Lectures notes

On

DESIGN OF MACHINE ELEMENTS

Course Code-TH2

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Chapter-1

Introduction

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* Machine

A machine can be defined as anything that reduces human efforts.

* Designing :-

The procedure in which various specification of a machine are finalized such as its dimensions, material selection, stress distribution, its working mechanism etc. is called designing.

* What is Machine design and classify it?

Machine design:-

→ It is the branch of engineering science that deals with the existing modification of the machine or development of the entirely new machine.

→ In othe words we can say that it is the subject which deals with machine.

Classification of Machine Design:

Machine design can be classified into three types. They are.

i- Adaptive Machine design

ii- Development Machine design

iii- New Machine design.

i) Adaptive Machine Design :-

→ The machine design where we are producing the replicas of the existing machine is known as Adaptive machine design.

→ This designing process requires technical skills to adapt the existing mechanism of a machine.

ii) Development Machine Design :-

→ The designing process where we are modifying an existing machine is known as development machine design.

→ This designing process involves a lot of creative thinking and technical skills required for the modification of the machine.

iii) New Machine Design :-

→ The machine design process where we are developing a totally new machine is known as new machine design.

→ Since we have to developed a new machine this process will involve a lot of research work related to the machine along with creative thinking and technical skills.

There are some other types of machine design, they are

1- Optimum Machine Design.
2- Industrial Machine Design.
3- Computer aided Machine Design
4- Element Machine Design.
5- System Machine Design
6- Rational Machine Design.

* Types of Designing Stresses:

There are basically three types of designing stress.

i- working stress
ii- yield stress
iii- Ultimate stress.

i- working stress:

the stress applied on a machine component where the machine can work properly with out any damage is known as working stress.

ii- Yield stress:

the stress applied on a machine component results in diformation of the machine components is known as yield stress.

iii- Ultimate stress:

The maximum stress, a machine components can withstand is known as ultimate stress.

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~~Design Stress~~

Factor of Safety (FOS)

→ The factor of safety can be defined as the ratio of maximum stress to the working stress/design stress.

$$FOS = \frac{\text{Maximum Stress}}{\text{Working Stress}}$$

→ It is unitless quantity.

→ In case of ductile materials such as mild steel, the FOS can be calculated by the formula

$$FOS = \frac{\text{Yield Stress}}{\text{working/design Stress}}$$

→ For brittle material such as cast iron, the FOS can be calculated by the formula..

$$FOS = \frac{\text{Ultimat Stress}}{\text{Working/design Stress}}$$

Q) A mild steel rod is subjected to a yield load of 3.5 KN having 10mm diameter. Calculate the FOS for the material, if the working stress is given 30 MPa.

Data given

Yield load = 3.5 KN

Diameter = 10mm

= 0.01m

$\therefore$ Area $= \frac{\pi}{4} \times d^2$

$= \frac{\pi}{4} \times (0.01)^2 m^2 = 7.85 \times 10^{-5} m^2$

Working stress $= 30$ MPa

$= 30 \times 10^6 N/m^2$

Yield force $= 3.5$ KN

$= 3.5 \times 10^3$ N

Yield stress $= \frac{\text{Yield force}}{\text{Area}}$

$= \frac{3.5 \times 10^3 N}{7.85 \times 10^{-5}}$

$= 4.45 \times 10^7$

$\therefore$ FOS $= \frac{\text{Yield stress}}{\text{working stress}}$

$= \frac{4.45 \times 10^7}{30 \times 10^6}$

$= 1.483$

Q-2

A cast iron rod having 15mm dia is subjected to a ultimate load of 6KN. Calculate the FOS for the material if the design stress is given 45 MPa.

Data given

Diameter = 15mm = 0.015m

Ultimate load = 6KN $= 6 \times 10^3$ N

$\therefore$ Area $= \frac{\pi}{4} \times d^2$

$= \frac{\pi}{4} \times (0.015)^2$

$= 1.77 \times 10^{-4} \ m^2$

Design Stress $= 45 \ MPa$

$= 45 \times 10^6 \ N/m^2$

$\therefore$ Ultimate Stress $= \dfrac{\text{Ultimate load}}{\text{Area}}$

$= \dfrac{6 \times 10^3}{1.77 \times 10^{-4}}$

$= 33898305.08 \ N/m^2$

$\therefore$ FOS $= \dfrac{\text{Ultimate stress}}{\text{design stress}}$

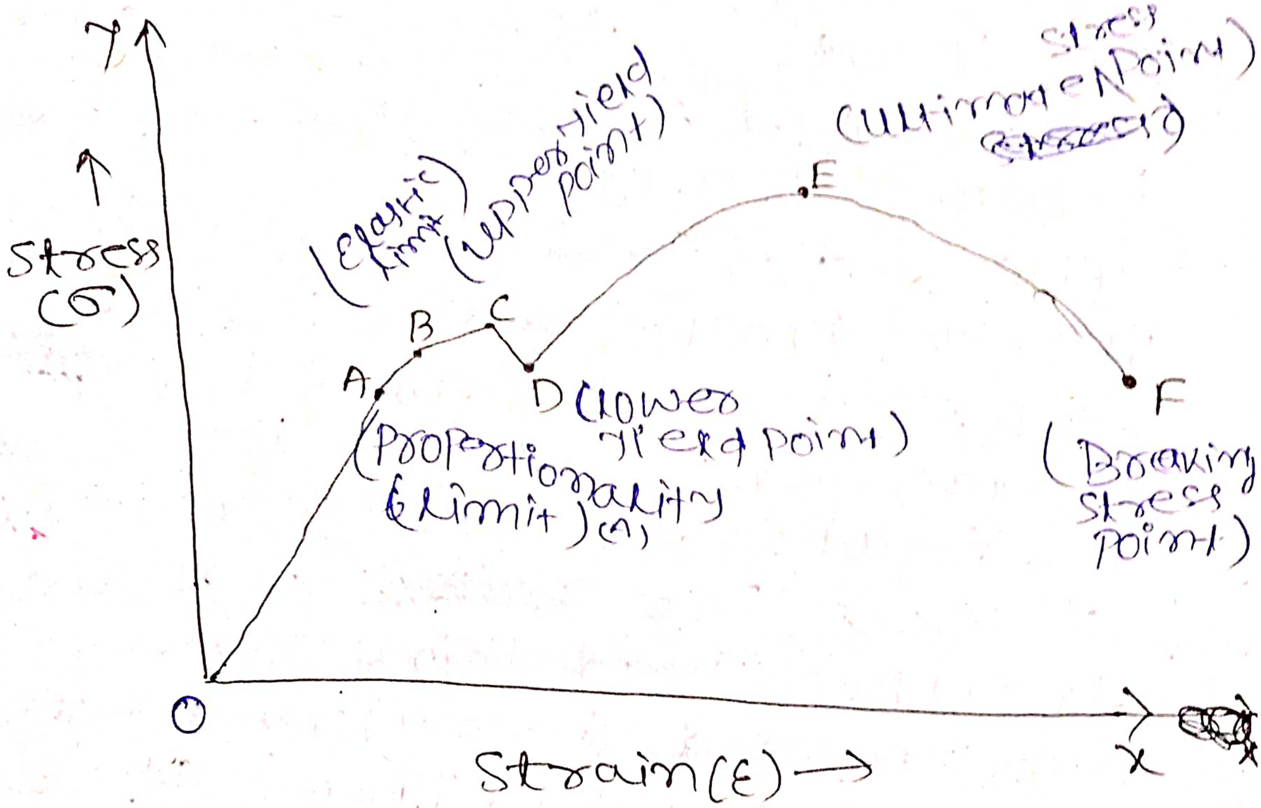
$= \dfrac{33898305.08}{45 \times 10^6}$

$= 0.75$

Stress-Strain Diagram :- 12

→ The stress-strain diagram can be defined as the graphical representation of stress with respect to strain. when they are plotted against each other in a two dimensional axis.

Stress-Strain Diagram for Ductile Material :- (mild steel) (M.S) (5)



- The stress strain diagram for ductile material can be plotted into different stages.
- Here we have plotted stress against strain in a two dimensional axis.

→ From the starting point 'O' till 'A' is the region which obeys Hooke's law, that means here the stress is directly proportional to strain. point 'A' is the proportionality limit and beyond 'A' the stress will not be directly proportional to strain.

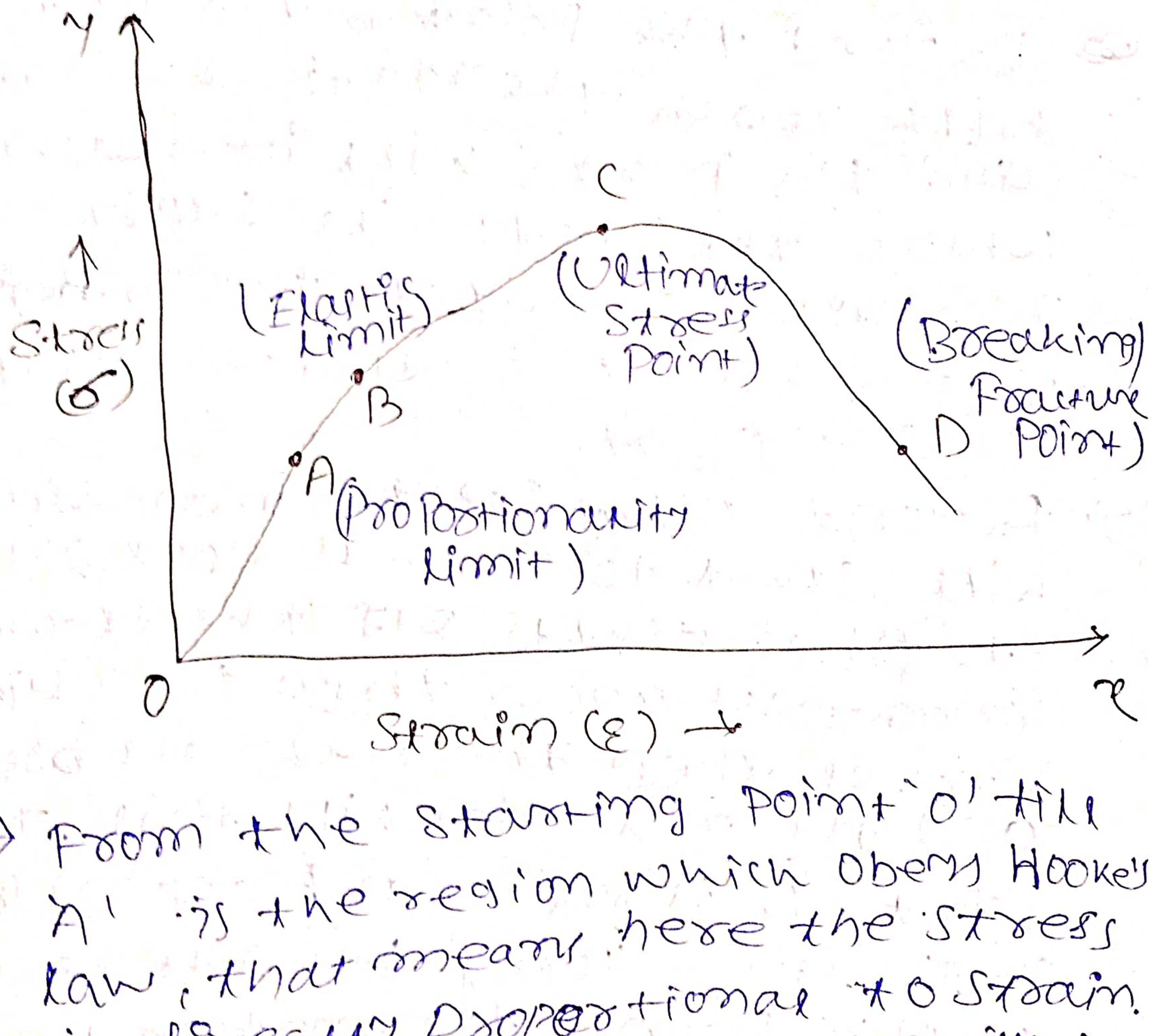
→ The region 'AB' is the elastic region where the body will deformed when the force is applied and it can again comback to its original shape and size after the removal of force. The point 'B' is the elastic limit and beyond 'B' the material will become plastic.

→ The region from 'B' to 'D' comes under plastic region where the deformation in the body is the material is always permanent. The material here retain the changes. The point 'C' is known as upper yield point since maximum deformation of the material takes place at 'C'. The point 'D' is known as lower yield point as minimum deformation takes place at point 'D'.

→ Beyond point 'D' a neck will start to form after it reaches the point 'E'. The point 'E' is known as ultimate stress point. As it is the point where a body can withstand maximum stress.

→ Beyond Point 'E' If the Stress increase then it will reach the Point 'F'. The Point 'F' is known as Breaking Stress Point, where a body or material fails due to excessive stress and Breaking of the neck of the material.

* Stress - Strain Diagram for Brittle Material :- (cast iron) (C.I) [5]



→ From the starting point 'O' till 'A' is the region which obeys Hooke's law, that means here the stress is directly proportional to strain. Point 'A' is the proportionality limit and beyond 'A' the stress will not be directly proportional to strain.

→ The region 'AB' is the elastic region where the body will deformed when the force is applied and it can again comback to its original shape and size after the removal of of force. The Point 'B' is the elastic limit and beyond 'B' the material will becomes Plastic.

→ In case of brittle materials there is absence of yield region so no deformation takes place in case of brittle materials.

Beyond the Point 'B' the material will reach the Plastic region and the Point 'C' will be reached where the body will sustain maximum amount of stress, Hense the Point 'C' is known as Ultimate Stress Point.

→ Beyond Point 'C' the material will start to form a neck on it. As a results if the stress increases more than the ultimate stress then the neck will break and the material will fail. The Point 'D' is known as Breaking Point or Fracture Point because here at this Point the material breaks and ultimately fails.

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General considerations in machinery or factors governing the machine design process.

15/10

There are various factors that are to be considered while designing a machine. They are

1- Cost of construction
2- forces & stresses subjected on machine parts
3- Motion of the machine or kinematics
4- Material selection.
5- form & size of the parts.
6- Frictional resistance & lubrication.

1- Cost of construction :-

-> The most important consideration is the cost of construction or the cost of production.
-> A design engineer should always try to minimize the cost of production.
-> The expenditure available place an important role in machine design.

2- forces & stresses subjected on machine parts -:

-> Stress calculation which is to be subjected on any machine component is crucial (imp) as it is directly related to proper functioning of the machine component. If the stress exceeds the permissible value then their is a risk of failure of the machine component..

3. Motion of the machine or kinematics:

→ Every machine components is design to execute a particular motion or mechanism.

→ So the design engineer always try to manage the motion of the machine parts in the desired manner.

4. Material selection:

→ Every material have different mechanical properties.

→ A design engineer should always try to select a proper material for a certain operation.

→ For example ductile material should be selected for manufacturing wire, hard material should be selected for metal cutting, resilient material should be selected for manufacturing springs.

5. Form & size of the parts:

→ A design engineer should always focus on manufacturing the parts in proper size and shape, so that the assembling can be done properly.

6/ Frictional Resistance & Lubrication :

→ Whenever two machine components comes in contact with each other it produces frictional resistance which is responsible for wear & tear of the machine components.

→ Hence, a design engineer should always use better and suitable lubrication process for the removal of excess heat, frictional heat.

7/ Use of standard Parts :

→ The standard parts are somehow more costly as compare to the regular parts by some amount of money. ~~But~~

→ But, by using standard parts we can avoid the replacement of parts in regular intervals.

8/ Safety of operation :

→ Whenever industries are set up, different types of machines are installed.

→ A design engineer should always ensure the safety of operation, so that their is no ~~accident~~ accidents ~~or~~ takes place while machine designing process.

9- Workshop facilities :

→ A good workshop facilities ensures better modification of a machine components.

→ A design engineer should always try to install better ~~works~~ machining facilities in the designing workshop.

10- Assembling :

→ Assembling is a one of the most imp. factor in machine design process.

→ Assembling refers to the process of collectively joining different machine components together in such a manner that it will provide unnessery function.

→ Assembling facilities ensures a better ~~machin~~ in machine designing process.

*Imp Design Procedure: 10/s Dt 25-10-2021

The designing procedure of machine consist of the following steps.

1- Need or Aim
2- Synthesis or Mechanism.
3- Analysis of forces.
4- Material selection.
5- Design of machine component
6- Modification.
7- Detailed drawing.
8- Production

1- Need or Aim:

→ The machine or component we are designing must be usefull and serve a certain purpose

2- Synthesis or Mechanism:

→ The component to be designed must have the desired motion in it.
→ The mechanism needed to move the component or machine should be finalised in this step.

3- Analysis of forces:
→ All the forces acting on each component of the machine should be calculated in order to avoid the risk of failure or breakdown of the machine.

4- <u>material selection</u>:-

→ The material best suited for all the components must be selected in order to get the purposefull result.

5- <u>Design of machine components</u>:-

In this steep, the sizes; shape and all the dimension of the different component of the machine are finalished.

6- <u>modification</u>:-

In this step all the changes ~~required~~ which are required are made by the design engineers.

7- <u>Detailed drawing</u>:-

After the modification are completed, a detailed drawing is prepared for all the components of the machine.

8- <u>Production</u>:-

Once the detailed drawing is prepared then we can produce the particular component or machine in large volume.

* Draw the flowchart for the design procedure. Explain the designing procedure in detail. 10

Need or Aim

↓

Synthesis of mechanism

↓

Analysis of forces

↓

Material selection

↓

Design of Machine component

↓

Modification

↓

Detailed drawing

↓

Production

* Mechanical Properties of Material. 10

There are various mechanical properties of material. They are.

1 - Strength
2 - Stiffness
3 - Elasticity
4 - Plasticity.
5 - Ductility
6 - Brittleness
7 - Malleability.
8 - Toughness
9 - Hardness
10 - Machinability.
11 - Resilience.
12 - Creep
13 - Fatigue.

1- Strength -:

It is the property of the material by virtue of which it can resist ~~[illegible]~~ any external load without ~~[illegible]~~ failure.

2- Stiffness -:

It is the property of the material to resist deformation ~~[illegible]~~ under external load.

3- Elasticity -:

It is the property of the material by vertue of which a material can change its shape or size and again regain its original size and shape after the removal of the deforming ~~forces~~ force ~~process~~.

4- Plasticity

The property of the material by virtue of which ~~the~~ a material change its shape and size but cannot regain its original shape and size after the removal ~~of~~ of the deforming force.

5- Ductility

It is the property of the material by virtue of which a material can be converted into thin wires.

6- Brittleness -:

It is the property of the material to resist any type of deformation under loads.

7- Malleability -:

It is the property of the material by virtue of which a material can be converted into thin sheets.

8- Toughness -:

It is the property of the material by virtue of which a material can resist shock under impact load.

9- Hardness -:

It is the property of the material by virtue of which a material can cut the other material. Ex- Tungsten carbide, silicon carbide, Dimonds.

10- Machinability -:

It is the property of the material by virtue of which a material can be machined.

11- Resilience -:

~~the pro~~ It is the property of the material by virtue of which a material can absorb ~~and~~ or stored energy.

12- Creep -:

When a material is subjected to constant stress ~~and~~ and high temp for a long period of time and it results it slow but ~~~~ permanent deformation, then the material is said to be in creep condition. Ex- Belt, cycle-chain

13- Fatigue -:

When a material is subjected to constant rounds of stress then it results in a conditions called fatigue.

DT·26-10·2021

* Modes of ~~Failure~~ Failure :

1- Elastic deflection
2- Failure due to Yield
3- Failure due to Fracture

1- Elastic deflection :

→ Elastic deflection is the situation in which transmission like shaft, structural members like beams and column are subjected to external forces and stress due to which they are deflected elastically.

→ Due to elastic deflection, a material is not able to perform the desired function for which it is designed and this can be considered as one of the reason for the ~~fail~~ failure of the material.

→ For the example, when a column is deflected elastically, then it cannot provide the support to the beams.

2- Failure due to Yield :

→ Yielding is the process in which ductile materials changes their shape and size when they are subjected to stresses.

→ Due to Yielding ductile materials are some time incapable of ~~to~~

forming the function for which they are designing. This enability ~~of the~~ or drawback of the productive material can be considered as one of the reason for the failure of the material.

→ For example, in belt drives when due to creep the length of the belt increases then it fails to do the power transmission as it continuously slips over the pulley.

3 – Failure due to fracture –:

→ In case of brittle materials there is no yielding region. So when they are subjected to ultimate stresses then instead of changing ~~there~~ their shape and size they break down into pieces. ~~This~~

~~→ This is b~~

→ This breakdown can be considered as the failure of the material.

→ Ex, When cutting tools like tungsten carbide, silicon carbide are subjected to ultimate stress they breakdown.

Design of Shafts and Keys.

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***Shafts :-** 2

Shaft is a rotating machine element that is use to transmit Power & motion.

***Properties of a good Shaft :-** 12/15

i - It should have high strength
ii - It should have high machinability
iii - It should have high wear resistance Property.
iv - It should have low sensitivity factor.
v - It should have high toughness.

Design of Shaft :-

Shaft can be designed according to the following ways :-

1 - According to strength
 i - According to shear stress
 ii - According to combined effect of twisting & bending moment.
2 - According to modulus of rigidity

1 - According to strength

i - According to shear stress :-

From the torsion equation we know that.

$$\frac{T}{J} = \frac{\tau}{r} = \frac{G\theta}{l}$$

so $\frac{T}{J} = \frac{\tau}{r}$

$$\Rightarrow \frac{T}{\frac{\pi}{32} \times d^4} = \frac{\tau}{\frac{d}{2}} \quad \left[\because J = \frac{\pi}{32} \times d^4 \quad \because r = \frac{d}{2}\right]$$

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

$$\Rightarrow T = \frac{\tau \times \cancel{2} \times \pi \times d^4}{\underset{16}{\cancel{32}} \times \cancel{d}}$$

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

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

Expression for torque for a solid shaft

* For hollow shaft

Design of shaft → calculation of Diameter

$$J = \frac{\pi}{32}\left[D^4 - d^4\right]$$

$$r = \frac{D}{2}$$

$$\therefore \frac{T}{J} = \frac{\tau}{r}$$

$$\Rightarrow \frac{T}{\frac{\pi}{32}\left[D^4 - d^4\right]} = \frac{\tau}{\frac{D}{2}}$$

$$\Rightarrow \frac{T \times 32}{\pi\left[D^4 - d^4\right]} = \frac{\tau \times 2}{D}$$

$$\Rightarrow T = \frac{\tau \times \cancel{2} \times \pi \times \left[D^4 - d^4\right]}{D \times \underset{16}{\cancel{32}}}$$

$$\Rightarrow \boxed{T = \frac{\pi}{16} \times \tau \times \left[\frac{D^4 - d^4}{D}\right]}$$

Expression for torque of hollow shaft.

Power transmitted by the shaft:-

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

where
P = Power transmitted
T = Torque
N = Speed in r.p.m

Q-1 A shaft is rotating at 200 r.p.m. to transmit power of 20 kW. Calculate the diameter of the shaft. If the shear stress is given 42 MPa.

Data given

$N = 200$ r.p.m
$P = 20\text{ kW} = 20 \times 10^3 \text{ W}$
$\tau = 42 \text{ MPa}$
$= 42 \text{ N/mm}^2$

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

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

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

$$\Rightarrow T = 954.92 \text{ N.m} = 954.92 \times 10^3 \text{ N.mm}$$

We know that

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

$$\Rightarrow 954.92 \times 10^3 = \frac{\pi}{16} \times 42 \times d^3$$

$$\Rightarrow d^3 = \frac{954.92 \times 10^3 \times 16}{\pi \times 42}$$

$$\Rightarrow d^3 = 115794.46$$

$$\Rightarrow d = \sqrt[3]{115794.46} = 48.74 \text{ mm}$$

Q-2 A solid shaft is transmitting 1 MW at 240 r.p.m. Determine the diameter of the shaft if the shear stress is given 60 MPa. Here the maximum torque exceeds the mean torque 20%.

Data given

$P = 1\text{ MW} = 1 \times 10^6 \text{ W}$
$N = 240$ r.p.m.
$\tau = 60 \text{ MPa} = 60 \text{ N/mm}^2$
$T_{max} = (100 + 20)\% \; T_{mean}$

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

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

$$= \frac{1 \times 10^6 \times 60}{2\pi \times 240}$$

$$= 39788.73 \text{ N.m}$$

$T_{max} = (100+20)\% \, T_{mean}$

$= \frac{120}{100} \times 39788.73$

$= 47746.47 \, N.m$

$= 47746.47 \times 10^3 \, N.mm$

We know that

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

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

$\Rightarrow d^3 = \frac{47746.47 \times 10^3 \times 16}{\pi \times 60}$

$\Rightarrow d = \sqrt[3]{\frac{47746.47 \times 10^3 \times 16}{\pi \times 60}}$

$= 157.43 \, mm$

Q-3 A shaft is rotating at 250 rpm to transmit 150 K.W. Calculate the dia of shaft if the shear stress is 70 M.Pa. The maximum torque exceeds the mean torque by 25%.

<u>Data given</u>

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

$P = 150 \, KW = 150 \times 10^3 \, w$

$\tau = 70 \, M.P = 70 \, N/mm^2$

$T_{max} = (100+25)\% \, T_{mean}$

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

$\Rightarrow 150 \times 10^3 = \frac{2\pi \times 250 \times T_{mean}}{60}$

$\Rightarrow T_{mean} = \frac{150 \times 10^3 \times 60}{2\pi \times 250}$

$\Rightarrow T_{mean} = 5729.57 \, N.m$

$\Rightarrow T_{max} = (100+25)\% \, T_{mean}$

$= \frac{125}{100} \times 5729.57$

$= 7161.97 \, N.m$

$= 7161.97 \times 10^3 \, N.mm$

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

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

$\Rightarrow d = \sqrt[3]{\frac{7161.97 \times 10^3 \times 16}{\pi \times 70}}$

$\Rightarrow d = 80.47 \, mm$

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Q-4

Find the diameter of a solid shaft transmitting 20KW at 200 r.p.m. The ultimate shear stress is given 360 mPa. Take FOS = 8.

Data given

$P = 20\,KW = 20\times10^3\,W$

$N = 200\,rpm$

Ultimate shear stress $= \tau_{u} = 360\,mp$

$= 360\,N/mm^2$

$FOS = 8$

$\therefore \tau = \frac{\tau_{ultimate}}{FOS}$

$= \frac{\cancel{360}^{45}}{\cancel{8}}$

$= 45\,N/mm^2$

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

$20\times10^3 = \frac{2\pi\times200\times T}{60}$

$\Rightarrow T = \frac{20\times10^3\times60}{2\pi\times200}$

$\Rightarrow T = 954.92\,N.m$

$= 954.92\times10^3\,N.mm$

We know that.

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

$\Rightarrow d = \sqrt[3]{\frac{T\times16}{\pi\times\tau}}$

$\Rightarrow d = \sqrt[3]{\frac{954.92\times10^3\times16}{\pi\times45}}$

$= 47.63\,mm$

Q-5

A hollow shaft is transmitting 25KW at 250 r.p.m. If the ultimate shear stress is given 480 mPa, calculate both the diameters of the hollow shaft considering the outside diameter to be twice of the inside diameter. Take FOS = 8.

Data given

$P = 25\,KW = 25\times10^3\,W$

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

$\tau_{ultimate} = 480\,mPa = 480\,N/mm^2$

$FOS = 8, \quad D = 2d$

$\therefore \tau = \frac{\tau_{ultimate}}{FOS} = \frac{\cancel{480}^{60}}{\cancel{8}} = 60\,N/mm^2$

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

$\Rightarrow 25\times10^3 = \frac{2\pi\times250\times T}{60}$

$\Rightarrow T = \dfrac{25\times10^3\times60}{2\pi\times2850}$

$\Rightarrow T = 954.92\ N.m$

$= 954.92\times10^3\ N.mm$

we know that

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

$\Rightarrow 954.92\times10^3 = \dfrac{\pi}{16}\times60\times\left(\dfrac{2d^4-d^4}{D}\right)$

$\Rightarrow 954.92\times10^3 = \dfrac{\pi}{16}\times60\times\left[\dfrac{16d^4-d^4}{D}\right]$

$\Rightarrow \dfrac{15d^4}{D} = \dfrac{954.92\times10^3\times16}{\pi\times60}$

$\cancel{\Rightarrow \dfrac{15d^4}{2d} = 81056.12}$

$\cancel{\Rightarrow \dfrac{d^4}{2d} = \dfrac{81056.12}{15}}$

$\cancel{\Rightarrow d = \sqrt[4]{5403.74}}$

$\cancel{\Rightarrow d = 8.57}$

$\therefore D = 2d$

$\cancel{= 2\times8.57}$

$\cancel{= 17.14}$

$\cancel{d = 8.54}$

$\Rightarrow \dfrac{15d^4}{2d} = \dfrac{954.92\times10^3\times16}{\pi\times60}$

$\Rightarrow 7.5d^3 = \dfrac{954.92\times10^3\times16}{\pi\times60}$

$\Rightarrow 7.5d^3 = 81056.12$

$\Rightarrow d^3 = \dfrac{81056.12}{7.5}$

$\Rightarrow d = \sqrt[3]{10807.48}$

$= 22.10\ mm$

$\therefore D = 2d$

$= 2\times22.10$

$= 44.20\ mm$

$d = 22.10\ mm$

$D = 44.20$

$d = 22.10$

Design of shaft → calculate diameter

ii- According to Combine effect of twisting and bending moment :

Let T = Torque produced in the shaft

M = Bending moment acting on shaft.

* Equivalent twisting moment (T_e)

Whenever a twisting force acts on a shaft then it produces an ~~twisting~~ equivalent twisting moment to offer resistance to the process of twisting.

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

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

* Equivalent Bending moment (M_e)

Whenever a bending force is acting on a shaft then the shaft produces an equivalent bending moment to ~~offer~~ offer resistance to the process of bending.

$$\boxed{M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right]}$$

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

σ_b = bending stress.

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Q-1

A solid circular shaft is subjected to a bending moment of 3000 N.m and a torque of 10,000 N.m. Design the shaft, if the ultimate tensile stress (Bending stress) is 700 MPa and ultimate shear stress = 500 MPa. Take FOS = 6.

Data given

Bending moment = M = 3000 N.m
$= 3000 \times 10^3$ N.mm
$= 3000 \times 10^3$ N.mm

Torque = T = 10,000 N.m
$= 10,000 \times 10^3$ N.mm

$\sigma_{b\,ultimate}$ = 700 MPa = 700 N/mm²

$\tau_{ultimate}$ = 500 MPa = 500 N/mm²

FOS = 6

Equivalent twisting moment

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

$$= \sqrt{3000^2 + 10000^2} = \sqrt{3000^2 + 10000^2}$$

$= 3605.55$ N.m $= 10440.30$ N.m

$= 3605.55 \times 10^3$ N.mm $= 10440.30 \times 10^3$ N.mm

$\tau_{ultimate}$ = 500 N/mm²

$$\therefore \tau = \frac{\tau_{ultimate}}{FOS}$$

$$= \frac{500}{6} = 83.33 \text{ N/mm}^2$$

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

$\Rightarrow 114.01 \times 10^3 = \frac{\pi}{16} \times 83.33 \times d^3$

$\Rightarrow d = \sqrt[3]{\frac{\cancel{114.01}\ 10440.39 \times 10^3 \times 16}{\pi \times 83.33}}$

$\Rightarrow d = 86.09$ mm

Equivalent Bending moment

$M_e = \frac{1}{2}[M + \sqrt{M^2 + T^2}]$

$= \frac{1}{2}[3000 + \sqrt{3000^2 + 10{,}000^2}]$

$= 6720.15$ N.m

$= 6720.15 \times 10^3$ N.mm

$\sigma_b = \frac{\sigma_{b\,ultimate}}{FOS}$

$= \frac{700}{6}$

$= 116.66$ N/mm²

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

$\Rightarrow 6720.15 \times 10^3 = \frac{\pi}{32} \times 116.66 \times d^3$

$\Rightarrow d = \sqrt[3]{\frac{6720.15 \times 10^3 \times 32}{\pi \times 16.66}} = 83.71$ mm

Taking the larger value of d, d = 86.09 mm

Q.2

A solid circular shaft is subjected to a bending moment of 2000 N.m and twisting moment of 8000 N.m. Design the shaft, if the ultimate bending stress 750 MPa and ultimate shear stress is 500 MPa. Take FOS = 5. [10]

Data given

$M_e = 2000$ N.m $= 2000 \times 10^3$ N.mm

$T_e = 8000$ N.m $= 8000 \times 10^3$ N.mm

$\sigma_{b\,ultimate} = 750$ MPa $= 750$ N/mm²

$\tau_{ultimate} = 500$ MPa $= 500$ N/mm²

Equivalent twisting moment

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

$\Rightarrow \tau = \frac{\tau_{ultimate}}{FOS}$

$= \frac{500}{5}$

$= 100$ N/mm²

$\Rightarrow 8000 \times 10^3 = \frac{\pi}{16} \times 100 \times d^3$

$\Rightarrow d = \sqrt[3]{\frac{8000 \times 10^3 \times 16}{\pi \times 100}} =$

Q. A solid circular shaft is subjected to a bending moment of 2000 N.m and twisting moment of 8000 N.m. Design the shaft if the ultimate bending stress is 750 MPa and ultimate shear stress is 500 MPa. Take FOS = 5.

Data given

$M = 2000$ N.m

$T = 8000$ N.m

$\sigma_{b_{ult}} = 750$ MPa $= 750$ N/mm²

$\tau_{ult} = 500$ MPa $= 500$ N/mm²

FOS $= 5$

Equivalent Twisting moment

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

$$= \sqrt{2000^2 + 8000^2}$$

$$= 8246.21 \text{ N.m}$$

$$= 8246.21 \times 10^3 \text{ N.mm}$$

$$\tau = \frac{\tau_{ult}}{FOS}$$

$$= \frac{500}{5} = 100 \text{ N/mm}^2$$

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

$$\Rightarrow 8246.21 \times 10^3 = \frac{\pi}{16} \times 100 \times d^3$$

$$\Rightarrow d = \sqrt[3]{\frac{8246.21 \times 10^3 \times 16}{\pi \times 100}}$$

$$= 74.88 \text{ mm.}$$

Equivalent Bending moment

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

$$= \frac{1}{2}[2000 + 8246.21]$$

$$= 5123.10 \text{ N.m}$$

$$= 5123.10 \times 10^3 \text{ N.mm}$$

$$\sigma_b = \frac{\sigma_{b_{ult}}}{FOS}$$

$$= \frac{750}{5}$$

$$= 150$$

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

$$\Rightarrow 5123.10 \times 10^3 = \frac{\pi}{32} \times 150 \times d^3$$

$$\Rightarrow d = \sqrt[3]{\frac{5123.10 \times 10^3 \times 32}{\pi \times 150}}$$

$$= 70.33 \text{ mm}$$

Taking the largest value of d, d = 74.88 mm

Q.3

A solid steel shaft transmit 20KW at 200 r.p.m. It carries a central load of 900 N and is simply supported between the bearings which are 2.5 m apart. Determine the diameter of the shaft, if the shear stress 42 MPa and the bending stress is 56 MPa. 10

Data given

$P = 20\ KW = 20 \times 10^3\ W$

$N = 200\ r.p.m$

$W = 900\ N$

$L = 2.5\ m$

$\tau = 42\ MPa = 42\ N/mm^2$

$\sigma_b = 56\ MPa = 56\ N/mm^2$

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

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

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

$$\Rightarrow \boxed{T = 954.92\ N.m}$$

* Simply supported beam subjected to central load, where

$$M = \frac{W \times L}{4}$$

$$\therefore M = \frac{W \times L}{4} = \frac{900 \times 2.5}{4}$$

$$\boxed{M = 562.5\ N.m}$$

Equivalent twisting moment

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

$$= \sqrt{562.5^2 + 954.92^2}$$

$$= 1108.27\ N.m$$

$$= 1108.27 \times 10^3\ N.mm$$

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

$$\Rightarrow 1108.27 = \frac{\pi}{16} \times 42 \times d^3$$

$$\Rightarrow d = \sqrt[3]{\frac{1108.27 \times 16}{\pi \times 42}}$$

$$= 51.22\ mm$$

Equivalent Bending moment

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

$$= \frac{1}{2}[562.5 + 1108.27]$$

$$= 835.38\ N.m = 835.38 \times 10^3\ N.mm$$

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

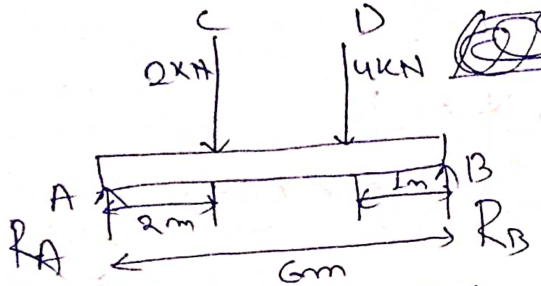
$$\Rightarrow 835.38 \times 10^3 = \frac{\pi}{32} \times 56 \times d^3$$

$$\Rightarrow d = \sqrt[3]{\frac{835.38 \times 10^3 \times 32}{\pi \times 56}} = 53.36\ mm$$

Taking the larger value of d,d = 53.36 mm

D+-08-11-2021

*



Taking 'A' as Reference

$R_B \times 6 = (2\times2) + (4\times5)$

$\Rightarrow R_B \times 6 = 4+20$

$\Rightarrow R_B \times 6 = 24$

$\Rightarrow R_B = \frac{24}{6}$

$\Rightarrow R_B = 4\ kN$

$R_A + R_B = C + D$

$\Rightarrow R_A + 4 = 2+4$

$\Rightarrow R_A + 4 = 6$

$\Rightarrow R_A = 6-4$

$\Rightarrow R_A = 2\ kN$

Bending moment at 'A' = 0

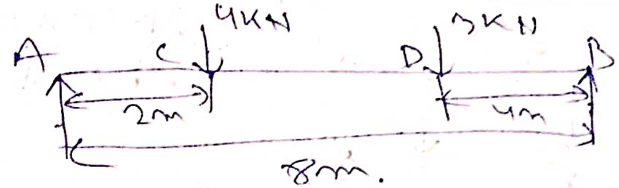
" " " 'B' = 0

Bending moment at 'C' $= (2\times2) - (2\times0)$

$= 4 - 0 = 4\ kN.m$

$(B.m)_D = (4\times2) - (4\times0) = 4-0 = 4\ kN.m$

*



Taking 'A' as reference

$R_B \times 8 = (4\times2) + (3\times4)$

$\Rightarrow R_B \times 8 = 8 + 12$

$\Rightarrow R_B = \frac{20}{8}$

$\Rightarrow R_B = 2.5\ kN$

$R_A + R_B = C + D$

$\Rightarrow R_A + 2.5 = 4 + 3$

$\Rightarrow R_A = 7 - 2.5$

$\Rightarrow R_A = 4.5\ kN$

$(B.m)_A = 0$

$(B.m)_B = 0$

$(B.m)_C = (4.5\times2) - (4\times0)$

$= 9 - 0$

$= 9\ kN.m$

$(B.m)_D = (2.5\times4) - (3\times0)$

$= 10 - 0$

$= 10\ kN.m$

~~08-11-2021~~

07-09-11-2021

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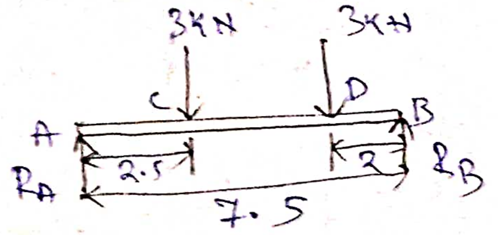


Taking 'A' as reference.

$R_B \times 7.5 = (3 \times 2.5) + (3 \times 5.5)$

$\Rightarrow R_B = \frac{24}{7.5}$

$\Rightarrow R_B = 3.2\ KN$

$R_A + R_B = C + D$

$\Rightarrow R_A + 3.2 = 3 + 3$

$\Rightarrow R_A = 6 - 3.2$

$= 2.8\ KN$

$(B.M)_A = 0$

$(B.M)_B = 0$

$(B.m)_C = (2.8 \times 2.5) - (3 \times 0)$

$= 7 - 0 = 7\ KNm$

$(B.m)_D = (3.2 \times 2) - (3 \times 0)$

$= 6.4 - 0 = 6.4\ KNm$

Q-

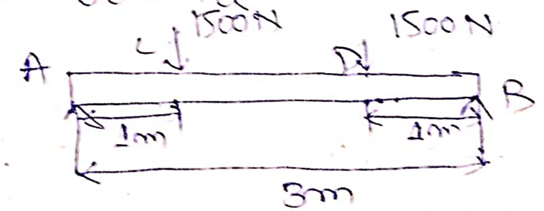
A Solid Steel Shaft is required to transmit 100 KW at 300 r.p.m. The Supported length of 'A' simply supported beam is 3m. It carries two loads 1500N at a distance of 1m from each end respectively. Calculate the diameter of the shaft considering equivalent twisting moment. Take shear stress $\tau = 60\ MPa$. [10]

Data given

$P = 100KW = 100 \times 10^3 W$

$N = 300\ r.p.m$

$\tau = 60\ MPa = 60\ N/mm^2$



$P = \frac{2\pi NT}{60}$

$\Rightarrow 100 \times 10^3 = \frac{2\pi \times 300 \times T}{60}$

$\Rightarrow T = \frac{100 \times 10^3 \times 60}{2\pi \times 300}$

$\Rightarrow \boxed{T = 3183.09\ N.m}$

$R_B \times 3 = (1500 \times 1) + (1500 \times 2)$

$\Rightarrow R_B = \frac{4500}{3} = 1500\ N$

$\boxed{R_B = 1500}$

$R_A + R_B = C + D$

$\Rightarrow R_A + 1500 = 1500 + 1500$

$\Rightarrow R_A = 3000 - 1500$

$\Rightarrow \boxed{R_A = 1500\,N}$

$(B.M)_A = 0$

$(B.M)_B = 0$

$(B.M)_C = (1500 \times 1) - (1500 \times 0)$

$= 1500$ N.m

$(B.M)_D = (1500 \times 1) - (1500 \times 0)$

$= 1500$ N.m.

$\therefore T = 3183.09$ N.m

$M = 1500$ N.m

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

$= \sqrt{1500^2 + 3183.09^2}$

$= 3518.81$ N.m

$= 3518.81 \times 10^3$ N.mm

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

$\Rightarrow 3518.81 \times 10^3 = \frac{\pi}{16} \times 60 \times d^3$

$\Rightarrow d = \sqrt[3]{\frac{3518.81 \times 10^3 \times 16}{\pi \times 60}}$

$\Rightarrow \boxed{d = 66.84\text{ mm}}$

Q-

A solid steel shaft transmits 20KW at 200 r.p.m. This simply supported beam carries a central load of 700N. The length of the beam is given 3m. Design the shaft according to combined effect of twisting moment and bending moment, if the shear stress is given 42MPa and the bending stress is given 60MPa. 10

Data given

$P = 20\,KW = 20 \times 10^3\,W$

$N = 200$ r.p.m

$W = 700\,N$

$L = 3m$

$\tau = 42\,MPa = 42\,N/mm^2$

$\sigma_b = 60\,MPa = 60\,N/mm^2$

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

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

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

$\Rightarrow \boxed{T = 954.92\ \text{N.m}}$

$$M = \frac{W \times L}{4}$$

$$= \frac{700 \times 3}{4}$$

$$= 525$$

$$\boxed{M = 525\,Nm}$$

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

$$= \sqrt{525^2 + 957.92^2}$$

$$= 1089.72\,N.m$$

$$\boxed{T_e = 1089.72 \times 10^3\,N.mm}$$

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

$$\Rightarrow d = \sqrt[3]{\frac{16 \times T_e}{\pi \times \tau}}$$

$$= \sqrt[3]{\frac{16 \times 1089.72 \times 10^3}{\pi \times 42}}$$

$$= 50.93\,mm$$

$$\boxed{d = 50.93\,mm}$$

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

$$= \frac{1}{2}\left[525 + \sqrt{525^2 + 957.92^2}\right]$$

$$= 807.36\,N.m$$

$$\boxed{M_e = 807.36 \times 10^3\,N.mm}$$

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

$$\Rightarrow 807.36 \times 10^3 = \frac{\pi}{32} \times 60 \times d^3$$

$$\Rightarrow d = \sqrt[3]{\frac{807.36 \times 10^3 \times 32}{\pi \times 60}}$$

$$\Rightarrow \boxed{d = 51.55\,mm}$$

Taking the larger value of d, ~~d=50.93~~

$d = 51.55mm$

Dt 15-11-2021

Design of Shaft According to Modulus of Rigidity :-

Modulus of rigidity

$$C/G = \frac{\tau}{\phi} = \frac{\text{Shear Stress}}{\text{Shear Strain}}$$

→ According to torsion equation

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\theta}{l}$$

$$\Rightarrow \frac{T}{J} = \frac{G\theta}{l}$$

Where
T = Torque
J = Polar moment of inertia
$= \frac{\pi}{32} \times d^4$
G = Modulus of rigidity
θ = Angle of twist :- (radian) $\left(x^{\circ} \times \frac{\pi}{180}\right)$
l = Length of shaft.

Q. A steel shaft transmits 4kW at 800 r.p.m. The angle of twist is 0.25°/m. Calculate the diameter of the shaft if the modulus of rigidity is given 84 GPa.

Data given

$P = 4kW = 4\times10^3 W$, $l = 1m = 1000mm$

$N = 800$ r.p.m

$\theta = 0.25^{\circ}/m = 4.36\times10^{-3}$ rad/m =

$G = 84 GPa$
$= 84\times10^9 \times 10^{-6}\ N/mm^2$
$= 84\times10^3\ N/mm^2$

$$P = \frac{2\pi NT}{60}$$

$$\Rightarrow 4\times10^3 = \frac{2\times\pi\times800\times T}{60}$$

$$\Rightarrow T = \frac{4\times10^3\times60}{2\times\pi\times800}$$

$\Rightarrow T = 47.74$ N.m

$$\boxed{T = 47.74\times10^3\ N.mm}$$

$$\frac{T}{J} = \frac{G\theta}{l}$$

$$\Rightarrow \frac{47.74\times10^3}{J} = \frac{84\times10^3\times4.36\times10^{-3}}{1000}$$

$$\Rightarrow \frac{47.74\times10^3}{J} = 0.36$$

$$\Rightarrow J = \frac{47.74\times10^3}{0.36}$$

$$\Rightarrow \boxed{J = 132611.11}$$

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

$$\Rightarrow 132611.11 = \frac{\pi}{32}\times d^4$$

$$\Rightarrow d = \sqrt[4]{\frac{132611.11\times32}{\pi}} = \boxed{34.09\ mm}$$

Q A solid shaft is transmitting 3kW at 400 r.p.m. The angle of twist is $\frac{1}{180}$th of the circumference, degree per metre, then calculate the dia of the shaft. Take G = 84 GPa

Data given

$P = 3kW = 3\times10^{3} W$

$N = 400$ r.p.m

$\theta = \frac{1}{180}\times360$

$= 2^{\circ}$

$= 2\times\frac{\pi}{180}$

$= 0.0349 \text{ rad/m}$

$l = 1 m$

$= 1000 mm$

$G = 84 GPa = 84\times10^{3} N/mm^{2}$

$\Rightarrow P = \frac{2\pi NT}{60}$

$\Rightarrow P\times60 = 2\pi NT$

$\Rightarrow T = \frac{P\times60}{2\pi N}$

$\Rightarrow T = \frac{3\times10^{3}\times60}{2\times\pi\times400}$

$\Rightarrow T = 71.61 \text{ N.m}$

$\Rightarrow \boxed{T = 71.61\times10^{3} \text{ N.mm}}$

$$\frac{T}{J} = \frac{G\theta}{l}$$

$$\Rightarrow T \times l = G\theta \times \pi \times d^4$$

$$\Rightarrow d = \sqrt[4]{\frac{T \times l \times 32}{G\theta \times \pi}}$$

$$\Rightarrow d = \sqrt[4]{\frac{71.61 \times 10^3 \times 1000 \times 32}{84 \times 10^3 \times 0.0\frac{3}{4} \times \pi}}$$

$$\cancel{\Rightarrow d = 23.19\,mm}$$

$$\Rightarrow \boxed{d = 22.48\,mm}$$

Checked
OAd

Chaper-3

Dt. 16-11-2021

KEYS & COUPLING

* Keys -:

Key can be defined as a piece of mild steel that is used to connect the shaft with the hub of the pulley, in order to prevent the relative motion between them.

* Types of Keys -:

Keys can be classified into many types. They are

1- Sunk Key or Rectangular Key
2- Saddle Key
3- Tangent Key
4- Round Key
5- Splines

* Strength of a Key -:

→ The maximum of force that a key can withstand without failure is known as strength of key.

→ The keys are always subjected to two forces - i- Shearing force
ii - Crushing force

* Design of Rectangular Sunk Key -:

Design a key → calculate length of key (l)

→ Design of key ~~refers~~ to calculation of length of key.

→ It can be calculated by equating the torque transmitted by the shaft with the shearing action and crushing action of the key.

Let

T = Torque transmitted by the shaft

τ = Shear stress

d = Diameter shaft

l = Length of key

t = thickness of key

w = width of key

σ_c = crushing stress

→ Torque transmitted by the shaft under shearing force

$$\boxed{T = l \times w \times \tau \times \frac{d}{2}} \quad \text{- - - - (i)}$$

→ Torque transmitted by the shaft under crushing force

$$\boxed{T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}} \quad \text{- - - - (ii)}$$

But, we know that when the shaft is running under the design shear stress then,

$$\boxed{T = \frac{\pi}{16} \times \tau \times d^3} \quad \text{- - - - (iii)}$$

Equating eqn (i) & (ii) respectively with eqn (iii) will provide two values for the length of the key. The larger value of the length of the key is finally considered.

equating eqn (i) & eqn (iii)

$T = l \times w \times \tau \times \frac{d}{2}$

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

equating eqn (ii) & (iii)

$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$

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

consider the larger value of l, l = __

Dt-17-11-2021

Q-1

Design a rectangular keyes for a shaft of 50 mm dia the shearing stress and crushing stress for the key is 42 MPa and 70 MPa respectively.

Data given

Shaft dia = d = 50 mm

τ = 42 MPa = 42 N/mm²

σ_c = 70 MPa = 70 N/mm²

for d = 50 mm

from data hand book.

t = 10 mm

w = 16 mm

Torque transmitted by shaft

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

$= \frac{\pi}{16} \times 42 \times (50)^3 \Rightarrow \boxed{T = 1030835.08 \text{ N.mm}}$

$= 1030835.08$ - - - (i)

~~→~~ According to shearing force

$$T = l \times w \times \tau \times \frac{d}{2}$$

$$\Rightarrow 1030835.08 = l \times 16 \times 42 \times \frac{50}{2} \quad \text{(from eq (i))}$$

$$\Rightarrow l = \frac{1030835.08 \times 2}{16 \times 42 \times 50}$$

$$\Rightarrow \boxed{l = 61.35\ mm} \text{ — (ii)}$$

According to crushing force

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\Rightarrow 1030835.08 = l \times \frac{10}{2} \times 70 \times \frac{50}{2} \quad \text{[from eq (i)]}$$

$$\Rightarrow l = \frac{1030835.08 \times 2 \times 2}{10 \times 70 \times 50}$$

$$\Rightarrow \boxed{l = 117.80\ mm}$$

∴ Taking the larger value of l, $l = 117.80$ mm

<u>Q-2</u> Design a rectangular key for a shaft of diameter 65mm, The ultimate crushing stress and ultimate shearing stress is given 480 mPa and 300 mPa respectively. Take FOS = 6.

According to crushing force

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\Rightarrow 4313799.41 = l \times \frac{12}{2} \times 50 \times \frac{65}{2}$$ [From eqn (i)]

$$\Rightarrow l = \frac{4313799.41 \times 2 \times 2}{12 \times 50 \times 65}$$

$$\Rightarrow l = 442.44$$

∴ Taking the larg

Data given

$d = 65\,mm$

$\tau_{ult} = 300\,MPa$

$\sigma_{c\,ult} = 480\,MPa$

$FOS = 6$

for $d = 65\,mm$, from data hand book

$W = 20\,mm$

$t = 12\,mm$

$$\tau = \frac{\tau_{ult}}{FOS} = \frac{300}{6} = 50\,MPa = 50\,N/mm^2$$

$$\sigma_c = \frac{\sigma_{c\,ult}}{FOS} = \frac{480}{6} = 80\,MPa = 80\,N/mm^2$$

Torque transmitted by shaft

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

$$= \frac{\pi}{16} \times 50 \times (65)^3$$

$$= 2696124.633\,N\text{-}mm$$

According to shearing force

$$T = l \times W \times \tau \times \frac{d}{2}$$

$$\Rightarrow 2696124.633 = l \times 20 \times 50 \times \frac{65}{2}$$

$$\Rightarrow l = \frac{2696124.633 \times 2}{20 \times 50 \times 65}$$

$$\Rightarrow \boxed{l = 82.95\,mm}$$

According to crushing force

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\Rightarrow 2696124.633 = l \times \frac{12}{2} \times 80 \times \frac{65}{2}$$

$$\Rightarrow l = \frac{2696124.66 \times 2 \times 2}{12 \times 80 \times 65}$$

$$\Rightarrow \boxed{l = 172.82\,mm.}$$

∴ Taking the larger value of l,

$l = 172.82\,mm$

* According to maximum shear stress theory :-

1- $\boxed{\tau_{shaft} = \frac{\text{Yield strength of shaft}}{2 \times FOS}}$

2- $\boxed{\tau_{key} = \frac{\text{Yield strength of key}}{2 \times FOS}}$

3- $\boxed{\sigma_{c\,key} = \frac{\text{Yield strength of key}}{2}}$

Dt-18-11-2021

Q A 45mm shaft is having a yield strength of 400MPa. The key is having the yield strength of 340MPa considering the maximum shear stress theory, design the rectangular key. Take FOS = 2

Data given

$d = 45mm$

Yield strength of shaft = 400MPa
= 400 N/mm²

Yield strength of key = 340MPa
= 340 N/mm²

FOS = 2

For d = 45 mm
From data hand book
W = 16
t = 10

$$\tau_{shaft} = \frac{\text{Yield strength of shaft}}{2 \times FOS}$$

$$= \frac{400}{2 \times 2}$$

$$\boxed{\tau_{shaft} = 100\ N/mm^2}$$

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

$$= \frac{\pi}{16} \times 100 \times 45^3 = 1789235.191\ N\,mm$$

$$\boxed{\cancel{T = 157.07\ N.mm}}$$

$$\tau_{key} = \frac{\text{Yield strength of key}}{2 \times FOS}$$

$$= \frac{340}{2 \times 2}$$

$$= 85\ N/mm^2$$

$$\boxed{\tau_{key} = 85\ N/mm^2}$$

$$\sigma_{c\,key} = \frac{\text{Yield strength of key}}{2}$$

$$= \frac{340}{2}$$

$$= 170\ N/mm^2$$

$$\boxed{\sigma_{c\,key} = 170\ N/mm^2}$$

According to shearing force

$$T = lw \times \tau \times \frac{d}{2}$$

$$\Rightarrow 1789235.191 = l \times 16 \times 85 \times \frac{45}{2}$$

$$\cancel{\Rightarrow l = \frac{157.07 \times 2}{16 \times 85 \times 45}}$$

$$\Rightarrow l = \frac{1789235.191 \times 2}{16 \times 85 \times 45}$$

$$\Rightarrow \boxed{l = 58.47\ mm}$$

According to crushing force

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\Rightarrow 1789235.191 = l \times \frac{10}{2} \times 170 \times \frac{45}{2}$$

$$\Rightarrow l = \frac{1789235.191 \times 2 \times 2}{10 \times 170 \times 45} = 93.55mm$$

∴ Taking the larger value of l, $l = 93.5\,mm$

Q.1 Design a rectangular sunk key for a shaft of 50mm dia. The shearing and crushing stresses are 50MPa and 85MPa respectively.

Q-2 Considering maximum shear stress theory design a key for a shaft of 70mm dia having yield strength of ~~350~~ 450 MPa. The key is having the yield strength of 350 MPa. Take FOS = 2

Q-1

Data given

$d = 50\,mm$

$\tau = 50\,MPa = 50\,N/mm^2$

$\sigma_c = 85\,MPa = 85\,N/mm^2$

For $d = 50\,mm$, from data handbook

$w = 16$

$t = 10$

Torque transmitted by the shaft

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

$$= \frac{\pi}{16} \times 50 \times (50)^3$$

$$= 1227184.63\ N.mm$$

$$\boxed{T = 1227184.63\ N.mm}$$

According to shearing action

$$T = l \times w \times \tau \times \frac{d}{2}$$

$$\Rightarrow 1227184.63 = l \times 16 \times 50 \times \frac{50}{2}$$

$$\Rightarrow l = \frac{1227184.63 \times 2}{16 \times 50 \times 50}$$

$$\Rightarrow \boxed{l = 61.35\,mm}$$

According to crushing action

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\Rightarrow 1227184.63 = l \times \frac{10}{2} \times 85 \times \frac{50}{2}$$

$$\Rightarrow l = \frac{1227184.63 \times 2 \times 2}{10 \times 85 \times 50}$$

$$\Rightarrow \boxed{l = 115.49\,mm}$$

∴ Taking the larger value of l, l
$l = 115.49\,mm$

Q-2 Data given

$d = 70\,mm$

Yield strength of shaft $= 450\,MPa = 450\,N/mm^2$

Yield strength of key $= 350\,MPa$
$= 350\,N/mm^2$

$FOS = 2$

For $d = 70\,mm$, from data handbook

$w = 22$

$t = 14$

Considering maximum shear stress theory -

$$\tau_{shaft} = \frac{\text{Yield strength of shaft}}{2 \times FOS}$$

$$= \frac{450}{2 \times 2}$$

$$= 112.5\ N/mm^2$$

$$\boxed{\tau_{shaft} = 112.5\ N/mm^2}$$

$$\tau_{key} = \frac{\text{Yield strength of key}}{2 \times FOS}$$

$$= \frac{350}{2 \times 2}$$

$$= 87.5\ N/mm^2$$

$$\boxed{\tau_{key} = 87.5\ N/mm^2}$$

$$\sigma_{c\,key} = \frac{\text{Yield strength of key}}{2}$$

$$= \frac{350}{2}$$

$$= 175\ N/mm^2$$

$$\boxed{\sigma_{c\,key} = 175\ N/mm^2}$$

Torque transmitted by the shaft

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

$$= \frac{\pi}{16} \times 112.5 \times (70)^3$$

$$= 7576637.908\ N.mm$$

According to shearing force

$$T = l \times w \times \tau \times \frac{d}{2}$$

$$\Rightarrow l = \frac{T \times 2}{w \times \tau \times d}$$

$$\Rightarrow l = \frac{7576637.908 \times 2}{22 \times 87.5 \times 70}$$

$$\Rightarrow \boxed{l = 112.45\ mm}$$

According to crushing force

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\Rightarrow l = \frac{T \times 2 \times 2}{t \times \sigma_c \times d}$$

$$\Rightarrow l = \frac{7576637.908 \times 2 \times 2}{14 \times 175 \times 70}$$

$$\Rightarrow \boxed{l = 176.71\ mm}$$

∴ Taking the larger value of l,

$l = 176.71\ mm$

* Failure of the Keys :-

→ Whenever the keys are subjected to excessive shearing stress and crushing stress, the keys fails.

→ Another reason for the failure of the key is key ways.

Dt- 22-11-2021

* The effect of keyways on a shaft

12

→ The keyways reduces the load carrying capacity of the shaft as some amount of material are removed from the cross-section of the shaft for making of keyways.

→ When keyway are formed, it give rise to various edges on the surface of the shaft, which leads to high stress concentration at the edges.

→ When the material are removed from the shaft surface for producing a keyway, then it gradually leads to reduction in strength of the shaft.

* Shaft Strength factor :- (e)

→ It can be defined as the ratio of strength of the shaft with keyways to the strength of the shaft without key way.

$$e = \frac{\text{Strength of shaft with keyways}}{\text{Strength of shaft without keyways}}$$

→ Shaft strength factor can be calculated by using formula

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

where

e = Shaft Strength factor
d = diameter of shaft
w = width of key
$h = \frac{\text{thickness of key}}{2} = \frac{t}{2}$
= Depth of key.

Q Calculate the shaft strength factor for a shaft having 78mm diameter.

Data given

for d = 78mm
w = 25mm
t = 14mm

$h = \frac{t}{2} = \frac{14}{2} = 7$

$$\therefore e = 1 - 0.2\left(\frac{25}{78}\right) - 1.1\left(\frac{\cancel{14}}{\cancel{2}\,78}\right)$$

$$= 0.83$$

Q-2 A shaft having 40mm dia transmits 15KW at 960 rpm. The shear stress and crushing stress are given 56 MPa and 112 MPa respectively. Design the rectangular sunk key for the shaft and also calculate the shaft strength factor.

Data given

$d = 40mm$

$P = 15 KW = 15\times10^3 W$

$N = 960$ r.p.m

$\tau = 56 MPa = 56 N/mm^2$

$\sigma_c = 112 MPa = 112 N/mm^2$

For d = 40mm, from data given,

$w = 14mm$

$t = 9mm$

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

$\Rightarrow T = \frac{P\times60}{2\pi N}$

$\Rightarrow T = \frac{15\times10^3\times60}{2\times\pi\times960}$

$\Rightarrow T = 149.20$ N.m

$\Rightarrow \boxed{T = 149.20\times10^3 \text{ N.mm}}$

According to shearing action

$T = l\times w\times\tau\times\frac{d}{2}$

$\Rightarrow 149.20\times10^3 = l\times14\times56\times\frac{40}{2}$

$\Rightarrow l = \frac{149.20\times10^3\times2}{14\times56\times40}$

$\Rightarrow \boxed{l = 9.51 mm}$

According to crushing action

$T = l\times\frac{t}{2}\times\sigma_c\times\frac{d}{2}$

$\Rightarrow l = \frac{T\times2\times2}{t\times\sigma_c\times d}$

$\Rightarrow l = \frac{149.20\times10^3\times2\times2}{9\times112\times40}$

$\Rightarrow \boxed{l = 14.80 \text{ mm}}$

∴ Taking the larger value of l, l = 14.80 mm

Shaft strength factor – e

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

$= 1 - 0.2\left(\frac{14}{40}\right) - 1.1\left(\frac{\frac{9}{2}}{40}\right)$

$= 0.80$

$\boxed{e = 0.80}$

COUPLING :- Dt-22-11-2021

*Coupling :- 12

→ It can be defined as a machine elements that is used to connect two or more pieces of shafts.

*Characteristic/Requirements of a good coupling :-

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→ It should have no projecting parts.

→ It should be easy to connect any disconnect.

→ It should keep the shafts in perfect alignment.

→ It should allow the shaft to deliver full power without any hamper.

→ It should not transmitte any kind of shock from one shaft to the another shaft.

~~Dt-23-11-2021~~

Dt-2-12-2021

*Function of coupling :- 15

→ A coupling is used for providing the connection between the two shafts.

→ The coupling is also used to avoid misalignment of shaft.

→ A coupling is used for the reduction of shocks due to loads from one shaft to another shaft.

→ A coupling is ~~used~~ can be used against overloads.

→ Coupling should also provide required strength to the shaft to resist shear stresses.

* Types of coupling / classification of coupling

Coupling can be classified according to the following types :-

1. Rigid coupling
   - i. ~~Split~~ Sleeve or muff coupling
   - ii. Clamp or compression coupling or split coupling.
   - iii – Flange coupling

2 – Flexible coupling
   - i – Bushed-pin type coupling
   - ii – Universal coupling.
   - iii – Oldham coupling.

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1- Sleeve or Muff Coupling :-

→ Muff coupling is a type of rigid coupling.

→ It consist of a sleeve or muff whose inner dia. is equal to the diameter of the shaft.

→ The muff coupling is joined with the shaft with the help of a key. ~~This coupling~~

→ This coupling allows the transmission of shaft smoothly.

Design of Muff coupling

D4 - 3-12-2021

1- **Diameter of the muff :-**

$$D = 2d + 13\,mm$$

where
D = dia of muff
d = dia of shaft

2- **Length of the muff :- (L)**

$$L = 3.5d$$

where
d = dia of shaft

* Design of muff or sleeve :-

→ The torque transmitted by the coupling during rotation can be calculated by using formula

$$\boxed{T = \frac{\pi}{16} \times \tau \times \left[\frac{D^4 - d^4}{D}\right]}$$

where

T = Torque transmitted by the sleeve
τ = Shear stress subjected on the sleeve/muff
D = outer dia of the coupling
d = inner dia of the coupling.

* In design of the sleeve/muff we have to find out the shear stress subjected on the coupling.

*Design of the key:-

i- Length of the key $(l) = \frac{L}{2}$

$L = 3.5d$

ii- By using dia of shaft (d) we can find out, W = width of key, t = Thickness of key

So,

Putting the value of l, W, t, d in the shearing & crushing eqn

$T = l \times w \times \tau \times \frac{d}{2}$ & $T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$

we can find out τ & σ_c for the key material

Q-1

Design a muff coupling which is used to connect 2 shafts transmitting 40kW at 350 r.p.m. The material of the shaft is subjected to shear stress and the material of the keys is subjected to shearing stress and crushing stress of 40MPa and 80 MPa respectively. The material of the muff has a shear stress value of 15 MPa.

Data given

$P = 40\,kW = 40 \times 10^3 W$

$N = 350$ r.p.m.

$\tau_{(k)} = 40 MPa = 40 N/mm^2$

$\sigma_{c(k)} = 80 MPa = 80 N/mm^2$

$\tau_{(m)} = 15 MPa = 15 N/mm^2$

~~Dia of the muff~~

~~$D = 2d + 13$ mm~~

i- Design of shaft:

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

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

$\Rightarrow T = \frac{40 \times 10^3 \times 60}{2\pi \times 350}$

$\Rightarrow T = 1091.34$ N.m

$\Rightarrow \boxed{T = 1091.34 \times 10^3 \text{ N.mm}}$

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

$\Rightarrow 1091.34 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$

$\Rightarrow d = \sqrt[3]{\frac{1091.34 \times 10^3 \times 16}{\pi \times 40}}$

$\Rightarrow \boxed{d = 51.79 \text{ mm}}$

$\Rightarrow \boxed{d \simeq 58 \text{ mm}}$

ii- Dia of the muff

$D = 2d + 13$ mm

$= (2 \times 58) + 13$ mm

$= 129$ mm

$\boxed{D = 129 \text{ mm}}$

iii- Length of muff

$L = 3.5d$

$= 3.5 \times 58$

$= 203$ mm

$\boxed{L = 203 \text{ mm}}$

iv- Design of the sleeve

$T = \frac{\pi}{16} \times \tau \times \left[\frac{D^4 - d^4}{D^4}\right]$

$\Rightarrow 1091.34 \times 10^3 = \frac{\pi}{16} \times \tau \times \left[\frac{(129)^4 - (58)^4}{(129)^4}\right]$

$\Rightarrow 1091.34 \times 10^3 = 404276.68\tau$

$\Rightarrow \tau = \frac{1091.34 \times 10^3}{404276.68}$

$\Rightarrow \boxed{\tau = 2.69 \text{ N/mm}^2}$

v- Design of key

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

$= \frac{203}{2}$

$= 101.5$ mm

From, shaft dia 58 mm

W of key = 18 mm

t of key = 11 mm

According to shearing action

$T = l \times w \times \tau \times \frac{d}{2}$

$\Rightarrow 1091.34 \times 10^3 = 101.5 \times 18 \times \tau \times \frac{58}{2}$

$\Rightarrow \tau = \frac{1091.34 \times 10^3 \times 2}{101.5 \times 18 \times 58}$

$\Rightarrow \boxed{\tau = 20.59 \text{ N/mm}^2}$

According to crushing action

$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$

$\Rightarrow \sigma_c = \frac{T \times 2 \times 2}{l \times t \times d}$

$\Rightarrow \sigma_c = \frac{1091.34 \times 10^3 \times 4}{101.5 \times 11 \times 58}$

$\Rightarrow \boxed{\sigma_c = 67.41 \text{ N/mm}^2}$

Q-2 Design a muff coupling connecting two shafts and transmitting 30 kW at 250 r.p.m. The shear stress for the shaft is 60 MPa. State whether designing of this coupling is safe or not. Consider the ultimate values are $\tau_{(m)} = 30$ MPa, $\tau_{(k)} = 65$ MPa, $\sigma_{ck} = 120$ MPa

Data given

$P = 30\,kW = 30\times10^3\,W$

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

$\tau = 60\,MPa = 60\,N/mm^2$

$\tau(m) = 30\,MPa$

$\tau(k) = 65\,MPa$

$\sigma_c(k) = 120\,MPa$

i. Design of Shaft

$$T = \frac{P\times60}{2\pi N}$$

$$= \frac{30\times10^3\times60}{2\times\pi\times250}$$

$$= 1145.91\,N.m$$

$$= 1145.91\times10^3\,N.mm$$

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

$$\Rightarrow d = \sqrt[3]{\frac{T\times16}{\pi\times\tau}}$$

$$\Rightarrow d = \sqrt[3]{\frac{1145.91\times10^3\times16}{\pi\times60}}$$

$$\Rightarrow d = 45.98\,mm$$

$$\Rightarrow \boxed{d \simeq 50\,mm}$$

For 50 mm dia, from datahandbook

$w = 16\,mm$ and $t = 10\,mm$

ii. Dia of the muff

$$D = 2d + 13\,mm$$

$$= 2\times50 + 13\,mm$$

$$= 113\,mm$$

$$\boxed{D = 113\,mm}$$

iii. Length of muff :-

$$L = 3.5d$$

$$= 3.5\times50$$

$$= 175\,mm$$

$$\boxed{L = 175\,mm}$$

iv. Design of Sleeve :-

$$T = \frac{\pi}{16}\times\tau\times\left[\frac{D^4 - d^4}{D}\right]$$

$$\Rightarrow T = \frac{\pi}{16}\times\tau\times\left[\frac{113^4 - 50^4}{113}\right]$$

$$\Rightarrow T = 272452.12\,\tau$$

$$\Rightarrow \tau = \frac{1145.91\times10^3}{272452.12}$$

$$\Rightarrow \boxed{\tau = 4.20\,N/mm^2}$$

v. Design of key

$$l = \frac{L}{2} = \frac{175}{2} = 87.5\,mm$$

According to Shearing action

$$T = l \times w \times \tau \times \frac{d}{2}$$

$$\Rightarrow \tau = \frac{T \times 2}{l \times w \times d}$$

$$\Rightarrow \tau = \frac{1145.91 \times 10^3 \times 2}{87.5 \times 16 \times 50}$$

$$\Rightarrow \boxed{\tau = 32.75\ N/mm^2}$$

According to crushing action

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\Rightarrow \sigma_c = \frac{T \times \cancel{2}2}{l \times t \times d}$$

$$\Rightarrow \sigma_c = \frac{1145.91 \times 10^3 \times 4}{87.5 \times 10 \times 50}$$

$$\Rightarrow \boxed{\sigma_c = 104.76\ N/mm^2}$$

~~Since the given values~~.

Dt 6-12-2021.

* Split Muff/clamp/ compression coupling :-

→ Split muff coupling is the type of rigid coupling, this coupling is known as split muff coupling because here the muff is divided into two parts.

→ The inner dia of the muff is equal to the dia of the shaft. one halves of the muff is fixed from it below and the other halves is fitted from the above. both the halves are held together by means of ~~no~~ mu bolts.

→ The no. bolts can be two, four or six.

→ The measure advantage of this coupling is that the position of the shaft is not required to be changed while assembling and de-assembling of the coupling.

→ This coupling is generally used to transmitted high or moderate speed.

* Design of split muff coupling :-

① Design of shaft :-

Power transmitted by the shaft

$$P = \frac{2\pi NT}{\cancel{60}60}$$

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

So, hence dia of shaft

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

$$\Rightarrow \boxed{d = \sqrt[3]{\frac{T \times 16}{\pi \times \tau}}}$$

2- Dia of muff (D):-

$$\boxed{D = 2d + 13 \text{ mm}}$$

where d = dia of shaft

3- Length of the muff (L):-

$$\boxed{L = 3.5d}$$

4- Design of muff:-

Torque transmitted by the muff

$$T = \frac{\pi}{16} \times \tau_m \times \left[\frac{D^4 - d^4}{D}\right]$$

$$\Rightarrow \boxed{\tau_m = \frac{16 \times T \times D}{\pi (D^4 - d^4)}}$$

5- Design of key:-

Length of the key (l)

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

where, l = length of ~~muff~~ key

L = Length of muff

→ From the dia of the shaft we can calculate the width of the key (w) & thickness of the key (t)

→ From the shearing eqn & crushing eqn we can calculate the values of shear stress & crushing stress respectively.

6- Design of bolts:-

$$T = \frac{\pi^2}{16} \times \mu (d_0)^2 \sigma_t \times n \times d$$

where

T = Torque transmitted by shaft

μ = Co-efficient of friction

σ_t = Tensile stress

d = diameter of shaft

d_0 = core diameter of bolt

n = no. of bolts.

From the above formula we can calculate the core dia of the bolt.

$$T = \frac{\pi^2}{16} \times \mu (d_0)^2 \sigma_t \times n \times d$$

$$\Rightarrow \boxed{d_0 = \sqrt{\frac{T \times 16}{\pi^2 \times \mu \times \sigma_t \times n \times d}}}$$

Dt. 7-12-2021

Q-

Design a clamp coupling transmitting 30kW at 100 r.p.m. The shear stress subjected is 40 MPa. The no. of bolts fixed on the coupling is 6. The tensile stress is found to be 70 MPa. Take the coefficient of friction between the shaft and the coupling is 0.3.

Data given :-

$P = 30\ kW = 30\times10^3\ W$

$N = 100\ r.p.m$

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

$n = 6$

$\mu = 0.3$

$\sigma_t = 70\ MPa = 70\ N/mm^2$

1 - Design of Shaft

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

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

$$= \frac{60\times 30\times10^3}{2\times\pi\times100}$$

$$= 2864.78\ N.m$$

$$\Rightarrow \boxed{T = 2864.78\times10^3\ N.mm}$$

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

$$\Rightarrow d = \sqrt[3]{\frac{T\times16}{\pi\times\tau}}$$

$$\Rightarrow d = \sqrt[3]{\frac{2864.78\times10^3\times16}{\pi\times40}}$$

$$\Rightarrow d = 71.44\ mm$$

$$\Rightarrow \boxed{d \simeq 75\ mm}$$

2 - Dia of muff :-

$$D = 2d+13$$
$$= 2\times75+13$$
$$= 163\ mm$$

$$\boxed{D = 163\ mm}$$

3 - Length of muff

$$L = 3.5d$$
$$= 3.5\times75$$
$$= 262.5\ mm$$

$$\boxed{L = 262.5\ mm}$$

4 - Design of muff

$$T = \frac{\pi}{16}\times\tau_m\times\left[\frac{D^4-d^4}{D}\right]$$

$$\Rightarrow T = \frac{\pi}{16}\times\tau_m\times\left[\frac{163^4-75^4}{163}\right]$$

$$\Rightarrow T = 812225.93\ \tau_m$$

$$\Rightarrow \tau_m = \frac{2864.78\times10^3}{812225.93}$$

$$\Rightarrow \boxed{\tau_m = 3.52\ N/mm^2}$$

5. Length of key

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

$= \frac{262.5}{2}$

$\Rightarrow \boxed{l = 131.25\,mm}$

For $d = 75\,mm$, from data handbook.

$w = 22\,mm$

$t = 14\,mm$

According to shearing action

$T = l \times w \times \tau \times \frac{d}{2}$

$\Rightarrow \tau = \frac{T \times 2}{l \times w \times d}$

$\Rightarrow \tau = \frac{2864.78 \times 10^3 \times 2}{131.25 \times 22 \times 75}$

$\Rightarrow \boxed{\tau = 26.45\,N/mm^2}$

According to crushing action

$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$

$\Rightarrow \sigma_c = \frac{T \times 2 \times 2}{l \times t \times d}$

$\Rightarrow \sigma_c = \frac{2864.78 \times 10^3 \times 4}{131.25 \times 14 \times 75}$

$\Rightarrow \boxed{\sigma_c = 83.15\,N/mm^2}$

6. Design of bolts :-

$T = \frac{\pi^2}{16} \times \mu \times d_0^2 \times \sigma_t \times n \times d$

$\Rightarrow d_0 = \sqrt{\frac{T \times 16}{\pi^2 \times \mu \times \sigma_t \times n \times d}}$

$\Rightarrow d_0 = \sqrt{\frac{2864.78 \times 10^3 \times 16}{\pi^2 \times 0.3 \times 70 \times 6 \times 75}}$

$\Rightarrow \boxed{d_0 = 22.16\,mm}$

Q. Design a clamp coupling transmitting 20KW at 200 r.p.m. The shear stress is given 45 MPa. The tensile stress subjected on the coupling is 75 MPa. The no. of bolts attached are 6. Take $\mu = 0.3$.

Data given :-

$P = 20\,KW = 20 \times 10^3\,W$

$N = 200\,r.p.m$

$\tau = 45\,MPa = 45\,N/mm^2$

$\sigma_t = 75\,MPa = 75\,N/mm^2$

$n = 6$

$\mu = 0.3$

1- Design of shaft

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

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

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

$\Rightarrow T = 954.92\ N.m$

$\Rightarrow \boxed{T = 954.92 \times 10^3\ N.mm}$

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

$\Rightarrow d = \sqrt[3]{\frac{T \times 16}{\pi \times \tau}}$

$\Rightarrow d = \sqrt[3]{\frac{954.92 \times 10^3 \times 16}{\pi \times 45}}$

$\Rightarrow d = 48.66\ mm$

$\boxed{d \simeq 50\ mm}$

2- Dia of Muff

$D = 2d + 13\ mm$

$= 2 \times 50 + 13$

$= 113\ mm$

$\boxed{D = 113\ mm}$

3- Length of muff

$L = 3.5d$

$= 3.5 \times 50$

$= 175\ mm$

$\boxed{L = 175\ mm}$

4- Design of muff:

$T = \frac{\pi}{16} \times \tau_m \times \left[\frac{D^4 - d^4}{D}\right]$

$\Rightarrow T = \frac{\pi}{16} \times \tau_m \times \left[\frac{113^4 - 50^4}{50}\right]$

$\Rightarrow T = 272452.12\ \tau_m$

$\Rightarrow \tau_m = \frac{954.92 \times 10^3}{272452.12}$

$\Rightarrow \boxed{\tau_m = 3.50\ N/mm^2}$

5- Length of key:

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

$= \frac{175}{2}$

$= 87.5\ mm$

$\boxed{l = 87.5\ mm}$

for $d = 50\ mm$, from data handbook

$w = 16\ mm$

$t = 10\ mm$

According to Shearing force

$T = l \times w \times \tau \times \frac{d}{2}$

$\Rightarrow \tau = \frac{T \times 2}{l \times w \times d}$

$\Rightarrow \tau = \frac{954.92 \times 10^3 \times 2}{87.5 \times 16 \times 50}$

$\Rightarrow \boxed{\tau = 27.28\ N/mm^2}$

According to Crushing action

$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$

$\Rightarrow \sigma_c = \frac{T \times 2 \times 2}{l \times t \times d}$

$\Rightarrow \sigma_c = \frac{954.92 \times 10^3 \times 4}{87.5 \times 10 \times 50}$

$\Rightarrow \boxed{\sigma_c = 87.30 \text{ N/mm}^2}$

Design of bolts:-

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

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

$$\Rightarrow d_o = \sqrt{\frac{954.92 \times 10^3 \times 16}{\pi^2 \times 0.3 \times 75 \times 6 \times 50}}$$

$$\Rightarrow \boxed{d_o = 15.14 \text{ mm}}$$

Chapter-4

Springs Dt 09-12-2021

* Spring

→ Spring can be defined as an elastic body which function is to distort (deform) when load is applied and regain its original configuration, when the loads are removed.

* Functions of Spring :-

→ To store energy.

→ To measure forces, such as a spring balances and engine indicators.

→ To maintain or control the motion between the cam and the follower.

→ To apply forces in brakes, clutches etc

→ To act as a cushion to absorbe externally produced shocks. (suspension system)

* Types of Spring :-

Springs can be classified according to the following ways. They are

1- Helical springs

2- Conical springs

3- Torsion springs

4- Disc springs.

5- Special purpose springs

6- Laminated or leaf springs.

1. Helical Springs:-

-> The springs which are bent into the form of a helix are known as Helical springs.

-> Helical springs can be classified into two types. They are

i- **Compression Helical Spring:-**

-> The Helical spring which are bent into the form of a helix and are subjected to compressive force are known as compression Helical spring.

ii- **Tension Helical spring**

-> The Helical spring which are subjected tensile force and bent into the form of a Helix are known as Tension Helical spring.

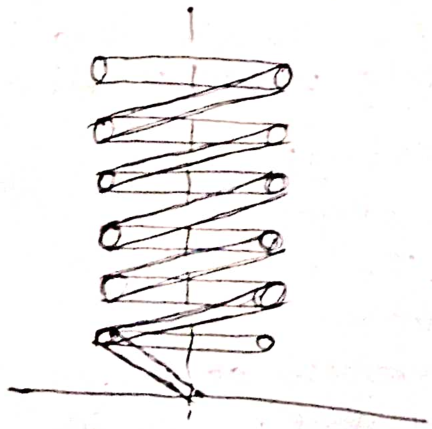


Fig- compression Helical spring.

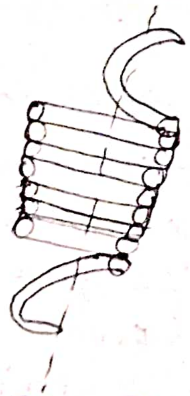


Fig- Tension helical spring

* **composition of compression Helical spring:-**

0.60 to 0.70% - carbon (strength)

0.60 to 1% - manganese (elastic property)

* **Terms Used in case of compression Helical springs:-**

i- **Solid Length:-**

-> The length of the compression spring when it comes in contact with each other is known as solid length.

-> In other words the length of the spring in fully loaded conditions is solid length.

-> solid length = $\boxed{l_s = n' d}$

where,

l_s = solid length

n' = no. of coils

d = diameter of wire.

ii- **Free Length:-**

-> The length of the compression spring when it is in unloaded conditions is known as free length.

iii- **Spring Index:- (C)**

-> It can be defined as the ratio of dia of the coil to the dia of the wire.

-> $\boxed{C = \frac{D}{d}}$

where

C = spring index

D = dia of coil

d = dia of wire

iv- Spring Rate / Stiffness (K)

→ It can be defined as the ratio of loads applied to the deflection produced in the springs.

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

Where

K = Spring rate or Stiffness

W = Load applied

δ = deflection produced..

v- Pitch -: (imp)

→ The Pitch of a compression spring can be defined as the total distance between one point of a helix to the same point present on another helix.

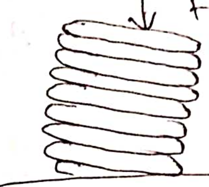
i- $\boxed{\text{Pitch} = \frac{\text{Free length}}{n'-1}}$ $(n' = n+2)$

ii- $\boxed{\text{Pitch} = \frac{n'd + \delta + 0.15\delta}{n'-1}}$

iii- $\boxed{\text{Free length} = n'd + \delta + 0.15\delta}$

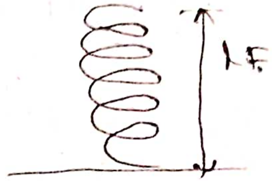
* What is the relationship between Pitch and Free length.

Full force or Loaded

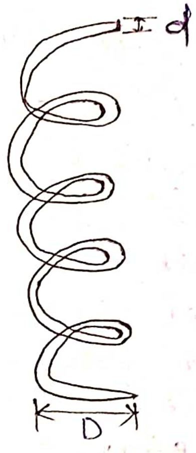


(Solid length)

No loads



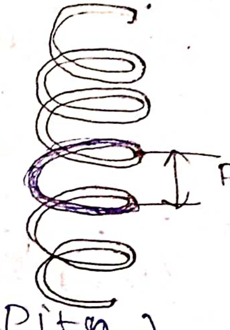
(Free length)



(Spring index)

Outer dia of Spring

$\boxed{D_0 = D + d}$.



(Pitch)

* Shear stress induced in compression Helical Spring -:

There are two ways in which the shear stress induced in the spring can be calculated.

i- Neglecting the curvature Effect

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

Where $K_s = 1 + \frac{1}{2C}$ = Spring constant = Stress concentration factor.

τ = Shear Stress on Spring

W = Load applied.

D = Dia of coil , d = dia of wire.

2- Considering the curvature effect

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

Where

τ = Shear stress on spring

K = Wahl's stress factor

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

c = spring index.

W = load applied.

D = dia of coil

d = dia of wire.

* What is Wahl's stress factor.

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

* Deflection in compression Helical spring:-

The deflection can be calculated by the formula

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

$$= \frac{8Wn}{Gd} \times \frac{D^3}{d^3}$$

$$= \frac{8Wnc^3}{Gd} \quad (\because c = \frac{D}{d})$$

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

Where,

δ = Deflection in springs

W = load applied

n = no. of coils/turns

D = dia of coil

d = dia of wire

C = Spring constant ($\frac{D}{d}$)

G = Modulus of rigidity

Q A compression helical spring having the following data.

Mean dia of coil = 50mm

wire dia = 5mm

No. of turns = 20

It is subjected to a load of 500N. Calculate the shear stress induced in the spring material neglecting the effect of curvature.

Data given:-

Dia of coil = D = 50 mm

wire dia = d = 5 mm

No of turns = n = 20

W = 500N

$\therefore c = \frac{D}{d}$

$= \frac{50}{5}$

$\boxed{c = 10}$

$\therefore K = 1 + \frac{1}{2c}$

$= 1 + \frac{1}{2 \times 10}$

$= 1 + \frac{1}{20}$

$= 1.05$

$\boxed{K = 1.05}$

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

$= 1.05 \times \frac{8 \times 500 \times 50}{\pi \times (5)^3}$

$= 534.76 \ N/mm^2$

$\boxed{\tau = 534.76 \ N/mm^2}$

* A compression Helical spring having mean dia of the coil 60mm, wire dia 6mm, number of turn 25 is subjected to a load of 550N. Calculate the shear stress induced in the spring considering the curvature effect. Take G = 84GPa and also calculate the deflection ~~Deflaction~~ produce in the spring.

Data given

$D = 60 \ mm$

$d = 6 \ mm$

$n = 25$

$W = 550 \ N$

$G = 84 GPa = 84 \times 10^9 \times 10^{-6} = 84 \times 10^3 \ N/mm^2$

$\therefore C = \frac{D}{d} = \frac{60}{6} = 10 \quad \boxed{C = 10}$

$K = \frac{4C-1}{4C-4} + \frac{0.4615}{C}$

$= \frac{4 \times 10 - 1}{4 \times 10 - 4} + \frac{0.615}{10}$

$= 1.14$

$\boxed{K = 1.14}$

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

$= 1.14 \times \frac{8 \times 550 \times 60}{\pi \times (6)^3}$

$\cancel{= 443.51 \ N/mm^2} = 443.51 \ N/mm^2$

$\boxed{\tau = 443.51 \ N/mm^2}$

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

$= \frac{8 \times 550 \times (60)^3 \times 25}{84 \times 10^3 \times (6)^4}$

$= 218.25 \ mm$

$\boxed{\delta = 218.25 \ mm}$

Q- A Helical spring having outer dia = 70 mm and wire dia 5 mm is subjected to a load of 800 N. Calculate the shear stress inducing in the spring considering the effect of curvature.

Data given

$D_o = 70\,mm$

$d = 5\,mm$

$W = 800\,N$

$D_o = D + d$

$\Rightarrow 70 = D + 5$

$\Rightarrow 70 - 5 = D$

$\Rightarrow \boxed{D = 65\,mm}$

$C = \frac{D}{d} = \frac{65}{5} = 13$

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

$= \frac{4\times13-1}{4\times13-4} + \frac{0.615}{C}$

$= 1.109$

$\boxed{K = 1.109}$

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

$= 1.109 \times \frac{8\times800\times65}{\pi\times(5)^3}$

$= 1174.802\,N/mm^2$

$\boxed{\tau = 1174.802\,N/mm^2}$

Q- A Helical spring having 6 mm wire dia and 75 mm outside dia is subjected to a shear stress of 350 MPa. Calculate the load in both the cases neglecting and considering the effect of curvature. Also calculate the deflection per turn in both the cases. Take G = 84 GPa.

Data given.

$D_o = 75\,mm$

$d = 6\,mm$

$\tau = 350\,MPa = 350\,N/mm^2$

$G = 84\,GPa$

$= 84\times10^9\times10^{-6}$

$= 84\times10^3\,N/mm^2$

$n = 1$

Neglecting the curvature effect

$D_o = D + d$

$\Rightarrow 75 = D + 6$

$\Rightarrow D = 69\,mm$

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

$\boxed{C = 11.5}$

$K = 1 + \frac{1}{2C}$

$= 1 + \frac{1}{2 \times 11.5}$

$= 1.043$

$\boxed{K = 1.043}$

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

$\Rightarrow 350 = 1.043 \times \frac{8 \times W \times 69}{\pi \times (6)^3}$

$\Rightarrow W = \frac{350 \times \pi \times (6)^3}{8 \times 1.043 \times 69}$

$\Rightarrow \boxed{W = 412.52 \text{ N}}$

$\delta = \frac{8WD^3 n}{G d^4}$

$= \frac{8 \times 412.52 \times (69)^3 \times 1}{84 \times 10^3 \times (6)^4}$

$= 9.95 \text{ mm}$

$\boxed{\delta = 9.95 \text{ mm}}$

considering curvature effect

$D_o = D + d$

$\Rightarrow 75 = D + 6$

$\Rightarrow D = 69 \text{ mm}$

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

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

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

$= 1.12$

$\boxed{K = 1.12}$

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

$\Rightarrow W = \frac{\tau \times \pi \times d^3}{K \times 8 \times D}$

$\Rightarrow W = \frac{350 \times \pi \times (6)^3}{1.12 \times 8 \times 69}$

$\Rightarrow \boxed{W = 384.16 \text{ N}}$

$\delta = \frac{8WnC^3}{Gd}$

$= \frac{8 \times 384.16 \times 1 \times (11.5)^3}{84 \times 10^3 \times 6}$

$= 9.27 \text{ mm}$

$\boxed{\delta = 9.27 \text{ mm}}$

Q- Design a spring to measure 1000 N the spring is subjected to a deflection of 80mm. Take G= 84GPa. Also calculate the shear stress induced in the spring if the wire diameter is given 4mm. The no. of turns is found to be 30

Data given

$W = 1000N$

$\delta = 80 mm$

$G = 84 GPa = 84 \times 10^3 N/mm^2$

$n = 30$

$d = 4mm$

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

$$\Rightarrow 80 = \frac{8 \times 1000 \times D^3 \times 30}{84 \times 10^3 \times (4)^4}$$

$$\Rightarrow D = \sqrt[3]{\frac{80 \times 84 \times 10^3 \times (4)^4}{8 \times 1000 \times 30}}$$

$$\Rightarrow \boxed{D = 19.28 mm}$$

$$\therefore C = \frac{D}{d}$$

$$= \frac{19.28}{4}$$

$$= 4.82$$

i- Neglecting curvature effect

$$K = 1 + \frac{1}{2C}$$

$$= 1 + \frac{1}{2 \times 482}$$

$$= 1.103$$

$$\therefore \tau = K_e \times \frac{8WD}{\pi d^3}$$

$$= 1.103 \times \frac{8 \times 1000 \times 19.28}{\pi \times (4)^3}$$

$$= 846.14 N/mm^2$$

$$\boxed{\tau = 846.14 N/mm^2}$$

ii- Considering curvature effect

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

$$= \frac{(4 \times 4.82)-1}{(4 \times 4.82)-4} + \frac{0.615}{4.82}$$

$$= 1.32$$

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

$$= 1.32 \times \frac{8 \times 1000 \times 19.28}{\pi \times (4)^3}$$

$$= 1012.607 N/mm^2$$

$$\boxed{\tau = 1012.607 N/mm^2}$$

Q. Imp A spring is having wire dia of 2mm, spring index (C) = 6, no. of turns = 18 is subjected to a load of 30 N.
Calculate, i- Shear stress induced in the spring considering the effect of curvature
ii- Calculate deflection (spring rate) (δ)
iii- Free length of spring
iv- Pitch of the spring.
Take G = 84 GPa.

Data given

$d = 2\,mm$

$C = 6$

$n = 18$

$W = 30\,N$

$G = 84\,GPa = 84 \times 10^3\,N/mm^2$

$\therefore C = \frac{D}{d}$

$\Rightarrow 6 = \frac{D}{2}$

$\Rightarrow \boxed{D = 12}$

$n = 18$

$\Rightarrow n' = n + 2$

$= 18 + 2$

$= 20$ $\boxed{n' = 20}$

i- Shear stress induced in the spring considering the effect of curvature :-

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

$$= \frac{4\times6-1}{4\times6-4} + \frac{0.615}{6}$$

$$= 1.25$$

$\boxed{K = 1.25}$

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

$$= 1.25 \times \frac{8\times30\times12}{\pi\times(2)^3}$$

$$= 143.23\,N/mm^2$$

$\boxed{\tau = 143.23\,N/mm^2}$

ii- $$\delta = \frac{8WD^3n}{Gd^4}$$

$$= \frac{8\times30\times(12)^3\times18}{84\times10^3\times(2)^4}$$

$$= 5.55\,mm$$

$\boxed{\delta = 5.55\,mm}$

iii- Free length of spring (l_f)

Free length $= n'd + \delta + 0.15\delta$

$= 20\times2 + 5.55 + 0.15\times5.55$

$\boxed{\text{Free length} = 46.38\,mm}$

iv- Pitch of the spring

$$\text{Pitch} = \frac{\text{Free length}}{n'-1}$$

$$= \frac{46.38}{20-1}$$

$$= \cancel{2.08}\ 2.44$$

~~Pitch = 2.08~~

$\boxed{\text{Pitch} = 2.44\,mm}$

Q- Design a helical compression spring for a maximum load of 1000N having deflection 25mm and spring index as 5. The maximum shear stress subjected in the spring 420mpa and modulus of rigidity = 84 kN/mm². Take wahl factor for calculation. n = 14

Data given

$W = 1000N$

$\delta = 25mm$

$c = 5$

$\tau = 420 \, mpa = 420 \, N/mm^2$

$G = 84 \, kN/mm^2 = 84 \times 10^3 \, N/mm^2$

$n = 14, \; n' = 2+n = 14+2 = 16$

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

$$\Rightarrow 25 = \frac{8 \times 1000 \times (5)^3 \times 14}{84 \times 10^3 \times d}$$

$$\Rightarrow d = \frac{8 \times 1000 \times (5)^3 \times 14}{84 \times 10^3 \times 25}$$

$$\Rightarrow \boxed{d = 6.66}$$

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

$$\Rightarrow 5 = \frac{D}{d}$$

$$\Rightarrow D = 5 \times 6.66$$

$$\Rightarrow \boxed{D = 33.3mm}$$

Free length $= l_f = n'd + \delta + 0.15\delta$

$= 16 \times 6.66 + 25 + 0.15 \times 25$

$= 135.31 \, mm$

$$\boxed{l_f = 135.31}$$

$$\text{Pitch} = \frac{\text{free length}}{n' - 1}$$

$$= \frac{135.31}{16 - 1}$$

$$= 9.02 \, mm$$

$$\boxed{\text{Pitch} = 9.02 \, mm}$$

Q- Design a compression helical spring for a load of 800N having deflection of 30mm. The spring index is 5. The maximum shear stress is 450 mpa and modulus of rigidity is 84 KN/mm². Take wahl factor into consideration.

Data given

$W = 800N$

$C = 5$

$\delta = 30mm$

$\tau = 450 mpa = 450 N/mm^2$

$G = 84 KN/mm^2 = 84 \times 10^3 N/mm^2$

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

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

$$= 1.31$$

$$\boxed{K = 1.31}$$

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

$$\Rightarrow \tau = K \times \frac{8W}{\pi d^2} \times \frac{D}{d}$$

$$\Rightarrow \tau = K \times \frac{8WC}{\pi d^2}$$

$$\Rightarrow d = \sqrt{K \times \frac{8WC}{\pi \times \tau}}$$

$$\Rightarrow d = \sqrt{1.31 \times \frac{8\times800\times5}{\pi\times450}}$$

$$\Rightarrow d = 5.44$$

$$\boxed{d = 5.44}$$

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

$$\Rightarrow D = C \times d$$
$$= 5 \times 5.44$$
$$= 27.2\ mm$$

$$\boxed{D = 27.2\ mm}$$

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

$$\Rightarrow n = \frac{\delta G d^4}{8WD^3}$$

$$\Rightarrow n = \frac{30 \times 84 \times 10^3 \times (5.44)^4}{8 \times 800 \times (27.2)^3}$$

$$\Rightarrow n = 17.13$$

$$\boxed{n \simeq 18}$$

$$\therefore n' = 2 + n$$
$$= 2 + 18$$
$$= 20$$

$$\boxed{n' = 20}$$

$$l_f = n'd + \delta + 0.15\delta$$
$$= 20 \times 5.44 \times 30 + 0.15 \times 30$$

$$\boxed{l_f = 143.3\ mm}$$

$$\text{Pitch} = \frac{l_f}{n'-1}$$
$$= \frac{143.3}{20-1}$$
$$= 7.54\ mm$$

$$\boxed{\text{Pitch} = 7.54\ mm}$$

Q- Design a closed helical compression spring having coil dia 75mm, wire dia 5mm, no. of turns = 16, load applied = 1000N, modulus of rigidity = 84 GPa, consider wahl factor for calculation.

Data given

$D = 75\ mm$

$d = 5\ mm$

$n = 16$

$W = 1000\ N$

$G = 84\ GPa$
$= 84 \times 10^3\ N/mm^2$

$$\therefore C = \frac{D}{d} = \frac{75}{5} = 15$$

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$
$$= \frac{4 \times 15 - 1}{4 \times 15 - 4} + \frac{0.615}{15}$$
$$= 1.094$$

$$\text{i-}\ \tau = K \times \frac{8WD}{\pi d^3}$$
$$= 1.094 \times \frac{8 \times 1000 \times 75}{\pi \times (5)^3}$$
$$= 1671.508\ N/mm^2$$

$$\boxed{\tau = 1671.508\ N/mm^2}$$

ii- $\delta = \frac{8WD^3}{Gd^4}$

$= \frac{8\times1000\times(75)^3}{84\times10^3\times(5)^4}$

$= 1028.57$ mm

$\boxed{\delta = 1028.57 \text{ mm}}$

iii- $l_f = n'd + \delta + 0.15\delta$

$= (2+16)\times5 + 1028.57 + 0.15\times1028.57$

$= 1272.85$ mm

$\boxed{l_f = 1272.85 \text{ mm}}$

iv- $\text{Pitch} = \frac{l_f}{n'-1}$

$= \frac{1272.85}{18-1}$

$= 74.87$ mm

$\boxed{\text{Pitch} = 74.87 \text{ mm}}$

*Surge in Springs L5 — Dt 23-12-2024

→ When helical springs are subjected to any types of load, then the coil of the spring which is directly in contact with the load suffers a high tension as well as high stress constration, which result in deflection ~~to~~ that produce in the spring.

→ During the process of loading the coils which are in direct contact in load ~~which one~~ takes the whole deflection & the deflection gradually propagate to the other coils of the spring. This result in the large deflection produced. The large deflection again leads to high ~~str~~ stress constration among the coil of the helical spring as the process of load application time becomes equal to the time required for waves deflection to travel. This leads to failure in springs which is known as surge in spring.

→ The ways by using which the surge in springs can be eliminated (avoided) are:

i- By using friction Dampers which will stop the wave propagation in spring.

ii - By using springs having high natural frequency.

iii- By using springs having different pitch near the ends of the coil.

DT- 23-12-2021

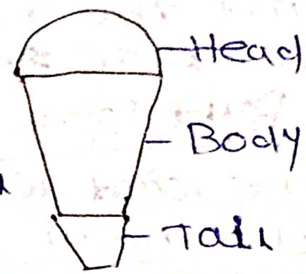
TYPES OF JOINTS

→ Joints are formed in materials for making different pieces a single unit.

→ The joints that are formed are generally temporary joint or permanent joint.

→ Welding joint and riveted joint are examples of permanent joints and screw joints ~~are~~ is example of temporary joint.

* Riveted ~~Joint~~ :-



→ A rivet is a piece of fastening material that is used to form a joint and a rivet is having a cylindrical body with a tail and head integral to it. (in addition)

* Riveted Joint

→ It can be defined as the joint which is form with the help of rivetes.

* Classification of Riveted Joints.

Riveted joint can be classified into two types
1. Lap joint
2. Butt Joint
   i- Single Strap Butt Joint
   ii- Double Strap Butt Joint.

1- Lap Joint-:

→ The join which is formed, where one plate overlaps the other plates and the two plates are riveted together is known as lap joint.

2- Butt Joint

The joint, which is formed when the main plates are ~~kepeed~~ kept in alignment with each other and the cover plate are placed on one or both the sides of the main plates and are riveted together is known as Butt Joint.

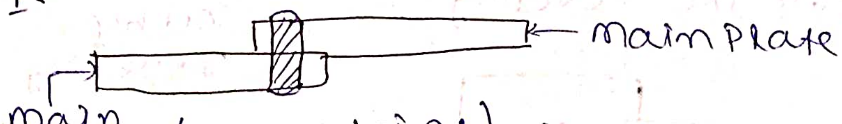
i- Single strap Butt Joint-:

The butt joint formed with the alignment of two main plates and one cover plates placed on any or sides of a main plates and reveting together is known as single strap Butt Joint.

ii- Double strap Butt Joint

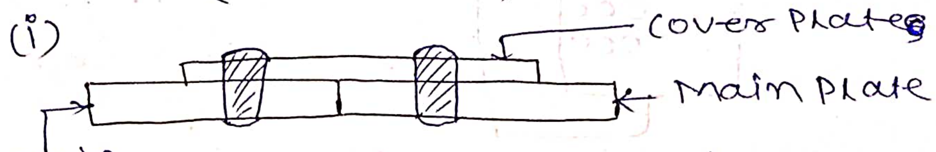
The butt joint formed with the alignment of two main plates and placing two cover plates on both the sides of the main plates and reveting them together is known as double strap Butt Joint.

1-



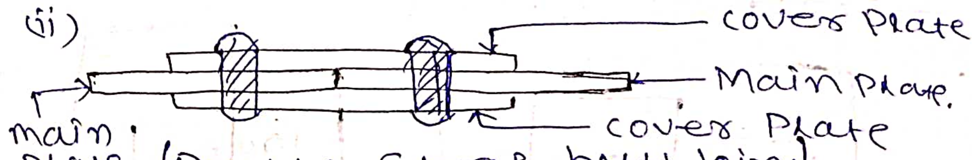
(Lap Joint.)

(i)



(Single strap butt Joint)

(ii)

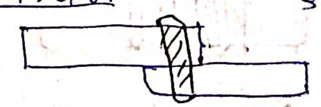


(Double strap butt Joint)

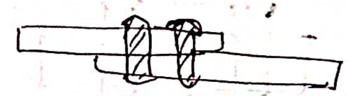
* Types of Reveting-:

i- According to no. of rows-:

Lap Joint



Single riveting

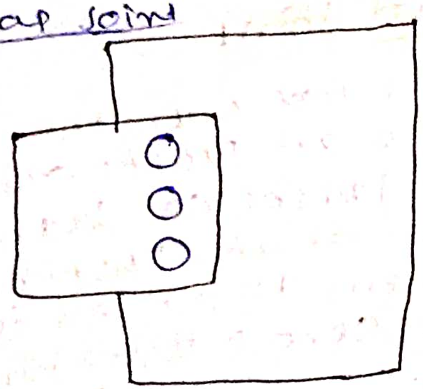


Double riveting.

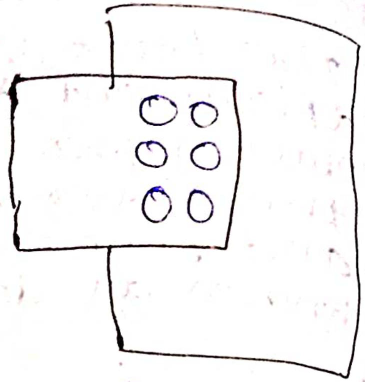
(side view) Triple riveting.

TOP view

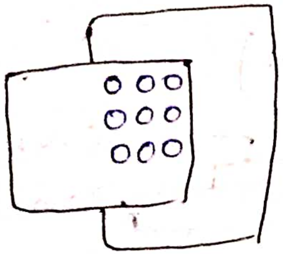
Lap joint



Single reveting

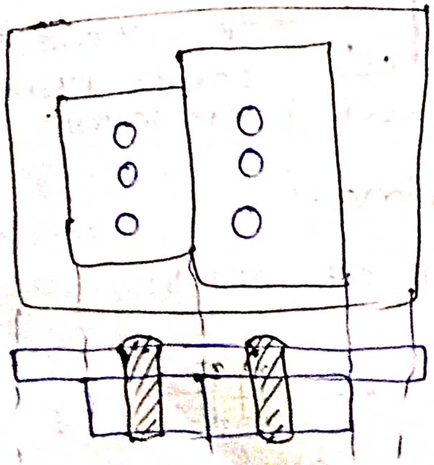


Double reveting

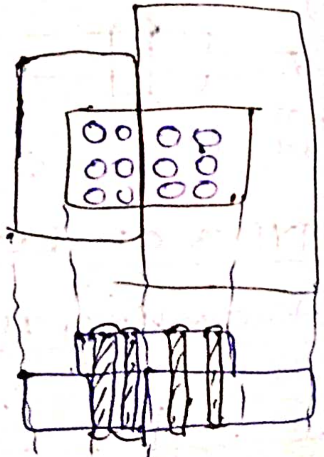


Triple reveting

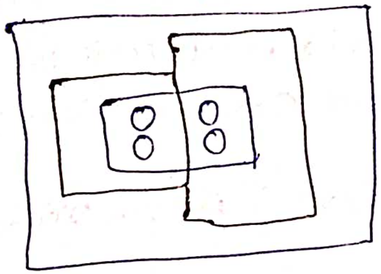
Butt joint



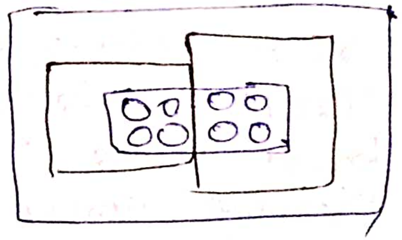
Single strap
Single riveting



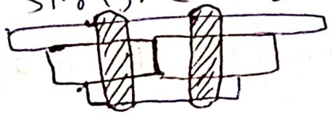
~~Double~~ single strap
double rivet

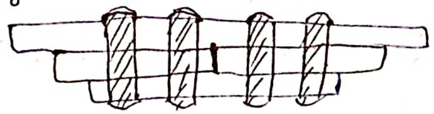


Double strap
single riveting



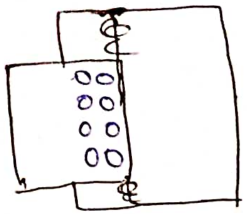
Double strap
double reveting.



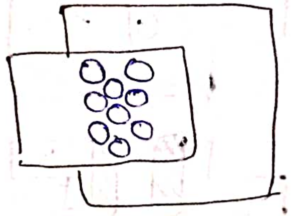


ii- According to Placement of Rivets :-

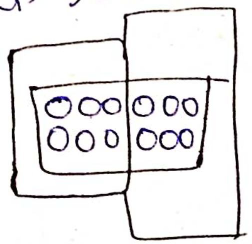
→ Chain riveting
→ Zig-zag riveting



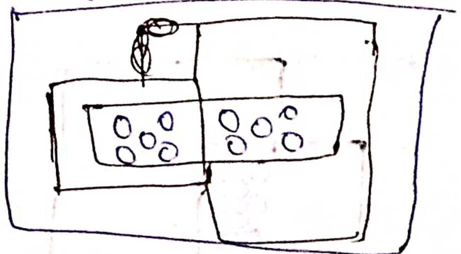
Double chain Riveting
Lap joint



Triple zig-zag Riveting.
Lap joint.



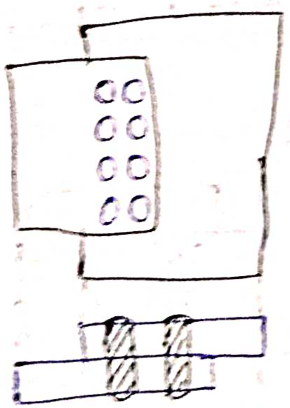
Single strap
~~chai~~ triple
chain riveting.



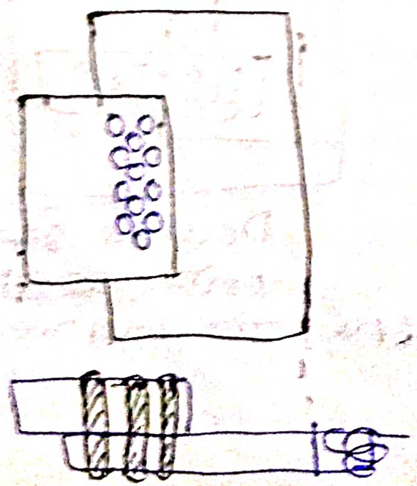
Double strap
triple ~~chain~~
zig-zag reveting

1- Double chain riveted lap joint.
2- Triple zig-zag riveted lap joint.
3- Single strap double chain riveted butt joint.
4- Double strap triple chain riveted butt joint.
5- Double strap triple zigzag riveted butt joint.

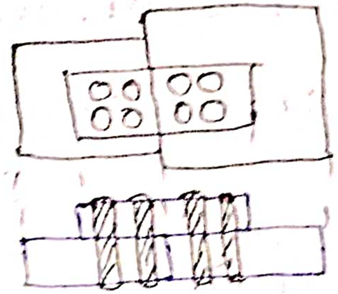
1- Double chain riveted lap joint



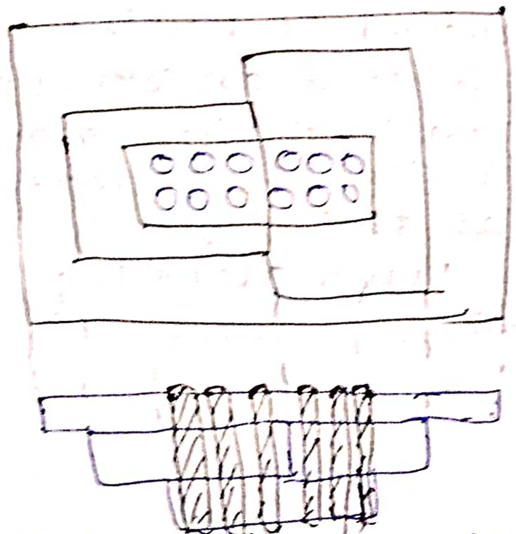
2- Triple zig-zag riveted lap joint.



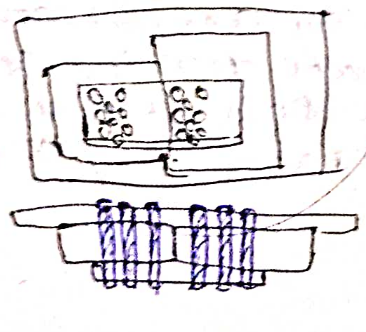
3- Single strap double chain riveted butt joint.



4- Double strap triple chain riveted butt joint.



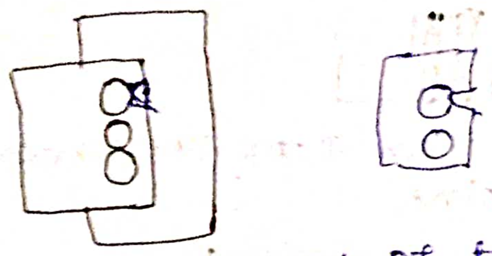
5 Double strap triple zigzag riveted butt joint



* Failures of riveted joint :- Dt-28/2/...

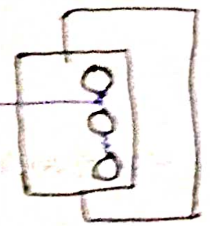
There are many reasons for the failure of a riveted joint. They are

i- Tearing of the Plates at an edge



→ When excess amount of tensile stress is subjected on rivets then, there is are possibilities that the plates which are riveted together may ~~...~~ break at the edge.

ii- Tearing of the Plates along the rows of rivet :-

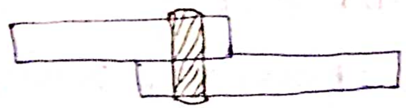


When the rivets are subjected to tensile stress which are greater than the ultimate tensile stress, then, there is a possibilities of tearing of the plates along the rows of rivets.

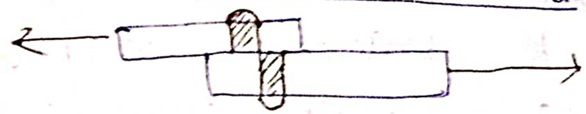
iii- Shearing of the rivets :-

→ When the rivets are subjected to shear stress greater than ultimate shear stress then, in that case the two plates which are riveted together. get sheared

Before Shearing

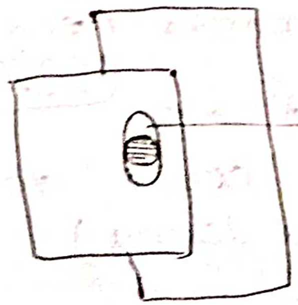


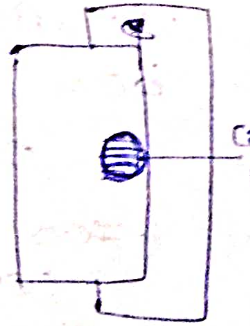
After Shearing :



iv- Crushing of the Plates :-

When excess amount of crushing force is subjected on the main plates than the holes created on the plates which are circular in shape gradually becomes oval in shape.





ine Shaft

* **Strength of a Rivete -:**

It can be defined as the maximum force a riveted Joint can transmitte without any failure.

1- **Tearing Resistance -: (P_t)**

$$P_t = (p-d)t \times \sigma_t$$

Where P_t = tearing resistance

p = Pitch of rivets

t = thickness of Plates

σ_t = tensile stress.

d = diameter of rivers.

Butt & Lap Joint

ii- **Shearing Resistance -: (P_s)**

a- $P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$

where P_s = Shearing resistance

n = no. of riveteing

d = diameter of rivets

τ = shear stress

for Lap & single strap butt Joint

b- $P_s = 2 \times n \times \frac{\pi}{4} \times d^2 \times \tau$

(for double strap / double cover butt Joint)

iii- **Crushing resistance -: (P_c)**

$$P_c = n \times d \times t \times \sigma_c$$

where P_c = crushing resistance

n = no. of riveting

d = diameter of rivets

t = thickness of Plates

σ_c = crushing stress

$\boxed{\text{Strength of a rived = Least value of } P_t, P_s \text{ or } P_c}$

* **Efficiency of the Rivet -: (η)**

$$\eta = \frac{\text{strength of rivet}}{p \times t \times \sigma_t}$$

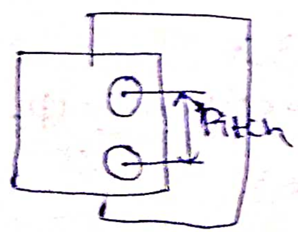
$$\boxed{\eta = \frac{\text{Least value of } P_t, P_s \text{ or } P_c}{p \times t \times \sigma_t}}$$

wher η = efficiency

p = Pitch of rivet

d = diameter of rivet

σ_t = tensile stress.



Ex Calculate the efficiency of single riveted lap joint of 6mm thick plates having 20mm dia rivet and a pitch of 50mm. Take $\sigma_t = 120$ MPa, $\tau = 90$ MPa, $\sigma_c = 180$ MPa. [10]

Data given

t = thickness = 6 mm

$d = 20$ mm

$p = 50$ mm

$\sigma_t = 120$ MPa $= 120\ N/mm^2$

$\sigma_c = 180$ MPa $= 180\ N/mm^2$

$\tau = 90$ MPa $= 90\ N/mm^2$

$n = 1$

i- Tearing Resistance (P_t)

$$P_t = (P-d)t \times \sigma_t$$
$$= (50-20)6 \times 120$$
$$= 21600\ N$$

ii- Shearing Resistance (P_s)

$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$$
$$= 1 \times \frac{\pi}{4} \times (20)^2 \times 90$$
$$= 28274.33\ N$$

iii- Crushing Resistance (P_c)

$$P_c = n \times d \times t \times \sigma_c$$
$$= 1 \times 20 \times 6 \times 180$$
$$= 21600\ N$$

∴ Strength of a rivet = Least value of P_t, P_s or P_c, $P_t = 21600$ N

$$\eta = \frac{\text{Strength of rivet}}{P \times t \times \sigma_t}$$
$$= \frac{21600}{50 \times 6 \times 120}$$
$$= 0.6$$
$$= 60\%$$

Ex Calculate the efficiency of double riveted lap joint of 6mm thick plates with 20mm dia rivet and a pitch of 65mm. Take $\sigma_t = 120$ MPa, $\tau = 90$ MPa, $\sigma_c = 180$ MPa.

Data given

$t = 6$ mm

$d = 20$ mm

$p = 65$ mm

$\sigma_t = 120$ MPa $= 120\ N/mm^2$

$\tau = 90$ MPa $= 90\ N/mm^2$

$\sigma_c = 180$ MPa $= 180\ N/mm^2$

$n = 2$

i- Tearing Resistance: (P_t)

$$P_t = (P-d)t \times \sigma_t$$
$$= (65-20) \times 6 \times 120$$
$$= 32400\ N$$

$$\boxed{P_t = 32400\ N}$$

ii- Shearing Resistance (P_s)

$P_s = n \times \frac{\pi}{4} d^2 \times \tau$

$= 2 \times \frac{\pi}{4} \times 20^2 \times 90$

$= 56548.66$ N

$\boxed{P_s = 56548.66 \text{ N}}$

iii- Crushing Resistance (P_c)

$P_c = n \times d \times t \times \sigma_c$

$= 2 \times 20 \times 6 \times 180$

$= 43200$

$\boxed{P_c = 43200 \text{ N}}$

Strength of rivet = Least values of P_t, P_s or P_c

$= P_t = 32400$ N

$\therefore \eta = \frac{\text{Strength of rivet}}{P \times t \times \sigma_t}$

$= \frac{32400}{65 \times 6 \times 120}$

$= 0.6923$

$= 69.23\%$

DT-03-1-2022

Q- Calculate the efficiency of a double riveted single strap butt joint having diameter of 20mm, width 10mm thick plates and a pitch of 50mm. Take σ_t = 100MPa, τ = 80MPa, σ_c = 180MPa

Data given

$n = 2$

$d = 20$ mm

$t = 10$ mm

$P = 50$ mm

$\sigma_t = 100$ MPa = 100 N/mm²

$\tau = 80$ MPa = 80 N/mm²

$\sigma_c = 180$ MPa = 180 N/mm²

i- Tearing resistance (P_t)

$\therefore P_t = (P-d) t \times \sigma_t$

$= (50-20) \times 10 \times 1000$

$= 30000$ N

$\boxed{P_t = 30000 \text{ N}}$

ii- Shearing resistance (P_s)

$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$

$= 2 \times \frac{\pi}{4} \times (20)^2 \times 80$

$= 50265.48$ N

$\boxed{P_s = 50265.48 \text{ N}}$

iii- Crushing resistance (P_c)

$P_c = n \times d \times t \times \sigma_c$

$= 2 \times 20 \times 10 \times 180$

$= 72000 N$

$\boxed{P_c = 72000 N}$

Strength = Least of $P_t, P_s, \& P_c$

i.e. $P_t = 30000 N$

$\therefore$ efficiency $= \dfrac{\text{Strength}}{P \times t \times \sigma_t}$

$= \dfrac{30000}{50 \times 10 \times 100}$

$= 0.6$

$= 60\%$

$\therefore \boxed{\eta = 60\%}$

Q. A double riveted double cover butt joint is having 20mm thick plates width 25mm diameter and 100mm pitch. Calculate the efficiency of the joint if $\sigma_t = 120$ mpa, $\tau = 100$ mpa and $\sigma_c = 150$ mpa.

Data given

$n = 2$

$t = 20mm$

$d = 25 mm$

$P = 100 mm$

$\sigma_t = 120$ mpa $= 120 N/mm^2$

$\tau = 100$ mpa $= 100 N/mm^2$

$\sigma_c = 150$ mpa $= 150 N/mm^2$

i- Tearing ~~force~~ resistance (P_t)

$P_t = (P-d) t \times \sigma_t$

$= (100 - 25) \times 20 \times 120$

$= 180000 N$

$\boxed{P_t = 180000 N}$

ii- ~~crushing~~ Shearing resistance (P_s)

$P_c = 2n \times \frac{\pi}{4} d^2 \times \tau$

$= 2 \times 2 \times \frac{\pi}{4} \times (25)^2 \times 100$

$= 196349.54 N$

$\boxed{P_c = 196349.504 N}$

iii- Crushing resistance (P_c)

$P_c = n \times d \times t \times \sigma_c$

$= 2 \times 25 \times 20 \times 150$

$= 150000\ N$

$\boxed{P_c = 150000\ N}$

Strength = least value of P_t, P_s & P_c

i.e. $P_c = 150000\ N$

∴ efficiency $= \frac{\text{Strength}}{p \times t \times \sigma_t}$

$= \frac{150000}{100 \times 20 \times 120}$

$= 0.625$

$= 62.5\%$

$\boxed{\eta = 62.5\%}$

Q A double riveted lap joint is made of 15mm thick plates having 25 mm diameter and 75mm pitch. Calculate the resistance induced in the join and also calculate the efficiency in the joint. Take $\sigma_t = 400\ MPa$, $\tau = 320\ MPa$, $\sigma_c = 640\ MPa$.

Data given

$n = 2$, $d = 25\ mm$

$t = 15\ mm$, $p = 75\ mm$

$\sigma_t = 400\ MPa = 400\ N/mm^2$

$\tau = 320\ MPa = 320\ N/mm^2$

$\sigma_c = 640\ MPa = 640\ N/mm^2$

i- Tearing resistance (P_t)

$P_t = (p-d)t \times \sigma_t$

$= (75-25) \times 15 \times 400$

$\boxed{P_t = 300000\ N}$

ii- Shearing resistance (P_s)

$P_s = 2n \times \frac{\pi}{4} d^2 \times \tau$

$= 2 \times \frac{\pi}{4} \times (25)^2 \times 320$

$= 314159.26\ N$

$P_s = 314159.26\ N$

iii- Crushing resistance (P_c)

$P_c = n \times d \times t \times \sigma_c$

$= 2 \times 25 \times 15 \times 640$

$= 480000\ N$

$\boxed{P_c = 480000\ N}$

∴ Strength = least value of P_t, P_s & P_c

i.e. $P_t = 300000\ N$

$\eta = \frac{\text{Strength}}{p \times t \times \sigma_t} = \frac{300000}{75 \times 15 \times 400}$

$= 0.6667$

$= 66.67\%$

$\boxed{\eta = 66.67\%}$

* Design of Boiler Joint :-

1. Thickness of Plates :-

$$\boxed{t = \frac{P \times D}{2 \times \sigma_t \times \eta} + 1\ mm}$$

Where P = Steam Pressure
D = Diameter of Boiler
t = thickness of Plates
σ_t = tensile stress
η = efficiency of joint.

2- Diameter of rivets :-

$$\boxed{d = 6\sqrt{t}}$$

Where t = thickness of Plates

3- Pitch of rivets (P)

$$\boxed{P = C \times t + 41.28\ mm}$$

where C = constant
t = thickness of Plates

or

Equate P_t & P_s

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

$\Rightarrow P = ?$

Taking smaller value of P = consider.

4- Distance between rows (D_R)

$$\boxed{D_R = 0.33P + 0.67d}$$

where P = Pitch of rivet
d = diameter of rivet.

5- Thickness of butt-strap :-

a) 1.125t Single strap-chain rivet)
b) $1.125t \left(\frac{P-d}{P-2d}\right)$ (single strap-zigzag riveting)
(c) 0.625t (Double strap chain riveting)
(d) $0.625t \left(\frac{P-d}{P-2d}\right)$ (Double strap-zig-zag riveting)

where t = thickness of Plates
P = Pitch of rivets
d = diameter of rivets.

6- Margin (m)

$$\boxed{m = 1.5d}$$

Where d = diameter of rivet.

* Assumption in Designing of Boiler Joint :-

→ The load should equally divided or distributed among all the rivets.
→ The tensile stress on all the rivet should be uniform.
→ The shear stress on all the riven should be uniform.

→ The crushing stress should be uniform.

→ The holes should fit properly with the diameter of the rivets.

→ The force of friction between the plate surface should be neglected.

Q A double riveted lap joint with zig zag riveting is to be designing for 13mm thick plates. Consider $\sigma_t = 80$ MPa, $\tau = 60$ MPa and $\sigma_c = 120$ MPa. Design the boiler joint. Calculate the efficiency of this joint.

<u>Data given</u>

$n = 2$

$t = 13$ mm

$\sigma_t = 80$ MPa $= 80$ N/mm²

$\tau = 60$ MPa $= 60$ N/mm²

$\sigma_c = 120$ MPa $= 120$ N/mm²

1- <u>Thickness of Plate</u>

$t = 13$ mm

2- <u>Diameter of rivets</u>

$d = 6\sqrt{t} = 6 \times \sqrt{13} = 21.63$ mm

3- <u>Pitch of rivets</u>

$P = 6 \times t + 41.28$

$= 2.62 \times 13 + 41.28 = 75.34$ mm

equate P_t & P_s

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

$\Rightarrow (P - 21.63) 13 \times 80 = 2 \times \frac{\pi}{4} \times (21.63)^2 \times 60$

$\Rightarrow P - 21.63 = \dfrac{2 \times \frac{\pi}{4} \times (21.63)^2 \times 60}{13 \times 80}$

$\Rightarrow P = 42.39 + 21.63$

$\Rightarrow P = 64.028$ mm

∴ Taking smaller value

$\boxed{P = 64.028 \text{ mm}}$

4- <u>Distance between Rows</u>

$D_R = 0.33P + 0.67d$

$= 0.33 \times 64.028 + 0.67 \times 21.63$

$= 35.62$ mm

5- <u>margin(m)</u>

$m = 1.5d$

$= 1.5 \times 21.63$

$= 32.445$ mm

i- Tearing resistance (P_t)

$P_t = (P-d)t \times \sigma_t$

$= (64.028 - 21.63) \times 13 \times 80$

$= 44093.92\ N$

$\boxed{P_t = 44093.92\ N}$

ii- Shearing resistance (P_s)

$P_s = n \times \frac{\pi}{4} d^2 \times \tau$

$= 2 \times \frac{\pi}{4} \times (21.63)^2 \times 60$

$= 44094.474\ N$

$\boxed{P_s = 44094.474\ N}$

iii - Crushing resistance (P_c)

$P_c = n \times d \times t \times \sigma_c$

$= 2 \times 21.63 \times 13 \times 120$

$= 67485.6\ N$

$\boxed{P_c = 67485.6\ N}$

Strength = Least of P_t, P_s & P_c

$= P_t$

$= 44093.92\ N$

$\therefore \eta = \frac{\text{Strength}}{P \times t \times \sigma_t} = \frac{44093.92}{64.028 \times 13 \times 80}$

$=$

Q. Design a double riveted butt joint with two cover plates for a boiler shell having 1.5m diameter subjected to steam pressure of 0.95 N/mm². Take the efficiency of the joint equal to 75%. Consider $\sigma_t = 90$ MPa, $\sigma_c = 140$ MPa, $\tau = 56$ MPa.

Data given

$n = 2$

$D = 1.5m = 1500\ mm$

$P = 0.95\ N/mm^2$

$\eta = 75\% = 0.75$

$\sigma_t = 90\ MPa = 90\ N/mm^2$

$\sigma_c = 140\ MPa = 140\ N/mm^2$

$\tau = 56\ MPa = 56\ N/mm^2$

1- Thickness of plates

$t = \frac{P \times D}{2 \times \sigma_t \times \eta} + 1$

$= \frac{0.95 \times 1500}{2 \times 90 \times 0.75} + 1$

$= 11.56\ mm$

$\boxed{t = 11.56\ mm}$

2- Diameter of rivets

$d = 6\sqrt{t}$

$= 6 \times \sqrt{11.56} = 20.4\ mm$

$\boxed{d = 20.4\ mm}$

3- <u>Pitch of rivets</u>

From data hand..
$C = 3.50$

$P = c \times t + 41.28$
$= 3.50 \times 11.56 + 41.28$
$= 81.74$ mm

equating P_t & P_s

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

$\Rightarrow (P-20.4) \times 11.56 \times 90 = 2 \times 2 \times \frac{\pi}{4} \times (20.4)^2 \times 56$

$\Rightarrow P - 20.4 = \frac{2 \times 2 \times \frac{\pi}{4} \times (20.4)^2 \times 56}{11.56 \times 90}$

$\Rightarrow P = 70.37 + 20.4$
$= 90.77$ mm

∴ Taking the smaller value

$\boxed{P = 81.74 \text{ mm}}$

4- <u>Distance between Rows</u>

$D_R = 0.33P + 0.67d$
$= 0.33 \times 81.74 + 0.67 \times 20.4$
$= 40.6422$ mm

$\boxed{D_R = 40.6422 \text{ mm}}$

5- <u>Thickness of butt strap</u>
i- For double strap chain riveting
Thickness of strap $= 0.625 \times t$
$= 0.625 \times 11.56$
$= 7.225$ mm

For double strap zigzag riveting

Thickness of strap $= 0.625t \times \left(\frac{P-d}{P-2d}\right)$
$= 0.625 \times 11.56 \times \frac{81.74 - 20.4}{81.74 - 2 \times 20.4}$
$= 10.82$ mm

6- <u>Margin</u>

$m = 1.5d$
$= 1.5 \times 20.4$
$= 30.6$ mm

*<u>Welded Joints</u> :-
→ A welded joint can be defined as a permanent joint which is formed by the fusion of edges of two plates with or without the application of pressure and filler material.

* <u>Types of welded joint</u> :-
There are two types of welded joint.
i- Lap joint
ii- Butt joint.

imp 15

* Advantages of welded joint over riveted joint.

- → The welded joint is always lighter than the riveted joint.
- → The welded joint has maximum efficiency as compare to riveted joint.
- → The welded joint has maximum strength as compare to the riveted joint.
- → It is possible to weld any part of a structure easily which is not possible in case of riveted joint.
- → The welded joint is smooth in appearance.
- → The welded joint always forms rigid ~~form~~ joint, which is not possible in case of riveted joint.
- → The failure of a riveted joint is more ~~light~~ likely to happen as compare to the welded joint.

* Disadvantages of welded joint over riveted joint.

- → To form welded joint skilled labours are required.
- → Welded joint requires proper fabrication process without which there is chances that the stresses will be induced in the joint.
- → Welded joint requires inspection work which is more difficult than riveting work.
- → The expension and contraction of the metal can also result in, the breakage or failure of the welded joint.

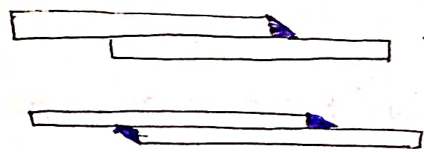
* Strength of transverse fillet welded joint:

Filleat -:

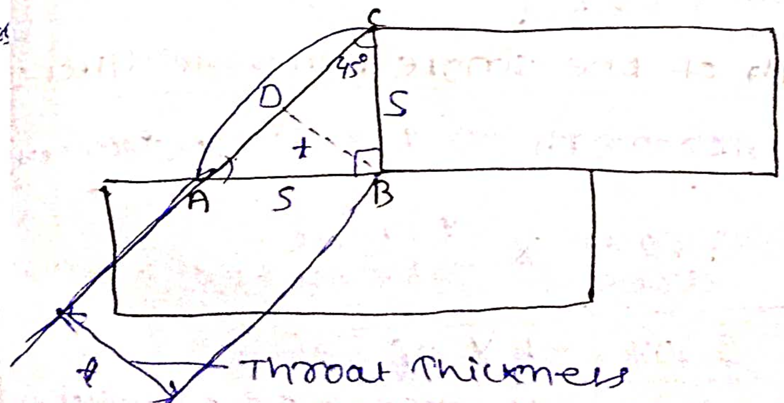
- → It is the area which is formed after the welding process.
- → The fillet is obtained by overlaping the plates and then welding the edges of the plates.
- → The transvers fillet are design to resist the tensile stress and they have high tensile strength.
- → There are two types of transverse fillet weld.

i - Single transverse fillet weld

ii - Double transverse fillet weld.



Strength of single transverse fillet weld -:



Let us take a right angle triangle ABC inside the fillet.

→ AB and BC will be the length of each sight of the weld which is known as leg or size of the weld. It is denoted as 's'.

→ The perpendicular distance from the ~~...~~ hypotenuse to the intersection of the legs is known as throat thickness. Here BD is the throat thickness. It is denoted by 't'.

In ABCD

$\sin C = \frac{BD}{BC}$

$\sin 45° = \frac{t}{s}$

$\Rightarrow t = s \times \sin 45°$

$\Rightarrow \boxed{t = 0.707 s}$

→ Area of the weld / throat area

A = Throat thickness × length of weld

$\Rightarrow A = t \times l$

$\Rightarrow \boxed{A = 0.707 s \times l}$

Strength of the single transverse fillet :-

~~Stress~~ Strength of the single transverse fillet

P = Throat area × tensile stress

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

Similarly ~~double~~ strength of the double transverse fillet

$\boxed{P = 2 \times 0.707 s \times l \times \sigma_t} = 1.414 s \times l \times \sigma_t$

* **Strength of parallel fillet welded joint :-**

→ Parallel fillet welded joints are design to resist the shear stress.

→ Parallel fillet welded joints have high shear strength.

→ Since parallel fillet welded joint are perform on both the ~~sides of~~ sides of the weld so

$$\boxed{\begin{aligned} P &= 2 \times 0.707 s \times l \times \tau \\ &= 1.414 s \times l \times \tau \end{aligned}}$$

Q A plate 100mm wide and 10mm thick is to be welded to another plate by means of double parallel fillet. The plates are subjected to a load of 80 kN. Find out the length of the weld, if the shear stress is given 55MPa.

Data given

Plate thickness = size of weld

$\Rightarrow s = 10\,mm$

$P = 80\,kN = 80 \times 10^3\,N$

$\tau = 55\,MPa = 55\,N/mm^2$

$P = A \times \tau$

$= 2 \times (t \times l) \times \tau$

$= 2 \times 0.707 s \times l \times 55$

$= 2 \times 0.707 \times 10 \times l \times 55$

$= 777.7\,l$

$\Rightarrow P = 777.7 l$

$\Rightarrow \frac{50\times10^3}{777.7} = l$

$\Rightarrow l = 102.86$ mm

For starting and stoping of weld run,

$l = 102.86 + 12.5$

$= 115.36$ mm

$\boxed{l = 115.36 \text{ mm}}$

Q A plate having 12mm thickness is to be welded to another plate by means of single transverse fillet weld. If the joint is subjected to a load of 90 KN and a tensile stress of 50 MPa. Calculate the length of the weld.

Data given

$P = 90$ KN $= 90\times10^3$ N

$\sigma_t = 50$ MPa $= 50$ N/mm²

$\cancel{s} = 12$ mm

$\therefore P = A \times \sigma_t$

$\Rightarrow 90\times10^3 = 0.707\, s \times l \times 50$

$\Rightarrow 90\times10^3 = 0.707 \times 12 \times l \times 50$

$\Rightarrow l = \frac{90\times10^3}{0.707\times12\times50}$

$\Rightarrow l = 212.16$ mm

$\therefore$ For starting and stoping of weld run,

$l = 212.16 + 12.5$

$= 224.66$ mm

$\boxed{l = 224.66 \text{ mm}}$

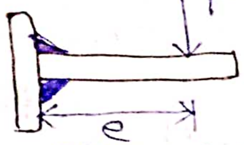
* Essentric loading in welded joint :-

→ The load which donot coinside with the axis of the body is known as essentric load.

→ The distance between the essentric load and the axis of the body is known as essentricity.

Case-1

P



e

essentricity

1- $A = 2\times t \times l$

$= 2\times0.707\, s\times l$

2- $P = A\times\sigma_t$

$= 2\times0.707\times s\times l\times\sigma_t$

3- $\tau = \frac{P}{A}$

4- Bending moment $(M) = P\times e$

5- section modulus $(Z) = \frac{s\times l^2}{4.242}$

6- Bending stress $(\sigma_b) = \frac{M}{Z}$

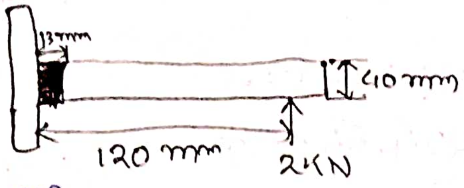
7- maximum normal stress $(\sigma_{n\,max})$

$\sigma_{n\,max} = \frac{\sigma_b}{2} + \frac{1}{2}\sqrt{\left(\frac{\sigma_b}{2}\right)^2 + 4\tau^2}$

8. maximum shear stress τ_{max}

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

Q A welded joint is shown in the figure. Calculate the maximum shear stress.



<u>Data given</u>

$P = 2KN = 2\times10^3 N$

$e = 120 mm$

$l = 40 mm$

$S = 13 mm$

1- Area

$A = t \times l$

$= 0.707 \times S \times 40$

$= 0.707 \times 13 \times 40$

$= 367.64 mm^2$ $= 735.28 mm^2$

2- $\tau = \frac{P}{A}$

$= \frac{2\times10^3}{735.28}$

$= 2.72 N/mm^2$

3- $M = P\times e = 2\times10^3\times120 = 240000 N.mm$

4- $Z = \frac{S\times l^2}{4.242} = \frac{13\times40^2}{4.242} = 4903.34 mm^3$

5- $\sigma_b = \frac{M}{Z} = \frac{240000}{4903.34} = 48.94 N/mm^2$

6- $\tau_{max} = \frac{1}{2}\sqrt{\left(\frac{\sigma_b}{2}\right)^2 + 4\tau^2}$

$= \frac{1}{2}\sqrt{\left(\frac{48.94}{2}\right)^2 + 4\times(2.72)^2}$

$= 12.53 N/mm^2$

$\therefore \boxed{\tau_{max} = 12.53 N/mm^2}$