DIGITAL CLASSROOM ENVIRONMENT

A NEW PEDAGOGY IS EMERGING

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C. Daksheeswara Reddy
Assistant Professor,
Mechanical Engineering

Bloom's Taxonomy - Cognitive

1 Remember

Behavior: To recall, recognize, or identify concepts

2 Understand

Behavior: To comprehend meaning, explain data in own words

3 Apply

Behavior: Use or apply knowledge, in practice or real life situations

4 Analyze

Behavior: Interpret elements, structure relationships between individual components

5 Evaluate

Behavior: Assess effectiveness of whole concepts in relation to other variables

6 Create

Behavior: Display creative thinking, develop new concepts or approaches

(Source: TASK Training material)

MACHINE DESIGN-I (R18A0314)

3RD YEAR B. TECH I- SEM, MECHANICAL ENGINEERING

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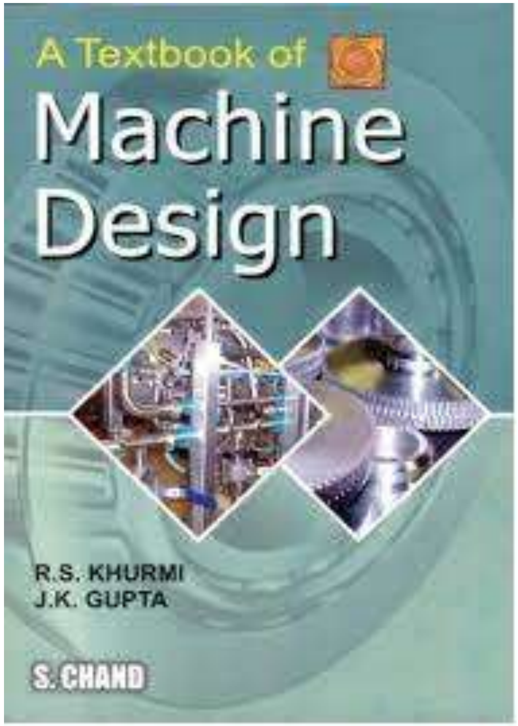
COURSE OBJECTIVES

UNIT - 1	**CO1:** The students should be able to understand. Types of loading on machine elements and allowable stresses. To apply different materials of construction and their properties and factors determining the selection of material for various applications.
UNIT - 2	**CO2:** To understand Stress concentration and the factors responsible. Determination of stress concentration factor; experimental and theoretical methods. Fatigue strength reduction factor and notch sensitivity factor.
UNIT - 3	CO3: To develop the Knowledge on Basic failure mechanisms of riveted joints. Concepts of design of a riveted joint, welded joints and Bolted Joints to determine the forces in welds and riveted joints and formulate design solution for size of weld and size of rivet

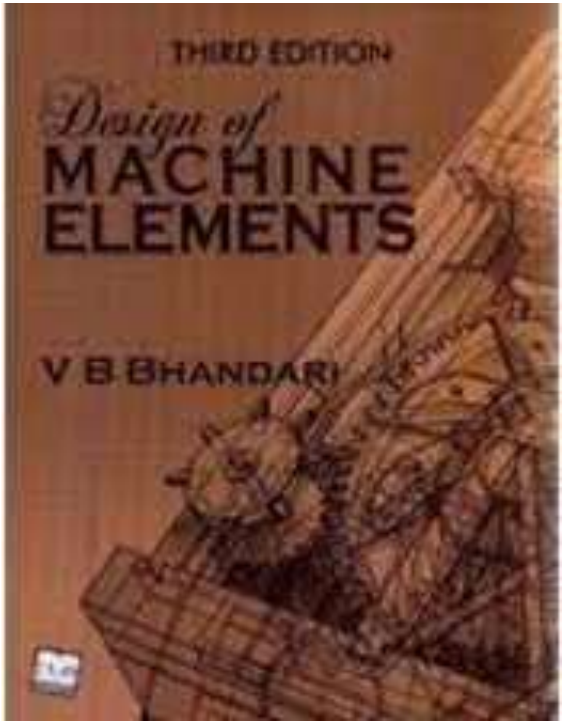
UNIT - 4	CO4: To learn the design Procedure for the different machine elements such as fasteners, couplings, keys, axially loaded joints etc.
UNIT - 5	CO5: To learn the design Procedure for the different Shafts under loading condition, able to know various shafts coupling.

TEXT BOOKS

1. A Textbook of Machine Design by R S Khurmi and J K Gupta

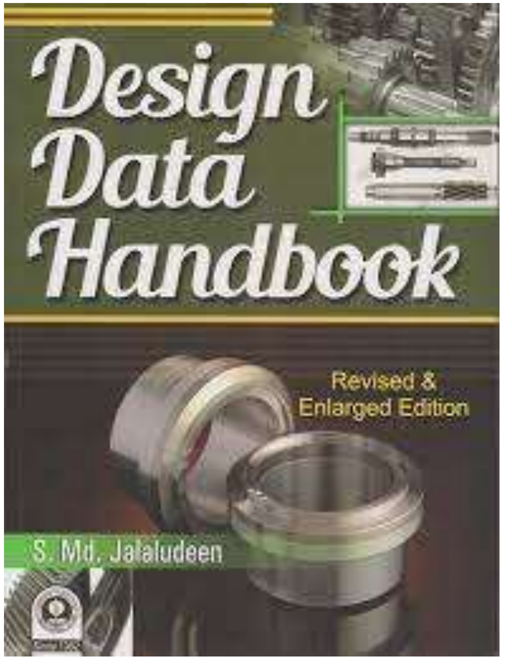


2. Design of Machine Elements by V.B.Bhandari



3.Machine Design by S.Md.Jalaludeen

4.Design Data Hand book by S.Md.Jalaludeen



UNIT 1

CO1: The students should be able to understand. Types of loading on machine elements and allowable stresses. To apply different materials of construction and their properties and factors determining the selection of material for various applications.

UNIT – I (SYLLABUS)

Introduction

- General Considerations in the Design of Engineering materials
- Properties of the materials
- Manufacturing Consideration in design
- BIS Codes for Steels

Design for Static Strength

- Simple and Combined stresses
- Torsional ,Bending and Impact stresses
- Various Theories of failure

- Factor of safety
- Design for strength and Regidity
- Concept of stiffnes in tension , Bending ,Torsion and combined situations

COURSE OUTLINE

UNIT -1

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
1	Introduction – Machine and Importance of Design	Definition of Design.	Understanding of Importance of Design (B2)
2	General considerations in the Design of Engineering Materials and mechanical Properties of Materials	Materials	• Understanding of importance of materials (B2) • Apply or select materials for different components(B3)
3	Manufacturing consideration in Design		• The major objective is to ensure that the product and the manufacturing processes are designed together(B2 & B3)
4	BIS codes of steels		• Recognize the materials(B1)

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
5	Simple & combines stresses	Definition of stress.	• Student able to Understand how to calculate Axial and combined stresses acting in an object(B2)
6	Torsional & Bending Stresses		• Student able to Understanding of how to calculate Bending and Torsional stresses in an object (B2)
7	Various Theories of Failure		• Student able to understand the principles behind various theories of failure and apply different materials(B3)

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LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
8	Factor of safety	Safety	• Student able to understand what is the importance of Factor of safety and how it considered or selected based on maerials(B2)
9	Design for strength & Rigidity		• Student able to understand mechanical properties of strength and stiffness (B2)
10	Concept of stiffness ,Bending ,Torsion and Combined situations		• Student able to understand when the loads applying an object indivisually and combined situations the component with standing or not (B3)

LECTURE 1

Introduction to Machine and Importance of Design

- Basic concept of design in general
- Concept of machine design and their types
- Factors to be considered in machine design
- Design is essentially a decision making process
- For every problem, we need to design a solution

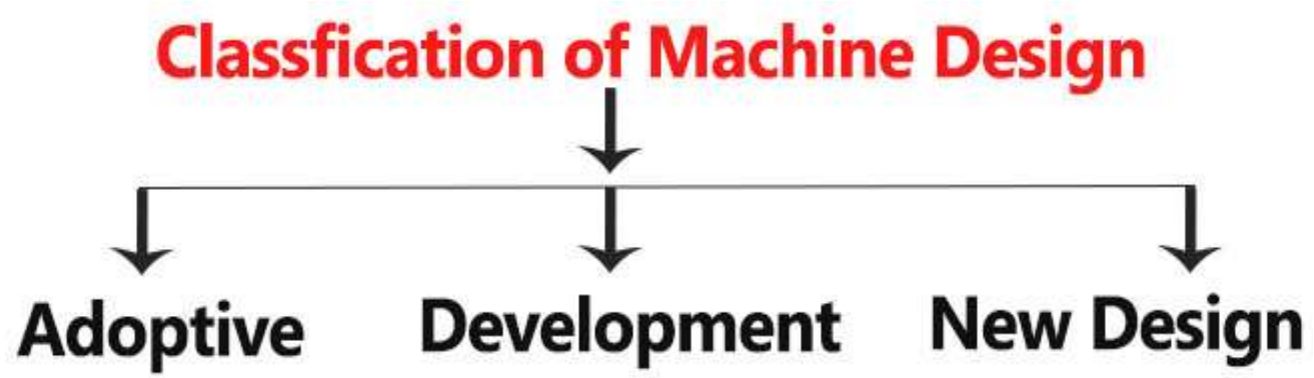
Defination :

Design is to formulate a plan to satisfy a particular need and to create something with a physical reality

(Or)

Machine Design is the creation of new and better machines and improving the existing ones. A new or better machine is one which is more economical in the overall cost of production and operation

Classification of machine design :



Adoptive Design :

In this design the work on existing product with adopt of existing design. These types of design need no special knowledge or skill.

Development Design:

In development design to modify the existing design into a new idea by adopting material different method.

New Design:

This type of design needs lots of research and technical knowledge.

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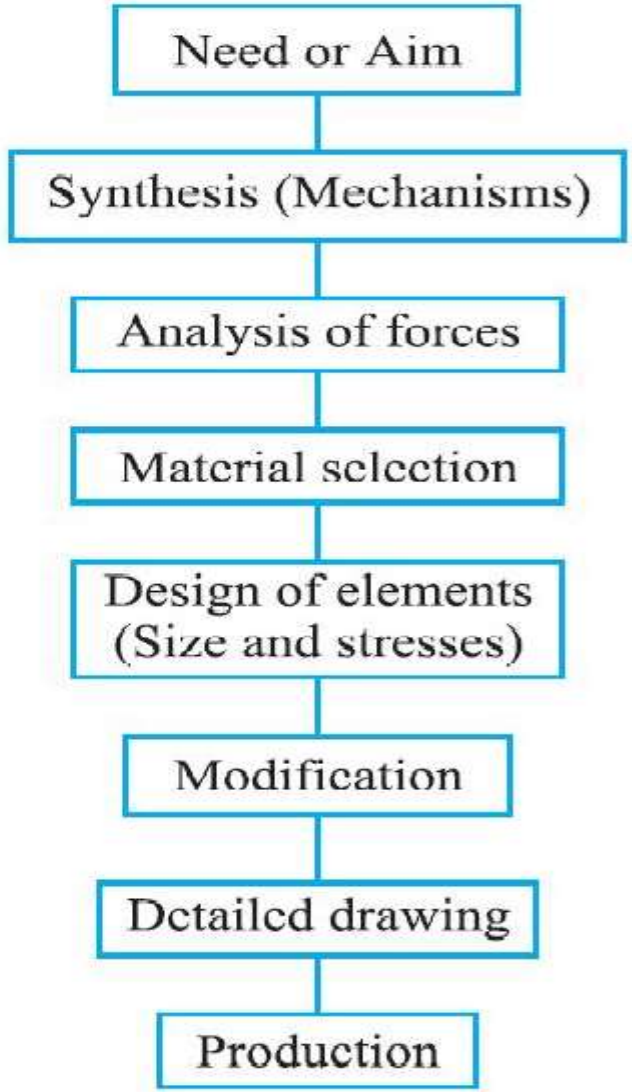
Basic procedure of Machine Design:



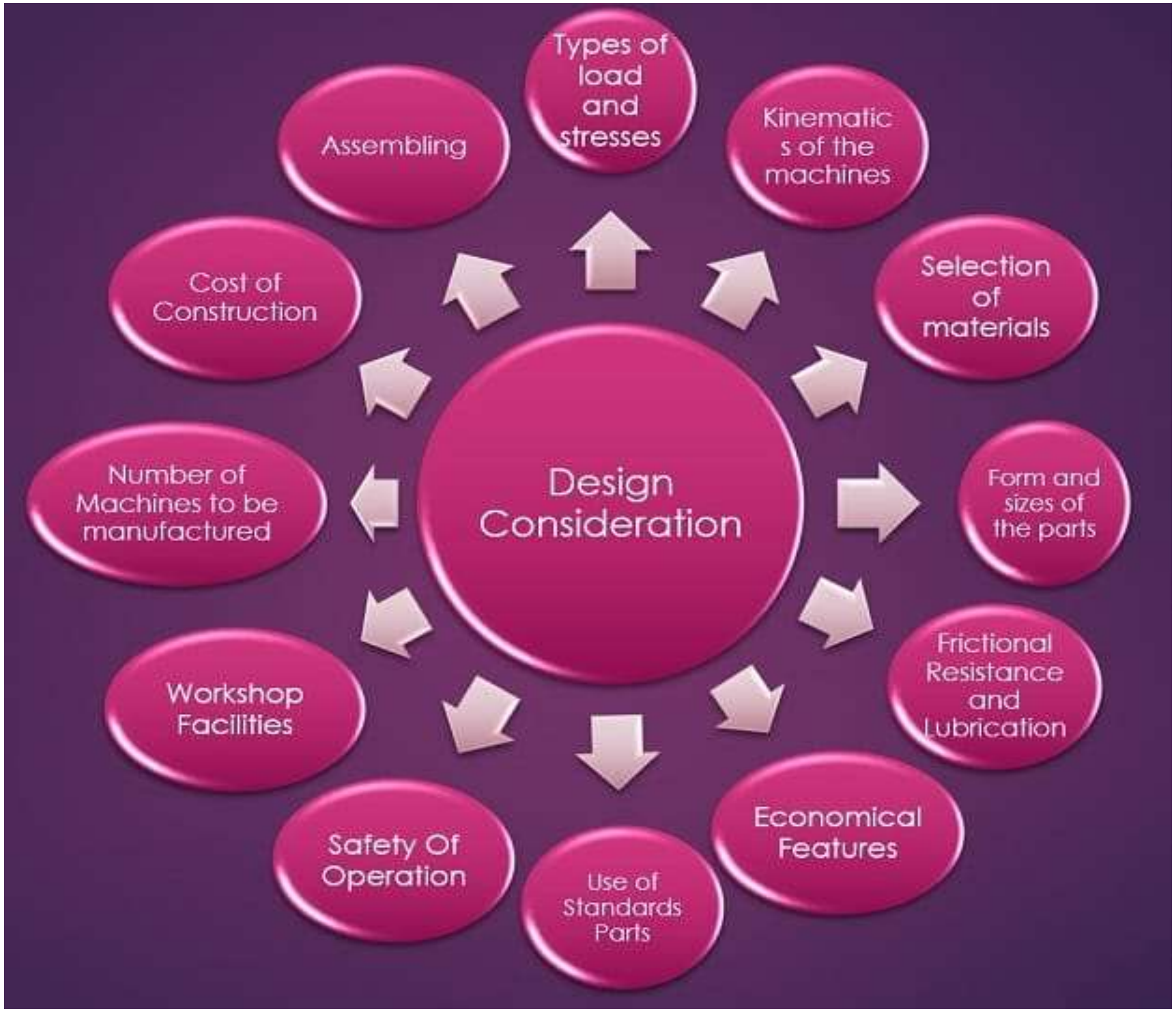
Revision Questions

1. What is Machine
2. What is Design
3. What is Adaptive design
4. What is Development Design
5. What is New Design
6. Explain Steps involved in Design

LECTURE 2

General considerations in Designing a Machine Component

General considerations in Designing a Machine Component:



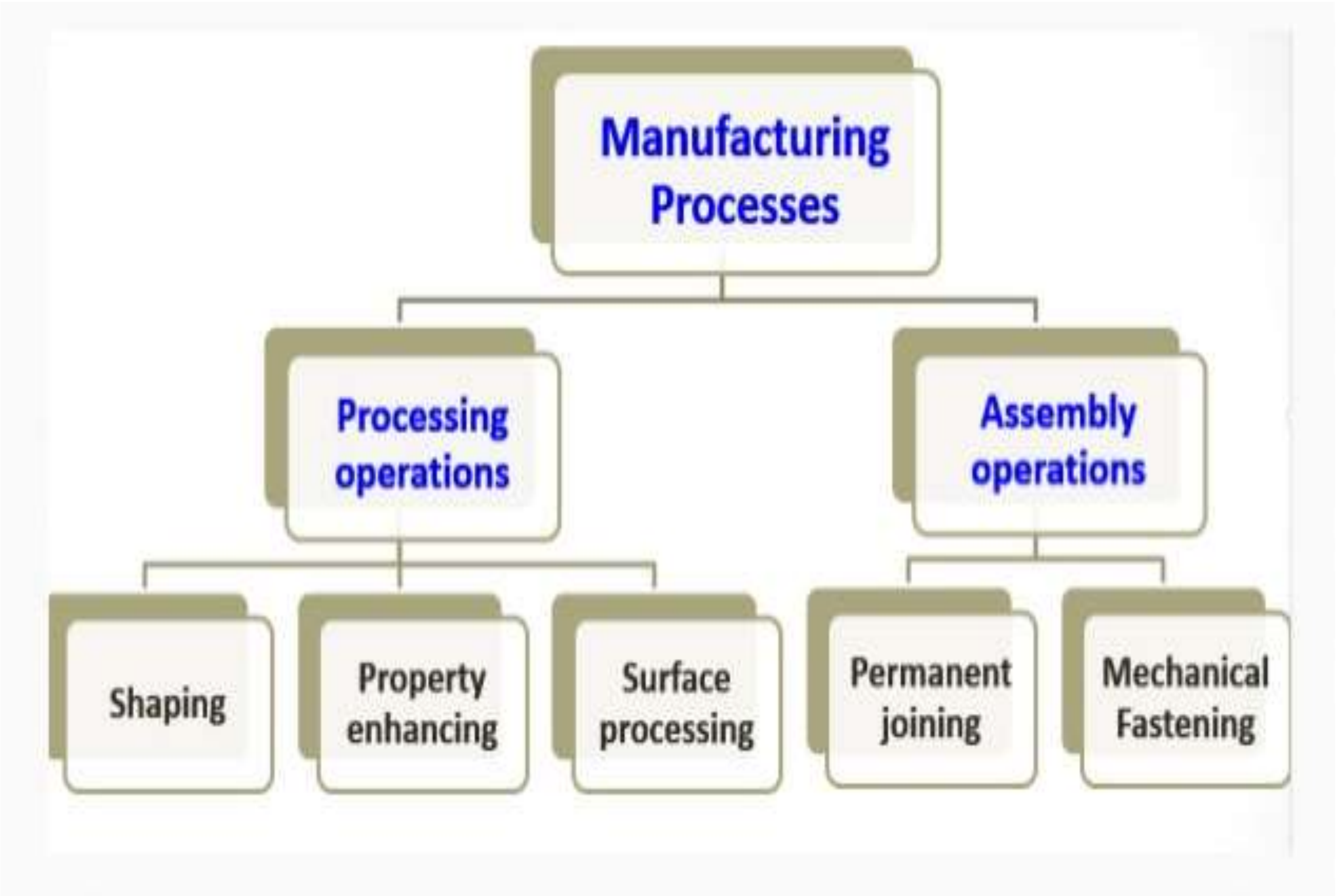
Revision Questions

1. What are the general considerations of Machine design

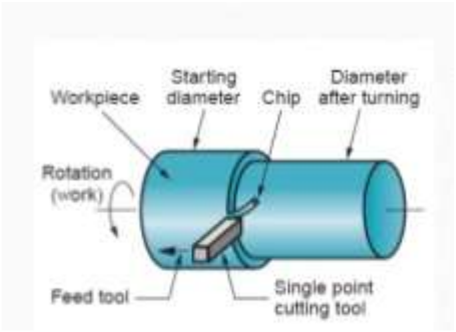
LECTURE 3

Manufacturing Consideration in machine Design

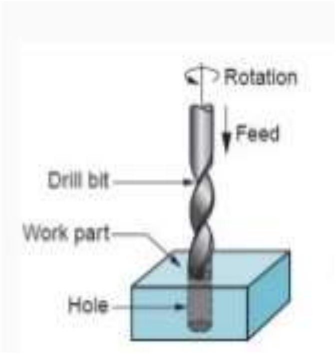
Manufacturing Consideration in machine Design:



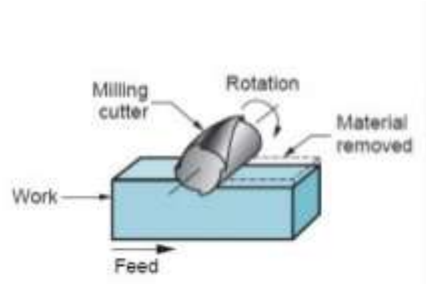
Shaping Operation:



fig(a) Turning

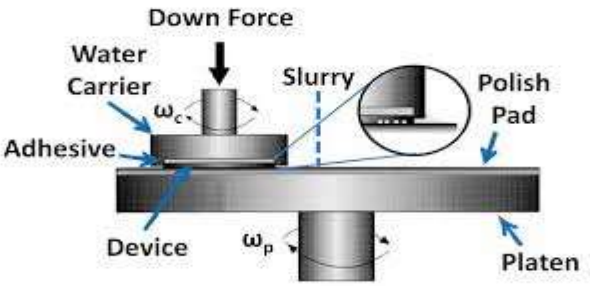


fig(b) Drilling



fig(c) Milling

Surface finishing Operation:



fig(d) Lapping

fig(e) Belt Grinding

Revision Questions

1. What are the manufacturing consideration in machine design

LECTURE 4

BIS Codes for Steel

BIS Codes for Steel:

According's to Bureau of Indian Standards, steels can be designated either based on letter symbols {IS: 1962 (Part I)—1974} or based on numerals {IS: 1962 (Part II)}. Minimum number of symbols is recommended to be used in designating any steel.

LECTURE 5

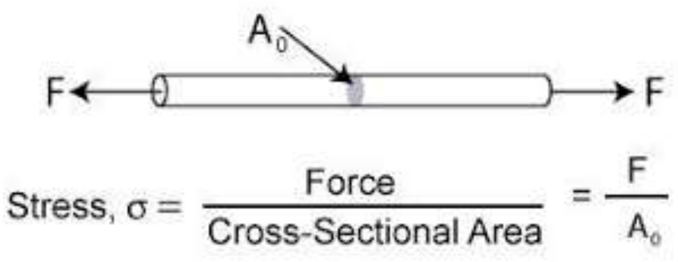
Simple & Combined stresses

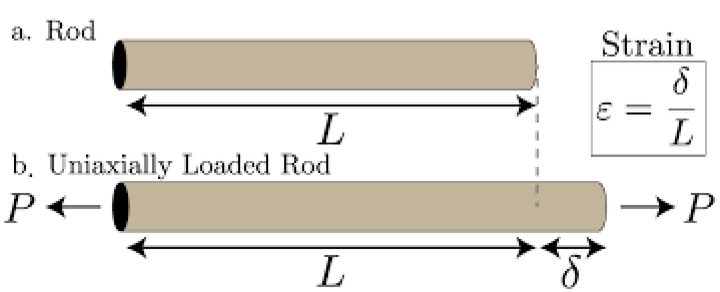
Design for Static Strength:

Static Strength is your ability to hold a pose without movement

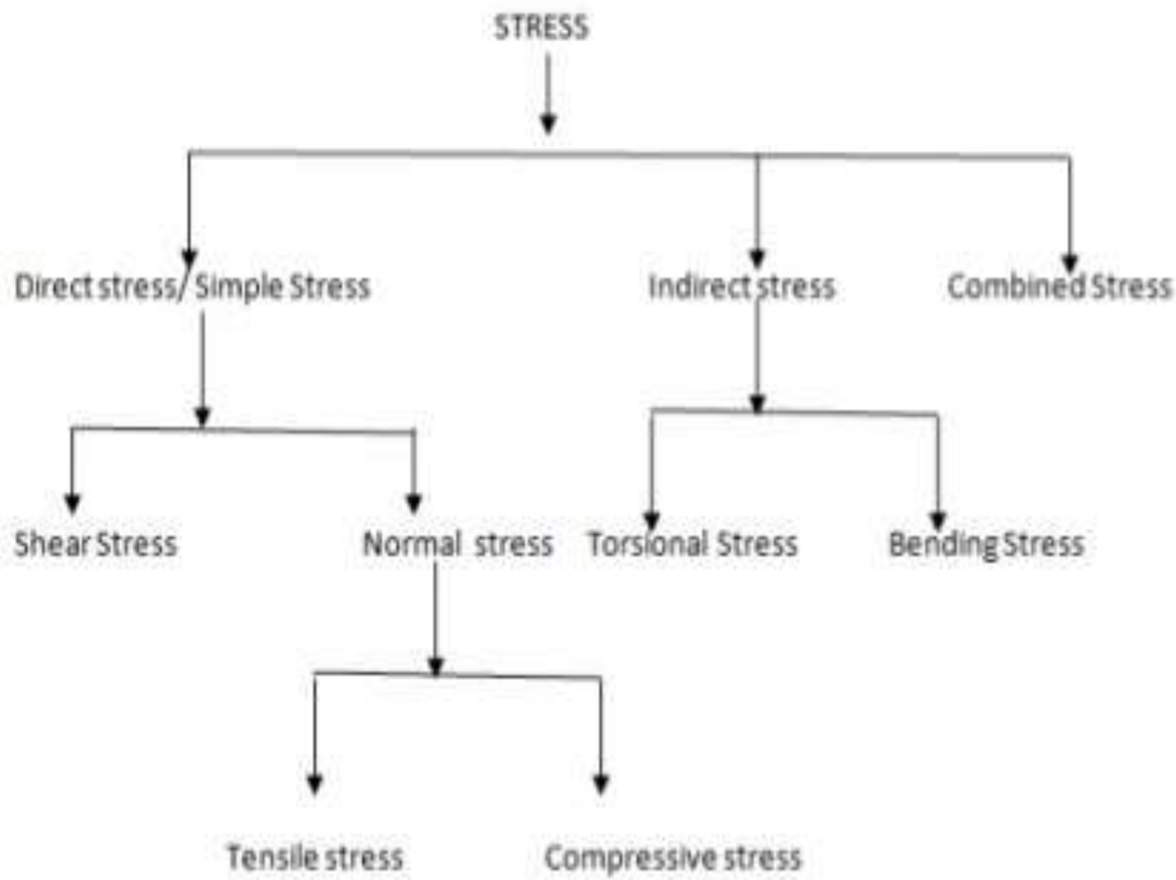
Simple Stress & Strain:

Where a **simple stress** is defined as the internal resistance force that opposes the external force per unit area. Where the **Strain** is defined as the deformation per unit length. All these **simple stress and strain** are briefly described below.





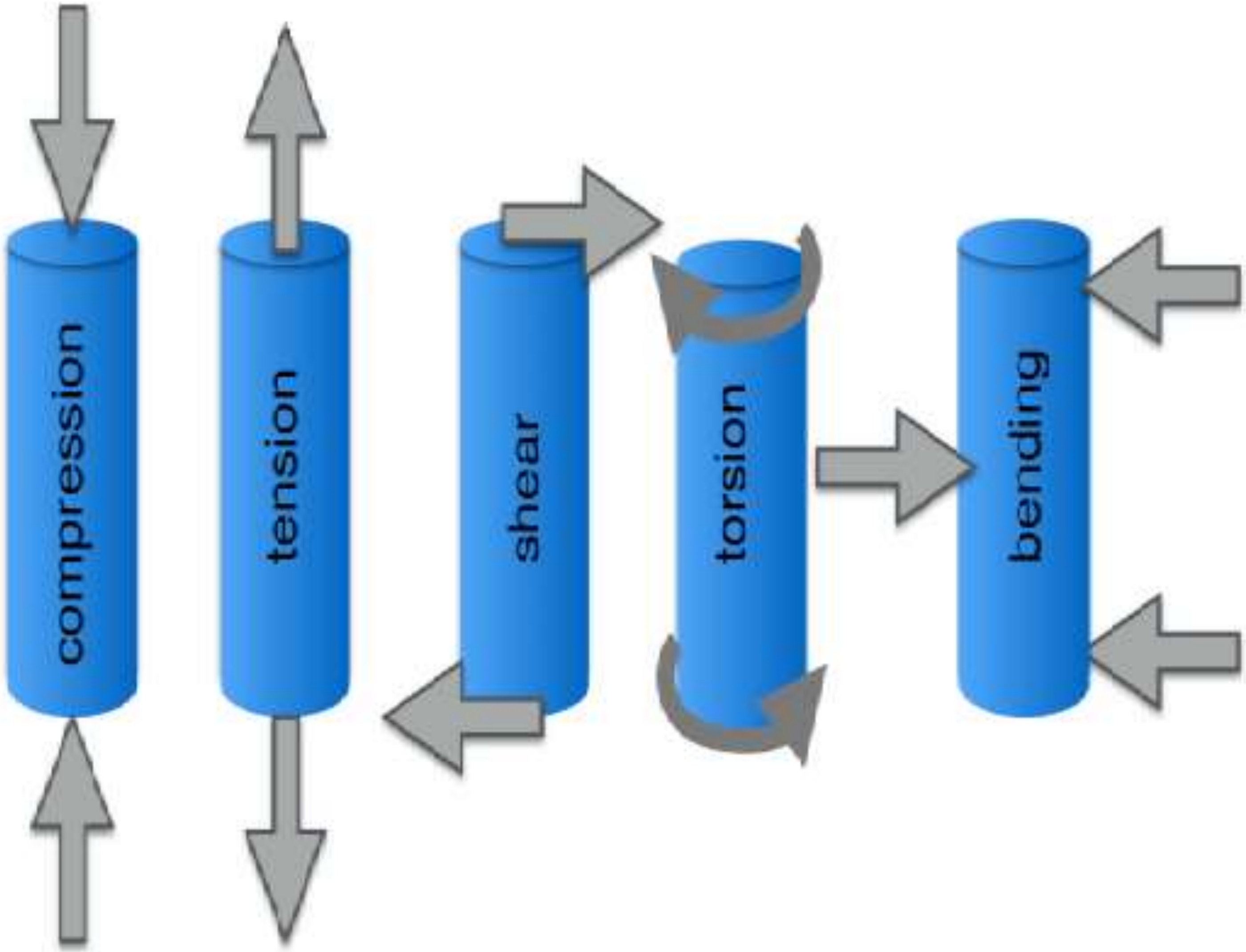
Types of stresses:



LECTURE 6

Torsional and Bending Stresses

Types of Stresses:



$$\tau = \frac{F}{A}$$

$\tau \longrightarrow$ Shear Stress

$F \longrightarrow$ force applied

$A \longrightarrow$ cross sectional area of material

$$\sigma_b = \frac{My}{I}$$

σ_b – Bending stress

M – Calculated bending moment

y – Vertical distance away from the neutral axis

I – Moment of inertia around the neutral axis

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

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LECTURE 7

Various Theories of Failure

Various Theories of Failure:

Predicting the failure stresses for members subjected to bi-axial or tri-axial stresses is much more complicated. In fact, the problem is so complicated that a large number of different theories have been formulated.The principal theories of failure for a member subjected to bi-axial stress are as follows:

- Maximum Principal Stress theory(RANKINE'S THEORY)
- Maximum Shear Stress theory(GUEST AND TRESCA'S THEORY)
- Maximum Principal Strain theory(St.VENANT'S THEORY)
- Total Starain Energy theory(HAIGH'S THEORY)
- Maximum Distorsion Energy theory(VONMISES AND HENCKY'S THEORY)

Maximum Principal Stress theory(RANKINE'S 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 simple tension test.

Mathematically

$\sigma_{t1} = \sigma_{yt}/F.S$ for ductile material

$= \sigma_u/F.S$ for brittle material

Where,

σ_{yt} = Yield point stress in tension as determined from simple tension test, and

σ_u = Ultimate stress.

Maximum Shear Stress theory(GUEST AND TRESCA'S 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

Conditionfor safedesign,

Maximum shear stress induced at a critical tensile point under triaxial combined stress

≤ Permissible shear stress (Tper)

$$\text{Absolute } \tau_{max} \leq \frac{(S_{ys})_{T.T}}{N} \quad \text{or} \quad \frac{S_{yt}}{2N}$$

2N

Maximum Principal Strain theory(St.VENANT'S 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 tension test.

$$\epsilon_{max}= (\sigma_{t1}/E)-(\sigma_{t2}/m*E)$$

According to the above theory,

$$\epsilon_{max}= (\sigma_{t1}/E)-(\sigma_{t2}/m.E)= \epsilon= (\sigma_{yt}/E*F.O.S)...(1)$$

Where,

σ_{t1} and σ_{t2} = Maximum and minimum principal stresses in a bi-axial stress system,

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

$1/m$ = Poisson's ratio,

E = Young's modulus, and

F.O.S. = Factor of safety.

From equation (*i*), we may write that

$$\sigma_{t1}-(\sigma_{t2}/m)= (\sigma_{yt}/F.O.S)$$

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

Maximum Starain Energy theory(HAIGH'S 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,

$$U_1=1/2E[\sigma_{t1}^2 + \sigma_{t2}^2 – ((2\sigma_{t1}* \sigma_{t2})/M)]$$

$$U_2=1/2E[\sigma_{yt}/F.O.S]^2$$

According to the above theory $U_1=U_2$

$$1/2E[\sigma_{t1}^2 + \sigma_{t2}^2 – ((2\sigma_{t1}* \sigma_{t2})/M)] =1/2E[\sigma_{yt}/F.O.S]^2$$

$$\text{Or } [\sigma_{t1}^2 + \sigma_{t2}^2 – ((2\sigma_{t1}* \sigma_{t2})/M)] = [\sigma_{yt}/F.O.S]^2$$

This theory may be used for ductile materials.

Maximum Distorsion Energy theory(VONMISES AND HENCKY'S THEORY)

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

$$(\sigma_{t1})^2 + (\sigma_{t2})^2 - 2\sigma_{t1} \times \sigma_{t2} = (\sigma_{yt} / F.O.S)^2$$

Where,
σ_{yt} is yield stress
F.O.S. = Factor of safety.

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

Problem:

A shaft is transmitting 100 kW at 160 r.p.m. Find a suitable diameter for the shaft, if the maximum torque transmitted exceeds the mean by 25%. Take maximum allowable shear stress as 70 MPa.

Solution. Given : $P = 100$ kW $= 100 \times 10^3$ W ; $N = 160$ r.p.m ; $T_{max} = 1.25\ T_{mean}$; $\tau = 70$ MPa $= 70$ N/mm^2

Let T_{mean} = Mean torque transmitted by the shaft in N-m, and

d = Diameter of the shaft in mm.

We know that the power transmitted (P),

$$100 \times 10^3 = \frac{2\ \pi\ N\ .\ T_{mean}}{60} = \frac{2\pi \times 160 \times T_{mean}}{60} = 16.76\ T_{mean}$$

$$\therefore \quad T_{mean} = 100 \times 10^3/16.76 = 5966.6 \text{ N-m}$$

and maximum torque transmitted,

$$T_{max} = 1.25 \times 5966.6 = 7458 \text{ N-m} = 7458 \times 10^3 \text{ N-mm}$$

We know that maximum torque (T_{max}),

$$7458 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 70 \times d^3 = 13.75\ d^3$$

$$\therefore \quad d^3 = 7458 \times 10^3/13.75 = 542.4 \times 10^3 \text{ or } d = 81.5 \text{ mm}$$ **Ans.**

Problem 2: A beam of uniform rectangular cross-section is fixed at one end and carries an electric motor weighing 400 N at a distance of 300 mm from the fixed end. The maximum bending stress in the beam is 40 MPa. Find the width and depth of the beam, if depth is twice that of width.

Solution. Given: $W = 400$ N ; $L = 300$ mm ;
$\sigma_b = 40$ MPa $= 40$ N/mm^2 ; $h = 2b$

The beam is shown in Fig. 5.7.

Let b = Width of the beam in mm, and

h = Depth of the beam in mm.

$\therefore$ Section modulus,

$$Z = \frac{b \cdot h^2}{6} = \frac{b\,(2b)^2}{6} = \frac{2\,b^3}{3} \text{ mm}^3$$

Maximum bending moment (at the fixed end),

$$M = W.L = 400 \times 300 = 120 \times 10^3 \text{ N-mm}$$

We know that bending stress (σ_b),

$$40 = \frac{M}{Z} = \frac{120 \times 10^3 \times 3}{2\,b^3} = \frac{180 \times 10^3}{b^3}$$

$\therefore$ $b^3 = 180 \times 10^3/40 = 4.5 \times 10^3$ or $b = 16.5$ mm **Ans.**

and $h = 2b = 2 \times 16.5 = 33$ mm **Ans.**

Revision Questions

1. Explain Importance of Failure Theory
2. What is maximum Principal stress theory
3. What is shear stress theory
4. What is maximum shear strain Theory
5. What is maximum Distorsion energy theory
6. What is maximum principal starin theory

LECTURE 8

Factor of Safety

The factor of safety is defined as the **ratio of ultimate to working stress**

Mathematically,

$$\text{Factor of safety} = \frac{\text{Maximum stress}}{\text{Working stress or design stress}}$$

For Brittle : It is the ratio of the Ultimate stress to the working or design stress

$$\text{Factor of safety} = \frac{\text{Ultimate stress}}{\text{Working stress or design stress}}$$

For Ductile: It is the ratio of the Yield stress to the working stress

$$\text{Factor of safety} = \frac{\text{Yield stress}}{\text{Working stress or design stress}}$$

Revision Questions

1. What is Factor of safety
2. Importance of Factor of safety

THANK YOU

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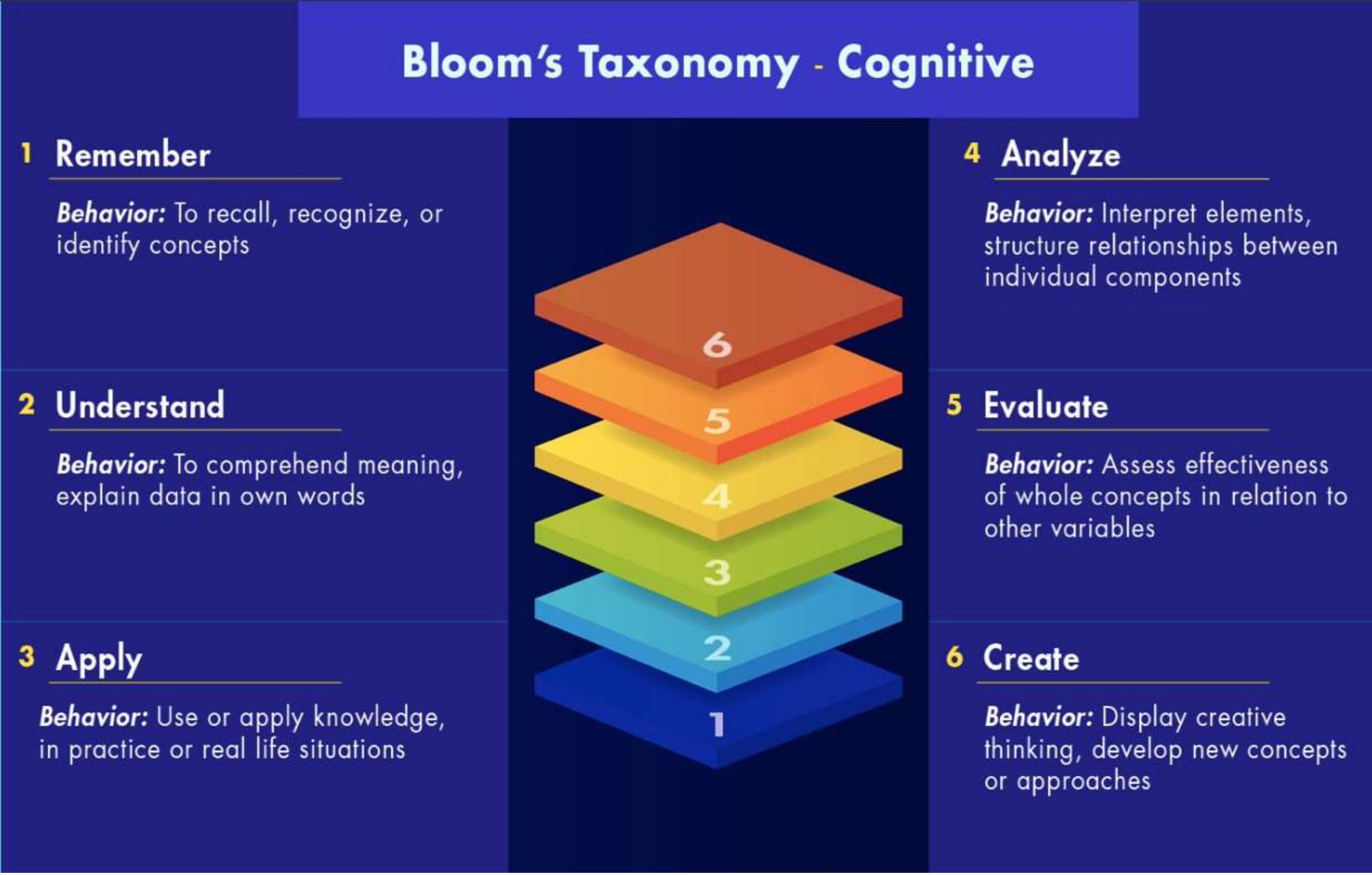


(Source: TASK Training material)

UNIT 2

CO2: To understand Stress concentration and the factors responsible. Determination of stress concentration factor; experimental and theoretical methods. Fatigue strength reduction factor and notch sensitivity factor

UNIT – II (SYLLABUS)

Introduction to Fatigue

- Stress Concentration ,Theoretical and Fatigue stress concentration Factor
- Design for fluctuating stresses ,Repeated and Reversed stress tor
- Fatigue Failure and Endurance Limit & Endurance limit estimation

Fatigue Failure Under Variable Loading

- Low cycle and High cycle fatigue finite & infinite life
- Goodman line, Soderberg line , Gerber line and modified Goodman line

COURSE OUTLINE

UNIT -2

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
1	**Introduction-Fatigue**	**Definition of Fatigue.**	• **Understand the definition of fatigue load(B2)**
2	Stress Concentration ,Theoretical and Fatigue stress concentration Factor		• Understand why stress concentration factor (B2) • Understand how to use and when to use stress concentration factor and difference between Theoretical & Fatigue Factor (
3	Design for fluctuating stresses ,Repeated and Reversed stress		• Students able to understand difference between Reversed and repeated stress(B2) • Understand when different types of stresses acting an object weather the object with standing or not (B3)

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
4	Fatigue Failure and Endurance Limit	Fatigue Failure	• understanding the difference between static and fatigue failure (B2) • Understand how many cycles it can with stand before failure (B3)
5	**Endurance limit estimation**		• **Student able to understand difference between standard and actual specimen life (B4)** • **Students able to understand what are the factors will effected on actual specimen life (B2)**
6	Low cycle and High cycle fatigue finite & infinite life		• Student able understand low and high cycle fatigue(B2) • Understanding difference between finite and infinite life (B2)

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
7	Goodman line, Soderberg line , Gerber line and modified Goodman line		•understanding the importance of all lines (B2) •Understand the difference between all lines and use these lines for calculating stress (B3) •Understand why the Goodman line modified and assess effectiveness(B4)

LECTURE 1

Introduction-Fatigue

UNIT II
DESIGN FOR FATIGUE STRENGTH

Introduction: A condition characterized by a lessened capacity for work and reduced efficiency of accomplishment

LECTURE 2

Stress Concentration

STRESS CONCENTRATION

STRESS CONCENTRATION:

Stress concentrations occur when there are irregularities in the geometry or material of a structural component that **cause** an interruption to the flow of **stress**. This arises from such details as holes, grooves, notches and fillets. **Stress concentrations** may also occur from accidental damage such as nicks and scratches.

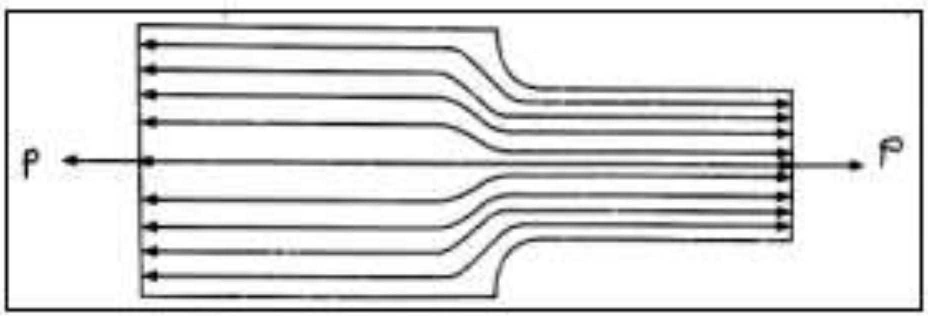


Fig. Stress concentration

THERETICAL STRESS CONCENTRATION FACTOR:

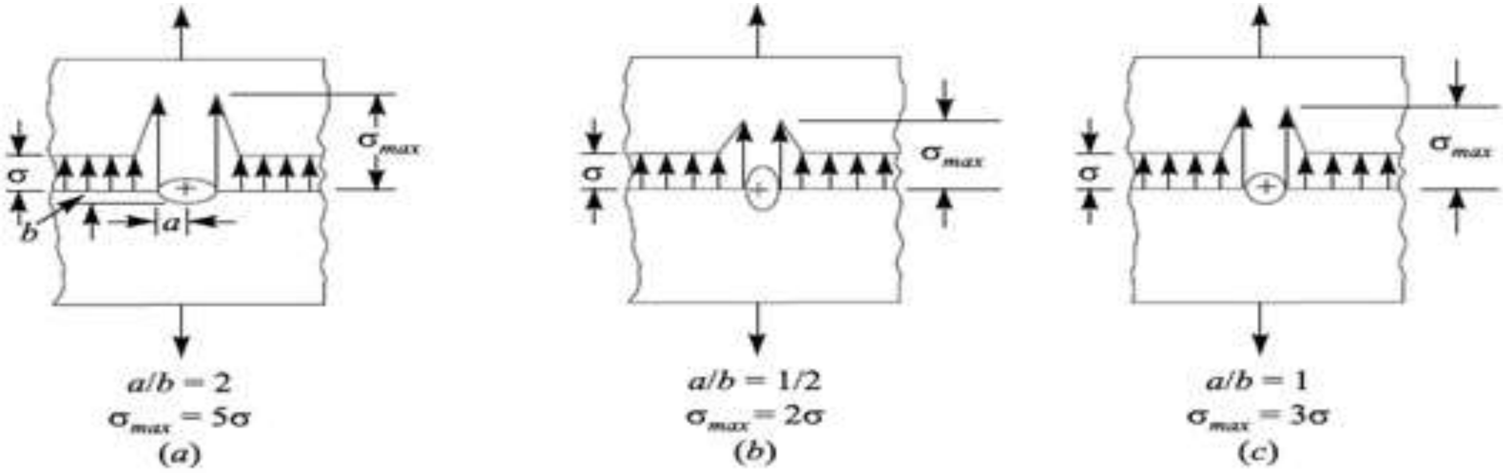
A stress concentration factor (Kt) is a dimensionless factor that is used to quantify how concentrated the stress is in a material. It is defined as the ratio of the highest stress in the element to the reference stress.

$$K = \frac{\sigma_{\max}}{\sigma_{ref}}$$

Reference stress is the total stress within an element under the same loading conditions without the stress concentrators, meaning the total stress on the material where the material is free from holes, cuts, shoulders or narrow passes.

STRESS CONCENTRATION DUE TO HOLES AND NOTCHES:

When inner forces go around **holes** or **notches**, they will concentrate near such “obstacles.” **Stress** concentrators are areas that tend to magnify the **stress** level within a part. **Stress** that is higher in one area than it is in surrounding regions can cause the part to fail.

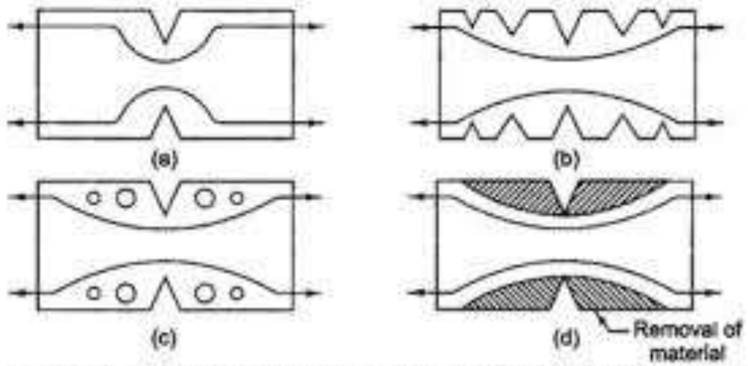


METHODS OF REDUCING STRESS CONCENTRATION:

The presence of stress concentration can not be totally eliminated but it may reduced to some extent

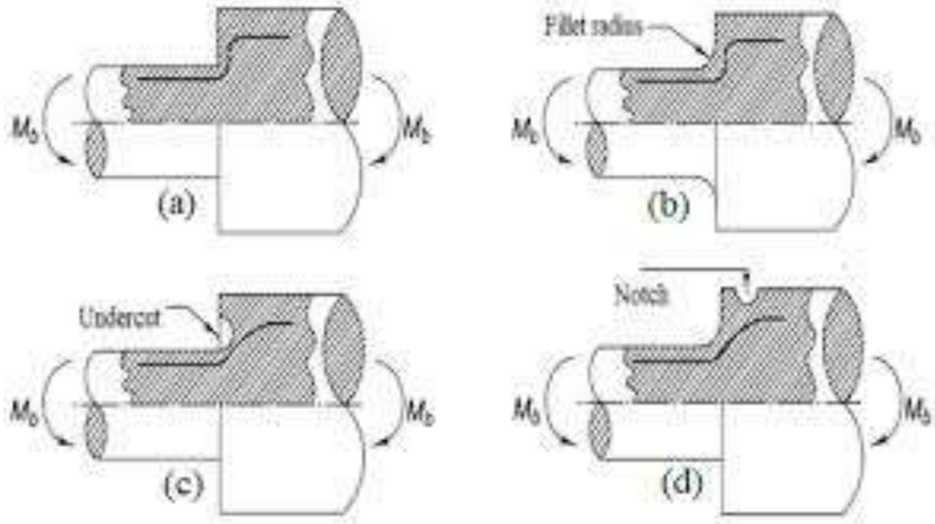
- Additional Notches and Holes in Tension member
- Fillet Radius ,Under Cutting and Notch for member in Bending
- Drilling Additional Holes for Shaft
- Reduction of Stress Concentration in Threaded members

Additional Notches and Holes in Tension member:

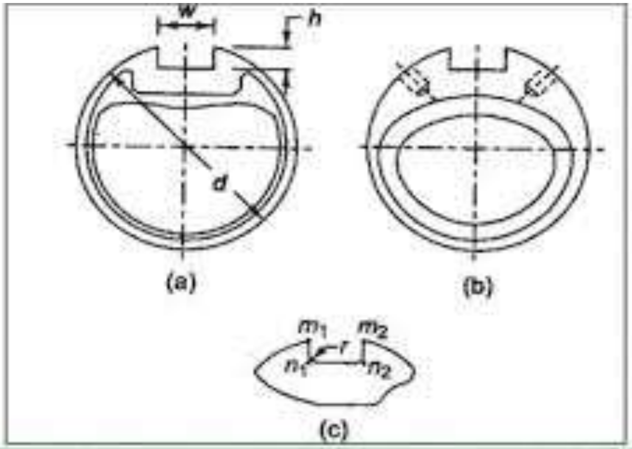


Reduction of Stress Concentration due to V-notch: (a) Original Notch (b) Multiple Notches(c) Drilled Holes (d) Removal of Undesirable Material

Fillet Radius ,Under Cutting and Notch for member in Bending:

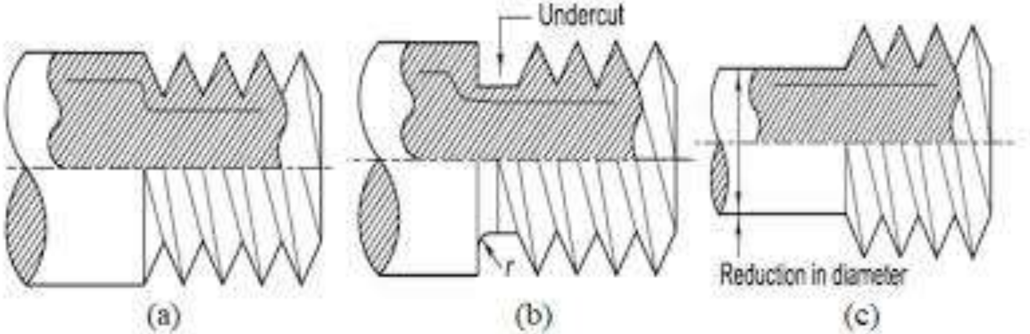


Drilling Additional Holes for Shaft:



Reduction of Stress Concentration in Shaft with Keyway: (a) Original Shaft (b) Drilled Holes (c) Fillet Radius

Reduction of Stress Concentration in Threaded members:



Revision Questions

1. What is Fatigue
2. What is stress concentration
3. What are reasons for stress concentration
4. How to reduce stress concentration and explain

LECTURE 3

Design For Fluctuating Stresses

<u>Fluctuating Stresses</u>:

The components are subjected to forces, which are not static, but vary in magnitude with respect to time. The stresses induced due to such forces are called Fluctuating Stresses.

The most popular model for stress–time relationship is the sine curve.

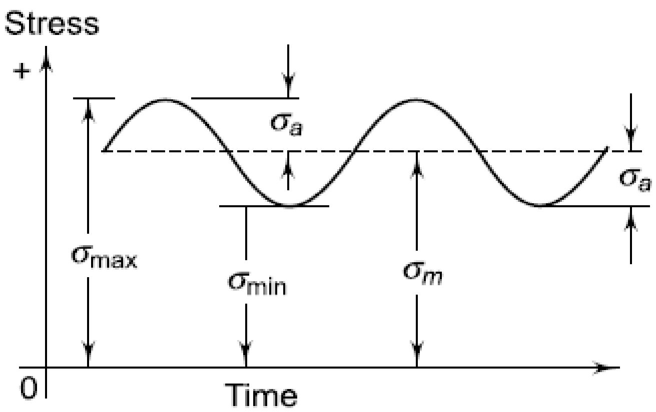
There are three types of mathematical models for cyclic stresses:

(a) Fluctuating or alternating stresses,

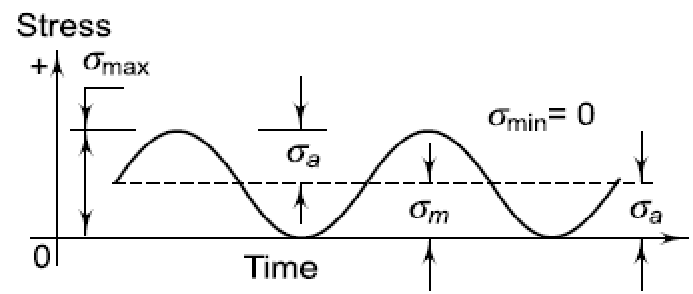
(b) Repeated stresses and

(c) Reversed stresses.

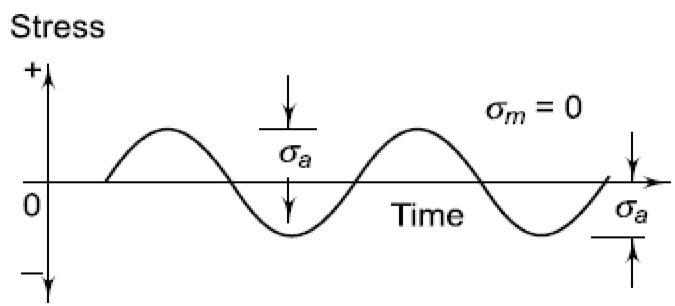
Stress–time relationships for these models are illustrated in Fig.



(a) Fluctuating stresses



(b) Repeated stresses



(c) Reversed stresses

Stress-Time Relations

Mean stress $\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$

Stress Amplitude $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$

Stress Ratio $R = \frac{\sigma_{min}}{\sigma_{max}}$

Amplitude Ratio $A = \frac{\sigma_a}{\sigma_m}$

(a) Fluctuating or alternating stresses (Ref Fig a):

The fluctuating or alternating stress varies in a sinusoidal manner with respect to time. It has some mean value as well as amplitude value.

It fluctuates between two limits—maximum and minimum stress.

The stress can be tensile or compressive or partly tensile and partly compressive.

(b) Repeated stresses (Ref Fig b):

The repeated stress varies in a sinusoidal manner with respect to time, but the variation is from zero to some maximum value. The minimum stress is zero in this case and therefore, amplitude stress and mean stress are equal.

(c) Reversed stresses (Ref Fig c):

The reversed stress varies in a sinusoidal manner with respect to time, but it has zero mean stress.

Revision Questions

1. What is fluctuating stress
2. What is reversed stress
3. What is repeated stress

LECTURE 4

Design for Fatigue Failure

Design for fatigue stress

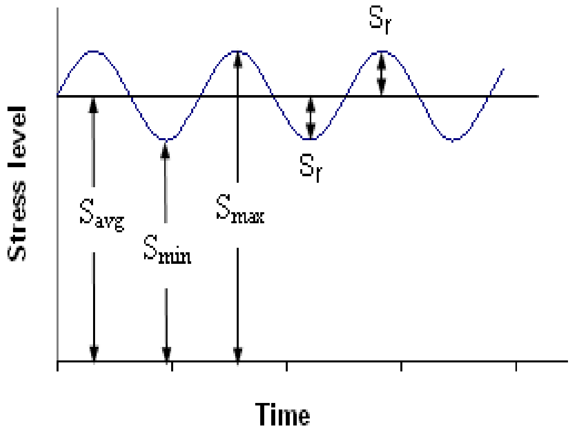
Stress-Time Relations

Mean stress $\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$

Stress Amplitude $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$

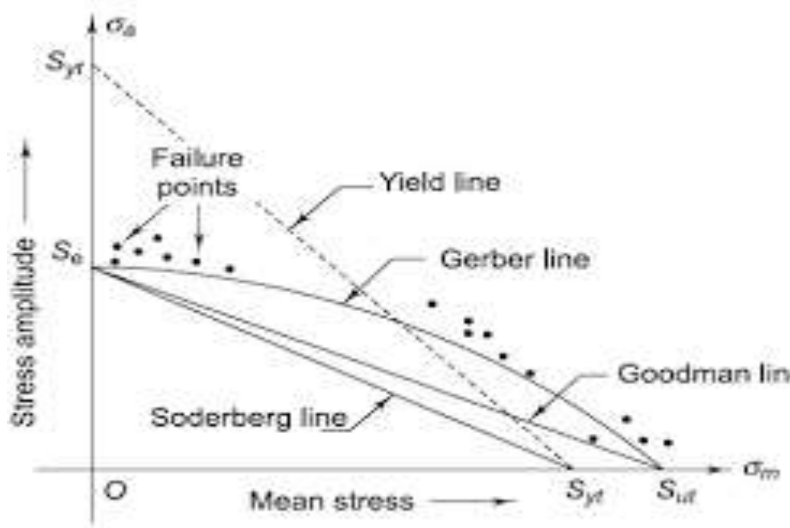
Stress Ratio $R = \frac{\sigma_{min}}{\sigma_{max}}$

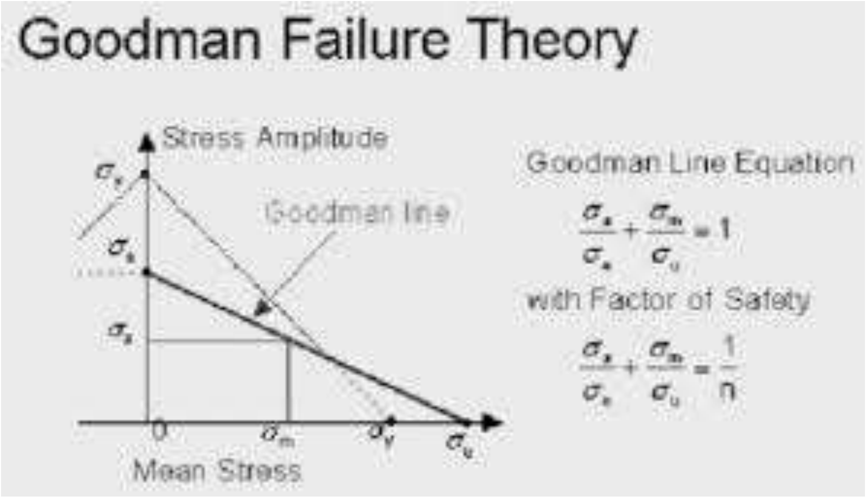
Amplitude Ratio $A = \frac{\sigma_a}{\sigma_m}$



Combined cyclic and steady stress

Goodman, Soderberg & Gerber Line:





Problem: Determine the thickness of a 120 mm wide uniform plate for safe continuous operation if the plate is to be subjected to a tensile load that has a maximum value of 250 kN and a minimum value of 100 kN. The properties of the plate material are as follows:Endurance limit stress = 225 MPa, and Yield point stress = 300 MPa. The factor of safety based on yield point may be taken as 1.5.

Let t = Thickness of the plate in mm.

$\therefore$ Area, $A = b \times t = 120\,t\ \text{mm}^2$

We know that mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{250 + 100}{2} = 175\ \text{kN} = 175 \times 10^3\ \text{N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{175 \times 10^3}{120t}\ \text{N/mm}^2$$

$$\text{Variable load, } W_v = \frac{W_{max} - W_{min}}{2} = \frac{250 - 100}{2} = 75\ \text{kN} = 75 \times 10^3\ \text{N}$$

$$\therefore \quad \text{Variable stress, } \sigma_v = \frac{W_v}{A} = \frac{75 \times 10^3}{120t}\ \text{N/mm}^2$$

According to Soderberg's formula,

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

$$\frac{1}{1.5} = \frac{175 \times 10^3}{120t \times 300} + \frac{75 \times 10^3}{120t \times 225} = \frac{4.86}{t} + \frac{2.78}{t} = \frac{7.64}{t}$$

$\therefore \quad t = 7.64 \times 1.5 = 11.46$ say 11.5 mm **Ans.**

Problem:Determine the diameter of a circular rod made of ductile material with a fatigue strength(complete stress reversal), σ_e = 265 MPa and a tensile yield strength of 350 MPa. Themember is subjected to a varying axial load from $W_{min} = -300 \times 10^3$ N to $W_{max} = 700 \times 10^3$ N and has a stress concentration factor = 1.8. Use factor of safety as 2.0.

Let d = Diameter of the circular rod in mm.

$$\therefore \quad \text{Area, } A = \frac{\pi}{4} \times d^2 = 0.7854\, d^2 \text{ mm}^2$$

We know that the mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{700 \times 10^3 + (-300 \times 10^3)}{2} = 200 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{200 \times 10^3}{0.7854\, d^2} = \frac{254.6 \times 10^3}{d^2} \text{ N/mm}^2$$

Variable load, $W_v = \dfrac{W_{max} - W_{min}}{2} = \dfrac{700 \times 10^3 - (-300 \times 10^3)}{2} = 500 \times 10^3$ N

$\therefore$ Variable stress, $\sigma_v = \dfrac{W_v}{A} = \dfrac{500 \times 10^3}{0.7854\, d^2} = \dfrac{636.5 \times 10^3}{d^2}$ N/mm^2

We know that according to Soderberg's formula,

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

$$\frac{1}{2} = \frac{254.6 \times 10^3}{d^2 \times 350} + \frac{636.5 \times 10^3 \times 1.8}{d^2 \times 265} = \frac{727}{d^2} + \frac{4323}{d^2} = \frac{5050}{d^2}$$

$\therefore$ $d^2 = 5050 \times 2 = 10\,100$ or $d = 100.5$ mm **Ans.**

Problem: A circular bar of 500 mm length is supported freely at its two ends. It is acted upon by a central concentrated cyclic load having a minimum value of 20 kN and a maximum value of 50 kN. Determine the diameter of bar by taking a factor of safety of 1.5, size effect of 0.85, surface finish factor of 0.9. The material properties of bar are given by: ultimate strength of 650 MPa, yield strength of 500 MPa and endurance strength of 350 MPa.

Solution. Given : $l = 500$ mm ; $W_{min} = 20$ kN $= 20 \times 10^3$ N ; $W_{max} = 50$ kN $= 50 \times 10^3$ N ; $F.S. = 1.5$; $K_{sz} = 0.85$; $K_{sur} = 0.9$; $\sigma_u = 650$ MPa $= 650$ N/mm^2 ; $\sigma_y = 500$ MPa $= 500$ N/mm^2 ; $\sigma_e = 350$ MPa $= 350$ N/mm^2

Let $\quad d$ = Diameter of the bar in mm.

We know that the maximum bending moment,

$$M_{max} = \frac{W_{max} \times l}{4} = \frac{50 \times 10^3 \times 500}{4} = 6250 \times 10^3 \text{ N-mm}$$

and minimum bending moment,

$$M_{min} = \frac{W_{min} \times l}{4} = \frac{20 \times 10^3 \times 500}{4} = 2550 \times 10^3 \text{ N-mm}$$

$\therefore$ Mean or average bending moment,

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{6250 \times 10^3 + 2500 \times 10^3}{2} = 4375 \times 10^3 \text{ N-mm}$$

and variable bending moment,

$$M_v = \frac{M_{max} - M_{min}}{2} = \frac{6250 \times 10^3 - 2500 \times 10^3}{2} = 1875 \times 10^3 \text{ N-mm}$$

Section modulus of the bar,

$$Z = \frac{\pi}{32} \times d^3 = 0.0982\, d^3 \text{ mm}^3$$

$\therefore$ Mean or average bending stress,

$$\sigma_m = \frac{M_m}{Z} = \frac{4375 \times 10^3}{0.0982\, d^3} = \frac{44.5 \times 10^6}{d^3} \text{ N/mm}^2$$

and variable bending stress,

$$\sigma_v = \frac{M_v}{Z} = \frac{1875 \times 10^3}{0.0982\, d^3} = \frac{19.1 \times 10^6}{d^3} \text{ N/mm}^2$$

We know that according to Goodman's formula,

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

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 650} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \qquad \text{...(Taking } K_f = 1)$$

$$= \frac{68 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{139 \times 10^3}{d^3}$$

$\therefore$ $\quad d^3 = 139 \times 10^3 \times 1.5 = 209 \times 10^3$ or $d = 59.3$ mm

and according to Soderberg's formula,

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

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 500} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \qquad \text{...(Taking } K_f = 1)$$

$$= \frac{89 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{160 \times 10^3}{d^3}$$

$\therefore$ $\quad d^3 = 160 \times 10^3 \times 1.5 = 240 \times 10^3$ or $d = 62.1$ mm

Taking larger of the two values, we have $d = 62.1$ mm **Ans.**

THANK YOU

DIGITAL CLASSROOM ENVIRONMENT

A NEW PEDAGOGY IS EMERGING

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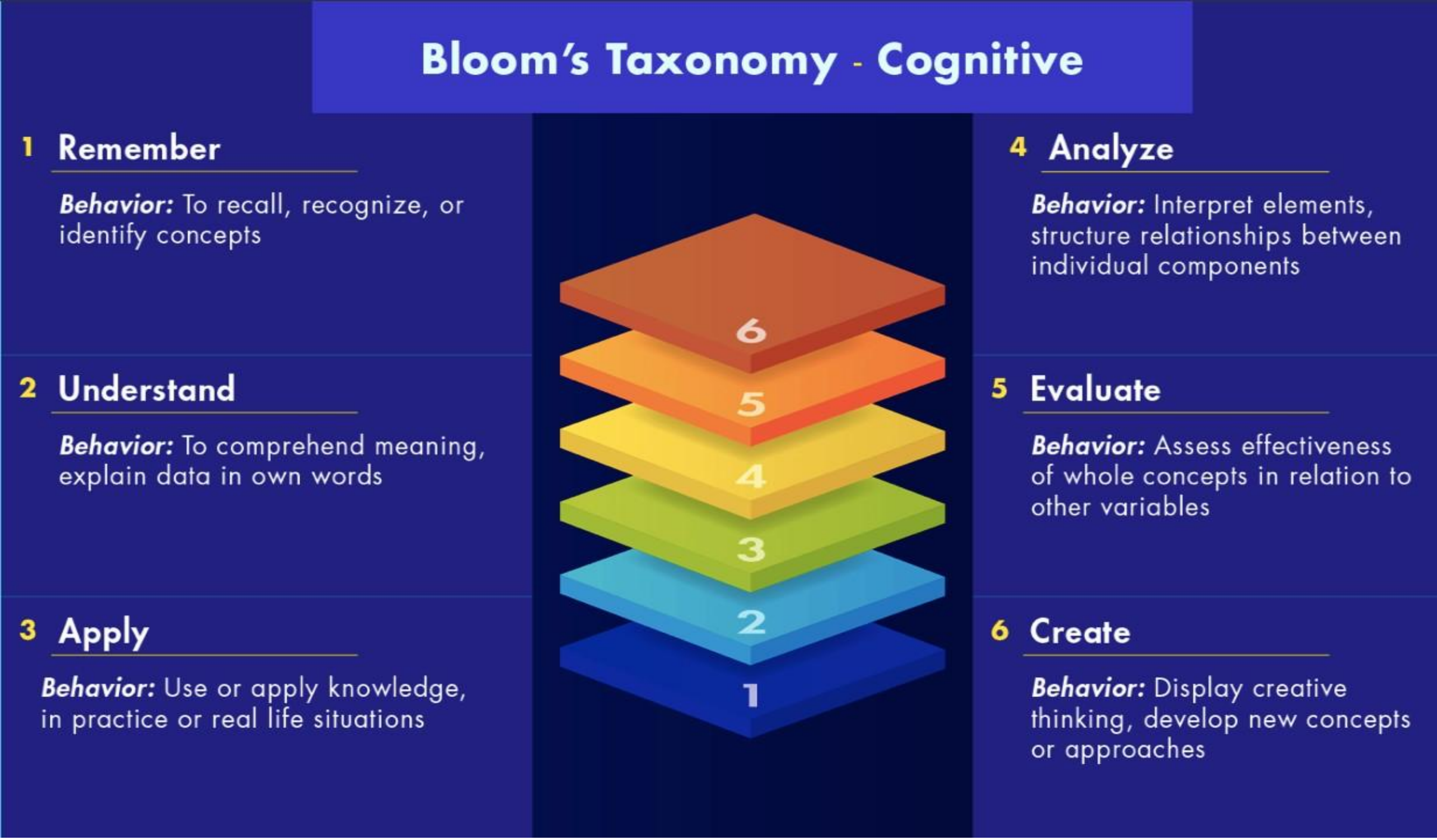
C. Daksheeswara Reddy
Assistant Professor,
Mechanical Engineering



(Source: TASK Training material)

UNIT 3

CO3: To develop the Knowledge on Basic failure mechanisms of riveted joints. Concepts of design of a riveted joint, welded joints and Bolted Joints to determine the forces in welds and riveted joints and formulate design solution for size of weld and size of rivet

UNIT – 3 (SYLLABUS)

Riveted Joints

- Introduction to joints and Types of joints
- Methods of Riveting and Failure
- Strength equations

Welded Joints

- Introduction to welded oints and Types of welded joints
- Design of fillet welds

- Design of axial and Eccentrically loaded welded joints both symmetrical and unsymmetrical sections

Bolted Joints

- Introduction to Bolted joints and Types of joints
- Bolt uniform strength
- Design of bolts under axial and Eccentrical loading conditions

COURSE OUTLINE

UNIT - 3

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
1	**Introduction to joints and Types of joints**	**Joints.**	• **Understand the importance of Joints (B2)** • **Understand difference b/w Temporary fasteners and permanent fasteners.(B2)**
2	Methods of Riveting and Failure		• Understand different types riveting methods(B2) • Understand the modes of Failures and Remedies(B2) • Understand importance of Riveted joints(B2)
3	Strength equations		• understand design procedure and calculating efficiency of Riveted joints(B3 &B4)

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
4	**Introduction to welded joints and Types of welded joints**	**Joints.**	• **Understand the importance of welded Joints (B2).**
5	Design of fillet welds		• Understand the difference between Transverse and Parallel filler welds.(B3)
6	Design of axial and Eccentrically loaded welded joints both symmetrical and unsymmetrical sections	section	• understand design procedure for both sections and calculating strength of joints (B4&B5)
7	Introduction to Bolted joints and Types of joints	Joints.	• Understand the importance of bolted Joints (B2).

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
8	**Bolt uniform strength**		• **Understand how to maintain the bolt in uniform manner and how to reduce stress concentration on bolt(B3)**
9	Design of bolts under axial and Eccentrical loading conditions	load	• Design of bolts under axial and Eccentrical loading conditions(B3&B4)

LECTURE 1

Introduction to joints and Types of joints

The student will be able to:

- Understand the procedure of design (w.r.t. Basic failures) under various loading condition.
- Understand the applications of joints in various loading condition.

RIVETED JOINTS

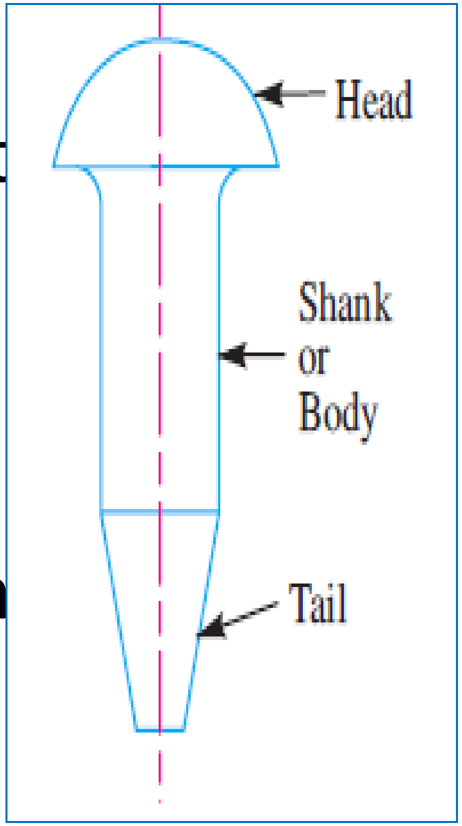
Introduction :

- The function of rivets in a joint is to make a connection that has strength & tightness.
- Strength is necessary to prevent failure of the joint while tightness is necessary in order to contribute to strength & to prevent leakage.
- Rivets may be driven by hand or by riveting machines.
- Rivets are commonly used in a) Tanks & pressure vessels b) bridges ,building, cranes and machinery c)Hulls of Ships

- A rivet is a short cylindrical bar with a head integral to it.
- The cylindrical portion of the rivet called ***shank or body*** and lower
- portion of shank is known as ***tail,***
  – ***as shown in Fig.*** 9.1. The fastenings may be classified into the following
- two groups :

  1. Permanent fastenings, and

  2. Temporary or detachable fastenings.



- The ***permanent fastenings are those fastenings which can not be disassembled without*** destroying the connecting components. The examples of permanent fastenings in order of strength are soldered, brazed, welded and riveted joints.
- The ***temporary or detachable fastenings are those fastenings which can be disassembled*** without destroying the connecting components. The examples of temporary fastenings are screwed, keys, cotters, pins and splined joints.

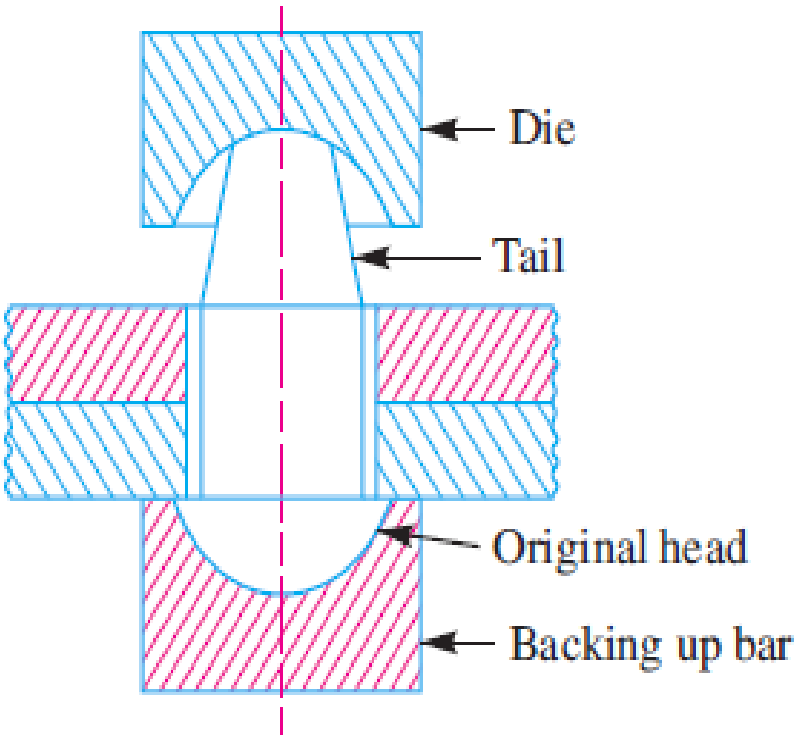
Revision Questions

1. Function of Rivet
2. What are parts of rivet
3. Difference between Temporary fastening and permanent fastening

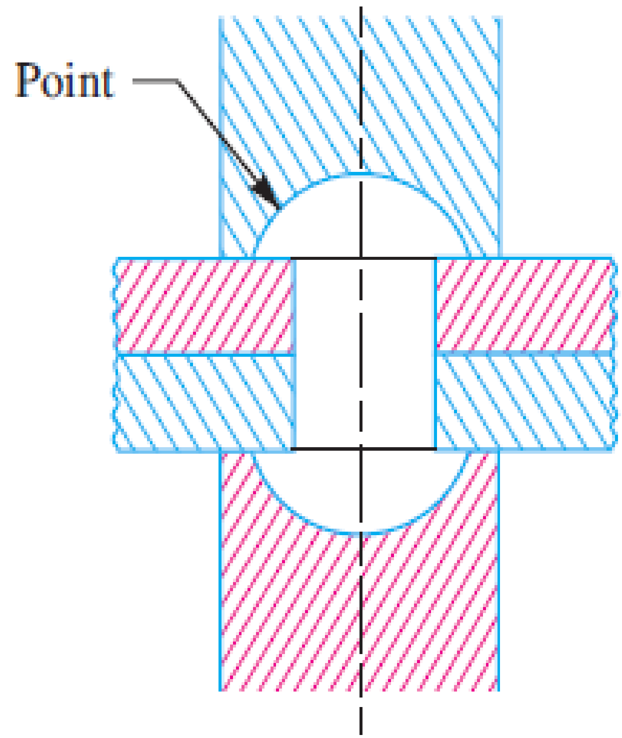
LECTURE 2

Methods of Riveting and Failure

- **Methods of Riveting:**



(*a*) Initial position.



(*b*) Final position.

- **Types of Riveted Joints**
- Following are the two types of riveted joints, depending upon the way in which the plates are connected.
- **1. Lap joint, and 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. The cover plate is then riveted together with the main plates. Butt joints are of the following two types :
- **1. Single strap butt joint, and 2. Double strap butt joint**.

- **Methods of Riveting:**

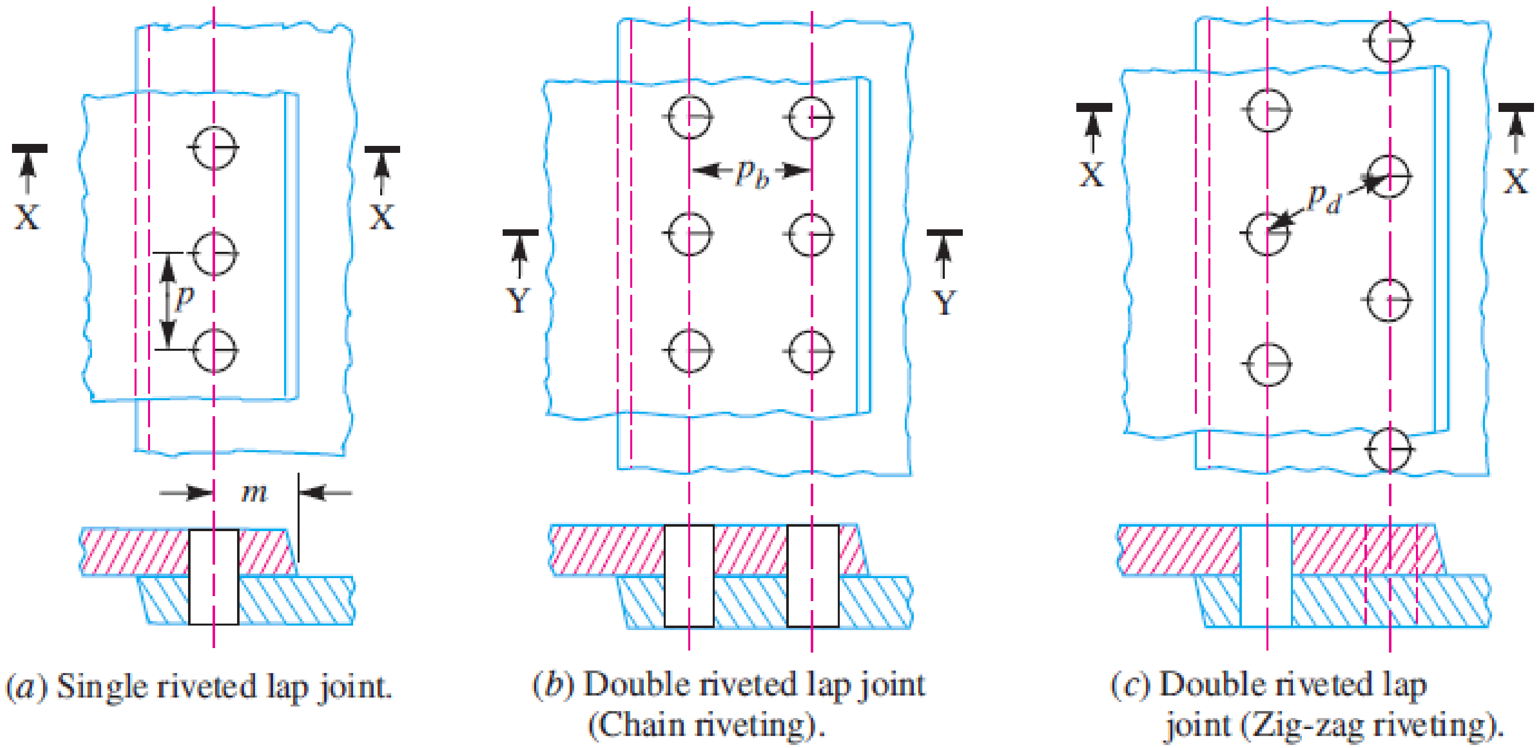
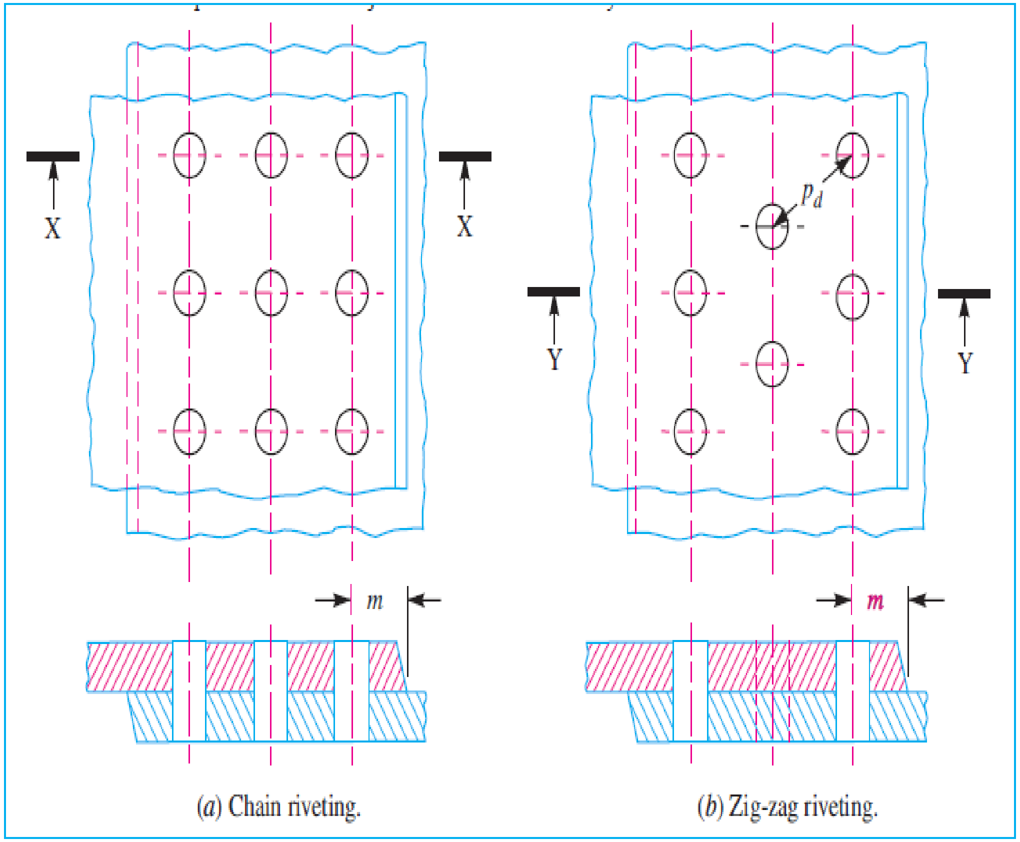


Fig: Single and double riveted lap Joint

- # Methods of Riveting:



(a) Chain riveting.
(b) Zig-zag riveting.

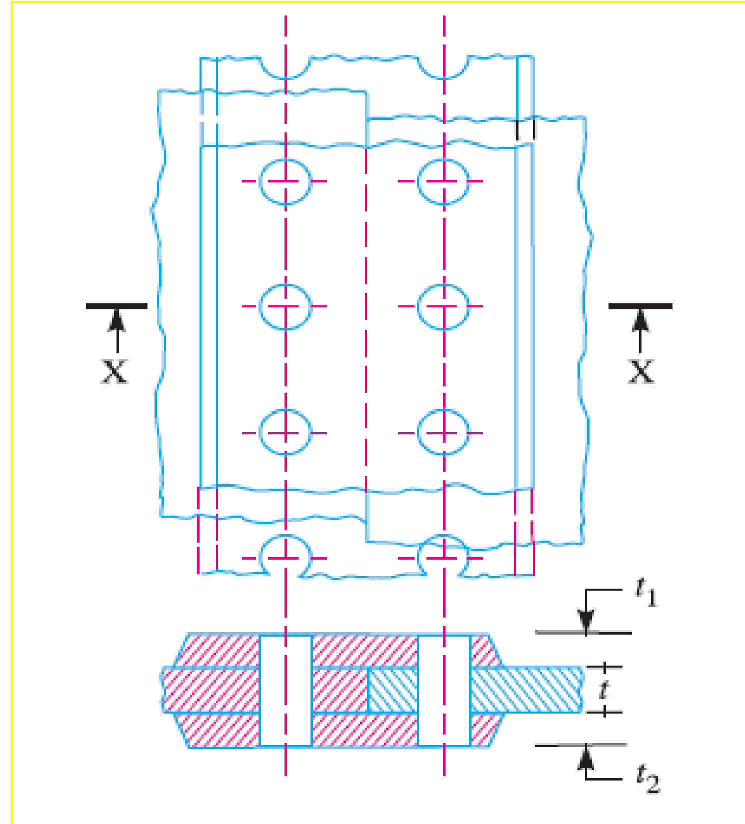


Fig: triple riveted lap Joint

Fig: single riveted double strap butt Joint

- **Methods of Riveting:**

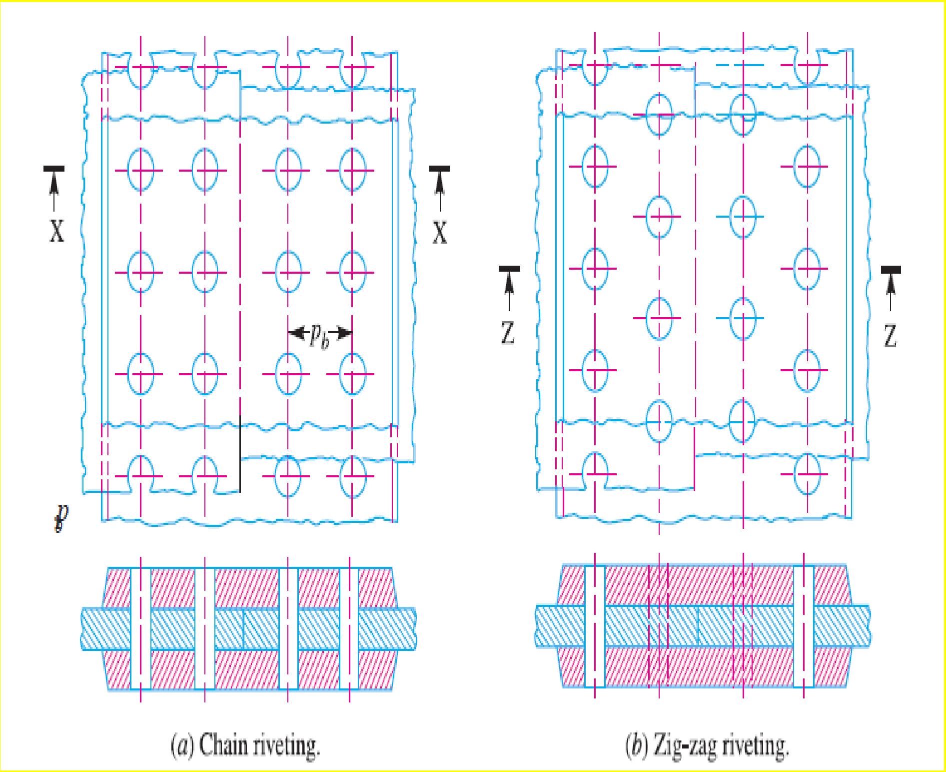


Fig: Double riveted double strap(equal) butt Joint

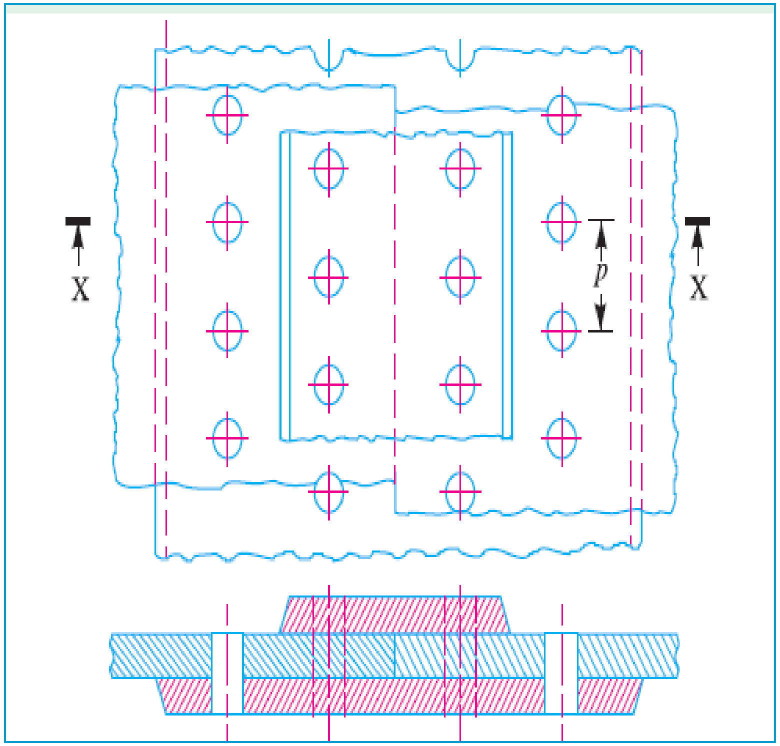


Fig: Double riveted double strap(unequal) butt Joint with zig-zag riveting

- **Important Terms Used in Riveted Joints**

1. ***Pitch***. *It is the distance from the centre of one rivet to the centre of the next rivet measured* parallel to the seam as shown in Fig. 9.6. It is usually denoted by *p.*

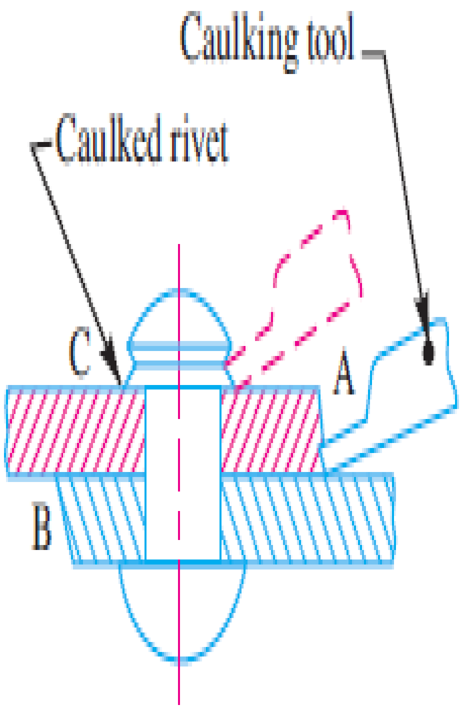
- **2. *Back pitch.*** *It is the perpendicular distance between the centre lines of the successive rows*. It is usually denoted by *pb.*

- **3. *Diagonal pitch.*** *It is the distance between the centres of the rivets in adjacent rows of zig-zag* riveted joint. It is usually denoted by *pd.*

- **4. *Margin or marginal pitch***. *It is the distance between the centre of rivet hole to the nearest* edge of the plate. It is usually denoted by *m.*

- **Caulking**

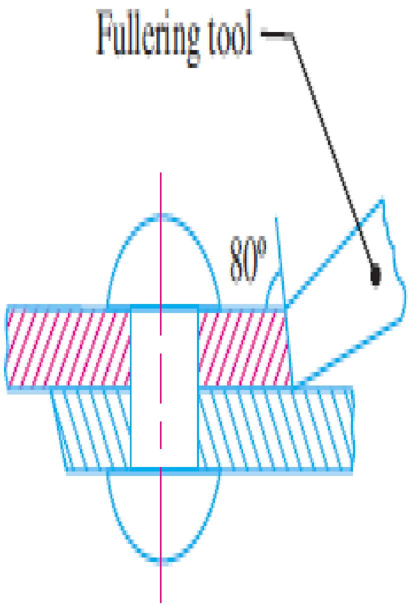
In order to make the joints leak proof or fluid tight in pressure vessels like steam boilers, air receivers and tanks etc. a process known as *caulking is employed. In this* process, a narrow blunt tool called caulking tool, about 5 mm thick and 38 mm in breadth, is used. The edge of the tool is ground to an angle of 80°. The tool is moved after each blow along the edge of the plate, which is planed to a bevel of 75° to 80° to facilitate the forcing down of edge.



(*a*) Caulking.

- **Fullering**

A more satisfactory way of making the joints staunch is known as ***fullering which has largely*** superseded caulking. In this case, a fullering tool with a thickness at the end equal to that of the plate is used in such a way that the greatest pressure due to the blows occur near the joint, giving a clean finish, with less risk of damaging the plate..

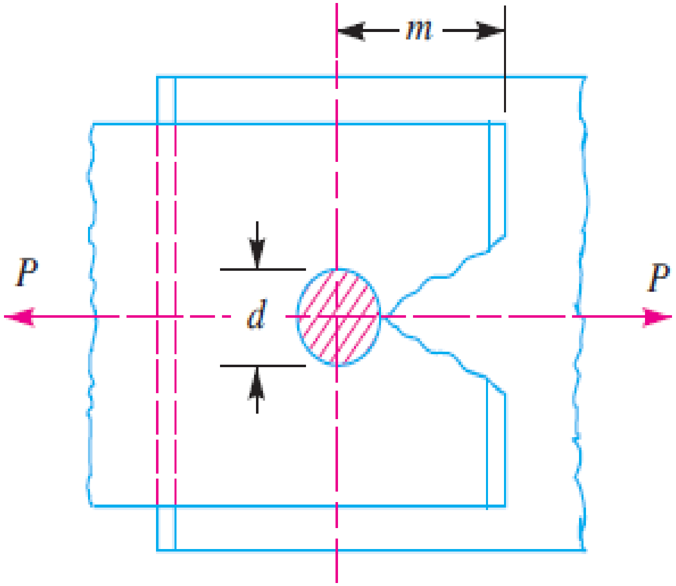


(*b*) Fullering.

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 . This can be avoided by keeping the margin, *m = 1.5d, where d is the diameter of* the rivet



2. ***Tearing of the plate across a row of rivets:*** *Due to the tensile stresses in the main plates, the* main plate or cover plates may tear off across a row of rivets

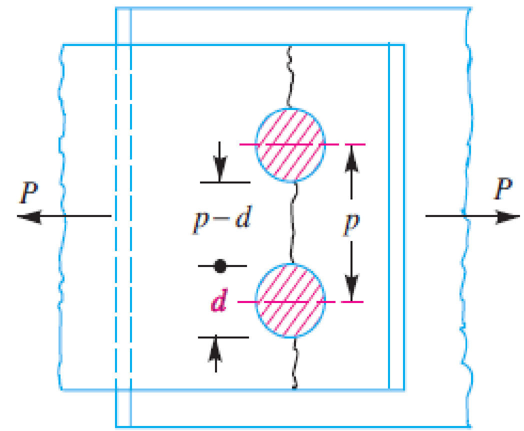
The resistance offered by the plate against tearing is known as ***tearing resistance or tearing strength or tearing value*** *of the plate.*

Let *p = Pitch of the rivets,*

d = Diameter of the rivet hole,

t = Thickness of the plate, and

σt = Permissible tensile stress for the plate material.



We know that tearing area per pitch length,

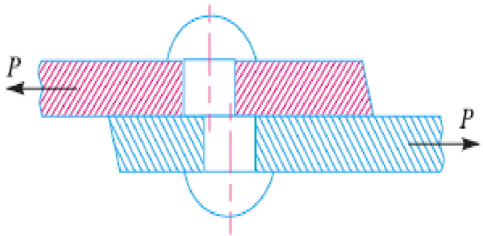
At = (p – d) t

∴ Tearing resistance or pull required to tear off the plate p length,

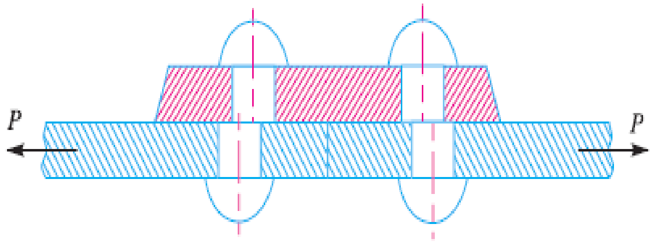
Pt = At.σt = (p – d)t.σt

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

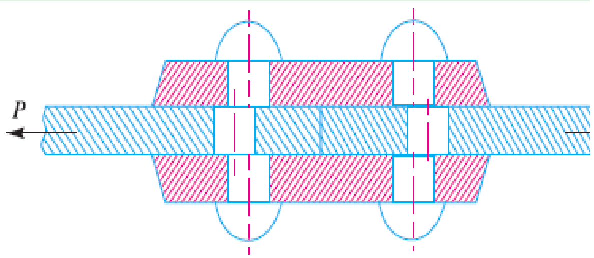
3. *Shearing of the rivets:* *The plates which are connected by the rivets exert tensile stress on* the rivets, and if the rivets are unable to resist the stress, they



(a) Shearing off a rivet in a lap joint.



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



c) Shearing off a rivet in double cover butt joint.

We know that shearing area,

$$A_s = \frac{\pi}{4} \times d^2 \qquad \text{...(In single shear)}$$

$$= 2 \times \frac{\pi}{4} \times d^2 \qquad \text{...(Theoretically, in double shear)}$$

$$= 1.875 \times \frac{\pi}{4} \times d^2 \qquad \text{...(In double shear, according to Indian Boiler Regulations)}$$

∴ Shearing resistance or pull required to shear off the rivet per pitch length,

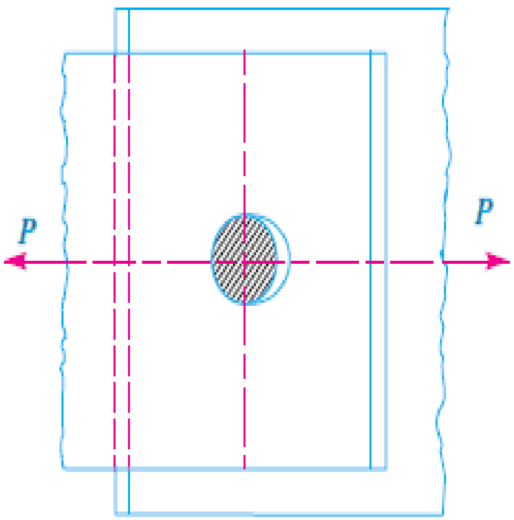
$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau \qquad \text{...(In single shear)}$$

$$= n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau \qquad \text{...(Theoretically, in double shear)}$$

$$= n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau \qquad \text{...(In double shear, according to Indian Boiler Regulations)}$$

When the shearing resistance (P_s) is greater than the applied load (P) per pitch length, then this type of failure will occur.

- **4. *Crushing of the plate or rivets:*** *Sometimes, the rivets do not actually shear off under the* under the tensile stress, but are crushed as shown in Fig
- Let *d = Diameter of the rivet hole,*
- *t = Thickness of the plate,*
- *σc = Safe permissible crushing stress for the*
- *rivet or* plate material, and
- *n = Number of rivets per pitch length under crushi*
- We know that crushing area per rivet
- *Ac = d.t*
- ∴ Total crushing area = *n.d.t*
- and crushing resistance or pull required to crush the rivet
- per pitch length,
- *Pc = n.d.t.σc*
- When the crushing resistance (*Pc) is greater than* the applied load (*P) per pitch length, then this type of* failure will occur.



Revision Questions

1. Explain methods of riveting
2. Explain Type of joints
3. Explain Failures in riveted joints

LECTURE 3

Design of Longitudinal Joints

Longitudinal joint

Design of Longitudinal Butt Joint for a Boiler

1. ***Thickness of boiler shell.:*** *First of all, the thickness of the boiler shell is determined by using* the thin cylindrical formula, *i.e.*

$$t = \frac{P.D}{2\,\sigma_t \times \eta_l} + 1 \text{ mm as corrosion allowance}$$

where *t = Thickness of the boiler shell,*
P = Steam pressure in boiler,
D = Internal diameter of boiler shell,
σ_t *= Permissible tensile stress, and*
η_l *= Efficiency of the longitudinal joint.*
The following points may be noted :
(*a) The thickness of the boiler shell should not be less than 7 mm.*
(*b) The efficiency of the joint may be taken from Data book.*

2. *Diameter of rivets. :* *After finding out the thickness of the boiler shell (t), the diameter of the* rivet hole (*d) may be determined by using Unwin's empirical formula, i.e.*

$$d = 6\sqrt{t}$$ *(when t is greater than 8 mm)*

But if the thickness of plate is less than 8 mm, then the diameter of the rivet hole may be calculated by equating the shearing resistance of the rivets to crushing resistance. In no case, the diameter of rivet hole should not be less than the thickness of the plate, because there will be danger of punch crushing.

3. *Pitch of rivets:*

- Pitch of rivet can be calculated by using formula

$$p = \frac{\pi}{4} d^2 \frac{(2n_2 + n_1)S_s}{tS_t} + d$$

- (From data book page no. 63, Table no V-2)

4. Distance between the rows of rivets.

- ***(a) For equal number of rivets in more than one row for lap joint or butt joint, the distance*** between the rows of rivets (*pb) should not be less than*
- **0.33 *p + 0.67 d*** *, for zig-zig riveting, and* **2 *d,*** *for chain riveting.*
- ***(b) For joints in which the number of rivets in outer rows is half the number of rivets in inner*** rows and if the inner rows are chain riveted, the distance between the outer rows and the next rows should not be less than
- **0.33 *p + 0.67 or 2 d***, *whichever is greater.*
- The distance between the rows in which there are full number of rivets shall not be less than 2*d.*
- ***(c) For joints in which the number of rivets in outer rows is half the number of rivets in inner*** rows and if the inner rows are zig-zig riveted, the distance between the outer rows and the next rows shall not be less than 0.2 *p + 1.15 d. The distance between the rows in which* there are full number of rivets (zig-zag) shall not be less than **0.165 *p + 0.67 d.***

- **5. *Thickness of butt strap.***

(*a*) The thickness of butt strap, in no case, shall be less than 10 mm.

(*b*) $t_1 = 1.125\, t$, for ordinary (chain riveting) single butt strap.

$t_1 = 1.125\, t \left(\dfrac{p-d}{p-2d}\right)$, for single butt straps, every alternate rivet in outer rows being omitted.

$t_1 = 0.625\, t$, for double butt-straps of equal width having ordinary riveting (chain riveting).

$t_1 = 0.625\, t \left(\dfrac{p-d}{p-2d}\right)$, for double butt straps of equal width having every alternate rivet in the outer rows being omitted.

(*c*) For unequal width of butt straps, the thicknesses of butt strap are

$t_1 = 0.75\, t$, for wide strap on the inside, and

$t_2 = 0.625\, t$, for narrow strap on the outside.

- 6. Margin : Margin (m) is taken as 1.5d.

LECTURE 4

Design of Circumferential Joints

- # Design of Circumferential Lap Joint for a Boiler

The following procedure is adopted for the design of circumferential lap joint for a boiler.

1. ***Thickness of the shell and diameter of rivets:*** *The thickness of the boiler shell and the* diameter of the rivet will be same as for longitudinal joint.

2. ***Number of rivets:*** *Since it is a lap joint, therefore the rivets will be in single shear.*

∴ Shearing resistance of the rivets, $P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$ (i)

where *n = Total number of rivets.*

Knowing the inner diameter of the boiler shell (*D), and the pressure of steam (P), the total* shearing load acting on the circumferential joint,

(ii)

$$W_s = \frac{\pi}{4} \times D^2 \times P$$

From equations (*i*) and (*ii*), we get

$$n \times \frac{\pi}{4} \times d^2 \times \tau = \frac{\pi}{4} \times D^2 \times P$$

$$\therefore \quad n = \left(\frac{D}{d}\right)^2 \frac{P}{\tau}$$

- **3. *Pitch of rivets :***

  If the efficiency of the longitudinal joint is known, then the efficiency of the circumferential joint may be obtained. It is generally taken as 50% of tearing efficiency in longitudinal joint, but if more than one circumferential joints is used, then it is 62% for the intermediate joints. Knowing the efficiency of the circumferential lap joint (η*c), the pitch of the rivets for the lap joint*

$$\eta_c = \frac{p_1 - d}{p_1}$$

4. *Number of rows:* *The number of rows of rivets for the circumferential joint may be obtained* from the following relation :

$$Number\ of\ rows = \frac{Total\ number\ of\ rivets}{Number\ of\ rivets\ in\ one\ row}$$

and the number of rivets in one row

$$\frac{\pi(D + t)}{p_1}$$ where D- Inner diameter

5. After finding out the number of rows, the type of the joint (*i.e. single riveted or double* riveted etc.) may be decided. Then the number of rivets in a row and pitch may be re-adjusted. In order to have a leak-proof joint, the pitch for the joint should be checked from Indian Boiler Regulations.

6. The distance between the rows of rivets (*i.e. back pitch) is calculated by* **$0.33\, p + 0.67\, d$**

7. After knowing the distance between the rows of rivets (*pb), the overlap of the plate may be* fixed by using the relation,

Overlap = (No. of rows of rivets – 1) $pb + m$

where m = *Margin.*

There are several ways of joining the longitudinal joint and the circumferential joint. One of the methods of joining the longitudinal and circumferential joint

Recommended Joints for Pressure Vessels:

The following table shows the recommended joints for pressure vessels or a Boiler

Diameter of shell (metres)	*Thickness of shell (mm)*	*Type of joint*
0.6 to 1.8	6 to 13	Double riveted
0.9 to 2.1	13 to 25	Triple riveted
1.5 to 2.7	19 to 40	Quadruple riveted

Design of uniform strength (Lozenge) Joint for structural Use :

A riveted joint known as ***Lozenge joint used for roof, bridge work or girders.*** In such a joint, diamond riveting is employed so that the joint is made of uniform strength.

Fig. shows a triple riveted double strap butt joint

Let *b = Width of the plate,*

t = Thickness of the plate, and

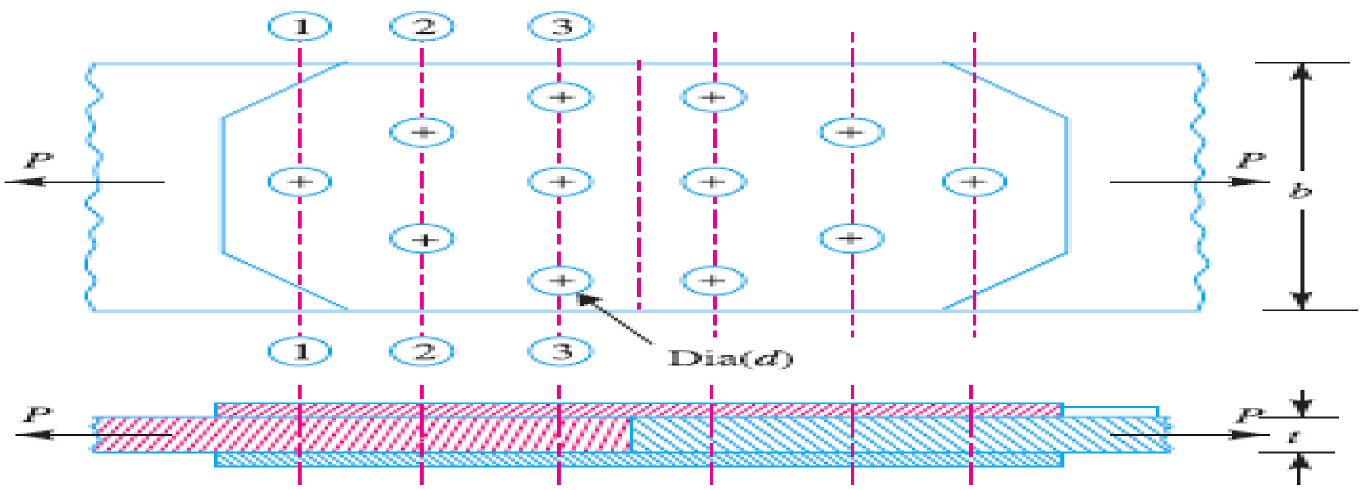
d = Diameter of the rivet hole.

In designing a Lozenge joint, the following procedure is adopted.

1. *Diameter of rivet*

The diameter of the rivet hole is obtained by using Unwin's formula, *i.e*

$$d = 6\sqrt{t}$$



2. *Number of rivets*

The number of rivets required for the joint may be obtained by the shearing or crushing resistance of the rivets.

Let *Pt = Maximum pull acting on the joint. This is the tearing resistance*

of the plate at the outer row which has only one rivet.

$= (b - d) t \times \sigma t$

and *n = Number of rivets.*

Since the joint is double strap butt joint, therefore the rivets are in double shear. It is assumed that resistance of a rivet in double shear is 1.75 times than in single shear in order to allow for possible eccentricity of load and defective workmanship.

∴ Shearing resistance of one rivet, $P_s = 1.75 \times \frac{\pi}{4} \times d^2 \times \tau$

and crushing resistance of one rivet,

$$P_c = d \times t \times \sigma_t$$

∴ Number of rivets required for the joint,

$$n = \frac{P_t}{least\ of\ P_t\ and\ P_c}$$

3. From the number of rivets, the number of rows and the number of rivets in each row is decided.

4. *Thickness of the butt straps:*

The thickness of the butt strap,

$$T_1 = 1.25\,t, \text{ for single cover strap}$$

$$= 0.75\,t, \text{ for double cover strap}$$

5. *Efficiency of the joint*

First of all, calculate the resistances along the sections 1-1, 2-2 and 3-3.

At section 1-1, there is only one rivet hole.

∴ Resistance of the joint in tearing along 1-1,

$P_{t1} = (b - d)\,t \times \sigma_t$

At section 2-2, there are two rivet holes.

∴ Resistance of the joint in tearing along 2-2,

$P_{t2} = (b - 2d)\,t \times \sigma_t$ *+ Strength of one rivet in front of section 2-2*

(This is due to the fact that for tearing off the plate at section 2-2, the rivet in front of section

2-2 *i.e. at section 1-1 must first fracture).*

Similarly at section 3-3 there are three rivet holes.

∴ Resistance of the joint in tearing along 3-3,

$P_{t3} = (b - 3d)\, t \times \sigma_t$ + Strength of 3 rivets in front of section 3-3

The least value of $P_{t1}, P_{t2}, P_{t3}, P_s$ or P_c *is the strength of the joint.*

We know that the strength of un-riveted plate,

$P = b \times t \times \sigma_t$

∴ Efficiency of the joint,

$$n = \frac{Least\ of\ P_{t1}, P_{t2}, P_{t3}, P_s, P_c}{P}$$

LECTURE 6

Design of Eccentrically loaded Riveted Joints

Design of Eccentric Loaded Riveted Joint :

When the line of action of the load does not pass through the centroid of the rivet system and thus all rivets are not equally loaded, then the joint is said to be an ***eccentric loaded riveted joint,*** The eccentric loading results in secondary shear caused by the tendency of force to twist the joint about the centre of gravity in addition to direct shear or primary shear.

Let *P = Eccentric load on the joint, and e = Eccentricity of the load i.e. the distance between the line of* action of the load and the centroid of the rivet system *i.e. G.*
The following procedure is adopted for the design of an eccentrically loaded riveted joint.

1. First of all, find the centre of gravity *G of the rivet system.*

Let *A = Cross-sectional area of each rivet,*

x1, x2, x3 etc. = Distances of rivets from OY, and

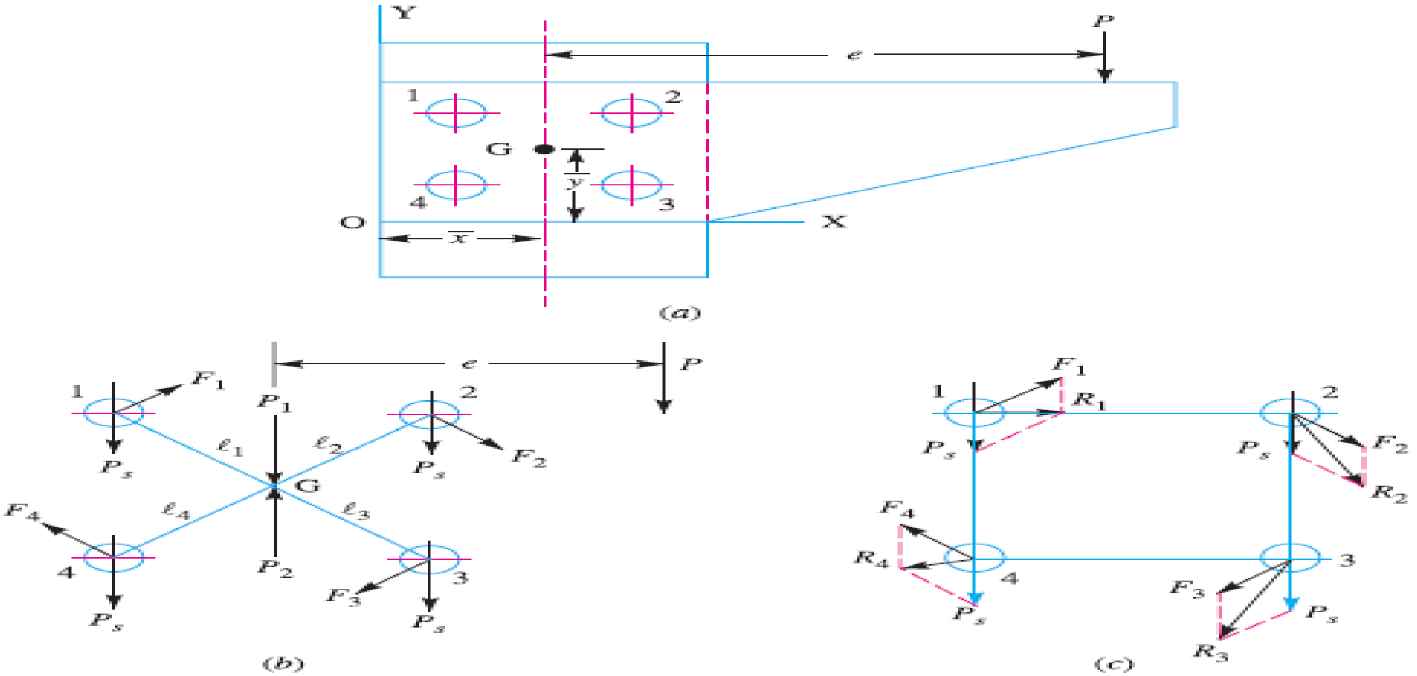
y1, y2, y3 etc. = Distances of rivets from OX.

• Design of Eccentric Loaded Riveted Joint :

We know that

$$\bar{X} = \frac{A_1X_1 + A_2X_2 + A_3X_3 + \cdots}{A_1 + A_2 + A_3 + \cdots}$$

$$\bar{Y} = \frac{A_1Y_1 + A_2Y_2 + A_3Y_3 + \cdots}{A_1 + A_2 + A_3 + \cdots}$$



2. **Introduce two forces *P1 and P2 at the centre of gravity 'G' of the rivet system. These forces*** are equal and opposite to *P as shown in Fig. (b).*

3. **Assuming that all the rivets are of the same size, the effect of *P1 = P is to produce direct shear*** load on each rivet of equal magnitude. Therefore, direct shear load on each rivet,

$P_s = \frac{P}{n}$,acting parallel to the load *P.*

4. **The effect of *P2 = P is to produce a turning moment of magnitude P × e which tends to rotate*** the joint about the centre of gravity '*G' of the rivet system in a clockwise direction. Due to the turning* moment, secondary shear load on each rivet is produced. In order to find the secondary shear load, the following two assumptions are made :

Design of Eccentric Loaded Riveted Joint :

(a) *The secondary shear load is proportional to the radial distance of the rivet under* consideration from the centre of gravity of the rivet system.

(***b***) *The direction of secondary shear load is perpendicular to the line joining the centre of the* rivet to the centre of gravity of the rivet system..

Let *F1, F2, F3 ... = Secondary shear loads on the rivets 1, 2, 3...etc.*

l1, l2, l3 ... = Radial distance of the rivets 1, 2, 3 ...etc. from the centre of gravity *'G' of the rivet system.*

∴ From assumption (*a*),

F1 ∝ l1 ; F2 ∝ l2 and so on

$$\frac{F_1}{l_1} = \frac{F_1}{l_1} = \frac{F_1}{l_1} = \cdots$$

$$F_1 = F_2 \times \frac{l_2}{l_1} \text{ and } F_3 = F_1 \times \frac{l_3}{l_1}$$

We know that the sum of the external turning moment due to the eccentric load and of internal resisting moment of the rivets must be equal to zero.

$$p \times e = F_1 l_1 + F_2 l_2 + F_3 l_3 + \cdots \ldots$$

$$= F_1 . l_1 + F_1 \times \frac{l_2}{l_1} \times l_2 + F_1 \times \frac{l_3}{l_1} \times l_3 + \ldots$$

$$= \frac{F_1}{l_1} \left[(l_1)^2 + (l_2)^2 + (l_3)^2 + \ldots \right]$$

From the above expression, the value of *F1 may be calculated and hence F2 and F3 etc. are* known. The direction of these forces are at right angles to the lines joining the centre of rivet to the centre of gravity of the rivet system, as shown in Fig. (*b), and should produce the moment in the* same direction (*i.e. clockwise or anticlockwise) about the centre of gravity, as the turning moment* (*P × e).*

5. The primary (or direct) and secondary shear load may be added vectorially to determine the resultant shear load (*R) on each rivet as shown in Fig. (c). It may also be obtained by using the* relation

$$R = \sqrt{(P_s)^2 + F^2 + 2\,P_s \times F \times \cos\theta}$$

Where

θ = Angle between the primary or direct shear load (P_s) and secondary shear load (F).

When the secondary shear load on each rivet is equal, then the heavily loaded rivet will be one in which the included angle between the direct shear load and secondary shear load is minimum. The maximum loaded rivet becomes the critical one for determining the strength of the riveted joint.

Knowing the permissible shear stress (τ), the diameter of the rivet hole may be obtained by using the relation,

$$Maximum\ Rusultant\ shear\ load(R) = \frac{\pi}{4} \times d^2 \times \tau$$

LECTURE 7

Introduction to welded joints

Welded Joint

Introduction

A welded joint is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together, with or without the application of pressure and a filler material. The heat required for the fusion of the material may be obtained by burning of gas (in case of gas welding) or by an electric arc (in case of electric arc welding). The latter method is extensively used because of greater speed of welding. Welding is extensively used in fabrication as an alternative method for casting or forging and as a replacement for bolted and riveted joints. It is also used as a repair medium *e.g. to reunite metal at a crack, to build up* a small part that has broken off such as gear tooth or to repair a worn surface such as a bearing surface.

Advantages and Disadvantages of Welded Joints over Riveted Joints

Advantages

1. The welded structures are usually lighter than riveted structures. This is due to the reason, that in welding, gussets or other connecting components are not used.

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

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

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

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

6. A welded joint has a great strength. Often a welded joint has the strength of the parent metal itself.

7. Sometimes, the members are of such a shape (*i.e. circular steel pipes) that they afford* difficulty for riveting. But they can be easily welded.

Disadvantages

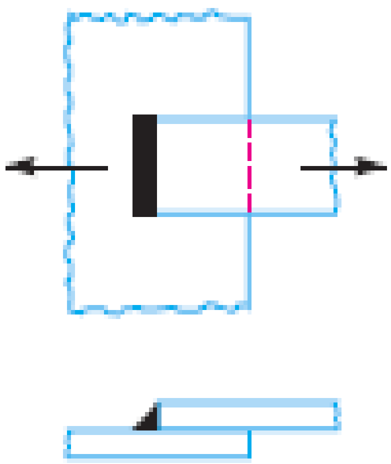
1. **Since there is an uneven heating and cooling during fabrication, therefore the members** may get distorted or additional stresses may develop.
2. **It requires a highly skilled labour and supervision.**
3. **Since no provision is kept for expansion and contraction in the frame, therefore there is a** possibility of cracks developing in it.
4. **The inspection of welding work is more difficult than riveting work.**

Types of Welded Joints

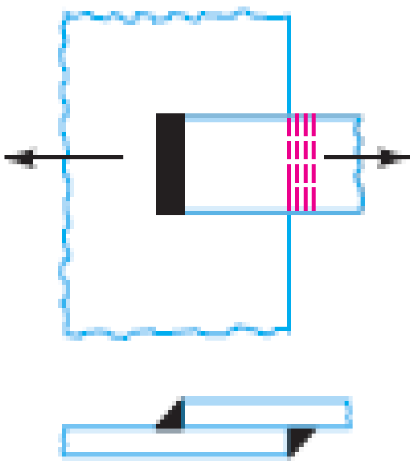
Following two types of welded joints are important from the subject point of view:

1. Lap joint or fillet joint, and 2. Butt joint.

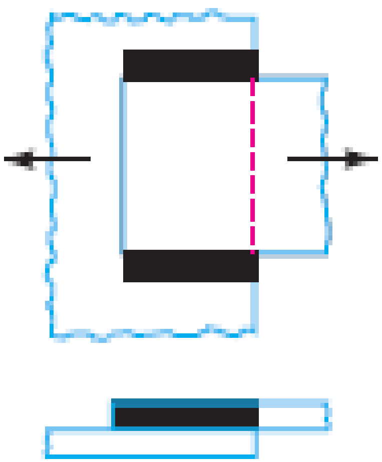
1. Lap Joint:



(a) Single transverse.

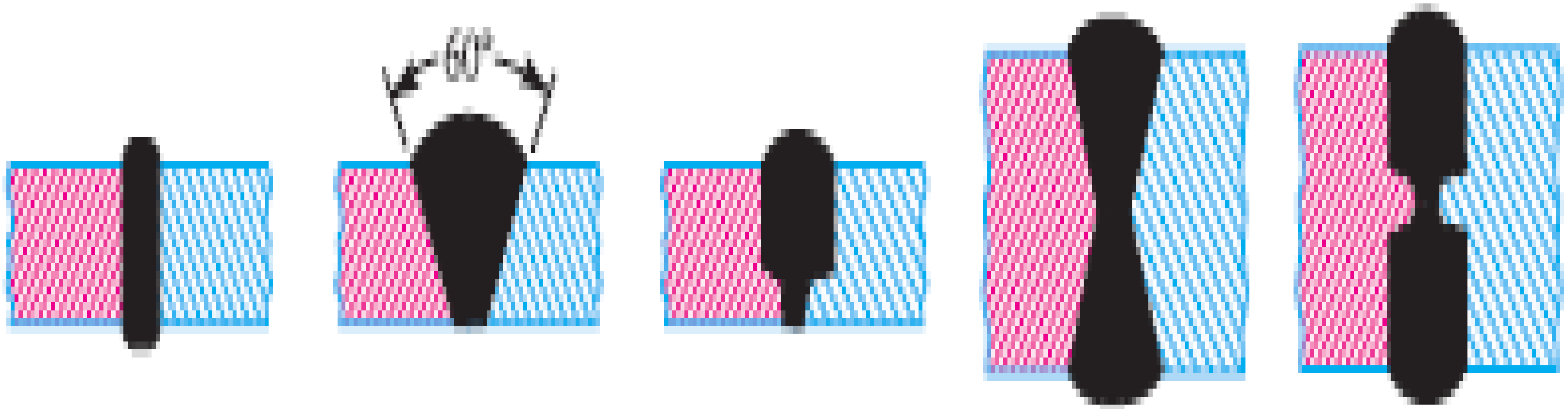


(b) Double transverse.



(c) Parallel fillet.

Butt Joint

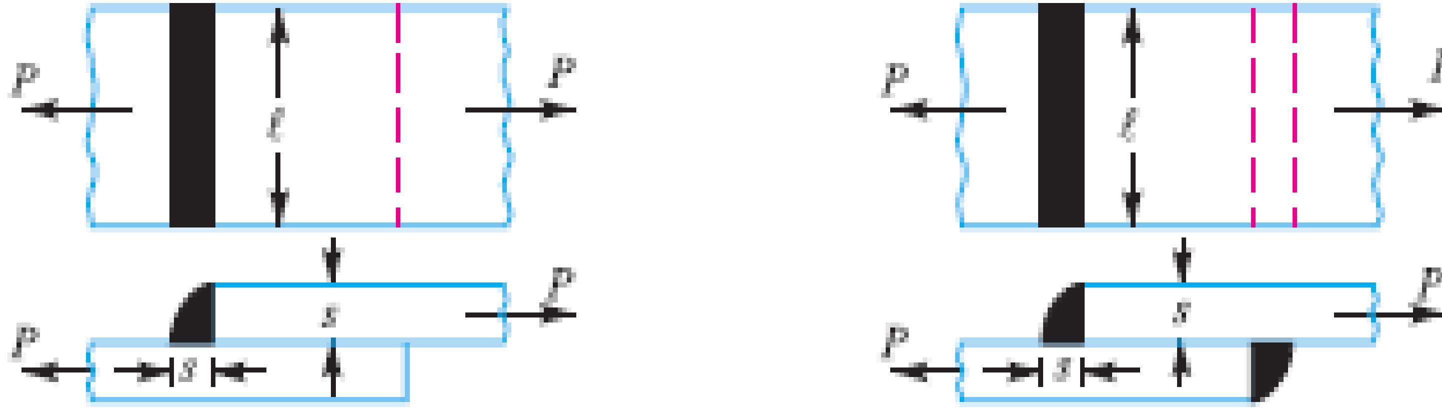


(*a*) Square butt joint. (*b*) Single *V*-butt joint. (*c*) Single *U*-butt joint. (*d*) Double *V*-butt joint. (*e*) Double *U*-butt joint.

LECTURE 8

Stresses Acting on Welded Joints

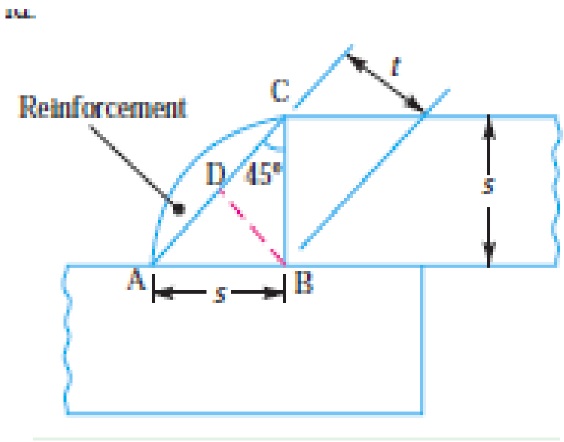
Strength of Transverse Fillet Welded Joints



(a) Single transverse fillet weld. (b) Double transverse fillet weld.

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 shown in Fig. The length of each side is known as ***leg or size of the weld and the perpendicular 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.

- Let t = Throat thickness (BD),
- s = Leg or size of weld,
- = Thickness of plate, and
- l = Length of weld,
- From Fig. , we find that the throat thickness,
- $t = s \times \sin 45° = 0.707\, s$
- ∴ *Minimum area of the weld or throat area,
- A = Throat thickness ×Length of weld $= t \times l = 0.707\, s \times l$



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

P = Throat area × Allowable tensile stress $= 0.707\, s \times l \times \sigma t$

and tensile strength of the joint for double fillet weld,

$P = 2 \times 0.707\, s \times l \times \sigma t = 1.414\, s \times l \times \sigma t$

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 Fig. *We have already discussed in the previous article, that the* minimum area of weld or the throat area,

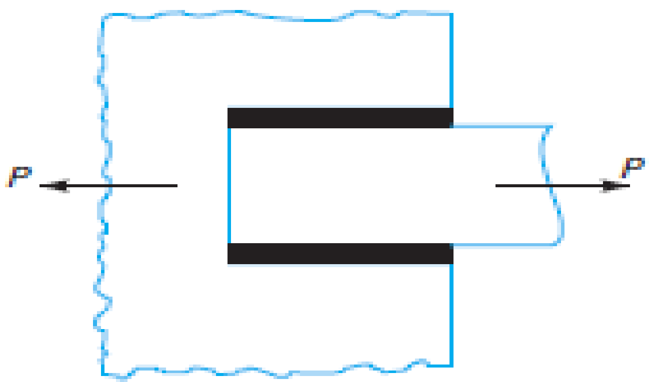
$A = 0.707\, s \times l$

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

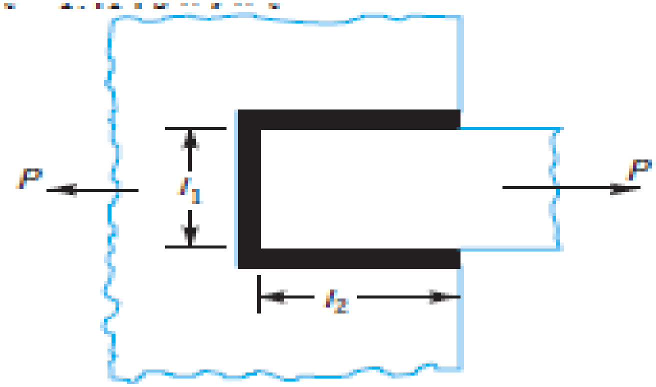
$P = \text{Throat area} \times \text{Allowable shear stress} = 0.707\, s \times l \times \tau$

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

$P = 2 \times 0.707 \times s \times l \times \tau = 1.414\, s \times l \times \tau$



(*a*) Double parallel fillet weld.



(*b*) Combination of transverse and parallel fillet weld.

Special Cases of Fillet Welded Joints

1. ***Circular fillet weld subjected to torsion. Consider a circular rod connected to a rigid plate*** by a fillet weld as shown in Fig. 10.9.

Let *d = Diameter 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 $= \dfrac{\pi t d^3}{4}$

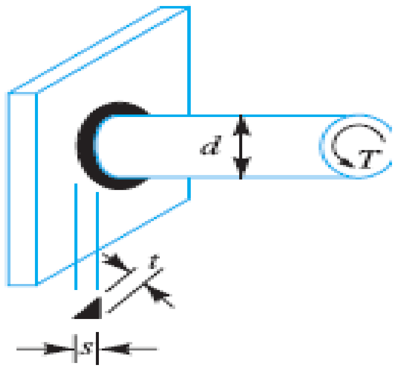


Fig. 10.9. Circular fillet weld subjected to torsion.

We know that shear stress for the material,

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

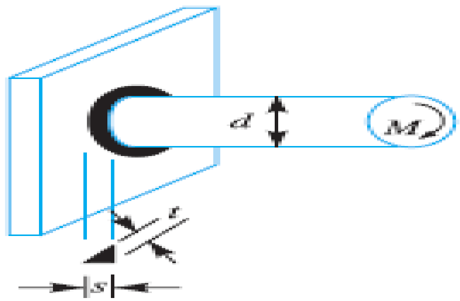
$$= \frac{T \times d/2}{\pi t d^3/4} = \frac{2T}{\pi t d^2}$$

This shear stress occurs in a horizontal plane along a leg of the fillet weld. The maximum shear occurs on the throat of weld which is inclined at 45° to the horizontal plane.

∴ Length of throat, $t = s \sin 45° = 0.707\, s$

2. *Circular fillet weld subjected to bending moment. Consider a circular rod connected to a*

rigid plate by a fillet weld as shown in Fig. 10.10.



Let *d = Diameter of rod,*

M = Bending moment acting on the rod,

s = Size (or leg) of weld,

t = Throat thickness,

***Z = Section modulus of the weld section*

$$= \frac{\pi t d^2}{4}$$

We know that the bending stress,

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

This bending stress occurs in a horizontal plane along a leg of the fillet weld. The maximum bending stress occurs on the throat of the weld which is inclined at 45° to the horizontal plane.

$\therefore$ Length of throat, $t = s \sin 45° = 0.707\, s$

and maximum bending stress,

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

3. *Long fillet weld subjected to torsion. Consider a* vertical plate attached to a horizontal plate by two identical fillet welds as shown in Fig. 10.11.

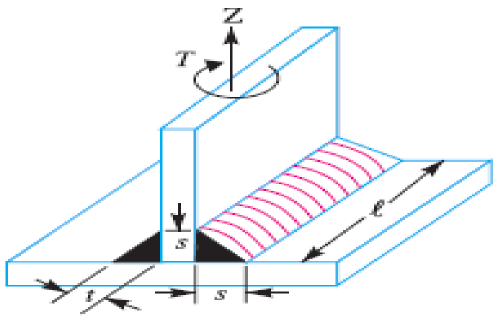


Fig. 10.11. Long fillet weld subjected to torsion.

Let *T = Torque acting on the 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

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

($\because$ of both sides weld)

It may be noted that the effect of the applied torque is to rotate the vertical plate about the Z-axis through its mid point. This rotation is resisted by shearing stresses developed between two fillet welds and the horizontal plate. It is assumed that these horizontal shearing stresses vary from zero at the Z-axis and maximum at the ends of the plate. This variation of shearing stress is analogous to the variation of normal stress over the depth (*l) of a beam subjected to pure bending.*

$\therefore$ Shear stress, $$\tau = \frac{T \times l/2}{t \times l^3/6} = \frac{3\,T}{t \times l^2}$$

The maximum shear stress occurs at the throat and is given by

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

LECTURE 9

Introduction-Bolted Joints

Bolted Joints:

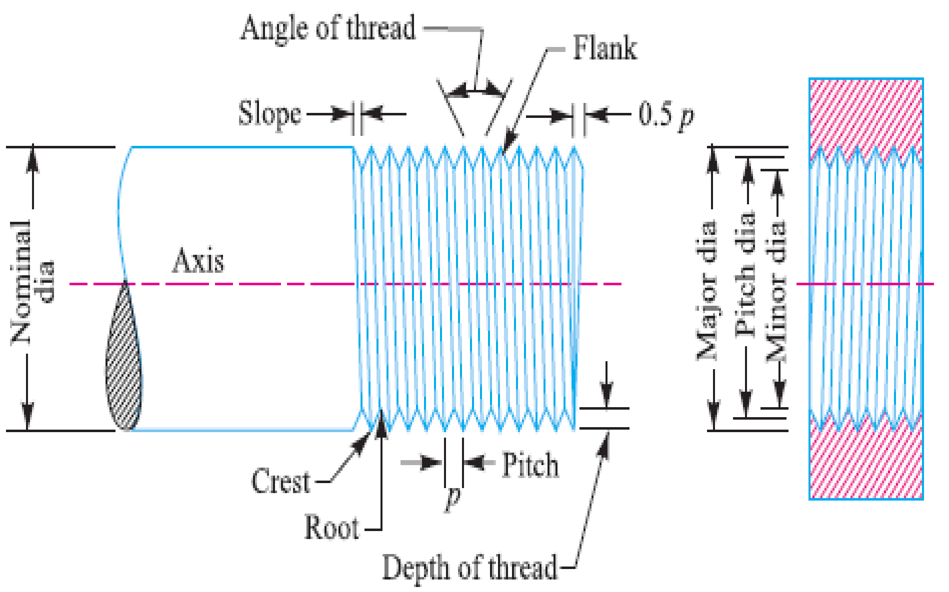
Introduction:

- 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 as 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 the machine parts are required to be readily connected or disconnected without damage to the machine or the fastening.

- **Advantages and Disadvantages of Screwed Joints**
- ***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 standardization and highly efficient manufacturing processes.
- ***Disadvantages***
- The main disadvantage of the screwed joints is the stress concentration in the threaded portions which are vulnerable points under variable load conditions.

Important Terms Used in Screw Threads

- Major diameter
- Minor diameter
- Pitch diameter
- Pitch
- Lead
- Crest
- Root
- Depth of thread
- Flank
- Angle of thread
- Slope

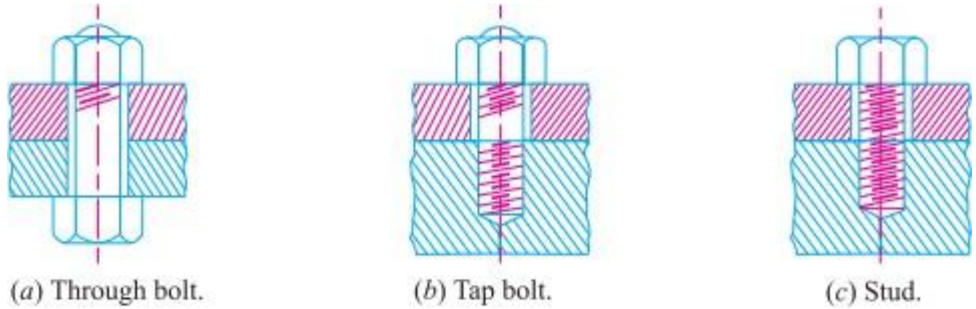


Forms of Screw Threads

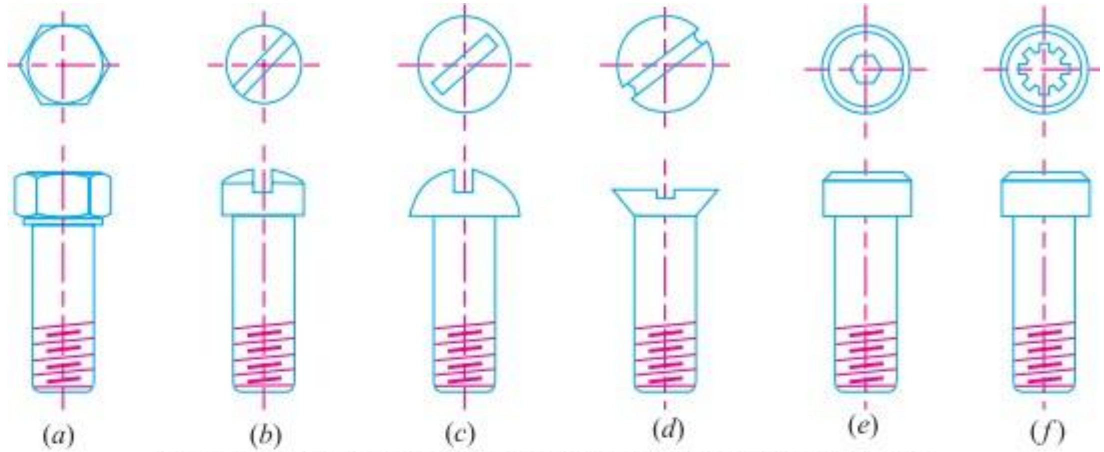
- **British standard Whitworth (B.S.W.) thread.**
- **British association (B.A.) thread**
- **American national standard thread**
- **Square thread**
- **Acme thread**
- **Knuckle thread**
- **Buttress thread**
- **Buttress thread**

Common Types of Screw Fastenings

- Through bolts
- Tap bolts
- Studs

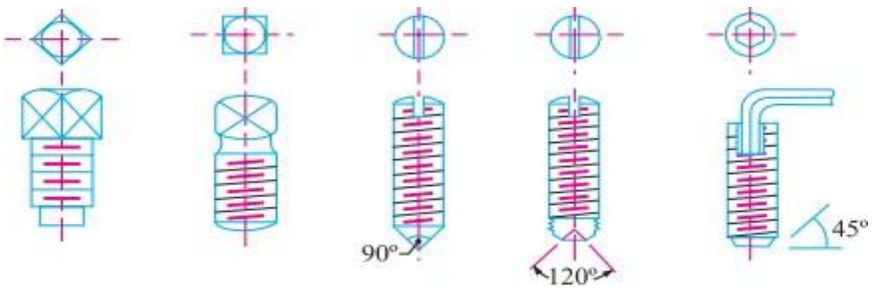


Cap screws



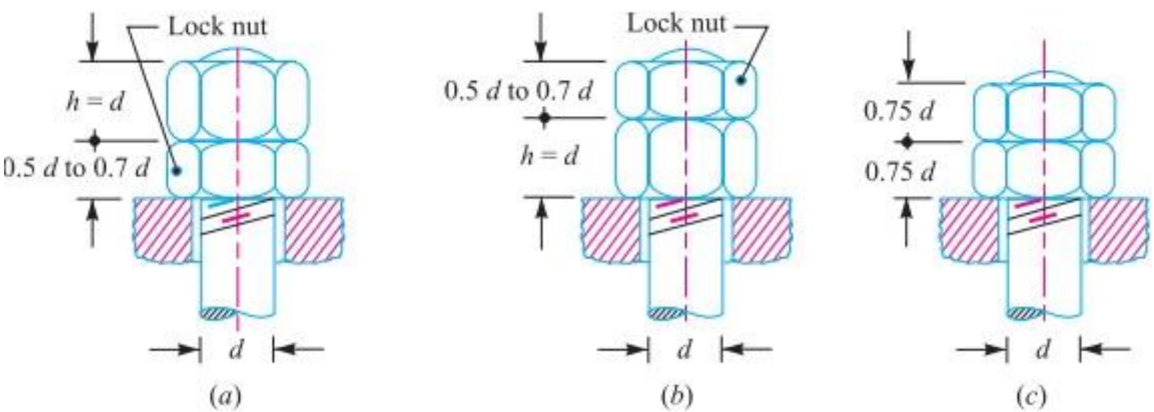
(a) Hexagonal head; (b) Fillister head; (c) Round head; (d) Flat head;
(e) Hexagonal socket; (f) Fluted socket.

- Machine screws
- Set screws



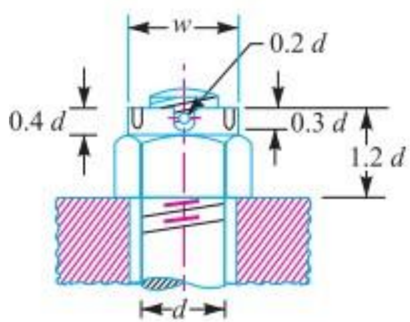
Locking Devices

Jam nut or lock nut

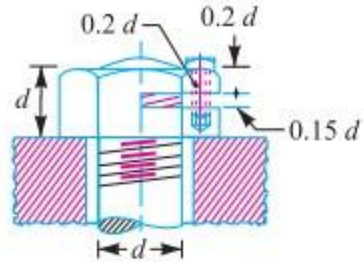


Jam nut or lock nut.

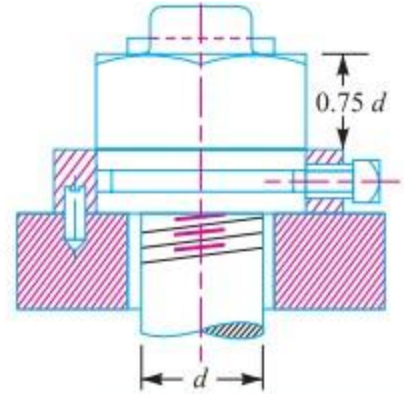
- Castle nut
- Sawn nut
- Penn, ring or grooved nut



Castle nut.



Sawn nut.

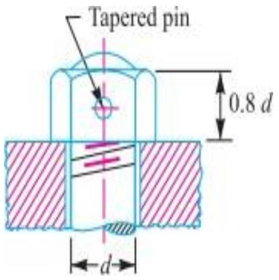


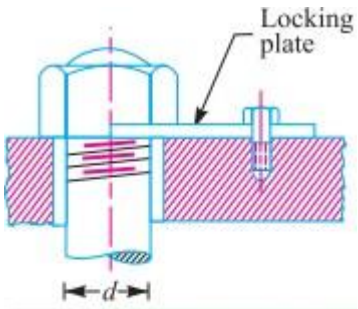
Penn, ring or grooved nut.

Locking with pin

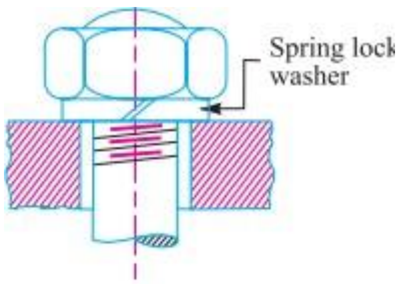
Locking with plate

Spring lock washer





Locking with plate.



Locking with washer.

LECTURE 10

Stresses on bolted joints

Stresses in Screwed Fastening due to Static Loading

- Internal stresses due to screwing up forces,
- Stresses due to external forces, and
- Stress due to combination of stresses at (1) and (2).

<u>Initial Stresses due to Screwing up Forces</u>

1. **Tensile stress due to stretching of bolt**

   The initial tension in a bolt, based on experiments, may be found by the relation

   $P_i = 2840\ d$ N

   P_i = Initial tension in a bolt, and

   d= Nominal diameter of bolt, in mm.

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. In such cases

Pi= 1420 d N

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

P= Permissible stress × Cross-sectional area at bottom of the thread.

$$\text{Stress area} = \frac{\pi}{4}\left(\frac{d_p + d_c}{2}\right)^2$$

where d_p = Pitch diameter, and

d_c = Core or minor diameter.

2. ***Torsional shear stress caused by the frictional resistance of the threads during its tightening.*** The torsional shear stress caused by the frictional resistance of the threads during its tightening may be obtained by using the torsion equation. We know that

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

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

where τ = Torsional shear stress,

T = Torque applied, and

d_c = Minor or core diameter of the thread.

It has been shown during experiments that due to repeated unscrewing and tightening of the nut, there is a gradual scoring of the threads, which increases the torsional twisting moment (T).

3. ***Shear stress across the threads.*** The average thread shearing stress for the screw (τ_s) is obtained by using the relation :

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

where b = Width of the thread section at the root.

The average thread shearing stress for the nut is

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

where d = Major diameter.

4. ***Compression or crushing stress on threads.*** The compression or crushing stress between the threads (σ_c) may be obtained by using the relation :

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

where d = Major diameter,

d_c = Minor diameter, and

n = Number of threads in engagement.

5. ***Bending stress if the surfaces under the head or nut are not perfectly parallel to the bolt axis.*** When the outside surfaces of the parts to be connected are not parallel to each other, then the bolt will be subjected to bending action. The bending stress (σ_b) induced in the shank of the bolt is given by

$$\sigma_b = \frac{x \,.\, E}{2l}$$

where x = Difference in height between the extreme corners of the nut or head,

l = Length of the shank of the bolt, and

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

Design of a Nut

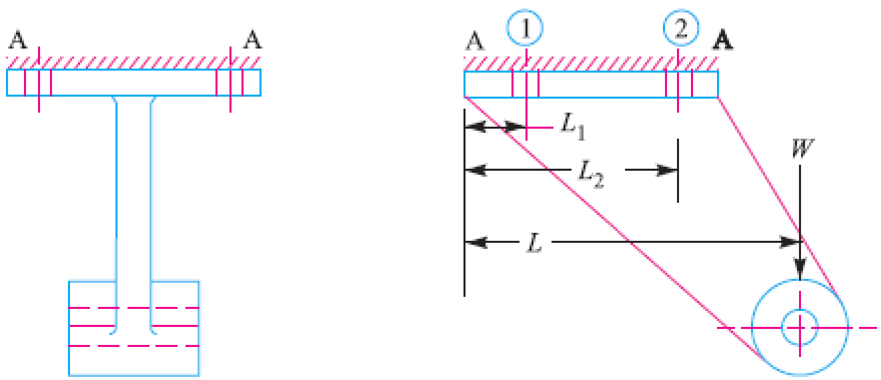
- When a bolt and nut is made of mild steel, then the effective height of nut is made equal to the nominal diameter of the bolt.
- If the nut is made of weaker material than the bolt, then the height of nut should be larger, such as 1.5 *d* for gun metal, 2 *d* for cast iron and 2.5 *d* for aluminium alloys (where *d* is the nominal diameter of the bolt).
- In case cast iron or aluminium nut is used, then *V*-threads are permissible only for permanent fastenings, because threads in these materials are damaged due to repeated screwing and unscrewing.

- **Bolted Joints under Eccentric Loading**
-
- There are many applications of the bolted joints which are subjected to eccentric loading such as a wall bracket, pillar crane, etc. The eccentric load may be
  - Parallel to the axis of the bolts,
  - Perpendicular to the axis of the bolts, and
  - In the plane containing the bolts.

Eccentric Load Acting Parallel to the Axis of Bolts

A little consideration will show that each bolt is subjected to a direct tensile load of

$$W_{t1} = \frac{W}{n}, \text{ where } n \text{ is the number of bolts.}$$



$\therefore$ Load on each bolt at distance L_1,

$$W_1 = w.L_1$$

and moment of this load about the tilting edge

$$= w_1.L_1 \times L_1 = w\,(L_1)^2$$

Similarly, load on each bolt at distance L_2,

$$W_2 = w.L_2$$

and moment of this load about the tilting edge

$$= w.L_2 \times L_2 = w\,(L_2)^2$$

$\therefore$ Total moment of the load on the bolts about the tilting edge

$$= 2w\,(L_1)^2 + 2w\,(L_2)^2 \qquad \text{...}(i)$$

... ($\because$ There are two bolts each at distance of L_1 and L_2)

Also the moment due to load W about the tilting edge

$$= W.L \qquad \text{...(ii)}$$

From equations (*i*) and (*ii*), we have

$$W.L = 2w(L_1)^2 + 2w(L_2)^2 \quad \text{or} \quad w = \frac{W.L}{2[(L_1)^2 + (L_2)^2]} \qquad \text{...(iii)}$$

It may be noted that the most heavily loaded bolts are those which are situated at the greatest distance from the tilting edge. In the case discussed above, the bolts at distance L_2 are heavily loaded.

$\therefore$ Tensile load on each bolt at distance L_2,

$$W_{t2} = W_2 = w.L_2 = \frac{W.L.L_2}{2[(L_1)^2 + (L_2)^2]} \qquad \text{... [From equation (iii)]}$$

and the total tensile load on the most heavily loaded bolt,

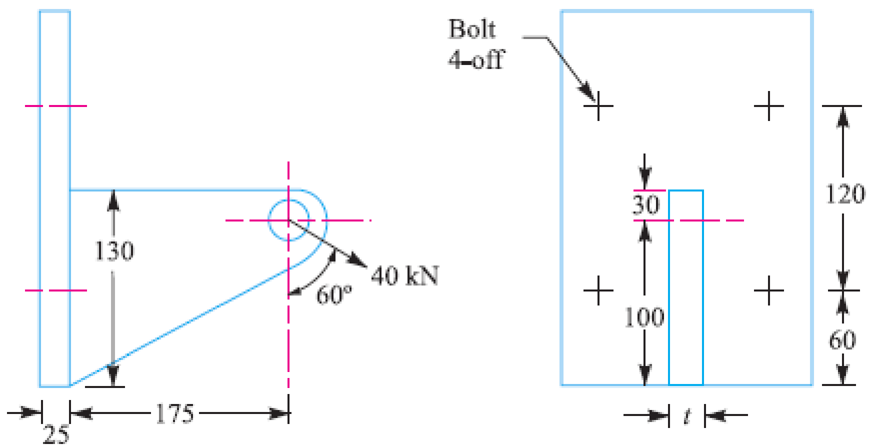
$$W_t = W_{t1} + W_{t2} \qquad \text{...(iv)}$$

If d_c is the core diameter of the bolt and σ_t is the tensile stress for the bolt material, then total tensile load,

$$W_t = \frac{\pi}{4}(d_c)^2 \sigma_t \qquad \text{...(v)}$$

From equations (*iv*) and (*v*), the value of d_c may be obtained.

Eccentric Load Acting Perpendicular to the Axis of Bolts



All dimensions in mm.

Problem: A double riveted lap joint is made between 15 mm thick plates. The rivet diameter and pitch are 25 mm and 75 mm respectively. If the ultimate stresses are 400 MPa in tension, 320 MPa in shear and 640 MPa in crushing, find the minimum force per pitch which will rupture the joint. If the above joint is subjected to a load such that the factor of safety is 4, find out the actual stresses developed in the plates and the rivets.

Solution. Given : $t = 15$ mm ; $d = 25$ mm ; $p = 75$ mm ; $\sigma_{tu} = 400$ MPa $= 400$ N/mm^2 ; $\tau_u = 320$ MPa $= 320$ N/mm^2 ; $\sigma_{cu} = 640$ MPa $= 640$ N/mm^2

Minimum force per pitch which will rupture the joint

Since the ultimate stresses are given, therefore we shall find the ultimate values of the resistances of the joint. We know that ultimate tearing resistance of the plate per pitch,

$$P_{tu} = (p - d)t \times \sigma_{tu} = (75 - 25)15 \times 400 = 300\ 000 \text{ N}$$

Ultimate shearing resistance of the rivets per pitch,

$$P_{su} = n \times \frac{\pi}{4} \times d^2 \times \tau_u = 2 \times \frac{\pi}{4}(25)^2\, 320 = 314\ 200 \text{ N} \quad \ldots(\because n = 2)$$

and ultimate crushing resistance of the rivets per pitch,

$$P_{cu} = n \times d \times t \times \sigma_{cu} = 2 \times 25 \times 15 \times 640 = 480\ 000 \text{ N}$$

From above we see that the minimum force per pitch which will rupture the joint is 300 000 N or 300 kN. **Ans.**

Actual stresses produced in the plates and rivets

Since the factor of safety is 4, therefore safe load per pitch length of the joint

$$= 300\ 000/4 = 75\ 000 \text{ N}$$

Let σ_{ta}, τ_a and σ_{ca} be the actual tearing, shearing and crushing stresses produced with a safe load of 75 000 N in tearing, shearing and crushing.

We know that actual tearing resistance of the plates (P_{ta}),

$$75\,000 = (p - d)\,t \times \sigma_{ta} = (75 - 25)15 \times \sigma_{ta} = 750\,\sigma_{ta}$$

$$\therefore \quad \sigma_{ta} = 75\,000 / 750 = 100 \text{ N/mm}^2 = 100 \text{ MPa} \quad \textbf{Ans.}$$

Actual shearing resistance of the rivets (P_{sa}),

$$75\,000 = n \times \frac{\pi}{4} \times d^2 \times \tau_a = 2 \times \frac{\pi}{4} (25)^2\,\tau_a = 982\,\tau_a$$

$$\therefore \quad \tau_a = 75000 / 982 = 76.4 \text{ N/mm}^2 = 76.4 \text{ MPa} \quad \textbf{Ans.}$$

and actual crushing resistance of the rivets (P_{ca}),

$$75\,000 = n \times d \times t \times \sigma_{ca} = 2 \times 25 \times 15 \times \sigma_{ca} = 750\,\sigma_{ca}$$

$$\therefore \quad \sigma_{ca} = 75000 / 750 = 100 \text{ N/mm}^2 = 100 \text{ MPa} \quad \textbf{Ans.}$$

Find the efficiency of the following riveted joints:
1. Single riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 50 mm. 2. Double riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 65 mm. Assume Permissible tensile stress in plate = 120 MPa Permissible shearing stress in rivets = 90 MPa Permissible crushing stress in rivets = 180 MPa.

Solution. Given : $t = 6$ mm ; $d = 20$ mm ; $\sigma_t = 120$ MPa $= 120$ N/mm^2 ; $\tau = 90$ MPa $= 90$ N/mm^2 ; $\sigma_c = 180$ MPa $= 180$ N/mm^2

1. *Efficiency of the first joint*

Pitch, $p = 50$ mm ...(Given)

First of all, let us find the tearing resistance of the plate, shearing and crushing resistances of the rivets.

(*i*) *Tearing resistance of the plate*

We know that the tearing resistance of the plate per pitch length,

$$P_t = (p - d)\, t \times \sigma_t = (50 - 20)\, 6 \times 120 = 21\ 600 \text{ N}$$

(*ii*) *Shearing resistance of the rivet*

Since the joint is a single riveted lap joint, therefore the strength of one rivet in single shear is taken. We know that shearing resistance of one rivet,

$$P_s = \frac{\pi}{4} \times d^2 \times \tau = \frac{\pi}{4}\ (20)^2\ 90 = 28\ 278 \text{ N}$$

(*iii*) *Crushing resistance of the rivet*

Since the joint is a single riveted, therefore strength of one rivet is taken. We know that crushing resistance of one rivet,

$$P_c = d \times t \times \sigma_c = 20 \times 6 \times 180 = 21\ 600 \text{ N}$$

$\therefore$ Strength of the joint

$$= \text{Least of } P_t, P_s \text{ and } P_c = 21\ 600 \text{ N}$$

$\therefore$ Strength of the joint

$$= \text{Least of } P_t, P_s \text{ and } P_c = 21\,600 \text{ N}$$

We know that strength of the unriveted or solid plate,

$$P = p \times t \times \sigma_t = 50 \times 6 \times 120 = 36\,000 \text{ N}$$

$\therefore$ Efficiency of the joint,

$$\eta = \frac{\text{Least of } P_t, P_s \text{ and } P_c}{P} = \frac{21\,600}{36\,000} = 0.60 \text{ or } 60\% \quad \textbf{Ans.}$$

2. *Efficiency of the second joint*

Pitch, $p = 65$ mm ...(Given)

(*i*) *Tearing resistance of the plate,*

We know that the tearing resistance of the plate per pitch length,

$$P_t = (p - d)\, t \times \sigma_t = (65 - 20)\, 6 \times 120 = 32\,400 \text{ N}$$

(*ii*) *Shearing resistance of the rivets*

Since the joint is double riveted lap joint, therefore strength of two rivets in single shear is taken. We know that shearing resistance of the rivets,

$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times \frac{\pi}{4}\,(20)^2\,90 = 56\,556 \text{ N}$$

(*iii*) *Crushing resistance of the rivet*

Since the joint is double riveted, therefore strength of two rivets is taken. We know that crushing resistance of rivets,

$$P_c = n \times d \times t \times \sigma_c = 2 \times 20 \times 6 \times 180 = 43\,200 \text{ N}$$

∴ Strength of the joint

$$= \text{Least of } P_t, P_s \text{ and } P_c = 32\,400 \text{ N}$$

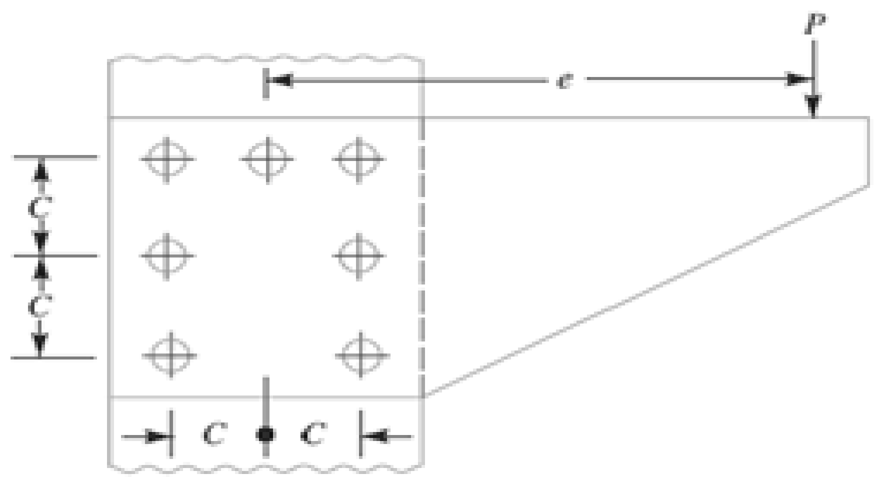
We know that the strength of the unriveted or solid plate,

$$P = p \times t \times \sigma_t = 65 \times 6 \times 120 = 46\,800 \text{ N}$$

∴ Efficiency of the joint,

$$\eta = \frac{\text{Least of } P_t, P_s \text{ and } P_c}{P} = \frac{32\,400}{46\,800} = 0.692 \text{ or } 69.2\% \quad \textbf{Ans.}$$

Problem: An eccentrically loaded lap riveted joint is to be designed for a steel bracket as shown in Fig. 2. The bracket plate is 25 mm thick. All rivets are to be of the same size. Load on the bracket, P = 50 kN ; rivet spacing, C = 100 mm; load arm, e = 400 mm. Permissible shear stress is 65 MPa and crushing stress is 120 MPa. Determine the size of the rivets to be used for the joint.



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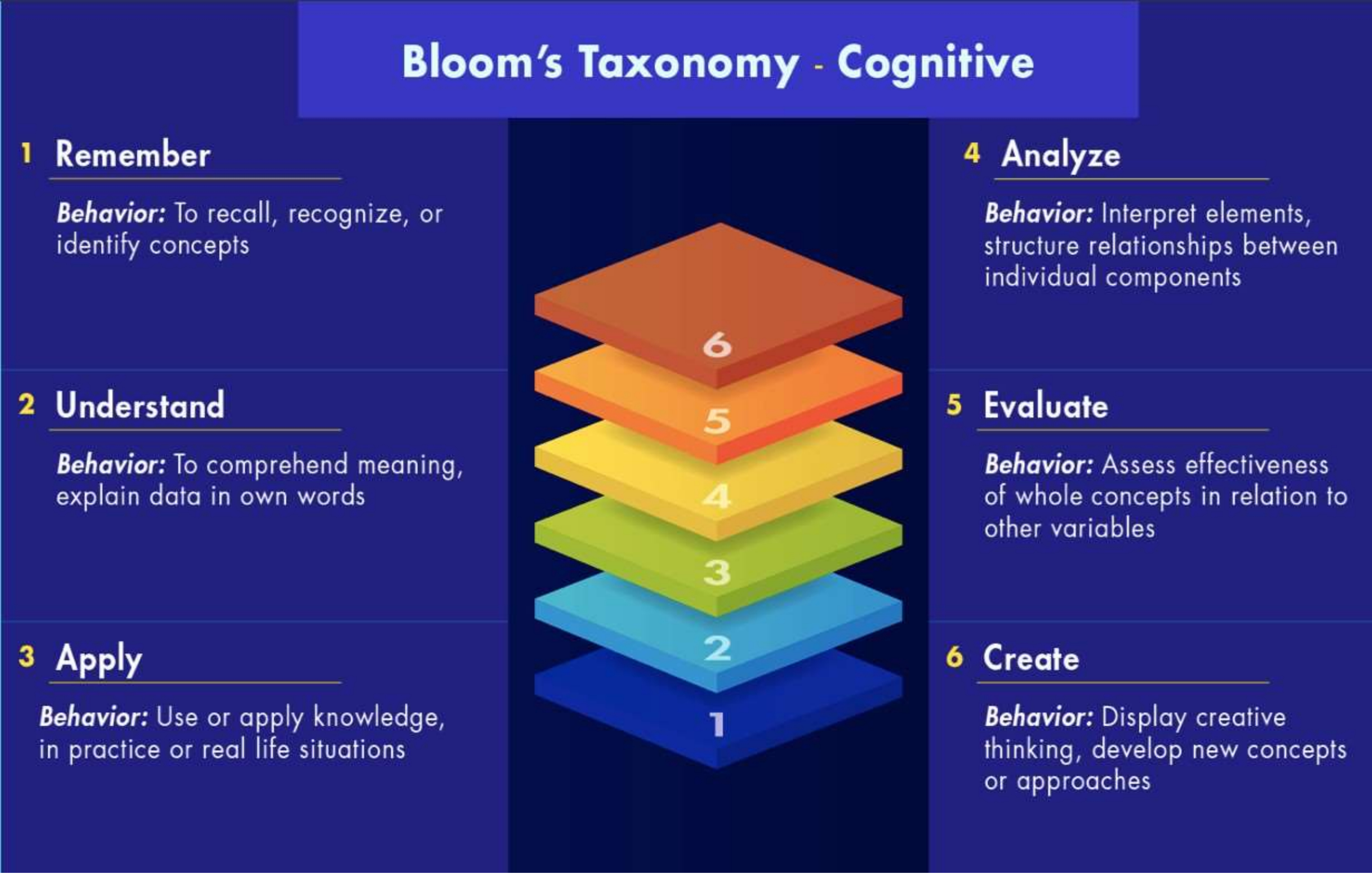


(Source: TASK Training material)

UNIT 4

CO4: To learn the design Procedure for the different machine elements such as fasteners, couplings, keys, axially loaded joints etc.

UNIT – IV (SYLLABUS)

Design of Keys

- Introduction to keys
- Types of keys
- Design of keys-stresses in keys

Design of Joints

- Cotter joints
- Spigot and socket, sleeve and cotter
- Jib and cotter joints-knuckle joints

COURSE OUTLINE

UNIT -4

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
1	**Introduction to keys and types of keys**	**keys**	• **Understand the importance of keys (B2)**
2	Stresses in keys	forces	• Understand forces acting on keys (B3)
3	Cotter joints and types of cotter joints		• understand importance of cotter joints(B2) • Understand difference between socket and spigot joints & sleeve and cotter joints(B2) • Understand design procedure and analysis of joints . (B3&B4)

LECTURE 1

Introduction-Keys

UNIT IV
KEYS, COTTERS AND KNUCKLE JOINTS

Introduction to Keys:

A key is a piece of mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them. It is always inserted parallel to the axis of the shaft.

Types of Keys:

There are five main types of keys: sunk, saddle, tangent, round, and spline.

- Sunk key. Types of sunk keys: rectangular, square, parallel sunk, gib-head, feather, and Woodruff.
- Saddle keys
- Tangent keys
- Spline keys

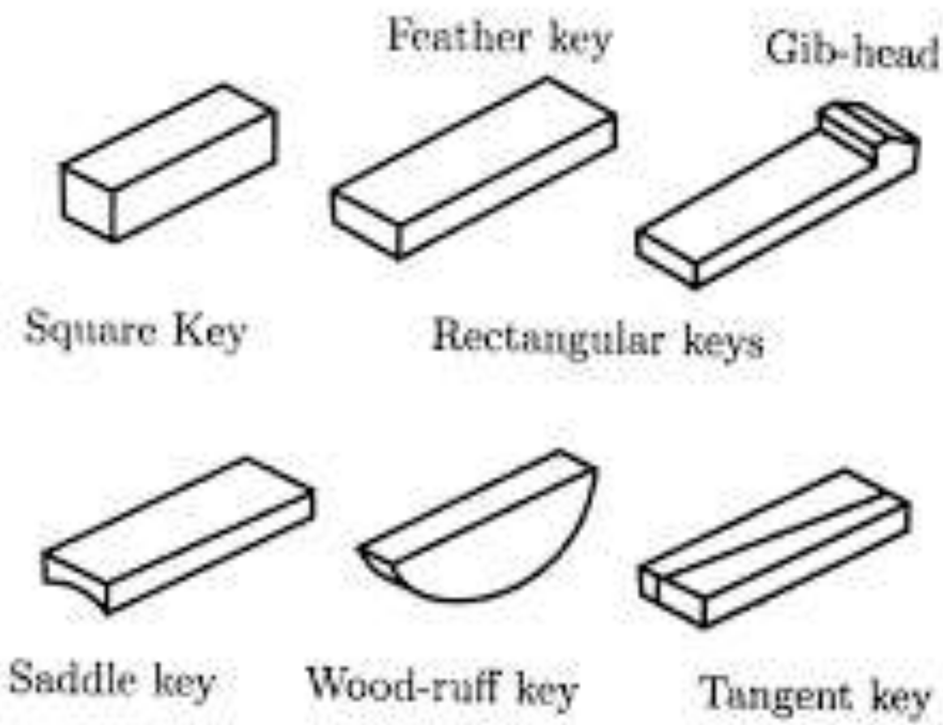


Figure 1: Types of Keys

Revision Questions

1. What is the function of key
2. Explain Types of keys

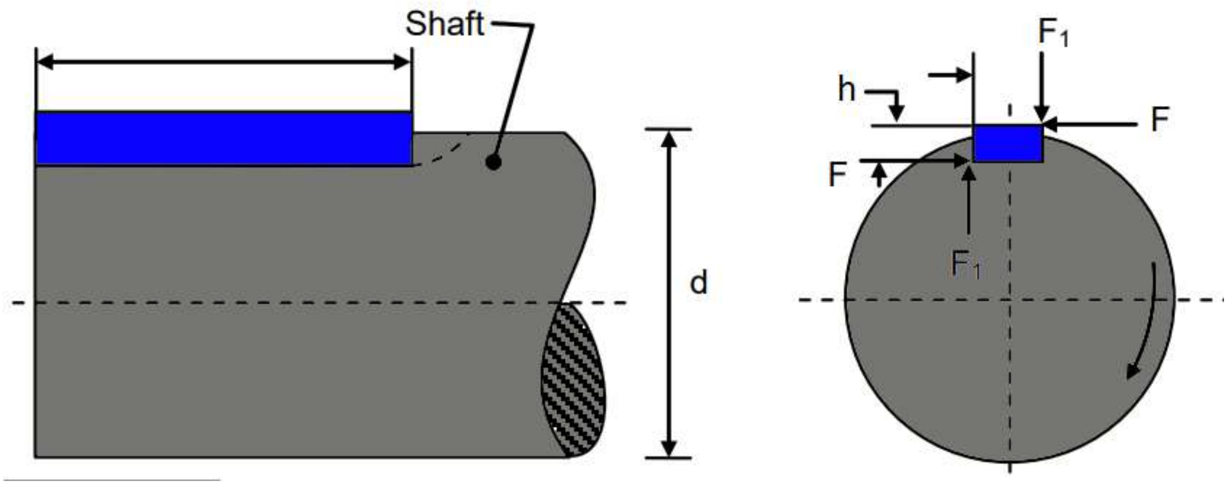
LECTURE 2

Stresses acting on keys

Forces acting on keys:

When a key is used in transmitting of torque from shaft to rotor or hub. The following two types of forces act on the key .There are many types of forces act on the key only square and flat keys are extensively used in practice.

1. Shearing stress
2. Compressive stress



Strengthy Of Sunk Key:

A key connecting the shaft and hub is shown in Fig.

Let T = Torque transmitted by the shaft,

F = Tangential force acting at the circumference of the shaft, d = Diameter of shaft,

l = Length of key, w = Width of key.

t = Thickness of key, and

τ and σc = Shear and crushing stresses for the material of key.

A little consideration will show that due to the power transmitted by the shaft, the key may fail due to shearing or crushing. Considering shearing of the key, the tangential shearing force acting at the circumference of the shaft,

F = Area resisting shearing × Shear stress = l × w × τ Therefore, Torque transmitted by the shaft,

$$T = F \times \frac{d}{2} = l \times w \times \tau \times \frac{d}{2} \qquad \ldots(i)$$

Considering crushing of the key, the tangential crushing force acting at the circumference of the shaft,

F = Area resisting crushing × Crushing stress

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

Therefore, Torque transmitted by the shaft,

$$T = F \times \frac{d}{2} = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2} \qquad \text{...(ii)}$$

The key is equally strong in shearing and crushing, if

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

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

In order to find the length of the key to transmit full power of the shaft, the shearing strength of the key is equal to the torsional shear strength of the shaft. We know that the shearing strength of key,

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

$$T = \frac{\pi}{16} \times \tau_1 \times d^3$$

$$i \times w \times \tau \times \frac{d}{2} = \frac{\pi}{16} \times \tau_1 \times d^3$$

$$l = \frac{\pi}{8} \times \frac{\tau_1 d^2}{w \times \tau} = \frac{\pi d}{2} \times \frac{\tau_1}{\tau} = 1.571\, d \times \frac{\tau_1}{\tau}$$

Revision Questions

1. What are the forces acting on keys

LECTURE 3

Design of cotter joints

Cotter Joints:

Cotter **joint** is widely used to connect the piston rod between the piston rod and the tailor pump rod, foundation bolt etc. ... A cotter is a wedge and crosshead of a steam engine, as a **joint**shaped piece made of a steel plate. It has uniform thickness and the width dimension is given a slight taper.

Cotter joint is a flat wedge shaped piece of steel. ... **cotter** is used to connect rigidly two rods which transmit motion in the axial direction, without rotation. • These **joints** may be subjected to tensile or compressive forces along the axes of the rods.

Types of Cotter Joints:

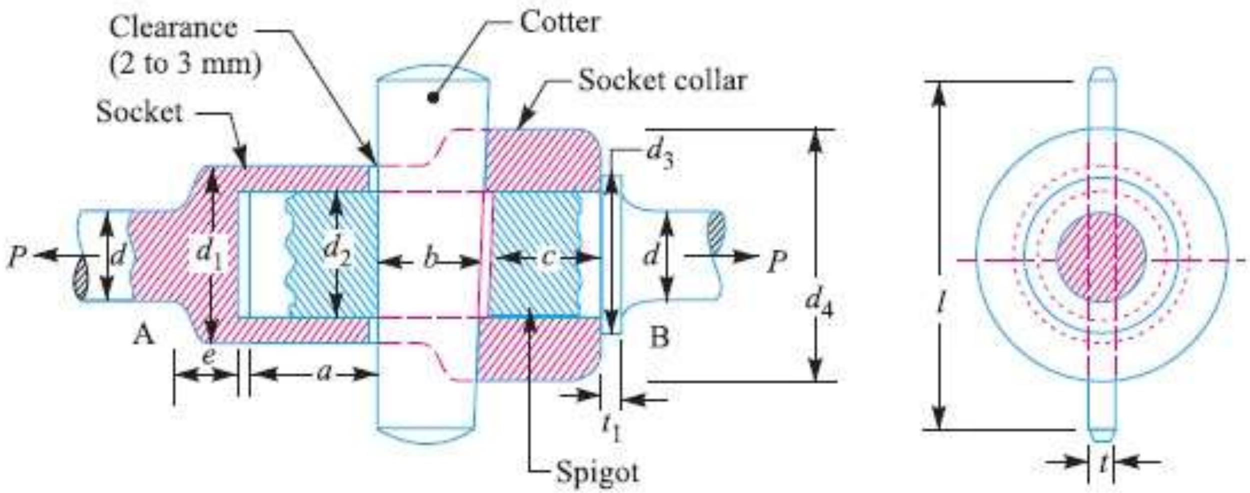
1. Socket and spigot cotter joint

1. Sleeve and cotter joint

1. Gib and cotter joint

Design of Socket and spigot joint:

A **spigot joint** is a type of pipe fitting connection that is inserted into another pipe fitting. The **spigot** end typically has the same outer diameter as the pipe and is usually fitted into another **joint** called a bell or **socket**. Together, these two elements form what is commonly known as a bell and **spigot joint**.



The socket and spigot cotter joint is shown in Fig. Let

P = Load carried by the rods,

d = Diameter of the rods,

d_1 = Outside diameter of socket,

d_2 = Diameter of spigot or inside diameter of socket, d_3 = Outside diameter of spigot collar,

t_1= Thickness of spigot collar, d_4 = Diameter of socket collar, c = Thickness of socket collar, b = Mean width of cotter,

t = Thickness of cotter, l = Length of cotter,

a = Distance from the end of the slot to the end of rod, σ_t = Permissible tensile stress for the rods material,

τ = Permissible shear stress for the cotter material, and σ_c = Permissible crushing stress for the cotter material.

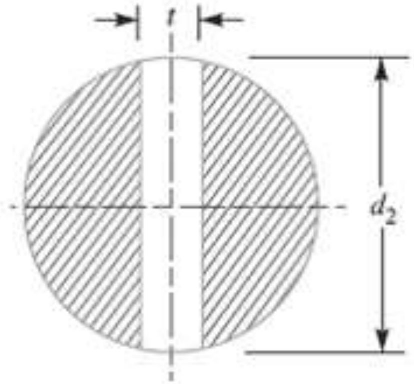
The dimensions for a socket and spigot cotter joint may be obtained by considering the various modes of failure as discussed below:

1. Failure of the rods in tension

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

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

From this equation, the diameter of spigot or inside diameter of socket d_2 may be determined. In actual practice, the thickness of cotter is usually taken as d_2 / 4.



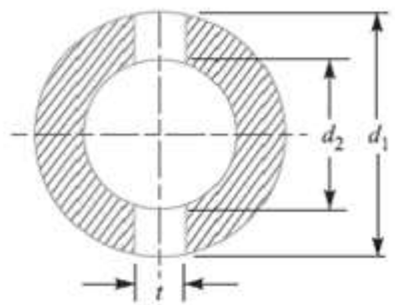
$$P = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t$$

3. Failure of the rod or cotter in crushing

$$P = d_2 \times t \times \sigma_c$$

From this equation , the induced crushing stress may be checked

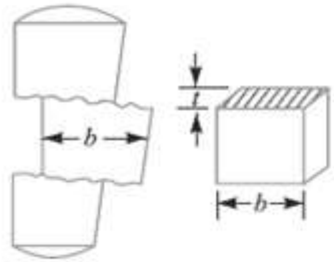
4. Failure of the socket in tension across the slot



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

From this equation, outside diameter of socket (d_1) may be determined

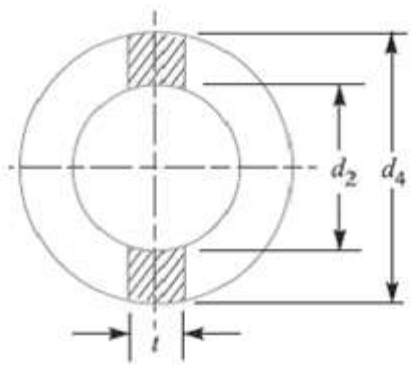
5. Failure of cotter in shear



$$P = 2b \times t \times \tau$$

From this equation, width of cotter (b) is determined.

6. Failure of the socket collar in crushing



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

From this equation, the diameter of socket collar (d_4) may be obtained.

7.Failure of socket end in shearing

$$P = 2\,(d_4 - d_2)\,c \times \tau$$

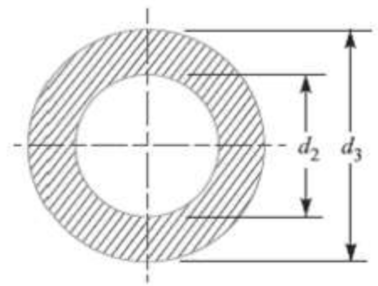
From this equation, the thickness of socket collar (c) may be obtained.

8. Failure of rod end in shear

$$P = 2\,a \times d_2 \times \tau$$

From this equation, the distance trom the end of the slot to the end of the rod (a) may be obtained.

9. Failure of spigot collar in crushing

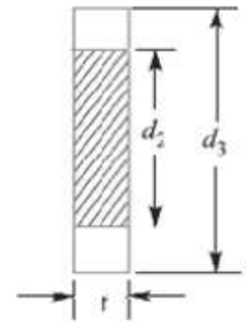


$$P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$$

From this equation, the diameter of the spigot collar (d_3) may be obtained.

10. Failure of the spigot collar in shearing

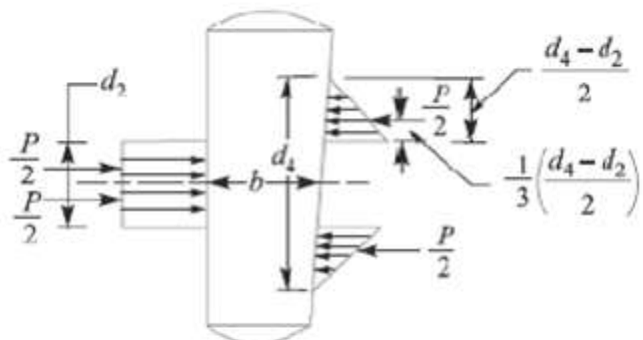
From this equation, the thickness of spigot collar (t_1) may be obtained.



$$P = \pi\, d_2 \times t_1 \times \tau$$

11. Failure of cotter in bending

The maximum bending moment occurs at the centre of the cotter and is given by



$$M_{max} = \frac{P}{2}\left(\frac{1}{3}\times\frac{d_4 - d_2}{2} + \frac{d_2}{2}\right) - \frac{P}{2}\times\frac{d_2}{4}$$

$$= \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{2} - \frac{d_2}{4}\right) = \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)$$

We know that section modulus of the cotter,

$$Z = t \times b^2 / 6$$

Bending stress induced in the cotter,

$$\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)}{t \times b^2 / 6} = \frac{P\,(d_4 + 0.5\,d_2)}{2\,t \times b^2}$$

- This bending stress induced in the cotter should be less than the allowable bending stress of the cotter.
- The length of cotter (l) in taken as 4 d.
- The taper in cotter should not exceed 1 in 24. In case the greater taper is required, then a locking device must be provided.
- The draw of cotter is generally taken as 2 to 3 mm.

GIB AND COTTER JOINT:

This joint is generally used to connect two rods of square or rectangular section. To make the joint; one end of the rod is formed into a U-fork, into which, the end of the other rod fits-in. When a cotter is driven-in, the friction between the cotter and straps of the U-fork, causes the straps open. This is prevented by the use of a gib.

Let *F* be the maximum tensile or compressive force in the connecting rod, and

b = width of the strap, which may be taken as equal to the diameter of the rod. *d*

h = height of the rod end

t_1 = thickness of the strap at the thinnest part

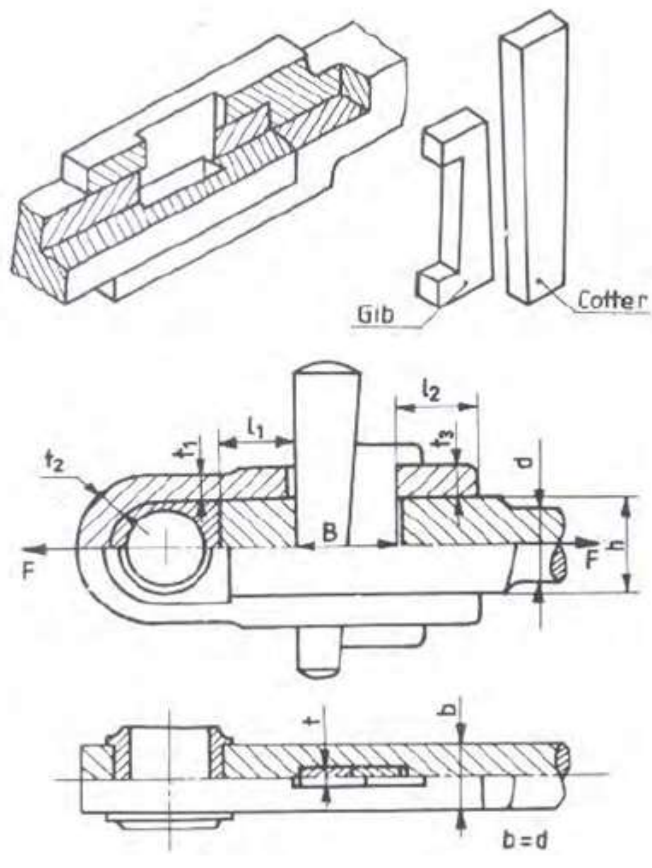
t_2 = thickness of the strap at the curved portion t_3 =thickness of the strap across the slot

l_1 = length of the rod end, beyond the slot 1_2 = length of the strap, beyond the slot

B = width of the cotter and gib

t = thickness of the cotter

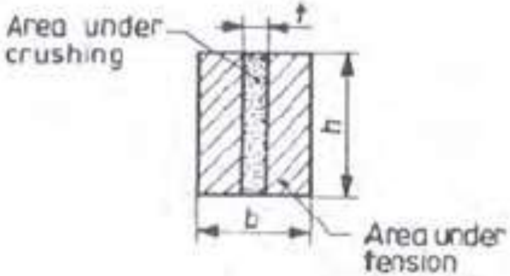
- Let the rod, strap, cotter, and gib are made of the same material with σ_c' σ_t' and τ :as the permissible stresses. The following are the possible modes of failure, and the corresponding design equations, which may be considered for the design of the joint:



1. Tension failure of the rod across the section of diameter (d)

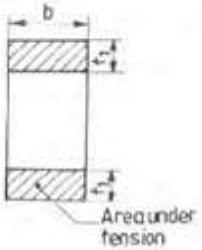
$$F = \frac{\pi d^2}{4} \times \sigma_t$$

2. Tension failure of the rod across the slot



$$F = (bh - ht)\,\sigma_t$$

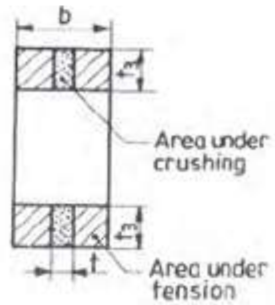
3. Tension failure of the strap, across the thinnest part



$$F = 2b\,t_1\,\sigma_t$$

4. Tension failure of the strap across the slot

The thickness, t2 may be taken
as (1.15 to 1.5) t], and Thickness
of the cotter, t = b/4.



$$F = 2\,b\,t_2 - 2\,t\,t_3 = 2t_3(b - t)\,\sigma_t$$

5. Crushing between the rod and cotter

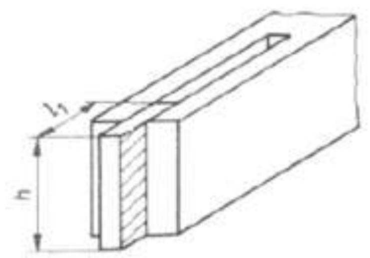
$$F = h\,t\,\sigma_c\ ;\ \text{and } h = 2t_3$$

6. Crushing between the strap and gib

$$F = 2\,t\,t_3\,\sigma_c$$

7. Shear failure of the rod end.

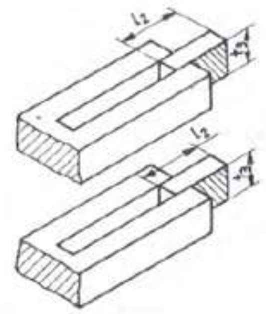
It is under double shear



$$F = 2l_1h\tau$$

8. Shear failure of the strap end.

It is under double shear



$$F = 4\, l_2\, t_3\tau$$

9. Shear failure of the cotter and gib. It is under double shear.

$$F=2Bt\tau$$

The following proportions for the widths of the cotter and gib may be followed: Width of the cotter =0.45 B Width of the gib = 0.55 B

KNUCKLE JOINT:

The following figure shows a knuckle joint with the size parameters and proportions indicated. In general, the rods connected by this joint are subjected to tensile loads, although if the rods are guided, they may support compressive loads as well.

Let F = tensile load to be resisted by the joint

d = diameter of the rods

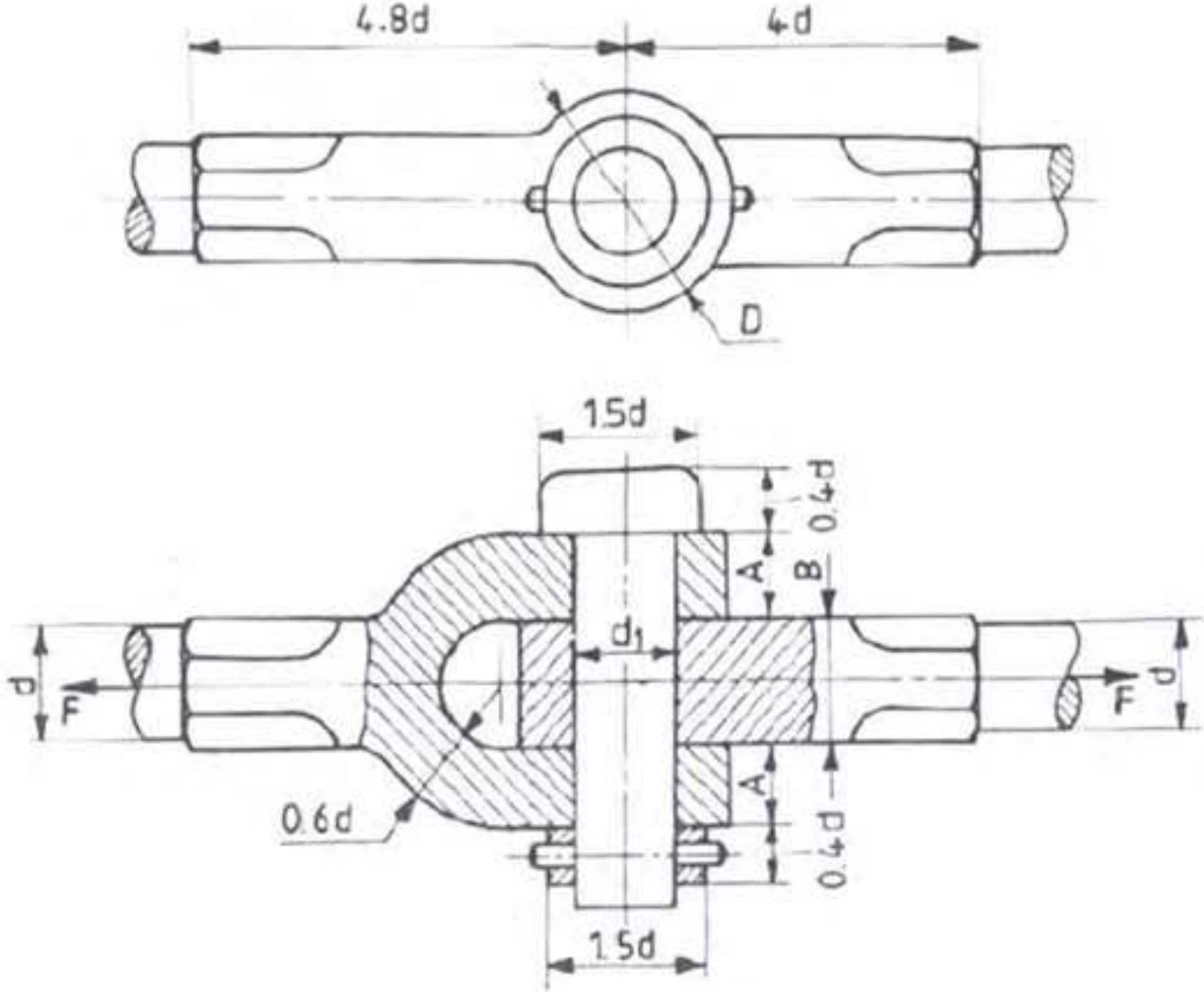
d_1 = diameter of the knuckle pin

D = outside diameter of the eye

A =thickness of the fork

B =thickness of the eye

Let the rods and pin are made of the same material, with σ_t, σ_c and τ as the permissible stresses. The following are the possible modes of failure, and the corresponding design equations, which may be considered for the design of the joint:

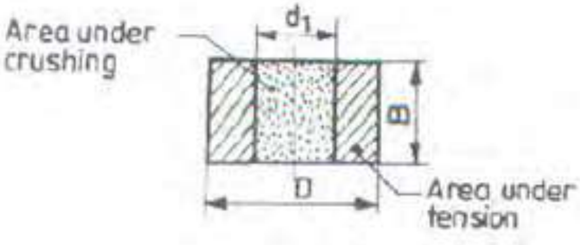


4.8d
4d
D
1.5d
0.4d
A
B
d
d_1
F
0.6d
1.5d

1. Tension failure of the rod, across the section of diameter, *d*

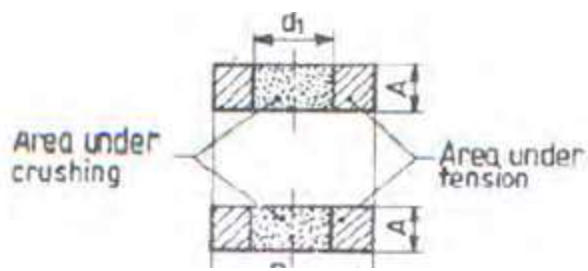
$$F = \frac{\pi d^2}{4} \times \sigma_t$$

2. Tension failure of the eye



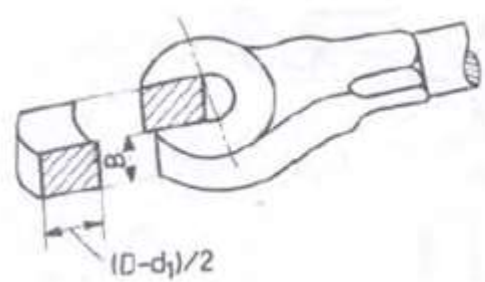
$$F = (D-d_1)\ B\ \sigma_t$$

3. Tension failure of the fork



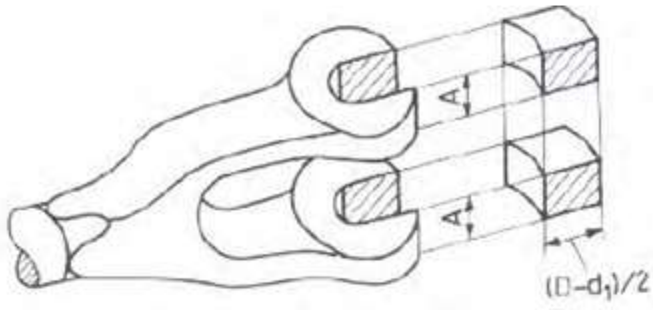
$$F = 2\ (D - d_1)\ A\ \sigma_t$$

4. Shear failure of the eye



$$F = (D-d_1)\ B\ \tau$$

5. Shear failure of the fork



$$F = 2\,(D-d_1)\,A\,\tau$$

6. Shear failure of the pin. It is under double shear.

$$F = 2 \times \frac{\pi}{4} \times (d_1)^2\,\tau$$

7. Crushing between the pin and eye

$$F = d_1\,B\,\sigma_c$$

8. Crushing between the pin and fork

$$F = 2\,d_1\,A\,\sigma_c$$

For size parameters, not covered by the above design equations; proportions as indicated in the figure may be followed.

Problem:

Design a knuckle joint to transmit 150 kN. The design stresses may be taken as 75 MPa in tension, 60 MPa in shear and 150 MPa in compression.

Solution. Given : $P = 150\text{ kN} = 150 \times 10^3\text{ N}$; $\sigma_t = 75\text{ MPa} = 75\text{ N/mm}^2$; $\tau = 60\text{ MPa} = 60\text{ N/mm}^2$; $\sigma_c = 150\text{ MPa} = 150\text{ N/mm}^2$

1. *Failure of the solid rod in tension*

Let d = Diameter of the rod.

We know that the load transmitted (P),

$$150 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 75 = 59\, d^2$$

$\therefore$ $d^2 = 150 \times 10^3 / 59 = 2540$ or $d = 50.4$ say 52 mm **Ans.**

Now the various dimensions are fixed as follows :

Diameter of knuckle pin,

$$d_1 = d = 52\text{ mm}$$

Outer diameter of eye, $d_2 = 2\,d = 2 \times 52 = 104$ mm

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\,d = 1.5 \times 52 = 78\text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\,d = 1.25 \times 52 = 65\text{ mm}$$

Thickness of fork, $t_1 = 0.75\,d = 0.75 \times 52 = 39$ say 40 mm

Thickness of pin head, $t_2 = 0.5\,d = 0.5 \times 52 = 26$ mm

2. *Failure of the knuckle pin in shear*

Since the knuckle pin is in double shear, therefore load (P),

$$150 \times 10^3 = 2 \times \frac{\pi}{4} \times (d_1)^2\,\tau = 2 \times \frac{\pi}{4} \times (52)^2\,\tau = 4248\,\tau$$

$\therefore \quad \tau = 150 \times 10^3 / 4248 = 35.3 \text{ N/mm}^2 = 35.3 \text{ MPa}$

3. *Failure of the single eye or rod end in tension*

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \sigma_t = (104 - 52)\, 65 \times \sigma_t = 3380\, \sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

4. *Failure of the single eye or rod end in shearing*

The single eye or rod end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \tau = (104 - 52)\, 65 \times \tau = 3380\, \tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

5. *Failure of the single eye or rod end in crushing*

The single eye or rod end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times t \times \sigma_c = 52 \times 65 \times \sigma_c = 3380\, \sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

6. *Failure of the forked end in tension*

The forked end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \sigma_t = (104 - 52)\, 2 \times 40 \times \sigma_t = 4160\, \sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

7. *Failure of the forked end in shear*

The forked end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \tau = (104 - 52)\, 2 \times 40 \times \tau = 4160\, \tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

8. *Failure of the forked end in crushing*

The forked end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times 2\,t_1 \times \sigma_c = 52 \times 2 \times 40 \times \sigma_c = 4160\,\sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 4180 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

From above, we see that the induced stresses are less than the given design stresses, therefore the joint is safe.

Problem:

Design and draw a cotter joint to support a load varying from 30 kN in compression to 30 kN in tension. The material used is carbon steel for which the following allowable stresses may be used. The load is applied sta ically. Tensile stress = compressive stress = 50 MPa ; shear stress = 35 MPa and crushing stress = 90 MPa.

Solution. Given : $P = 30 \text{ kN} = 30 \times 10^3 \text{ N}$; $\sigma_t = 50 \text{ MPa} = 50 \text{ N/mm}^2$; $\tau = 35 \text{ MPa} = 35 \text{ N/mm}^2$; $\sigma_c = 90 \text{ MPa} = 90 \text{ N/mm}^2$

1. *Diameter of the rods*

Let d = Diameter of the rods.

Considering the failure of the rod in tension. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 50 = 39.3\,d^2$$

$$\therefore \quad d^2 = 30 \times 10^3 / 39.3 = 763 \quad \text{or} \quad d = 27.6 \text{ say } 28 \text{ mm} \quad \textbf{Ans.}$$

2. *Diameter of spigot and thickness of cotter*

Let d_2 = Diameter of spigot or inside diameter of socket, and

t = Thickness of cotter. It may be taken as $d_2 / 4$.

Considering the failure of spigot in tension across the weakest section. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]50 = 26.8\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 30 \times 10^3 / 26.8 = 1119.4 \quad \text{or} \quad d_2 = 33.4 \text{ say } 34 \text{ mm}$$

and thickness of cotter, $t = \dfrac{d_2}{4} = \dfrac{34}{4} = 8.5$ mm

Let us now check the induced crushing stress. We know that load (P),

$$30 \times 10^3 = d_2 \times t \times \sigma_c = 34 \times 8.5 \times \sigma_c = 289\,\sigma_c$$

$$\therefore \quad \sigma_c = 30 \times 10^3 / 289 = 103.8 \text{ N/mm}^2$$

Since this value of σ_c is more than the given value of $\sigma_c = 90$ N/mm^2, therefore the dimensions $d_2 = 34$ mm and $t = 8.5$ mm are not safe. Now let us find the values of d_2 and t by substituting the value of $\sigma_c = 90$ N/mm^2 in the above expression, *i.e.*

$$30 \times 10^3 = d_2 \times \frac{d_2}{4} \times 90 = 22.5\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 30 \times 10^3 / 22.5 = 1333 \quad \text{or} \quad d_2 = 36.5 \text{ say } 40 \text{ mm} \quad \textbf{Ans.}$$

and $\quad t = d_2 / 4 = 40 / 4 = 10$ mm **Ans.**

3. *Outside diameter of socket*

Let d_1 = Outside diameter of socket.

Considering the failure of the socket in tension across the slot. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}\left\{(d_1)^2 - (d_2)^2\right\} - (d_1 - d_2)\,t\right]\sigma_t$$

$$= \left[\frac{\pi}{4}\left\{(d_1)^2 - (40)^2\right\} - (d_1 - 40)\,10\right]50$$

$$30 \times 10^3/50 = 0.7854\,(d_1)^2 - 1256.6 - 10\,d_1 + 400$$

or $(d_1)^2 - 12.7\,d_1 - 1854.6 = 0$

$$\therefore \quad d_1 = \frac{12.7 \pm \sqrt{(12.7)^2 + 4 \times 1854.6}}{2} = \frac{12.7 \pm 87.1}{2}$$

$$= 49.9 \text{ say } 50 \text{ mm Ans.} \qquad \text{...(Taking +ve sign)}$$

4. *Width of cotter*

Let b = Width of cotter.

Considering the failure of the cotter in shear. Since the cotter is in double shear, therefore load (P),

$$30 \times 10^3 = 2\,b \times t \times \tau = 2\,b \times 10 \times 35 = 700\,b$$

$$\therefore \quad b = 30 \times 10^3 / 700 = 43 \text{ mm Ans.}$$

5. *Diameter of socket collar*

Let d_4 = Diameter of socket collar.

Considering the failure of the socket collar and cotter in crushing. We know that load (P),

$$30 \times 10^3 = (d_4 - d_2)\,t \times \sigma_c = (d_4 - 40)10 \times 90 = (d_4 - 40)\,900$$

$$\therefore \quad d_4 - 40 = 30 \times 10^3 / 900 = 33.3 \text{ or } d_4 = 33.3 + 40 = 73.3 \text{ say } 75 \text{ mm Ans.}$$

6. *Thickness of socket collar*

Let c = Thickness of socket collar.

Considering the failure of the socket end in shearing. Since the socket end is in double shear, therefore load (P),

$$30 \times 10^3 = 2(d_4 - d_2)\,c \times \tau = 2\,(75 - 40)\,c \times 35 = 2450\,c$$

$$\therefore \quad c = 30 \times 10^3 / 2450 = 12 \text{ mm Ans.}$$

7. *Distance from the end of the slot to the end of the rod*

Let a = Distance from the end of slot to the end of the rod.

Considering the failure of the rod end in shear. Since the rod end is in double shear, therefore load (P),

$$30 \times 10^3 = 2\,a \times d_2 \times \tau = 2a \times 40 \times 35 = 2800\,a$$

$$\therefore \quad a = 30 \times 10^3 / 2800 = 10.7 \text{ say } 11 \text{ mm Ans.}$$

8. *Diameter of spigot collar*

Let d_3 = Diameter of spigot collar.

Considering the failure of spigot collar in crushing. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c = \frac{\pi}{4}\left[(d_3)^2 - (40)^2\right]90$$

or
$$(d_3)^2 - (40)^2 = \frac{30 \times 10^3 \times 4}{90 \times \pi} = 424$$

$\therefore$
$$(d_3)^2 = 424 + (40)^2 = 2024 \quad \text{or} \quad d_3 = 45 \text{ mm } \textbf{Ans.}$$

9. *Thickness of spigot collar*

Let t_1 = Thickness of spigot collar.

Considering the failure of spigot collar in shearing. We know that load (P),

$$30 \times 10^3 = \pi\, d_2 \times t_1 \times \tau = \pi \times 40 \times t_1 \times 35 = 4400\, t_1$$

$\therefore$
$$t_1 = 30 \times 10^3 / 4400 = 6.8 \text{ say } 8 \text{ mm } \textbf{Ans.}$$

10. The length of cotter (l) is taken as 4 d.

$\therefore$
$$l = 4\,d = 4 \times 28 = 112 \text{ mm } \textbf{Ans.}$$

11. The dimension e is taken as 1.2 d.

$\therefore$
$$e = 1.2 \times 28 = 33.6 \text{ say } 34 \text{ mm } \textbf{Ans.}$$

Problem:

Design a cotter joint to connect piston rod to the crosshead of a double acting steam engine. The diameter of the cylinder is 300 mm and the steam pressure is 1 N/mm^2. The allowable stresses for the material of cotter and piston rod are as follows: σ_t = 50 MPa ; τ = 40 MPa ; and σ_c = 84 MPa

Solution. Given : $D = 300$ mm ; $p = 1$ N/mm^2 ; $\sigma_t = 50$ MPa $= 50$ N/mm^2 ; $\tau = 40$ MPa $= 40$ N/mm^2 ; $\sigma_c = 84$ MPa $= 84$ N/mm^2

We know that maximum load on the piston rod,

$$P = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4}(300)^2\, 1 = 70\ 695 \text{ N}$$

The various dimensions for the cotter joint are obtained by considering the different modes of failure as discussed below :

1. *Diameter of piston rod at cotter*

Let d_2 = Diameter of piston rod at cotter, and

t = Thickness of cotter. It may be taken as $0.3\, d_2$.

Considering the failure of piston rod in tension at cotter. We know that load (P),

$$70\ 695 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - 0.3\,(d_2)^2\right] 50 = 24.27\,(d_2)^2$$

$\therefore$ $(d_2)^2 = 70\ 695 / 24.27 = 2913$ or $d_2 = 53.97$ say 55 mm **Ans.**

and $t = 0.3\, d_2 = 0.3 \times 55 = 16.5$ mm **Ans.**

2. *Width of cotter*

Let b = Width of cotter.

Considering the failure of cotter in shear. Since the cotter is in double shear, therefore load (P),

$$70\ 695 = 2\, b \times t \times \tau = 2\, b \times 16.5 \times 40 = 1320\, b$$

$\therefore$ $b = 70\ 695 / 1320 = 53.5$ say 54 mm **Ans.**

3. Diameter of socket

Let $\quad d_3$ = Diameter of socket.

Considering the failure of socket in tension at cotter. We know that load (P),

$$70\ 695 = \left\{\frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] - (d_3 - d_2)\ t\right\}\sigma_1$$

$$= \left\{\frac{\pi}{4}\left[(d_3)^2 - (55)^2\right] - (d_3 - 55)\ 16.5\right\}50$$

$$= 39.27\,(d_3)^2 - 118\ 792 - 825\ d_3 + 45\ 375$$

or $\quad (d_3)^2 - 21\ d_3 - 3670 = 0$

$$\therefore \quad d_3 = \frac{21 + \sqrt{(21)^2 + 4 \times 3670}}{2} = \frac{21 \pm 123}{2} = 72 \text{ mm} \quad \text{...(Taking + ve sign)}$$

Let us now check the induced crushing stress in the socket. We know that load (P),

$$70\ 695 = (d_3 - d_2)\,t \times \sigma_c = (72 - 55)\ 16.5 \times \sigma_c = 280.5\,\sigma_c$$

$$\therefore \quad \sigma_c = 70695/280.5 = 252 \text{ N/mm}^2$$

Since the induced crushing is greater than the permissible value of 84 N/mm^2, therefore let us find the value of d_3 by substituting σ_c = 84 N/mm^2 in the above expression, *i.e.*

$$70\ 695 = (d_3 - 55)\ 16.5 \times 84 = (d_3 - 55)\ 1386$$

$$\therefore \quad d_3 - 55 = 70\ 695/1386 = 51$$

or $\quad d_3 = 55 + 51 = 106$ mm **Ans.**

We know the tapered length of the piston rod,

$$L = 2.2\,d_2 = 2.2 \times 55 = 121 \text{ mm} \quad \textbf{Ans.}$$

Assuming the taper of the piston rod as 1 in 20, therefore the diameter of the parallel part of the piston rod,

$$d = d_2 + \frac{L}{2} \times \frac{1}{20} = 55 + \frac{121}{2} \times \frac{1}{20} = 58 \text{ mm} \quad \textbf{Ans.}$$

and diameter of the piston rod at the tapered end,

$$d_1 = d_2 - \frac{L}{2} \times \frac{1}{20} = 55 - \frac{121}{2} \times \frac{1}{20} = 52 \text{ mm} \quad \textbf{Ans.}$$

THANK YOU

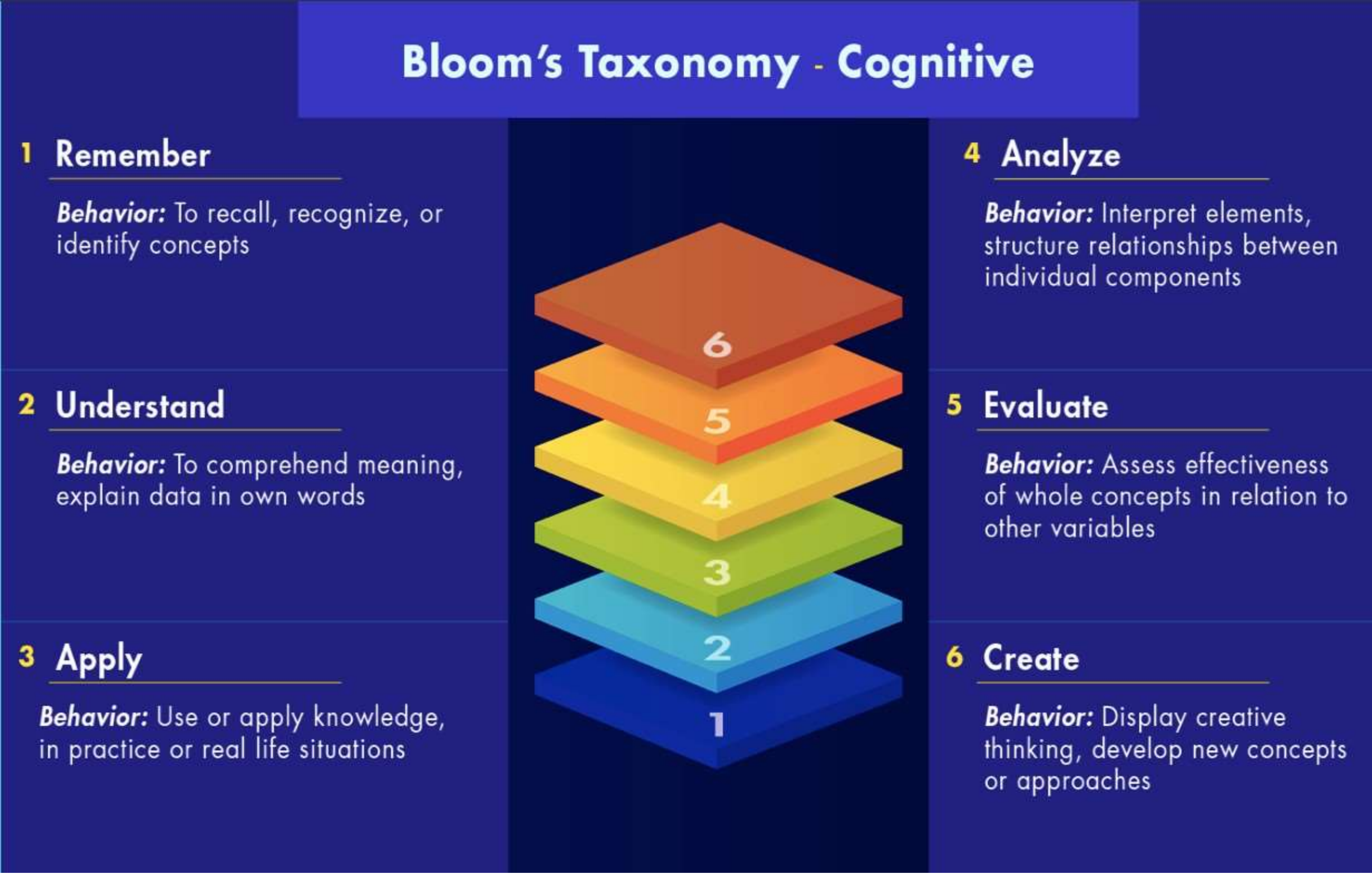
DIGITAL CLASSROOM ENVIRONMENT

A NEW PEDAGOGY IS EMERGING

MRCET CAMPUS

UGC Autonomous

C. Daksheeswara Reddy
Assistant Professor,
Mechanical Engineering



(Source: TASK Training material)

UNIT 5

CO5: To learn the design Procedure for the different Shafts under loading condition, able to know various shafts coupling.

UNIT – V (SYLLABUS)

Design of Shafts

- Design of solid and hollow shafts for strength and rigidity
- Design of shafts for combined bending and axial loads
- Shaft sizes

Design of Couplings

- Rigid couplings
- Muff, Split muff and Flange couplings
- Flexible couplings – Flange coupling (Modified)

COURSE OUTLINE

UNIT -5

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
1	**Introduction to shafts**		• **Understand the importance of shafts (B2)** • **Understand selection of materials for shaft (B2)**
2	Design of solid and hollow shafts	forces	• Understand difference between solid and hollow shafts(B2) • Understand forces acting solid and hollow shaft(B3)
3	Design of shafts for combined bending and axial loads	LOAD	• understand how to estimate shaft diameters for different segments along length(B4&B5) • Understand how design couplings for shafts. (B3)

LECTURE	LECTURE TOPIC	KEY ELEMENTS	Learning objectives (2 to 3 objectives)
4	**Introduction to couplings**		• Understand the importance of couplings(B2) •Understand selection of materials for couplings (B2)
5	**Types of couplings**	forces	• **Understand different types of coupling(B2)** • **Understand what type of forces acting on couplings(B3)**
6	Design of couplings	LOAD	• Understand how to design couplings for shafts.(B4&B5) • Understand which coupling is suitable for selected shafts (B5)

LECTURE 1

Introduction-Shafts

UNIT V
SHAFTS AND SHAFT COUPLINGS

Introduction to Shafts:

A shaft is a rotating machine element which is used to transmit power from one place to another. The power is delivered to the shaft by some tangential force and the resultant torque (or twisting moment) set up within the shaft permits the power to be transferred to various machines linked up to the shaft. In order to transfer the power from one shaft to another, the various members such as pulleys, gears etc., are mounted on it. These members along with the forces exerted upon them causes the shaft to bending.

Types of Shafts:

1. Transmission shafts
2. Machine shafts

Revision Questions

1. Define shaft
2. Classify the shafts

LECTURE 2

Design of solid and hallow Shafts

Stresses in Shafts

The following stresses are induced in the shafts:

1. Shear stresses due to the transmission of torque (*i.e.* due to torsional load).
2. Bending stresses (tensile or compressive) due to the forces acting upon machine elements like gears, pulleys etc. as well as due to the weight of the shaft itself.
3. Stresses due to combined torsional and bending loads.

Design of Shafts

The shafts may be designed on the basis of

1. Strength, and **2.** Rigidity and stiffness.

In designing shafts on the basis of strength, the following cases may be considered:

1. Shafts subjected to twisting moment or torque only,
2. Shafts subjected to bending moment only,
3. Shafts subjected to combined twisting and bending moments, and
4. Shafts subjected to axial loads in addition to combined torsional and bending loads.

Shafts Subjected to Twisting Moment Only

a) Solid shaft:

When the shaft is subjected to a twisting moment (or torque) only, then the diameter of the shaft may be obtained by using the torsion equation. We know that

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

Where T = Twisting moment (or torque) acting upon the shaft,

J = Polar moment of inertia of the shaft about the axis of rotation,

τ = Torsional shear stress, and

r = Distance from neutral axis to the outer most fibre

= d / 2; where d is the diameter of the shaft.

We know that for round solid shaft, polar moment of inertia,

$$J = \frac{\pi r^4}{2} = \frac{\pi d^4}{32}$$

then we get

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

b) Hollow Shaft:

We also know that for hollow shaft, polar moment of inertia,

$$J = \frac{\pi}{2}(R^4 - r^4) = \frac{\pi}{32}(D^4 - d^4)$$

Where d_o and d_i = Outside and inside diameter of the shaft, and $r = d_0 / 2$.

Substituting these values in equation **(*i*)**, we have

$$\frac{T}{\frac{\pi}{32}\left[(d_o)^4 - (d_i)^4\right]} = \frac{\tau}{\frac{d_o}{2}} \quad \text{or} \quad T = \frac{\pi}{16} \times \tau \left[\frac{(d_o)^4 - (d_i)^4}{d_o}\right]$$

Let k = Ratio of inside diameter and outside diameter of the shaft = d_i / d_o

- Now the equation **(*iii*)** may be written as

$$T = \frac{\pi}{16} \times \tau \times \frac{(d_o)^4}{d_o}\left[1 - \left(\frac{d_i}{d_o}\right)^4\right] = \frac{\pi}{16} \times \tau (d_o)^3 (1 - k^4)$$

1. The hollow shafts are usually used in marine work. These shafts are stronger per kg of material and they may be forged on a mandrel, thus making the material more homogeneous than would be possible for a solid shaft. When a hollow shaft is to be made equal in strength to a solid shaft, the twisting moment of both the shafts must be same. In other words, for the same material of both the shafts,

$$T = \frac{\pi}{16} \times \tau \left[\frac{(d_o)^4 - (d_i)^4}{d_o} \right] = \frac{\pi}{16} \times \tau \times d^3$$

$$\therefore \qquad \frac{(d_o)^4 - (d_i)^4}{d_o} = d^3 \quad \text{or} \quad (d_o)^3 (1 - k^4) = d^3$$

2. The twisting moment (T) may be obtained by using the following relation:
   We know that the power transmitted (in watts) by the shaft,

   Where T = Twisting moment in N-m, and

   N = Speed of the shaft in r.p.m

$$P = \frac{2\pi N \times T}{60} \quad \text{or} \quad T = \frac{P \times 60}{2\pi N}$$

3. In case of belt drives, the twisting moment (T) is given by

$$T = (T_1 - T_2)\, R$$

Where T_1 and T_2 = Tensions in the tight side and slack side of the belt respectively, and R = Radius of the pulley.

Shafts Subjected to Bending Moment Only

a) Solid Shaft:

When the shaft is subjected to a bending moment only, then the maximum stress (tensile or compressive) is given by the bending equation. We know that

$$\sigma_b = \frac{My}{I}$$

Where M = Bending moment,

I = Moment of inertia of cross-sectional area of the shaft about the axis of rotation, σ_b = Bending stress, and

y = Distance from neutral axis to the outer-most fibre.

We know that for a round solid shaft, moment of inertia,

$$I = \frac{\pi}{64} \times d^4 \quad \text{and} \quad y = \frac{d}{2}$$

Substituting these values in equation

$$\frac{M}{\frac{\pi}{64} \times d^4} = \frac{\sigma_b}{\frac{d}{2}} \quad \text{or} \quad M = \frac{\pi}{32} \times \sigma_b \times d^3$$

From this equation, diameter of the solid shaft (*d*) may be obtained.

b) Hollow Shaft:We also know that for a hollow shaft, moment of inertia, And $y = d_0/ 2$

$$I = \frac{\pi}{64}\left[(d_o)^4 - (d_i)^4\right] = \frac{\pi}{64}(d_o)^4 (1 - k^4) \quad \text{...(where } k = d_i / d_o)$$

Again substituting these values in equation, we have d_o determined.

$$\frac{M}{\frac{\pi}{64}(d_o)^4 (1 - k^4)} = \frac{\sigma_b}{\frac{d_o}{2}} \quad \text{or} \quad M = \frac{\pi}{32} \times \sigma_b (d_o)^3 (1 - k^4)$$

LECTURE 3

Shafts subjected to combined bending & twisting

Shafts Subjected to Combined Twisting Moment and Bending Moment

When the shaft is subjected to combined twisting moment and bending moment, then the shaft must be designed on the basis of the two moments simultaneously. Various theories have been suggested to account for the elastic failure of the materials when they are subjected to various types of combined stresses. The following two theories are important from the subject point of view:

1. Maximum shear stress theory or Guest's theory. It is used for ductile materials such as mild steel.
2. Maximum normal stress theory or Rankine's theory. It is used for brittle materials such as cast iron.

`Let τ = Shear stress induced due to twisting moment, and

` σ_b = Bending stress (tensile or compressive) induced due to bending moment.

Solid Shaft:

According to maximum shear stress theory, the maximum shear stress in the shaft,

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

Substituting the values of σ_b and τ

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

$$\text{or} \qquad \frac{\pi}{16} \times \tau_{max} \times d^3 = \sqrt{M^2 + T^2}$$

We know that equivalent twisting moment

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

Now according to maximum normal stress theory, the maximum normal stress in the shaft,

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

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

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

or $$\frac{\pi}{32} \times \sigma_{b\,(max)} \times d^3 = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right]$$

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

Hollow shaft:

In case of a hollow shaft, the equations **(*ii*)** and **(*v*)** may be written as

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times \tau (d_o)^3 (1 - k^4)$$

$$M_e = \frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right) = \frac{\pi}{32} \times \sigma_b (d_o)^3 (1 - k^4)$$

Problem:

A steel spindle transmits 4 kW at 800 r.p.m. The angular deflection should not exceed 0.25° per metre of the spindle. If the modulus of rigidity for the material of the spindle is 84 GPa, find the diameter of the spindle and the shear stress induced in the spindle.

Solution. Given : $P = 4$ kW $= 4000$ W ; $N = 800$ r.p.m. ; $\theta = 0.25° = 0.25 \times \frac{\pi}{180} = 0.0044$ rad ; $L = 1$ m $= 1000$ mm ; $G = 84$ GPa $= 84 \times 10^9$ N/m^2 $= 84 \times 10^3$ N/mm^2

Diameter of the spindle

Let d = Diameter of the spindle in mm.

We know that the torque transmitted by the spindle,

$$T = \frac{P \times 60}{2\pi N} = \frac{4000 \times 60}{2\pi \times 800} = 47.74 \text{ N-m} = 47\ 740 \text{ N-mm}$$

We also know that $\frac{T}{J} = \frac{G \times \theta}{L}$ or $J = \frac{T \times l}{G \times \theta}$

or $$\frac{\pi}{32} \times d^4 = \frac{47\ 740 \times 1000}{84 \times 10^3 \times 0.0044} = 129\ 167$$

$\therefore$ $$d^4 = 129\ 167 \times 32 / \pi = 1.3 \times 10^6 \quad \text{or} \quad d = 33.87 \text{ say } 35 \text{ mm } \textbf{Ans.}$$

Shear stress induced in the spindle

Let τ = Shear stress induced in the spindle.

We know that the torque transmitted by the spindle (T),

$$47\ 740 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times \tau\ (35)^3 = 8420\ \tau$$

$\therefore$ $$\tau = 47\ 740 / 8420 = 5.67 \text{ N/mm}^2 = 5.67 \text{ MPa } \textbf{Ans.}$$

LECTURE 4

Introduction to couplings

Shaft Coupling:

Shafts are usually available up to 7 meters length due to inconvenience in transport. In order to have a greater length, it becomes necessary to join two or more pieces of the shaft by means of a coupling.

Advantages:

- Shaft couplings are used in machinery for several purposes, the most common of which are the following:
- To provide for the connection of shafts of units those are manufactured separately such as a motor and generator and to provide for disconnection for repairs or alternations.
- To provide for misalignment of the shafts or to introduce mechanical flexibility.
- To reduce the transmission of shock loads from one shaft to another.
- To introduce protection against overloads.
- It should have no projecting parts.

Revision Questions

1. What is the function of coupling
2. Classify the couplings

LECTURE 5

Types of couplings

Types of Shafts Couplings:

Rigid coupling It is used to connect two shafts which are perfectly aligned. Following types of rigid coupling are important from the subject point of view:

I. Sleeve or muff coupling.
II. Clamp or split-muff or compression coupling, and
III. Flange coupling.

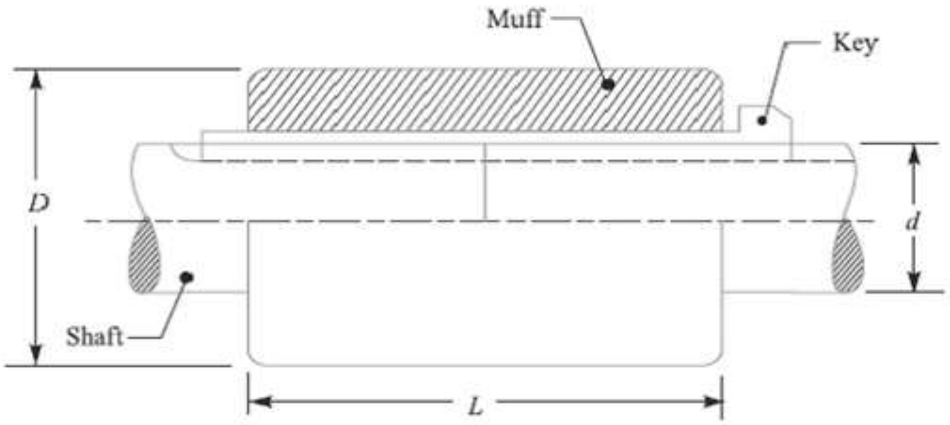
Flexible coupling**.** It is used to connect two shafts having both lateral and angular misalignment. Following types of flexible coupling are important from the subject point of view:

i. Bushed pin type coupling,
ii. Universal coupling, and
iii. Oldham coupling.

LECTURE 6

Design of couplings

Sleeve or Muff-coupling:



1. Design for sleeve

The sleeve is designed by considering it as a hollow shaft

Let T = Torque to be transmitted by the coupling, and

τ_c = Permissible shear stress for the material of the sleeve which is cast iron. The safe value of shear stress for cast iron may be taken as 14 MPa. We know that torque transmitted by a hollow section, From this expression, the induced shear stress in the sleeve may be checked.

$$T = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D} \right) = \frac{\pi}{16} \times \tau_c \times D^3 \, (1 - k^4) \qquad \ldots (\because k = d/D)$$

2. Design for key

The key for the coupling may be designed in the similar way as discussed in Unit-5. The width and thickness of the coupling key is obtained from the proportions. The length of the coupling key is at least equal to the length of the sleeve (i.e. 3.5 d). The coupling key is usually made into two parts so that the length of the key in each shaft,

$$l = \frac{L}{2} = \frac{3.5\,d}{2}$$

After fixing the length of key in each shaft, the induced shearing and crushing stresses may be checked. We know that torque transmitted,

$$T = l \times w \times \tau \times \frac{d}{2} \quad \text{... (Considering shearing of the key)}$$

$$= l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2} \quad \text{... (Considering crushing of the key)}$$

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

Solution.

Given: P = 40 kW = 40 × 10^3 W; N = 350 r.p.m.; τ_s = 40 MPa = 40 N/mm2; σ_{cs} = 80 MPa = 80 N/mm^2; σ_c = 15 MPa = 15 N/mm^2.

$$T = \frac{P \times 60}{2\,\pi N} = \frac{40 \times 10^3 \times 60}{2\,\pi \times 350} = 1100 \text{ N-m}$$

$$= 1100 \times 10^3 \text{ N-mm}$$

We also know that the torque transmitted (T),

$$1100 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\, d^3$$

$$\therefore \quad d^3 = 1100 \times 10^3/7.86 = 140 \times 10^3 \text{ or } d = 52 \text{ say } 55 \text{ mm } \textbf{Ans.}$$

2.Design for sleeve

We know that outer diameter of the muff,

D = 2d + 13 mm = 2 × 55 + 13 = 123 say 125 mm **Ans.**

and length of the muff,

L = 3.5 d = 3.5 × 55 = 192.5 say 195 mm **Ans.**

Let us now check the induced shear stress in the muff. Let τ_c be the induced shear stress in the muff which is made of cast iron. Since the muff is considered to be a hollow shaft, therefore the torque transmitted (T),

$$1100 \times 10^3 = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D} \right) = \frac{\pi}{16} \times \tau_c \left[\frac{(125)^4 - (55)^4}{125} \right]$$

$$= 370 \times 103\ \tau_c$$

$$\therefore \quad \tau_c = 1100 \times 10^3 / 370 \times 10^3 = 2.97 \text{ N/mm}^2$$

Since the induced shear stress in the muff (cast iron) is less than the permissible shear stress of 15 N/mm2, therefore the design of muff is safe.

3.Design for key

From Design data Book, we find that for a shaft of 55 mm diameter,

Width of key, w = 18 mm **Ans.**

Since the crushing stress for the key material is twice the shearing stress, therefore a square key may be used.

Then, Thickness of key, t = w = 18 mm **Ans.**

We know that length of key in each shaft,

l = L / 2 = 195 / 2 = 97.5 mm **Ans.**

Let us now check the induced shear and crushing stresses in the key. First of all, let us consider shearing of the key. We know that torque transmitted (T),

$$1100 \times 10^3 = l \times w \times \tau_s \times \frac{d}{2} = 97.5 \times 18 \times \tau_s \times \frac{55}{2} = 48.2 \times 10^3 \, \tau_s$$

$$\tau_s = 1100 \times 10^3 / 48.2 \times 10^3 = 22.8 \text{ N/mm}^2$$

Now considering crushing of the key. We know that torque transmitted (*T*),

Since the induced shear and crushing stresses are less than the permissible stresses, therefore the design of key is safe.

$$1100 \times 10^3 = l \times \frac{t}{2} \times \sigma_{cs} \times \frac{d}{2} = 97.5 \times \frac{18}{2} \times \sigma_{cs} \times \frac{55}{2} = 24.1 \times 10^3 \, \sigma_{cs}$$

$$\sigma_{cs} - 1100 \times 10^3 / 24.1 \times 10^3 - 45.6 \text{ N/mm}^2$$

Clamp or Compression Coupling or split muff coupling

It is also known as **split muff coupling.** In this case, the muff or sleeve is made into two halves and are bolted together as shown in Fig. The halves of the muff are made of cast iron. The shaft ends are made to a butt each other and a single key is fitted directly in the keyways of both the shafts. One-half of the muff is fixed from below and the other half is placed from above. Both the halves are held together by means of mild steel studs or bolts and nuts. The number of bolts may be two, four or six. The nuts are recessed into the bodies of the muff castings. This coupling may be used for heavy duty and moderate speeds.

In the clamp or compression coupling, the power is transmitted from one shaft to the other by means of key and the friction between the muff and shaft. In designing this type of coupling, the following procedure may be adopted.

1. Design of muff and key

The muff and key are designed in the similar way as discussed in muff coupling.

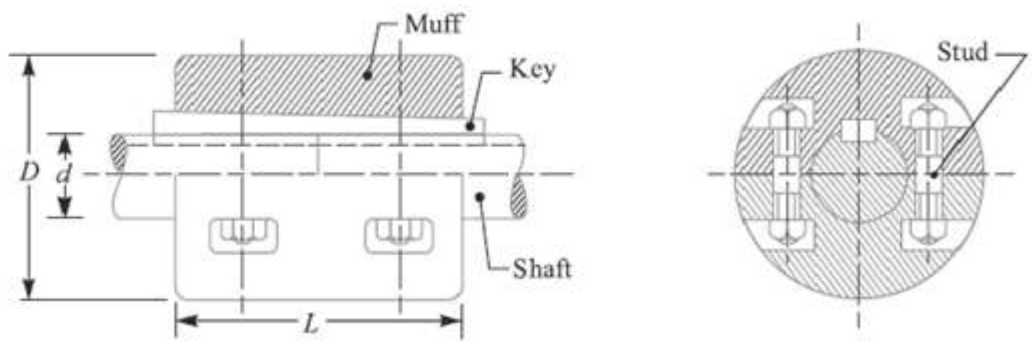
2. Design of clamping bolts

Let T = Torque transmitted by the shaft, d = Diameter of shaft,

d_b = Root or effective diameter of bolt, n = Number of bolts,

σ_t = Permissible tensile stress for bolt material,

μ = Coefficient of friction between the muff and shaft, and L = Length of muff.



We know that the force exerted by each bolt

$$= \frac{\pi}{4} (d_b)^2 \sigma_t$$

Then, Force exerted by the bolts on each side of the shaft

$$= \frac{\pi}{4} (d_b)^2 \sigma_t \times \frac{n}{2}$$

Let p be the pressure on the shaft and the muff surface due to the force, then for uniform pressure distribution over the surface,

$$p = \frac{\text{Force}}{\text{Projected area}} = \frac{\frac{\pi}{4} (d_b)^2 \sigma_t \times \frac{n}{2}}{\frac{1}{2} L \times d}$$

Then, Frictional force between each shaft and muff,

$$F = \mu \times \text{pressure} \times \text{area} = \mu \times p \times \frac{1}{2} \times \pi d \times L$$

$$= \mu \times \frac{\frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2}}{\frac{1}{2} L \times d} \times \frac{1}{2} \pi d \times L$$

$$= \mu \times \frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2} \times \pi = \mu \times \frac{\pi^2}{8}(d_b)^2 \sigma_t \times n$$

And the torque that can be transmitted by the coupling,

$$T = F \times \frac{d}{2} = \mu \times \frac{\pi^2}{8}(d_b)^2 \sigma_t \times n \times \frac{d}{2} = \frac{\pi^2}{16} \times \mu\, (d_b)^2 \sigma_t \times n \times d$$

From this relation, the root diameter of the bolt (d_b) may be evaluated.

Flange Coupling

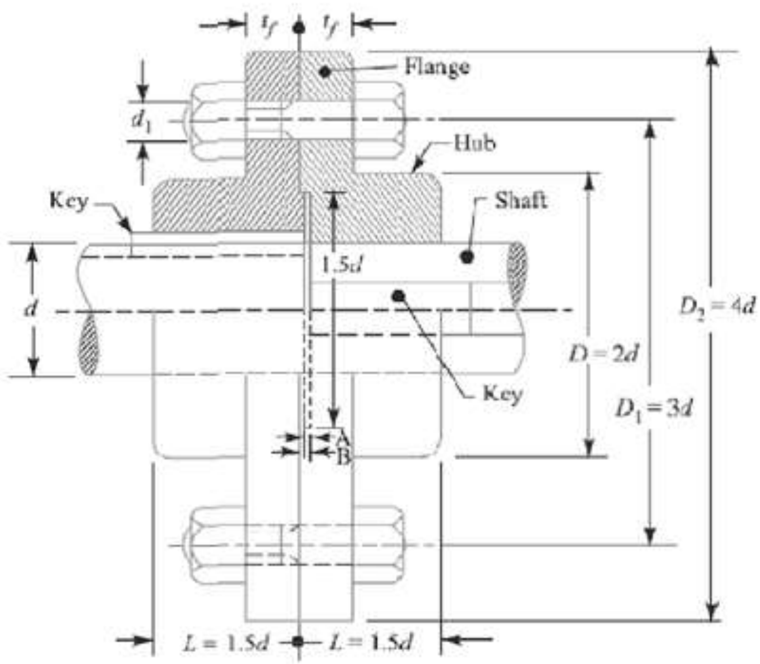
A flange coupling usually applies to a coupling having two separate cast iron flanges. Each flange is mounted on the shaft end and keyed to it. The faces are turned up at right angle to the axis of the shaft. One of the flanges has a projected portion and the other flange has a corresponding recess. This helps to bring the shafts into line and to maintain alignment. The two flanges are coupled together by means of bolts and nuts. The flange coupling is adapted to heavy loads and hence it is used on large shafting.

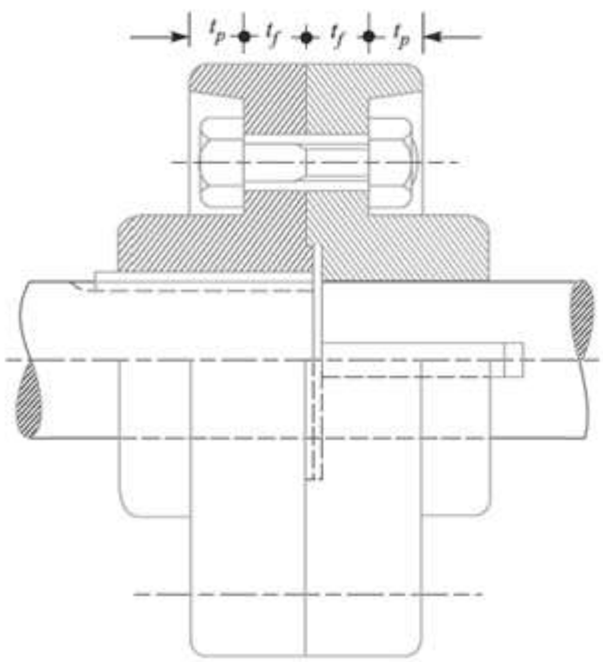
Unprotected type flange coupling

In an unprotected type flange coupling, as shown in Fig.1, each shaft is keyed to the boss of a flange with a counter sunk key and the flanges are coupled together by means of bolts. Generally, three, four or six bolts are used. The keys are staggered at right angle along the circumference of the shafts in order to divide the weakening effect caused by keyways.

Protected type flange coupling.

In a protected type flange coupling, as shown in Fig.2, the protruding bolts and nuts are protected by flanges on the two halves of the coupling, in order to avoid danger to the workman. The thickness of the protective circumferential flange (t_p) is taken as 0.25 d. The other proportions of the coupling are same as for unprotected type flange coupling





Design of Flange Coupling

Consider a flange coupling as shown in Fig.1 and Fig.2. Let d = Diameter of shaft or inner diameter of hub,

D = Outer diameter of hub,

D_1 = Nominal or outside diameter of bolt, D_1 = Diameter of bolt circle,

n = Number of bolts,

t_f = Thickness of flange,

τ_s, τ_b and τ_k = Allowable shear stress for shaft, bolt and key material respectively τ_c = Allowable shear stress for the flange material i.e. cast iron,

σ_{cb}, and σ_{ck} = Allowable crushing stress for bolt and key material respectively.

The flange coupling is designed as discussed below:

1. Design for hub

The hub is designed by considering it as a hollow shaft, transmitting the same torque (T) as that of a solid shaft.

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

The outer diameter of hub is usually taken as twice the diameter of shaft. Therefore from the above relation, the induced shearing stress in the hub may be checked.

The length of hub (L) is taken as 1.5 d.

2.Design for key

The key is designed with u … nd then checked for shearing and crushing stresses. Th … s usually the same as that of shaft. The length of key is … ength of hub.

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

3.Design for flange

The flange at the junction of the hub is under shear while transmitting the torque. Therefore, the torque transmitted,

T = Circumference of hub × Thickness of flange × Shear stress of flange × Radius of hub

$$= \pi D \times t_f \times \tau_c \times \frac{D}{2} = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

The thickness of flange is usually taken as half the diameter of shaft. Therefore from the above relation, the induced shearing stress in the flange may be checked.

4.Design for bolts

The bolts are subjected to shear stress due to the torque transmitted. The number of bolts (n) depends upon the diameter of shaft and the pitch circle diameter of bolts (D_1) is taken as 3 d. We know that

Load on each bolt

$$= \frac{\pi}{4} (d_1)^2 \; \tau_b$$

Then, Total load on all the bolts

$$= \frac{\pi}{4}(d_1)^2 \ \tau_b \times n$$

And torque transmitted,

$$T = \frac{\pi}{4}(d_1)^2 \ \tau_b \times n \times \frac{D_1}{2}$$

From this equation, the diameter of bolt (d_1) may be obtained. Now the diameter of bolt may be checked in crushing.

We know that area resisting crushing of all the bolts = $n \times d_1 \times t_f$ And crushing strength of all the bolts = $(n \times d_1 \times t_f)\ \sigma_{cb}$

Torque,

$$T = (n \times d_1 \times t_f \times \sigma_{cb})\ \frac{D_1}{2}$$

From this equation, the induced crushing stress in the bolts may be checked

Problem: Design a cast iron protective type flange coupling to transmit 15 kW at 900 r.p.m. from an electric motor to a compressor. The service factor may be assumed as 1.35. The following permissible stresses may be used :

Shear stress for shaft, bolt and key material = 40 Mpa, Crushing stress for bolt and key = 80 Mpa, Shear stress for cast iron = 8 Mpa. Draw a neat sketch of the coupling

Solution. Given: P = 15 kW = 15 × 103 W; N = 900 r.p.m. ; Service factor = 1.35 ; $\tau_s = \tau_b = \tau_k$= 40 MPa = 40 N/mm^2 ; $\sigma_{cb} = \sigma_{ck}$ = 80 MPa = 80 N/mm^2 ; τ_c = 8 MPa = 8 N/mm^2. The protective type flange coupling is designed as discussed below:

Design for hub

First of all, let us find the diameter of the shaft (d). We know that the torque transmitted by the shaft,

$$T = \frac{P \times 60}{2\,\pi N} = \frac{15 \times 10^3 \times 60}{2\,\pi \times 900} = 159.13 \text{ N-m}$$

Since the service factor is 1.35, therefore the maximum torque transmitted by the shaft, T_{max}= 1.35 × 159.13 = 215 N-m = 215 × 103 N-mm

We know that the torque transmitted by the shaft (T),

$$215 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\, d^3$$

$$d^3 = 215 \times 10^3 / 7.86 = 27.4 \times 10^3 \quad \text{or} \quad d = 30.1 \text{ say } 35 \text{ mm Ans.}$$

We know that outer diameter of the hub,

D = 2d = 2 × 35 = 70 mm Ans.

And length of hub, L = 1.5 d = 1.5 × 35 = 52.5 mm Ans.

Let us now check the induced shear stress for the hub material which is cast iron. Considering the hub as a hollow shaft. We know that the maximum torque transmitted (T_{max}).

$$215 \times 10^3 = \frac{\pi}{16} \times \tau_c \left[\frac{D^4 - d^4}{D}\right] = \frac{\pi}{16} \times \tau_c \left[\frac{(70)^4 - (35)^4}{70}\right] = 63\,147\, \tau_c$$

Then, τ_c = 215 × 103/63 147 = 3.4 N/mm2 = 3.4 MPa

Since the induced shear stress for the hub material (i.e. cast iron) is less than the permissible value of 8 MPa, therefore the design of hub is safe.

Design for key

Since the crushing stress for the key material is twice its shear stress (i.e. $\sigma_{ck} = 2\tau_k$), therefore a square key may be used. From DDB, we find that for a shaft of 35 mm diameter,

Width of key, w = 12 mm Ans.

And thickness of key, t = w = 12 mm Ans.

The length of key (l) is taken equal to the length of hub. Then, l = L = 52.5 mm Ans.

Let us now check the induced stresses in the key by considering it in shearing and crushing. Considering the key in shearing. We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 - l \times w \times \tau_k \times \frac{d}{2} = 52.5 \times 12 \times \tau_k \times \frac{35}{2} - 11\,025\ \tau_k$$

Then, τ_k = 215 × 103/11 025 = 19.5 N/mm2 = 19.5 MPa

Considering the key in crushing. We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2} = 52.5 \times \frac{12}{2} \times \sigma_{ck} \times \frac{35}{2} = 5512.5\, \sigma_{ck}$$

Σ_{ck} = 215 × 103/ 5512.5 = 39 N/mm^2 = 39 MPa.

Since the induced shear and crushing stresses in the key are less than the The thickness permissible stresses, therefore the design for key is safe.

Design for flange

The thickness of flange (t_f) is taken as 0.5 d.

Then, t_f = 0.5 d = 0.5 × 35 = 17.5 mm Ans.

Let us now check the induced shearing stress in the flange by considering the flange at the junction of the hub in shear.

We know that the maximum torque transmitted (T_{max}),

τ_c = 215 × 103/134 713 = 1.6 N/mm2 = 1.6 MPa

$$215 \times 10^3 - l \times w \times \tau_k \times \frac{d}{2} = 52.5 \times 12 \times \tau_k \times \frac{35}{2} - 11\,025\ \tau_k$$

Since the induced shear stress in the flange is less than 8 MPa, therefore the design of flange is safe.

Design for bolts

Let d_1 = Nominal diameter of bolts.

Since the diameter of the shaft is 35 mm, therefore let us take the number of bolts, n = 3 and pitch circle diameter of bolts,

D_1 = 3d = 3 × 35 = 105 mm

The bolts are subjected to shear stress due to the torque transmitted. We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 = \frac{\pi}{4}(d_1)^2\ \tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4}(d_1)^2\ 40 \times 3 \times \frac{105}{2} = 4950\ (d_1)^2$$

$(d_1)^2 = 215 \times 103/4950 = 43.43$ or $d_1 = 6.6$ mm

Assuming coarse threads, the nearest standard size of bolt is M 8. Ans. Other proportions of the flange are taken as follows:

Outer diameter of the flange,

$D_2 = 4\ d = 4 \times 35 = 140$ mm Ans.

Thickness of the protective circumferential flange,

$t_p = 0.25\ d = 0.25 \times 35 = 8.75$ say 10 mm Ans.

THANK YOU