COURSE FILE

Course Name: Diploma in Mechanical engineering

Subject : Design of Machine elements

Subject Code : MEC604

Semester : 6th

Hours : 42

Full Marks : 80+20=100

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Content-

Chapter	Name of the Topic	Hours
01	**Introduction to Design** **1.1** Fundamentals: -Types of loads, concepts of stress, Strain, Stress-Strain Diagram for Ductile and Brittle Materials, Types of Stresses such as Tension, Compression, Shear, Bearing pressure Intensity, Crushing, bending and torsion, Principle Stresses (Simple Numerical) **1.2** Fatigue, Creep, S-N curve, Endurance Limit, Factor of Safety and Factors governing selection of factor of Safety. Stress Concentration–Causes & Remedies **1.3** Converting actual load or to rqueinto design load ortorque using design factor slike velocity factor, factor of safety & service factor. Properties of Engineering materials, Designation of materials as per IS and introduction to International standards & advantages of standardization, use of design data book . **1.4** Theories of Elastic Failures Principal normal stress theory, Maximum shear stress theory & maximum distortion energy theory.	08
02	**Design of Shafts, Keys and Couplings** **3.1** Types of Shafts, Shaft materials, Standard Sizes, Design of Shafts (Hollow and Solid) using strength and rigidity criteria, ASME code of design for lines hafts supported between bearings with one or two pulleys in between or one over hung pulley **3.2** Design of Sunk Keys, Effect of Key ways on strength of shaft. **3.3** Design of Couplings– Muff Coupling, Flange Coupling, Bush-pin type flexible coupling.	08

03	**Design of simple machine parts** **2.1** Cotter Joint. Knuckle Joint. **2.2** Design of Levers:-Hand/Foot Lever & Bell Crank Lever **2.3** Design of C Clamp. Off-set links. Over hang Crank. Arm of Pulley	06
04	**Design of Power Screws and Spur Gear** **4.1** Thread Profiles used for power Screws, relative merits and demerits of each, self-locking and over hauling property. Torque required to over come thread friction, efficiency of power screws, types of stresses induced. **4.2** Design of Screw Jack. Toggle Jack.	08
05	**Design of springs** **5.1** Classification and Applications of Springs. Spring Terminology, materials and specifications Stresses in springs. Wahl's correction factor. Deflection of springs. Energy store din springs. **5.2** Design of Helical springs subjected to uniformly applied loads. **5.3**Leafsprings constructionandapplication	05
06	**Design of Fasteners** **6.1** Riveted Joints-Design of riveted joints, efficiency and frictional resistance of riveted joints. **6.2** Welded Joints- Representation of welds. Design of welded joints for static loads. strength of welds at varying load. **6.3** Stresses in Screwed fasteners, bolts of Uniform Strength. Design of Bolts subjected to fatigue loading.	05
07	**Ergonomics & Aesthetic consideration in design** **7.1 Ergonomics** of Design–Man–Machine relationship. Design of Equipment for control, environment & safety. Aesthetic considerations regarding shape. size. color & surface finish.	02
	Total	42

Course objectives

- To introduce students to the design and theory of common machine elements and to give students experience in solving design problems involving machine elements.
- To synergize forces, moments, torques, stress and strength information to develop ability to analyze, design and/or select machine elements – with attention to safety, reliability, and societal and fiscal aspects.
- To require the student to prepare professional quality solutions and presentations to effectively communicate the results of analysis and design

Course Outcomes

1. After completing this course, student will get a good understanding of various practical design concepts and able to design Machine Component.
2. They will be able to design a Machine component at various types of loading such as static and dynamic loading

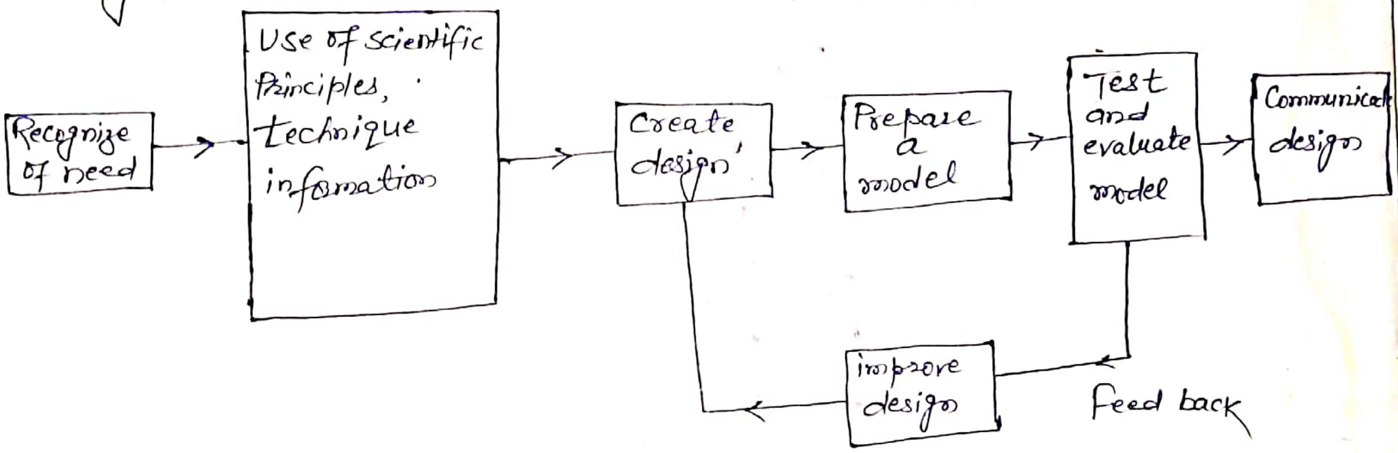
Day-①

Chapter - 1.

Hizamuddin Ansari
Diploma 2nd (shift)
ME - 6th sem.
Sub - Design of Machine element

* Machine design :-

Introduction :-
The subject machine design is the creation of new and better machines and improving the existing ones. A new or better machines is one which is more economical in the overall cost of production and operation. The process of design is a longitudional time consuming. From the study of existing ideas, a new ideas has to be conceived.
The idea is then reformed in terms of dimensions and suitable drawing.



• Machine design may also defined as the use of scientific principles technical information and imagination in the description of a machine or mechanical system to perform specific function.

* General procedure or steps involved in designing a machine Component

There are following steps in the process of designing a machine Component

(i) Recognition of need :- First of all, make a complete statement of the problem, indicate the need, aim or purpose for which the machine is to be designed.

(ii) Mechanism :- select the possible mechanism or group of mechanism which will give the desired motion.

(iii) Analysis of forces :- Find the forces acting on each member of the machine.

(iv) Material selection :- select the material best suited for each member.

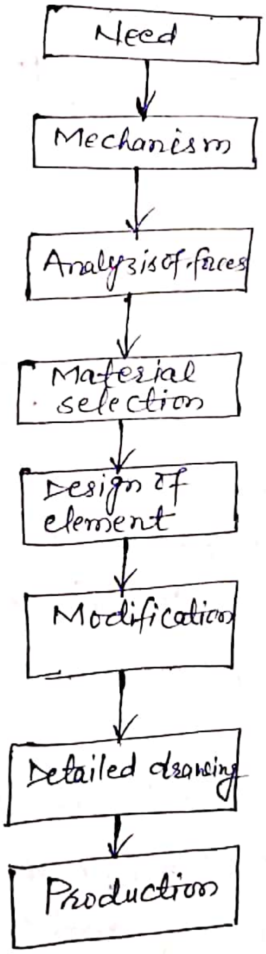
1/6/18

(v) Design of machine elements :- Find size of each member by considering forces acting on the member and permissible stress for the material used.

(vi) Modification :- Modify the size of member to agree with the best experiance and judgement.

(vii) Detailed drawing :- Draw detailed drawing of each component and assembly of machine with complete specification for the manufacturing process.

(viii) Production :- The component as per the drawing is manufactured in the workshop.



steps involved in design process

* General consideration in machine design:-

There are following consideration in machine design.

(i) Type of loads and stresses caused by the load.

(2) Mechanism (Motion of the parts or kinematics of machines).

(3) Selection of material.

(4) Convenient and economical features.

(5) Use of standard Part.

(6) Safety of operation.

(7) Work shop facility.

(8) Number of machine to be manufactured.

(9) Cost of construction and assembly.

(10) Frictional resistance and lubrication.

* Basic design requirement

The basic design requirements in machine design can be listed as below.

(i) Strength

(ii) Rigidity

(iii) Wear resistance

(iv) Manufacturability

(v) Operational safety

(vi) Standardisation

(vii) Reliability

(viii) Maintainability

(ix) Durability

(x) shape and size

(xi) Efficiency

(xii) Simplicity

(xiii) Easy to assemble

(xiv) Lubrication

(xv) Cost

(xvi) Material.

Ductile material → Steel, Copper, Aluminium.

Brittle material → Cast iron.

Machine :- A machine is a thing that is created by people to make work easier.

Strain energy :-

It is the energy stored in an elastic body under loading.

* Types of loads.

(a) Dead or steady load:- it is the gradually applied load which donot change in magnitude, direction or point of application with respect to time.
Example – Load acting on column or struct.

(b) Variable or Fluctuating load:- it varies in magnitude, direction or point of application w.r.to time. it is also called fatigue load. Ex – Force on i.c. engine valve spring, Bending moment on rotating shafts.

(c) Impact load:-
it is suddenly applied ~~or removed load~~ with initial velocity.
Ex – ~~Leaf~~ Blows of hammer.

(d) Shock load:-
it is applied or removed suddenly
Ex – Leaf spring of automobile.

- Dead or steady load :- A load is said to be dead or steady load when it does not change in magnitude or direction.

- Live or variable load :- A load is said to be live or variable load when it changes continuously.

- Suddenly applied or shock load :- A load is said to be suddenly applied or shock load when it is suddenly applied or removed

- Impact load :- A load is said to be an impact load, when it is applied with some initial velocity.

* Design stress :-

While designing machine parts, it is desirable to keep the stress lower than the maximum stress, at which failure take place. This stress is known as working stress or design stress.

Mathematically,
$$\boxed{\text{Design stress} = \frac{\text{Maximum stress}}{\text{Factor of safety}}}$$

* Factor of safety :-

While designing a component, it is necessary to provide sufficient reserve strength in case of an accident. This is achieved by taking a suitable factor of safety (FOS).

$$\therefore \boxed{FOS = \frac{\text{Maximum stress}}{\text{Working stress or Allowable stress}}}$$

The allowable stress is the stress value, which is used in design to determine the dimension of component.

- For ductile material, the allowable stress (σ) is obtained by the following relationship

$$\boxed{\sigma = \frac{\sigma_{yt}}{FOS}}$$

$\sigma_{yt} \rightarrow$ yield point tensile stress

$\sigma_{ut} \rightarrow$ ultimate tensile stress.

- For brittle material,

$$\boxed{\sigma = \frac{\sigma_{ut}}{FOS}}$$

* Factors to be consider while selecting factor of safety :-

(i) Reliability of properties of material and change of these properties during service.
(ii) Reliability of applied load.
(iii) The certainity as to exact mode of failure.
(iv) Extent of stress-concentration
(v) The extent of initial stresses set up during manufacturing.
(vi) The extent of loss of life, if failure occurs.
(vii) The reliability of test results to actual machine parts.

* In what cases, the value of factor of safety is taking high.

Ans:- (1) Impact loads and accidental loads.

(2) Non - uniformity of material.

(3) Corrosion and high temperature

(4) Reliability of parts.

(5) In Case of possibilities of accidents and damage to property and life due to design failure.

* Value of factor of safety may be taken low.

(1) Less frequently used parts.

(2) No damage to property and life in case of design failure.

(3) Analysis of loads with respect to magnitude and nature is easy to estimate.

(4) In case of failure, little time and low money is needed for rectification.

*

Type of loads	factor of safety
Static load	1.5 to 3
Variable load	3 to 6
Impact/shock load	6 to 12

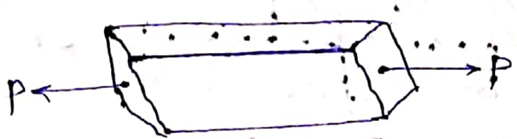
1.8
1.5
1.4

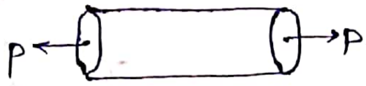
Day-2.

* Types of Stresses

The different types of stresses are –

(i) **Tensile stress** :– When a body is subjected to equal and opposite axial pull forces, the stress produced is called as tensile stress.



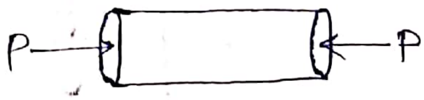


$$\boxed{\sigma_t = \frac{P}{A}}$$

$$\boxed{\sigma_t = \frac{P}{\frac{\pi}{4}d^2}}$$

2. **Compressive force** :– When a body is subjected to equal and opposite axial push forces, the stress is produced is called compressive force.

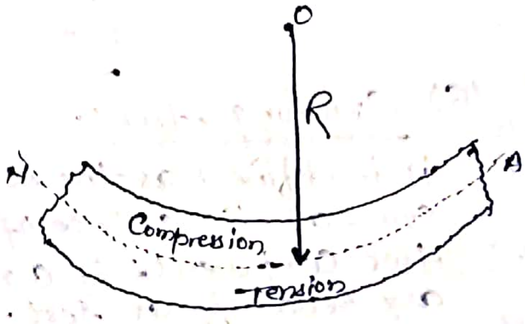
$$\boxed{\sigma_c = \frac{P}{\frac{\pi}{4}d^2}}$$



3. **Bending stress** :- When a beam is subjected to bending moment (M), the stress induced is called bending stress.

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

$$\boxed{\sigma_b = \frac{M}{I} \times y}$$

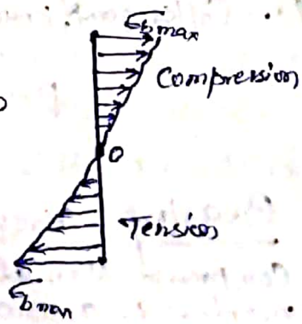


where I = M.I. of crossection.

M = Bending moment acting in the beam.

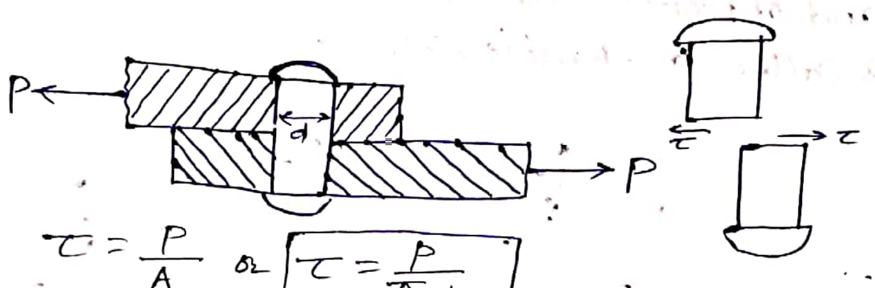
y = D/s of extreme fibre from neutral axis.

Z = Section Modulus = $\frac{I}{y}$



(4) Transverse shear stress :-

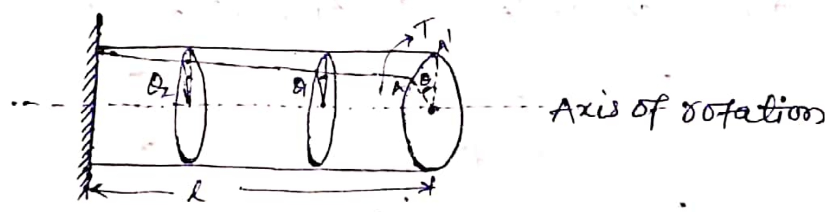
when a section is subjected to two equal and opposite forces acting tangentially across the section such that, it tends to shear off across the section. The stress Produced is called as transverse shear stress.



$\tau = \frac{P}{A}$ or $\boxed{\tau = \frac{P}{\frac{\pi}{4}d^2}}$

(5) Torsional Shear stress :-

when a machine component is under the action of two equal and opposite couples i.e twisting moment or torque, then component is said to be torsional shear stress.



$\theta > \theta_1 > \theta_2$ $\boxed{\theta \propto l}$

Torsional eqn.

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

where T = Torque applied. ~~Twisting or Torsional Shear stress~~

J = Polar M.I.

τ = Torsional shear stress.

r = Radius of the shaft.

G = modulus of rigidity

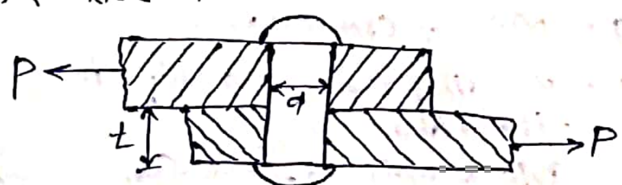
θ = Angle of twist

l = length of the shaft.

(6) Crushing stress:-

it is defined as the localized compressive stress at the surface of contact between two components of a machine. which are relatively at rest.

Ex.- Cotter joint, knuckle joint.



$$\boxed{\sigma_{ck} = \frac{P}{d \cdot t \cdot n}} = \frac{\text{Load}}{\text{Projected area of contact}}$$

where d = dia. of rivet.

t = Thickness of plate

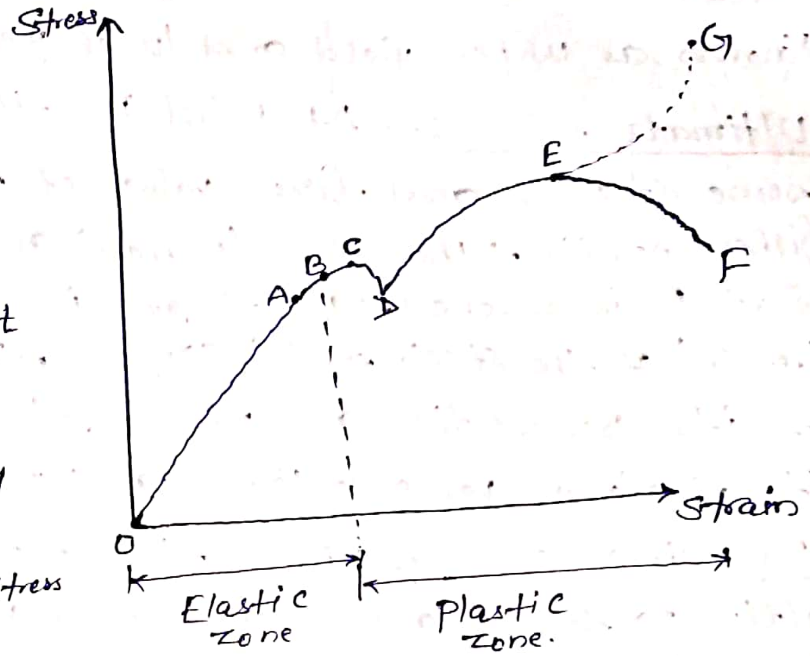
n = no's of rivets.

(7) Bearing Stress:-

it is defined as the localised compressive stress at the surface of contact between two members of a machine part which have relative motion b/w them.

$\frac{P}{l \cdot d} =$ Bearing stress (Pressure) $= \frac{\text{Load}}{\text{Projected area of contact}}$

Stress - strain diagram for ductile material (Mild steel)



OA → Proportional Limit

B → Elastic limit

C → upper yield Point

D → Lower yield Point

E → ultimate stress

F → Failure zone.

1. Proportional limit :- We see from the diagram that from point O to A is a straight line, which represents that the stress is proportional to strain. it is thus obvious that Hook's law holds good up to point A. and it is known as proportional limit.

2. Elastic limit :- it may be noted that even if the load is increased beyond point A up to point B, the material will regain its shape and size when the load is removed. this means that material has elastic properties up to point B. This properties is known as elastic limit. It is defined as the stress developed in the material without any permanent deformation.

3. yield point :- if the material is stressed beyond point B, the plastic stage will reach i.e on the removal of the load, the material will not be able to recover its original shape and size. From ~~point~~ fig. we see that beyond point B, the strain increases faster rate with any increase in stress until the point C is reached.

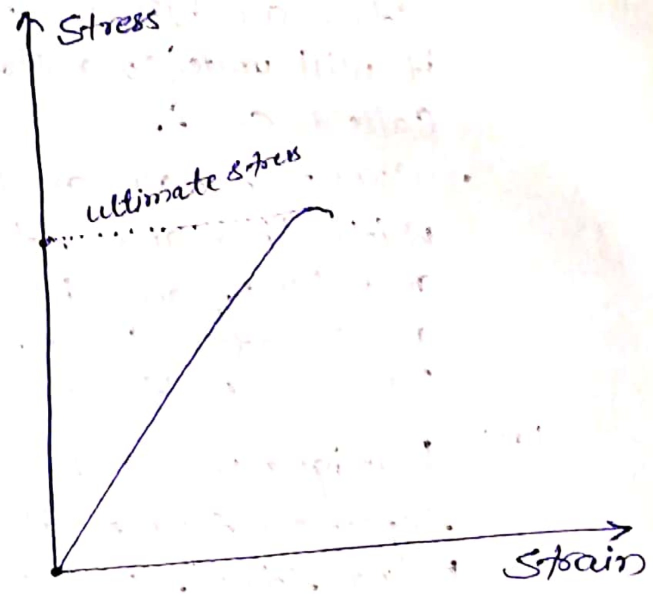
At this point, the material yields before the load and there is an appreciable strain without any increase in stress. Hence there are two yield points C and D and known as upper yield and lower yield points.

4. Ultimate Stress :- At point D, the specimen regains some strength and higher values of stresses are required for higher strain. The stress (or load) goes on increasing till the point E is reached. The gradual increase in the strain (length) of the specimen is followed with the uniform reduction of its cross-section area. At E, the stress which attains its maximum value is known as ultimate stress.

5. Breaking Stress :- After the specimen has reached the ultimate stress, a neck is formed which decreases the crossectional area of the specimen. A little consideration will show that the stress necessary to break away the specimen is less than the maximum stress. The stress is therefore reduced until the specimen breaks away at point F. The stress corresponding to point F is known as breaking stress.

Note :- The breaking stress (i.e stress at F which is less than E) appears to be some what misleading. As the formation of a neck take place at E which reduces the crossectional area, it causes the specimen suddenly to fail at F. if each value of the ~~specimen~~ strain between E and F, the tensile load is divided by the reduced crossectional area at the narrowest part of the neck, then the true-stress-strain curve will follow the dotted line EG.

* Stress- strain for Brittle material. (cast iron)



The usual dividing line between ductility and brittleness is 5% elongation. A material having less than 5% elongation is known as brittle material and one which elongation is more than 5% is termed as ductile material.

* Properties of the material:-

(1) Strength
(2) Stiffness → K = P/δ
(3) Elasticity
(4) Plasticity
(5) Ductility
(6) ~~Stiffness → K = P~~ Malleability → To convert sheet metal.
(7) Toughness → Resist fracture due to high impact load like hammer blows.
(8) ~~Malleability~~ Machinability.
(9) Resiliance :- It is the property of a material to absorb strain energy with in elastic limit without any permanent deformation. This is desirable in case of spring material.

- Proof resiliance :- It is the max^m strain energy that can be stored in the body upto elastic limit.

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(10) Creep :— When a Component is subjected to Constant stress at high temperature over a long period of time it will undergo a slow and Permanent deformation Called creep.

- Creep is slow and Progressive deformation of material with time and stress. Creep is a function of stress and temp. and becomes important for Component operating at elevated temp.

Example — Bolts and pipes in thermal power plant.

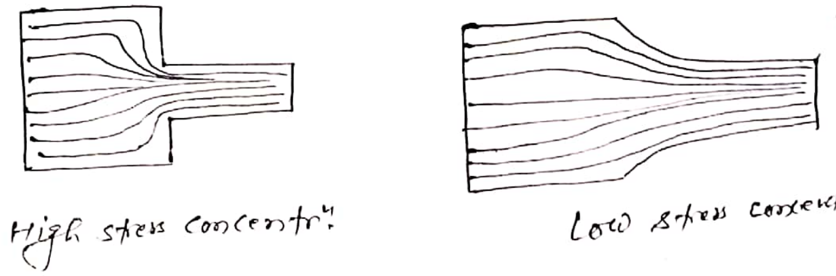
(11) Fatigue :— When a material is subjected to repeated stresses, it fails at stress below the yield point stress. Such type of failure of material is Known as fatigue.

(12) Hardness :— it is very important of the material it is resistance to wear, scratching, deformation and machinability etc.

* Methods of Reducing Stress-Concentration:-

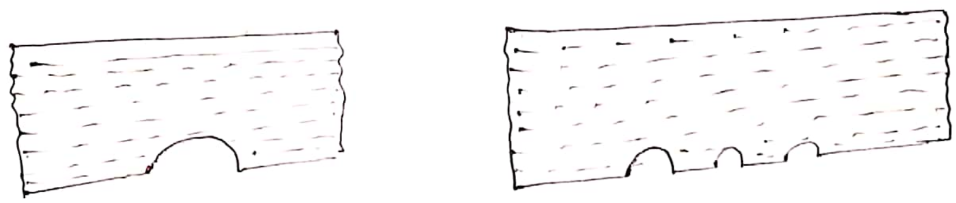
There are following methods are available to reduce the stress concentration.

1. Provide a fillet radius so that the crossection may change gradually.



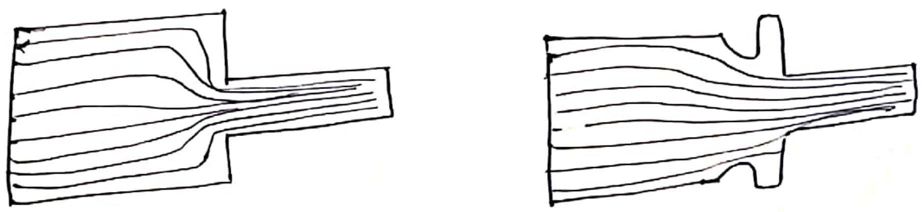
(2). Some time an elliptical fillet is used.

(3) If notch is unavoidable it is better to provide a number of small notches rather than long one. This reduces the stress-concentration to a large extent.



(4) If a Projection is unavoidable from design Consideration, it is preferance to provide a narrow notch than a width notch.

(5) Stress releiving groove are sometimes provided.



* Stress-Concentration :-

Stress-Concentration is defined as the location of high stresses due to the irregularities present in the component and abrupt changes of the cross-section.

In order to consider the effect of stress concentration and find out localised stresses, a factor called stress-concentration factor is used. It is denoted by K_t and defined as-

$$K_t = \frac{\text{Highest value of actual stress near discontinity}}{\text{Normal stress obtained by elementary equation for min}^{m} \text{ crossection.}}$$

* Causes of Stress-Concentration :-

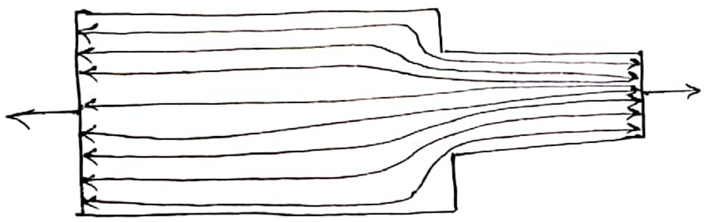
(i) Change in crossection such as stepped axle, key-ways, grooves threaded holes.

(ii) Concentrated load applied at minimum area of machine parts, such as contact between gear teeth, a beam and its supports.

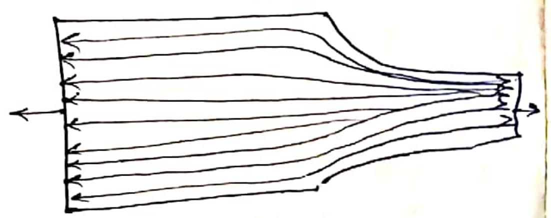
(iii) Variation of Mechanical properties of materials from point to point due to cavities, cracks or air pockets.

(iv) surface irregularities or poor surface finish.

(v) it occurs for all kinds of stresses in presence of fillets, notches, holes, key ways etc.



Stress concentration

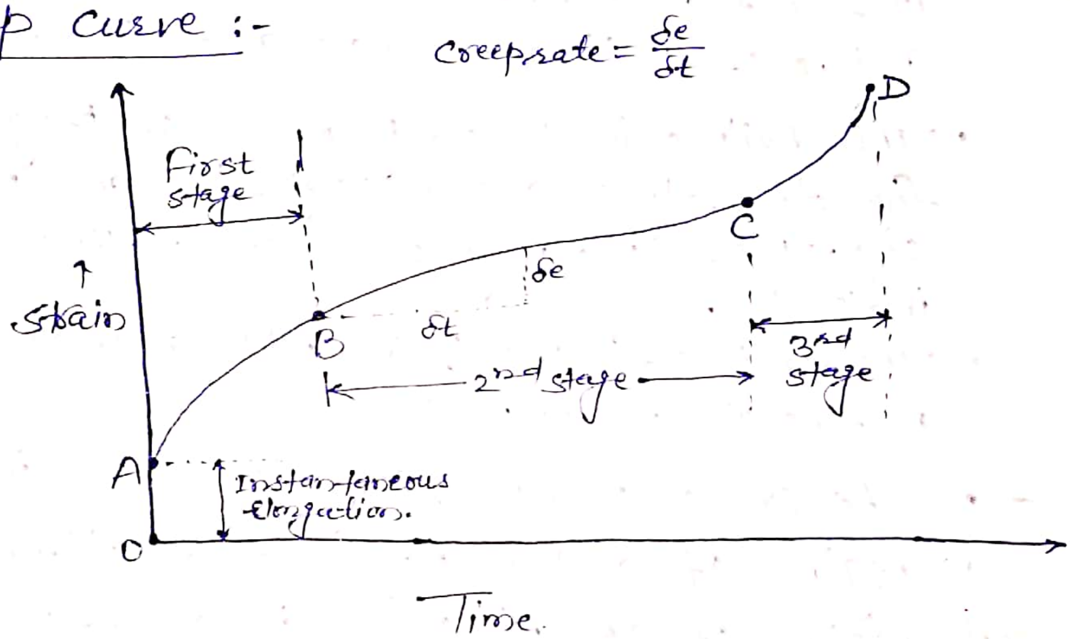


Preffered

* Creep and creep curve :-

→ When a component is subjected to constant stress at high temperature over a long period of time, it will undergo a slow and permanent deformation called creep. Creep is slow and progressive deformation of material with time under constant stress.

* Creep curve :-



When the load is applied at the beginning of creep test, instantaneous elastic deformation OA occurs.
This elastic deformation is followed by creep curve ABCD.
· Creep occurs in three stages.

(a) First Stage :-

It is called primary creep, During this, creep rate decreases i.e slope of creep curve increases from A to B Progressively with time, Hence, the metal strain hardens to support the external load.

(b) Second Stage :-

It is called secondary creep, Here the creep rate is almost constant. This stage occupies a major portion of the life of components.

(c) Third stage:-

It is called as tertiary creep. During this, creep rate is accelerated due to necking and finally results into fracture at point D.

* Fatigue.

When a material is subjected to repeated stresses, it fails at stresses below yield point stresses. such a type of failure is called as fatigue failure.

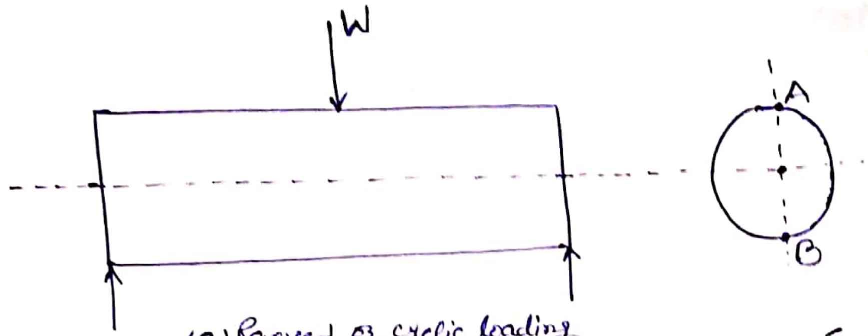
- This failure is caused by means of progressive crack formation, that are actually fine and microscopic size. The failure may occur even without any prior indication.
- The fatigue failure begins with a crack at some point in the component.

→ • The crack is more likely to occur in the following regions.

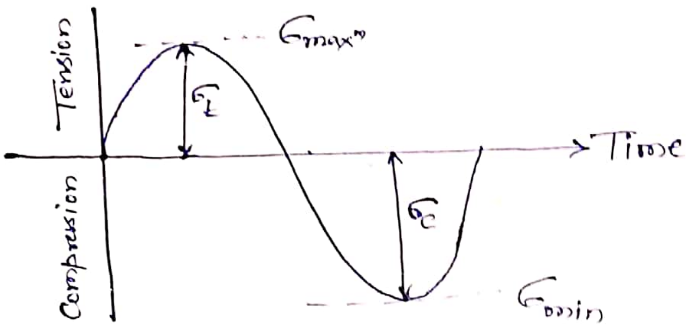
(i) Regions of discontinuity such as oil holes or keyways.
(ii) Regions of abrupt change in crossection.
(iii) Regions of irregularities in ~~material~~ machining operation such as machining scratches, stamp mark or inspection marks.
(iv) Internal cracks in materials like blow holes.

Endurance limit :-

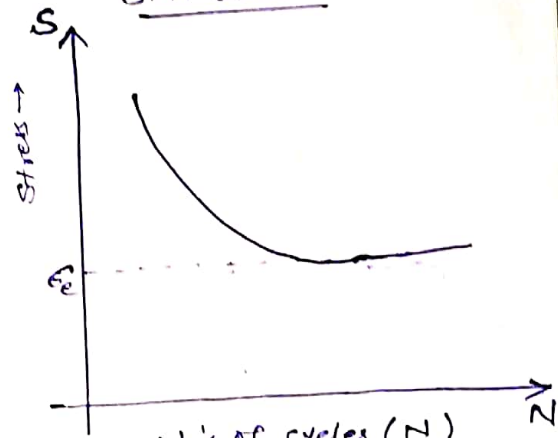
* S-N Curve :-



(a) Reversed or cyclic loading



(b) Complete reverse cycle



(c) Endurance or fatigue load (S-N curve).

- Endurance or fatigue limit is defined as maximum value of completely reversed bending stress, which a standard specimen can withstand without failure, for infinite numbers of cycles of loads.

- A little consideration will show that, if the stress is kept lower than the dotted line value, the material will not fail, whatever may be the number of cycle. This dotted line stress is known ~~tha~~ as endurance or fatigue limit.

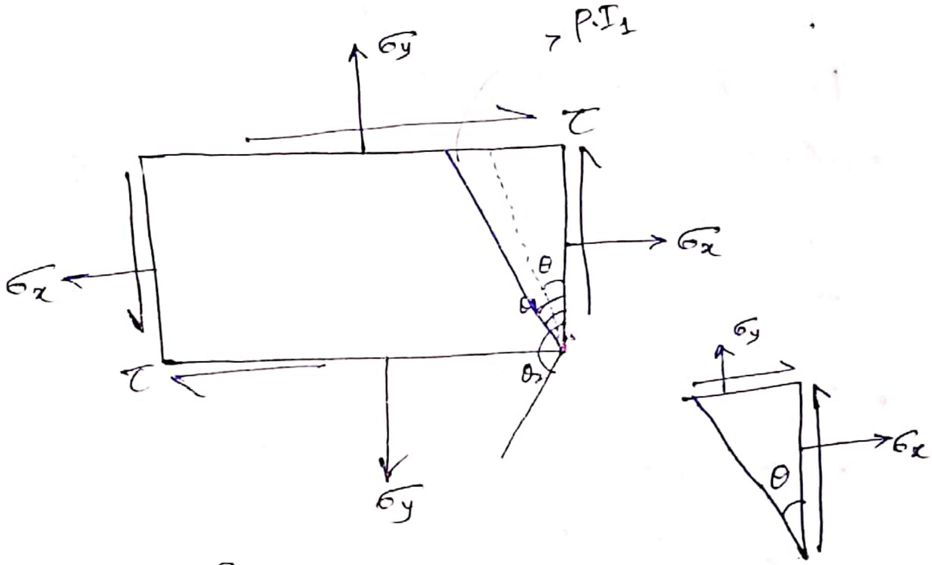
Day 5

Principale stress and strain

When ever a multiload situation is given and it is desired to design given Component then max^m normal and max^m shear stress in the component must be known. These two values can only be Calculated by using Principle stress Concept.

Principale stress are the normal stresses located on Principle plane.

Principle plane is a plane inside a loaded component on which there is no shear stress Component.



Normal stress at any Angle θ.

$$\sigma_\theta = \frac{\text{Normal force}}{\text{Resisting force}} = \frac{\sigma_x+\sigma_y}{2} + \frac{\sigma_x-\sigma_y}{2}\cos 2\theta + \tau \sin 2\theta$$

shear stress at any angle θ.

$$\tau_\theta = \frac{\text{Tanj. force.}}{\text{Resisting area}} = \frac{\sigma_x-\sigma_y}{2}\sin 2\theta - \tau\cos 2\theta$$

For example:-

If $\sigma_x = 50$ MPa.
$\sigma_y = 30$ MPa
$\tau = 10$ MPa
$\theta = 15^\circ$

$\therefore \sigma_\theta = 53.66$ MPa.

$\tau_\theta = -3.66$ MPa.

-ve sign means opposite dir^n which Consider.

For Principle plane

$\tau_{\theta_1} = 0$

$$\frac{\sigma_x - \sigma_y}{2} \sin 2\theta_1 - \tau \cos 2\theta_1 = 0$$

$$\frac{\sigma_x - \sigma_y}{2} \sin 2\theta_1 = \tau \cos 2\theta_1$$

$$\tan 2\theta_1 = \frac{2\tau}{\sigma_x - \sigma_y}$$

$$\boxed{\theta_1 = \frac{1}{2}\tan^{-1}\left(\frac{2\tau}{\sigma_x - \sigma_y}\right)}$$

* when there is no involvement of angle -

$$\sigma_1 = \frac{\sigma_x+\sigma_y}{2} + \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + \tau^2} \longrightarrow \text{max}^m \text{ P-stress}$$

$$\sigma_2 = \frac{\sigma_x+\sigma_y}{2} - \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + \tau^2} \longrightarrow \text{min}^m \text{ P-stress}$$

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2}$$

From previous example.

- $\theta_1 = 22.5°$

  $\sigma_{\theta_1} = 54.11$ MPa. $\longrightarrow$ This is σ_1 (Max^m principle stress)

  $\tau_{\theta_1} = 0$ MPa.

  اس سے بڑی کوئی value of stress اس Component کے اندر نہیں ہو سکتی ہے۔

- $\theta_2 = \theta_1 + 90°$ [$\because$ they are $\perp$r. to each other]

  $\therefore \sigma_{\theta_2} = 25.85$ MPa $= \sigma_2$ (Min^m principle stress)

  اس سے چھوٹی کوئی value of normal stress نہیں ہو سکتی ہے۔

  $\tau_{\theta_2} = 0$ MPa.

- $\theta_3 = \theta_1 + 45°$

  $\therefore \tau_{max} = \tau_{\theta_3}$

  & $\sigma_{\theta_3} = 40$ MPa.

- $\theta_4 = \theta_3 + 90$

  $\tau_{\theta_4} = -14.14$ MPa.

θ_3 is the location of max^m shear stress

θ_4 " " " min^m " "

$\boxed{\theta_4 \perp r\ \theta_3}$

جہاں system کی max^m shear stress ہوتی ہے وہاں σ کوئی ضروری نہیں کہ 0 ہو جائے۔ 0 ہو بھی سکتی ہے اور نہیں بھی ہو سکتی ہے۔

$\sigma_{\theta_1} = \sigma_1 \longrightarrow$ Means any plane which is at θ_1 & max^m normal stress $= \sigma_1$

$\sigma_{\theta_2} = \sigma_2$ "

$$\theta_1 = \frac{1}{2}\tan^{-1}\left[\frac{2\tau}{\sigma_x - \sigma_y}\right]$$

$\theta_2 = \theta_1 + 90° \longrightarrow \theta_2$ is the location of min^m principle stress.

Day-6. 1.11

* Theories of failure:-

(1) Max^m principle stress theory (Rankine theory) → square
(2.) Max^m shear stress theory (Guest theory or Tresca) → Hexagon.
(3) Max^m principle strain theory (Venant theory) → Parallelogram.
(4) Max^n shear strain energy theory (von-misses) → Ellipse.
(5) Max^m strain energy theory (Haigh) → Ellipse.

* Max^m principle or normal stress theory or Rankine theory:

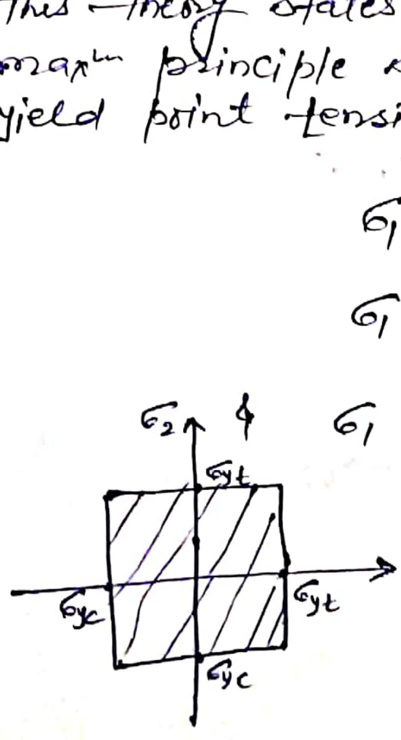
Suitable for brittle material.

This theory states that failure is going to occur when max^m principle stress of multiload situation exceeds yield point tensile stress of simple test (single load situation).

$$\sigma_1 > \sigma_{yt} \longrightarrow \text{Failure.}$$

$$\sigma_1 = \frac{\sigma_{yt}}{FOS} \longrightarrow \text{safe.} \longrightarrow \text{for ductile material.}$$

$$\sigma_1 = \frac{\sigma_{ut}}{FOS} \longrightarrow \text{safe} \longrightarrow \text{for brittle material.}$$



* Any point, inside or on the square is safe.

(2.) Max^m shear stress theory (Guest theory) :-

suitable for ductile material

This theory states that failure is going to occur when max^m shear stress of multiload situation exceed the simple tensile test

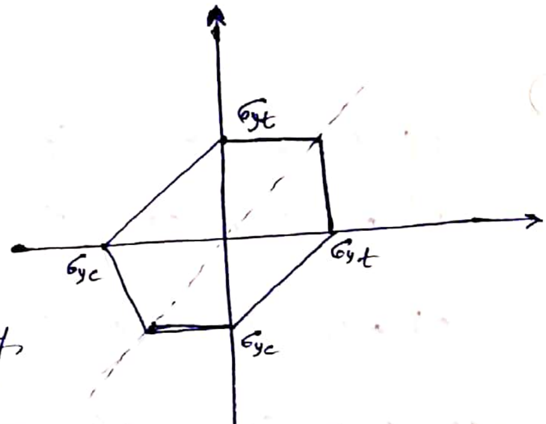
$$\cancel{\tau_{max^m} > \sigma_y}$$

$$\tau_{max} > \frac{\tau_y}{Fos}$$

where τ_{max} = max^m shear stress
τ_y = yield point shear stress
σ_{yt} = yield point ~~strength in~~ stress in tension.

$$\frac{\sigma_1 - \sigma_2}{2} = \frac{\sigma_{yt}}{Fos}$$

$$\boxed{\sigma_1 - \sigma_2 = \frac{\sigma_{yt}}{Fos}}$$



(3) Max^m distortion energy theory or Haigh theory:

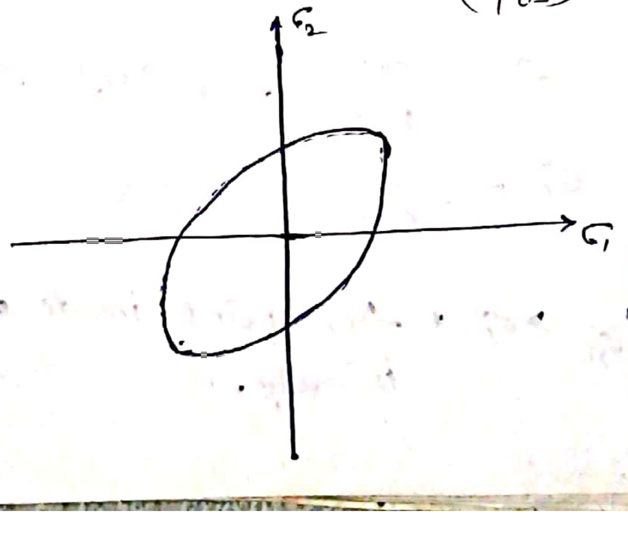
Ductile material only.

This theory states that failure is going to occur when max^m strain energy in multiload situation exceed strain energy of simple test.

$$U_{max} > U_{yt}$$

$$U_{max} = U_{yt}$$

$$\sigma_1^2 + \sigma_2^2 - \frac{2\sigma_1\sigma_2}{m} = \left(\frac{\sigma_{yt}}{Fos}\right)^2$$



* Standardisation :-

Standardisation is defined as the process of establishing standards so as to minimize the varieties in the ~~Com~~ characteristics include materials, dimensions, quality etc. In India ISI is responsible for evolving all types of technical standards.

* ~~Uses of~~ Benefits of Standardisation:-

(i) It helps in manufacture of component quickly and economically.

(ii) It makes the mass production possible, reducing the manufacturing costs and labour requirement.

(iii) The standardisation of specifications and methods of testing the machine elements help in improving quality and hence their service life.

(iv) The repair and maintanance of the machines is simplified since the worn out ~~and~~ or damaged parts can easily be replaced by standard ones.

(v) It reduces the time and efforts needed to create and manufacture new machines since the standard elements and units can be used to assemble a new machine.

Day-7.4.8.

Nizamuddin Ansari
Diploma 2nd shift
ME- 6th sem
Sub- Design of M/c element

Chapter - 2.

Design of simple machine parts :-

• Introduction of Cotter Joint :-

→ A Cotter Joint is a temporary fastening and it is used to connect two co-axial rods or bars, which are subjected to axial tensile / or compressive forces.

→ A Cotter is a flat wedge shaped piece of rectangular cross-section and its width is tapered (either on one side or both sides) from top to bottom for an easy adjustment. The taper in cotter varies from 1 in 48 to 1 in 24.

→ The cotter is usually made of mild steel and wrought iron.

* Application of Cotter Joint

(i) Connection of the piston rod to the cross head of a reciprocating steam engine.

(ii) Valve rod and its stem.

(iii) Strap end of connecting rod.

(iv) Lewis foundation bolt and other cottered foundation bolts.

(v) Piston rod to the tail end (rod) in an air pump.

* Advantages of Cotter joint :-

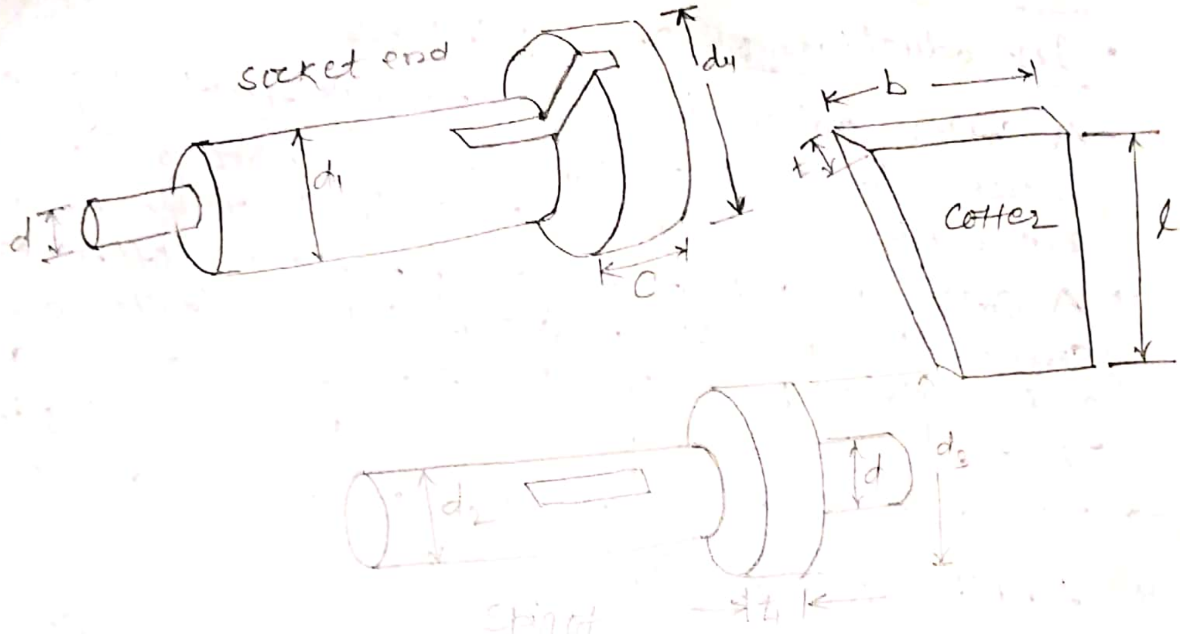
(i) Quick to assemble and dissamble parts.

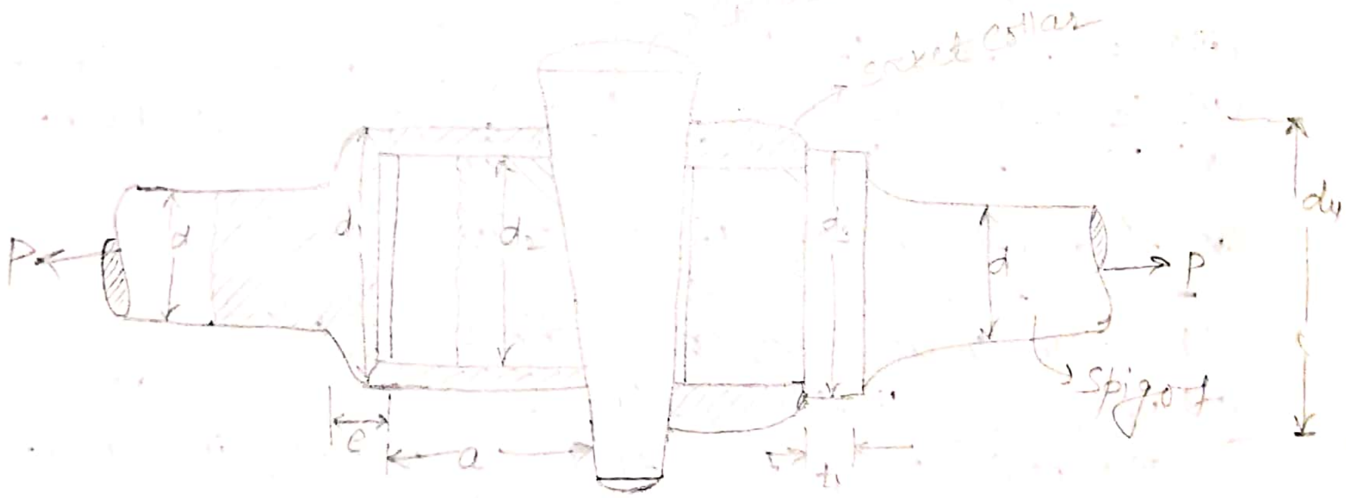
(ii) Very high tightening force due to the wedge action is developed, which prevents loosing of parts in service.

(iii) The joint is simple to design.

(iv) The Joint is easy to manufacture.

* Design of socket and spigot Cotter Joint :-





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Design procedure :-

Let P = Load Carried by the rods.

d = Diameter of the rods.

d_1 = outside diameter of the socket.

d_2 = Diameter of the spigot or inside dia. of the socket.

d_3 = outside diameter of spigot collar.

d_4 = Diameter of socket collar.

t_1 = Thickness of spigot collar.

c = Thickness of socket collar.

b = Mean width of cotter.

l = Length of cotter.

a = D/s from the end of the slot to the end of rod.

σ_t, σ_c, τ = Tensile, Compressive and shearing stresses.

Assumptions :-

(i) There is no stress concentration.

(ii) The load is uniformly distributed over each part of the joint.

- The dimensions for a socket and spigot cotter joint may be obtained by considering the various modes of failure as discused below.

1. Failure of rod in tension.

The rods may be fail in tension due to the tensile load (P)

$$\boxed{P = \frac{\pi}{4} d^2 \sigma_t}$$

$d = ?$

2. Failure of spigot in tension across the weakest section (or slot).

Since the weakest section of the spigot is that section which has a slot in it for cotter.

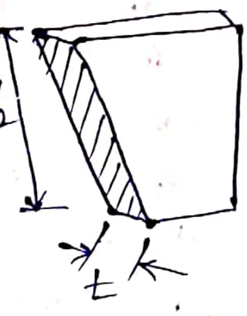
$$P = \left[\frac{\pi}{4}(d_2)^2 - d_2 t\right] \sigma_t$$

$d_2 = ?$

Note:- In actual practice, the thickness of cotter is usually taken as $\frac{d_2}{4}$.

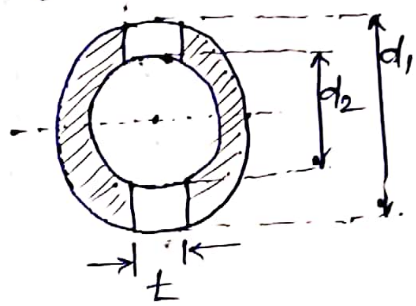
(3) Failure of the rod or cotter in crushing.

$$P = d_2 t \cdot \sigma_c$$



(4) Failure of the socket in tension across the slot.

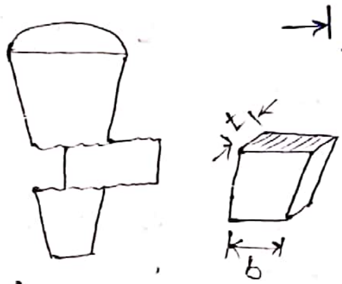
$$P = \frac{\pi}{4}\left[d_1^2 - d_2^2\right] - \left[d_1 - d_2\right] t$$



(5) Failure of cotter in shear

~~$P = (d_4 - d_2) t \times \sigma_c$~~

$$\boxed{P = 2bt \times \tau}$$



(6) Failure of the socket collar in crushing.

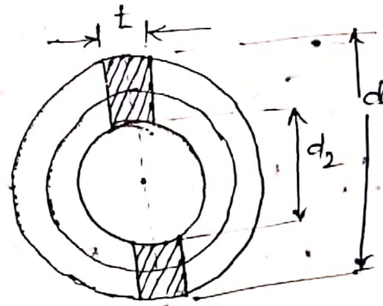
$$\boxed{P = (d_4 - d_2) t \times \sigma_c}$$

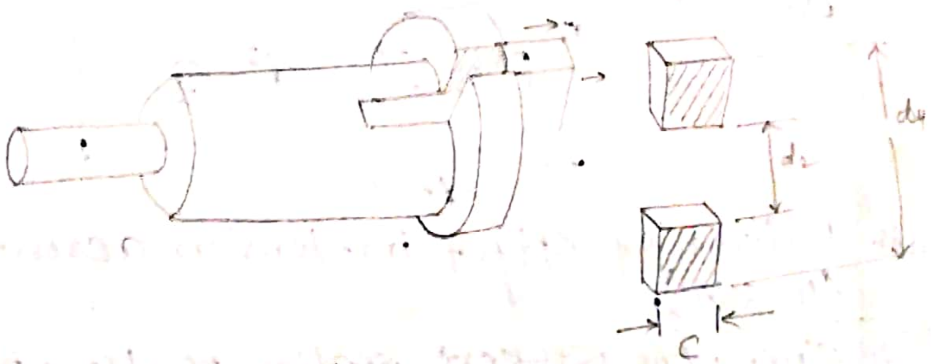
d_4 ?

(7) Failure of socket end in shearing.

$$\boxed{P = 2(d_4 - d_2) \cdot c \times \tau}$$

c = ?





(8). Failure of spigot end in shear (shear failure of spigot end beyond slot.)

$$\boxed{P = 2a \times d_2 \times \tau}$$

(9) Failure of spigot collar in crushing

$$\boxed{P = \frac{\pi}{4}\left[d_3^2 - d_2^2\right] \sigma_c}$$

(10) Failure of the spigot collar in shearing

$$\boxed{P = \pi d_2 \times t_1 \times \tau}$$

Note:→ when all the parts of the joint are made of steel, the following properties in terms of diameter of rod are generally used.

$d_1 = 1.75d$, $d_2 = 1.21d$, $d_3 = 1.5d$, $d_4 = 2.4d$

$a = c = 0.75d$, $b = 1.3d$, $l = 4d$, $t = 0.31d$, $t_1 = 0.45d$, $e = 1.2d$.

* Knuckle joint

A knuckle joint is used to connect two circular rods, which are under action of ~~tension~~ tensile loads. A knuckle joint may be easily disconnected for adjustments or repair. It allows relative angular movement of the rod in the plane about axis of pin.

* Difference between Cotter joint and Knuckle joint :-

Cotter joint	Knuckle joint
(i) Capable of taking both tensile / compressive loads.	(i) Can take only tensile load.
(ii) Can not permit angular movement between rods.	(ii) Can permit limited angular movement between rods.
(iii) Cotter is not subjected to bearing failure.	(iii) Knuckle pin is subjected to bearing failure.
(iv) Since there is no forked end, it is not known as forked pin joint	(iv) It is known as forked pin joint.
(v) Taper and clearence is provided in the Cotter.	(v) No taper or clearence is provided
(vi) Cotter is rectangular in section	(vi) Knuckle pin is circular in section.

* Application

Cotter joint	Knuckle joint
(a) Cotter foundation bolt.	(a) Tie bar and roof truss.
(b) Big end of the connecting rod of a steam engine	(b) Links of suspension bridge.
(c) Joining piston rod with cross-head.	(c) Valve mechanism.
(d) Joining two rods with a pipe.	(d) Fulcrum of the lever
	(e) Links of bicycle chain
	(f) Joint for rail shifting mechanism.

* Reason's for Taper provided on Cotter:-

- Cotter is a flat wedge shape piece of rectangular cross-section and its width is tapered (either on one side or both sides) from one end to another. Reason's for providing taper are –

(a) it helps the easy removal.
(b) Due to it, Cotter remains in its positions.
(c) it Provides maximum friction area.

- Taper provided is usually between 1 in 48 to 1 in 24. Higher taper than this increases the risk of slipping back.
- Generally taper is provided only on one side, as machining taper on both sides is rather difficult.
- Taper provided is usually small, so that self locking can be achieved. if taper is more and joint is subjected to variable loading, then Cotter may be provided with some locking arrangement, so as to keep it in position.

* Difference between Cotter and Key:-

Keys	Cotter
(i) Key's are driven parallel to the axis of shaft	(i) Cotter is normally driven at right angle to the axis of connected parts.
(ii) it is subjected to torsional shear stress and crushing stresses.	(ii) The Cotter is subjected crushing stress and shear stress.
(iii) it resists shear over longitudional section.	(iii) The cotter resist shear over transverse sections.

* Purpose of Providing Clearance in Cotter Joint:-

- A clearance of 2 to 3 mm is provided between the slots and Cotter. When Cotter is driven in slots, the two rods are drawn together, until the spigot rests on the socket collar.

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