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Chapter-1
Introduction.

Machine design:- It may be define as the creation of new and better machines. The existing machine are also made to improve. But these activities should be more economical. in overall cost of the production and operation.

Types of Machine Designs:-

1. New design.
2. Development design.
3. Adaptive design.

(1.) New designs:- It need a lot of technical ability, creativity and a lot of research work.

(2.) Development Designs:- It need considerable scientific training and design ability in order to modify the existing design into a new design.

(3.) Adaptive Designs:- Here the work of the designer is concerned with adaptation of existing design. No specific knowledge required.

(4.) Computer Adied Designs:- this type of design depends upon the use of computer system to creating and modification of design.

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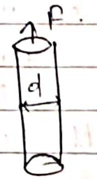
5. Empirical Design:- This type of design depends upon empirical formula based on the practice and past experience.



$$\sigma = \frac{F}{\frac{\pi}{4} d^2} \quad N/m^2$$

6. Element of Design:- In this design any element of mechanical system (for example connecting road piston crank shaft) are design.

7. Optimum design:- this type of design maximum weightage considered also keep its adverse effect.

8. Industrial design:- In this design industrial and production aspects to manufacturing any machine part in factory.

9. System Design:- It is the design of any complex mechanical system like train.

10. Rational Design:- This design depends upon mathematical formula of mechanics.

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Necessity of Machine Design:-

1. To convert the existing old designs into new design.
2. For faster production.
3. For automation of industries.
4. For innovation of new products.
5. To encourage the market with new ideas and technologies inserted in new designs.

Design Procedure:-

1. First of all prepare the drawing of the Job showing its use.
2. Draw the diagram of the Mechanism in simple form.
3. Make a written statement of the problem completely and clearly.
4. Select the mechanism of the ~~mechanism~~ M/C in simple form.
5. During machining operation, calculate the force acting on the mechanism

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Characteristics of a good Designer :-

1. Imagination :- A good designer should have good imagination power and it should be correct ~~logical thinking~~.

2. Logical thinking :- A good designer should have good have good logical thinking i.e. Better idea should come to his mind.

3. Good Memory :- A good designer must have good memory of design ~~added~~ and its application.

4. Good concentration :- A good designer should have good concentration so that he can produce good design work.

5. Good Judgment :- A good designer should Judge all the situation properly so that he take right design ~~des~~ Judgement.

6. Cooperative Nature :- A good designer should have cooperative nature i.e he can work in a team with coordination with his senior and ~~subordinate~~ subordinates.

7. Environmental knowledge :- A good designer should have good knowledge of Environment which will help in good Environ-mental friendly design.

8. Self confidence :- A good designer have self confidence so that he can perform his duties well with responsibility.

9. Sense of Responsibility :- he must have a good sense of Responsibility towards its work.

10. Positive attitude :- He should have good positive attitude towards the problem.

11. Good Communication Skills :- He should have good communication skill towards work.

12. Energetic :- He should have energetic towards work.

Comparision of Design work and undesign work

Design work	undesigned work
1. Better control cost	1. Lack of cost control.
2. Better quality control.	2. no control on quality
3. More output	3. less output.
4. more efficient	4. less efficient.

5. More reliable	5. less reliable.
6. Research and development is easy to conduct.	6. Research development will be difficult.
7. After sale service are better.	7. After sale service are poor.
8. Durability of product is high	8. Durability is less or low of product

Design Terminologies.

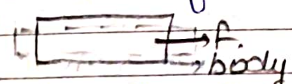
1. load :- Any external applied force is called load. (Newton) (N) ⇒ $F = ma$
dimensional formula $[M^1 L^1 T^{-2}]$

2. Classification of load :-

Acc^n to effort produce on the body.

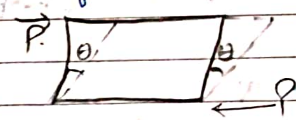
1. Tensile load.
2. shear load
3. compressive load.

1. Tensile load :- This increase the length of the body and its applied $\perp^r$ to the surface. normal to the surface.



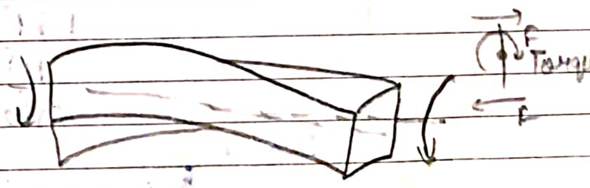
2. compressive load :- it decrease the length of the and applied normal to surface is called compressive load.

3. Shear load :- It causes sliding of one surface of the body relative to other and acts parallel to the surface.



4. Bending stress or load :- It produces certain curvature in the body is known as Bending load.

5. Twisting load :-

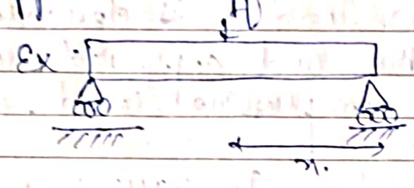


When two equal and opposite force or couple acts on two ends points of the bar the result will be twisting.

Acc^n to the Manner of Application of load :-

(1) Static load (Dead load) :- The magnitude direction and point of application is fixed.

Ex:



(2) Live load :- (Fluctuating load) :- Magnitude, direction and point where of application are not fixed.
Ex → Bridge or river.

Imp Stress :- Stress may be defined as internal resistance (force) per unit area offer by body against deformation. It denoted by σ which

$\boxed{\sigma = \frac{F}{A}}$ N/m^2 (Pascal) → SI unit.

Imp

1 Pascal = $1 N/m^2$.

$1 kPa = 10^3 N/m^2$

$1 MPa = 10^6 N/m^2$ or $1000000 N/m^2$

$1 GPa = 10^9 N/m^2$

Also.

$\boxed{1 MPa = 1 N/mm^2}$

1 bar = 10^5 Pascal for Pressure.

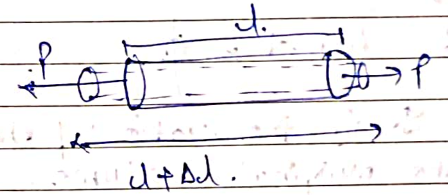
Types stress :-

(1) Tensile stress
(2) Compressive stress
(3) Shear stress.

} N/m^2

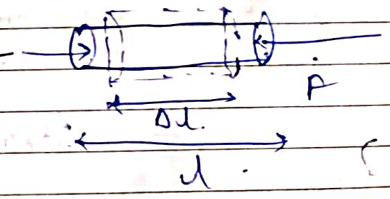
Strain :- It is ratio of change in dimension to original dimension.

(1) Tensile strain :- $\epsilon_T = \frac{\Delta l}{l}$ = $\frac{\text{Change in length}}{\text{original length}}$ = (Tensile Strain)



unit :- (unitless Because of its ratio).

(2) Compressive strain :-



$\epsilon_C = \left(\frac{\Delta L}{L}\right)$ (− ve denotes dimension has decrease)

No unit

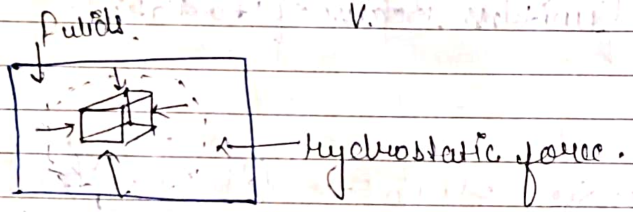
When axial push applied on the crossection area of the body its dimension decrease therefore the ratio of decrease in length and original length is called as compressive strain.

Shear strain :- When a body subjected to equal and opposite force, which are not in same line / non-collinear. It tend to shear of cross the resisting section.
the ratio of angular deformation to original length along the force is called shear strain therefore.

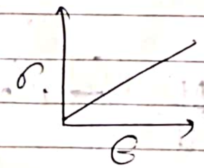
$$\epsilon_s = \tan\phi \approx \phi$$

Volumetric strain :- It is the ratio of change in volume to the original volume.

$$\epsilon_v = \frac{\Delta V}{V}$$



Hook's law :- Strain $\epsilon \propto$ stress σ (with in Elastic limit).



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Factor of safety :- It is the ratio ultimate stress to the working stress.

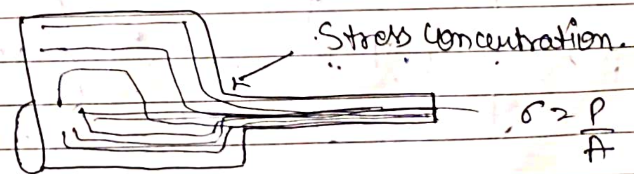
$$\frac{\text{Ultimate stress}}{\text{Working stress}}$$

unitless.

Factor affecting factor of safety :-

1. Type of load.
2. Frequency of loading.
3. Degree of safety required.
4. Degree of economy required.
5. dependability of structure.
6. life of structure.

Stress Concentration :- The shape of cross section of a machine component the simple stress change / the simple stress distribution is not valid. The irregular distribution due to sudden change in shape of the workpiece or component leads to stress concentration.



$$\sigma = \frac{P}{A}$$

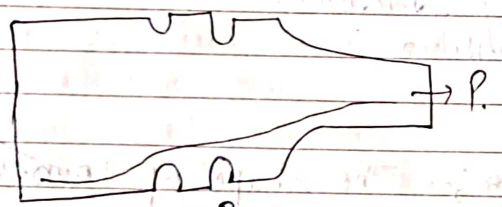
$$\sigma \propto \frac{1}{A}$$

Teacher's Sign ....

Methods to Reduces Stress Concentration :- Stress concentration can be elemenated by following methods by :-

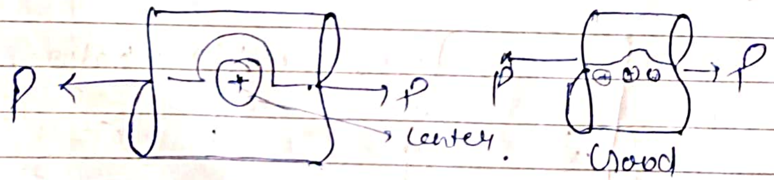
(1) Providing fillets.

Poor. good.



Best

(2) By providing holes in cylinderical member and components.

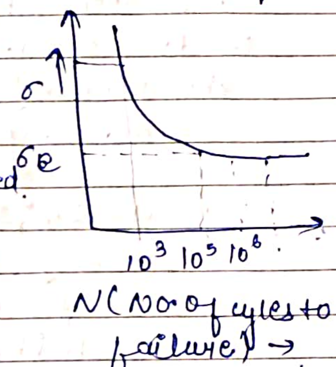


H.W. Mechanical properties of material, Genral consideration in M/c Design.

SN curve, fatigue limit or Endurance limit :-

It is ~~pure~~ plot b/w stress amplitude and the no. of cycles to failure.

* Fatigue limit or Endurance :-



When a m/c element or a part subjected cyclic loading its failure occurs much below the yield stress. such failure much below the yield stress Such failure is called fatigue limit.

If the the max cyclic stress is less sigma e i.e endurance limit the element will not break on infinite no. of cycles meet.

Selection of Material :-

→ It depends upon

- Abavailbility of Material.
- strength.
- Durability.
- Resistance to corrosion.
- Cost.
- Ease of Handling and M/c.

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Mechanical properties of Materials :-

(1.) Elasticity :- The ability of a material which has been deformed in some to return to its original shape and size after the removal of deforming force is called elasticity.

(2.) Plasticity :- The ability of a material to flow to new shape under pressure and to retain its new form is termed as plasticity.

(3.) Malleability :- The ability of a material to be deformed by being hammered, rolled or pressed without fracture through cold or hot working is termed as its malleability. example gold, silver etc.

(4.) Ductility :- The ability of a material to be drawn into wires without rupture and without losing much strength is known as ductility.

(5.) Toughness :- The property of a material to offer resistance to fracture under shock load such as hammer blows is called its toughness.

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(6.) Hardness :- The property of resistance to Scratching, abrasion, surface wear or indentation by harder bodies is called hardness.

(7.) Resilience :- The property of material by virtue of which it stores energy and resists impact and shock loads is called Resilience.

(8.) Fatigue :- The phenomenon that leads to fracture in metal and alloys under repeated fluctuating or alternating load or stress too small to produce permanent deformation when applied statically is called fatigue.

(9.) Creep :- The property of a material due to which it undergoes slow plastic deformation under prolonged loading, usually at high temperature is called creep.

(10.) Strength :- The ability of the material to withstand or supported external loads without any structural damage is called its strength.

Jyeeta

* Codes and standards :-
A code is a set of specificate or precisions made for the purpose of analysis, design and manufacture of product.

Standard is a set of specification made in oder to bring uniformly efficiency and quality.

In India standardisation and codes are given by UIS that is ______ of Indian standard.

BIS :-
ASME :- American society for mech. engineers.
ATTM :- Americian society for testing and material.

m.

Nominal stress :- It is the ratio of maximum stress to nominal stress.

$$K_p = \frac{\text{Maximum Stress}}{\text{Nominal stress}}$$

Theory of failure :- When member is subjected to uniaxial stress the prediction of its failure is simple but when the member is subjected to biaxial or triaxial (complex) so as a large no. of theory are formulate. for this.

* General design considerations :-

1. Motion of part and stress induced by load.
2. Selection of material.
3. Form and size of component.
4. Friction resistance and lubrication.
Whenever there is a mechanical ~~two~~ contact b/w two part, friction comes into the picture. The loss of power due to friction reduces the efficiency.

5. Ecomic expect :- The cost of manufacturing and construction should be ecomically feasible.
6. Use of standard parts.
9. Safety.
7. Workshop facility.
8. No. of parts produced.

Theory of failure :-

1) Maximum principle stress theory (Rankine theory) :-

It consider the failure of ~~two~~ material will occurs when principle stress at any point in the corss section reaches a maximum value regardless of other stresses, the critical value of this stress for ductile material is yeild point stress, while for brittle materials it is ultimate stress this theory best suited for brittle material.

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

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Chapter-2.
Design Failure.

Principle stress :-

These mutually perpendicular planes which carry no shear stress are called principle planes. The direct stress along these planes are called principal stresses.

$\boxed{\sigma_t = \sigma_{yielding}}$ for ductile Material

$\boxed{\sigma_t = \sigma_{ultimate}}$ for Brittle Material.

beside Page.

(2.) Maximum shear stress theory :- also known as Guest theory / Tresca's theory.

Accⁿ to this theory the failure or yield at point in a member occurs when maximum shear stress reaches a magnitude equal to shear stress reaches a magnitude equal to shear stress at yielding point of the materials therefore

$$\tau_{max} = \tau_{yielding}.$$

↳ shear Stress.

This theory is used for ductile material.

Spiral

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(3) Maximum Principle strain theory :-

$\epsilon = \frac{\Delta l}{l}$

This theory also known as Saint Venant's theory.

acc. to this theory the yielding at any point in the material begins when the strain in material exceeds the strain corresponding to yielding point in biaxial state of stress.

...lding. $\sigma \propto \epsilon$

$\sigma = E\epsilon$.

...lding $\geq \epsilon = \frac{\sigma_x}{E} - \frac{\mu \sigma_y}{E}$

...gy theory :- It also ...ory the failure ...oint in a member ... energy per unit ...tting is strain ... energy yield ...e of Material ...tile material.

$\sigma_x^2 + \sigma_y^2 - 2\sigma_x\sigma_y\mu$

...ergy at yielding.

...itude ...gnitude ...point

...lding. ...g ...e material.

Strain energy $= \frac{\sigma^2}{2E} \times \frac{\text{Volume}}{\text{Volume}}$

(5) Maximum distortion energy theory :- also known as von mises theory in this theory the failure or yielding at a point of a material will occurs w... the maximum distortion strain energy per unit volume absorb by the materia... is equal to yield point stress as obtained by simple tensile test (for ductile)

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ϵ yielding. $\sigma \propto \epsilon$
$\sigma = E\epsilon$.

$$\epsilon \text{ yielding} \geq \epsilon = \frac{\sigma_x}{E} - \frac{\mu \sigma_y}{E}$$

(4) Maximum strain energy theory :- It also known Haigh's theory the failure or the yielding a point in a member in occurs when strain energy per unit volume reaches limiting strain energy that is strain energy yield point present volume of Material it is preffered for ductile material.

Strain energy per unit volume $= \frac{1}{2E}\left[\sigma_x^2 + \sigma_y^2 - 2\sigma_x \times \sigma_y \mu\right]$

is less than < strain energy at yielding for safe design.

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ϵ yielding . $\sigma \propto E$

$\sigma = E\epsilon$.

ϵ yielding $\geq \epsilon = \frac{\sigma_x}{E} - \frac{\mu \sigma_y}{E}$.

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Chapter-2

Design shaft

#Shafts:- A shaft is a rotating element of circular cross section used for power transmission the power is delivered to the shaft by trangential force or torque.

Type of shafts -

- M/C shaft
- Power transmitted shaft

(1) M/C shafts:- A M/C shaft is a integral part M/C itself for example crank shaft is M/C shaft.

(2) Power transmitted shafts:- this shaft are use to transferred power counter shaft line shaft a Example power transmission shaft. For Example propeller shaft, from the engine to wheel in the car.

Standard size dowsheets of transmission shafts:-

(1) Acc^n to length:- 5m, 6m, 7meter,
(2) Acc^n to diameter 25 mm to 60 mm, 60 mm tollowing

Material use for shafts:-

(1) high corrosion resistance.
(2) High wear Resistance.
(3) High strength
(4) good Machinability.
(5) good heat treatment properties.

Design of shafts:- Shaft subjected to twisting moment only we have,

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

Tersion equation.

Where T = Torque.
J = Polar moment of inertia.
G = Modulus of Rigidity.
θ = Angle of twist (Radian) (always.)
L = length of shaft
τ = Shear stress.
R = Radius / distance from centroidal axis / Neutral axis to the outermost fibre. and for shaft $r = \frac{d}{2}$. (d = diameter of shaft)

For solid shafts:-

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

$$\frac{T}{\frac{\pi}{\cancel{32}_{16}} d^{\cancel{4}3}} = \frac{\tau}{\cancel{d}/2}$$

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$$\frac{T}{\left(\frac{\pi}{16}d^3\right)} = \frac{\tau}{1}$$

$$\boxed{T = \frac{\pi d^3 \times \tau}{16}}$$ — (2)

For Hollow shafts :-

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

d_o = outer diameter of shaft.

$$\frac{T}{\frac{\pi}{\cancel{32}\,16}\left(d_o^4 - d_i^4\right)} = \frac{\tau}{\frac{d_o}{2}}$$

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

$$\tau = \frac{16\,T d_o}{\pi\left(d_o^4 - d_i^4\right)}$$ — (2)

Power, P

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

$$\boxed{\omega = \frac{2\pi N}{60}}$$

T = Torque.
N = Rpm.

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When shaft subjected to bending moment only :-

$$\frac{\sigma_B}{y} = \frac{M}{I} = \frac{E}{R}.$$

σ_B = Bending stress.
y = Distance from neutral axis / Axis.
R. outermost fibre from neutral axis.
M = Bending Moment
I = Moment of interia.
E = Modulus of elasticity.

M. Imp
i. Solid Shaft $\boxed{\sigma_b = \frac{32M}{\pi d^3}}$.

Hollow shafts :- on the hollow shaft $\frac{\pi \times \sigma_b \times d_o^3}{32}(1-K^4)$.

where $K = \frac{d_i}{d_o}$ d_i internal dia
d_o outer dia

$$\boxed{\sigma_b = \frac{32M}{\pi d_o^3(1-K^4)}}$$

Shaft Subjected to Bending Moment and Torsion Moment Both :- (Combined loading).

When shaft is subjected Bending moment & Torque as well, it become a case of complex state of stress. so we need theories of failure of shaft design. In case ductile material mild steel we use max.

{1 MPa = 10^6 ...
1 MPa = 1 N/mm²

shear stress theories.

when.
max principle stress theory therefore acc^y to max shear stress theory

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

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

$$\tau_{max} = \frac{16}{\pi d^3} \times \cancel{\frac{1}{2}} \times \cancel{2}\sqrt{M^2+T^2}$$

Imp.

$$\boxed{\tau_{max} = \frac{16}{\pi d^3}\sqrt{M^2+T^2}}$$

Imp.
Qu1. Find the suitable dia of shaft required to transfer power 440 KW at 60 rpm. τ_{max} is 100 mega pascal $\theta = 1°$, length 1.5 meter if the shaft ~~As 25~~ T_{max} have 25% of mean torque. $G = 8.5 \times 10^4$ Mega pascal? find $d = ??$

$$\tau = \frac{16T}{\pi d^3} \quad -(1)$$

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

$$T = \frac{P \times 60}{2\pi N} = \frac{\overset{220}{\cancel{440}} \times 10^3 \times \cancel{60}}{\cancel{2} \times \pi \times \cancel{60}}$$

$$= \frac{220 \times 10^3}{\pi}$$

$$= 70.06 \times 10^3 \, Nm.$$

$T_{mean} = 70.06 \times 10^3$ N-m

$T_{max} = 1.25 \times 70.06 \times 10^3$
$= 87.57 \times 10^3$ N.m.

$$\tau = \frac{16T}{\pi d^3} \quad -(1)$$

$$100 \times \frac{10^6}{10^3} = \frac{16 \times 87.5 \times 10^3}{\pi \times d^3}$$

$$d^3 = \frac{16 \times 87.5}{\pi \times 100 \times 10^3}$$

$$d = \sqrt[3]{\frac{16 \times 87.5}{\pi \times 100 \times 10^3}}$$

$d = 0.164$ mm.
$= 164$ mm.

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

$$\frac{87.5 \times 10^3}{\frac{\pi}{32}d^4} = \frac{8.5 \times 10^4 \times 10^6 \times \pi}{1.5 \times 180}$$

$$\frac{87.5 \times 10^3}{\pi \times d^4} \times 32 = \frac{8.5 \times 10^4 \times 10^6 \times \pi}{1.5 \times 180}$$

$$\frac{87.5 \times 32 \times 1.5 \times 180}{\pi^2 \times 8.5 \times 10 \times 10^6} = d^4$$

$d = 0.17326$ Meter
$d = 173$ mm approx

Acc^n to MPST (Maximum principle stress theory) in shaft for Brittle Materials:-

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

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

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

$$(\sigma_b)_{max} = \frac{16}{\pi d^3}\left[M + \sqrt{M^2 + T^2}\right].$$

∴ Equivalent Bending Moment

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

Also Equivalent Twisting Moment

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

Note:- $1^\circ = \frac{\pi}{360}$ Radian.

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

A shaft transmit a power of 100 kwat, 200 rpm. if the allowable shear stress $\tau_{max} = 60 N/mm^2$ find suitable diameter of shaft when the shaft do not twist more than 1°. length of shaft is 3 meter, modulus of Rigidity $G = 810 \times 10^3$ MPa or 80 GPa?

$$\frac{T}{J} = \frac{\tau}{R} = \text{strength}.$$

$$\frac{G\theta}{L} = \frac{T}{J} = \text{rigidity}.$$

$\therefore J = \frac{\pi}{32} d^4 \rightarrow$ solid shaft

$J = \frac{\pi}{32}(d_o^4 - d_i^4) \rightarrow$ Hollow shaft

The maximum stress in a propeller shaft of external diameter is 500mm internal dia is 250mm. Subjected to twisting moment of 500 N-m the Modulus of Rigidity is 90GPa How much is the twist in the length which is 25 times the External diameter?

Ans. $d_o = 500mm$

$d_i = 250mm$

$T = 5000\ N\text{-}m$

$G = 90GPa$

$l = 25 \times d_o$

$l = 25 \times 500$

$= 12500$

$12.5\ m$

$\tau_{max} = ?$

$\theta = ?$

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

$$J = \frac{\pi}{32}(d_o^4 - d_i^4)$$

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

$$\tau = \frac{16T \times \cancel{d_o}}{\pi(d_o^4 - d_i^4)}$$

Spiral

Teacher's Sign ....

$\tau_{max} = \cancel{\dfrac{16 T \times d_0}{\pi (d_0^4 - d_i^4)}}$

$\cancel{\tau_{max} \; 16 \times}$

$J = \dfrac{\pi}{32} (d_0^4 - d_i^4)$

$J = 5.752 \times 10^9 \, mm^4$

$\dfrac{5000 \times 10^3}{5.752 \times 10^9} = \dfrac{\tau}{250}$ → outer dia

$\tau = \dfrac{5000 \times 10^3 \times 250}{5.752 \times 10^9}$

$\boxed{\tau = .0.217 \, N/mm^2}$

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

$\dfrac{5000 \times 10^3}{5.752 \times 10^9} = \dfrac{90 \times 10^3 \times \theta}{12500}$

$\theta = \dfrac{5000 \times 10^3 \times 12500}{90 \times 10^3 \times 5.75}$

$\theta = \dfrac{6.25 \times 10}{90 \times 10^3 \times 5.75}$

$\cancel{\dfrac{1.21}{10000}}$ $\quad \theta_r = \dfrac{.500}{51750} \quad \cancel{6/75/}$

$\boxed{\theta = 1.21 \times 10^{-4}}$

Spiral Teacher's Sign ....

$1 N/mm^2 = 1 M Pascal = 10^6 \dfrac{N}{m^2}$ Pascal

$\theta = 0.000121$ radian.

$= \dfrac{0.000121 \times 180}{\pi} = 0.006$.

Que Find the diameter of solid shaft transmitting 19 kW of power at N=220 rpm and τ_{max} = 355 N/mm² and f.o.s is 8.5 if a hollow shaft is to be used in place of solid shaft find the diameter of hollow shaft in a diameter of shaft given that $\dfrac{d_i}{d_o} = 0.6$.

$\tau_{max} = 355 \, N/mm^2$

$N = 220$ rpm

$FOS = 8.5$.

$P = 19$ kW.

$\dfrac{d_i}{d_o} = 0.6$.

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

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

$19 \times 10^3 = \dfrac{2 \times \pi \times 220 \times T}{60}$

$19 \times 10^3 \times 60 = 2 \times \pi \times 220 \times T$

$\dfrac{19 \times 10^3 \times 60}{2 \times \pi \times 220} = T$

Spiral Teacher's Sign ....

$\tau_{max} = \frac{16T \times d_0}{\pi (d_0^4 - d_i^4)}$

$\tau_{max} \quad 16 x$

$J = \frac{\pi}{32}(d_0^4 - d_i^4)$

$J = 5.752 \times 10^9 mm^4$

$\frac{5000 \times 10^3}{5.752 \times 10^9} = \frac{\tau}{250}$ → outer dia

$\tau = \frac{5000 \times 10^3 \times 250}{5.752 \times 10^9}$

$\boxed{\tau = 0.217 N/mm^2}$

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

$\frac{5000 \times 10^3}{5.752 \times 10^9} = \frac{90 \times 10^3 \times \theta}{12500}$

$\theta = \frac{5000 \times 10^3 \times 12500}{90 \times 10^3 \times 5.75}$

$\theta = \frac{6.25 \times 10}{90 \times 10^3 \times 5.75}$

$\frac{1}{10000}$ $\theta_r = \frac{500}{\cancel{6175}\ 5175\ 00}$

$\boxed{\theta = 1.21 \times 10^{-4}}$

Teacher's Sign ....

Spiral

$\theta = 0.000121$ radian.

$= \frac{0.000121 \times 180}{\pi} = 0.006.$

Que Find the diameter of solid shaft transmitting 19KW of power at N=220 rpm and τ_{max} = 355 N/mm² and f.o.s is 8.5 if a hollow shaft is to be used in place of solid shaft find the diameter of hollow shaft in a diameter of shaft given that $\frac{d_i}{d_o} = 0.6$.

$\tau_{max} = 355 N/mm^2$

$N = 220 rpm$

$FOS = 8.5$

$P = 19 KW.$

$\frac{d_i}{d_o} = 0.6.$

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

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

$19 \times 10^3 = \frac{2 \times \pi \times 220 \times T}{60}$

$19 \times 10^3 \times 60 = 2 \times \pi \times 220 \times T$

$\frac{19 \times 10^3 \times 60}{2 \times \pi \times 220} = T$

Spiral

Teacher's Sign ....

Dt. Page

$T = 825.13\ N{\cdot}m.$

$\tau = \dfrac{16T}{\pi N^3}$

$41.76 = \dfrac{16 \times 825.13}{\pi d^3}$

$d = \sqrt[3]{\dfrac{16 \times 825.13 \times 10^3}{41.76 \times 3.14}}$

$d^3 = \dfrac{13202.08 \times 10^3}{131.1264}$

$d = \sqrt[3]{100.68209 \times 10^3}$

~~d = 46.5 mm~~

$d = \sqrt[3]{100682.09}$

$d = 46.5\ mm = 47\ mm$

$\dfrac{T}{J} = \dfrac{\tau}{r}$

$\dfrac{T}{\frac{\pi}{\cancel{32}_{16}}\left[d_o^4 - d_i^4\right]} = \dfrac{\tau}{\frac{d_o}{\cancel{2}}}$

Dt. Page

$\dfrac{16T}{\pi\left[d_o^4 - d_i^4\right]} = \dfrac{\tau}{d_o}$

$\dfrac{16T}{\pi d_o^{\cancel{4}3}\left[1 - \dfrac{d_i}{d_o^4}\right]} = \dfrac{\tau}{\cancel{d_o}}$

$\dfrac{10^3 \times 16 \times 825.4 = 41.76}{\pi d_o^3 (1 - (0.6)^4)}$

$\dfrac{16 \times 825.4 \times 10^3}{\pi (1 - (0.6)^4) \times 41.76} \cancel{= d_o^3} = d_o^3.$

$\dfrac{13206400}{114.1324186}$

$d_o^3 = 115711.2078.$

$d_o = 48.7\ mm.$

$d_i = 29.23\ m = 30\ mm,$

Dt. Page

Material for high strength shaft. — Nickel steel
Chrome-vanadium steel
Nickel chromium steel.

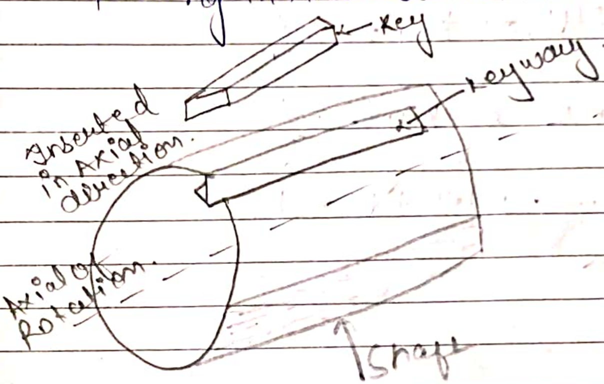
Dt. Page

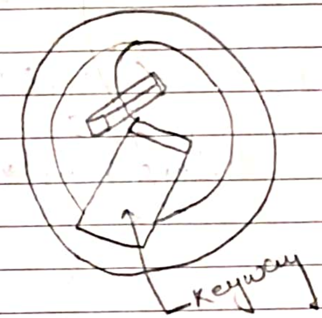
24/02/25.

Jay Shree Ram

Chapter-4. Unit - III
Design of Key.

Key:- A Key is a piece of Mild steel inserted b/w shaft and hub of mating part such as crank, pulling in Axial direction.

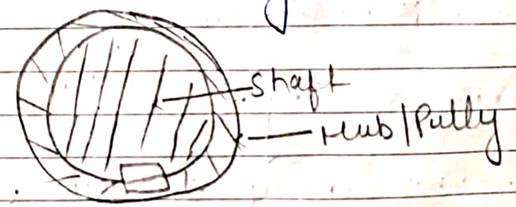




Types of Key:-

1. Sunk Key.
2. Saddle Key.

1. Sunk Key:- The type of key in which the key is partially in the key way of shaft and partial in the keyway of hub of mating member are called as Sunk key.



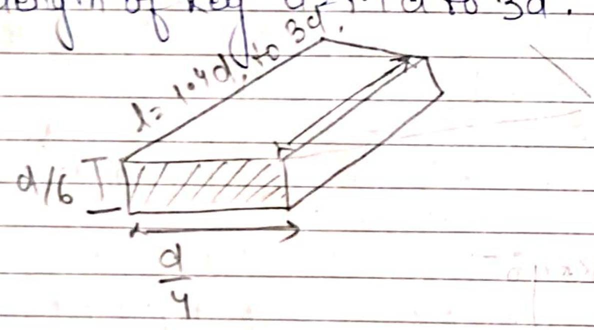
(a) Rectangular parallel sunk key :- In rectangular sunk key the area of cross-section is rectangular.

let diameter of shaft = d.

then width of key $w = \frac{d}{4}$.

Thickness of key $t = \frac{d}{6}$

length of key $l = 1.4d$ to $3d$.



There is no tapper on rectangular sunk key.

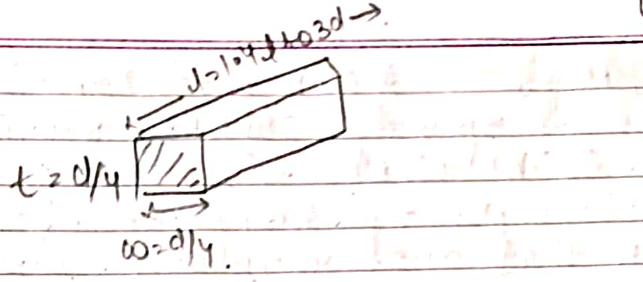
(b) Square sunk key :- In square of sunk key the area of cross-section depth = width (thickness).

Let diameter of shaft = d

then width of key $w = \frac{d}{4}$

Thickness of key $t = d/4$. (thickness)

there is no Tapper on square sunk key.



(c) Tapper sunk key In tapper sunk key cross-section may be rectangular or square. Its upper surface tapper slightly while ~~let diameter of shaft = d.~~ water surface is straight.

let diameter of shaft = d.

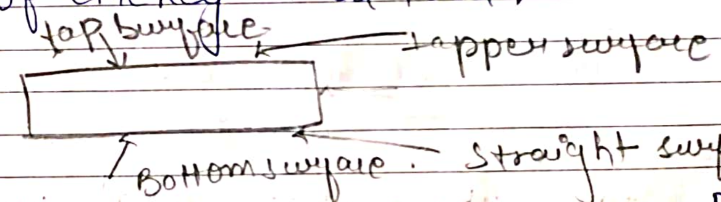
then width of key $w = \frac{d}{4}$

Thickness of key $= \frac{d}{4}$ [For square cross-section]

Thickness of key $t = \frac{d}{6}$. [for rectangular cross-section]

Tapper is 1 : n100 (1 : 100).

length of the key = 1.3d to 3d.



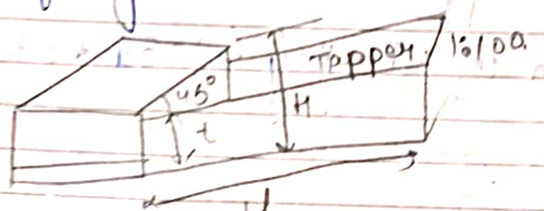
then tapper 1:100, then key is fitted ~~partial~~ partly in the Hub. the following are design parameter of the tapper sunk key.

(d) Gib head key :- A Gib head key is a rectangular tappered sunk key provided a gib head which gives its name. gib head. this type of key is use for the mating parts that are regularly separated for maintance or other purpose. the gib head key is easy to reduce inserted because of its concave construction and tapper.

Let diameter shaft $= d$.
the width of key $w = \frac{d}{4}$.

Thickness of over Headed $H = 1.75t$
Thickness of key $t = \frac{d}{6}$.

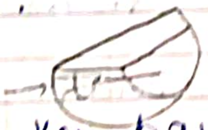
Tapper is 1 in 100 (1:100)
Length of key $= d$ to $1.5d$.



Wood ruff key :- wood ruff key is in the form of segment of wood ruff key. It is a circular disc of uniform thickness this type of key is mainly use on m/c and Automobile work.

Let diameter of shaft $= d$.
width of key $= d/4$

thickness $t = 0.4d$.
Length $= l = 1.5d$ to $3d$
Taper $= 1:100$.

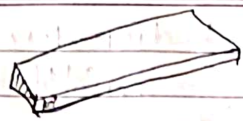


(f) Round key :- this type of key have circular cross section these type of key are fitted in drill holes partial in the shaft partially in Hub the dimension of Round key are as below

Let diameter shaft $= d$.
Diameter of key $d_1 = 0.16d$
Length $= l = 1.4d$ to $3d$.
Tapper 1:100.

(g) Saddle key :- A saddle key is simply mounted on the shaft and inserted in the key way on the hub. there is no keyway on the shaft the saddle key is Generally use for light power transmission. therefore No requirement deliated key slot on the shaft.
It is of two type.
1) Flat saddle key
2) hollow saddle key.

1.) Flat saddle key :-



d = diameter of shaft

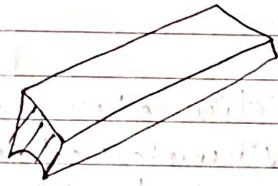
width of key $w = \frac{d}{4}$

thickness of key $t = \frac{d}{12}$

length of key $l = 1.4d$ to $3d$.

Taper is same = 1:100

2.) Hollow saddle key :-



(1.) this type of key is use for very light loads.

(2.) It is use in cams as temprary. It is bottom face such as that the design parameter are as follows.

d = diameter

width of key $w = \frac{d}{4}$

thickness of key $t = \frac{d}{12}$

length of key $l = 1.4d$ to $3d$.

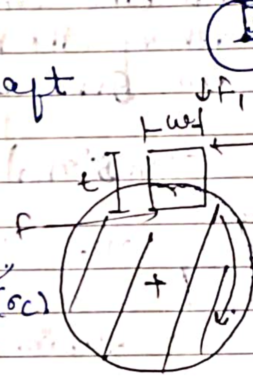
taper = 1:100.

Strength of ~~of~~ sunk key :- ~~let the T~~

let T = Torque transmitted by shaft

d = diameter of shaft.

F = Tangential force on the shaft



τ = shear stress strength of shaft.

σ_c = crushing stress.

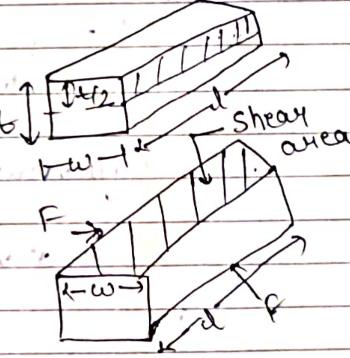
F = Area of resisting × crushing stress (σ_c)

$F = l \times \frac{t}{2} \times \sigma_c$

T = Torque = $F \times \perp^r$ distance.

$T = F \times \frac{d}{2}$

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



Therefore, the key is equal strong in shearing and crushing if.

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

$\frac{\sigma_c}{\tau} = \frac{2w}{t}$ or $\boxed{\frac{w}{t} = \frac{\sigma_c}{2\tau}}$

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Page

therefore the crushing stress for all key material is atleast twice the allowable shear stress. If ~~ratio~~ we take width = thickness (square cross-section)

To find the length of the key. The shear strength of the key should be equal shear strength of the shaft.

Torsional strength of key.

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

Torsional strength of shaft

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

equating (1) and (2)

$$l \times w \times \tau \times \frac{d}{\cancel{2}} = \frac{\pi}{\cancel{16}_{8}} \times \tau_s \times d^{\cancel{3}\,2}$$

$$l = \frac{\pi}{8} \times \frac{\tau_s \times d^2}{\tau \times w}$$

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

$$l = 1.571 \times d \times \frac{\tau_s}{\tau}$$

Dt.
Page

If the material of shaft and key is same.

$$\tau_s = \tau$$

$$\boxed{l = 1.571 \times d}$$

Que 1. $T = 504 N/m^2$ for shaft.

$T = 2500 N/m$

$w = \frac{d}{4}$.

$L = 1.5D$ in shear.

$T = 42.0 MN/m^2$

$\sigma_c = 560 N/m^2$

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

$$d^3 = \frac{16 \times 2500 \times 10^{-6}}{50 \times 3.14}$$

$$\frac{40000}{157} = 254.77 \times 10^{-6} = 0.064 m$$

$$w = \frac{d}{4} = \frac{64}{4} = 16mm$$

for key. $F = 2500$.

$$T = F \times \frac{d}{2}$$

$$F = \frac{2500 \times 2 \times 10^3}{64} = 78125.0$$

$$\tau = \frac{F}{A} = \frac{78125}{w \times L} = \frac{78125}{16 \times 96}.$$

$\tau_{induced} = 50.86$ MPa

$\sigma_c = 2\tau = 50.8 \times 2$

f.o.s crushing $= \frac{560}{101.72} = 5.5$

f.o.s shearing $= \frac{420}{50.8} = 8.25$ Ans.

Que 2. A rectangular sunk key

$t = 12mm$

$l = 80mm$

is required to transfer torque of shaft

$T = 25$ kN/m

from 100mm dia shaft

$\tau_f = 100$ m

Induced shear and crushing

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

$t = \frac{d}{2}$

$6 \times 12 = d$

$72 = d$

$T = L \times w \times \tau$

$\therefore T = F \times \frac{d}{2}$; $T = 20 \times 10^3$ N/m.

$\tau_{induced} = \frac{F}{w \times l}$

key $\sigma_c = F / H_2 / l$

$T = F \times \frac{d}{2}$

$F = \frac{10^3 \times 2500 \times 2}{100} = 500000$

$F = 500000$

$\tau = \frac{25 \times 10^5}{6 \times 80} = 1041.66$ N/m

$\tau = \frac{5 \times 10^5}{16 \times 80} = 390.62$ Nmm.

Chapter-8.

Design coupling :- The process of Joining and connecting transmission of motion two or more shaft is called Design coupling.

Neccesity of coupling :- due to restriction of for transmission of power when distance b/w two point is large,

- To provide misalignment of to shaft and, also provide flexiability.
- To provide connection of shaft which are manufactured seprately.
- To provide protection against overloads.
- To reduce transmission shocks.
- To reduce vibration during power transmission.

Advantages of coupling :-

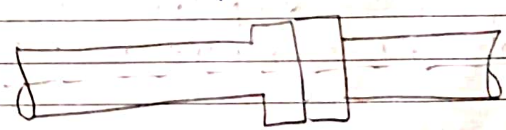
1. It is use to connect two or more shaft for power transmission.
2. It provide easy connection and disconnect of shaft
3. It provide ability of to Increase length of shaft.
4. Flange coupling provides accurate rigid and, strong connection of shaft.
5. Coupling provides correct alignment of shaft.
6. Flanges coupling is used for heavy loads.

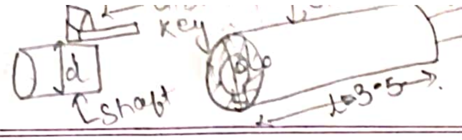
Types of coupling.

Rigid coupling
- Sleeve or muff coupling
- Clamping Coupling
- Flange Coupling
  - Non Protected type F.C
  - Protected type F.C
  - Marine Flange coupling

Flexible coupling
- Pin type Flexible
- Universal Coupling
- Oldham's Coupling.

Rigid coupling :- I this type of coupling provide perfect axial alignment of two connecting road shaft.





di = inner,
do = outer

Muff coupling or sleeve coupling :- • It is simplest type of coupling.

- It consist of hollow sleeve or muff of whose internal diameter D is same as shaft diameter.
- It is fitted over the end of two shaft by gib head key.
- this power is transmitted from one shaft to another shaft with help of key and sleeve.
- As the power is transmitted from one shaft is another shaft with of key and sleeve. it should be strong enough to transfer the desired torque.
- let out-let diameter of shaft sleeve $D = 2d + 13$. length of the sleeve is equal to 3.5 times of diameter where d = diameter of shaft.

Design of muff coupling shafts :-

let d = diameter of shaft
D = outer diameter of sleeve
L = length of sleeve
τ = permissible / Allowable shear stress,

Torque transmitted by sleeve $\rightarrow \dfrac{T}{J} = \dfrac{\tau}{\frac{do}{2}}$

$$T = \frac{\tau \times J}{\frac{do}{2}}.$$

$$T = \frac{\tau \times \frac{\pi}{32}(do^4 - d_i^4)}{\frac{do}{2}}$$

$$\boxed{T = \frac{\tau \times \pi(do^4 - d_i^4)}{16 \times do}}$$

*. length of ~~coupling~~ key = length of coupling (Note two keys are used)

Ques. Design a muff coupling / sleeve coupling which is use to connect to shaft transmitting 40KW of power at 400rpm the material use for key and shaft carbon steel having allowable shear stress of 42MPa and crushing stress of 84 MegaPascal the material of sleeve cast iron having allowable shear stress of 15MPascal?

Solu. $P = 40KW = 40 \times 10^3$.
$N = 400$ r.p.m
$\tau_s = 42MPa$ $42 N/mm^2$
$\tau_c = 84MPa$.
$\tau_c = 15MPa$ $15 N/mm^2$

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

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

$\frac{60 \times 40 \times 10^3}{2 \times \pi \times 400}$

$\frac{2400,00}{2513.27} = 954.93$

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

$954.93 = \frac{\pi}{16} \times 42 \times 10^6 \times d^3$

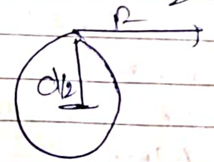
$d^3 = \frac{955 \times 16}{\pi \times 42 \times 10^6}$

$d = \sqrt{\frac{16 \times 955}{\pi \times 42 \times 10^6}} = 48.7 mm \approx$

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

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

$l = \frac{3.5d}{2} = 87.5$



$T = F \times \frac{d}{2}$

$F = \frac{955}{\frac{25}{1000}}$

$F = \frac{955 \times 10^3}{25}$

$F = \frac{38.2 \times 10^3}{25}$

$\boxed{F = 38.2 \times 10^3 N.}$

$\tau_{induced} = \frac{F}{w \times l} = \frac{38.2 \times 10^3}{12.5 \times 87.5} = 34.9 MPa$

$T = 955 Nm$

$\sigma_{cs} = \frac{F}{\frac{t}{2} \times l} = \frac{38.2 \times 10^3}{6.25 \times 87.5} = 69.8 MPa,$

sleeve design:-

$l = 3.5d = 175 mm$

$d_o = 2d + 13 = 113 mm$

$T = \frac{\pi}{16} \times \tau \times (d_o^4 - d_i^4)$

$T = \frac{\cancel{T} = 16 \times T \times d_o}{\pi(d_o^4 - d_i^4)},$

$N\text{-}mm\ mm = \frac{N\text{-}mm^2}{mm^4} = \frac{N}{mm^2} MPa$

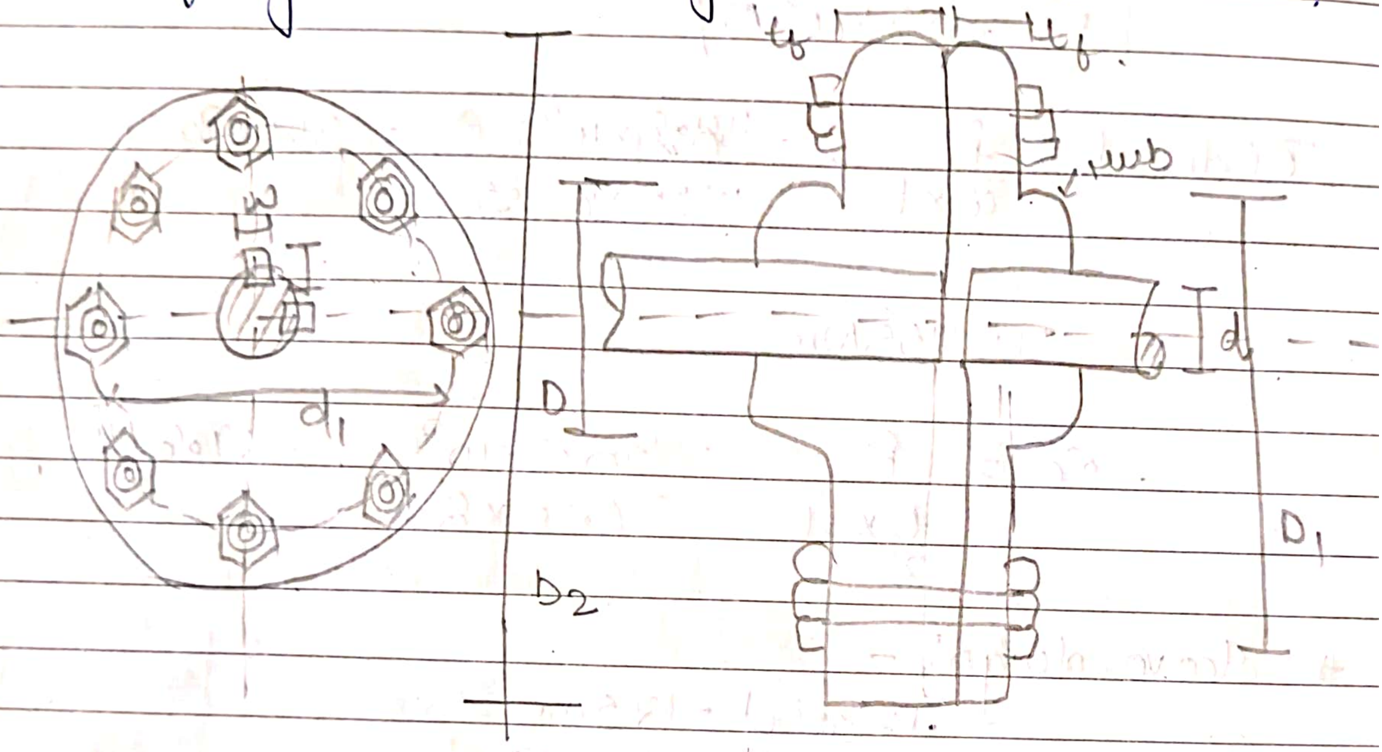
$\tau_{induced} = \frac{16 \times 955 \times 113 \times 10^3}{\pi(113^4 - 50^4)} = 3.5.$

$\tau_{permissible} = 13 MPa \rightarrow$ dus (safedesign),

* Flange couplings – It is made up of cast iron.

- It consist two flange, two shaft two key.

the flanges held together nut and bolt.



diameter of shaft = d.

diameter of hub D = 2d.

Length of hub L = 1·4d

Diameter of pitch circle D_1 = 3d.

Outer diameter of flanges D_2 = 4d.

Thickness of t_f = 0·5d.

No. of Bolt = n.

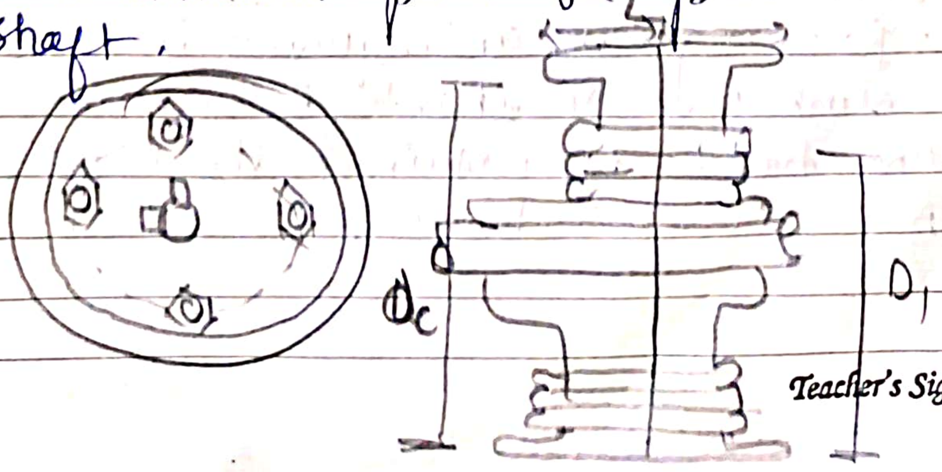
Dt.
Page

If $d = n = 3$ for up to 40mm
$= 4$ for 100mm
6 180mm.

τ_s Allowable shear stress of shaft
τ_b = " " " " Bolt
τ_K = " " " " Key

τ_c = " " " " Flange material
Cast iron.
σ_c = Allowable crushing of stress for key.
Bolt.

* Protected type Flange coupling
In ordinary flange coupling the bolt head are protected beyond flanges which are viable to ~~cast~~ cause injuries to the workers, material and m/c around them. to avoid All this protected type flanges coupling is used it consist of two flanges made up of caste iron and the shaft and key is made up of mild the up of ~~caste iron~~ ~~and the shaft and mild~~ key is provided to prevent relative motion and a taper of is to 100 is given on the key. the dimension of the extruded part for protection is $t_p = 0.25d$ of shaft.



Spiral
Teacher's Sign ....

Dt.
Page

Chapter-8 unit-IV
Design of weld Joints.

There are two types of Joints:-

(i) Permanent Joints.
(ii) Temporary Joints.

(i) Temporary Joints:- The Joint can ~~assemble~~ disassemble without distroying the connected parts. for example Nut and Bolt Joints by Nut and shut Joint by nut screws, Joint pins, Joint by key.

(ii) Permanent Joints:- In this type Joint part can not be assemble easily without distroying for example Reviting welding Brazing, Soldering (Soldering < 400°C)

Weld Joints:- weld Joint is a permanent Joint application of made by with or without heating without or without ~~distroying~~ pressure force with or without use of filler material. Example Explosion welding use of without heating.

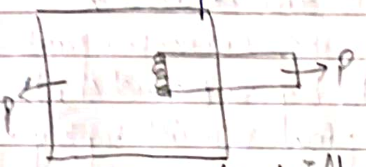
weld Joint

- Lap Joint or fillet Joint
- Butt Joint.

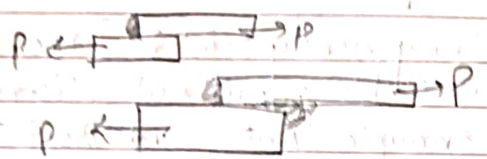
Lap Joint
- Single transverse. fillet Joint
- Double transverse. fillet Joint
- parallel fillet Joint

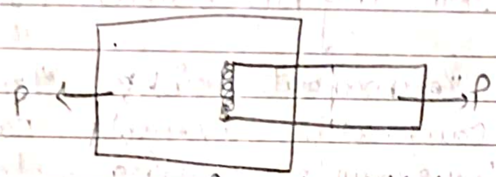
Butt Joint
- Double V-Butt Joint
- Square V butt Joint
- Single V-Butt Joint
- Single U Butt Joint
- Double U Butt Joint

Lap Joint :-



single fillet





Double fillet

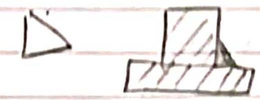
parallel fillet/Lap

Butt Joint :-

Squary Butt Joint

V Butt Joint

Basic weld Symbols :-

(1.) Fillet Butt 

(2.) Square Butt 

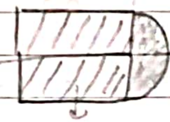
(3.) Single U Butt 

(4.) Double U Butt 

(5.) Single V-Butt 

(6.) Double V Butt 

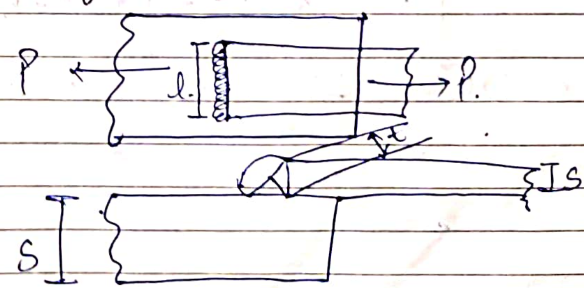
(7.) Eye Edge Joint 



(8.) Spot weld 

Strength of welded Joint :-

1. Case I for transverse fillet weld



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s = thickness of plate to be welded.
or
leg of weld.
t = throat thickness.

$t = s \times \sin 45°$
$t = s \times \frac{1}{\sqrt{2}} = 0.707s$

$\boxed{\text{throat thickness} = 0.707s}$

length of weld = l.

Now, Minimum Area of weld $A = t \times l = 0.707$

∴ length of weld.

Now if permissible Tensile Stress is.

$\boxed{l = \frac{P}{\sigma \times 0.707 \times s}}$

$P = \sigma \times A$
$\sigma \times 0.707 \times s \times l.$

Now, for double fillet weld

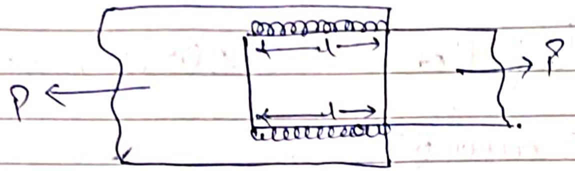
$P = \sigma \times 2 \times A$

$\boxed{P = \sigma \times 2 \times 0.707 \times s \times l.}$

Teacher's Sign ....

Dt. Page

Case 2 :- For parallel fillet weld.



Now minimum Area of weld $A = 2 \times t \times l = 0.707 s \times l \times 2$

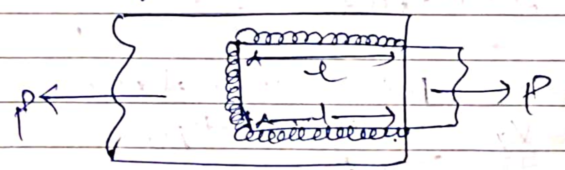
Now if the permissible shear stress is τ, maximum load.

$$P = \tau \times 0.707 \times s \times l \times 2$$

$$\boxed{l = \frac{P}{2 \times \tau \times 0.707 \times s}}$$

$$\boxed{P = (\sigma \times 0.707 s \times l) + (\tau \times 0.707 s \times 2l)}$$

Case 3. For combined parallel single transverse fillet weld.



P for transverse fillet.

$$P = \sigma \times 0.707 \times s \times l \text{ — (1)}$$

Now minimum Area of weld

$$A = 2 \times t \times l = 0.707 \times S \times l \times 2$$

Now if parallel fillet $P = \tau \times 0.707 \times S \times l \times 2$ — (2)

from eq (1) and (2)

$$\boxed{P_{net} = (6 \times 0.707 \times S \times l) + (\tau \times 0.70 \times S \times 2l)}$$

Qu. A Plate of 80mm wide and 12mm thickness is welded to another plate by double parallel fillet weld. the plate is subjected to 80KN find the length of weld if permissible shear stress 50MPa?

Ans. Re to $W = 80mm$

$S = 12mm$

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

$\tau = 50MPa$

$$P = 2 \times 0.707 S \times l \times \tau$$

$$80 \times 10^3 = 1.414 \times 12 \times l \times 50$$

$$l = \frac{80 \times 10^3}{1.414 \times 12 \times 50}$$

$$l = \frac{80 \times 10^3}{848.4} = 94.2 mm$$

$l = 95mm + 12.5 \rightarrow$ for weld stop.

$= 107.5 mm.$

Ques. A Plate 90mm wide and 12mm thick is to be weld to another plate by means of parallel fillet welds. The plate are subjected to a static load of 50KN. Find the length of the weld required so that maximum shear stress does not exceed 60MPa.

Ans. width = 90mm.

$S = 12mm.$

$P = 50KN = 50 \times 10^3 N.$

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

$l =$ length of weld.

Maximum load which the plates can carry for double parallel fillet welds,

$$P = 1.414 S \times l \times \tau$$

$$50 \times 10^3 = 1.414 \times 12 \times l \times 60 = 1018l.$$

$$l = 49mm$$

Adding 12.5 for starting and stoping of weld run,

$$l = 49 + 12.5 = 61.5mm.$$

Ques. A plate 120mm wide and 15mm thick is to be welded to another plate by means of single transverse and double parallel fillet weld. Determine the length of each parallel fillet weld if the Joint is subjected to load. Take allowable Stress in tension as 70 N/mm² and in shear as $60 N/mm^2$.

Sol^n. width $= 120mm$ $S = 15mm$ $\sigma_t = 76 N/mm^2$
$\tau = 60 N/mm^2$ $l_1 = 120 - 12.5 = 107.5 mm$
Let l_2 = length of each parallel fillet weld.

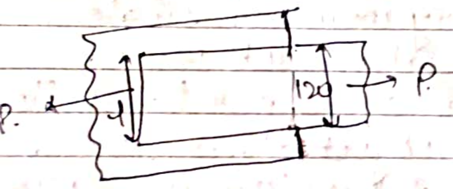
Maximum load to be carried by the plate.
$P = A \times \sigma$
$120 \times 15 \times 76$
$136800 N$.

Load carried by single transverse weld.

$P_1 = 0.707 S \times l_1 \times \sigma_t$
$0.707 \times 15 \times 107.5 \times 76$
$= 86642.85 N$

Load carried by double parallel filled weld.

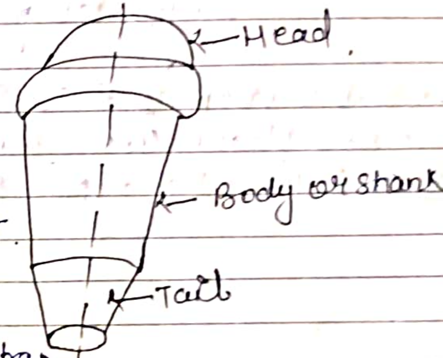
$P_2 = 1.414 S \times l_2 \times \tau$
$= 1.414 \times 15 \times l_2 \times 60$
$1272.6 l_2 N$
$P = P_1 + P_2$
$136800 = 86642.85 + 1272.6 l_2$
Adding 12.5mm for starting and stopping of weld.
$l_2 = 39.4 + 12.5 = 51.9mm$ say $52mm$
Ans.



Chapter-7

Design of Riveted Joint

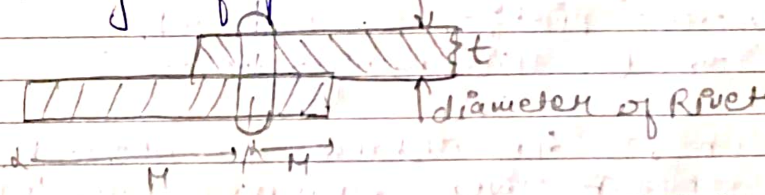
* Rivet:- A Rivet is a small simple road having a head at one end and tail another end, its made with forging the cylinderical portion is called Shank. It is made up of ductile, tough material. Example Aluminium, Bass, copper and nickel steel.



Rivet.

*. Types of Rivet Joints:-
- Lap Joint
- Butt Joint
  - Single Strap Butt Joint
  - Double Strap Butt Joint

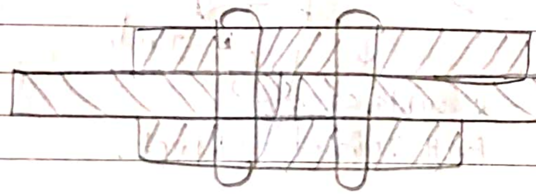
Lap Joint:- When edges of plates to riveted overlap on each other the joint is called lap Joint.
When edge of plate to riveted.



*. Butt Joint :- In Butt Joint two plates are placed side by side parallel and a additional plate is over them now Reviting is done this type of Butt Joint with single coverplate is known single strap riveted Joint.

Single strap Butt Joint.

*. Double strap Butt Joint :- In this Joint two cover plate are used to obtained riveted Butt Joint this type of Joint is known Butt Joint.

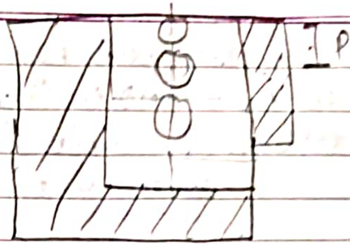


Double strap Butt Joint,

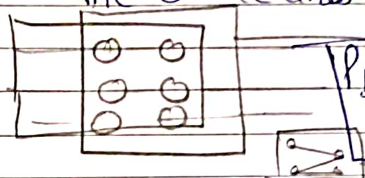
Terminology :-

1) Diameter of Rivet :- $d = 6\sqrt{t}$ (mm)
where t = thickness of plate (mm)

2) Pitch :- the distance b/w the centers of adjacent rivets measured parallel to seam. $\boxed{p = 3d}$ denoted by p.



3. Back Pitch :- the perpendicular distance b/w the centrelines of adjacent rows of rivets.



$P_b = 2d + 6$ (for chain type riveting)
$P_b = 2d$ (for zigzag riveting.)

4. Diagonal Pitch :- the distance b/w centres of rivets in adjacent rows in zig zag riveted Joints.

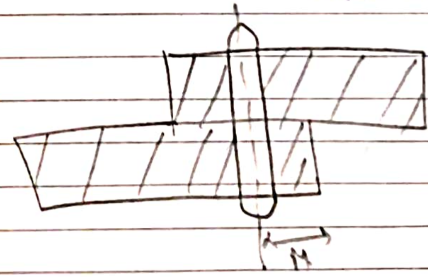
$$\boxed{P_d = \frac{2p + d}{3}}$$

5. Marginal Pitch or Margins :-

the distance b/w center of rivet to the nearest edge of the plate.

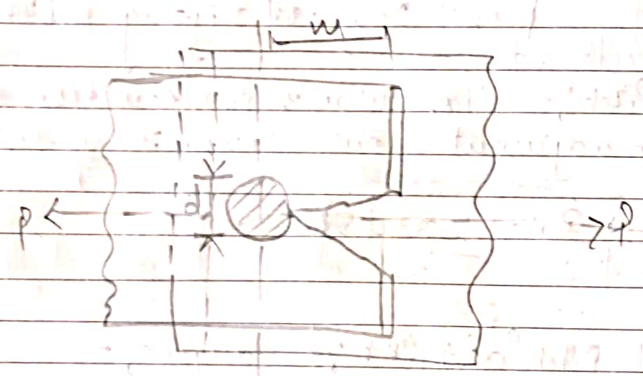
$$\boxed{m = 1.5d}$$

M means Margin



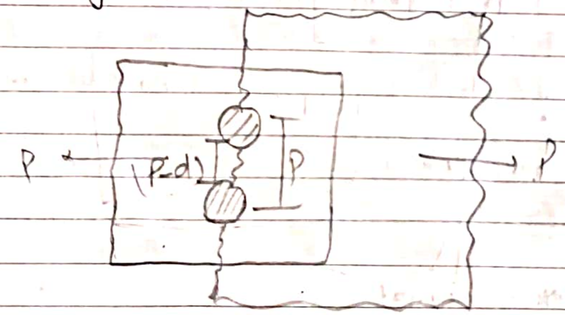
(5) thickness of cover plate :- the thickness of cover plate is equal to $t_1 = 1.125t$ (Single strap Butt Joint)
$t_2 = 0.625t$ (Double strap Butt Joint)

failure of Rivet Joint :- The Rivet Joint can be fail because of tearing of the plate at an edge Rivet Joint so to avoid margin is provided. $\boxed{m = 1.5d}$



Tearing of the Plate at an edge.

(i) Tearing of the Plate at an edge :-



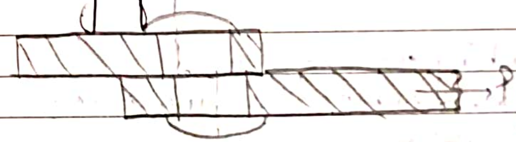
(ii) Tearing of the

Resistance offered by Plate against tearing is known as tearing Resistance.

$$P_t = A_t \times \sigma_t = (P-d) \times t \times \sigma_t$$

Where $= A_t =$ Area $= (P-d) \times t$
$t =$ thickness of plate
$P =$ Pitch of rivets
$d =$ diameter of Rivet hole.

(2) Shearing of Rivets :-

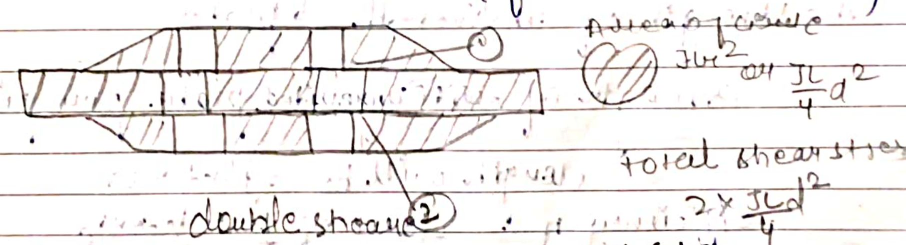


Shearing resistance = shear stress × shear Area

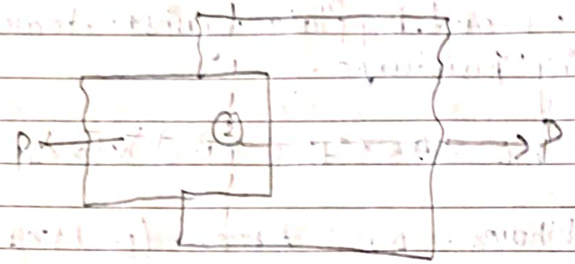
τ permissable/shear strength of material

$$\tau = \tau \times \frac{\pi}{4} d^4 \text{ (for single shear)}$$

(for double shear)



(4.) crushing of the Rivets :-



The resistance offered by a rivet to be crushed is called as crushing resistance or crushing strength,

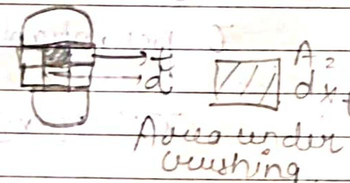
which is $P_c = n \cdot d \times t \times \sigma_c$

Where crushing Area = $d \times t$

σ_c = permissible crushing stress

t is thickness of plate.

N = No. of Rivets per pitch length.



$P_c = d \times t \times \sigma_c$

efficiency of Rivets Joint are define as ratio of strength of Rivets Joint to the strength of unriveted plate.

Strength of unriveted plate per pitch length will be = $P \times t \times \sigma_t$

efficiency η : $\dfrac{\text{least of } P_t \text{ or } P_s \text{ or } P_c}{P \times t \times \sigma_t}$

Ques Find the efficiency of Riveted Joint

(a) Single Riveted lap Joint thickness of plate 9mm 18m of dia of Rivets having pitch 54mm

(b) Double Rivited lap Joint for 9mm with 18 mm dia. and pitch is 60mm.

(c) The maximum permissible shear in rivets = 100MPa

(d) The Permissible crushing stress in rivets = 200MPA Permissible tensile stress in the plates = 150MPa.

$P_t = (P-d) t \times \sigma_t = (54-18) 9 \times 150 = 48600N.$

$P_s = n \times \frac{\pi}{4} d^2 = 1 \times \frac{\pi}{4} (18)^2 \times 100 = 25447N.$

$P_c = d \times t \times \sigma_c = 18 \times 9 \times 200$
$= 32400N$

Least strength of the Joint is 25447N.

$P = p \times t \times \sigma_t = 54 \times 9 \times 150 = 72900$

$\frac{25447}{72900} = 0.349 = 34.9\%.$

Efficiency of the second Joint :-

$P_t = (P-d) t \times \sigma_t = (60-18) 9 \times 150$
$= 42 \times 9 \times 150$
$= 56700N.$

$P_s = n \times \frac{\pi}{4} d^2 = 2 \times \frac{\pi}{4} d^2 \tau = 2 \times \frac{\pi}{4} \times (18)^2 \times 100$

$= 50893.8\,N$

$P_c = n \times d \times t \times \sigma_c = 2 \times 18 \times 9 \times 200$ $[\because n = 2]$

$= 64800\,N$

Least Strength Joint $= \underline{50893.8}$

$P = p \times t \times \sigma_t = 60 \times 9 \times 150 = 81000\,N$

$\frac{50893.8}{81000} = 0.628 \times 100 = 62.8\%$

Que A double riveted lap joint is to be designed for 18mm of plate the riveting is of chain type the permissible stress.

$\sigma_t = 100\,MPa$ $\tau = 85\,MPa$. $\sigma_c = 150\,MPa$.

find the efficiency.

given:- $\tau \cdot 85\,MPa$

$\sigma_c = 150\,MPa$

$\sigma_t = 100\,MPa$.

$t = 18mm$

$d = 6\sqrt{t} \cdot 6\sqrt{18} = 26$

$p = 3d = 3 \times 26 = 78$

Tearing resistance $P_t = (p - d) \times t \times \sigma_t$

$= (78 - 26) \times 18 \times 100 = 93600\,N$

$n = 2$

Shearing Resistance of Rivet $= n \times \frac{\pi}{4} d^2 \times \tau$

$2 \times \frac{\pi}{4} \times 676 \times 85$

$= 90246.7\,N.$

Chrushing Resistance of Rivet $n = 2$

$n \times d \times t \times \sigma_c$

$= 2 \times 18 \times 150 = 140400\,N$

Strength of unriveted plate $p \times t \times \sigma_c = 78 \times 18 \times 150$

$= 210600$

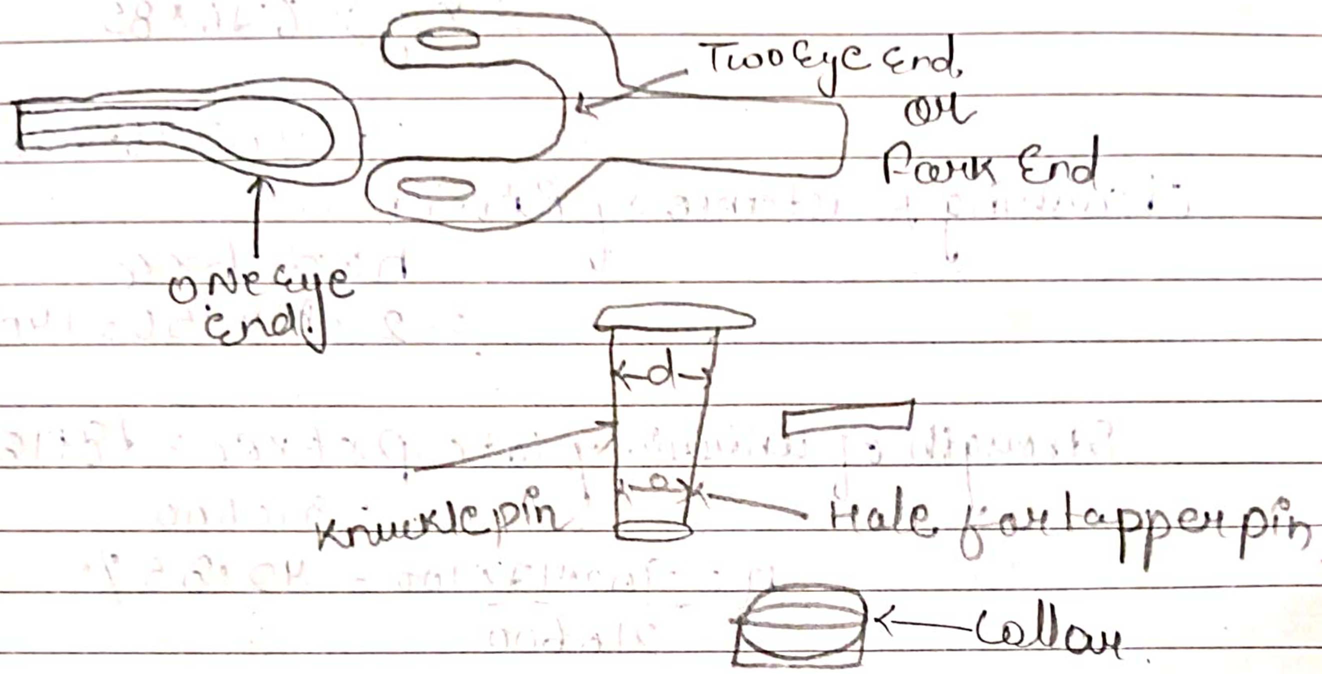
$\eta = \frac{90247 \times 100}{210600} = 42.85\%$

Caulking:- To make the riveted joints leak proof or fluid tight in pressure vessel like steam boiler etc, joint are made by. Caulking process it is done with help of caulking tool which is narrow bulnt.

It is 5mm thick and 40mm width.

Fullering:- The process of buring down holes of the thickness of the plate with the help of fullering tool. It is done to prevent. leakage of gases or fluid. the fluring tool is operated by with the of hammer ~~[illegible]~~. It is also called as ~~superseald~~ Superseald Caulking.

Design of Knuckle Joint :-



A knuckle Joint is use to connect two rods which may be in straight line or may not be straight line, the Joint have small Angular movement of one end related to another