

UNIT - I

The knowledge of materials and their properties is of great significance for a design engineer. The machine elements should be made of such a material which has properties suitable for the conditions of operation.

Classification of Engineering Materials.

The engineering materials are mainly classified as:

1. Metals and their alloys, such as Iron, steel, copper and aluminium etc.
2. Non metals such as glass, rubber, plastic, etc.

The Metals may be further classified as

- a) Ferrous metals,
- b) Non-ferrous metals.

The ferrous metals are those which have the iron as their main constituent, such as cast iron, wrought iron and steel.

The non-ferrous metals are those which have a metal other than iron as their main constituent, such as copper, aluminium, brass, tin, zinc, etc.

Selection of Material for Engineering purpose.

The selection of a proper material for engineering purpose, is one of the most difficult problem for the designer. The best material is one which serve the desired objective at the minimum cost. The following factors should be considered while selecting material.

- Availability of the materials.
- Suitability of the materials for the working conditions in service.
- The cost of the materials.

Properties of Materials

1. Physical Properties of metals:-

The physical properties of the metals include luster, colour, size and shape, density, electric and thermal conductivity, and melting point.

2. Mechanical Properties of metals:-

The mechanical properties of the metals are those which are associated with the ability of the materials to resist mechanical forces and loads. These mechanical properties of the metal include strength, stiffness, elasticity, plasticity, ductility, brittleness, malleability, toughness, resilience, creep and hardness.

→ Strength :- It is the ability of a material to resist the external applied forces without breaking or yielding. The internal resistance offered by a part to an externally applied force is called "stress".

→ Stiffness :- It is the ability of a material to resist deformation under stress.

→ Elasticity :- It is the property of a material to regain its original shape after deformation when the external forces are removed. This property is desirable for materials used in tools and machines. It may be noted that steel is more elastic than rubber.

→ Plasticity:- It is property of a material which retains the deformation produced under load permanently. This property of the material is necessary for forgings, in stamping images on coins and in ornamental work.

→ Ductility:- It is the property of material enabling it to be drawn into wire with the application of tensile force. A ductile material must be both strong and plastic. The ductility is usually measured by the terms, percentage elongation and percentage reduction in area. The ductile materials commonly used in engineering practice are mild steel, copper, aluminium, nickel, zinc, tin and lead.

→ Brittleness:- It is the property of material opposite to ductility. It is the property of breaking of material with little permanent distortion. Brittle materials when subjected to tensile loads, snap off forth out giving any sensible elongation. Cast iron is brittle material.

→ Mallability:- It is a special case of ductility which permits materials to be rolled or hammered into thin sheets. A malleability material should be plastic but it is not essential to be so strong. The malleability materials commonly used in engineering practice are lead, soft steel, wrought iron, copper and aluminium.

→ Toughness:- It is the property of a material to resist fracture due to high impact loads like hammer blows. The toughness of the material decrease when it is heated. It is measured by the amount of energy that a unit volume of the material has absorbed after being stressed upto the point of fracture. This property is desirable in parts subjected to shock and impact loads.

→ Machinability:- It is the property of a material which refers to a relative ease with which a material can be cut. The machinability of material can be measured in a number of ways such as comparing the tool life for cutting different materials or thrust required to remove the material at some given rate or the energy required to remove a unit volume of the material. It may be noted that brass can be easily machined than steel.

→ Resilience:- It is the property of a material to absorb energy and to resist shock and impact loads. It is measured by the amount of energy absorbed per unit volume upto its elastic limit. This property is essential for spring materials.

→ Creep:- When a part is subjected to constant stress at high temperature for a long period of time, it will undergo a slow and permanent deformation called creep. This property is considered in designing internal combustion engines, boilers and turbines.

→ Fatigue:- When a material is subjected to repeated stresses, it fails at stresses below the yield point stresses. Such type of failure of a material is known as fatigue. The failure is caused by means of a progressive crack formation which are usually fine and of microscopic size. This type of property is considered in the design of shafts, connecting rods, springs etc.

→ Hardness:- It is very important property of the metals and it embraces many different properties such as resistance to wear, scratching, deformation and machinability etc. It also means the ability of a (material) metal to cut another metal. The hardness is usually expressed in numbers which are dependent on the method of making the test. The hardness of metal may be determined

By the following tests.

a → Brinell hardness test.

b → Rockwell hardness test.

c → Vickers hardness test (called Diamond Pyramid test)

d → Shore scleroscope.

Manufacturing Processes:-

The knowledge of manufacturing process is of great importance for a design engineer. The following are the various manufacturing processes used in mechanical engineering.

1. Primary shaping Processes:- The processes used for the preliminary shaping of the machine component are known as primary shaping processes. The common operations used for this process are casting, forging, extruding, rolling, drawing, bending, shearing, spinning, powder metal forming, squeezing, etc.

2. Machining Processes:- The processes undergone used for giving final shape to the machine component, according to planned dimensions are known as machining processes. The common operations used for this process are turning, planning, shaping, drilling, boring, reaming, sawing, broaching, milling, grinding, hobbing, etc.

3. Surface-finishing Processes:- The processes used to provide a good surface finish for the machine component are known as surface-finishing processes. The common operations used for this process are polishing, buffing, honing, lapping, abrasive belt grinding, barrel tumbling, electro plating, super-finishing, sheradizing, etc.

4. Joining Processes:- The processes used for joining machine components are known as joining processes. The common operations used for this process are welding, riveting, soldering, brazing, screw fastening, pressing, sintering etc.

5. Processes effecting change in properties:- These processes are used to impart certain specific properties to the machine components so as to make them suitable for the particular operations or uses. Such processes are heat treatment, hot working, cold working and shot peening.

Casting :- It is the one of most important manufacturing process used in mechanical engineering. The castings are obtained by remelting of ingots in a cupola or some other foundry furnace and then pouring this molten metal into metal or sand moulds. The various important casting processes are as follows.

- Sand mould casting
- Permanent mould casting
- Slush casting
- Die casting

Advantages:- Die casting Advantages

- The production rate is high, ranging up to 700 casting per hour.
- It gives better surface smoothness.
- The dimensions may be obtained with in tolerance.
- The die retains its trueness and life for longer periods. For example, the life of a die for zinc base casting is upto one million castings, for copper base alloys upto 75000 castings and for aluminium base alloys upto 500000 castings.

- It requires less floor area for equivalent production than by other casting methods
- By die casting, thin and complex shapes can be easily produced.
- The holes up to 0.8 mm can be cast.

Disadvantages:-

- The diecasting units are costly.
- Only non-ferrous alloys are casted more economically.
- It requires special skill for maintain maintenance and operation of die casting machine.

Forging: It is the process of heating a metal to desired temperature in order to acquire sufficient plasticity, followed by operations like hammering, bending and pressing etc.

Mechanical working of metals:

The mechanical working of metals is defined as an intentional deformation of metals plastically under the action of externally applied forces.

The mechanical working of metal is described as hot working and cold working depending upon whether the metal is worked above or below the recrystallisation temperature. The metal is subjected to mechanical working for the following purpose.

- To reduce the original block or ingot into desired shapes.
- To refine grain size.
- To control the direction of flow line.

Hot working:-

The working of metals above the recrystallisation temperature is called hot working. This temperature should not be too high to reach the solidus temperature, otherwise the metal will burn and become unsuitable for use.

Advantages:-

- The porosity of the metal is largely eliminated.
- The grain structure of the metal is refined.
- The impurities like slag and sponges are squeezed into fibres and distributed throughout the metal.
- The mechanical properties such as toughness, ductility, percentage of elongation, percentage reduction in area, and resistance to shock and vibration are improved due to the refinement of grains.

Disadvantages:-

- It requires expensive tools.
- It produces poor surface finish, due to the rapid oxidation and scale formation on the metal surface.
- Due to the poor surface finish, close tolerance cannot be maintained.

Hot working Processes:-

1. Hot rolling
2. Hot forging.
3. Hot spinning
4. Hot extrusion
5. Hot drawing or cupping
6. Hot piercing.

Cold working

(5)

The working of metals below their recrystallisation temperature is known as cold working. Most of the cold working processes are performed at room temperature. The cold working distorts the grain structure and does not provide an appreciable reduction in size. It requires much higher pressure than hot working. The extent to which a metal can be cold worked depends upon its ductility. The higher the ductility of the metal, the more it can be cold worked. During cold working severe stresses known as residual stresses are setup. Since the presence of these stresses is undesirable, therefore, a suitable heat treatment may be employed to neutralise the effect of these stresses. The cold working is usually used as finishing operation, following the shaping of metal by hot working. It also increases tensile strength, yield strength and hardness of steel but lowers its ductility. The increase in hardness due to cold working process is called work-hardening.

Effects due to cold working processes;

- The stresses are setup in the metal which remain in the metal unless they are removed by subsequent heat treatment.
- A distortion of the grain structure is created.
- The strength and hardness of the metal are increased with a corresponding loss in ductility.
- The recrystalline temperature for steel is increased.
- The surface finish is improved.
- The close dimensional tolerance can be maintained.

Cold working processes:-

- 1. Cold rolling
- 2. Cold forging
- 3. Cold spinning
- 4. Cold extrusion
- 5. Cold drawing
- 6. Cold bending
- 7. Cold peening.

Interchangeability:-

The term interchangeability is normally employed for the mass production of identical items with in the prescribed limits of sizes. A little consideration will show that in order to maintain the sizes of the part with in a close degree of accuracy, a lot of time ~~is~~ is required. But even then there will be small variations. If the variations are within certain limits, all parts of equal equivalent size will be equally fit for operating in machines and mechanisms. Therefore, certain variations are recognised and allowed in the size of the mating parts to give the required fitting. This facilitates to select at random from a large number of parts for an assembly and results in a considerable saving in the cost of production.

In order to control the size of finished part part, with due allowance for error, for interchangeable parts is called limit system.

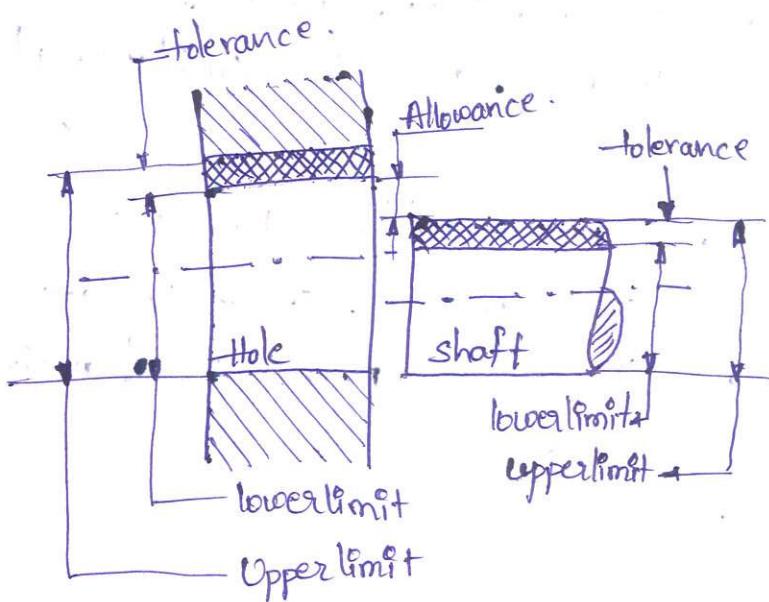
It may be noted that when an assembly is made of two parts the part which enters into the other, is known as enveloped surface (or shaft for cylindrical part) and the other in which one enters is called enveloping surface (or hole for cylindrical part).

Important terms used in Limit System:-

The following terms used in Limit System or Interchangeable system.

1. Normal Size:-

It is the size of a part specified in the drawing as a matter of convenience.



2. Basic Size:-

It is the size of a part to which all limits of variation (i.e., tolerances) are applied to arrive at final dimensioning of the mating parts. The normal or basic size of part is often the same.

3. Actual Size:-

It is the actual measured dimension of the part. The difference between the basic size and the actual size should not exceed a certain limit, otherwise it will interfere with the interchangeability of the mating parts.

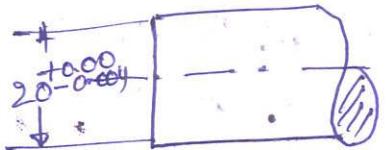
4. Limiting of Sizes:-

There are two extreme permissible sizes for a dimension of the part as shown in figure. The largest permissible size for a dimension of the part is called upper or high or maximum limit, whereas the smallest size of the part is known as lower or minimum limit.

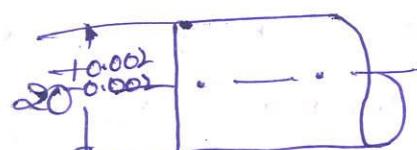
5. Allowance:-

It is the difference between the basic dimensions of the mating parts. The allowance may be positive or negative. When the shaft size is less than the hole size, then the allowance is positive and when the shaft size is greater than the hole size, then the allowance is negative.

6. Tolerance is the difference between the upper limit and lower limit of a dimension. In other words, it is the maximum permissible variation in a dimension. The tolerance may be unilateral or bilateral. When all the tolerance is allowed on one side of the nominal size, e.g. $20^{+0.000}_{-0.004}$ then it is said to be unilateral system of tolerance. The unilateral system is mostly used in industries as it permits changing the tolerance value while still retaining the same allowance or type of fit.



a) Unilateral tolerance



b) Bilateral tolerance.

When the tolerance is allowed on the both sides of the nominal size, e.g $20^{+0.002}_{-0.002}$,

* Theories of failure under static load

- * Maximum principal stress theory
- * Maximum shear stress theory
- * Maximum principal strain theory.
- * Maximum strain energy theory
- * Maximum distortion energy theory.

* Maximum principal or normal stress theory.

According to this theory, the failure or yielding occurs at a point in a member when the maximum principal or normal stress in a bi-axial stress system reaches the limiting strength of the material in a simple tension test.

Since the limiting strength for ductile materials is yield point stress and for brittle materials the limiting strength is ultimate stress, therefore according to the above theory, taking factor of safety into consideration, the maximum principal or normal stress (σ_i) in a bi-axial stress system is given by:

$$\sigma_i = \frac{\bar{\sigma}_t}{F.S}, \text{ for ductile materials}$$

$$= \frac{\bar{\sigma}_u}{F.S}, \text{ for brittle materials}$$

where

$\bar{\sigma}_t$ = yield point stress in tension as determined from simple tension test, and

$\bar{\sigma}_u$ = ultimate stress.

Since the maximum principal or normal stress theory is based on failure in tension or compression and ignores the possibility of failure due to shearing stress, therefore it is not used for ductile materials.

However, for brittle materials which are relatively strong in shear but weak in tension or compression this theory is generally used.

* maximum shear stress theory

According to this theory, the failure or yielding occurs at a point in a member when the maximum shear stress in a bi-axial stress system reaches a value equal to the shear stress at yield point in a simple tension test. Mathematically,

$$\tau_{\max} = \frac{\tau_{yt}}{F.S} \quad \dots \textcircled{1}$$

where

τ_{\max} = maximum shear stress in a bi-axial stress system.

τ_{yt} = shear stress at yield point as determined from simple tension test.

F.S = Factor of safety.

since the shear stress at yield point in a simple tension test is equal to one-half the yield stress in tension, therefore the equation (i) may be written as

$$\tau_{\max} = \frac{\sigma_y}{2 \times F.S}$$

This theory is mostly used for designing members of ductile materials.

* Maximum principal strain theory

According to this theory, the failure or yielding occurs at a point in a member when the maximum principal strain in a bi-axial stress system reaches the limiting value of strain as determined from a simple tensile test. The maximum principal strain in a bi-axial stress system is given by.

$$\epsilon_{max} = \frac{\epsilon_1}{E} - \frac{\epsilon_2}{m \cdot E}$$

∴ According to the above theory.

$$\epsilon_{max} = \frac{\epsilon_1}{E} - \frac{\epsilon_2}{m \cdot E} = \epsilon = \frac{\epsilon_y E}{F.S} \quad \text{--- (i)}$$

where

ϵ_1 and ϵ_2 = maximum and minimum principal stresses in a bi-axial stress system,

ϵ = strain at yield point as determined from simple tension test,

m = poisson's ratio

E = young's modulus, and

$F.S$ = Factor of safety.

From equation (i) we may write that

$$\epsilon_1 - \frac{\epsilon_2}{m} = \frac{\epsilon_y E}{F.S}$$

This theory is not used, in general, because it only gives reliable results in particular cases.

* Maximum strain energy theory

According to this theory, the failure or yielding occurs at a point in a member when the strain energy per unit volume in a bi-axial stress system reaches the limiting strain energy per unit volume as determined from simple tension test.

We know that strain energy per unit volume in a bi-axial stress system.

$$U_s = \frac{1}{2}E \left[(\epsilon_1)^2 + (\epsilon_2)^2 - \frac{2\epsilon_1 \times \epsilon_2}{m} \right]$$

and limiting strain energy per unit volume for yielding as determined from simple tension test,

$$U_2 = 1/2e \left[\frac{\bar{y}t}{F.S} \right]^2$$

According to the above theory, $U_1 = U_2$

$$\therefore 1/2e \left[(\bar{\epsilon}_1)^2 + (\bar{\epsilon}_2)^2 - \frac{2\bar{\epsilon}_1 \times \bar{\epsilon}_2}{m} \right] = 1/2e \left[\frac{\bar{y}t}{F.S} \right]^2$$

$$(\bar{\epsilon}_1)^2 + (\bar{\epsilon}_2)^2 - \frac{2\bar{\epsilon}_1 \times \bar{\epsilon}_2}{m} = \left[\frac{\bar{y}t}{F.S} \right]^2$$

→ This theory may be used for ductile materials.

* maximum distortion energy theory

According to this theory, the failure or yielding occurs at a point in a member when the distortion strain energy per unit volume in a bi-axial stress system reaches the limiting distortion energy per unit volume as determined from a simple tension test. Mathematically, the maximum distortion energy theory for yielding is expressed as.

$$(\bar{\epsilon}_1)^2 + (\bar{\epsilon}_2)^2 - 2\bar{\epsilon}_1 \times \bar{\epsilon}_2 = \left[\frac{\bar{y}t}{F.S} \right]^2$$

→ This theory is mostly used for ductile materials in place of maximum strain energy theory.

1. The load on a bolt of an axial pull of 10 kN together with a transverse shear force of 5 kN determine the diameter of the bolt. According to maximum principle stress theory, strain theory, strain energy theory, stress distortion theory. take permissible tensile stress at elastic limit 100 MPa. and Poisson's ratio = 0.3.

Given data:

$$\Rightarrow \sigma_t = \frac{\sigma_y t}{F.S}$$

$$\Rightarrow \gamma_{max} = \frac{\gamma_y t}{2 \times F.S}$$

$$\Rightarrow \frac{\sigma_{t1}}{\epsilon} - \frac{\sigma_{t2}}{\epsilon} = \frac{\sigma_y t}{\epsilon \times F.S}$$

$$\Rightarrow \frac{1}{2} \epsilon [(\sigma_{t1})^2 + (\sigma_{t2})^2 - 2 \times \frac{\sigma_{t1} \times \sigma_{t2}}{m}] = \frac{1}{2} t \frac{(\sigma_y t)^2}{F.S}$$

$$\Rightarrow (\sigma_{t1})^2 + (\sigma_{t2})^2 - 2 \times \sigma_{t1} \times \sigma_{t2} = \left[\frac{\sigma_y t}{F.S} \right]^2$$

$$\Rightarrow \sigma_{t1} = \frac{\sigma_1 + \sigma_2}{2} - \frac{1}{2} \sqrt{(\sigma_1 - \sigma_2)^2 + 4 \gamma^2}$$

$$\gamma_{max} = \frac{\sigma_{t1} - \sigma_{t2}}{2} = \frac{1}{2} \sqrt{(\sigma_1 - \sigma_2)^2 + 4 \gamma^2}$$

$$\text{tensile Stress } (P_t) = 10 \text{ kN} = 10 \times 10^3 \text{ N}$$

$$\text{tensile Stress } (\sigma_1) = P_t / A = \frac{10 \times 10^3}{\pi/4 d^2} \text{ N/mm}^2$$

$$= \frac{4 \times 10^4}{\pi d^2} \text{ N/mm}^2 = \frac{12732.33}{d^2} \text{ N/mm}^2$$

$$\text{Shear Stress} = \frac{\text{Shear Load}}{\text{Shear area}}$$

$$= \frac{5 \times 10^3}{\pi/4 d^2} \text{ N/mm}^2$$

$$\gamma = \frac{4 \times 5 \times 10^3}{\pi d^2} \text{ N/mm}^2 \Rightarrow \frac{6366.19}{d^2} \text{ N/mm}^2$$

$$\sigma_{t_1} = \frac{12732.33}{2d^2} + \frac{1}{2} d^2 \sqrt{\frac{(12732.33)^2}{d^2} + \frac{(6366.19)^2}{d^2}}$$

$$= \frac{12732.33}{2d^2} + \frac{1}{2} d^2 \sqrt{(12732.33)^2 + 4 \times (6366.19)^2}$$

$$= \frac{12732.33}{2d^2} + \frac{1}{2} d^2 \times 18006.412$$

$$\sigma_{t_1} = \frac{1}{2} d^2 (15369.235) \text{ N/mm}^2$$

$$\sigma_{t_2} = \frac{12732.33}{2d^2} - \frac{1}{2} \sqrt{\frac{(12732.33)^2}{d^2} + 4 \frac{(6366.19)^2}{d^2}}$$

$$= \frac{12732.23}{2d^2} - \frac{1}{2} d^2 \times 18006.412$$

$$\sigma_{t_2} = \frac{1}{2} d^2 (-2636.906) \text{ N/mm}^2$$

$$\begin{aligned}\gamma_{\max} &= \frac{\sigma_{t_1} - \sigma_{t_2}}{2} = \frac{1}{2} d^2 \left[\frac{15369.23 + 2636.906}{2} \right] \\ &= \frac{1}{2} d^2 \times 9003.0705 \text{ N/mm}^2.\end{aligned}$$

* According to maximum principle stress theory

$$\sigma_{t_1} = \frac{\sigma_y t}{F.S}$$

Stress at elastic limit = 100 MPa = 100 N/mm²

$$= \frac{15369.235}{d^2} = 100$$

$$d = \sqrt{\frac{15369.235}{100}} \text{ unit-1, pg-18/61}$$

$$d = 12.39 \text{ mm}$$

* According to maximum shear stress theory

$$\begin{aligned}\gamma_{\max} &= \frac{\gamma_y t}{F.S} \\ &= \frac{9003.0705}{d^2} = 100 \\ d &= 9.49 \text{ mm}\end{aligned}$$

* According to strain theory

$$\begin{aligned}\frac{\sigma_{t_1}}{\epsilon} - \frac{\sigma_{t_2}}{\epsilon} &= \frac{\sigma_y t}{\epsilon \times F.S} \\ &= \frac{1}{d^2} (15369.5 - (-2636.906)) \\ d &= 15.543 \text{ mm}\end{aligned}$$

* According to strain energy theory

$$\begin{aligned}(\sigma_{t_1})^2 + (\sigma_{t_2})^2 - \frac{2 \times \sigma_{t_1} \times \sigma_{t_2}}{m} &= \left[\frac{\sigma_y t}{F.S} \right]^2 \\ \frac{(15369.235)^2 + (-2636.906)^2}{d^2} - 2 \times 15369.2 \times (-2636.906) &= \frac{100^2}{0.3 \times d^4} \\ d^4 &\in [513348177.7] = 100^2\end{aligned}$$

$$d^4 = \frac{513348177.7}{(100)^2}$$

$$d^4 = 51334.817$$

$$= \sqrt[4]{51334.817}$$

$$d = 15.02 \text{ mm}$$

* maximum principle stress distortion energy theory

$$\left[\frac{1536.236}{d^2} \right]^2 + \left(-\frac{2636.906}{d^2} \right)^2 - 2 \times \frac{15369.236}{d^2}$$

$$X - \frac{(2636.906)}{d^2} = (100)^2$$

$$= \frac{1}{d^4} [3242113.7] = 100^2$$

$$d^4 = \frac{3242113.7}{100^2}$$

$$d^4 = 3242113.7$$

$$d = \sqrt[4]{3242113.7}$$

$$d = 13.41 \text{ mm}$$

completely reversed (or) cyclic stresses.

The stresses which are vary from one value of compressive to the same value of tensile or vice versa, are known as completely reversed or cyclic stresses.

- The stresses which are vary from minimum value to a maximum value in same nature (i.e tensile or compressive) are called fluctuating stresses.
- The stresses which are vary from zero to maximum value are called repeated stresses.
- The stresses which are vary from a minimum value to maximum value of the opposite nature (i.e from a certain minimum compressive to a certain maximum tensile or from a minimum tensile to a maxi compressive), are called alternating stresses.

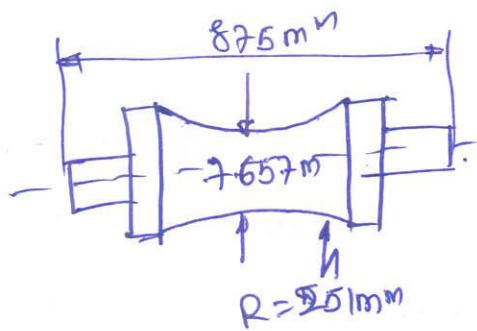
Fatigue and Endurance limit;

It has been found experimentally that when a material is subjected to repeated stresses, it fails at stresses below the yield point stresses.

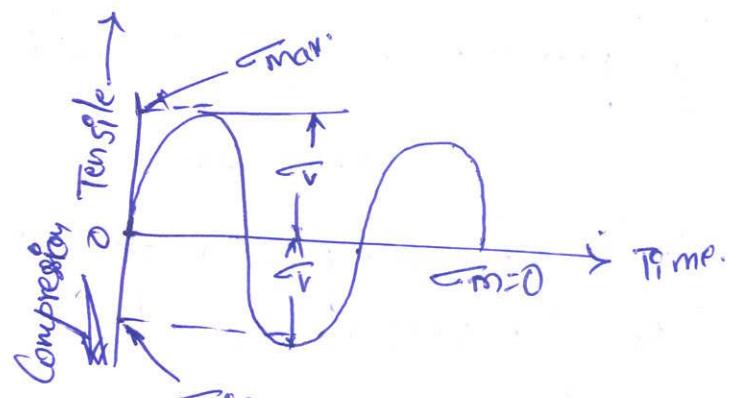
Such type of failure is known as fatigue.

Endurance limit or fatigue limit is defined as maximum value of the completely reversed bending stress which a polished standard specimen can withstand without failure for infinite no. of cycles. (usually 10^7 cycles)

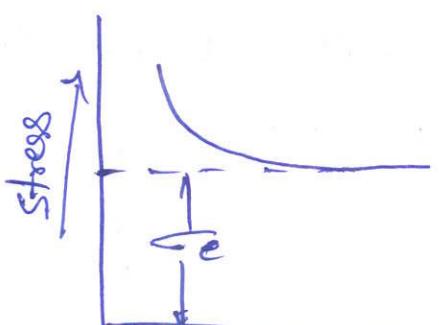
The term endurance limit is used for reversed bending only while for other type of loading, the term endurance strength may be used when referring the fatigue strength.



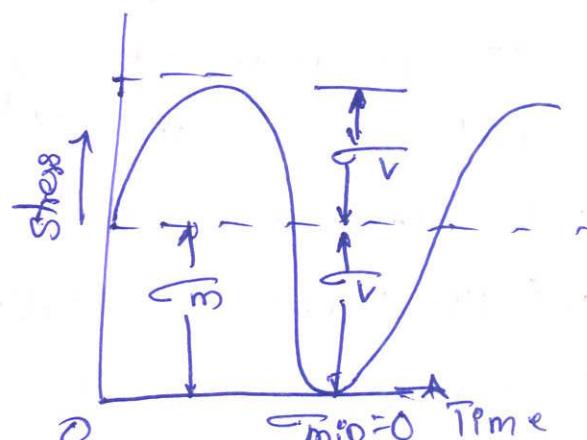
a) Standard Specimen



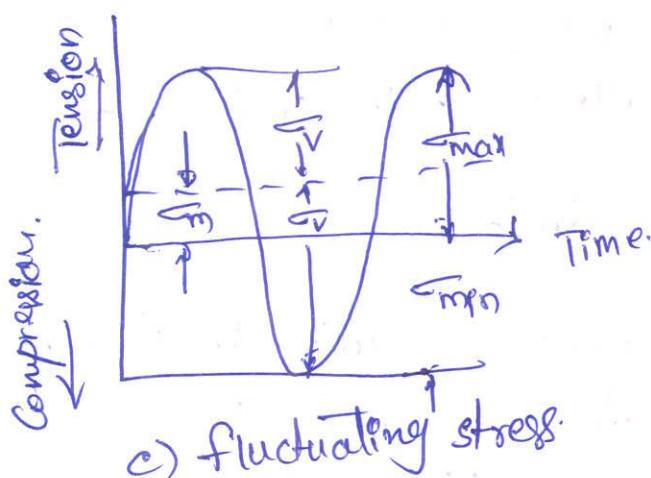
b) Completely reversed stress.



c) Endurance (or) fatigue limit.



d) Repeated stresses.



c) Fluctuating stress.

1. Mean (or) Average stress.

$$\bar{\sigma}_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

2. Reversed stress component (or) alternating (or) variable stress.

$$\bar{\sigma}_v = \frac{\sigma_{max} - \sigma_{min}}{2}$$

Note: For repeated loading, the stresses varies from zero to maxm. (i.e. $\sigma_{min}=0$)

$$\therefore \bar{\sigma}_m = \bar{\sigma}_v = \frac{\sigma_{max}}{2}$$

3. Stress ratio $R = \frac{\sigma_{max}}{\sigma_{min}}$

U. Relation b/w endurance limit and stress ratio.

$$\sigma_e' = \frac{3\sigma_e}{2-R}$$

σ_e' = Endurance limit for any stress range represented by R .

$$\sigma_e = " "$$

R = stress ratio.

Effect of loading on Endurance Limit - load factor

There are many machine members which are subjected to loads other than reversed bending loads. Thus endurance limit will differ for different loading.

Types of loads:-

K_b = load correction factor for the reversed bending load.
Its value is usually taken as unity.

K_a = load correction factor for the reversed axial load.
Its value may be taken as 0.8

K_s = load correction factor for reversed torsional or shear load, its value taken as 0.55 for brittle ductile material, 0.8 for brittle material.

∴ Endurance limit for reversed bending load $\sigma_e = \sigma_{eb} \cdot K_b$

$$\sigma_{eb} = e.$$

Endurance limit for reversed axial load $\tau_{ea} = \sigma_{eb} \cdot K_a$ [$\because K_b = 1$]

" " " " torsional load $\tau_e = \sigma_{eb} \cdot K_s$.

Effect of surface finish on Endurance Limit -

- Surface Finish Factor:

K_{SFR} = Surface Finish Factor.

for reversed bending load $\sigma_e = \sigma_{eb} \cdot K_{SFR}$

$$= \sigma_e \cdot K_b \times K_{SFR}$$

$$= e \times K_{SFR}.$$

for reversed axial load

$$e_c = \sigma_{ea} \cdot K_{SFR}$$

$$= e \times K_a \times K_{SFR}$$

for reversed torsional load

$$= \tau_e \times K_{SFR}$$

$$= e \times K_s \times K_{SFR}.$$

Effect of size on endurance limit - size factor.

K_{Siz} = size factor.

endurance limit for reversed

for reversed bending load = $\sigma_e \times K_{Siz}$.

$$= \sigma_e \times K_{D} \times K_{SW} \times K_{Sz}$$

for reversed axial load = $\sigma_e \times K_{Siz}$

$$= \sigma_e \times K_a \times K_{SW} \times K_{Sz}$$

for reversed torsional load = $\sigma_e \times K_{Sz}$

$$= \sigma_e \times K_{SW} \times K_{Sz}$$

$$= \sigma_e \times K_s \times K_{SW} \times K_{Sz}$$

Effect of miscellaneous factors on endurance limit

K_r = reliability factor

K_t = temperature.

K_i = impact factor

which are effecting on endurance limit.

1. for reversed bending load

$$\sigma_e' = \sigma_b \times K_{SW} \times K_{Sz} \times K_r \times K_t \times K_i$$

2. for reversed axial load.

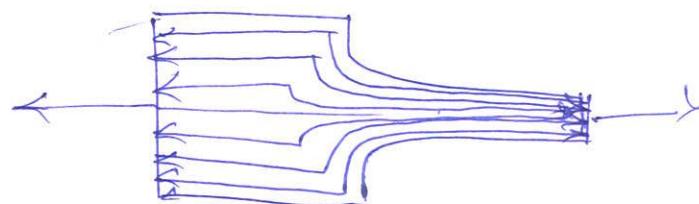
$$\sigma_e' = \sigma_a \times K_{SW} \times K_{Sz} \times K_r \times K_t \times K_i$$

3. for reversed torsional or shear load

$$\sigma_e' = \sigma_e \times K_{SW} \times K_{Sz} \times K_r \times K_t \times K_i$$

Stress Concentration
is

whenever a machine component changes the shape of its cross-section the simple stress distribution no longer hold good and neighbourhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration.



stress concentration

Theoretical or Form Stress Concentration Factor

it is defined as the ratio of the maximum stress in a member (at a notch or a fillet) to the nominal stress at the same section based upon net area.

$$k_t = \frac{\text{maximum stress}}{\text{Nominal stress.}}$$

The value of k_t depends upon the material and geometry of the part.

Fatigue Stress Concentration Factor

Fatigue Stress Concentration Factor denoted by k_f .

$$\therefore k_f = \frac{\text{Endurance limit without stress concentration}}{\text{Endurance limit with stress concentration.}}$$

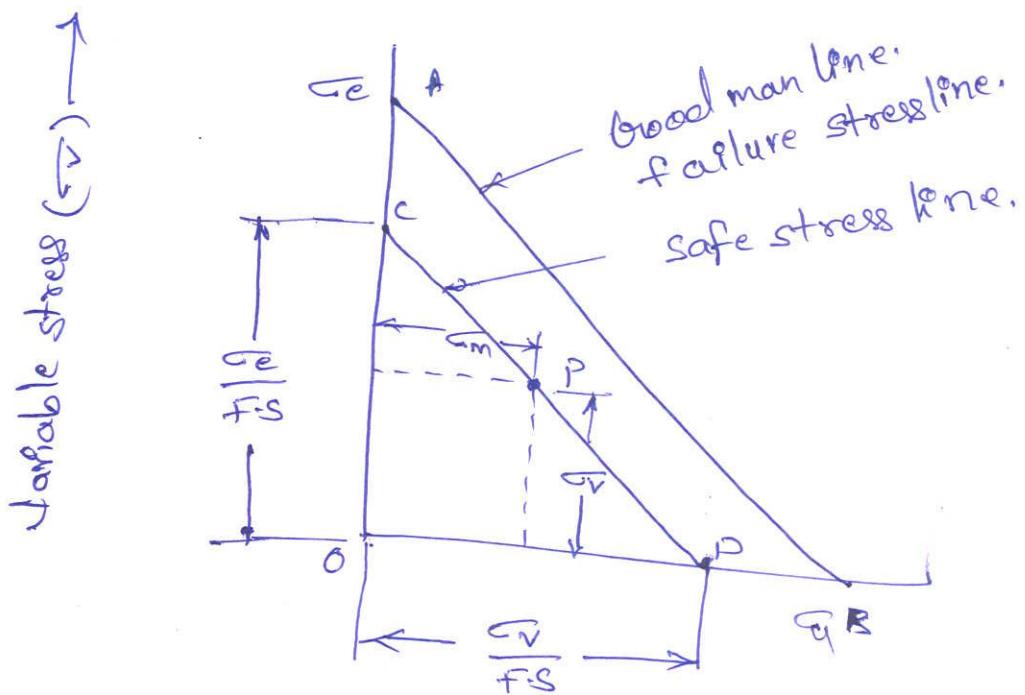
Notch Sensitivity

In cyclic loading, the effect of the notch or the fillet is usually less than predicted by the use of the theoretical factors as discussed before. The difference depends upon the stress gradient in the region of the stress concentrations and on the hardness of the material. The term notch sensitivity is applied to this behaviour, it may be defined as the degree to which the theoretical effect of stress concentration is actually reached.

Notch sensitivity factor q is used for cyclic loading, then fatigue stress concentration factor may be obtained from the following relation,

$$q = \frac{k_f - 1}{k_t - 1}$$
$$k_f = 1 + q(k_t - 1) \quad (\text{for tensile or bending stress})$$
$$k_{fs} = 1 + q(\cancel{k_t} - 1) \quad (\text{for shear stress})$$

Goodman Method for Combination of Stresses.



- A straight line connecting the endurance limit (σ_e) and the ultimate strength (σ_u), as shown in fig.
- follows the suggestion of good man. A good man line is used when the design is based on ultimate strengths and may be used for ductile or brittle materials.

In this criteria is used when ultimate stress is considered while designing. Line AB connecting σ_e and σ_u is called Goodman's failure stress line. If a suitable factor of safety (F.S) is applied to endurance limit and ultimate strength, a safe stress line CD may be drawn parallel to the line AB. Let us consider a design point P on the line CD.

Now from similar triangles COD and POQ

$$\frac{PO}{CO} = \frac{QD}{OD} = \frac{OD - OQ}{OD} = 1 - \frac{OQ}{OD}$$

$$\frac{\sigma_u}{\sigma_e} \frac{1}{F.S} = 1 - \frac{\sigma_m}{\sigma_u} \frac{1}{F.S}$$

$$\sigma_v = \frac{\sigma_e}{F.S} \left[1 - \frac{\sigma_m}{\sigma_u} \right] = \sigma_e \left[\frac{1}{F.S} - \frac{\sigma_m}{\sigma_u} \right]$$

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v}{\sigma_e}$$

Fatigue stress concentrations factor, k_f is used to multiply with variable stress.

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times k_f}{\sigma_e}$$

Consider other factors like load factor, surface finish factor, size factor the eqn may written as

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times k_f}{\sigma_e \times k_b \times k_{sur} \times k_{sz}} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times k_f}{\sigma_e \times k_b \times k_{sur} \times k_{sz}}$$

$F.S$ = Factor of safety

σ_m = mean stress.

σ_u = ultimate stress

σ_v = variable stress

σ_e = endurance limit.

k_f = fatigue stress concentrations factor

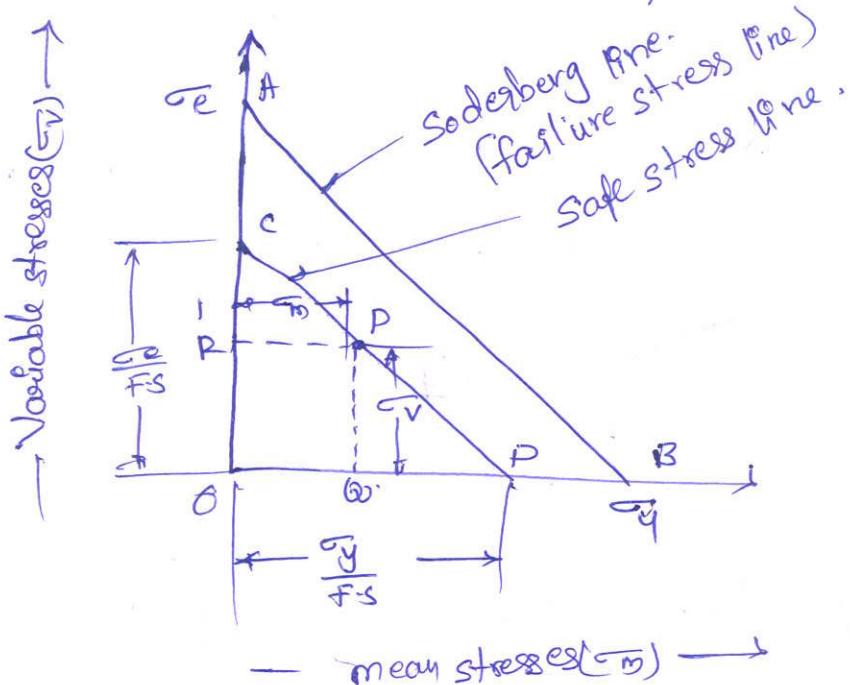
k_{sur} = surface finish factor

k_{sz} = size factor

k_b = load factor for reversed bending load.

Soderberg Method for Combination of Stresses.

A straight line connecting the endurance limit (σ_e) and the yield strength (σ_y), as shown by the line AB, follows the suggestion of Soderberg's line. The line PS based on yield strength



proceeding in the same way as discussed

in here the line AB connecting σ_e and σ_y is called Soderberg's failure stress line.

If a suitable factor of safety $F.S$ is applied to the endurance limit and yield strength, a safe stress line CD may be drawn parallel to AB. Let us consider a design point P on the line CD. Now from similar triangles, ΔCOD and ΔPOD .

$$\frac{PQ}{OC} = \frac{OD - OQ}{OD}$$

$$\frac{PQ}{OC} = \frac{OD - OQ}{OD}$$

$$\frac{PQ}{OC} = 1 - \frac{OQ}{OD}$$

$$\frac{\sigma_r}{\sigma_e F.S} = 1 - \frac{\sigma_m}{\sigma_y F.S}$$

$$\sigma_v = \frac{\sigma_e}{F.S} \left[1 - \frac{c_m x F.S}{\sigma_y} \right]$$

$$\sigma_v = \frac{\sigma_e}{F.S} \times F.S \left[\frac{1}{F.S} - \frac{c_m}{\sigma_y} \right]$$

$$\frac{\sigma_v}{\sigma_e} = \frac{1}{F.S} - \frac{c_m}{\sigma_y}$$

$$\boxed{\frac{1}{F.S} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v}{\sigma_e}}$$

Consider load factor, surface finish factor and size factor applied to variable stress and endurance limit then it may be written as

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_{e.b} \times K_{sw} \times K_{st}} \quad \text{--- (1)}$$

→ This method is used for ductile material only so the eqn (1) may be applied for ductile material to reversed bending load (tension on compressive)

→ When the component is subjected to axial load the eqn may be written as,

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_{e.a} \times K_{sw} \times K_{st}} \quad [\sigma_{e.a} = \sigma_{e.t}]$$

→ When the component is subjected to shear load then the eqn may be written as

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_y} + \frac{\gamma_v \times K_f}{\sigma_{e.s} \times K_{sw} \times K_{st}} \quad [\gamma_v = \sigma_{e.s} \cdot K_s]$$

A cantilever beam made of cold drawn carbon steel are circular cross sections. in fig. is subjected to a load which varies from 0 to 3F. determine max. load that this member can withstand for an indefinite life using F.S = 2 the theoretical stress concentration factor is 1.42 and notch-sensitivity 0.9 Assume the following values.

$$\text{Ultimate stress} = 550 \text{ MPa}$$

$$\text{yield stress} = 470 \text{ MPa}$$

$$\text{Endurance limit} = 275 \text{ MPa}$$

$$\text{size factor} = 0.85$$

$$\text{surface finish factor} = 0.89$$

A)

$$w_{\min} = -F$$

$$w_{\max} = 3F$$

$$F.S = 2$$

$$q = 0.9$$

$$K_t = 1.42$$

$$\sigma_u = 550 \text{ N/mm}^2$$

$$\sigma_y = 470 \text{ N/mm}^2$$

$$\sigma_e = 275 \text{ N/mm}^2$$

$$k_{\text{size}} = 0.85$$

$$k_{\text{surf}} = 0.89$$

$$M = c(-z)$$

$$c = \frac{m}{z}$$

$$c_{\max} = \frac{3F \times 125 \times 32}{\pi (13)^3}$$

$$c_{\max} = 1.7386 F$$

$$c_{\min} = \frac{-F \times 125 \times 32}{\pi (13)^3}$$

$$c_{\min} = -0.5795 F$$

$$c_{\text{mean}} = \frac{c_{\max} + c_{\min}}{2}$$

$$= \frac{1.7386 F - 0.579 F}{2}$$

$$c_{\text{mean}} = 0.5798 F$$

$$\sigma_v = \frac{c_{\max} - c_{\min}}{2}$$

$$= \frac{1.7386 F + 0.579 F}{2}$$

$$\sigma_v = 1.1588 F$$

$$q = \frac{k_t^{-1}}{k_f - 1}$$

$$k_f = 1 + q (k_t^{-1})$$

$$k_f = 1 + 0.9 (1.42^{-1})$$

$$K_f = 1.378$$

According to Goodman line method:-

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times K_f}{\sigma_e \times K_{sur} \times K_{size}}$$

$$\frac{1}{2} = \frac{0.5798F}{550} + \frac{1.1588F \times 1.378}{275 \times 0.85 \times 0.89}$$

$$\frac{1}{2} = 8.7298 \times 10^{-3} \times F$$

$$F = \frac{1}{2 \times 8.729 \times 10^{-3}}$$

$$F = 57.27 N$$

Soderberge line method:-

$$\frac{1}{2} = \frac{0.5798F}{470} + \frac{1.1588 \times 1.378}{275 \times 0.85 \times 0.89}$$

$$\frac{1}{2} = 8.9092 \times 10^{-3} \times F$$

$$F = \frac{1}{2 \times 8.9092 \times 10^{-3}}$$

$$F = 56.12 N$$

- ② A simply supported beam has a concentrated load at the centre which fluctuates from a value of P to 4P the span of beam is 500mm and c/s is circular with a Ø60mm taking for the beam material an ultimate stress of 400MPa and a yield stress of 330MPa. Endurance limit of 330MPa for reversed bending and F.O.S. 1.3 calculate the max value of P take a size factor of 0.85 and surface finish factor of 0.89?

$$A) \sigma_{max} = 4P$$

$$\sigma_{min} = P$$

span of beam (L) = 500mm

c/s of circular beam (D) = Ø60mm

$$\sigma_u = 700 \text{ MPa}$$

$$= 700 \text{ N/mm}^2$$

$$\sigma_y = 500 \text{ N/mm}^2$$

$$\sigma_e = 330 \text{ N/mm}^2$$

$$F.S = 1.3$$

$$K_{S2} = 0.85$$

$$K_f = 0.9$$

$$M = \sigma(z)$$

$$\sigma = \frac{M}{z}$$

$$\sigma_{max} = \frac{4P}{2} \times \frac{500}{2}$$

$$= \frac{4P \times 500}{4} = 500P$$

$$\sigma_{max} = \frac{500P}{\frac{\pi}{3}(D^3)} = \frac{500P \times 32}{\pi(60)^3} = 0.0235P.$$

$$\text{min load} = \frac{P \times 500}{4} = \frac{500P}{4}$$

$$\sigma_{min} = \frac{M}{z} = \frac{500P \times 32}{4 \times \pi(60)^3} = 5.894 \times 10^{-3} P.$$

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$= \frac{0.0235P + 5.894 \times 10^{-3} P}{2}$$

$$= 0.0146P.$$

$$\sigma_v = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$= \frac{0.0235P - 5.894 \times 10^{-3} P}{2} = 8.803 \times 10^{-3} P.$$

According to goodman method

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times K_f}{\sigma_v \times K_{sw} \times K_{sz}}$$

$$\frac{1}{1.3} = \frac{0.0146P}{500} + \frac{8.803 \times 10^{-3} P \times 1}{330 \times 0.89 \times 0.85}$$

$$\frac{1}{1.3} = 5.610 \times 10^{-5} P$$

$$0.76 = 5.610 \times 10^{-5} P$$

$$P = \frac{0.76}{5.610 \times 10^{-5}}$$

$$P = 13547.23 N$$

Soderberg's method :-

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_v \times K_{sw} \times K_{sz}}$$

$$0.76 = \frac{0.0146P}{500} + \frac{8.803 \times 10^{-3} \times 1}{330 \times 0.89 \times 0.95}$$

$$0.76 = 6.075 \times 10^{-5} P$$

$$P = \frac{0.76}{6.075 \times 10^{-5}} = 12510.28 N$$

- ③ Determine the dia of a tensile member of circular c/s of following data is given.
- max tensile load = 10 kN
 max compressive load = 5 kN

Ultimate tensile Strength = 600 MPa.

yield Point = 380 MPa.

endurance limit = 290 MPa

$$F.O.S = 4$$

stress concentration factor = 2.2

A) $\sigma_{max} = P/A = \frac{10 \times 10^3}{\pi k_1 (d^2)} = \frac{10 \times 10^3 \times 4}{\pi d^2} = 12732.39/d^2$

$$\sigma_{min} = -P/A = -\frac{5 \times 10^3 \times 4}{\pi d^2} = -\frac{6366.19}{d^2}$$

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{\frac{12732.39}{d^2} + \left(\frac{-6366.19}{d^2}\right)}{2} = \frac{3184.1}{d^2}$$

$$\sigma_v = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{9549.29}{d^2}$$

good man :-

$$\frac{1}{f.s} = \frac{\sigma_m}{\sigma_v} + \frac{\sigma_v \times K_f}{\sigma_e}$$

$$\frac{1}{4} = \frac{\frac{3184.1}{d^2}}{600} + \frac{\frac{9549.29}{d^2} \times 2.2}{290}$$

$$d^2 = 310.99 \text{ mm}^2$$

$$d = 17.63 \text{ mm}$$

Soderberg's method :-

$$\frac{1}{f.s} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_e}$$

$$\frac{1}{4} = \frac{\frac{3184.1}{d^2}}{380} + \frac{\frac{9549.29}{d^2} \times 2.2}{290}$$

$$d^2 = 323.27$$

$$d = 17.97 \text{ mm}$$

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 (i.e. touching) each other and a cover plate (i.e. strap) is placed either on one side or on both sides of the main plates.

Two Types of Butt joints

1. Single strap butt joint
2. Double strap butt joint.

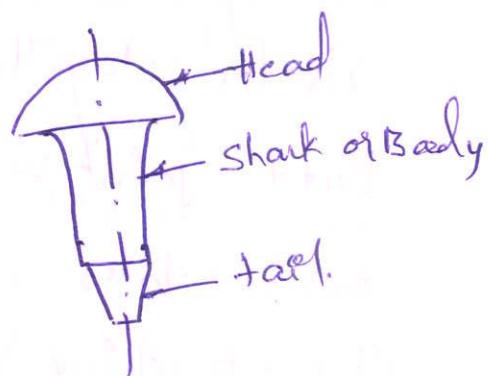
Important terms used in Riveted Joint

1. Pitch (a_f) is the distance from the centre of one rivet to the centre of the next rivet measured parallel to the seam. Usually denoted by P .
2. Back pitch (a_b) is the distance b/w the centre lines of the successive rows. It is usually denoted by P_b .
3. Diagonal pitch (a_d) is the distance b/w the centres of the rivets in adjacent rows of zig-zag riveted joint. It is usually denoted by P_d .
4. Margin (or) Marginal pitch (a_m) is the distance b/w the centre of hole to the nearest edge of the plate. It is usually denoted by m .

Riveted Joints

A rivet is a short cylindrical bar with a head integral to it. The cylindrical 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 structural work, ship building, bridges, tanks and boiler shells.



Material of Rivets:-

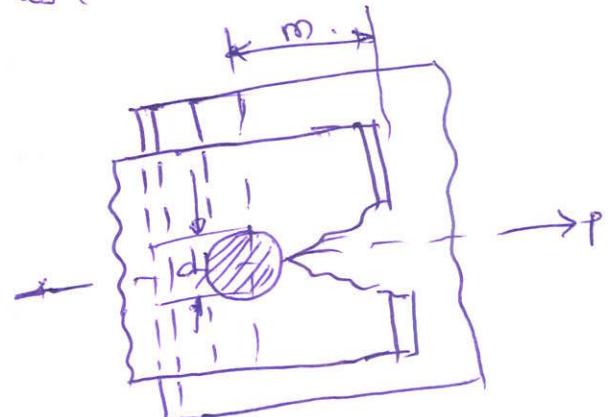
The material of rivets must be tough and ductile. They are usually made of steel (low carbon steel or nickel steel), brass, aluminium (or) copper.

Essential Qualities of a Rivet:-

According to Indian standard, the material of a rivet must have a tensile strength not less than 40 N/mm^2 and elongation not less than 20%. The material must be of such quality that when in cold condition, the shank shall be bent on itself through 180° without cracking and after being heated to 650°C and quenched, it must pass the same test. The rivet when hot must flatten without cracking to a dia. 2.5 times the dia of shank.

Failures of a Riveted Joint:-

1. ~~Tearing of the plate at an edge~~ A joint may fail due to tearing of the plate at an edge as shown in figure. This can be avoided by keeping the margin. $m = 1.5d$, where d is the diameter of rivet hole.

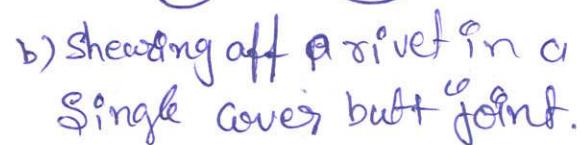


2. ~~Tearing of the plate across a row of rivets~~

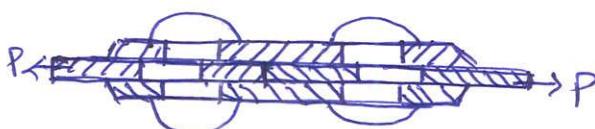
Due to the tensile stresses in the main plates the main plate or cover plate may tear off across a row of rivets. In such case we consider only one pitch length of the plate, since every rivet is responsible for that much length of the plate only.



a) Shearing off a rivet in a lap joint



b) Shearing off a rivet in a single cover butt joint.



Shearing off a rivet in double cover butt joint.

\therefore Shearing resistance or pull required to shear off the rivet per pitch length

$$P_s = n \times \frac{\pi}{4} \times d^2 \times \gamma \quad \text{--- for single shear.}$$

$$= n \times 2 \times \frac{\pi}{4} \times d^2 \times \gamma \quad \text{--- for Double shear.}$$

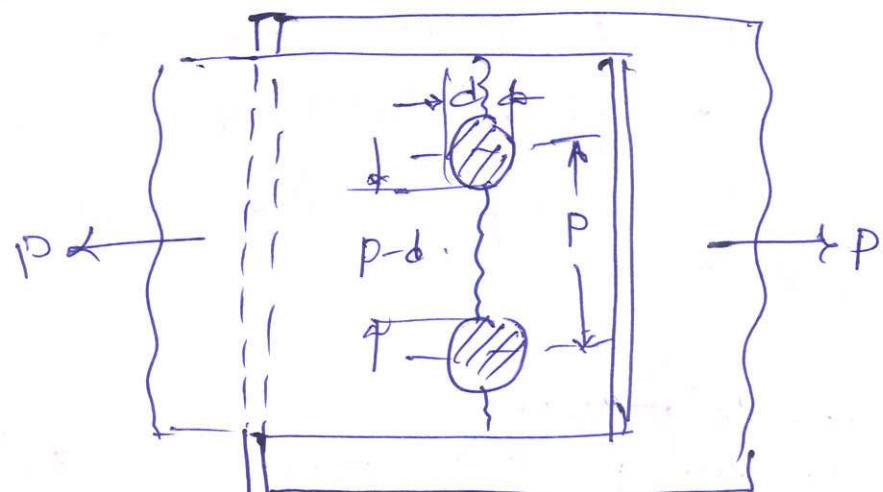
where d = diameter of the rivet hole.

γ = Safe permissible shear stress for rivet material.

n = no. of rivets per pitch length.

$$P_s = n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \gamma \quad (\text{double shear, according to Indian Boiler Regulation}).$$

3. Peeling of the plate



Tearing resistance or pull required to tear off the plate per pitch length

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

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

if the joint is continuous as in case of boilers, the strength is calculated per pitch length. But if the joint is small, the strength is calculated for the whole length of the plate.

Efficiency of a riveted Joint:-

The efficiency of a riveted joint is defined as the ratio of the strength of riveted joint to the strength of the unriveted or solid plate.

i. Strength of riveted joint = least. of P_s , P_t and P_c

strength of the un-riveted or solid plate per pitch length $p = pxt \times \sigma_t$.

ii. Efficiency of the riveted joint

$$\eta = \frac{\text{least of } P_t, P_s \text{ and } P_c}{Pxt \times \sigma_t}$$

p = pitch of the rivets

t = thickness of the plate

σ_t = permissible tensile stress of the plate material.

4. Crushing of the plate or rivets:-

Some times the rivets do not actually shear off under the tensile stress, but are crushed as shown in figure. Due to this, the rivet hole becomes of an oval shape and hence the joint becomes loose. This is also known as bearing failure.

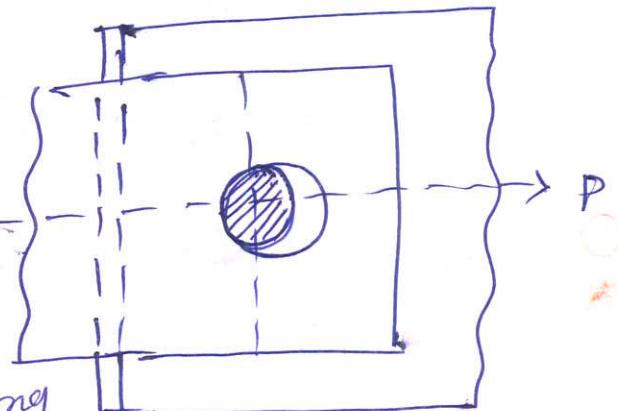
The resistance offered by a rivet to be crushed is known as crushing resistance or crushing strength.

$$P_c = n \cdot d \cdot t \cdot \sigma_c$$

where d = dia of rivet hole. t = thickness of plate

σ_c = Safe permissible crushing stress for the rivet.

n = No. of rivets per pitch length.



Strength of a riveted joint.

The strength of a joint may be defined as the maximum force, which it can transmit without causing it to fail. We see that P_t , P_s and P_c are the pulls required to tear off the plate, shear off the rivet and crushing of the rivet. A little consideration will show that if we go on increasing the pull on a riveted joint, it will fail when the least of these three pulls is reached, because a higher value of the other pull will never reach since the joint has failed, either by tearing off the plate, shearing off the rivet or crushing off the rivet.

welding Introduction

A welded joint is a permanent joint which is obtained by the fusion of edges of the two parts to be joined together, with or without application of pressure and a filler material. The heat required for the fusion of the (filler) material may be obtained by burning of gas (in case of gas welding) or by an electric arc (in case of electric arc welding).

Advantages:-

- welding structures are usually lighter than riveted structures.
- The welded joints provides maximum efficiency
- Alterations and additions can be easily made in the existing structures.
- Welded structure is smooth in appearance.
- welded joint has a greater strength
- it is possible to weld any part of a structure at any point.
- The process of welding takes less time than riveting.

Disadvantages:-

- Since there is an uneven heating and cooling during fabrication, therefore the members may get distorted or additional stresses may develop.
- It requires a highly skilled labour and supervision.
- Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it.

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

welding processes ↴

1. welding processes that use heat alone e.g. fusion welding
2. welding processes that use a combination of heat and pressure e.g. forge welding.

Fusion welding: In case of fusion welding, the parts to be joined are held in position while the molten metal is supplied to the joint. The molten metal may come from the parts themselves (i.e. parent metal) or filler metal which normally have in the composition of the parent metal. The joint surface become plastic or even molten because of heat from the molten filler metal or other source. When the molten metal solidifies or fuses the joint is formed.

According to method of heat generated, may be classified as.

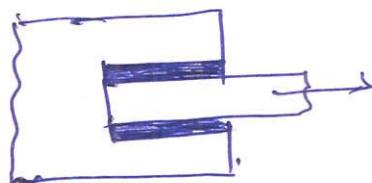
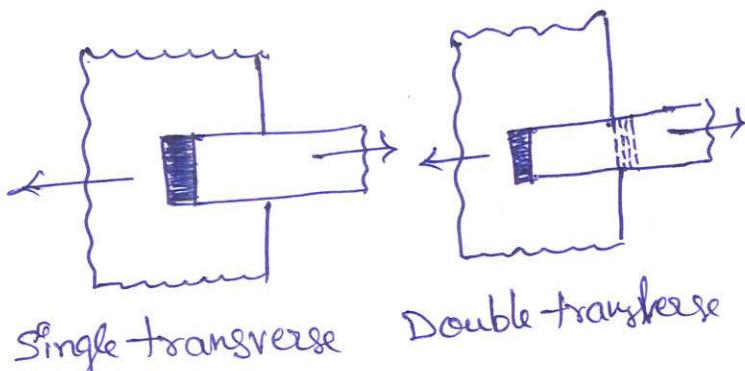
1. Thermite welding.
2. Gas welding.
3. Electric arc welding.

Forge welding: In forge welding, the parts to be joined 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 welded Joints

Two types of welded joints in welded joints

1. Lap Joint or fillet Joint.
- 2.



parallel fillet.

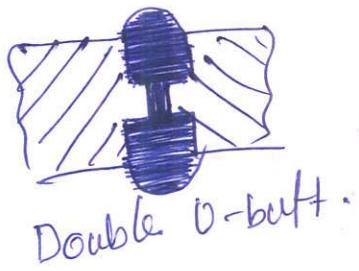
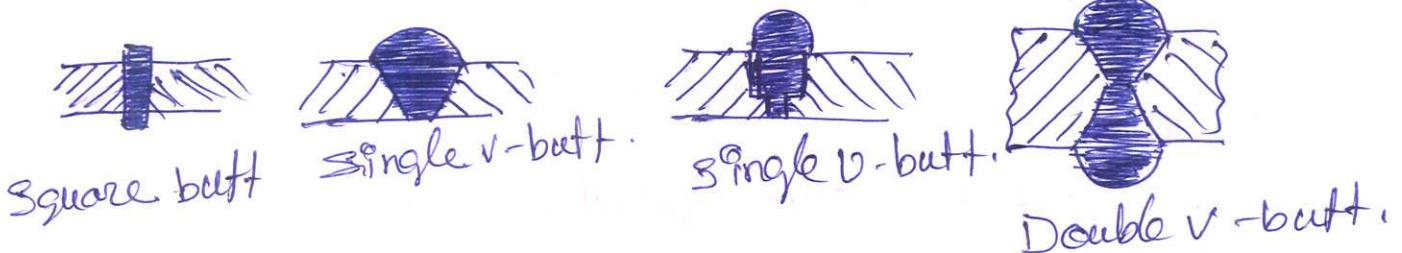
The lap joint or the fillet joint is obtained by overlapping the plates and then welding the edges of the plates. The cross-section of the fillet is approximately triangular. The fillet joints may be.

1. Single-transverse fillet
2. Double-transverse fillet.
3. parallel fillet

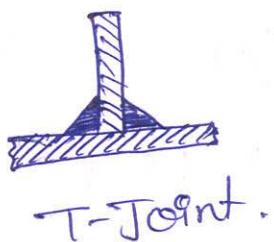
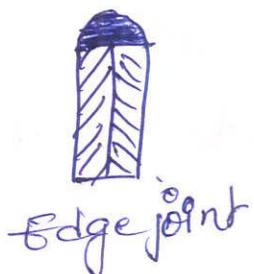
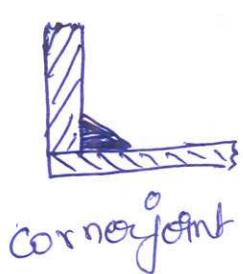
A single-transverse fillet joint has the disadvantage that the edge of the plate which is not welded can buckle or warp out of shape.

2. Butt joint:

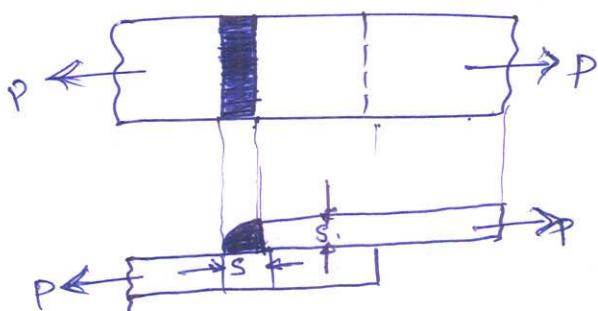
The butt joint is obtained by placing the plates edge to edge. In butt welds the plate edges do not require bevelling if the thickness of plate is less than 5mm. On the other hand, if the plate thickness is 5mm to 12.5mm, the edges should be bevelled to V or U groove on both sides.



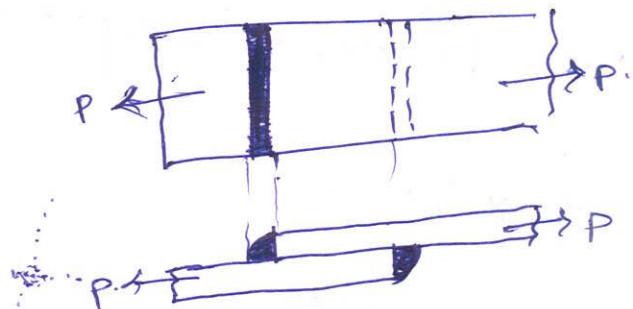
Other types of butt welded joints



Strength of Transverse fillet welded Joints
The transverse fillet welds are designed for tensile strength. Let us consider a single and double-transverse-fillet welds as shown in figure.



Single transverse-fillet



Double transverse-fillet

In order to determine the strength of the fillet joint, it is assumed that the section of fillet is a right angled triangle ABC with hypotenuse AC making equal angles with other two sides AB and BC. The enlarged view of the fillet is

is shown in figure. The length of each side is known as leg or size of the weld and perpendicular or distance of the hypotenuse from the intersection of legs (i.e. BD) is known as throat thickness. The minimum area of the weld is obtained at the throat BD, which is given by the product of the throat thickness and length of weld.

$$t = \text{Throat thickness (BD)}$$

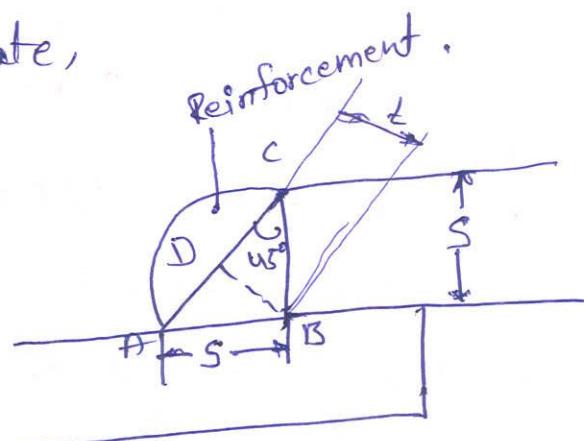
$$s = \text{Leg or size of weld.}$$

= Thickness of plate,

$$l = \text{length of weld.}$$

We find that the throat thickness

$$t = s \times \sin 45^\circ = 0.707s.$$



\therefore Minimum area of the weld or throat area

$$A = \text{throat thickness} \times \text{length of weld.}$$

$$= l \times t = 0.707s \times l.$$

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

$$P = \text{Throat area} \times \text{allowable tensile stress}$$

$$= 0.707s \times l \times \sigma_E.$$

And tensile strength of the joint for double fillet weld.

$$P = 2 \times 0.707s \times l \times \sigma_E = 1.414s \times l \times \sigma_E.$$

Note:- Since the weld is weaker than the plate due to slag and blow holes, therefore the weld is given a reinforcement which may be taken as 10% of the plate thickness.

Strength of parallel fillet welded joints

The parallel fillet welded joints are designed for shear strength. Consider a double parallel fillet welded joint as shown in figure. We have already discussed in the previous article, that the minimum area of weld or throat area.

$$A = 0.7075 \times d$$

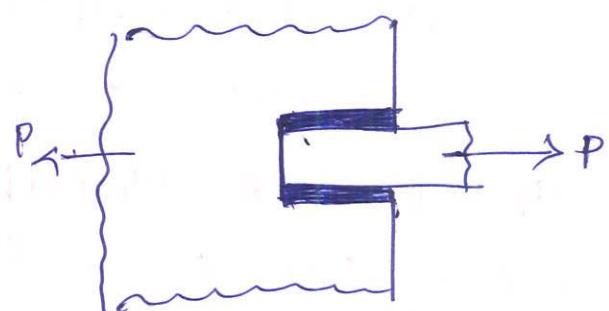
If σ_y is allowable shear stress for the weld metal, then the shear strength of the joint for single parallel fillet weld.

$$P = \text{throat area} \times \text{allowable shear stress.}$$

$$= 0.7075 \times d \times \sigma_y$$

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

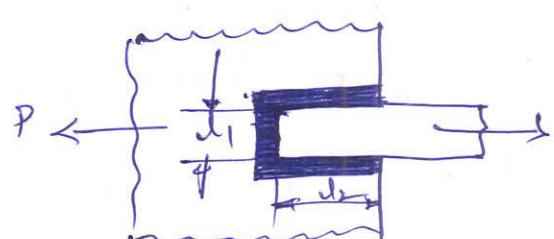
$$P = 2 \times 0.7075 \times d \times \sigma_y = 1.4145 \times d \times \sigma_y$$



If there is a combination of single transverse and double parallel fillet welds as shown in figure, then the strength of the joint is given by the sum of strengths of single transverse and double parallel fillet welds.

$$P = 0.7075 \times d_1 \times \sigma_y + 1.4145 \times d_2 \times \sigma_y$$

where d_1 is normally width of plate



→ In order to allow for starting and stopping of the bead, 12.5mm should be added to the length of each weld obtained by the above expression.

→ For reinforced fillet welds, the throat dimension may be taken as $0.85t$.

Special cases of fillet welded joints

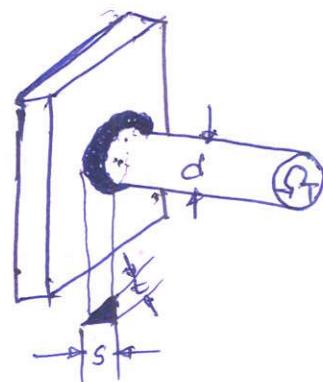
1. Circular fillet weld subjected to torsion
Consider a circular rod connected to a rigid plate by fillet weld.

We know the shear stress of

material

$$\tau_f = \frac{T \cdot r}{J} = \frac{T \times d/2}{J}$$

$$= \frac{T \times d/2}{\pi t d^3/4} = \frac{2T}{\pi t d^2}$$



length of throat $t = s \sin 45^\circ$

$$t = 0.707s$$

The maximum shear stress

$$\tau_{max} = \frac{\sigma T}{\pi \times 0.707 \times s \times d^2} = \frac{2.83T}{\pi s d^2}$$

where d = dia of rod

r = radius of rod

T = Torque acting on the rod

s = size (or leg) of weld.

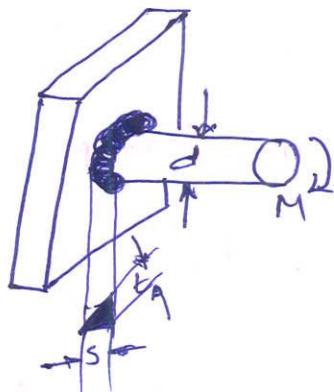
t = throat thickness.

J = polar moment of inertia of the weld section $\frac{\pi t d^3}{48}$

2: Circular-fillet weld subjected to bending moment & consider a circular rod connected to a rigid plate by a fillet weld.

We know that the bending stress

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



$$\therefore \text{length of throat } t = s \sin 45^\circ = 0.707 s.$$

\therefore The maximum bending stress.

$$\sigma_b(\max) = \frac{4M}{\pi \times 0.707 s \times d^2} = \frac{5.66 M}{\pi s d^2}$$

where

d : Dia of rod.

M : Bending moment acting on the rod,

s : size (or leg) of weld.

t : throat thickness.

Z : section modulus of weld section

$$= \pi t d^2 / 4$$

3. Long-filleted weld subjected to torsion

Consider a vertical plate attached to a horizontal plate by two identical fillet weld as shown in fig.

T = torque acting on vertical plate

l = length of weld.

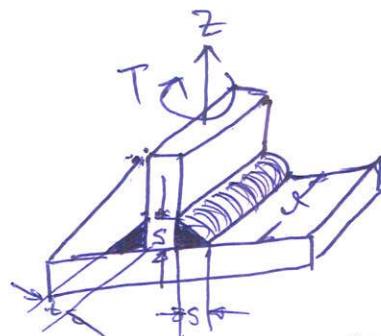
s = size (or leg) of weld

t = throat thickness, and

J = polar moment of inertia of the weld section

$$= 2 \times \frac{t \times l^3}{12} = \frac{t \times l^3}{6}$$

$$\therefore \text{shear stress } \gamma = \frac{T \times l/2}{t \times l^3/6} = \frac{3T}{t \times l^2}$$



The maximum shear stress occurs at the throat and given by

$$T_{\max} = \frac{3T}{0.707 s \times l^2} = \frac{4.242 T}{s \times l^2}$$

Screwed Joints

A screw thread is formed by cutting a continuous helical groove on a cylindrical surface. A screw made by cutting a single helical groove on the cylinder is known a single threaded (or single-start) screw and if a second thread is cut in the space between the grooves of the first, a double-threaded (or double-start) screw is formed. Similarly, triple and quadruple (i.e. multiple-start) threads may be formed. The helical grooves may be cut either right hand or left hand.

A screwed joint is mainly composed of two elements i.e. a bolt and nut. The screwed joints are widely used where machine parts are required to be readily connected or disconnected without damage to machine or the fastening.

Advantages and Disadvantages

Advantages:-

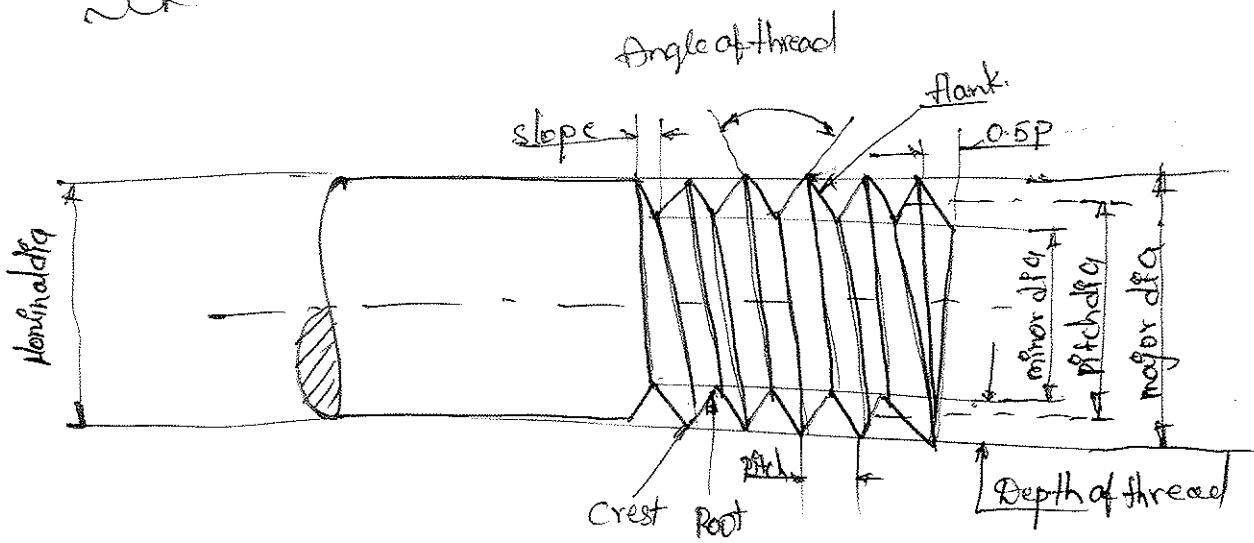
- Screwed joints are highly reliable in operation.
- Screwed joints are convenient to assemble and disassemble.
- A wide range of screwed joints may be adopted to various operating conditions.
- Screws are relatively cheap to produce due to standardisation and highly efficient manufacturing processes.

Disadvantages:-

- The main disadvantage of the screwed joint is the stress concentration in the threaded portions which are vulnerable points under variable load conditions.

Note:- Strength of screwed joints is not comparable with that of riveted joints or welded joints.

Important terms used in Screw Threads



1. Major dia. - It is the largest dia of the external or internal screw thread. The screw is specified by this diameter. It is also known as outside or nominal dia.
2. Minor dia. - It is the smallest dia of an external or internal screw thread. It is also known as core or root diameter.
3. Pitch diameter. - It is the diameter of an imaginary cylinder on a cylindrical screw thread, the surface of which would pass through the thread at such points as to make equal the width of thread and width of the space b/w the threads. It is also known as effective diameter.
4. Pitch. - It is the distance from a point on one thread to the corresponding point on the next.

$$\text{Pitch } P = \frac{1}{\text{No. of threads per unit length of screw.}}$$

5. Lead. - It is defined as the distance which a screw thread advances axially in one rotation of the nut.

6. Crest. - It is the top surface of the thread.
7. Root. - It is the bottom surface created by the two adjacent flanks of the threads.

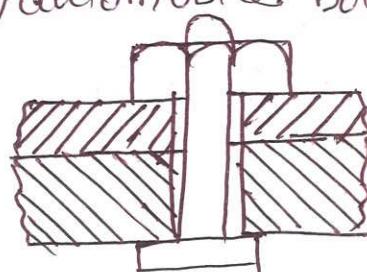
8. Depth of thread :- it is the perpendicular distance between the crest and root.
9. Flank :- it is the surface joining the crest & root.
10. Angle of thread :- it is the angle included by the flanks of the thread.
11. Slope :- it is half the pitch of the thread.

Forms of Screw threads:-

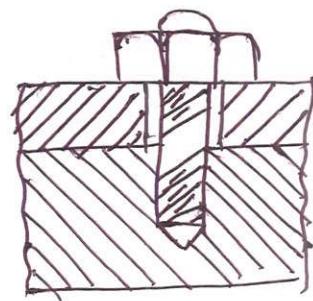
1. British standard with worth (B.S.W) thread.
2. British association (B.I.A) thread.
3. American national standard thread.
4. Unified standard thread
5. Square thread.
6. ACME thread.
7. Knuckle thread
8. Buttress thread
9. Metric thread.

Common Types of screw fastening

1. Through bolts :- it is a cylindrical bar with threads for the nut at one end and head at the other end. The cylindrical part of the bolt is known as shank. It is passed through drilled holes in the two parts to be fastened together and clamped them securely to each other as the nut is screwed on to the threaded end. The through bolts according to their use may be known as machine bolts, carriage bolts, automobile bolts, eye bolts etc.

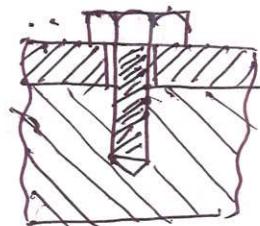


2. Tap bolt → A tap bolt or screw differs from a bolt. It is screwed into a tapped hole of one of the parts to be fastened with out the nut.



3. Studs:

A stud is a round bar threaded at both ends. One end of the stud is screwed into a tapped hole of the parts to be fastened, while the other end receives a nut on it, as shown in fig. Studs are chiefly used instead of tap bolts for securing various kinds of covers e.g. covers of engine and pump cylinders, valves, chest etc.



4. Cap screws: The cap screws are similar to tap bolts except that they are of small size and a variety of shapes of heads are available.

5. Machine screws: These are similar to cap screws with the head slotted for a screw driver. They are generally used with a nut.

6. Set screws: These are used to prevent relative motion b/w the two parts. A set screw is screwed through a threaded hole in one part so that its point passes against the other part. This resists the relative motion b/w the two parts. They may be used instead of key to prevent the relative motion.

Locking Devices

Ordinary thread fastenings, generally, remain tight under static loads, but many of these fastenings become loose under the action of variable loads or when machine is subjected to vibrations. The loosening of fastening is very dangerous and must be prevented. In order to prevent this, a large number of locking devices are available, some of which are discussed below.

1. Jam nut or lock nut
2. castle nut
3. sawn nut
4. lock ring or grooved nut
5. Locking with pin
6. Locking with plate
7. Spring lock washers.

Designation of Screw threads:-

1. Size designation:- The size of the screw thread is designated by the letter 'M' followed by the diameter and pitch, the two being separated by the sign x. When there is no indication of the pitch, it shall mean that a coarse pitch is implied.
2. Tolerance designation:- This shall include:
 - a) A figure designating tolerance grade as indicated below '7' for fine grade, '8' for normal (medium) grade and '9' for coarse grade.
 - b) A letter designating the tolerance position as indicated below: 'f1' for unit thread, 'd' for bolt thread with allowance, and 'n' for bolt thread without allowance.

Stresses in screwed fastening due to static loading:-

1. Internal stresses due to screwing up forces.

2. Stresses due to external force.

3. Stress due to combination of stresses due to initial tightening and External loads.

1. Initial stresses due to screwing up forces:-

a) tensile stress due to stretching of bolt, since none of the above mentioned stresses are accurately determined, therefore bolts are designed on the basis of direct tensile stress with a large factor of safety in order to account for indeterminate stresses.

2. The initial tension in the bolt

$$P_i = 2840 d \text{ N}$$

P_i = Initial tension in a bolt,

d = Nominal dia of bolt. in mm.

Above relation is used for making a joint fluid tight.

When the joint is not required as tight as fluid tight joint, then the initial tension in a bolt may be reduced to half of the above value.

$$P_i = 1420 d \text{ N.}$$

The small dia bolts may fail during tightening, therefore bolts of smaller dia are not permitted in making fluid tight joint.

If the bolt is not initially stressed, then the max. safe axial load which may be applied to it is given by

$P = \text{permissible stress} \times \text{cross-sectional area at bottom of the thread (i.e. stress area)}$

$$\text{stress area} = \frac{\pi}{4} \left(\frac{dp + dc}{2} \right)^2$$

dp = pitch diameter

dc = core or minor diameter.

2. Torsional shear stress caused by frictional resistance of the threads during its tightening.

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

$$\tau = \frac{T}{J} \times r = \frac{\pi}{32} (dc)^4 \times \frac{dc}{2} = \frac{16 T}{\pi (dc)^3}$$

τ = Torsional shear stress.

T = Torque applied,

dc = minor or core diameter of the thread.

3. Shear stress across the threads

$$\tau_s = \frac{P}{\pi d_c \times b \times h}$$

b = width of the thread section at the root.

for Nut

$$\tau_n = \frac{P}{\pi d \times b \times h}$$

d = major dia

4. Compression or Crushing stress on threads

$$\sigma_c = \frac{P}{\pi [d^2 - (dc)^2] n}$$

d = major diameter

dc = minor diameter

n = no. of threads in engagement.

5. Bending stress if the surface under the head or nut are not perfectly parallel to the bolt.

$$\sigma_b = \frac{\alpha \cdot E}{2l}$$

α = Difference in height b/w the extreme corners of the nut or head.

l = length of the shank of the bolt.

E = Young's modulus for the material of the bolt.

Stresses due to external forces:-

1. Tensile stress) The bolts, studs are screws usually carry a load in the direction of the bolt axis which induces a tensile stress in the bolt.

d_c = Root or core dia of thread
 σ_t = permissible tensile stress for the bolt material

We know that external load applied

$$P = \frac{\pi}{4} (d_c)^2 \sigma_t \quad \text{or} \quad \sigma_c = \sqrt{\frac{4P}{\pi d_c^2}}$$

→ If the external load is taken up by n no. of bolts, then

$$P = \frac{\pi}{4} (d_c)^2 \times \sigma_t \times n.$$

→ In case of the standard table not available take coarse threads, $d_c = 0.84 d$, where d is the normal d_c .

2. Shear stress:- Bolts are used to prevent the relative movement of two or more parts, as in case of flange coupling, then the shear stress is induced in the bolts.

d = major dia of bolt.

n = no. of bolts.

1. Shear load carried by the bolts

$$P_s = \frac{\pi}{4} d^2 \times \gamma \times n.$$

$$\text{or } d = \sqrt{\frac{4 P_s}{\pi \gamma n}}$$

3. Combinations of tension and shear stress :-
When the bolt is subjected to both tension and shear load, as in case of coupling bolts or bearings,

Then maximum principal shear stress

$$\gamma_{\max} = \frac{1}{2} \sqrt{(\sigma_t)^2 + 4 \gamma^2}$$

and max. principal tensile stress

$$\sigma_t(\max) = \frac{\sigma_t}{2} + \frac{1}{2} \sqrt{(\sigma_t)^2 + 4 \gamma^2}$$

Stress due to combined forces:-

When the connected members are very yielding as compared with the bolt, which is a soft gasket, then the resultant load on the bolt is approximately equal to the sum of the initial tension and the external load. In order to determine the resultant axial load (P) on the bolt, the following eqn may be used

$$P = P_1 + \left(\frac{q}{1+q} \right) P_2 = P_1 \times K.P_2$$

P_1 = Initial tension due to tightening of the bolt

P_2 = External load on the bolt.

α = Ratio of elasticity of connected part to the elasticity of bolt.