\frac{\pi}{4} d_b^2 \sigma_t \frac{n}{2}}{\frac{1}{2} L d}$$

∴ Frictional force between each shaft & muff

$$F = \mu \times \text{pressure} \times \text{area}$$

$$= \mu p \times \left(\frac{1}{2} \pi d L\right)$$

$$= \mu \frac{\frac{\pi}{4} d_b^2 \sigma_t \frac{n}{2}}{\frac{1}{2} L d} \times \frac{1}{2} \pi d L$$

$$= \mu \frac{\pi}{4} d_b^2 \sigma_t \frac{n}{2} \pi$$

$$= \mu \frac{\pi^2}{8} d_b^2 \sigma_t n$$

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

$$\text{or } T = \mu \frac{\pi^2}{8} d_b^2 \sigma_t n \frac{d}{2}$$

From this d_b can be found out take ($\mu = 0.3$).

Q. Design a ~~ab~~ clamp coupling to transmit 30 kN at ~~100 rpm~~ 100 rpm. The allowable shear stress for shaft & key is 40 MPa & no. of bolts connecting the two halves are 6. The permissible tensile stress for the bolts is 70 MPa. μ betn muff & ~~muff~~ shaft surface is 0.3

Soln Given $P = 30\,kN = 30\times10^3\,N$

$N = 100\,rpm$

$\tau = 40\,MPa$

$n = 6$

$\sigma_t = 70\,MPa$

$\mu = 0.3$

1) Design for shaft

$$T = \frac{P\times60}{2\pi N} = \frac{30\times10^3\times60}{2\pi\times100} = 2865\times10^3\,N\text{-}mm$$

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

$\Rightarrow 2865\times10^3 = \frac{\pi}{16}\times40\times d^3$

$\Rightarrow d^3 = 365\times10^3$

$\Rightarrow d = 71.4 \approx 75\,mm$ (standardize) (Ans)

2) Design for muff

Dia of muff $= D = 2d + 13\,mm = (2\times75)+13 = 163 \approx 165\,mm$

Total length of the muff $L = 3.5d$

$= 3.5\times75 = 262.5\,mm$ Ans.

3) Design for key

From data book for shaft dia $d = 75\,mm$

width of key $= w = 22\,mm$

Thickness " $= t = 14\,mm$

& length " = Total length of muff $= 262.5\,mm$

4) Design for bolts

Let $d_b \rightarrow$ root or core dia of bolt

we know $T = \frac{\pi^2}{16}\mu d_b^2\sigma_t n d$

$\Rightarrow 2865\times10^3 = \frac{\pi^2}{16}\times0.3\times d_b^2\times70\times6\times75$

$\Rightarrow d_b^2 = 492$

$\Rightarrow d_b = 22.2$ mm

From data book, standard core dia of the bolt for ~~coar~~ coarse series is 23.22mm & nominal dia of bolt is 27mm (M27) Ans

END

All formula for Design of coupling

Muff coupling

(6)

(1) Outer dia of sleeve $D = 2d + 13$mm

length of the sleeve $L = 3.5d$

$d \rightarrow$ shaft dia

(2) T = Torque transmitted by the coupling

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

(4) τ_c = permissible shear stress for (cast iron)

For hollow shaft

$$T = \frac{\pi}{16} \tau \left(\frac{D^4 - d^4}{D} \right)$$

Design for clamp coupling

(1) Design of shaft $\Rightarrow$

(2) " " muff

(3) " for key $\rightarrow$

d = shaft dia

w = width of key

t = thickness of key

l = Total length = muff length

(4) d_b = root or core dia of bolt

$$T = \frac{\pi^2}{16} \mu d_b^2 \times \sigma_t n d$$

DESIGN OF CLOSED COIL HELICAL SPRING (CHAPTER-5)

Spring

It is defined as an element which change its shape or deflects when loaded and regains its original shape after removal of the load. Different applications of Spring are

i) To absorb or control energy under shock load or vibration

ii) To apply Force as required like brakes, clutches, Spring loaded valves etc.

iii) To control motion by having contact betn two elements like cam & Follower.

iv) To measure Forces as required like Spring balance, engine indicators etc

v) To store energy as per need like watch, toy etc

Types of Spring :-

1. Helical Spring
2. Conical or volute spring
3. Torsion "
4. Leaf "
5. Disc "
6. Special purpose "

Helical Spring

→ They are made up of wire coiled in the Form of helix It is used to carry tensile or compressive loads. The cross-section of the wire is circular, Square, or Rectangular.

→ It is available in two Forms.

(a) compression helical Spring (b) tensile helical spring

→ It is again divided into 2 type

(i) ~~Closed~~ closely coiled helical Spring

(ii) open " " "

→ Springs are called as closed coil when the helix angle of the spring is < 10°. Here the spring wire is so close that the plane containing each turn is nearly at 90° to the axis of the helix & the wire is subjected to torsion. So, shear stress is produced here due to twisting. The load applied is parallel ~~to~~ or along the axis of the spring.

→ Spring are called as open coiled when helix angle is > 10°. Here the gap betn two consecutive turn is large.

Advantages of helical spring

→ These are easy to manufacture
→ " , available in wide variety
→ " " more reliable
→ " " constant spring rate
→ " " accurate in performance
→ By changing dimensions, certain characteristic can be changed.

Material used for helical spring :-

The selection of spring ~~a~~ material depends upon
i) Load acting on the spring
ii) The range of stress through which the spring operates
iii). Limitations on mass & volume of spring
iv) environmental cond, in which the spring will operate i.e temp & corrosive atm.

→ It should have high fatigue, strength, high ductility, high resilience & creep resistance.

→ It depends on its service
(a) Severe service → rapid continuous loading e.g automotive valve spring
(b) Avg. " → governor spring, automotive suspension spring
(c) Light " → where spring subjected to static or very infrequently varied e.g. safety valves.

→ Spring are generally made from oil-tempered carbon steel wire containing 0.60 to 0.70% carbon & 0.6 to 1.0 % Mn
→ Non-Ferrous materials like phosphor bronze, beryllium Cu, monel metal, brass etc may be used in special case to ↑ fatigue resistance, temp resistance & corrosion resistance
→ Helical Spring are either cold formed or hot formed depends on the size of wire, small → cold forms, large size → hot form
→ Smaller size wire have greater strength & less ductility

Standard Size of Spring wire (SWG)

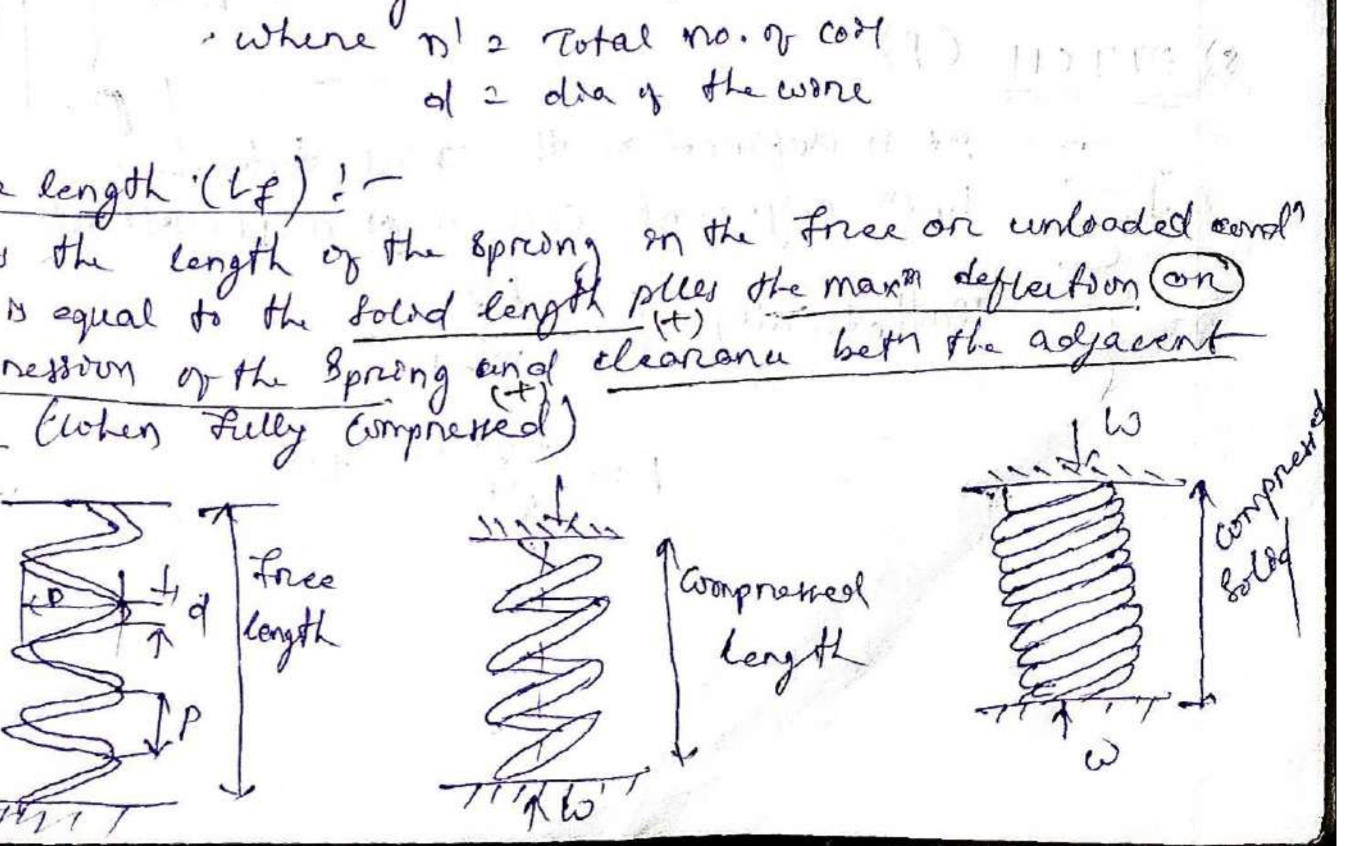
The Spring wires are standardised to have easy availability in market for easy exchange and for easy compression etc. It is represented by SWG (Standard wire Gauge) number & corresponding diameter of Spring wire

V. IMP. Terms used in compression Spring :-

1) Solid length (L_s) :-
When the compression spring is compressed until the coils comes in contact with each other, then the length of spring is called solid length. It is the product of total no. of coils & the dia of the wire.

mathematically $L_s = n'd$
where n' = Total no. of coil
d = dia of the wire

2) Free length (L_f) :-
It is the length of the spring in the free or unloaded cond^n. It is equal to the solid length plus (+) the max^m deflection or compression of the spring and (+) clearance betn the adjacent coils (when fully compressed)



5/12/22 Mathematically L_f = Solid length + max^n compression + ~~cle~~ clearance betn adjecent coils

$= n'd + \delta_{max} + 0.15\,\delta_{max}$ ~~L_f~~

(For compression spring)

or $L_f = n'd + \delta_{max} + (n'-1)\,1\text{ mm}$ (" tension spring)

2 mark

3) SPRING INDEX (c)

It is defined as the ratio of the mean dia of the coil to the dia of the wire

2 mark

Mathematically $c = \frac{D}{d}$ (always greater than 1)

where $D \rightarrow$ mean dia of coil $= \frac{D_i + D_o}{2}$

$d \rightarrow$ dia of wire

6/12/22

4) SPRING RATE / STIFFNESS (K) / Spring constant

It is defined as the load required to produce unit deflection.

Mathematically $K = \frac{W}{\delta}$

5) PITCH (P)

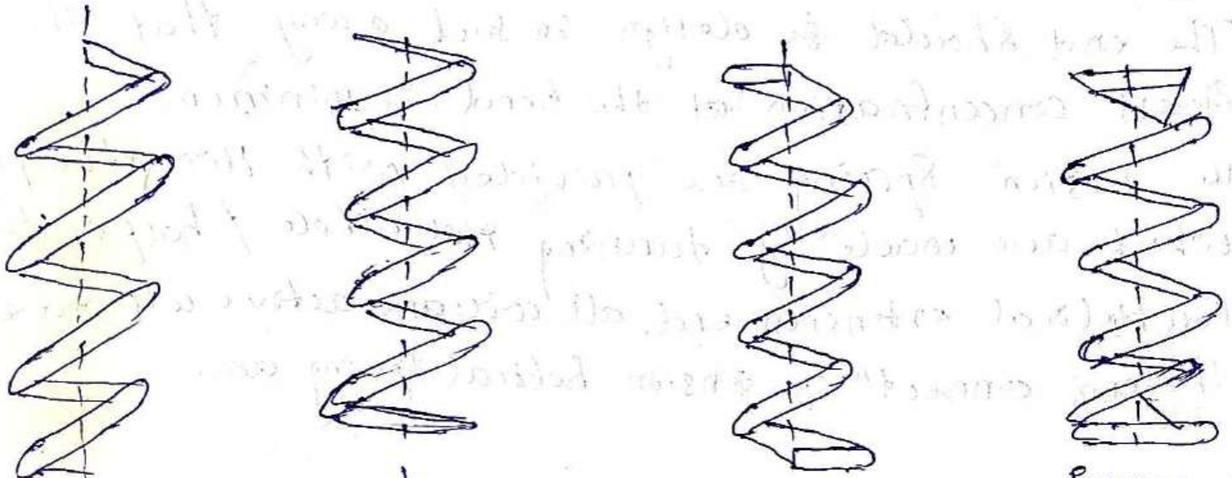
It is defined as the axial distance betn adjecent coil in ~~un~~ compressed state

Mathetically $P = \frac{L_f}{n'-1}$

$P = \frac{L_f - L_s}{n'} + d$

mark(6) END CONNECTION FOR COMPRESSION SPRING

- There are 4 commond method which are used in forming the ends of the helical compression spring
- These are (a) plain ends
  (b) plain ~~ends~~ & Ground ends
  (c) Squared ends
  (d) Squared & ground ends.



plain end — plain & grounded end — Square end — Square & grounded end

- The terms at the 2 end to not affect the deflectn calculated by the load deflection equn. So while calculating the no. of active terms, the end terms should be subtracted from the total no. of terms.

ACTIVE TERMS

Terms in the spring which contribute to spring action, support the external load and deflect under the action load

INACTIVE TERMS / COIL

A portion of the end coil which is in contact with the seat, does not contribute to spring action are called inactive terms.

6/12/22

Types of end	No. of active turns
1 - plain end	$\longrightarrow N_t \rightarrow$ total
2 - plain & grounded	$\longrightarrow (N_t - \frac{1}{2})$
3 - Square end	$\longrightarrow N_t - 2$
4 - Square & grounded	$\longrightarrow N_t - 2$

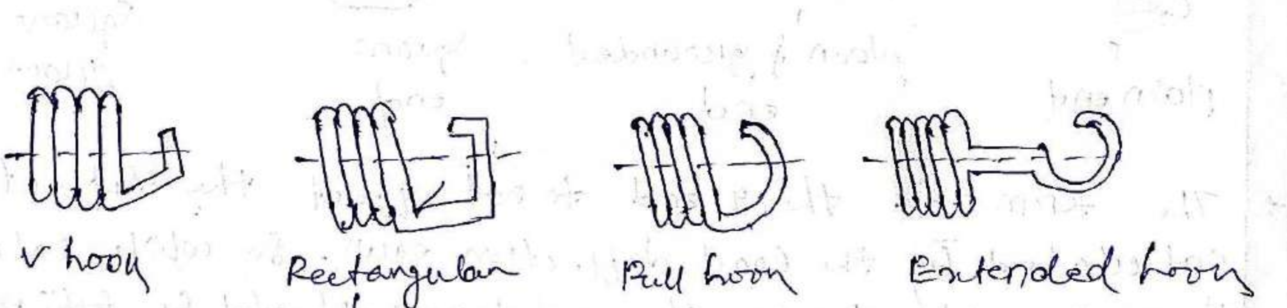
Not imp

END CONNECTION for TENSION HELICAL SPRING

→ The end should be design in such a way that the stress concentration at the bend is minimum.

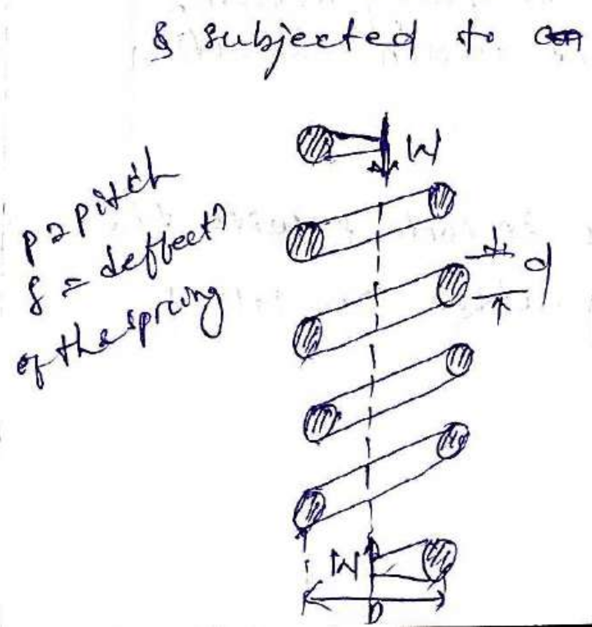
→ The tension spring are provided with Hooks/Loops which are made by turning whole / half of the coil.

→ For Helical extension end, all coils are active coil and the end connect of tension helical spring are



STRESSES IN HELICAL SPRING OF CIRCULAR WIRE.

→ Let us consider a compression helical spring of circular wire & subjected to an axial load of W.



C = Spring index = $\frac{D}{d}$

Let D = mean dia of spring coil

d = dia of spring wire

n = No. of active coil

G = Modulus of rigidity of spring material

W = Axial load on the spring

τ = Max^m shear stress induced in the wire

Due to the applied load W, it tends to rotate the ends of the wire because of the twisting moment set-up in the wire so, torsional shear stress is induced in the wire.

The spring will be in equ^n under the action of two forces W & T

$$T = W\times\frac{D}{2} = \frac{\pi}{16}\tau_1 d^3$$

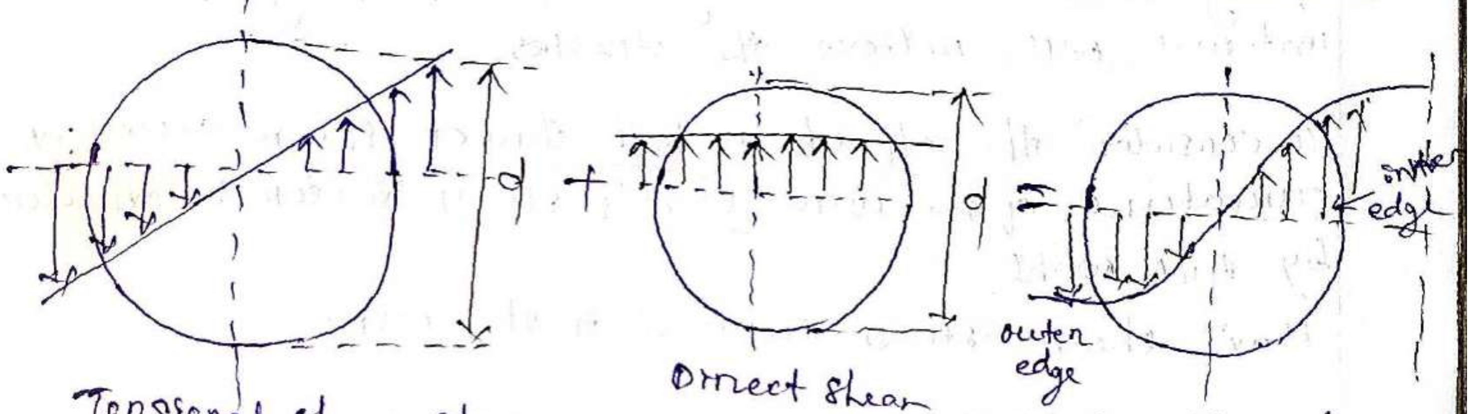
$$\Rightarrow \tau_1 = W\times\frac{D}{2}\times\frac{\cancel{16}\,8}{\pi d^3} = \boxed{\frac{8WD}{\pi d^3} = \tau_1}$$

Along with the torsional shear stress (τ_1) the following stresses will be act on the wire

i) Direct shear stress due to load W &

ii) stress due to curvature of wire

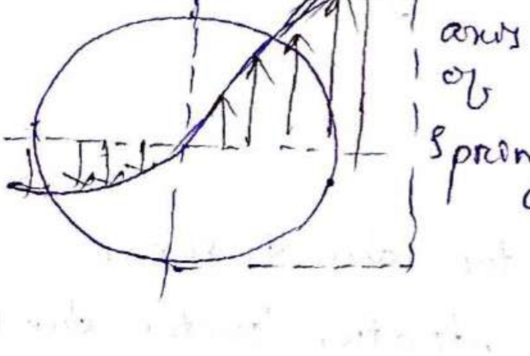
Direct shear stress $= \tau_2 = \dfrac{W}{\frac{\pi}{4}d^2} \Rightarrow \boxed{\tau_2 = \frac{4W}{\pi d^2}}$



Torsional shear stress diagram

Direct shear stress diagram

Resultant torsional shear & direct shear stress diagram



Resultant torsional shear & direct shear & curvatures shear stresses diagram

* Now resultant ~~force~~ shear stress induced in the wire

$$\tau,\ \tau_1 \pm \tau_2 = \frac{8WD}{\pi d^3} \pm \frac{4W}{\pi d^2}$$

+ve sign is for inner ~~at~~ edge & -ve sign is for outer edge of wire.

Max^m shear stress induced in the wire = Torsional shear + direct shear

$$\tau_{max} = \frac{8WD}{\pi d^3} + \frac{4W}{\pi d^2}$$

$$= \frac{8WD}{\pi d^3}\left(1 + \frac{d}{2D}\right)$$

$$= \frac{8WD}{\pi d^3}\left(1 + \frac{1}{2c}\right)$$

$$= K_s \times \frac{8WD}{\pi d^3}$$

Where K_s = Shear stress factor $= 1 + \frac{1}{2c}$

when springs are subjected to static load the effect of spring curvature can be neglected as yielding of the material will relieve the stresses.

To consider the effects of both direct shear as well as curvature of the wire, wahl's stress factor is introduced by A.M. wahl.

Max^m shear stress induced in the wire.

$$\tau = K \times \frac{8WD}{\pi d^3} = K \times \frac{8Wc}{\pi d^2}$$

where $\boxed{K = \frac{4c-1}{4c-4} + \frac{0.615}{c}}$

So $K \uparrow$ when $c \downarrow$

Again $\boxed{K = K_s + K_c}$

where K_s = stress factor due to shear

K_c = stress concentration factor due to curvature

Deflection of helical springs of circular wire →

Total length of wire = l = length of one coil × no. of active

$$\boxed{l = \pi D \times n}$$

Let θ = Angular deflection of the wire when torque T is acting

∴ Axial deflection of the spring $\boxed{\delta = \theta \times \frac{D}{2}}$

From torsion eqn

$$\frac{T}{J} = \frac{\tau}{D/2} = \frac{G\theta}{l} \Rightarrow \theta = \frac{Tl}{GJ} = \frac{\left(W \times \frac{D}{2}\right) \times \pi D n}{G \frac{\pi}{32} d^4}$$

$$\text{So now } \delta = \theta \times \frac{D}{2} = \frac{32 W D \pi D n}{2 G \pi d^4} \times \frac{D}{2} = \frac{8 W D^3 n}{G d^4}$$

$$= \frac{8W(Cd)^3 n}{Gd^4} \quad \left(\because C = \frac{D}{d}\right) \Rightarrow \frac{8WC^3 d^3 n}{Gd^4} \Rightarrow \frac{8WC^3 n}{Gd}$$

$$\boxed{\delta = \frac{8WC^3 n}{Gd}}$$

$$\& \text{ Stiffness} = k = \frac{W}{\delta} = \frac{W}{\left(\frac{8WC^3 n}{Gd}\right)} = \frac{Gd}{8C^3 n} = \text{const.}$$

Eccentric loading of spring

→ When the axial load does not ~~coi~~ coincide with the axis of spring then due to eccentric loading safe load decrease & k also affected

The eccentric load on the spring increases stress on one side & decrease stress on other side

For eccentricity e, the safe load on spring

$$= W \times \left(\frac{D}{2e + D}\right)$$

D — mean dia of spring; $\frac{D}{2e+D}$ — Factor

Buckling of compression springs

When $L_f > 4D$ then spring behaves like a column which may fail due to buckling

Critical buckling load $= W_{cr} = k \times K_B \times L_f$

Where k = Spring const.

K_B = buckling factor depends on ratio $\frac{L_f}{D}$ (can be referred to data)

L_f = Free length of spring

IMP Surge in Springs:-

→ When one end of a helical spring is resting on a rigid support & the other end ~~of~~ is loaded suddenly, then all the coils of the spring will not suddenly deflects equally because some time is required for propagation of stress along the spring wire.

At begining, the end coils of the spring are in contact with applied load, takes the whole deflection & then it transmit it to the adjacent coils in this way a wave of compression ~~propagation~~ propagates through the coils to support end from where it is reflected back to the deflected end

for 2 mark → When one end of the spring is suddenly loaded with other end ~~as~~ resting on a rigid support then all the coils will not deflect equally. At starting the coil in contact with load absorbs the total deflection & transmits the same to the adjacent coil. Thus a wave of compression propagates through the coils. /

If the applied load is ~~fluctuated~~ fluctuating type & if the time interval betn load applied is equal to time required, for wave to travel from one end to other, then resonance will occurs. This will result in very large deflection of coil & this will produce high stresses, so due to this the spring, may ~~be~~ fail, which is called as surge

So f_n of spring should ≥ 20 (frequency of periodic load) to avoid resonance.

$$f_n = \frac{d}{2\pi D^2 n}\sqrt{\frac{6Gg}{\rho}} \text{ cycle/sec}$$

f_n → natural frequency
ρ → density of the material
g → accn due to gravity

G → modulus of rigidity
n → no. of active turns
d → wire dia
D → coil dia

Surge in spring can be eliminated by

i) By using friction dampers ~~on~~ on the centre coils so that wave propagation restricts.

ii) By using spring of high f_n

iii) " " " with pitch of the coil near the end different from pitch of the spring at the centre.

Q A compression coil spring made of an alloy steel with $D = 50mm$, $d = 5mm$, $n = 20$, if it is subjected to an axial load of 500N, calculate max^m τ (neglect curvature effect) to which the spring material is subjected.

Sol^n $C = \frac{D}{d} = \frac{50}{5} = 10$

$k_s = 1 + \frac{1}{2c}$, $1 + \frac{1}{2 \times 10} = 1.05$

$\tau_{max} = k_s \times \frac{8WD}{\pi d^3} = 1.05 \times \frac{8 \times 500 \times 50}{\pi \times 5^3} = 534.7 \, N/mm^2$

2) A helical spring is made from a wire of 6mm dia & has outside dia of 75mm if the permissible $\tau = 350MPa$, & $G = 84 \, KN/mm^2$ Find the axial load & deflect^n per active turn.

10 complet

Sol^n Given $d = 6mm$ $\tau = 350 \, MPa$ $W = ?$

$D_o = 75mm$ $G = 84 \times 10^3 \, MPa$ $\delta/n = ?$

Mean dia $D = D_o - d = 75 - 6 = 69mm$

$C = \frac{D}{d} = \frac{69}{6} = 11.5$

neglecting curvature effect

$k_s = 1 + \frac{1}{2c} = 1 + \frac{1}{2 \times 11.5} = 1.043$

$\tau = k_s \frac{8WD}{\pi d^3} = 1.043 \times \frac{8W \times 69}{\pi \times 6^3} = 0.848 \, W$

$\Rightarrow W = \frac{350}{0.848} = 412.7 \, N$

we know $\delta = \frac{8WD^3 n}{Gd^4}$

$\delta/n = \frac{8WD^3}{Gd^4} = \frac{8 \times 412.7 (69^3)}{84 \times 10^3 \times 6^4} = 9.96mm$

Considering curvature effect

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\times 11.5-1}{4\times 11.5-4} + \frac{0.615}{11.5} = 1.123$$

$$\tau = K\frac{8WC}{\pi d^2} = 1.123\times\frac{8W\times 11.5}{\pi 6^2} = 0.913W \Rightarrow W = 383.4\ N$$

$$\delta/n = \frac{8WD^3}{Gd^4} = \frac{8\times 383.4\,(69)^3}{84\times 10^3\times 6^4} = 9.26\ mm$$

Q2 Design a spring for a balance to measure '0' to 1000 N over a scale of length 80mm. The spring is to be enclosed in a casting of 25mm dia. The approximate no. of turns is 30. G = 85 kN/mm². Also calculate the max^m shear stress induced.

Sol^n $W = 1000\ N$, $n = 30$

$\delta = 80mm$, $G = 85\ kN/mm^2$

As spring is to be enclosed in a casting of 25 mm dia so (D_o = Ded) should be less than 25mm

$$\delta = \frac{8WC^3n}{Gd} \Rightarrow 80 = \frac{8\times 1000\times C^3\times 30}{85\times 10^3\times d} \Rightarrow \frac{C^3}{d} = 28.3$$

Assume $d = 4mm$, $C^3 = 28.3d \Rightarrow C = 4.84 \Rightarrow \frac{D}{4} = 4.84$

$\Rightarrow D = 19.36\ mm$

So $D_o = D + d = 19.36 + 4 = 23.36\ mm$

As $D_o = 23.36$ mm which is less than the casting dia of 25mm, So assumed dimension $d = 4mm$ is correct

$$\tau_{max} = K\times\frac{8WD}{\pi d^3} = 1.322\times\frac{8\times 1000\times 19.36}{\pi\times 4^3} = 1018.2\ MPa$$

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = 1.322$$

Q. Design a helical compression spring for a maxm load of 1000 N for a deflection of 25mm using spring index as 5 max permissible shear stress for spring wire is 420 MPa & G = 84 kN/mm² Take wahl's stress factor

Soln $G = 84\ kN/mm^2 \quad c = 5 \qquad \delta = 25mm$

$W = 1000\ N, \quad \tau = 420\ MPa,$

1) mean dia of spring

$$K = \frac{4c-1}{4c-4} + \frac{0.615}{c} = \frac{4\times5-1}{4\times5-4} + \frac{0.615}{5} = 1.31$$

$$\tau = K \times \frac{8WD}{\pi d^3} \Rightarrow 420 = 1.31 \times \frac{8\times1000\times c}{\pi d^2} = 1.3 \times \frac{8\times1000\times5}{\pi d^2}$$

$$\Rightarrow d = 6.3\ mm$$

From data book take standard wire of size SWG 3 having $d = 6.401\ mm$ mean dia of spring coil $= D = cd = 5\times6.401$
$= 32.005\ mm$

$D_o = D + d = 32.005 + 6.401 = 38.406\ mm$

2) No of turn of coil

$$\delta = \frac{8Wc^3n}{Gd} \Rightarrow 25 = \frac{8\times1000\times5^3\times n}{84\times10^3\times6.401} =$$

$$\Rightarrow n = 13.44 \approx 14$$

For square & ground end, $n = n+2 = 14+2 = 16$

3) Free length of spring (L_f)

$L_f = n'd + \delta_{max} + 0.15\delta$

$= (16\times6.401) + (1.15\times25) = 131.2\ mm$

4) pitch of ~~coil~~ coil

$$\frac{\text{Free length}}{n-1} = \frac{131.2}{16-1} = 8.75\ mm$$