DESIGN OF MACHINE ELEMENTS

5TH SEMESTER
MECHANICAL ENGINEERING

LECTURE NOTE

By

SRI BIPIN KUMAR DASH
Senior Lecturer
Department of Mechanical Engg.
Government polytechnic,Kandhamal
Phulbani-762001

Design of machine elements

Dt:- 03/11/21

(TH-2)

→ Introduction of DME

Clasification of Machine Design
- → Adaptive
- → Development
- → New

→ Adaptive design :-

* This type of design is done from the existing design.
* A little changes or modification can be accepted in the existing design of the product.
* This type of design doesnot need special knowledge or skill.

→ Development design :-

* This type of design needs considerable scientific training and design ability of the designer.
* The design concept created (developed) from the existing design but the final product may differ from the original product.

→ New design :-

* In this design, the designer needs
  i. Lot of research
  ii. Technical ability
  iii. Creative thinking.

→ Design procedure :-

* There are basically 8 no. of steps followed for a basic machine design.

Recognition of need
↓
Synthesis mechanism
↓
analysis of process
↓
Selection of the material
↓
Design of elements
↓
Modification
↓
Detailed drawing
↓
production.

i) Recognition of need :-

* The first step of design is to mode a complete statement of the problem.

* That problem should indicate the need or purpose for which the machine is to be designed.

ii) Synthesis mechanism :-

* There are a no. of possible mechanisms available for designing. But we have to select the best suitable mechanism from the group of mechanisms which will fullfill the desired motion.

iii) Analysis of process :-

* In this step forces acting on different elements

are to be analized.

* Here the energy transmitted by the member (element) are also analized.

iv) Selection of the material :-

* The designer should choose the best suitable material for each member (element) of the machine.

v) Design of elements :-

* when the elements (parts of the machine).
* First we have to calculate different stress acted upon different elements of the machine.
* Then we have to choose the required shape and size of the elements so that it can transmit required motion.
* while designing the elements one thing should kept in mind that it should n't go the deformation beyond the permissible limit of deformation.

VI) Modification :-

* Modify the design of the elements if required. On the basis of the manufacturer requirment.

VII) Detailed Drawing :-

* After completing the design calculations of the machine components, by using different design softwares or by the use of drafter (manual) drawing should be drawn.

VIII) Production :-

* After detailed drawing it is sent to the shop fl[oor] (machine) for production.
* As per the drawing the product is manufactured.

⇒ Classification of engineering material.

Engineering materials

- Metals and their alloys (Iron, Al, cu...)
  - Ferrous (Iron as main constituents)
  - Non-ferrous (absent of Fe) (Al, cu, An)
- Non-metals (glass, wood, Rubber)

→ Physical properties

* Luster (Polished or finishing surface)
* colour
* Shape and size
* density
* Electrical conductivity
* Thermal conductivity
* melting point

→ Mechanical properties

various types of mechanical properties are

* Strength
* Stiffness
* Elasticity
* plasticity
* Ductility
* Brittleness
* Malleability
* Toughness
* Machinability
* Resilience
* Creep
* Fatigue
* Hardness.

Q. Explain different mechanical properties of metals?

ans:- These are some of the mechanical properties of metals.

i) Strength
ii) Stiffness
iii) Elasticity
iv) plasticity
v) Ductility
vi) Brittleness
vii) Malleability
viii) Toughness
ix) Machinability
x) Resilience
xi) Creep
xii) Fatigue
xiii) Hardness

i) Strength :-

* It is the property of metal by virtue of which it will resist externally applied loads without breaking.

ii) Stiffness :-

* It is ability of the material to resist deformati which is caused by stress.

Note :- The modulus of elasticity (E) is the measure of stiffness.

iii) Elasticity :-

* Elasticity is the property of the material by virtue of which it can regain its original shape and size after the removal of load (external load

iv) Plasticity :-

* plasticity is the property of the material by virtue of which it can regain its original shape and size after the removal of external load.

v) Ductility :-

* It is the property of material by virtue of which it can be drawn into wires.

vi) Brittleness :-

* It is the property of material by virtue of which it cannot drawn into wires.

* Brittleness property is just opposite to the ductility

* with little permanent deformation the material will

break with the application of external load.

VII) Malleability :-

* It is a property of material by virtue of which it can turn into sheets by the application of external load.

VIII) Toughness :-

* It is the property of a material which can resist fracture due to high impact load.

IX) Machinability :-

* It is the property of material which refers to ability to cut the material.

X) Resilience :-

* It is the property of material to absorb energy with in elastic limit.

* The maximum resilence occurs at elastic limit and this known as proof resilience.

* Modulas of resilience = $\frac{\text{Proof resilience}}{\text{unit volume}}$

XI) Creep :-

* Creep is a testing method where a material is Subjected to

→ constant stress

→ High temperature

→ For a long period of time.

XII) Fatigue :-

* when a material is subjected to repeated stresses the material will fail before yield point. that means the material can not be reused this is known as fatigue.

* Generally the fatigue failure occurs due to the progressive growth of the crack.

XIII) Hardness :-

* Hardness includes different properties such as

→ Resistance to wear

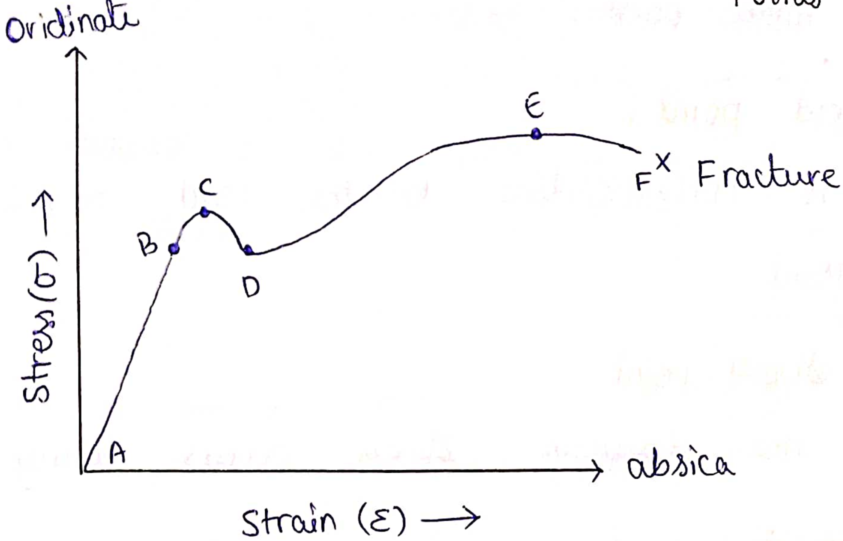
→ Resistance to scracthing

→ Resistance to deformation

→ Resistance to machinability etc.

* The hardness is generally expressed in numbers.

Q. Explain stress versus strain diagram of mild steel Specimen with its different Salient points ?



AB → Proportional limit

B → Elastic limit

C → upper yield point

D → lower yield point

E → Maximum stress

F → Fail / breaking stress.

i. Proportional limit :-

* From the origin A to the point B called proportional limit, the stress strain diagram curve is a straight line.
* It state that within the proportional limit the stress is directly proportional to strain.

II. Elastic limit :-

* The limit in which the material will return to its original shape and size when the load is removed.

III. yield point :-

* yield point is the point at which the material will have an appreciable elongation.

upper yield point :-

* which is corresponding to the load reached Just before yield point.

Lower yield point :-

* which is corresponding to the load required yield Point.

ultimate stress point :-

* It is the maximum stress occurs during the test.

Fracture point :-

* It is also known as failure point at which the materi breaks and it reaches to its maximum strain limit.

Working stress :-

* The stress ~~to~~ to which material may be safely subjected in the course of ordinary use.

yield stress :-

* It is the value of stress below which a material will deform elastically and will return to its original shape when the applied stress is removed.

→ Factor of Safety :- (F.O.S)

$$F.O.S = \frac{U.S}{W.S}$$

where, U.S = ultimate stress
W.S = working stress.

* F.O.S is defined as the ratio of maximum stress to the working stress.

Note :-

For ductile material, $F.O.S = \frac{\text{yield stress}}{\text{working stress}}$

For Brittle material, $F.O.S = \frac{\text{ultimate stress}}{\text{working stress}}$

→ Factors effecting machine design :-

i. Types of load and stresses caused by the load.

ii. Motion of the parts or kinematics of the machine.

iii. Selection of materials

iv. Form and size of the parts.

v. Frictional resistance and lubrication.

vi. convinient and economical features.

vii. use of standard parts.

viii. Safety of operation

ix. workshop facilities

x. No. of machines to be manufactured

xi. Cost of constructions.

xii. Assembling.

Write short notes on

→ Types of load and stresses

Types of load :-

* Load is defined as any external force acting upon a machine part.

Dead or steady load :-

* which does not change in magnitude and direction.

Continious or variable :-

* when the load is applied continiously is called continious or vaiable load.

Suddenly applied load :-

* when the load is applied or removed suddenly is called suddenly applied load.

Impact Load :-

* when the load is applied with some initial velocity is called impact load.

Stress :-

* when ever a body is subjected to the load an internal resisting force will develop to oppose the defor- -mation this resisting force per unit cross-sectional area is known as stress.

* It is denoted as 'σ' unit N/mm^2

$$\text{Stress} = \frac{\text{Resisting force}}{\text{C.S. Area}}$$

Tensile stress :-

* when ever a body is subjected to equal and opposi
pull forces it elongates, this increament of the length
per original length is called tensile stress.

Compressive stress :-

* when a body is subjected to equal and opposite
push forces it contracts, This decrement of the length
per original length is called compressive stress.

Shear stress :-

* when the external force on the body tries to
shear on the body this shear force per unit cro
sectional area is known as shear stress.

Bearing stress :-

* If we apply a compressive stress on a body,
then there will be a reaction as internal stress
in the material of the body. This internal stress
is known as the bearing stress.

SHAFT DESIGNING :-

* The shaft may be designed on the basis of
  i) strength.
  ii) stiffness or rigidity

* while designing the shaft on the basis of strength the following cases may be taken into consideration.

→ Shaft subjected to Torque (Twisting moment only).

→ Shaft subjected to Bending moment only.

→ Shaft subjected to Combined twisting and bending moment.

→ Shaft subjected to axial load in addition to combined twisting and bending moment.

Q.1) what is shaft?

ans:- It is a rotating member usually of circular cross-section to transmit power. It is subjected to transmit torque and bending moment. It is used to transmit power in engines, machines and equipments.

Q.2) what is axle?

ans:- It is a stationary machine element and it is used for transmission of bending moment only. It simply as a support for some rotating body such as holding drum of a car wheel, pulley. It may be rotating or machine element.

Q.3) what is key?

ans:- A key can be defined as the machine element it is used to connect transmission shaft to rotating machine element, like pulley, gear, spocket or flywheel.

→ Types of Shaft

1. Transmission shaft :-

* these shaft transmit power between the source and the machine absorbing power. The counter shafts, line shaft, over head shaft and all factory shafts are transmission shafts.

II. Machine shaft :-

* These shaft for an integral part of the machine itself. The crank shaft is an example of machine shaft.

→ standard sizes of transmission shafts.

* The standard sizes of shaft are:-

25 mm to 60 mm with 5 mm steps

60 mm to 110 mm with 10 mm steps

110 mm to 140 mm with 15 mm steps

140 mm to 500 mm with 20 mm steps

→ The standard length of shaft are [5m, 6m and 7m].

→ Stress in Shaft :-

I. Shear stress :- Due to transmission of torque.

II. Bending stress :- due to (transmission) force acting upon machine element like gear, pulley's etc. as well as due to the weight of the shaft itself.

III. Stress due to combined torsional and bending load.

→ Shaft subjected to twisting moment only.

$$\frac{T}{J} = \frac{\tau}{R} = \frac{C\theta}{l} \quad \text{(pure torsional equation)}.$$

Let shaft is solid shaft of diameter d.

J = Polar moment of inertia.

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

For solid shaft $\boxed{T = \frac{\pi}{16} \tau d^3}$

For hollow shaft = $J_{hollow} = \frac{\pi}{32} (D^4 - d^4)$

Q. A line shaft is rotating at 200 r.p.m is to transmit 20 kW. The shaft may be assumed to be made up of mild steel with an allowable shear stress of 42 MPa. Determine the diameter of the shaft. Neglating Bending moment effect.

Soln:- Given data

$N = 200$ r.p.m

Shear stress

$= \tau = 42$ MPa

$1 \, Pa = 1 \, N/m^2$

$\Rightarrow 1 \, MPa = 1 \, N/mm^2$

$1 \, MPa = 10^6 \times \frac{N}{(10^3)^2 \, mm^2}$

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

$1 \, MPa = 1 \, N/mm^2$

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

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

$T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2 \times \pi \times 200} = 955 \, N\text{-}m$

$= 955 \times 10^3 \, N\text{-}mm$

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

$\Rightarrow d = \left(\frac{T \times 16}{\pi \tau}\right)^{1/3} = \left(\frac{955 \times 10^3 \times 16}{\pi \times 42}\right)^{1/3}$

$d = 48.7 \, mm$

$\cong 50 \, mm$

Q.2) A solid shaft is transmitting 1 MW at 240 r.p.m determine the diameter of the shaft if the maximum torque transmited exceeds the mean torque by 20%. Take the maximum allowable shear stress as 60 MPa.

Soln:- Given data

power (p) = 1 MW = 10^6 watt

N = 240 r.p.m

$T_{max} = 1.2\, T_{mean}$

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

d = ?

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

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

$$= \frac{10^6 \times 60}{2 \times \pi \times 240} = 39784 \text{ N-m}$$

$$T_{mean} = 39784 \times 10^3 \text{ N-mm}$$

$$T_{max} = 1.2\, T_{mean}$$

$$= 1.2 \times 39784 \times 10^3$$

$$= 47741 \times 10^3 \text{ N-mm}$$

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

$$47741 \times 10^3 = \frac{\pi}{16} \times 60 \times d^3$$

$$d^3 = \frac{16 \times 47741 \times 10^6}{\pi \times 60}$$

$$d^3 = 4053 \times 10^3$$

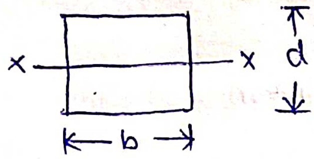
$$d = \sqrt[3]{4053 \times 10^3}$$

$$= 159.4 \text{ mm} \simeq 160 \text{ mm}.$$

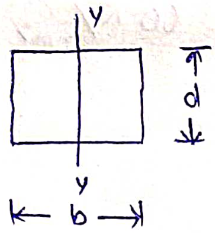
→ Shaft subjected to Bending moment

$$\frac{M}{I} = \frac{\sigma_b}{y} = \frac{E}{R}$$

$$I_{xx} = \frac{1}{12} b \cdot d^3$$

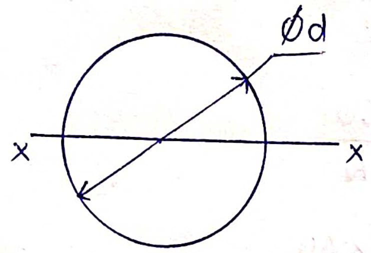


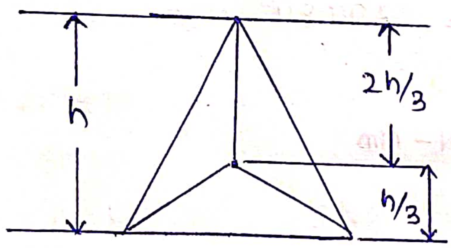
$$I_{yy} = \frac{1}{12} d \cdot b^3$$



$$I_{xx} = \frac{\pi}{64} d^4$$

$$I_{yy} = \frac{\pi}{64} d^4$$





→ Shaft subjected to T and M (Torque and Bending moment)

If T → Twisting moment
M → Bending moment (is known)

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

T_e → equivalent twisting moment

$$M_e = \frac{1}{2}\left[M + T_e\right]$$

$$M_e = \frac{1}{2}\left[M + \sqrt{T^2 + M^2}\right] = \frac{\pi}{32} \sigma_b d^3$$

M_e → equivalent bending moment.

Q.) A line shaft is rotating at 200 r.p.m is to transmit 20 KW. The shaft may be assumed to be made up of mild steel. with an allowable shear stress of 42 MPa Determine the diameter of the shaft neglating bending moment effect. (20)

Soln:- Given data

$N = 200$ r.p.m

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

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

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

$$T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2 \times \pi \times 200} = 955 \text{ N-m}$$

$$= 955 \times 10^3 \text{ N/mm}$$

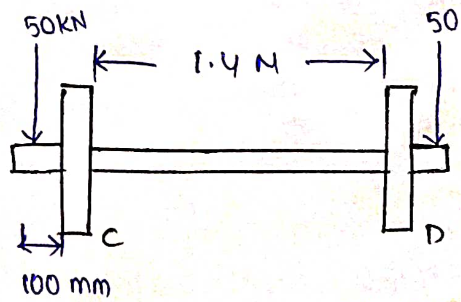
$$T = \frac{\pi}{16} \tau d^3$$

$$d = \left(\frac{T \times 16}{\pi \tau}\right) = \left(\frac{955 \times 10^3 \times 16}{\pi \times 42}\right)^{1/3}$$

$$\Rightarrow d = 48.7 \text{ mm}$$

$$\cong 50 \text{ mm}$$

Q.) A pair of wheels of a railway wagon carries ~~a~~ a load of ~~100~~ 50 KN on each, axle box. acting at a distance of 100 MM outside the wheel base. the gauge of the rail is 1.4 M. Find the diameters of axle between the wheels. If the stress is not to exceed 100 MPa.



Soln:-

$$M = W \cdot L = 50 \times 10^3 \times 100 = 5 \times 10^6 \text{ N-mm}$$

d = diameter of axle.

$$M = 5 \times 10^6 = \frac{\pi}{32} \times \sigma_b \times d^3$$

$$= \frac{\pi}{32} \times 100 \times d^3 = 9.82 d^3$$

$$d^3 = \frac{5 \times 10^6}{9.82} = 0.51 \times 10^6 \text{ or}$$

$$d = 79.8 \cong 80 \text{ mm}$$

→ what is key?

ans:- A key is a piece of mild steel inserted between the shaft and hub of the pulley to connect these in order to prevent relative motion between them.

* It is always inserted parallel to the axis of the shaft.

→ what type of stresses are induced in keys?

ans:- Two types of stresses are induced in keys

i) Crushing stress
ii) Shearing stress

→ Types of keys :-

i) Sunk keys
ii) Saddle keys
iii) Tangent keys
iv) Round keys.
v) Splines.

1. Sunk keys :-

* The sunk keys are provided half in the keyway of the shaft and half in the keyway of the hub or boss of the pulley.

* The sunk keys are of the following types.

1. Rectangular sunk key.
2. Square sunk key.
3. Parallel sunk key.
4. Gib - head key.
5. Feather key.
6. woodruff key.

2. Saddle keys :-

* The Saddle keys are of the following two types.

i. Flat Saddle key.

ii. Hollow Saddle key.

i. Flat Saddle key :- It is a taper key which fits in the keyway in the hub and is flat on the shaft. It is likely to slip round the shaft under load. Therefore it is used for comparatively light loads.

ii. Hollow Saddle key :- It is a taper key which fits in a keyway in the hub and the bottom of the key is shaped to fit the curved surface of the shaft. since hollow Saddle keys hold on by friction, therefore these are suitable for light loads. It is usually used as a temporary fastening in fixing and setting eccentrics, cams etc.

3. Tangent keys :-

* The tangent keys are fitted in pair at right angles. Each key is to withstand torsion in one direction only. These are used in large heavy duty shafts.

4. Round keys :-

* The round keys are circular in section and fit into holes drilled partly in the shaft and partly in the hub. they have the advantage that their keyways may be drilled and reamed after the mating parts have been assembled. Round keys are usually considered to be more appropriate for low power drives.

5. Splines :-

* Sometimes, keys are made integral with the shaft which fits in the keyways broached in the hub. Such shafts are known as splined shafts.
* These shafts usually have four, six, ten or sixteen splines. The splined shafts are relatively stronger than shafts having a single keyway.

Q→ Design the rectangular key for a shaft of a 50 m.m diameter. The shearing and crushing stress for the key material are 42 MPa, 70 MPa.

Soln:- Given data,

$d = 50\ mm$

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

$\sigma_c = 70\ N/mm^2$

Apporoximate proportions of sunk key.

$$w = \frac{d}{4} = \frac{50}{4} = 12.5\ mm$$

$$w = 12.5\ mm$$

$$t = \frac{2}{3} w = \frac{d}{6}$$

$$= \frac{50}{6} = 8.33 = \frac{2}{3} \times 12.5 = 8.33\ mm.$$

Torque transmitted by the sunk key

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

$$= l \times 12.5 \times 42 \times \frac{50}{2}$$

$$T_{shear} = 13125 \cdot l$$

Crushing stress

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

$$\frac{8.33}{2} \times 70 \times \frac{50}{2}$$

$$T_c = 7288.75 . l$$

Torque transmitted by the shaft $= \frac{\pi}{16} \times \tau \times d^3$

$= 1080835 \ N/mm$

$$= L = \frac{T_{shaft}}{T_{shear}} = \frac{1030835}{13125} = 78.5 \cong 79 \text{ mm}.$$

$$= L = \frac{T_s}{T_c} = \frac{1030835}{7288.75} = \boxed{141.4 \simeq 141 \text{ mm}}$$

∴ From the design we got length of the key $l = 80$ mm (considering shear stress), $l = 141$ mm (considering crushing stress)

For safer design we have to take the larger length of the key. That means $l = 142$ mm.

Q→ Differentiate between shaft and key

Shaft	Key
i. Rotating member	i. Non-rotating member.
ii. used to transmit the Torque and support the transmission element like → gear, pulley etc.	ii. Only used to support the transmission element like :- wheels, pulleys, etc.
iii. It is subjected to Torque bending moment and axial force.	iii. It is subjected to bending moment.
iv. Example :- Shaft, counter-shaft, spindle, crank shaft.	iv. Example :- Front axle of car, wheel axle of motor etc.

→ what is coupling ?

* Shafts are usually available upto 7 meters long due to inconvinience in transportation. In order to have a greater length, it becomes necessary to joint two or more pieces of the shaft by means of coupling.

→ why we will use coupling ?

* Shaft coupling are used in machinary for several purposes.

i. To provide for the connection of shaft of unit that are manufactured separatly such as motor and generator and to provide disconnection for repair or alternation.

ii. To provide for misalignment of the shaft on to introduce mechanical flexibility.

iii. To reduce the transmission of shock load from one shaft to another.

iv. To introduce protection aganist overloads.

v. It should ~~at~~ have no projecting parts.

→ Requirement of a good shaft coupling.

i. It should be easy to connect or disconnect.

ii. It should transmit the full power from one shaft to the other shaft without any lose.

iii. It should hold the shaft with perfect alignment.

iv. It should reduce the transmission of shock loads from one shaft to another shaft.

v. It should have no projecting parts.

→ Types of coupling :-

- Coupling
  - Rigid coupling
    - Sleeve or muff coupling.
    - Clamp or split-muff coupling.
    - Flange coupling
  - Flexible coupling.
    - Bushed pin type coupling
    - Oldhum coupling
    - universal coupling

→ Rigid coupling :-

* It is used to connect two shafts which are perfectly aligned.

→ Flexible coupling :-

* It is used to connect two shafts having both lateral and angular misalignment.

Q. Design and make a neat diamensional sketch of a muff coupling which is used to connect two steel shafts transmitting 40 KW at 350 r.p.m. The material for the shafts and key is plain carbon steel for which allowable shear and crushing stresses are taken as 40 MPa and 80 MPa respectively. The material for the muff is cast iron for which the allowable shear stress may be assumed as 15 MPa.

Sol^n:- Given data,

Power transmitting = P = 40 KW

Speed of shaft = n = 350 r.p.m

Material of the shaft and key is plain carbon steel.

Shear stress = τ_1 = 40 MPa = 40 N/mm^2

Crushing stress = 80 MPa = 80 N/mm^2

τ_2 = 15 MPa = 15 N/mm^2.

$$T = \frac{60 \times 10^6 \times P}{2\pi N}$$

$$T = \frac{60 \times 10^6 \times 40}{2 \times \pi \times 350} = 1.31 \times 10^{12}$$

$$d = \left[\frac{16T}{\pi \tau_1}\right]^{1/3} = \left[\frac{16 \times 1.31 \times 10^{12}}{\pi \times 40}\right]^{1/3}$$

d = 51.79 mm.

D = 2d

= 51.79 × 2

D = 103.58 mm

L = 3d = 155.37 − 181.26 mm

$$w = d/4 = \frac{51.79}{64} = 12.94 \text{ mm}$$

$$t = d/6 = \frac{51.79}{6} = 8.63 \text{ mm}$$

$$T = \frac{\pi}{16} \tau_2 D^3 (1-K^4)$$

K = d/D

K = 0.5

$$= \frac{\pi}{16} \times 15 \times (103.58)^3 (1 - 0.5^4).$$

T = 3068457.73

SPRING

Q→ Design a helical compression spring for a maximum load of 1000 N for a deflection of 25 mm using the value of spring index as 5.

Take maximum permissible shear stress for spring wire is 420 MPa and modulus of rigidity is 84 kN/mm²

Take wahl's factor,

$$k = \frac{4c-1}{4c-4} + \frac{0.615}{c}$$, where c = spring index.

Soln:- Given data,

Linear deflection = δ_{max} = 25 mm

Spring index = c = 5

wahl's factor $k = \frac{4c-1}{4c-4} + \frac{0.615}{c}$

$$= \frac{4 \cdot 5 - 1}{4 \cdot 5 - 4} + \frac{0.615}{5}$$

$$k = 1.3105$$

Maximum shear stress = τ = 420 MPa

$= 420 \text{ N/mm}^2$

Modulus of rigidity = G = 84 kN/mm²

$= 84 \times 10^3 \text{ N/mm}^2$

Maximum load = P = 1000N

$$\tau = \frac{8K\,PC}{\pi d^2}$$

$$420 = \frac{8 \times 1.3105 \times 1000 \times 5}{\pi d^2}$$

$$= d^2 = \frac{8 \times 1.3105 \times 1000 \times 5}{\pi \times 420}$$

$d^2 = 39.72$

$d = \sqrt{39.72}$

$= 6.30$

eye diameter $(d) = 6.30$ mm

For compression spring

$N =$ Total no. of coils $= n$

$$\delta = \frac{8 \times PC^3 n}{Gd}$$

$$25 = \frac{8 \times 1000 \times 5^3 \cdot n}{84 \times 10^3 \times 6.30}$$

$$= n = \frac{8 \times 1000 \times 5^3 \times 25}{84 \times 10^3 \times 6.30}$$

$n = 47.24$

$N = n$, $N = 47.24$

$L_s = Nd = 47.24 \times 6.30$

$L_s = 297.61$

solid length of spring $= L_s = 297.61$ mm.

Free length of spring $= L_f$

$L_f = Nd + 1.15 \times \delta_{max}$

$= 297.61 + 1.15 \times 25$

$= 326.36$ mm

$C = \frac{D}{d} = 5 = \frac{D}{6.30}$

$D = 6.30 \times 5$

$= 31.5$ mm

* Designed data :-

$D = 31.5$ mm

$d = 6.30$ mm

$l_s = 297.61$ mm

$l_f = 326.36$ mm

SPRING

→ what is spring ?

ans:- A spring is defined as an elastic body whose function is to when loaded and recover its original shape when the load is removed.

→ Application of spring :-

1. To cushion, absorb or control energy due to eith shock or vibration as in as spring, railway buffers, air-craft landing gears, shock absorbers and vibration damper.

2. To ~~apply forces~~ control motion, as in maintaining contact between two elements as in cams and followers.

3. To apply forces, as in brakes, clutches and spring loaded value.

4. To measure forces as in spring balances and engine - indicator.

5. To store energy as in watches, toys etc.

→ Types of spring :-

* Helical spring.
* conical and volute spring.
* Torsion spring.
* Laminated or leaf spring.
* Disc. or bellevible spring.
* Special purpose spring.

Helical spring

compression — Tension.

→ Terms used in compression spring.

1. Solid length
2. Free length
3. spring index (c) $c = \frac{D}{d}$
4. Spring rate :- It is defined as the load required per unit deflection of the spring ($k = \frac{W}{\delta}$) N/mm
5. pitch.

Q. what do you mean by wahl's stress factor for helical spring?

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

where,

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

Q.) A compression coil spring made of an alloy steel is having the following specifications :-

Mean diameter of coil = 50 mm ;
wire diameter = 5 mm;
Number of active coils = 20.

If the spring is subjected to an axial load of 500 N ; Calculate the maximum shear stress (neglect the curvature effect) to which the spring material is subjected.

Soln:- Given : D = 50 mm ; d = 5 mm ; n = 20, W = 500 N

spring index $C = \frac{D}{d} = \frac{50}{5} = 10$

∴ shear stress factor,

$$K_s = 1 + \frac{1}{2C} = 1 + \frac{1}{2\times10} = 1.05$$

Max shear stress

$$\tau = K_s \times \frac{8W.D}{\pi d^3} = 1.05 \times \frac{8\times500\times50}{\pi\times5^3} = 534.7 \ N/mm^2.$$

$$= 534.7 \ MPa.$$

Q. A helical spring is made from a wire of 6mm diameter and has outside diameter of 75 mm. If the permissible shear stress is 350 MPa and modulus of rigidity 84 kN/mm², find the axial load which the spring can carry and the deflection per active turn.

Soln:- Given :- $d = 6mm$, $D = 75 \ mm$; $\tau = 350 \ MPa$; $G = 84 \ kN/mm^2$
$= 84\times10^3 \ N/mm^2$

D mean of spring,

$$D = D_o - d = 75 - 6 = 69 \ mm$$

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

W = axial load

δ/n = Deflection per active turn.

1. Neglecting the effect of curvature.

Shear stress factor;

$$K_s = 1 + \frac{1}{2C} = 1 + \frac{1}{2\times11.5} = 1.043$$

Maxm shear stress (τ).

$$350 = K_s \times \frac{8W.D}{\pi d^3} = 1.043 \times \frac{8W\times69}{\pi\times6^3} = 0.848 \ W$$

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

Deflection of spring,

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

∴ Deflection per active turn,

$$\frac{\delta}{n} = \frac{8W.D^3}{G.d^4} = \frac{8\times412.7\,(69)^3}{84\times10^3\times6^4} = 9.96 \ mm$$ and

Joints
- Temporary
- Permanent

→ Temporary Joint :-

* while ditaching of the joint, if there will be no effect on parent materials then the joint is known as temporary joint.

Example :- Nut and Bolt, Screw joint, key joint, coupling etc.

→ Permanent Joint :-

* During the ditachment of the joint, if there will be some effect on parent materials. then the joint is known as Permanent Joint.

→ Welding :-

* Fusion of two materials by adding heat with or without filler materials and with or without pressure is known as welding.

→ Advantages and disadvantages of welded Joints over riveted Joints :-

* Following are the advantages and disadvantages of welded joints over riveted joints.

Advantages :-

1. The welded structures are usually lighter than riveted structures.

2. The welded joints provide maximum efficiency (may be 100%) which is not possible in case of riveted joints.

3. Alterations and additions can be easily mode in the existing structures.

4. As the welded structures is smooth in appearance, therefore it looks pleasing.

5. In welded connections, the tension members are not weakned as in the case of riveted Joints.

6. A welded joint has a great strength.

7. The process of welding takes less time than the riveting.

8. It is possible to weld any part of a structure at any point. But riveting requires enough clearance.

9. The welding provides very rigid joints.

→ Disadvantages :-

1. Since there is an uneven heating and cooling during fabrication, therefore the members may get distorted or additional stresses may develop.

2. It requires a highly skilled labour and supervision.

3. Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it.

4. The inspection of welding work is more difficult than riveting work.

→ Types of welding process :-

i) Fusion welding (only heat).

ii) Forge welding (heat and pressure).

i) Fusion welding :-

* In case of fusion welding, the parts to be jointed are held in position while the molten metal is supplied to the joint. The molten metal may come from the parts themselves.

* The fusion welding, according to the method of heat generated, may be classified as:

i. thermit welding

ii. Gas welding

iii. electric arc welding.

ii) Forge welding :-

* In forge welding, the parts to be jointed are first heated to a proper temperature in a furnace or forge and then hammered. This method of welding is rarely used now-a-days. An electric - resistance welding is an example of forge welding.

* Types of forge welding :-

i. Spot welding

ii. Seam welding

iii. Projection welding

iv. upset and flash welding.

→ Types of welded Joints
* Lap Joint
* Butt Joint

→ Lap Joint :-

The lap joint or the fillet Joint is obtained by overlapping the plates and then welding the edges of the plates.
The cross section of the fillet Joint is approximately triangular.
The fillel joints may be

i. Single transverse fillet
ii. Double transverse fillet
iii. Parallel fillet joints.

→ Butt Joint :-

The butt joint obtained by placing the plates edge to edge. In butt welds, the plate edges do not require bevelling if the thickness of plate is less than 5mm. On the other hand, if the plate thickness is 5 mm to 12.5 mm the edges should be bevelled to V or U - groove on both sides.
The butt Joints may be.

i. Square butt Joint
ii. Single V-butt Joint
iii. Single U-butt Joint
iv. Double V-butt Joints
v. Double U-butt Joints.

Q. A plate 100 mm wide and 10 mm thick is to be welded to another plate by means of double parallel fillet. The plates are subjected to a static load of 80 KN. Find the length of weld if the permissible shear stress in the weld does not exceed 55 MPa.

Soln:- Given data;

width = 100 mm

Thickness = 10 mm

$P = 80\ KN = 80 \times 10^3\ N$

$\tau = 55\ MPa = 55\ N/mm^2$.

Let l = Length of weld, and

S = Size of weld = plate thickness = 10 mm (Given).

$(P) = 80 \times 10^3 = 1.414 \times S \times l \times \tau = 1.414 \times 10 \times l \times 55 = 778\,l$.

$\therefore \quad l = 80 \times 10^3 / 778 = 103\ mm$.

Adding 12.5 mm for starting and stoping of weld run, we have

$l = 103 + 12.5 = 115.5$ ans

→ what is a Rivet ?

ans:- A rivet is a short cylindrical bar with a head integral to it. The cylinder portion of the rivet is called shank or body and lower portion of shank is known as tail.

* The rivets are used to make permanent fastening between the plates such as in structural work, ship ~~buld~~ buildings, bridges, tanks and boiler shells.

* The riveted joints are widely used for joining light metals.

→ The fastenings may be classified into the following two groups :-

1. Permanent fastenings,
2. Temporary or detachable fastenings.

1. Permanent fastenings

* The permanent fastenings are those fastenings which can not be disassembled without destroying the connecting component. the ~~ex~~ examples of permanent fastenings ~~to~~ in order of strength or soldered, brazed, welded and riveted joints.

2. Temporary or ditachable fastenings :-

* The temporary or ditachable fastenings are those which can be disassembled without destroying the connecting component. The examples of temporary fastenings are screwed, keys, cotters, pins and splined joints.

→ Methods of Riveting :-

* There are two types of riveting
i) cold riveting.
ii) Hot riveting.

→ Types of Riveted joints :-
1. Lap Joint
2. Butt joint

→ Lap Joint :-

A lap joint is that in which one plate overlaps the other and the two plates are then riveted together.

→ Butt Joint :-

A butt joint is that in which the main plates are kept in alignment butting each other and a cover plate is placed either on one side or on both sides of the main plates.

→ Butt joints are of the following two types :-
1. single strap butt joint.
2. double strap butt joint.

1. In a single strap butt joint, the edges of the main plates butt aganist each other and only one cover plate is placed on one side of the main plates and then riveted togeter.

2. In a double strap butt joint, the edges of the main plates butt aganist each other and two cover plates are placed on both sides of the main plates and then riveted together.

→ Important terms used in riveted Joints.

1. Pitch
2. Back pitch
3. Diagonal pitch
4. Margin or marginal pitch.

1. Pitch :-

It is the distance from the centre of one rivet to the centre of the next rivet measured parallel to the seam. It is usally denoted by p.

2. Back pitch :-

It is the perpendicular distance between the centre lines of the successive rows. It is usually denoted by P_b.

3. Diagonal pitch :-

It is the distance between the centres of the rivets in adjacent rows of zig-zag riveted joints. It is usually denoted by P_d.

4. Marginal pitch :-

It is the distance between the centre of rivet hole to the nearest edge of the plate. It is usually denoted by m.

Q. A double riveted lap joint is made between 15 mm thick plates. The rivet diameter and pitch are 25 mm and 75 mm respectively. if the ultimate stresses are 400 MPa in tension, 320 MPa in shear and 640 MPa in crushing

① find the minimum force per pitch which will rupture the joint.

b. If the above joint is subjected to a load such that the factor of safety is 4, find out the actual stresses developed in the plates and rivets. (42)

Soln:- Given data

$t = 15$ mm, $d = 25$ mm; $p = 75$ mm, $\sigma_{tu} = 400$ MPa $= 400\ N/mm^2$

$\tau_u = 320$ MPa $= 320\ N/mm^2$

$\sigma_{cu} = 640$ MPa $= 640\ N/mm^2$.

a) Minimum force per pitch which will rupture the joint.

$$P_{tu} = (p-d)t \times \sigma_{tu} = (75-25)15 \times 400 = 300000\ N$$

ultimate shearing resistance of the rivets per pitch

$$P_{su} = n \times \frac{\pi}{4} \times d^2 \times \tau_u = 2 \times \frac{\pi}{4}(25)^2\, 320 = 314200\ N$$

ultimate crushing resistance of the rivets per pitch.

$$P_{cu} = n \times d \times t \times \sigma_{cu} = 2 \times 25 \times 15 \times 640 = 480000\ N$$

$$P_{min} = 300\ KN$$

b) Actual stresses produced in the plates and rivets.

Since the factor of safety is 4,

∴ Safe load per pitch length of the joint

$$= 300000/4 = 75000\ N$$

$$P_{ta} = 75000 = (p-d)\, t \times \sigma_{ta} = (75-25)\, 15 \times \sigma_{ta} = 750\, \sigma_{ta}$$

$$\sigma_{ta} = 75000/750 = 100\ N/mm^2 = 100\ MPa$$

$P_{sa} =$

$$75000 = n \times \frac{\pi}{4} \times d^2 \times \tau_a = 2 \times \frac{\pi}{4}(25)^2 \tau_a = 982\, \tau_a.$$

$\tau_a = 75000/982 = 76.4 \ N/mm^2 = 76.4 \ MPa.$ (43)

$(P_{ca}) =$

$75000 = n \times d \times t \times \sigma_{ca} = 2 \times 25 \times 15 \times \sigma_{ca} = 750 \ \sigma_{ca}.$

$\sigma_{ca} = 75000/750 = 100 \ N/mm^2 = 100 \ MPa$ //