Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Mechanical Techniques

Learning package

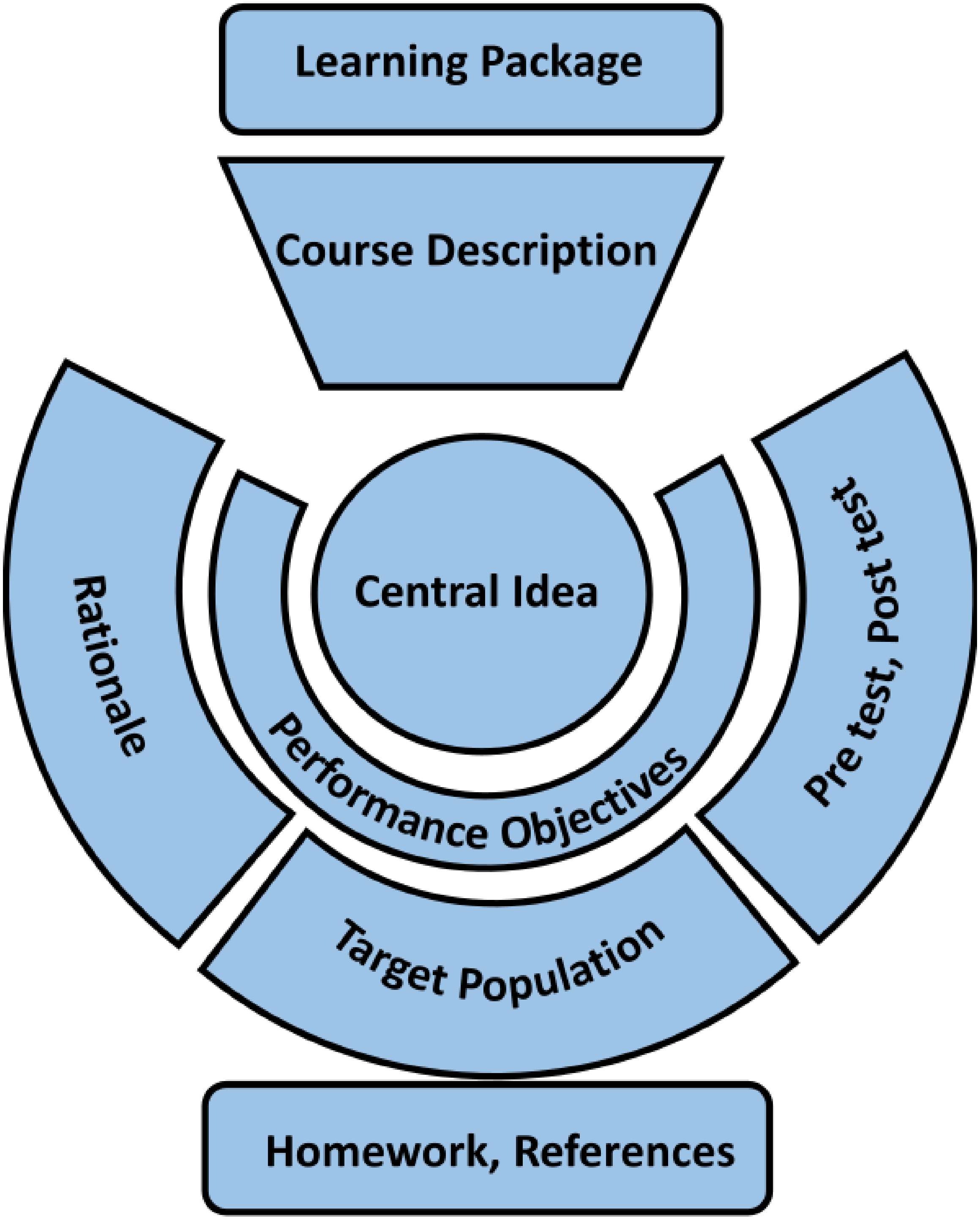
Machine Parts Technology

For

Second year students

By

Mr. Saleh Kudhair Jebur
Assistant Lecturer
Dep. Of Mechanical Techniques
2025


Learning Package
Course Description
Central Idea
Rationale
Performance Objectives
Pre test, Post test
Target Population
Homework, References

Course Description

Course Name:
Machine parts technology
Course Code:
N/A
Semester / Year:
Semester
Description Preparation Date:
25/ 06/ 2025
Available Attendance Forms:
Attendance only
Number of Credit Weeks (Total) / Number of Units (Total)
30 weeks/3 hour weekly/ 3 Units for each semester
Course administrator's name (mention all, if more than one name)
Name: Mr. Saleh Kudhair Jebur Email: salih.kudhair@stu.edu.iq
Course Objectives
1. Understanding Fundamental Machine Components Students will develop comprehensive knowledge of basic machine elements including bearings, gears, shafts, couplings, and fasteners, enabling them to identify, classify, and understand the function of each component in mechanical systems. 2. Design Principles and Selection Criteria Students will learn to apply engineering principles to select appropriate machine parts based on load requirements, operating conditions, material properties, and cost consideratio preparing them for real-world engineering decisions. 3. Material Properties and Applications Students will gain understanding of various engineering materials used in machine parts manufacturing, including metals, polymers, and composites, and their specific applications based on mechanical properties, durability, and environmental factors. 4. Failure Analysis and Maintenance Concepts Students will develop skills to analyze common failure modes of machine components such as fatigue, wear, corrosion, and overload, and understand preventive maintenance strategies to extend component life and ensure system reliability. 5. Practical Assembly and Troubleshooting Skills Students will acquire hands-on experience in assembling, disassembling, and troubleshooting machine parts, developing practical skills essential for technical

careers in mechanical maintenance, manufacturing, and engineering support roles.

Teaching and Learning Strategies

1. Cooperative Concept Planning Strategy.
2. Brainstorming Teaching Strategy.
3. Note-taking Sequence Strategy.

Course Structure

Weeks	Hours	Unit or subject name	Learning method	Evaluation method
1	3	Simple Stress and Strain (Review of Strength of Materials)	Visual Learning through Data Show Presentati and Technical Catalogs 1. Students learn through detailed PowerPoint presentations displayed via data show projecto featuring high-resolution images, technical drawings, and manufacturer catalogs of variou machine components to understand their specifications, applications, and selection crite 2. Theoretical Problem-Solving with Real Indu Examples Students solve practical calculation problems a case studies using textbooks and handouts, focusing on component selection, load calculations, and failure analysis based on actu industrial scenarios and manufacturer specifications. 3. Group Discussions and Technical Drawing Analysis Students participate in classroom discussions v analyzing technical drawings, assembly diagra and component specifications sheets to develo understanding of machine parts relationships a assembly procedure	Weekly, Monthly, Daily, and Written Exams, and Final Term Exam.
2	3	Simple Stress and Strain (Review of Strength of Materials)		
3	3	Discussion		
4	3	Welded Joints		
5	3	Welded Joints		
6	3	Welded Joints		
7	3	Discussion		
8	3	Keys		
9	3	Keys		
10	3	Discussion		
11	3	Belts		
12	3	Belts		
13	3	Discussion		
14	3	Riveted Joints		
15	3	Riveted Joints		
16	3	Discussion		
17	3	Shafts		
18	3	Shafts		
19	3	Discussion		
20	3	Screwed Joints		
21	3	Screwed Joints		
22	3	Discussion		
23	3	Springs		
24	3	Springs		
25	3	Discussions		

26	3	Clutches		
27	3	Clutches		
28	3	Discussion		
29	3	Gear		
30	3	Discussion		

Course Evaluation
Distribution as follows: 30 points for Midterm Theoretical Exams for the each semester, 10 points fo Daily Exams and Continuous Assessment, and 60 points for the Final Exam.

Learning and Teaching Resources	
Required textbooks (curricular books, if any)	Machine Elements in Mechanical Design" by Robert L. Mott
Recommended books and references (scientific journals, reports...)	Illustrated Sourcebook of Mechanical Components" by Robert O. Parmle
Electronic References, Websites	1. MIT OpenCourseWare - Elements of Mechanical Design. 2. URL: https://www.machinedesign.com/learning-resources

Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

Learning package

In

Simple Stress and strain

For

Students of Second Year

By

Mr. Saleh Khudhair Jebur
Assistant Lecturer

Dep. Of Mechanical Techniques

1. Overview:

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

1 / B –Rationale:-

Studying the stress and strain behavior is crucial for comprehensive knowledge of design criteria.

1 / C –Central Idea:-

1 – Stress definition

2 – strain definition

3 –Young's Modulus Relationship

1 / D – Performance Objectives

After studying the first unit, the student will be able to:-

1. Know the Stress and strain Concepts
2. Stress and Strain Curves for variables materials
3. Derive the Young's Modulus (Modulus of Elasticity)

2. Stress & Strain Relations

Machine parts are subjected to various forces which may be due to either one or more of the followings:

1 – Energy transmitted
2 – Weight of machine
3 – Frictional resistance
4 – Inertia of recipracating part
5 – Change of temperture

The different forces acting on a machine part produce various types of stresses.

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

Types of loads :
1 – dead or steady load
2 – live or varying load
3 – suddenly applied or shock load

Stress (σ) :
The internal force per unit area at any section of the body .

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

σ : stress ***(kg / cm^2) or (N / mm^2)***
P : load or force ***(kg) or (N)***
A : cross-sectional area of the body . ***(mm^2)***

Strain (Є) :
The deformation per unit length .

$$\in = \frac{\delta l}{L}$$

ϵ : Strain ***(without units)***

δl : change in length ***(mm)***

L : original length ***(mm)***

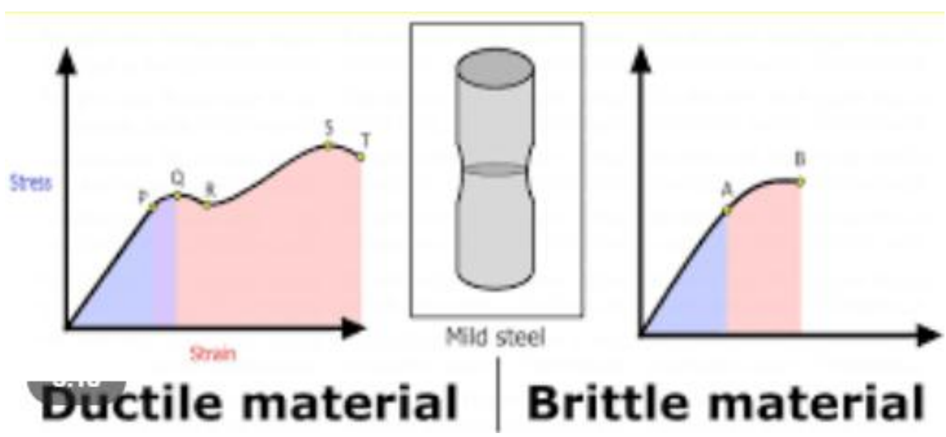
Young Modulus :(E)

Hook's law states that when the material is loaded within elastic limit , the stress is proportional to the strain .

$\sigma \ \alpha \ \epsilon \sigma = E \ \epsilon$

$E = \sigma / \epsilon$

$$E = \frac{P / A}{\delta l / L}$$



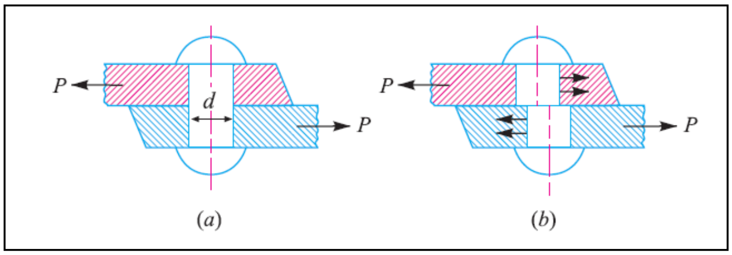
E : Young Modulus , or modulus of elasticity ***(N / mm^2)***

Shear stress : (τ)

When a body is subjected to two equal and opposite forces acting tangentially across the resisting section .

$$\text{Shear stress } (\tau) = \frac{\text{Tangential forces}}{\text{Resisting area}}$$

$$\tau = \frac{P}{A} = \frac{P}{\frac{\pi}{4}(d^2)}$$



Bearing stress (Crushing stress) : (σ_c or σ_b)

A localized compressive stress at the area of contact between two members such as (riveted joint) .

σ_c or $\sigma_b = P / d \,.\, t \,.\, n$

t = thickness of the plate in riveted joints

n = number of rivets in crushing

Safety factor : (S.F)

The ratio of the maximum stress to the working stress or design stress .

$$S.F = \frac{\sigma_{max}}{\sigma_{work}} = \frac{\text{yield point stress}}{\sigma_{design}} \qquad \textit{for ductile materials}$$

$$S.F = \frac{\sigma_{max}}{\sigma_{work}} = \frac{\text{ultimate point stress}}{\sigma_{design}} \qquad \textit{for brittle materials}$$

Working stress : (σ_w)

It is lower than the maximum or ultimate stress at which failure of the material take place .

Example (1) :

A load of 5 KN is to be raised with the help of a steel wire . Find the minimum diameter of the steel wire if the stress is not to exceed 100 MN / m^2 .

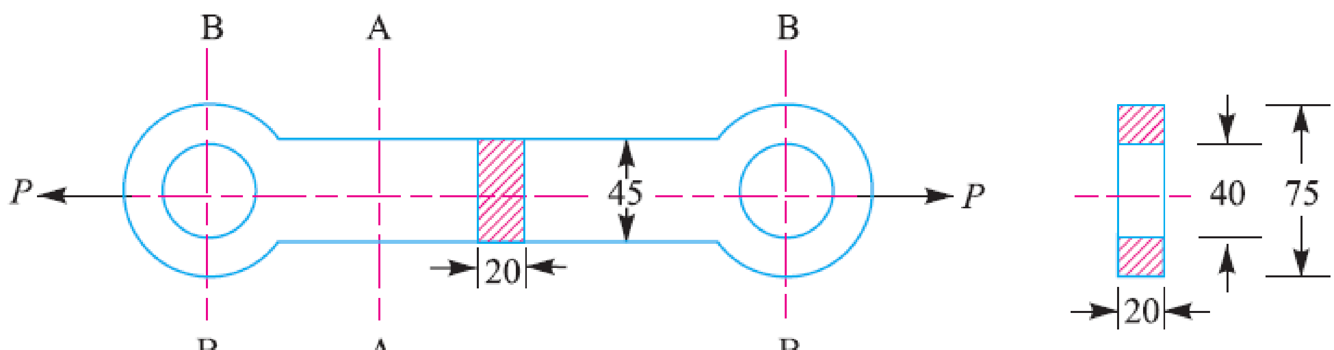
Solution :

P = 5 KN = 5000 N , σ = 100 MN / m^2 = 100 N / mm^2

$$\sigma = \frac{P}{A} \Longrightarrow 100 = \frac{5000}{\frac{\pi}{4}(d^2)} \Longrightarrow d = 7.98 \approx 8\ mm$$

Example (2):

A cast iron link, as shown in Fig. below, is required to transmit a steady tensile load of 45 kN. Find the tensile stress induced in the link material at sections A-A and B-B.



All dimensions in mm.

Solution :

Given : $P = 45\ \text{kN} = 45 \times 10^3\ \text{N}$

Tensile stress induced at section A-A

We know that the cross-sectional area of link at section *A-A*,

$$A_1 = 45 \times 20 = 900\ \text{mm}^2$$

∴ Tensile stress induced at section *A-A*,

$$\sigma_{t1} = \frac{P}{A_1} = \frac{45 \times 10^3}{900} = 50\ \text{N/mm}^2 = 50\ \text{MPa}$$ **Ans.**

Tensile stress induced at section B-B

We know that the cross-sectional area of link at section B-B,

$$A_2 = 20\ (75 - 40) = 700\ \text{mm}^2$$

∴ Tensile stress induced at section *B-B,*

$$\sigma_{t2} = \frac{P}{A_2} = \frac{45 \times 10^3}{700} = 64.3\ \text{N/mm}^2 = 64.3\ \text{MPa}\ \textbf{Ans.}$$

Example (3):

The piston rod of a steam engine is 50 mm in diameter and 600 mm long. The
diameter of the piston is 400 mm and the maximum steam pressure is 0.9 N/mm²$^{}$. Find the compression of the piston rod if the Young's modulus for the material of the piston rod is 210 kN/mm²$^{}$.

Solution :

Given : $d = 50$ mm ; $l = 600$ mm ; $D = 400$ mm ; $p = 0.9\ \text{N/mm}^2$; $E = 210\ \text{kN/mm}^2 = 210 \times 10^3\ \text{N/mm}^2$

Let δl = Compression of the piston rod.

We know that cross-sectional area of piston,

$$= \frac{\pi}{4} \times D^2 = \frac{\pi}{4}\ (400)^2 = 125\ 680\ \text{mm}^2$$

∴ Maximum load acting on the piston due to steam,

$$P = \text{Cross-sectional area of piston} \times \text{Steam pressure}$$

$$= 125\ 680 \times 0.9 = 113\ 110\ \text{N}$$

We also know that cross-sectional area of piston rod,

$$A = \frac{\pi}{4} \times d^2 = \frac{\pi}{4} (50)^2$$

$$= 1964 \text{ mm}^2$$

and Young's modulus (E),

$$210 \times 10^3 = \frac{P \times l}{A \times \delta l}$$

$$= \frac{113\,110 \times 600}{1964 \times \delta l} = \frac{34\,555}{\delta l}$$

$$\therefore \quad \delta l = 34\,555 / (210 \times 10^3)$$

$$= 0.165 \text{ mm}$$ **Ans.**

Example (4):

Calculate the force required to punch a circular blank of 60 mm diameter in a plate of 5 mm thick. The ultimate shear stress of the plate is 350 N/mm²^2.

Solution :

Given: $d = 60$ mm ; $t = 5$ mm ; $\tau_u = 350$ N/mm^2

We know that area under shear,

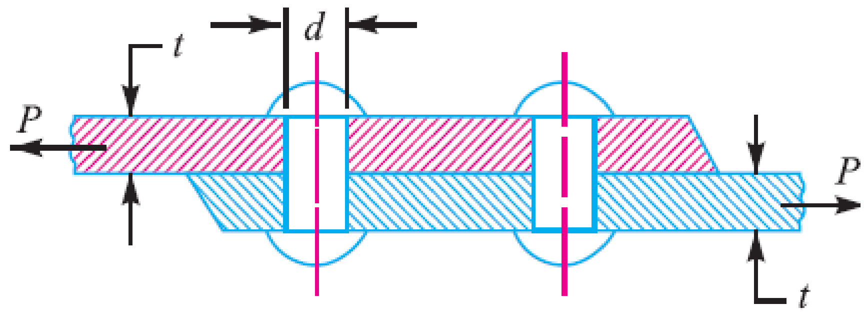
$$A = \pi\, d \times \tau = \pi \times 60 \times 5 = 942.6 \text{ mm}^2$$

and force required to punch a hole,

$$P = A \times \tau_u = 942.6 \times 350 = 329\,910 \text{ N} = 329.91 \text{ kN}$$ **Ans.**

Example (5):

Two plates 16 mm thick are joined by a double riveted lap joint as shown in Fig. below. The rivets are 25 mm in diameter. Find the crushing stress induced between the plates and the rivet, if the maximum tensile load on the joint is 48 kN.



Solution :

Given : $t = 16$ mm ; $d = 25$ mm ;
$P = 48$ kN $= 48 \times 10^3$ N

Since the joint is double riveted, therefore, strength of two rivets in bearing (or crushing) is taken. We know that crushing stress induced between the plates and the rivets,

$$\sigma_c = \frac{P}{d.t.n} = \frac{48 \times 10^3}{25 \times 16 \times 2} = 60 \text{ N/mm}^2 \text{ **Ans.**}$$

HOME WORK :

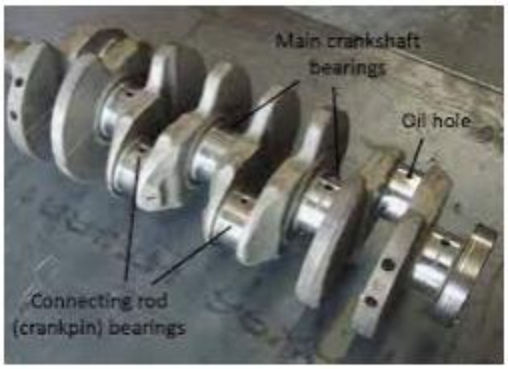
1. A wrought iron rod is under a compressive load of 350 kN. If the permissible stress for the material is 52.5 N/mm², calculate the diameter of the rod. [Ans. 95 mm]

2. A square tie bar 20 mm × 20 mm in section carries a load. It is attached to a bracket by means of 6 bolts. Calculate the diameter of the bolt if the maximum stress in the tie bar is 150 N/mm² and in the bolts is 75 N/mm². [Ans. 13 mm]

3. The diameter of a piston of the steam engine is 300 mm and the maximum steam pressure is 0.7 N/mm². If the maximum permissible compressive stress for the piston rod material is 40 N/mm², find the size of the piston rod. [Ans. 40 mm]

4. Find the minimum size of a hole that can be punched in a 20 mm thick mild steel plate having an ultimate shear strength of 300 N/mm². The maximum permissible compressive stress in the punch material is 1200 N/mm². [Ans. 20 mm]

5. The crankpin of an engine sustains a maximum load of 35 kN due to steam pressure. If the allowable bearing pressure is 7 N/mm², find the dimensions of the pin. Assume the length of the pin equal to 1.2 times the diameter of the pin. [Ans. 64.5 mm; 80 mm]



Ministry of high Education and Scientific Research
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Department of Electronic Techniques

Learning package In Welded Joints

For

Second year students

By

Mr. Saleh Kudhair Jebur
Assistant Lecturer
Dep. Of Mechanical Techniques
2025

1. Overview

1 / A –Target population :-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

1 / B –Rationale:-

Explore the welding technologies types, and evaluation the welded joints strength.

1 / C –Central Idea:-

Welding strategies investigation, the most proper testing plan and analyzing the strength of joints.

1 / D – Performance Objectives
After studying this unit, the student will be able to distinguish:-

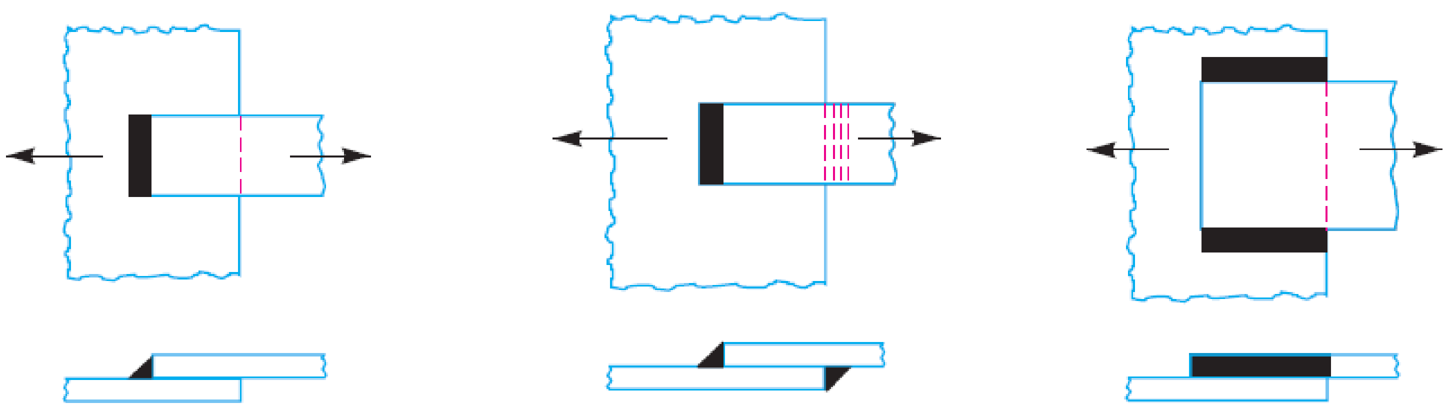
Welding technologies types, and evaluation the welded joints strength.

2. Welded Joints

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 fillet material .***

Uses of welded joints : Welding is used in :
1 - fabrication as an alternative method for casting or forging .
2 – replacement for bolted and riveted joints .
3 – repair medium e.g. to reunite metal at a crack .
4 – build up a small parts that has broken off such as gear tooth .

Types of welding joints :
1 – Lap joint :
The lap joint of the fillet joint is obtained by over lapping the plates and then welding the edges of the plates , the cross – section of the fillet is approximately triangular .
There are three types of lap joint :
a – single transverse fillet
b – double transverse fillet
c – parallel fillet joint

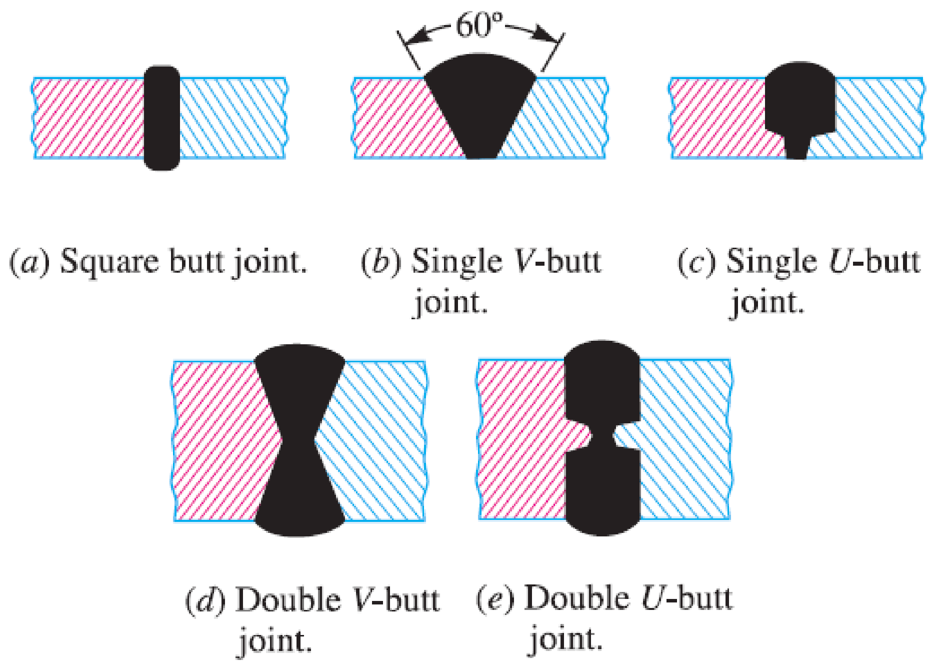


(*a*) Single transverse. (*b*) Double transverse. (*c*) Parallel fillet.

Types of lap or fillet joints.

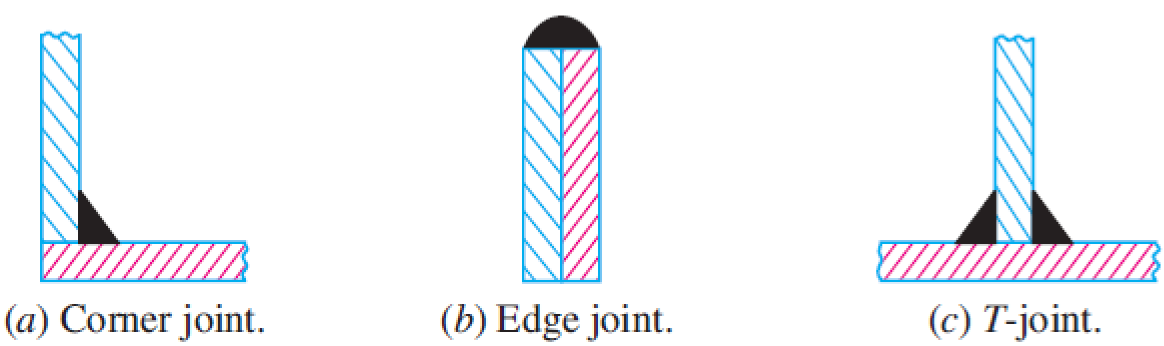
2 – Butt joint :
The butt joint is obtained by placing the plates edge to edge .
There are many types of butt joint :

a – square butt joint
b – single V-butt joint
c – single U-butt joint
d – double V-butt joint
e – double U-butt joint



Types of butt joints.

The other types of welded joints are :
a- corner joint ***b – edge joint*** ***c – T-joint***



Other types of welded joints.

Strength of transverse fillet welded joints :
To determine the strength of transverse fillet welded joint ,

Let t = Throat thickness (BD),

s = Leg or size of weld,

= Thickness of plate, and

l = Length of weld,

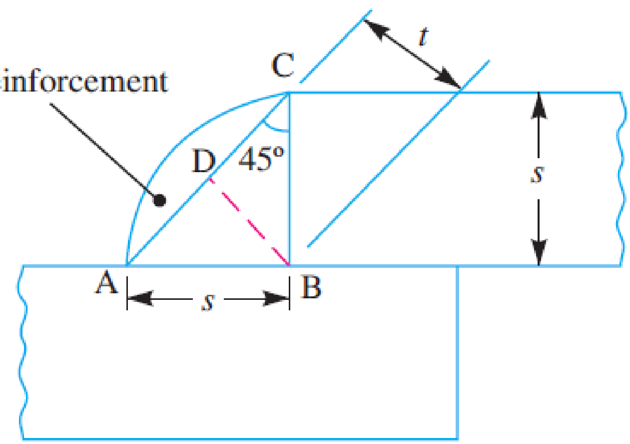
From Fig. 10.7, we find that the throat thickness,

$$t = s \times \sin 45° = 0.707\, s$$

∴ *Minimum area of the weld or throat area,

A = Throat thickness × Length of weld

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



Enlarged view of a fillet weld.

$Pt = 0.707 \times \sigma_t \times s \times l$	for single transverse fillet
$Pt = 1.414 \times \sigma_t \times s \times l$	for double transverse fillet

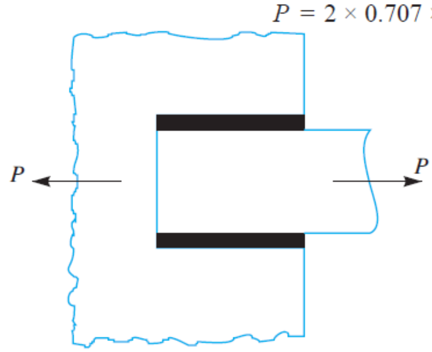
Strength of parallel fillet welded joints :

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

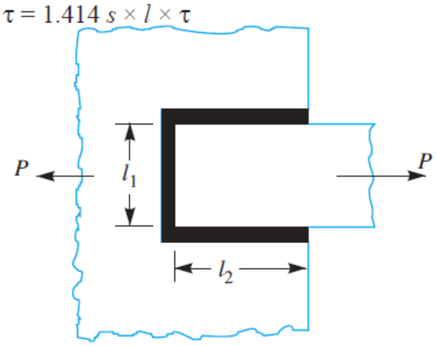
P = Throat area × Allowable shear stress = $0.707\, s \times l \times \tau$

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.

$$\boxed{Ps = 1.414 \times \tau \times s \times l}$$

Notes: 1. If there is a combination of single transverse and double parallel fillet welds as shown in Fig. (*b*), then the strength of the joint is given by the sum of strengths of single transverse and double parallel fillet welds. Mathematically,

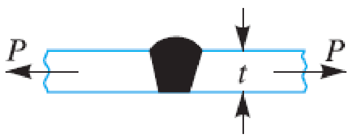
$$P = 0.707s \times l_1 \times \sigma_t + 1.414\, s \times l_2 \times \tau$$

where l_1 is normally the width of the plate.

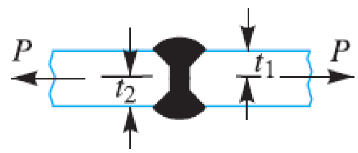
2. In order to allow for starting and stopping of the bead, 12.5 mm should be added to the length of each weld obtained by the above expression.

3. For reinforced fillet welds, the throat dimension may be taken as 0.85 *t*.

Strength of butt fillet welded joints :



(*a*) Single *V*-butt joint.



(*b*) Double *V*-butt joint.

In case of butt joint, the length of leg or size of weld is equal to the throat thickness which is equal to thickness of plates.

∴ Tensile strength of the butt joint (single-*V* or square butt joint),

$$P = t \times l \times \sigma_t$$

where l = Length of weld. It is generally equal to the width of plate.

and tensile strength for double-*V* butt joint as shown in Fig. (*b*) is given by

$$P = (t_1 + t_2)\, l \times \sigma_t$$

where t_1 = Throat thickness at the top, and

t_2 = Throat thickness at the bottom.

Example (1) :

plate 100 mm wide and 12.5 mm thick is to be welded to another plate by means of parallel fillet welds. The plates are subjected to a load of 50 kN. Find the length of the weld so that the maximum stress does not exceed 56 MPa.

Solution :

Given: *Width = 100 mm ; Thickness = 12.5 mm ;
P = 50 kN = 50×10^3N ; τ = 56 MPa = 56 N/mm^2

Let l = Length of weld, and

s = Size of weld = Plate thickness

= 12.5 mm ... (Given)

We know that the maximum load which the plates can carry for double parallel fillet welds (P),

$$50 \times 10^3 = 1.414\, s \times l \times \tau$$

$$= 1.414 \times 12.5 \times l \times 56 = 990\, l$$

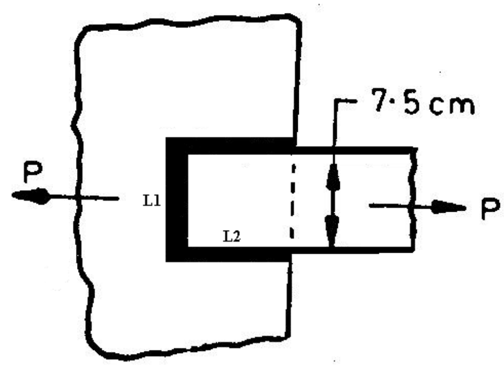
$$\therefore \quad l = 50 \times 10^3 / 990 = 50.5 \text{ mm}$$

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

$$l = 50.5 + 12.5 = 63 \text{ mm}$$ **Ans.**

Example (2) :

A plate 75 mm wide and 12.5 mm thick is joined with another plate by a single transverse weld and a double parallel fillet welds as shown in figure , the maximum tensile and shear stresses are 70 N / mm² and 56 N / mm² respectively . Find the length of each parallel fillet .



Solution :

Given : Width = 75 mm ; Thickness = 12.5 mm ;

σ_τ = 70 MPa = 70 N/mm^2 ; τ = 56 MPa = 56 N/mm^2.

The effective length of weld (l_1) for the transverse weld may be obtained by subtracting 12.5 mm from the width of the plate.

$\therefore \quad l_1 = 75 - 12.5 = 62.5$ mm

Let $\quad l_2$ = Length of each parallel fillet.

We know that the maximum load which the plate can carry is

$$P = \text{Area} \times \text{Stress} = 75 \times 12.5 \times 70 = 65\ 625 \text{ N}$$

Load carried by single transverse weld,

$$P_1 = 0.707\, s \times l_1 \times \sigma_t = 0.707 \times 12.5 \times 62.5 \times 70 = 38\ 664 \text{ N}$$

and the load carried by double parallel fillet weld,

$$P_2 = 1.414\, s \times l_2 \times \tau = 1.414 \times 12.5 \times l_2 \times 56 = 990\, l_2 \text{ N}$$

$\therefore$ Load carried by the joint (P),

$$65\ 625 = P_1 + P_2 = 38\ 664 + 990\, l_2 \quad \text{or} \quad l_2 = 27.2 \text{ mm}$$

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

$$l_2 = 27.2 + 12.5 = 39.7 \text{ say } 40 \text{ mm } \textbf{Ans.}$$

Homework :

1 – A plate 10 cm wide and 10 mm thick is to be welded with another plate by means of transverse welds at the ends , if the plates are subjected to a load of 70 kN , Find the size of the weld , the permissible tensile stress should not exceed 70 N / mm^2 .

(Ans. 8.32 cm)

2 – if the plates in question 1 are joined by double parallel fillets and the shear stress is not to exceed 56 N / mm^2 , Find the length of the weld .

(Ans. 9.1 cm)

Ministry of high Education and Scientific Research
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Department of Electronic Techniques

Learning package
In

Keys (Keyed Joints)

For

Second year students

By

Mr. Saleh Kudhair Jebur
Assistant Lecturer
Dep. Of Mechanical Techniques
2025

1. Overview

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

1 / B –Rationale:-

Explore the Keys Design Principle, types, and evaluation the Keyed joints strength.

1/ C –Central Idea :-

Mechanical fasteners that prevent relative rotation between shafts and mounted components while transmitting torque through positive mechanical connection.

1 / D – Performance Objectives

Effectively transfer rotational forces, maintain precise angular positioning, distribute loads evenly, allow easy assembly/disassembly, and provide long-term durability under repeated loading cycles.

2. 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 .

Uses of keys :

Keys are used as temporary fastening and are subjected to considerable crushing and shearing stresses .

A key way :

A key way is a slot or recess in a shaft and hub of the pulley to accommodate a key .

Types of keys :

1 – Sunk keys :

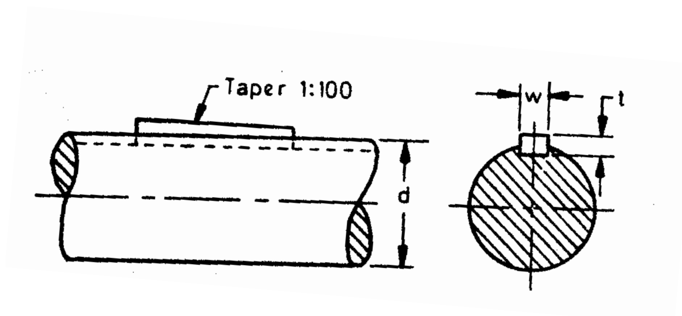
a – Rectangular sunk key

width of key , w = d / 4

thickness of key , t = ⅔ . w

d = diameter of the shaft

= diameter of the hole in hub



b –Squar sunk key :

w = t = d / 4

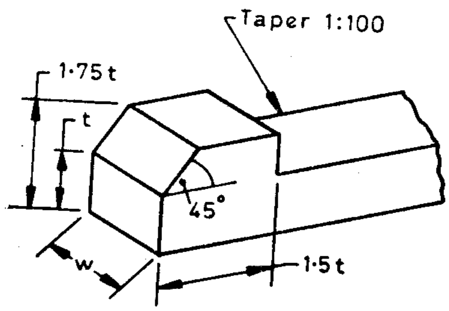
c –Parallel sunk key :

it is a taperless and is used where the pulley , gear or other mating piece is required to slide along the shaft .

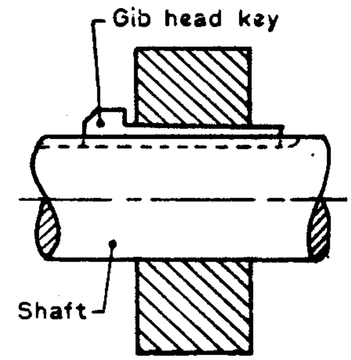
d – Gib-head key :

it is rectangular sunk key with a head at one end

•



(a)



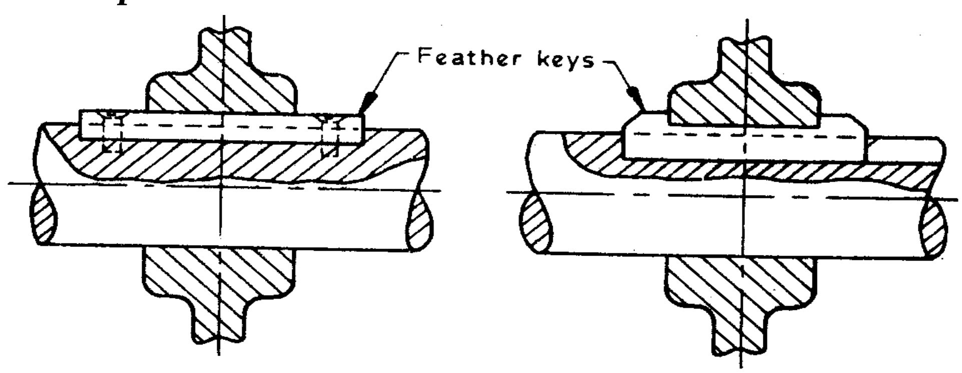
(b)

a - Gib-head key ***b - Gib-head key use***

width , w = d / 4
thickness of key , t = 2w/3 . w = d / 6

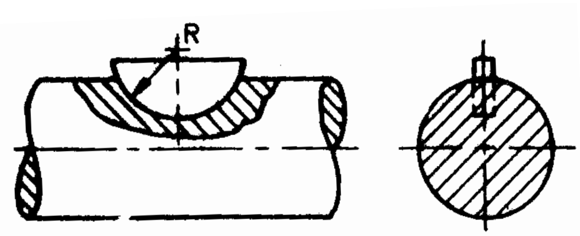
e-Feather key :

it is a special type of parallel key which transmits a turning moment and also permits axial movement .



(a) (b)

f – Woodruff key :
this key is largely used in machine tool and automobile construction .

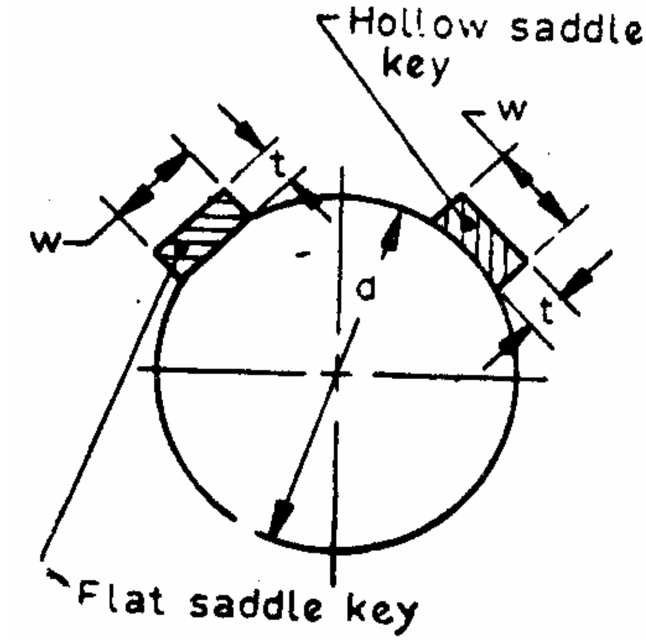


2 – *Saddle keys :*

the saddle keys are of two parts :

a – flat saddle key : used for comparatively light loads .

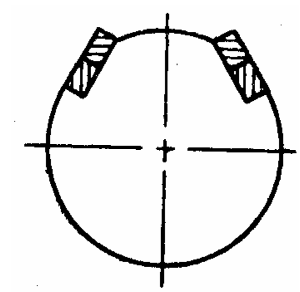
b – hollow saddle key : used as a temporary fastening in fixing and setting eccentrics cams … etc .



$$t = w / 3 = d / 12$$

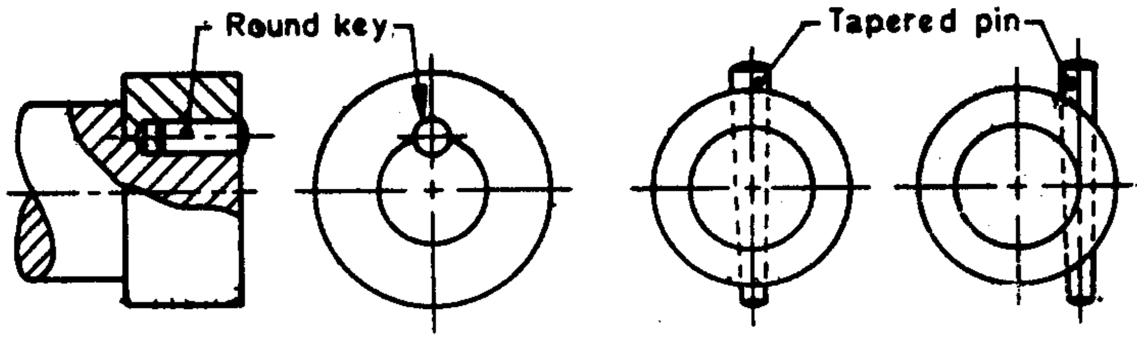
3 – *Tangent keys :*

these are used in large heavy duty shafts .



4 – *Round keys :*

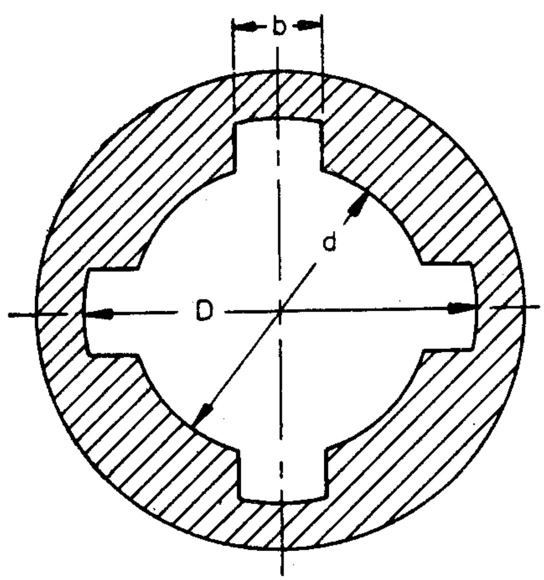
these are usually used for low power drives .



(*a*) (*b*)

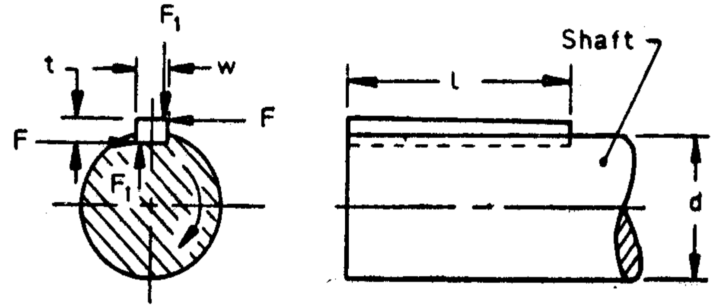
5 – Splines keys :

these keys used with the shafts which have (4) , (6) , (10) and (16) splines to transmite large power and moments as in automobiles and sliding gears .



$D = 1{\cdot}25\, d$, and $b = 0{\cdot}25\, D$

STRENGTH OF *SUNK KEYS* :



T : Torque transmitted by the shaft
F: Tangential force acting at the circumference of the shaft .
d : diameter of the shaft
l : length of the key
w : width of the key
t : thickness of the key
τ : shear stress for the material of key
σ_c : crushing stress for the material of key

In shearing :
F = area resisting shearing X shear stress
$= l \cdot w \cdot \tau$
$T = F \cdot (d/2) = l \cdot w \cdot \tau \cdot (d/2)$

In crushing :
F = area resisting crushing X crushing stress
$= l \cdot (t/2) \cdot \sigma_c$
$T = F \cdot (d/2) = l \cdot (t/2) \cdot \sigma_c \cdot (d/2)$

The key is equally strong in shearing and crushing , if :
$l \cdot w \cdot \tau \cdot (d/2) = l \cdot (t/2) \cdot \sigma_c \cdot (d/2)$
when $w = t$ in square key , then :
$\tau \cdot l \cdot w \cdot (d/2) = \sigma c \cdot l \cdot (w/2) \cdot (d/2)$

$\boxed{\therefore \ \tau = 2 \cdot \sigma_c}$

The length of key :
To find the length of the key to transmit full power of the shaft , the shearing strength of the shaft is equal to the torsional shear strength of the shaft :
we know that the shearing strength of the key :
$T = l \cdot w \cdot \tau \cdot (d/2)$
and the torsional shear strength of the shaft :
$T = (\pi/16) \cdot \tau_1 \cdot d^3$ ***(taking τ_1 = shear stress for the shaft material)***
$l \cdot w \cdot \tau \cdot (d/2) = (\pi/16) \cdot \tau_1 \cdot d^3$

$$l = \frac{\pi}{8} \times \frac{\tau_1 \times d^3}{w \times \tau_s}$$

$$= \frac{\pi . d}{2} \times \frac{\tau_1}{\tau} \qquad (taking \quad w = \frac{d}{4} \quad)$$

$$= 1.571 \; d \times \frac{\tau_1}{\tau}$$

Special case : when the key material is same as that of the shaft , then : τ = τ₁

$$l = \frac{\pi . d^2}{8w} = \frac{\pi}{2} \times d = 1.571 \times d \qquad (taking \quad w = \frac{d}{4} \quad)$$

Example (1) :

Design the rectangular key for a shaft of 50 mm diameter , the shearing and crushing stresses in the key are limited to 42 N/mm² and 70 N/mm² .if the width of key is 16 mm and thickness of the key is 10mm

Solution :

Diameter of the shaft , d=50 mm

Shear stress in the key, τ = 42 N/mm² , Crushing stress in the key, σ_c =70 N/mm²

shearing of the key :

$T = l \,.\, w \,.\, \tau \,.\, (d / 2)$

$(\pi / 16) \,.\, \tau \,.\, d^3 = l \,.\, w \,.\, \tau \,.\, (d / 2)$

$(\pi / 16) \,.\, d^3 = (l \,.\, w) / 2$

$$l = \frac{\pi . \; d^2}{8w} = \frac{\pi \times 50^2}{8 \times 16} = 61.4 \quad mm$$

crushing of the key :

$T = l \,.\, (t / 2) \,.\, \sigma_c \,.\, (d / 2)$

$(\pi / 16) \,.\, d^3 = l \,.\, (t / 2) \,.\, \sigma_c \,.\, (d / 2)$

$$l = \frac{\pi}{4} \times \frac{\tau \times d^2}{t \times \sigma_c} = \frac{\pi \times 42 \times 5^2}{4 \times 1 \times 70} = 118\,mm \approx 120\,mm$$

Taking larger of the two values, we have length of key,
l = 120 mm

Example (2) : A 15 kW, 960 r.p.m. motor has a mild steel shaft of 40 mm diameter and the extension being 75 mm. The permissible shear and

crushing stresses for the mild steel key are 56 MPa and 112 MPa. Design the keyway in the motor shaft extension.

Solution :

$$T=\frac{P\times 60}{2\,\pi\,N}=\frac{15\times 60}{2\times 3.14\times 960}=149\quad N.m=149\quad\times 10^{3}N.mm$$

Design of keyway :

T = l . w . σ $_s$ ***. (d / 2)***

$$149\quad\times 10^{3}=75\times w\times 56\times\frac{40}{2}$$

$$w=\frac{149\quad\times 10^{3}\times 2}{75\times 56\times 40}=\ 1.7\ \ mm$$

this width of keyway is too small , the width of keyway should be at least (d / 4)

w = (d / 4) = 40 / 4 = 10 mm , since σ $_c$ ***= 2 τ***

therefore , a square key is adopted , then

t = w …….. t = 10 mm .

Example (3) :

A 200 h.p. , 960 r.p.m motor a mild steel shaft of 50 mm diameter , key width is 16 mm with thickness of 14 mm , the permissible shear and crushing stresses for the material key are 42 N / mm² and 70 N / mm², Design the key ?

Solution :

$$T=\frac{P\times 45\times 10^{6}}{2\times\pi\times N}=\frac{200\times 45\times 10^{6}}{2\times\pi\times 960}=1492078\quad N.mm$$

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

$$1492078=42\times 16\times l\times\frac{50}{2}$$

$$l=88.81\quad mm$$

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

$$1492078=70\times\frac{14}{2}\times l\times\frac{50}{2}$$

$$l=121.8\quad mm$$

$$\therefore\ \ l=121.8\ \ mm\approx 122\ \ mm$$

Example (4) :

A flat key 10 mm wide , 8 mm thickness and 75 mm long is required to transmit a torque of 10^6 N .mm from a 50 mm diameter shaft , investigate to determine whether the is sufficient , use a design stress in shear of 50 N / mm^2 and in crushing of 130 N / mm^2, if the length is not sufficient redesign the length of the key ?

Solution :

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

$$10^6 = 50 \times 10 \times l \times \frac{50}{2}$$

$$l = 80 \quad mm$$

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

$$10^6 = 130 \times \frac{8}{2} \times l \times \frac{50}{2}$$

$$l = 76.9 \quad mm$$

$$\therefore \quad l = 80 \quad mm$$

Example (5) :

Prove that the strong is equal in shearing and crushing in square key .

Solution :

$$\because F_s = F_c$$

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

$$2\tau \times w = \sigma_c \times t$$

$$\because w = t$$

$$\therefore 2\tau = \sigma_c$$

Homework :

1. ***Design a key to transmit 40 kW power by a shaft of 50 mm diameter which rotates 1200 r.p.m , if the permissible shear and crushing stresses are 35 N/mm^2 and 65 N/mm^2 respectively , take the key length 30 mm . [Ans. w = 13 mm; t = 14 mm]***
2. ***A belt pulley is fastened to 80 mm diameter shaft transmitting 75 kW at 200 r.p.m by means of key 22 mm width and 14 mm thickness. Determine the length of the key. Take $\tau = 50$ N/mm^2 , : $\sigma_c = 130$ N/mm^2 [Ans. l = 128 mm]***
3. ***A shaft of 80 mm diameter transmits power at maximum shear stress of 63 MPa. Find the length of a 20 mm wide key required to mount a pulley on the shaft so that the stress in the key does not exceed 42 N/mm^2 . [Ans. 152 mm]***
4. ***A shaft of 30 mm diameter is transmitting power at a maximum shear stress of 80 N/mm^2 if a pulley is connected to the shaft by means of a key.***
   Find the dimensions of the key so that the stress in the key is not to exceed 50 N/mm^2 and the length of the key is (4) times the width . [Ans. l = 126 mm]

Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

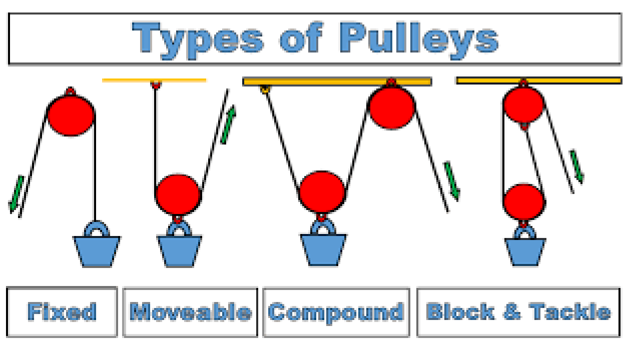
Learning package

In

Belts (Pulleys)

For

Second year students



By

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Dep. Of Mechanical Techniques

2025

1. Overview

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

1 / B –Rationale:-

Belts provide a cost-effective, flexible solution for power transmission that allows for speed variation, shock absorption, and overload protection while accommodating shaft misalignments and reducing maintenance requirements compared to rigid coupling systems.

1/ C –Central Idea:-

Flexible power transmission elements that transfer motion and power between rotating shafts through friction or positive engagement for speed control and power transmission.

1 / D – Performance Objectives

Efficiently transmit mechanical power with minimal losses, maintain desired speed ratios, absorb shock and vibrations, tolerate minor misalignments, and provide overload protection through controlled slippage.

2. Belts

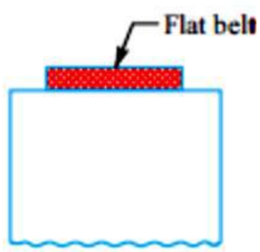
The belts or ropes are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or at different speeds. The amount of power transmitted depends upon the following factors :

1. The velocity of the belt.
2. The tension under which the belt is placed on the pulleys.
3. The arc of contact between the belt and the smaller pulley.
4. The conditions under which the belt is used .

Types of Belts

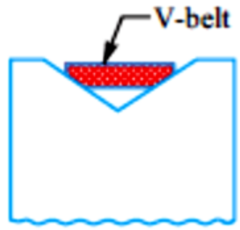
Though there are many types of belts used the following:

1.Flat belt: The flat as shown in Fig (a), is mostly used in the factories and Workshops ,where a moderate amount of power is to be transmitted, from one pulley to another when the two pulleys are not more than 8 metres apart.



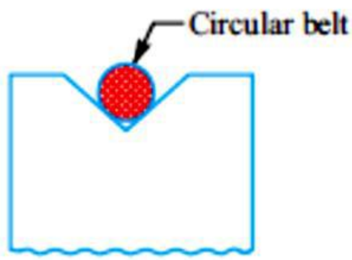
(a) Flat belt.

2.V- belt. The V-belt as shown in Fig. (b), is mostly used in the factories and workshops where a great amount of power is to be transmitted, from one pulley to another, when the two pulleys are very near to each other



(*b*) V-belt.

3.Circular belt or rope: The circular belt or rope as shown in Fig (c) is mostly used in the factories and workshops, where a great amount of power is to be transmitted, from one pulley to another, when the two pulleys are more than 8 metres apart.



(*c*) Circular belt.

1- Velocity Ratio of Belt Drive

It is the **ratio between the velocities of the driver and the follower or driven.** It may be expressed, mathematically, as discussed below :

Let d_1 = Diameter of the driver,

d_2 = Diameter of the follower,

N_1 = Speed of the driver in r.p.m., and

N_2 = Speed of the follower in r.p.m.

∴ Length of the belt that passes over the driver, in one minute

$$= \pi d_1.N_1$$

Similarly, length of the belt that passes over the follower, in one minute

$$= \pi d_2 . N_2$$

Since the length of belt that passes over the driver in one minute is equal to the length of belt that passes over the follower in one minute, therefore

$$\pi\, d_1 \,.\, N_1 \;=\; \pi\, d_2 \,.\, N_2$$

$\therefore$ Velocity ratio, $\dfrac{N_2}{N_1} = \dfrac{d_1}{d_2}$

When the thickness of the belt (t) is considered, then velocity ratio,

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}$$

Note: The velocity ratio of a belt drive may also be obtained as discussed below :

We know that peripheral velocity of the belt on the driving pulley,

$$v_1 = \frac{\pi d_1 . N_1}{60} \text{ m/s}$$

and peripheral velocity of the belt on the driven or follower pulley,

$$v_2 = \frac{\pi d_2 . N_2}{60} \text{ m/s}$$

When there is no slip, then $v_1 = v_2$.

$$\therefore \qquad \frac{\pi d_1 . N_1}{60} = \frac{\pi d_2 . N_2}{60} \qquad \text{or} \qquad \frac{N_2}{N_1} = \frac{d_1}{d_2}$$

2- Length of an Open Belt Drive

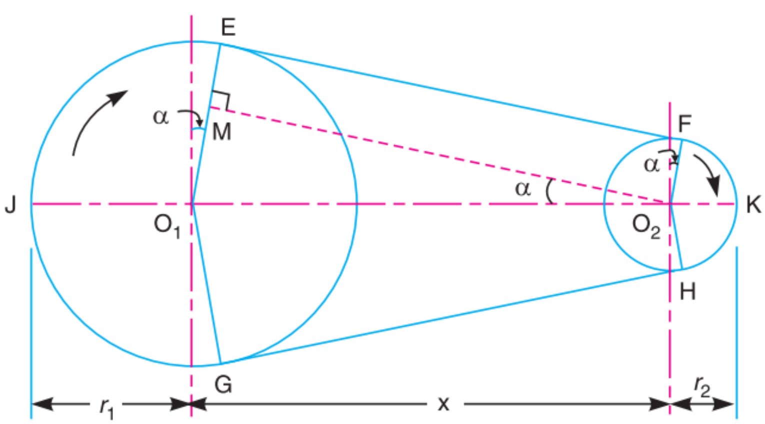


Fig. 11.11. Length of an open belt drive.

Let r_1 and r_2 = Radii of the larger and smaller pulleys,

x = Distance between the centres of two pulleys (*i.e.* $O_1 O_2$), and

L = Total length of the belt.

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x}$$

3- Length of a Cross Belt Drive

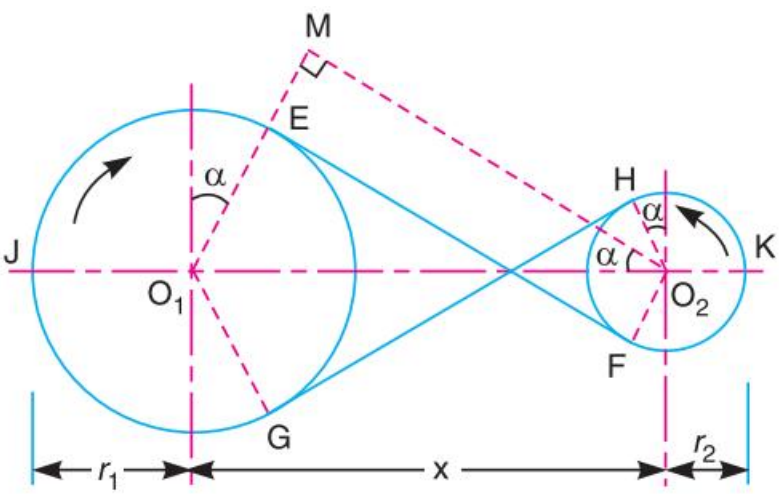


Fig. 11.12. Length of a cross belt drive.

Let r_1 and r_2 = Radii of the larger and smaller pulleys,

x = Distance between the centres of two pulleys (*i.e.* $O_1 O_2$), and

L = Total length of the belt.

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x}$$

4- Power Transmitted by a Belt

Let T_1 and T_2 = Tensions in the tight and slack side of the belt respectively in newtons,

r_1 and r_2 = Radii of the driver and follower respectively, and

v = Velocity of the belt in m/s.

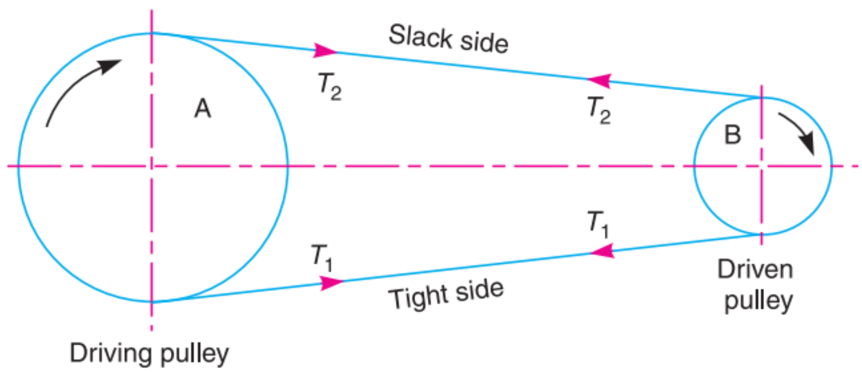


Fig. 11.14. Power transmitted by a belt.

The effective turning (driving) force at the circumference of the follower is the difference between the two tensions (*i.e.* $T_1 - T_2$).

$\therefore$ Work done per second = $(T_1 - T_2)\, v$ N-m/s

and power transmitted, $P = (T_1 - T_2)\, v$ W ...($\because$ 1 N-m/s = 1 W)

A little consideration will show that the torque exerted on the driving pulley is $(T_1 - T_2)\, r_1$. Similarly, the torque exerted on the driven pulley *i.e.* follower is $(T_1 - T_2)\, r_2$.

5- Ratio of Driving Tensions For Flat Belt Drive

Consider a driven pulley rotating in the clockwise direction as shown in Fig. 11.15.

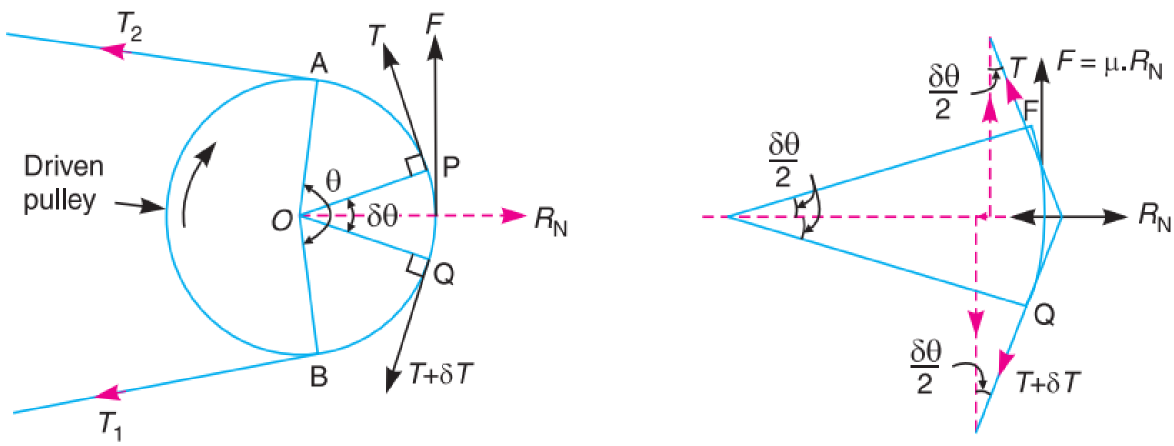


Fig. 11.15. Ratio of driving tensions for flat belt.

Let
T_1 = Tension in the belt on the tight side,
T_2 = Tension in the belt on the slack side, and
θ = Angle of contact in radians (*i.e.* angle subtended by the arc AB, along which the belt touches the pulley at the centre).

$$2.3\log\left(\frac{T_1}{T_2}\right) = \mu.\theta \quad \text{or} \quad \frac{T_1}{T_2} = e^{\mu.\theta}$$

6- Determination of Angle of Contact

Let
r_1 = Radius of larger pulley,
r_2 = Radius of smaller pulley, and
x = Distance between centres of two pulleys (*i.e.* $O_1 O_2$).

From Fig. 11.16 (*a*),

$$\sin\alpha = \frac{r_1 - r_2}{x}$$

open belt

Angle of contact or lap, $\theta = (180° - 2\alpha)\dfrac{\pi}{180}$ rad

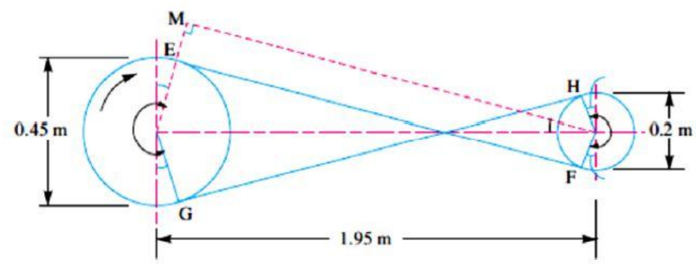
$$\sin\alpha = \frac{r_1 + r_2}{x}$$

crossed belt

Angle of contact or lap, $\theta = (180° + 2\alpha)\dfrac{\pi}{180}$ rad

Example. 1 Two pulleys, one(450 mm) diameter and the other (200 mm) diameter, on parallel shafts (1.95 m) apart are connected by a crossed belt. Find the length of the belt required and the angle of contact between the belt and each pulley. What power can be transmitted by the belt when the larger pulley rotates at (200 rev/min), if the maximum permissible tension in the belt is (1 kN), and the coefficient of friction between the belt and pulley is 0.25?

Solution *.Given :* $d_1 = 450$ *mm* $= 0.45$ *m or* $r_1 = 0.225$ *m ;* $d_2 = 200$ *mm* $= 0.2$ *m or* $r_2 = 0.1$ *m ;* $x = 1.95$ *m ;* $N_1 = 200$ *r.p.m. ;* $T_1 = 1$ *kN* $= 1000$ *N ;* $\mu = 0.25$



Length of the cross belt

$$L_C = \pi (r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x}$$

$$= \pi (0.225 + 0.1) + 2 \times 1.95 + \frac{(0.225 + 0.1)^2}{1.95}$$

$$= 1.02 + 3.9 + 0.054 = 4.974 \text{ m} \quad \textbf{Ans.}$$

$$\sin\alpha = \frac{r_1 + r_2}{x} = \frac{0.225 + 0.1}{1.95} = 0.1667$$

$$\alpha = 9.6°$$

$$\theta = 180° + 2\alpha = 180 + 2 \times 9.6 = 199.2°$$

$$= 199.2 \times \frac{\pi}{180} = 3.477 \text{ rad} \quad \textbf{Ans.}$$

Power transmitted

$$\frac{T_1}{T_2} = e^{\mu\theta} \Rightarrow \frac{1000}{T_2} = e^{0.25\times 3.47} \Rightarrow T_2 = \frac{1000}{2.387} = 419\ N$$

We know that the velocity of belt

$$v = \frac{\pi\, d_1 N_1}{60} = \frac{\pi \times 0.45 \times 200}{60} = 4.713 \text{ m/s}$$

∴ **Power transmitted,**

$$P = (T_1 - T_2)\, v = (1000 - 419)\, 4.713 = 2738 \text{ W} = 2.738 \text{ kW} \quad \textbf{Ans.}$$

Example 2. Find the power transmitted by a belt running over a pulley of 600 mm diameter at 200 r.p.m. The coefficient of friction between the belt and the pulley is 0.25, angle of lap 160° and maximum tension in the belt is 2500 N.

Solution. Given : $d = 600$ mm $= 0.6$ m ; $N = 200$ r.p.m. ; $\mu = 0.25$; $\theta = 160° = 160 \times \pi / 180$ $= 2.793$ rad ; $T_1 = 2500$ N

We know that velocity of the belt,

$$v = \frac{\pi d . N}{60} = \frac{\pi \times 0.6 \times 200}{60} = 6.284 \text{ m/s}$$

Let $T_2 =$ Tension in the slack side of the belt.

We know that
$$2.3 \log \left(\frac{T_1}{T_2} \right) = \mu . \theta = 0.25 \times 2.793 = 0.6982$$

$$\log \left(\frac{T_1}{T_2} \right) = \frac{0.6982}{2.3} = 0.3036$$

$\therefore$
$$\frac{T_1}{T_2} = 2.01 \qquad \text{...(Taking antilog of 0.3036)}$$

and
$$T_2 = \frac{T_1}{2.01} = \frac{2500}{2.01} = 1244 \text{ N}$$

We know that power transmitted by the belt,

$$P = (T_1 - T_2)\, v = (2500 - 1244)\, 6.284 = 7890 \text{ W}$$
$$= 7.89 \text{ kW} \quad \textbf{Ans.}$$

Ministry of high Education and Scientific Research
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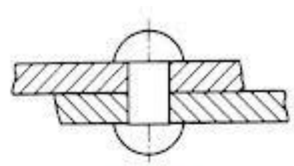
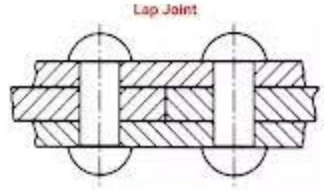
Department of Electronic Techniques

Learning package In Riveted Joints For

Second year students



Lap Joint



Butt Joint

By

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Dep. Of Mechanical Techniques

2025

1. Overview

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

1 / B –Rationale:-

Rivets offer permanent, high-strength joints that are ideal for applications requiring vibration resistance and structural integrity where welding is impractical or where disassembly is not required throughout the component's service life.

1/ C –Central Idea:-

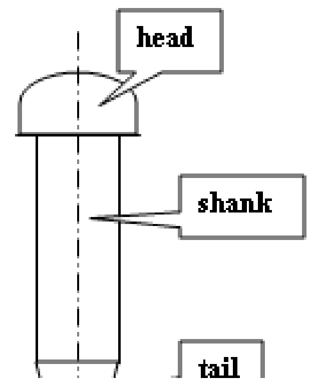
Permanent mechanical fasteners that join materials together by creating strong, non-removable connections through deformation of the rivet material to form mechanical locks

1 / D – Performance Objectives

Create permanent joints with high shear and tensile strength, resist vibration and dynamic loading, provide corrosion resistance, and maintain joint integrity throughout the component's service life.

2. Riveted Joints:

A rivet is a short cylindrical bar with a head integral with it ,The cylindrical portion of the rivet is called " shank " or " body "and the lower portion of shank is known as " tail "as shown in figure.



Uses of rivets :

Rivets are used to make permanent fastening between the plates such as structural work , tanks , and boilers .

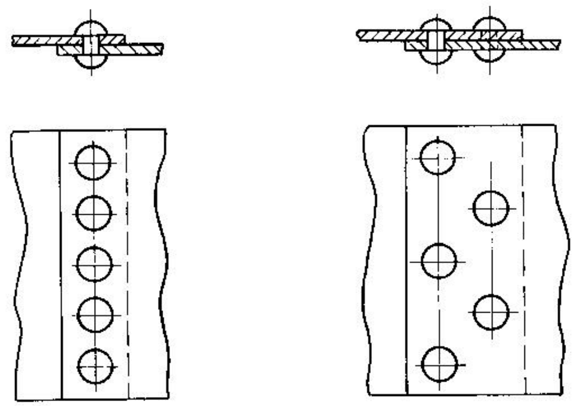
Types of riveted joints :

A – Lap joint :

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

1 - Single riveted joint

2 – Double riveted joint



SINGLE RIVETED LAP JOINT

DOUBLE RIVETED LAP JOINT

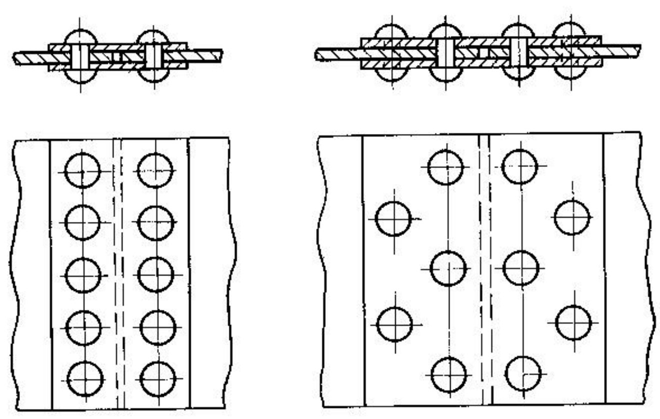
(A) LAP JOINTS

B – 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 plate .

1 - Single riveted joint

2 – Double riveted joint

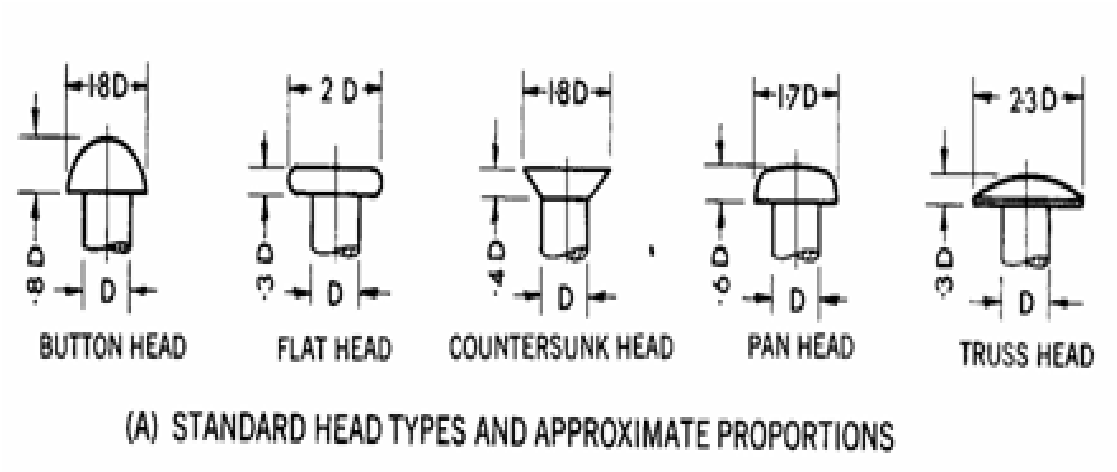


SINGLE RIVETED BUTT JOINT

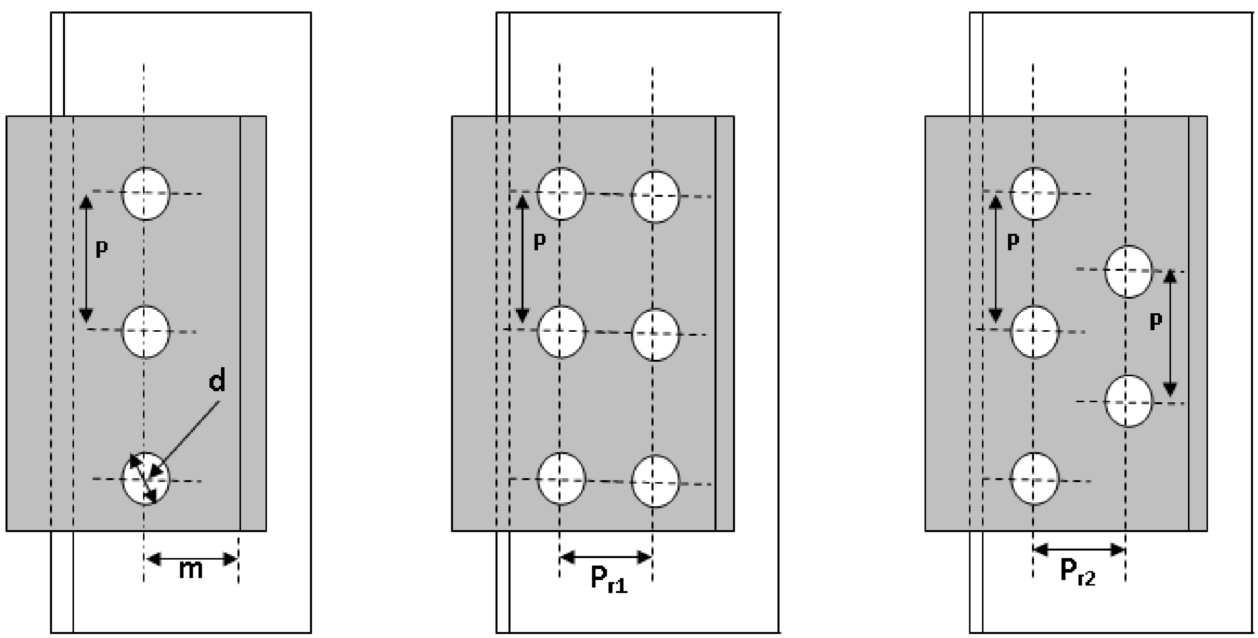
DOUBLE RIVETED BUTT JOINT

(B) BUTT JOINTS

Types of Standard rivet heads:



The main dimensions of riveted joints :



$$m\,(\text{margin}) = 1.5 \times d$$

$$d \;\; (diameter \;\, of \;\, rivet) = 6\sqrt{t} \qquad\qquad when \;\; t = thickness \;\, of \;\, the \;\, plates$$

$$P \;\; (pitch) = 3 \times d$$

$$\text{Pr}_1 = 2d + 6\,mm$$

$$\text{Pr}_2 = 2d$$

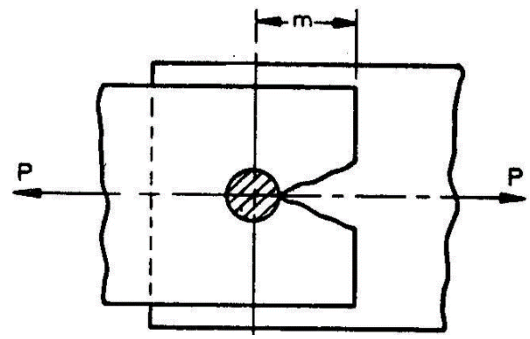
<u>Failure of riveted joints :</u>

A riveted joints may fail in the following ways :

1 – Tearing of the plate at an edge :

This case can be avoided by keeping the margin ,

m = 1.5 d



<u>2 - Tearing of the plate across a row of rivets :</u>

P : pitch of the rivets

d : diameter of the rivets

t :thickness of the plate

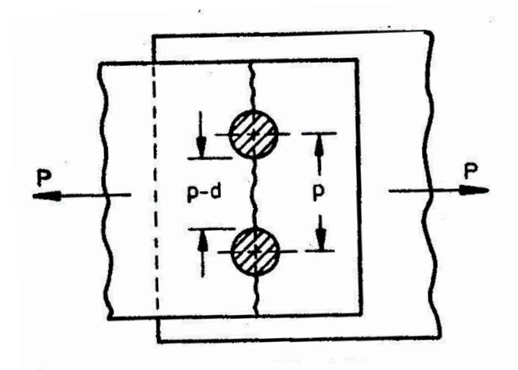
σ_t :tensile stress for the plate material

A_t : tearing area per pitch length

$A_t = (p – d)\ t$

P_t : tearing resistance (pull force) required to tear off the plate per pitch length

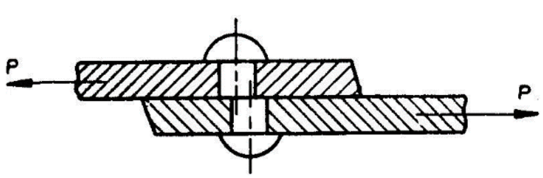
$P_t = \sigma_t \,.\, A_t = \sigma_t\ (p – d)\ .\ t$

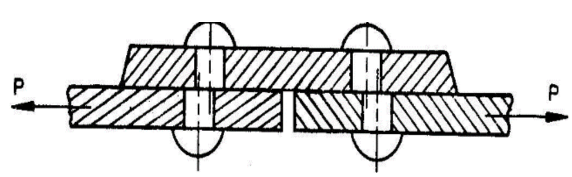


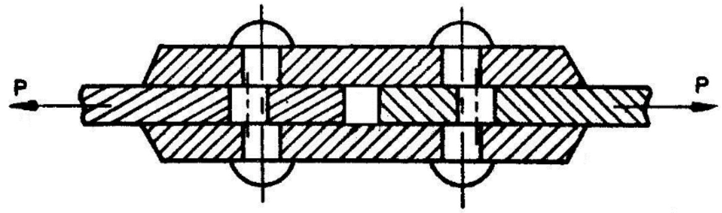
<u>3 – Shearing of the rivets :</u>

a – shearing of a rivet in a lap joint .

b – shearing of a rivet in a single cover butt joint







d : diameter of the rivet
τ: safe permissible shear stress for the rivet material
n : number of rivets per pitch length
As : shearing area

$$As = \frac{\pi}{4}.d^2$$ ***(in single shear)***

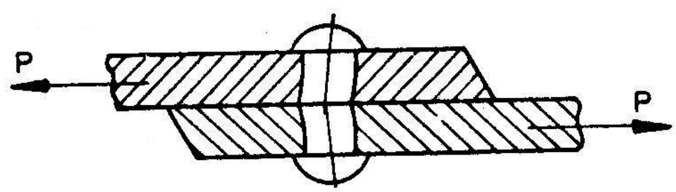
$$As = 2\times\frac{\pi}{4}\times d^2$$ ***(in double shear)***

Ps :shearing resistance or pull required to shear of the rivet per pitch length .

$$Ps = \frac{\pi}{4}\times d^2 \times\tau\times n$$ ***(in single shear)***

$$Ps = 2\times\frac{\pi}{4}\times d^2 \times\tau\times n$$ ***(in double shear)***

<u>4 – Crushing of the rivets :</u>



d : diameter of the rivet
t : thickness of the plate
σ_c : safe permissible crushing stress for the rivet material
n : number of rivets per pitch length under crushing .

$$Ac = d\times t$$

$$\therefore \text{ total crushing area }, \; Ac = n\times d\times t$$

Pc :crushing resistance or pull required to crush the rivet per pitch length .

$$Pc = n\times d\times t\times\sigma_c$$

Efficiency of a riveted joint

The efficiency of a riveted joint is the ratio of the strength of the joint to the strength of the un-riveted or solid plate .

The strength of the riveted joint = Least of P_t , Ps , Pc

P : strength of the un-riveted or solid plate

p = pitch of the rivets × thickness of the plate × tensile stress

$P = p \cdot t \cdot \sigma_t$

$$\therefore Efficiency\,of\ a\ riveted\ joint(\eta) = \frac{Least\ of\ \ Pt\ ,\ Ps\ ,\ Pc}{P}$$

<u>Example (1) :</u>

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.

<u>Solution :</u>

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,

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

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.**

<u>Example (2) :</u>

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

<u>Solution :</u>

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

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

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,

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

$\therefore$ 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,

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

$\therefore$ Efficiency of the joint,

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

Example (3) :

A double riveted double cover butt joint in plates 20 mm thick is made with 25 mm diameter rivets at 100 mm pitch. The permissible stresses are :

σt = 120 MPa; τ = 100 MPa; σc = 150 MPa

Find the efficiency of joint, taking the strength of the rivet in double shear as twice than that of single shear.

Solution :

Given : t = 20 mm ; d = 25 mm ; p = 100 mm ; σ_t = 120 MPa = 120 N/mm^2 ; τ = 100 MPa = 100 N/mm^2 ; σ_c = 150 MPa = 150 N/mm^2

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

(i) Tearing resistance of the plate

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

$$P_t = (p - d)\, t \times \sigma_t = (100 \times 25)\, 20 \times 120 = 180\ 000 \text{ N}$$

(ii) Shearing resistance of the rivets

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

$$P_s = n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times 2 \times \frac{\pi}{4}\,(25)^2\, 100 = 196\ 375 \text{ N}$$

(iii) Crushing resistance of the rivets

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

$$P_c = n \times d \times t \times \sigma_c = 2 \times 25 \times 20 \times 150 = 150\ 000 \text{ N}$$

∴ Strength of the joint

= Least of P_t, P_s and P_c

= 150 000 N

Efficiency of the joint

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

$$P = p \times t \times \sigma_t = 100 \times 20 \times 120$$
$$= 240\,000 \text{ N}$$

$\therefore$ Efficiency of the joint

$$= \frac{\text{Least of } P_t, P_s \text{ and } P_c}{P} = \frac{150\,000}{240\,000}$$
$$= 0.625 \text{ or } 62.5\% \textbf{ Ans.}$$

Example (4) :

A double riveted double cover butt joint in plates 20 mm thick is made with 25 mm diameter rivets at 100 mm pitch. The permissible stresses are :
σt = 120 MPa; τ = 100 MPa; σc = 150 MPa
Find the efficiency of joint, taking the strength of the rivet in double shear as twice than that of single shear.

Solution :

Homework :

1- ***A single riveted lap joint is made in 15 mm thick plates with 20 mm diameter rivets. Determine the strength of the joint, if the pitch of rivets is 60 mm. Take σt = 120 MPa; τ = 90 MPa and σc = 160 MPa. [Ans. 28 280 N]***
2- ***Two plates 16 mm thick are joined by a double riveted lap joint. The pitch of each row of rivets is 90 mm. The rivets are 25 mm in diameter. The permissible stresses are as follows : σt = 140 MPa ; τ = 110 MPa and σc = 240 MPa***
   Find the efficiency of the joint. [Ans. 53.5%]
3- ***A double riveted lap joint with chain riveting is to be made for joining two plates 10 mm thick. The allowable stresses are : σt = 60 MPa ; τ = 50 MPa and σc = 80 MPa. Find the rivet diameter, pitch of rivets and distance between rows of rivets. Also find the efficiency of the joint.***
   [Ans. d = 20 mm ; p = 73 mm; pb = 38 mm; η = 71.7%]

4- Design a triple riveted double strap butt joint with chain riveting for a boiler of 1.5 m diameter and carrying a pressure of 1.2 N/mm2. The allowable stresses are :

σt = 105 MPa ; τ = 77 MPa and σc = 162.5 MPa.

[Ans. d = 20 mm; p = 50 mm]

Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

Learning pack
In
Shafts

For

Second year students

By

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Assistant Lecturer
Dep. Of Mechanical Techniques

1. Overview

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

/ B –Rationale:-

Shafts are fundamental machine elements that enable the transmission of rotational power and motion while providing structural support for rotating components, making them indispensable in virtually all rotating machinery applications

1/ C –Central Idea:-

Rotating machine elements that serve as the backbone of machinery, transmitting power and motion while supporting various components and accommodating multiple types of loads.

1 / D – Performance Objectives

Transfer rotational power efficiently, support radial and axial loads without excessive deflection, maintain precision rotation with minimal run out, provide secure component mounting, and resist fatigue failure.

2. Shafts

Shaft is a rotating machine element which is used to transmit power from one place to another.

<u>Types of shafts :</u>

1- **Transmission shafts**

   These shafts carry machine parts such as pulleys, gears, therefore they subject to bending in addition to twisting.

2- **Machine shafts**

   These shafts form an integral part of machine itself Grand shaft. Cam shaft

<u>Design of shafts</u>

The shafts may be designed on the basis of:

a. **Strength**
b. **Rigidity and stiffness**

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

a. **shafts subjected to twisting moment or torque only**
b. **Shafts subjected to bending moment only.**
c. **Shafts subjected to combined torsion and bending.**
d. **shafts subjected to axial loads in addition to c**

<u>a - shafts subjected to twisting moment</u>

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

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

T **: Torque**

d **: diameter of shaft**

τ**: allowable shear stress .**

$$\tau_{all} = \frac{\tau_{ult}}{s.f}$$

$$d = \sqrt[3]{\frac{16\,T}{\pi\ \tau}} \qquad \text{for solid shaft}$$

$$d_o = \sqrt[3]{\frac{16\,T}{\pi \times \tau \times (1-k^4)}} \qquad \text{for hollow shaft} \qquad k = \frac{di}{do}$$

The twisting moment (T) may be obtained by using the following relation :

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

If P in h.p then $T = \frac{P \times 45000}{2\pi N}$ **N.m**

If P in kW then $T = \frac{P*60000}{2\pi N}$ **N.m**

<u>Example (1) :</u>

A line shaft rotating at 200 r.p.m. is to transmit 20 kW. The shaft may be assumed to be made of mild steel with an allowable shear stress of 42 MPa. Determine the diameter of the shaft, neglecting the bending moment on the shaft.

Solution. Given : $N = 200$ r.p.m. ; $P = 20$ kW $= 20 \times 10^3$ W; $\tau = 42$ MPa $= 42$ N/mm^2

Let d = Diameter of the shaft.

We know that torque transmitted by the shaft,

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

We also know that torque transmitted by the shaft (T),

$$955 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 42 \times d^3 = 8.25\ d^3$$

$\therefore$ $d^3 = 955 \times 10^3 / 8.25 = 115\ 733$ or $d = 48.7$ say 50 mm **Ans.**

<u>Example (2) :</u>

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

Solution. Given : $P = 1$ MW $= 1 \times 10^6$ W ; $N = 240$ r.p.m. ; $T_{max} = 1.2\ T_{mean}$; $\tau = 60$ MPa $= 60$ N/mm^2

Let d = Diameter of the shaft.

We know that mean torque transmitted by the shaft,

$$T_{mean} = \frac{P \times 60}{2\pi N} = \frac{1 \times 10^6 \times 60}{2\pi \times 240} = 39\ 784 \text{ N-m} = 39\ 784 \times 10^3 \text{ N-mm}$$

$\therefore$ Maximum torque transmitted,

$$T_{max} = 1.2\ T_{mean} = 1.2 \times 39\ 784 \times 10^3 = 47\ 741 \times 10^3 \text{ N-mm}$$

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

$$47\ 741 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 60 \times d^3 = 11.78\ d^3$$

$\therefore$ $d^3 = 47\ 741 \times 10^3 / 11.78 = 4053 \times 10^3$

or $d = 159.4$ say 160 mm **Ans.**

Example (3) :

Find the diameter of a solid steel shaft to transmit 20 kW at 200 r.p.m. The ultimate shear stress for the steel may be taken as 360 MPa and a factor of safety as 8. If a hollow shaft is to be used in place of the solid shaft, find the inside and outside diameter when the ratio of inside to outside diameters is 0.5

Solution. Given : $P = 20$ kW $= 20 \times 10^3$ W ; $N = 200$ r.p.m. ; $\tau_u = 360$ MPa $= 360$ N/mm^2 ; $F.S. = 8$; $k = d_i / d_o = 0.5$

We know that the allowable shear stress,

$$\tau = \frac{\tau_u}{F.S.} = \frac{360}{8} = 45 \text{ N/mm}^2$$

Diameter of the solid shaft

Let d = Diameter of the solid shaft.

We know that torque transmitted by the shaft,

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

We also know that torque transmitted by the solid shaft (T),

$$955 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 45 \times d^3 = 8.84\, d^3$$

$\therefore$ $d^3 = 955 \times 10^3 / 8.84 = 108\,032$ or $d = 47.6$ say 50 mm **Ans.**

Diameter of hollow shaft

Let d_i = Inside diameter, and

d_o = Outside diameter.

We know that the torque transmitted by the hollow shaft (T),

$$955 \times 10^3 = \frac{\pi}{16} \times \tau\, (d_o)^3\, (1 - k^4)$$

$$= \frac{\pi}{16} \times 45\, (d_o)^3\, [1 - (0.5)^4] = 8.3\, (d_o)^3$$

$\therefore$ $(d_o)^3 = 955 \times 10^3 / 8.3 = 115\,060$ or $d_o = 48.6$ say 50 mm **Ans.**

and $d_i = 0.5\, d_o = 0.5 \times 50 = 25$ mm **Ans.**

b. shafts subjected to bending moment

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

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

$$I = \frac{\pi}{64}\, d^4$$

$$M = \frac{\pi}{32}\, \sigma_b\, d^3$$

$$d = \sqrt[3]{\frac{32M}{\pi \times \sigma_b}} \qquad \textit{For solid shaft}$$

σ_b **: Bending stress (Compression tension)**

$$d_o = \sqrt[3]{\frac{32M}{\pi \times \sigma_b \times (1-k^4)}} \quad \textit{For Hollow shaft}$$

Example (4) :

A pair of wheels of a railway wagon carries a load of 50 kN on each axle box, acting at a distance of 100 mm outside the wheel base. The gauge of the rails is 1.4 m. Find the diameter of the axle between the wheels, if the stress is not to exceed 100 MPa.

Solution. Given : $W = 50$ kN $= 50 \times 10^3$ N ; $L = 100$ mm ; $x = 1.4$ m ; $\sigma_b = 100$ MPa $= 100$ N/mm^2

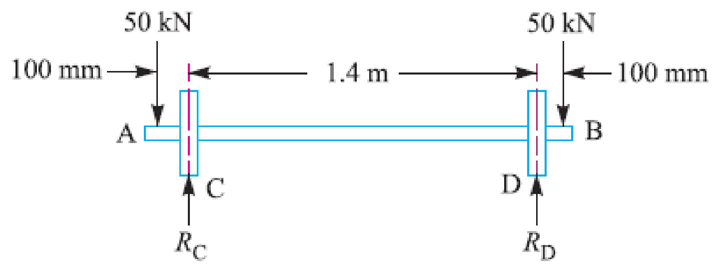


Fig. 14.1

The axle with wheels is shown in Fig. 14.1.

A little consideration will show that the maximum bending moment acts on the wheels at C and D. Therefore maximum bending moment,

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

Let d = Diameter of the axle.

We know that the maximum bending moment (M),

$$5 \times 10^6 = \frac{\pi}{32} \times \sigma_b \times d^3 = \frac{\pi}{32} \times 100 \times d^3 = 9.82\, d^3$$

$\therefore$ $d^3 = 5 \times 10^6 / 9.82 = 0.51 \times 10^6$ or $d = 79.8$ say 80 mm **Ans.**

C - shafts subjected to combined twisting moment and bending moment :

when the shaft is subjected to combined twisting moment and bending . Then the shaft must be designed on the basis of the two moment simultaneously , various theories have been suggested to account for elastic failure of the materials when they are subjected to various types of combined stresses .

The two theories .

1- **Max shear stress theory or Guests theory . it used for ductile material 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

σ_b : **Bending stress induced due to bending moment**

<u>According to theory (1) :</u>

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

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

Te = equiralent twisting moment

$$d = \sqrt[3]{\frac{16 \times Te}{\pi \times \tau_{max}}}$$

<u>According to theory (2) :</u>

maximum normal stress theory

$$\sigma_{b_{max}} = \frac{1}{2}\,\sigma_b + \sqrt{(\frac{1}{2}\,\sigma_b)^2 + (\tau)^2} = \frac{1}{2} \times \frac{32\,M}{\pi d^3} + \sqrt{(\frac{1}{2} \times \frac{32\,M}{\pi d^3})^2 \,.\, (\frac{16\,T}{\pi d^3})^2}$$

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

Me = **equivalent bending moment** $= \frac{1}{2}\,(M + \sqrt{M^2 + T^2})$

$$\sigma_{b_{\max}} = \sigma_{b_{all}}$$

$$Me = \frac{\pi}{32}\,\sigma_b\,d^3$$

$$d = \sqrt[3]{\frac{32Me}{\pi.\sigma_b}}$$ **for solid shaft**

$$Me = \frac{\pi}{32}\,d_o^3\,(1 - k^4)$$

$$d_o = \sqrt[3]{\frac{32Me}{\pi.\sigma_b(1-k^4)}}$$ **for hollow shaft**

<u>Example (5) :</u>

A solid circular shaft is subjected to bending moment of 3 x 10^6 N.mm and a torque of 10^7 N.mm The shaft is made of steel having ultimate tensile stress of 700 N/mm^2 and ultimate shear stress of 500 N/mm^2 . Assuming a factor of safety as (6), Determine the diameter of shaft .

$$\sigma_{b_{all}} = \frac{\sigma_{bult}}{S.F.} = \frac{700}{6} = 116.6\ N/mm^2$$

$$\tau_{all} = \frac{500}{6} = 83.3\ N/mm^2$$

$$Te = \sqrt{M^2 + T^2} = \sqrt{(3 \times 10^6)^2 + (10^7)^2}$$

$$Te = 10.44 \times 10^6\ \text{N.mm}$$

$$d = \sqrt[3]{\frac{16Te}{\pi.\tau_{all}}} = \sqrt[3]{\frac{16 \times 10.44 \times 10^6}{\pi \times 83.3}} = 86\ mm$$

Example (6):

A shaft made of mild steel is required to transmit 120 h.p at 300 r.p.m . The supported Length of the shaft is 3 m. It carries two pulleys each weighing 150 kg supported at a distance of lm from the ends respectively. Assuming the safe value of stress . determine the dia . Of shaft

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

$$T = \frac{120\times45\times10^6}{2\pi\times300} = 2.86 \times 10^6 \ N.mm$$

Max. Bending moment at C or D

M = 1500 × 1000 = 1.5 × 10^6 N.mm

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

$$Te = \sqrt{(1.5 \times 10^6)^2 + (2.86 \times 10^6)^2} = 3.2 \times 10^6 \text{ N .mm}$$

$$d = \sqrt[3]{\frac{16\,Te}{\pi\,\tau_{all}}}$$

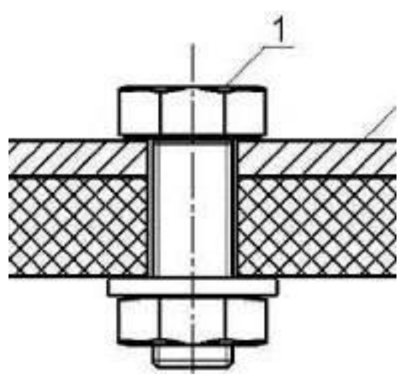
$$d = \sqrt[3]{\frac{16\times3.2\times10^6}{\pi\times60}} = 64.9 \text{ mm} \approx 65 \text{ mm}$$

Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

Learning package
In

Screwed Joints
For

Second year students



By

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2025

1. Overview

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

/ B –Rationale:-

Screws provide versatile, removable fastening solutions that allow for precise adjustment, easy maintenance access, and controlled clamping forces, making them ideal for applications requiring frequent assembly and disassembly operations.

1/ C –Central Idea:-

Threaded fasteners that create removable mechanical joints by converting rotational motion into linear motion, providing clamping force and adjustable positioning capabilities.

1 / D – Performance Objectives

Generate adequate clamping force for secure joints, allow for easy assembly and disassembly, resist loosening under vibration, provide precise positioning control, and maintain thread integrity over multiple uses

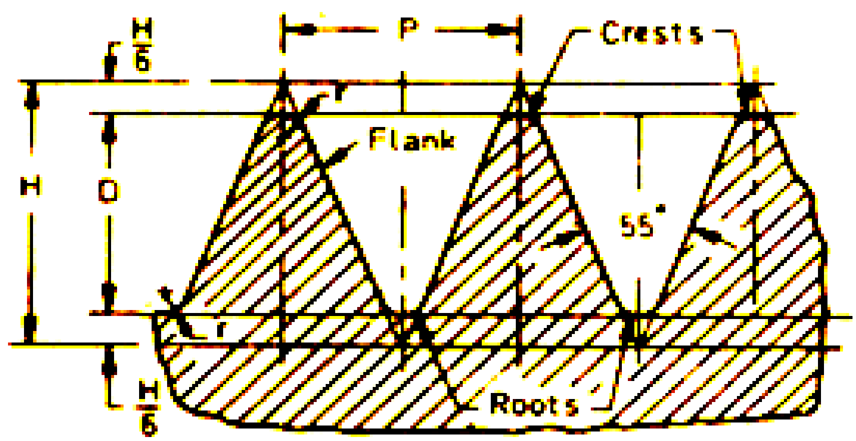
2. Screwed Joints

A screwed joint is mainly composed of two elements: bolt and nut. 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.

<u>Forms of screw threads :</u>

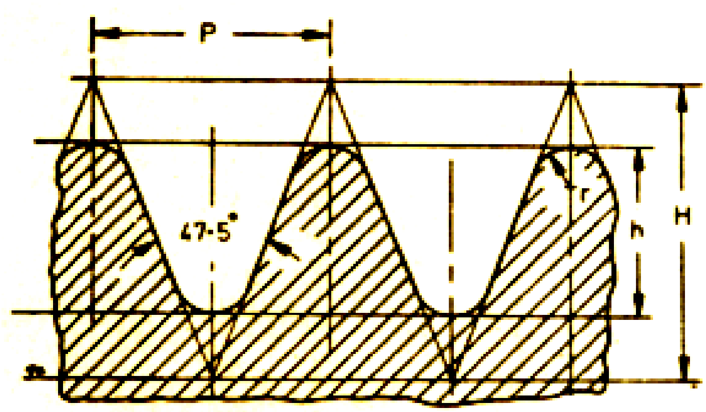
The following are the various forms of screw threads:

1. ***British standard whiteworth (B . S . W .) thread :***
   These threads are found on bolts and screwed fastenings for several purpose , and used for steel and iron pipes and tubes carrying fluids, in external pipe threading , the threads are specified by the bore of the pipe .



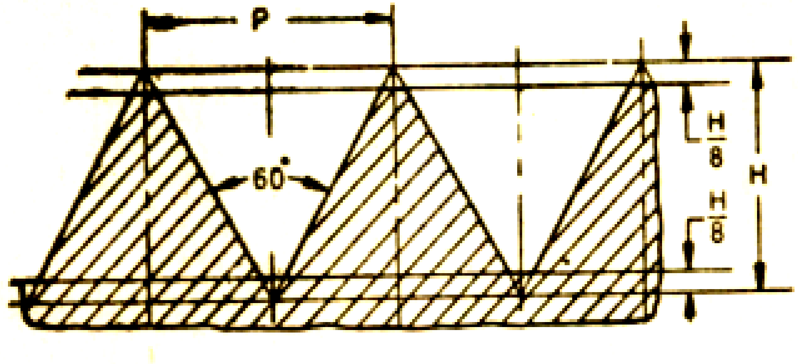
$H = 0{\cdot}96\,P$; $D = 0{\cdot}64\,P$; $r = 0{\cdot}1373\,P$

2. ***British Association (B.A.) thread :***
   These threads are used on screws for precision work .



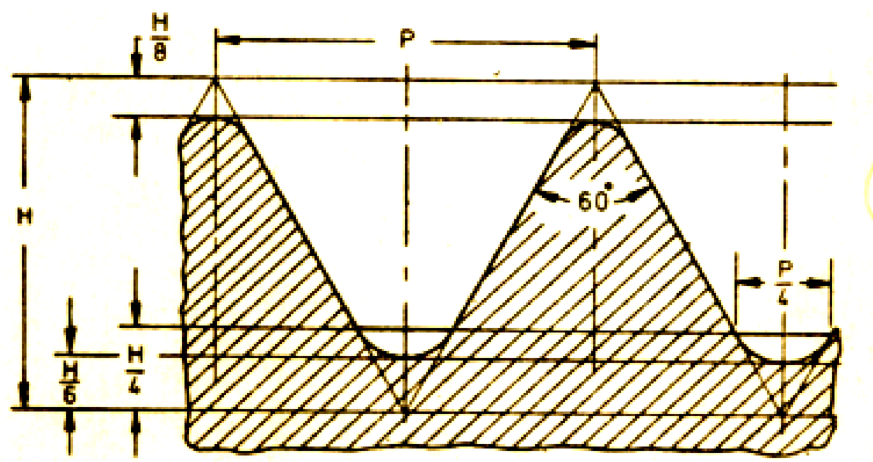
$H = 1{\cdot}13634P$; $h = 0{\cdot}6P$; $r = 0{\cdot}18083P$

3. ***American National standard thread :*** *These threads are used for general purposes e.g. on bolts , nuts , screws and tapped holes .*



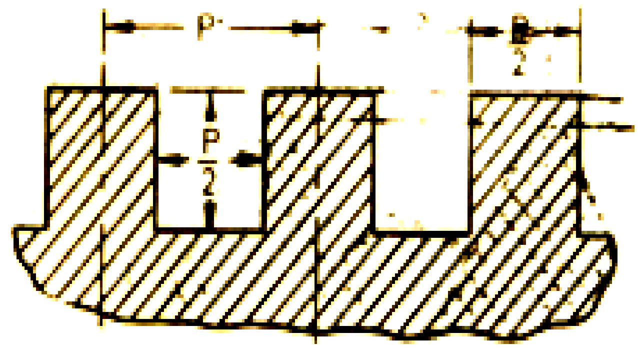
H = 0·866P

4. ***Unified standard thread :*** *This thread has rounded crests and roots , used for general purposes e.g. on bolts , nuts , screws.*

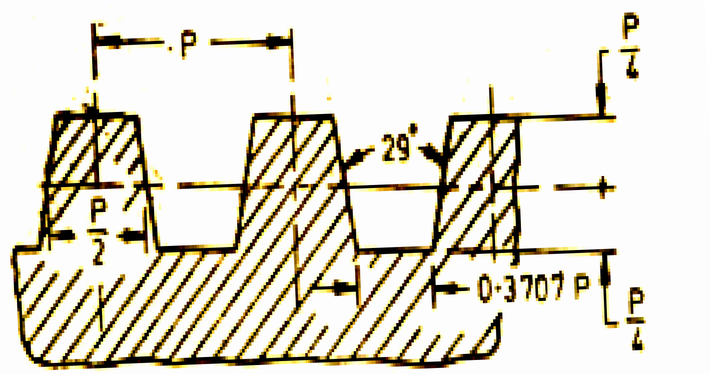


H = 0·866 P

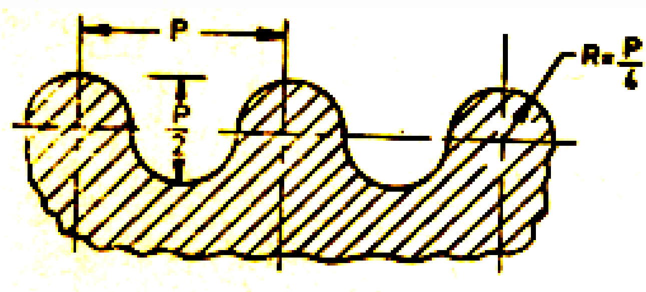
5. *square thread :*
   Because of their high efficiency , are widdey used for transmission of power in either direction such type of threads are usually found on the feed mechanisms of machine tools , valves , spindles , screw jacks



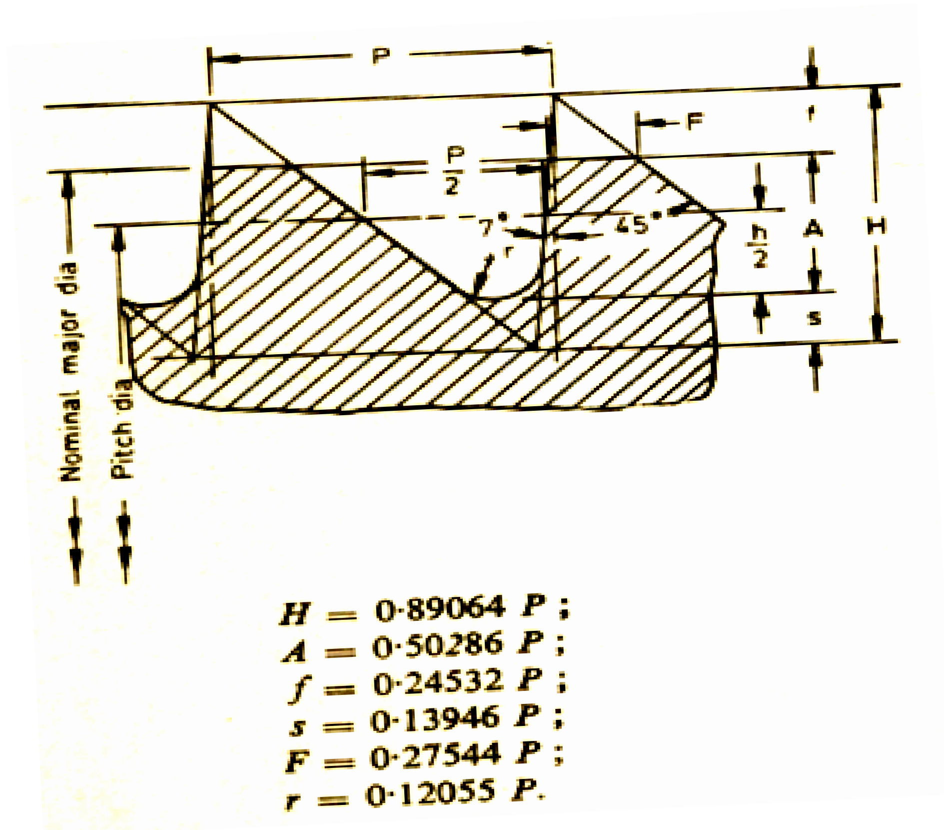
6. *Acme thread :*
   It is a modification of square thread . It is much stronger than square thread and can be easily produced . These threads are frequently used on screw cutting lathes, brass valves cock and bench vices .



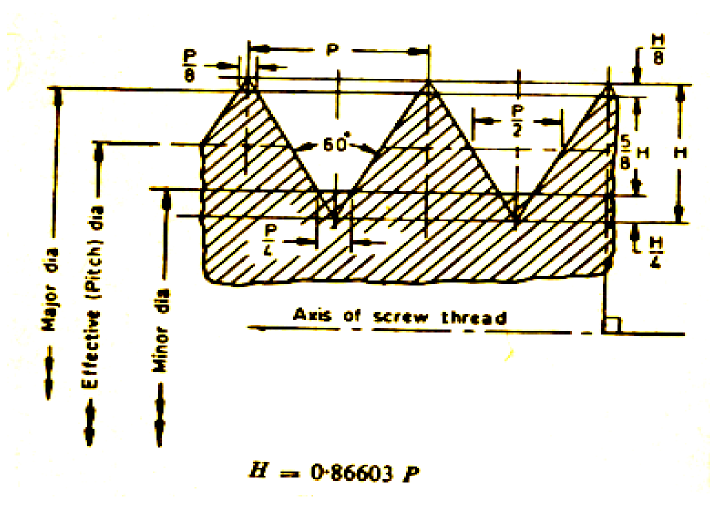
7. *Knuckle thread :*
   It is also a modification of square thread . These threads are used for rough and ready work . They are usually found on hydrants and large moulded insulators used in electrical trade , necks of glass bottles , railway carriage couplings .



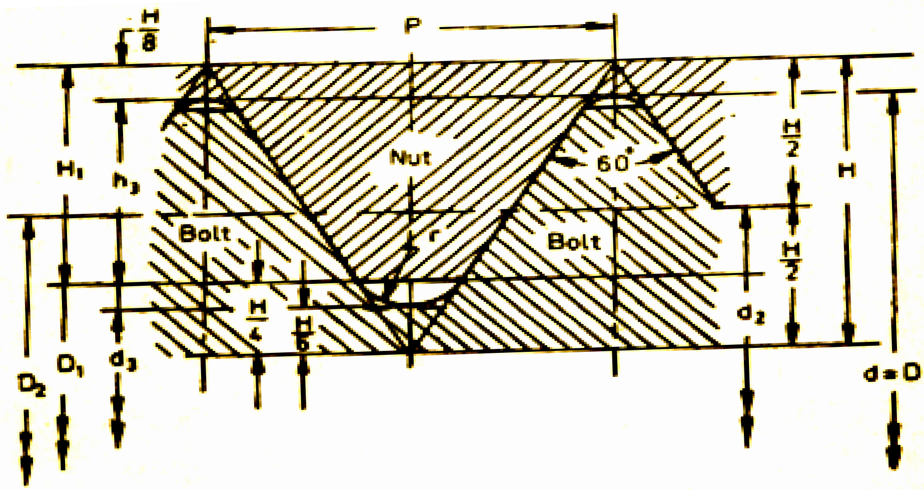
8. *Buttress thread :*
   It is used for transmission of power in one direction only . The spindles of bench vices are usually provided with buttress thread , because it has low frictional resistance .



9. ***Metric thread :***
It is an indian standard thread and is similar to B.S.W. thread , it has an included angle of 60º instead of 55º .



The basic profile of the thread



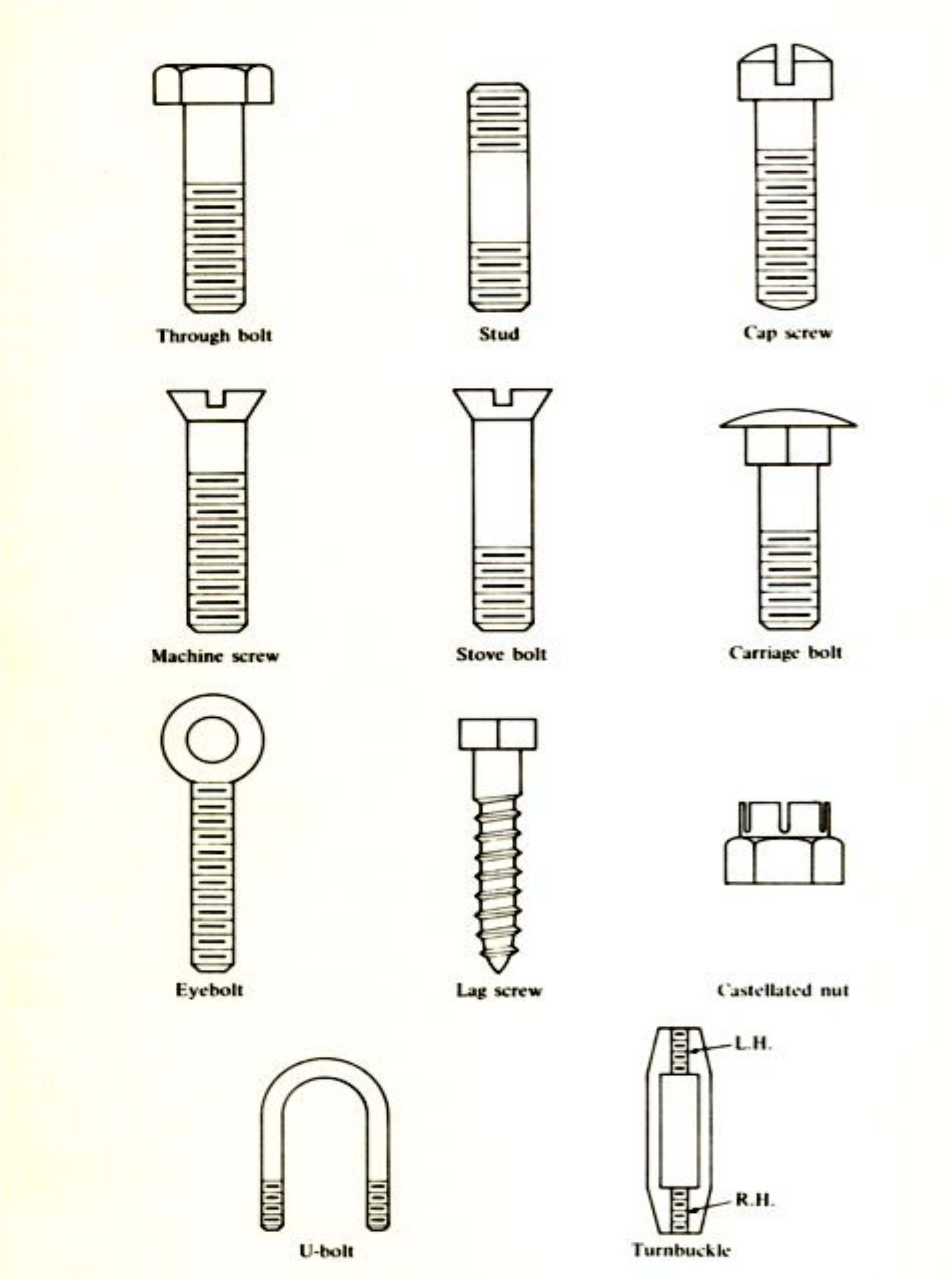
$H = 0{\cdot}86603\ P;\quad D_2 = d_2 = d - 0{\cdot}6495\ P;$

$D_1 = d - 1{\cdot}0825\ P;\ d_3 = d - 1{\cdot}2268\ P;$

$H_1 = \frac{5}{8} H;\quad h_3 = \frac{17}{24} H;\ r = \frac{H}{6}$

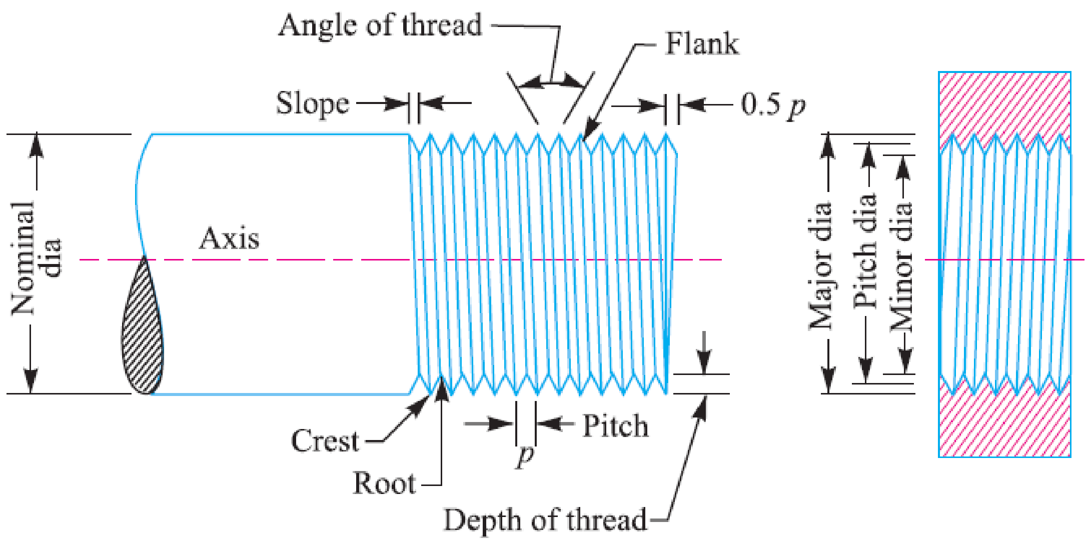
The design profile of the nut and bolt

Types of bolts :



<u>Stresses in screwed fastening due to static loading</u>

The following stresses in screwed fastening due to static loading are important from the subject point of view.



Terms used in screw threads.

1. ***<u>Intrnal stresses due to screwing up forces.</u>***

   a -Tensile stress due to stretching of the bolt.

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

   $P_i = 2\,840\ d$ ***(N)*** ***(in S.I.Units)***

   P_i ***= Initial tension in a bolt .***

   d = Nominal diameter of bolt , in (mm)

   $P_i = 284\ d$ ***(kg)*** ***If (d) in (cm)***

   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 (i .e. stress area)

   The stresses area may be obtained from Table 1 or it may be found by using the relation,

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

   d_p ***= Pitch diameter.***

   dc = Core or minor diameter.

b -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.

(c) Shear stress across the threads

The average thread shearing stress for the screw is obtained by using the relation :

$$\tau = \frac{P}{\pi \,.\, dc \,.\, b \,.\, n}$$

b = Width of the thread section at the root.

The average thread shearing stress for the nut is :

$$\tau = \frac{P}{\pi \,.\, d \,.\, b \,.\, n}$$

d = Major diameter.

d –Crushing stress on thread :

The crushing stress between the threads may be obtained by using the relation :

$$\sigma_{cr} = \frac{P}{\pi(d^2 - dc^2).n}$$

where:

d = Major diameter

d_c = Minor diameter

n = No. of threads in engagement.

e - Bending stress :

The bending stress induced in the shank of the bolt is given by:

$$\sigma_{bending} = \frac{x.E}{2.l}$$

where:

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

l = Length of the shank of the bolt

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

Table(1) : Design dimensions of screw threads , bolts and nuts

Designation	Pitch(mm)	Major or Nominal diameter Nut &Bolt (d=D)mm	Effective or pitch diameter Net & bolt (dp) mm	Minor or core diameter (dc)mm		Depth of Thread (bolt) mm	Stress area mm^2
				bolt	nut		
M1	0.25	1.000	0.838	0.693	0.729	0.153	0.460
M2	0.40	2.000	1.740	1.509	1.567	0.245	2.07
M4	0.70	4.000	3.545	3.141	3.242	0.429	8.78
M5	0.80	5.000	4.480	4.019	4.134	0.491	14.2
M6	1.00	6.000	5.350	4.773	4.918	0.613	20.1

M7	*1.00*	*7.000*	*6.350*	*5.773*	*5.918*	*0.613*	*28.9*
M8	*1.25*	*8.000*	*7.188*	*6.466*	*6.647*	*0.707*	*36.6*
M10	*1.50*	*10.00*	*9.026*	*8.160*	*8.876*	*0.920*	*58.3*
M12	*1.75*	*12.00*	*10.863*	*9.858*	*10.106*	*1.074*	*84.0*
M14	*2.00*	*14.00*	*12.701*	*11.546*	*11.835*	*1.227*	*115*
M20	*2.50*	*20.00*	*18.376*	*16.933*	*17.294*	*1.534*	*245*
M30	*3.50*	*30.00*	*27.727*	*25.706*	*26.211*	*2.147*	*561*
M42	*4.50*	*42.00*	*39.077*	*36.416*	*37.129*	*2.760*	*1.104*
M52	*5.00*	*52.00*	*48.752*	*45.795*	*49.587*	*3.067*	*1.755*
M60	*5.50*	*60.00*	*56.428*	*53.177*	*54.046*	*3.374*	*2.360*

<u>Example (1) :</u>

Determine the safe tensile load for a bolt of M 30, assuming a safe tensile stress of 42 MPa.

<u>Solution :</u>

Given : Size of bolt = M 30

∴ Major diameter of bolt : d = 30 mm

Safe tensile stress, σ_s = 42 N/mm^2

From Table (1) , stress area i.e. cross-sectional area at the bottom of the thread corresponding to M 30

$$= 561\ mm^2$$

∴ Safe tensile load = Stress area × σ_t

$$= 561 \times 42 = 23562\ N$$

Note: In this example, we have assumed that the bolt is not initially stressed.

<u>Example (2) :</u>

Two machine parts are fastened together tightly by means of a 14 mm tap bolt. If the load tending to separate these parts is neglected, find the stress that is set up in this bolt by the initial tightening.

<u>Solution :</u>

Given: Nominal diameter of the bolt , d = 14 mm

From Table (1) , the core diameter of the thread, corresponding to M 14.

d_e =11. 546 mm

Stress set up in the bolt by the initial tightening , Let σ_t = Stress set up in the bolt ,

We know that, initial tension in the bolt,

P = 2840 d = 2840 X 14= 39760 N

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

$$39760 = \frac{\pi}{4} \times 11.546^2 \times \sigma_t$$

$$\sigma_t = \frac{39760 \times 4}{\pi \times 11.546^2} = 379.94 \quad N/mm^2$$

2. <u>stress due to external forces.</u>

The following stresses are induced in a bolt when it is subjected to an external load.

1. ***Tensile stress***
2. ***Shear stress***
3. ***Combined tensile and Shear stress.***

We shall now discuss these stresses in detail as below:

<u>1. Tensile stress</u>

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

Let P = External load applied

d_e = Root or core diameter of the thread

σ_t = Permissible tensile stress for the bolt material.

$$P = \frac{\pi}{4} d_c{}^2 \times \sigma_t \quad \Longrightarrow \quad d_c = \sqrt{\frac{4P}{\pi \, \sigma_t}}$$

Now from Table (1) , for I.S.O. metric thread, the value of the nominal diameter of bolt corresponding to the value of d_c obtained or stress area $\left(\frac{\pi}{4} d_c{}^2\right)$ may be fixed.

<u>Notes</u> : 1. If the external load is taken up by a number of bolts, then

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

2. In case the standard table is not available, then for coarse threads

$d_c = 0.84\, d$ *d = Nominal diameter of bolt*

2. Shear stress

Sometimes , the bolts are used to prevent the relative movement of two or more parts, as in case of flange coupling , then the shear stress is induced in the bolts. The shear stresses should be avoided as far as possible. It should be noted that when the bolts are subjected to direct shearing loads, they should be located in such a way that the shearing loads comes upon the body (i.e. shank) of the bolt and not upon the threaded portion. In some cases , the bolts may be relieved of shear load by using shear pins. When a number of bolts are used to share the shearing load, the finished bolts should be fitted to the reamed holes.

Let : P_s ***= Shearing load carried by the bolt***

d = Major diameter of the bolt,

n = Number of bolts.

Then :

$$P_s = \frac{\pi}{4} d^2 \times \tau \times n \quad \Longrightarrow \quad d = \sqrt{\frac{4P_s}{\pi .\tau .n}}$$

3. Combined tension and shear stress

When the bolt is subjected to both tension and shear loads, as in case of coupling bolts or bearing , then the diameter of the shank of the bolt is obtained from the shear load and that of threaded part from the tensile load. A diameter slightly larger, than that required for either shear or tension can be assumed and stresses due to combined load should be checked for the following principal stresses.

Maximum principal shear stress,

$$\sigma_{s(\max)} = \sqrt{\tau^2 + \left(\frac{\sigma_t}{2}\right)^2}$$

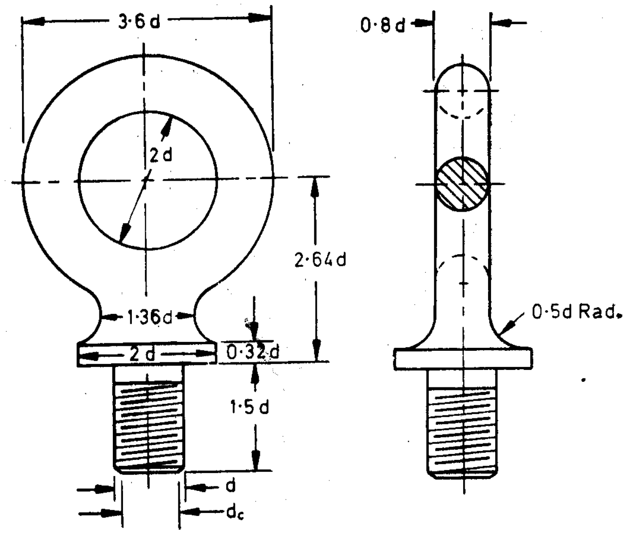
and maximum 0b principal tensile stress.

$$\sigma_{t(\max)} = \frac{\sigma_t}{2} + \sqrt{\tau^2 + \left(\frac{\sigma_t}{2}\right)^2}$$

These stresses should not exceed the safe permissible values of the stresses.

Example (3) :

An eye bolt is to be used for lifting a load of 60 kN. Find the nominal diameter of the bolt if the tensile stress is not to exceed 100 N/mm². Assume coarse threads.



Solution :

Given: Load to be lifted , P = 60 kN

= 60 X 10³ N

Permissible tensile stress , σ_t = 100 N/mm²

Let d = Nominal diameter of the bolt

d_c = Core diameter of the bolt,

Using the relation

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

$$d_e = \sqrt{\frac{4P}{\pi\,\sigma_t}} = \sqrt{\frac{4 \times 60 \times 10^3}{\pi \times 100}}$$ ***= 27.64= 33 mm***

<u>Example (4) :</u>

Two shafts are connected by means of a flange coupling to transmit torque of 25 N. m. The flanges of the coupling are fastened by four bolts of the same material at a radius of 30 mm.

Find the size of the bolts if the allowable shear stress for the bolt material is 30 N/mm².

Solution. Given : $T = 25$ N-m $= 25 \times 10^3$ N-mm ; $n = 4$; $R_p = 30$ mm ; $\tau = 30$ MPa $= 30$ N/mm²

We know that the shearing load carried by flange coupling,

$$P_s = \frac{T}{R_p} = \frac{25 \times 10^3}{30} = 833.3 \text{ N} \qquad ...(i)$$

Let d_c = Core diameter of the bolt.

∴ Resisting load on the bolts

$$= \frac{\pi}{4}(d_c)^2 \tau \times n = \frac{\pi}{4}(d_c)^2\ 30 \times 4 = 94{\cdot}26\,(d_c)^2 \qquad ...(ii)$$

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

$$(d_c)^2 = 833.3 / 94.26 = 8.84 \qquad \text{or} \qquad d_c = 2.97 \text{ mm}$$

From Table 1 (coarse series), we find that the standard core diameter of the bolt is 3.141 mm and the corresponding size of the bolt is M 4. **Ans.**

Homework :

1 – Determine the safe tensile load for bolts of M 20 and M 36. Assume that the bolts are not initially stressed and take the safe tensile stress as 200 MPa. [Ans. 49 kN; 16.43 kN]

2 - An eye bolt carries a tensile load of 20 kN. Find the size of the bolt, if the tensile stress is not to exceed 100 MPa. Draw a neat proportioned figure for the bolt. [Ans. M 20]

3 –An engine cylinder is 300 mm in diameter and the steam pressure is 0.7 N/mm²^2***. If the cylinder head is held by 12 studs, find the sizeof the stud. Assume safe tensile stress as 28 MPa. [Ans. M 24]***

4 ***– Find the size of 14 bolts required for a C.I. steam engine cylinder head. The diameter of the cylinder is 400 mm and the steam pressure is 0.12 N/mm2. Take the permissible tensile stress as 35 MPa. [Ans. M 24]***

Ministry of high Education and Scientific Research
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Department of Electronic Techniques

Learning package
In
Springs
For

Second year students

By

Mr. Saleh Kudhair Jebur
Assistant Lecturer
Dep. Of Mechanical Techniques
2025

1. Overview

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

/ B –Rationale:-
Springs are crucial for energy storage, shock absorption, and force control in mechanical systems, providing essential functions such as maintaining contact pressure, returning components to neutral positions, and isolating vibrations.

1/ C –Central Idea:-
Mechanical systems, providing essential functions such as maintaining contact pressure, returning components to neutral positions, and isolating vibrations.

1 / D – Performance Objectives
Store and release energy efficiently, provide consistent force deflection characteristics, absorb shock and vibration, maintain elastic properties over millions of cycles, and return to original shape after load removal.

2. Springs

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

<u>The various important applications of springs are :</u>

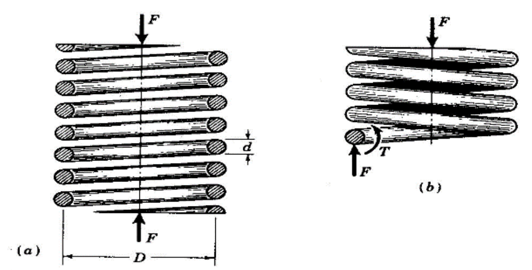
1 – To apply forces , as in brakes and clutches and spring loaded valves .

2 – To measure forces , as in spring balances .

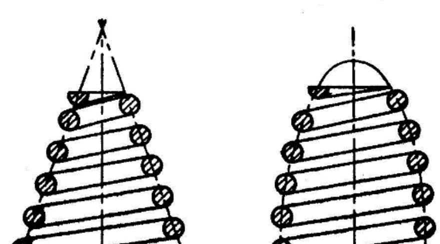
3 – To store energy , as in watch springs .

4 – To absorb shock and vibrations as in car springs and railway buffer.

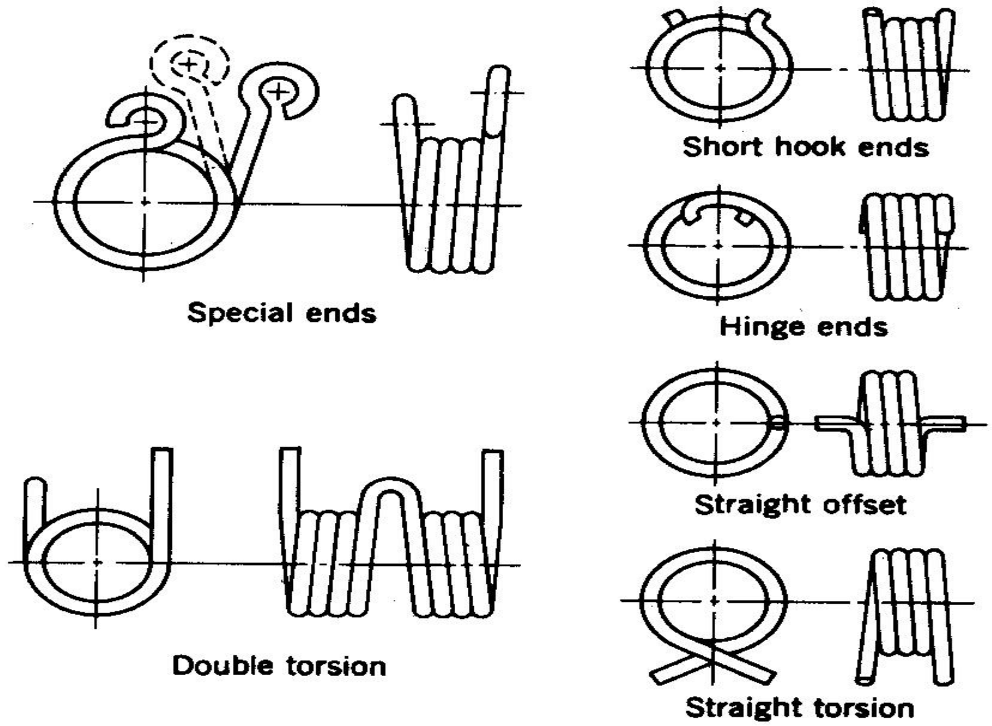
<u>Types of springs :</u>



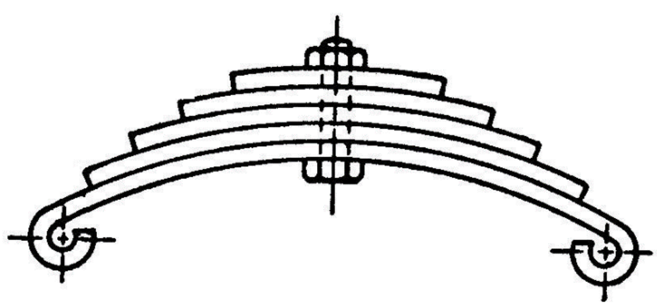
1 – Helical springs .



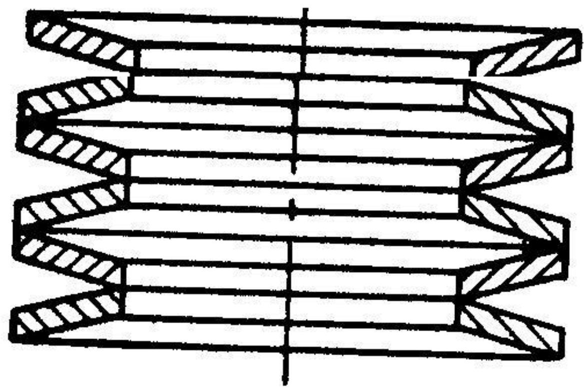
2 – Conical and volute springs .



3 – Torsion springs .

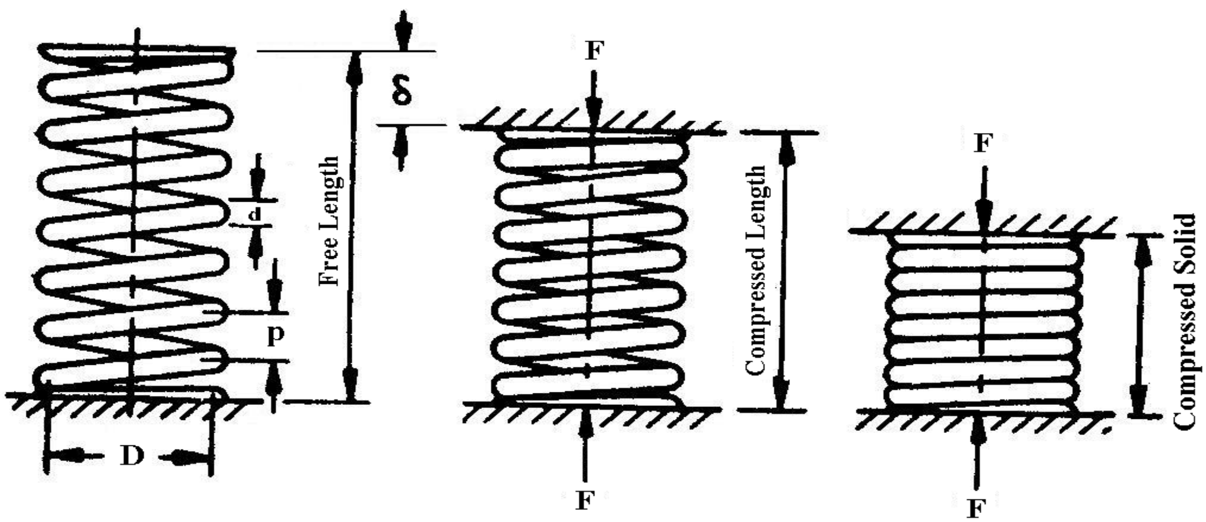


4 – Laminated or leaf springs.



5 *– Disc springs.*

<u>Terms used in connection with compression spring :</u>



Spring index : is defined as the ratio of the diameter of coil to the diameter of wire .

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

C : spring index
D : diameter of coil $= D_o - d$
d : diameter of wire
D_o ***: outside diameter of coil***

Spring rate : (stiffness OR spring constant : is defined as the load required per unit deflection of spring .

$$K = \frac{F}{\delta}$$

K : spring rate

F : applied load

δ :deflection of the spring

Total active length of spring wire (L)

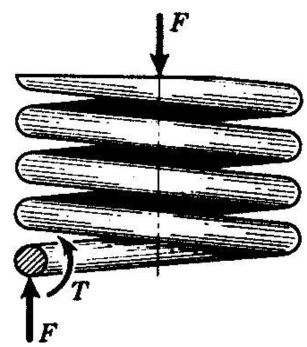
$$L = \pi.D.n$$

n : number of active turns

<u>Stresses in helical spring of circular wire :</u>

<u>1 – shear stress induced in the wire due to the twisting moment :</u>

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



<u>2 – direct shear stress due to the load (F) :</u>

$$\tau = \frac{F}{\frac{\pi}{4}.\, d^2}$$

<u>3 – Stress due to curvature of spring :</u>

a – neglecting the curvature effect :

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

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

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

$$\tau = \frac{F \times \frac{D}{2} \times \frac{d}{2}}{\frac{\pi}{32} \times d^4} = \frac{\frac{F.D.d}{4}}{\frac{\pi.d^4}{32}} = \frac{8FD}{\pi\ d^3}$$

b – considering the curvature effect :

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$ ***When K : Wahl's correction factor***

$$\tau = K \times \frac{8 . F . D}{\pi . d^3}$$

Deflection of helical spring of circular wire :

G : modulus of rigidity

δ : deflection of the spring

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

ملاحظات :

1 – في حالة وجود قيمة (C) في السؤال ، نحول القوانين بدلالة (C) وكما يأتي :

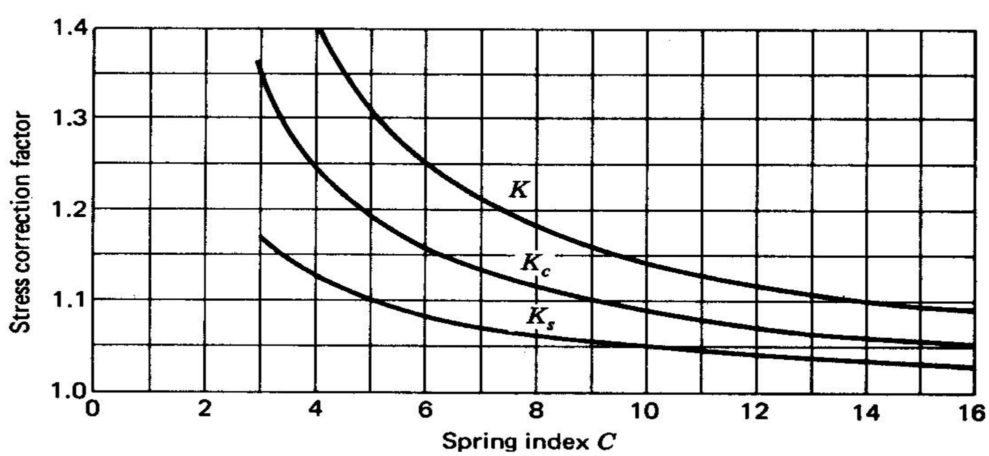
$$\tau = \frac{8 . F . D}{\pi . d^3} = \frac{8 . F . C}{\pi . d^2}$$

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

2 – اذا كان المطلوب في السؤال تصميم نابض (design spring) فأنه يتطلب ايجاد (p , n , D , d , l .)

3 – ان النابض يتعرض الى الانحناء عندما تكون : l > 4 D.

The relationship between spring index and stress correction factor :



Example (1) :

Design a helical compression spring for maximum load of 1000 N , with a deflection of 25 mm using the value of spring index as 5 , G = modulus of rigidity = 84000 N/mm^2 , maximum permissible shear stress for spring wire is 420 N / mm^2 .

Solution :

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} \qquad\qquad K = \frac{4\times5-1}{4\times5-4} + \frac{0.615}{5} = 1.31$$

$$\tau = K \times \frac{8.F.C}{\pi.d^2}$$

$$420 = 1.31 \times \frac{8\times1000\times5}{\pi.d^2} \quad \Rightarrow \quad d = 6.3 \quad mm$$

$$C = \frac{D}{d} \quad \Rightarrow \quad 5 = \frac{D}{6.3} \quad \Rightarrow \quad D = 31.5 \quad mm$$

$$\delta = \frac{8.F.\ C^3.\ n}{G.d}$$

$$25 = \frac{8\times1000\times\ 5^3.\ n}{84000\times6.3} \quad \Rightarrow \quad n = 13.2 \quad turn = 14 \quad turn$$

n' = n + 2 = 14 + 2 = 16

free length (Lf) = n' . d + δ + (n' – 1) 0.1

= 16 X 6.3 + 25 + (16 – 1) X 0.1 = 140.8 mm

Pitch of spring (p) = (Lf) / (n' – 1) = 140.8 / (16 – 1) = 9.4 mm

Example (2) :

A helical spring is made from a wire of 6 mm diameter and has outside diameter of 7.5 cm , if the permissible shear stress is 3500 kg / cm^2 , the modulus of rigidity 8.4 * 10^5 kg / cm^2 , find the axial load which the spring can carry , and the deflection per active turn :

a – neglecting the effect of curvature.

b – Considering the effect of curvature.

Solution :

d = 6 mm = 0.6 cm , D_o = 7.5 cm , $\tau = 3500$ kg / cm², G = 8.4 * 10^5 kg / cm² ,

D = D_o – d = 7.5 – 0.6 = 6.9 cm .

a - neglecting the effect of curvature:

$$\tau = \frac{8 . F . D}{\pi . d^3} \Rightarrow F = \frac{\pi \times \tau \times d^3}{8 \times D} = \frac{3.14 \times 3500 \times (0.6)^3}{8 \times 6.9} = 43 \quad kg$$

$$\frac{\delta}{n} = \frac{8 . F . D^3}{G . d^4} = \frac{8 \times 43 \times (6.9)^3}{8.4 \times 10^5 \times (0.6)^4} = 1.038$$

b – considering the effect of curvature :

$$C = \frac{D}{d} = \frac{6.9}{0.6} = 11.5$$

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} \qquad K = \frac{4 \times 11.5 - 1}{4 \times 11.5 - 4} + \frac{0.615}{11.5} = 1.123$$

$$\tau = K \times \frac{8 . F . D}{\pi . d^3} \quad \Rightarrow \quad F = \frac{\pi \times \tau \times d^3}{8 \times K \times D} = \frac{3.14 \times 3500 \times (0.6)^2}{8 \times 1.123 \times 6.9} = 38.3 \quad kg$$

$$\frac{\delta}{n} = \frac{8 . F . D^3}{G . d^4} = \frac{8 \times 38.3 \times (6.9)^3}{8.4 \times 10^5 \times (0.6)^4} = 0.9245$$

Example (3) :

Design a compression helical spring to carry a load of 50 kg with a deflection of 2.5 cm the spring index may be taken as (8) , Assum the following data for the spring material

Permissible shear stress = 3500 kg / cm²

Modulus of rigidity = 8.4 X 10^5 kg / cm²

Solution :

$$\text{Wahl's factor} \quad K=\frac{4C-1}{4C-4}+\frac{0.615}{C} \qquad K=\frac{4\times 8-1}{4\times 8-4}+\frac{0.615}{8}=1.184$$

$$\tau = K \times \frac{8.F.C}{\pi . d^2}$$

$$3500=1.184 \times \frac{8\times 50\times 8}{3.14 . d^2} \quad \Rightarrow \quad d=0.58 \quad cm$$

$$C=\frac{D}{d} \quad \Rightarrow \quad 8=\frac{D}{0.58} \quad \Rightarrow \quad D=4.697 \quad cm$$

$$\delta=\frac{8.F.\ C^3.\ n}{G.d}$$

$$2.5=\frac{8\times 50\times\ 8^3.\ n}{8.4\times 10^5\times 0.58} \quad \Rightarrow \quad n=6.1 \ \ turn \ =7 \ \ turn$$

n' = n + 2 = 7 + 2 = 9

free length (Lf) = n' . d + δ ***+ (n' – 1) 0.1***

= 9 X 0.58 + 2.5 + (9 – 1) X 0.1 = 1.578 cm

Pitch of spring (p) = (Lf) / (n' – 1) = 1.578 / (9 – 1) = 0.197 cm

Example (4) :

Compute the deflection of helical spring to withstand 500 N , by 12 turns , if you know that the diameter of coil is 100 mm and the diameter of wire is 10 mm , using the modulus of rigidity as 80 Gpa .

Solution :

$$\delta=\frac{8.F.\ D^3.\ n}{G.d^4}=\frac{8\times 500\times (100)^3\times 12}{80\times 10^3\times (10)^4}=60 \quad mm$$

Example (5) :

Findout the number of turns for helical spring subjected to 500 N force , if the spring rate is 18.898 N / mm and the coil diameter is 126 mm with the wire diameter of 12.6 mm taking G = 84 Gpa .

Solution :

Spring rate (K) :

$$K = \frac{F}{\delta} \quad \Rightarrow \quad \delta = \frac{F}{K} = \frac{500}{18.898} = 26.457 \quad mm$$

$$\delta = \frac{8 . F . \ D^3 . \ n}{G . d^4} \quad \Rightarrow \quad n = \frac{\delta \ . \ G . \ d^4}{8 . F . \ D^3} = \frac{26.457 \times 84000 \times (12.6)^2}{8 \times 500 \times (126)^3} = 7 \ \ turns$$

***Homework :** Do as required in the table below depending on the given data :*

serial	*Given data*	*Required*
1	***D = 300 mm , d = 30 mm F = 5 KN , n = 6 turns , G = 80 GPa***	σ_S , δ , ***Lf***
2	***D = 14 cm , d = 12 mm ,*** $\sigma_S = 100 KN$	***F***
3	***F = 2750 N ,*** δ ***= 6 mm , C= 5 ,*** σ_S ***=420 N / mm²*** ***, G = 84 KN / mm²***	***Design the spring : D , d , n , Lf , p***
4	***D = 40 mm , C = 5 ,*** σ_S ***= 80 N / mm², G = 8.4 KN / mm², K = 23 N / mm***	***Chack whether the spring under the curvature or not .***

Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

Learning package
In

Clutches

For

Second year students

By

Mr. Saleh Kudhair Jebur
Assistant Lecturer
Dep. Of Mechanical Techniques
2025

1. Overview

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

/ B –Rationale:-

Clutches enable controlled engagement and disengagement of power transmission, allowing for smooth starting, stopping, and speed changes while protecting machinery from overload conditions and enabling selective operation of multiple driven components.

1/ C –Central Idea:-

Mechanical devices that selectively engage and disengage power transmission between driving and driven components, allowing controlled connection and disconnection of rotating elements

.1 / D – Performance Objectives

Provide smooth engagement and disengagement of power transmission, handle specified torque loads without slippage, operate reliably under varying speeds and loads, minimize wear during operation, and respond quickly to control inputs.

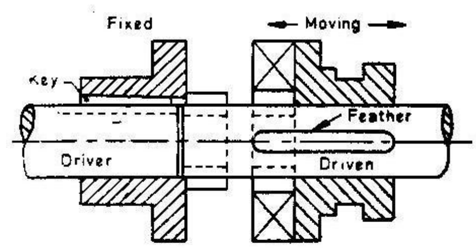
2. Clutches

A clutch is a machine member used to connect a driving shaft to a driven shaft so that the driven shaft may be started OR stopped at without stopping the driving shaft .

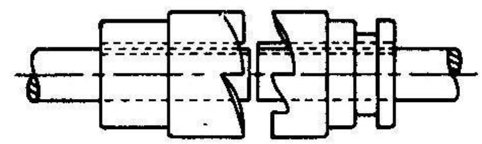
The uses of the clutch is mostly found in automobiles

Types of clutches :

1. **Positive clutches (figure (a & b))**
2. **Friction clutches (figure (C))**



(a)



(b)

The friction clutches are :

- **Plate (Disc) clutches (figure (C))**

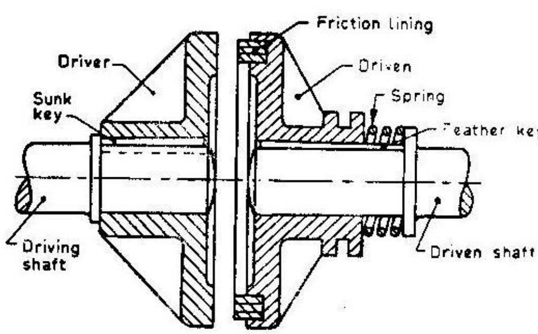
The disc clutch may be a single disc clutch OR multiple disc clutch . Since bolt sides of each disc are normally effective , then

n = number of pairs of contact surface

for single disc clutch n = 2

for multiple disc clutch $n = n_1 + n_2 - 1$

n_1 n_2 = number of disc on drining and driven shaft



(C)

- **<u>Considering uniform pressure</u>**

Pr : intensity of Pressure

$$Pr = \frac{F}{\pi\,(r_1^2 - r_2^2)}$$

F : Axial thrust with which the contact surface are held together

r_1 , r_2 : External and internal radii of Friction faces .

T : Torque transmitted [Total friction torque]

$$T = n.\mu.F.r$$

μ : Coefficient of friction

r : Mean radius of the friction face $r = \frac{2}{3}\left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right)$

- **<u>Considering uniform wear</u>**

$Pr.x = C$ **[constant]**

x : distance from the axis of the clutch

$$F = 2\pi C(r_1 - r_2)$$

$$"'r = \frac{r_1 + r_2}{2}$$

Notes

a. **In case of a new clutch the intensity of pressure approximately uniform .**
   In case of an old clutch the uniform wear theory is more approximate.

b. **The uniform pressure theory gives a higher friction torque than wear theory .**

<u>Example (1) :</u>

A Friction clutch is to transmit 15 h.p at 3000 r.p.m. it is to be of single plate type with both sides of the plate effective . The axial pressure being Limited to 0.9 kg/cm^2 . If the external diameter of friction Lining is 1.4 times the internal diameter . find the required dimensions of friction Lining . Assume uniform wear conditions . The coefficient of friction may be taken as 0.3

<u>Solution :</u>

$$T = \frac{P}{2\pi N} = \frac{15 \times 4500}{2 \times 3.14 \times 3000} = 3.58\ kg.cm$$

$Pr.x = C$ **[Pr is maximum at r_2]**

$$Pr.r_2 = C$$

$$C = 0.9\, r_2$$

$$F = 2\pi C(r_1 - r_2) \quad = 2\pi\, 0.9\, r_2(1.4\, r_2 - r_2) \; = 2.2\, r_2^2$$

$$T = n\mu F\left(\frac{r_1 + r_2}{2}\right)$$

$$358 = 2\ \times 0.3\ \times 2.2\, r_2^2\left(\frac{1.4\, r_2 + r_2}{2}\right) = 1.356\, r_2^2\left(\frac{2.4\, r_2}{2}\right)$$

$$= 1.627\, r_2^3$$

(n=2 for both sides of plate effective)

$r_2^3 = 220$ ………. $r_2 = 6.07\ cm$ …………… $r_1 = 1.4 \times 6.07 = 8.45\ cm$

<u>Example (2) :</u>

A single disc clutch with both sides of the disc effective is used to transmit 12 horse power at 900 r.p.m. The axial pressure is limited to 0.85 kg/cm^2 . If the external diameter of the friction lining is 1.25 times to the internal diameter,Find the required dimensions of the friction

lining. Assume uniform wear conditions . The coefficient of friction may be taken as 0.3 .

<u>Solution :</u>

$$T = \frac{P*4500}{2\pi N} = \frac{12*4500}{2\pi*900} = \frac{54000}{2\pi*900} = 9.549\ kg.m = 955.4\ \mathrm{kg.cm}$$

$$C = Pr.r_2 = 0.85\ \mathrm{r}_2$$

$$F = 2\pi c(r_1 - r_2) = 2\pi * 0.85 r_2(1.25\, r_2 - r_2) = 1.335\, r_2^2$$

$$r = \frac{r_1+r_2}{2} = \frac{1.25\, r_2+r_2}{2} = 1.125\, r_2$$

$$T = n.\mu.F.r$$

$$955.4 = 2 * 0.3 * 1.335\, r_2^2 * 1.125\, r_2$$

$$r_2^3 = 1061.11\ \ \ldots\ldots\ldots\ \ \ r_2 = 10.195\ cm$$

$$r_1 = 12.74\ cm$$

$$F = 2\pi C(r_1 - r_2) = 2 * 3.14 * 0.85 * 10.195(12.74 - 10.195)$$

$$= 138.788\ kg$$

<u>Example (3) :</u>

Multi – disc clutch has three discs on the driving shaft and two on the driven shaft . the outside diameter of the contact surface is 240mm and inside diameter 120 mm Assuming uniform wear and coefficient of friction 0.3 find the max . axial intensity of pressure between the discs for transmitting 25kw at 1575 r.p.m.

<u>Solution :</u>

$$T = \frac{P}{2\pi N} = \frac{25 \times 10^3 \times 60}{2\pi \times 1575} = 151.6\ N.m = 151.6 \times 10^3\ N.mm$$

N=n1 + n2 – 1 = 3 + 2 – 1 = 4

$$T = n.\mu.F.r\ = n.\mu.F\left(\frac{r_1+r_2}{2}\right)$$

$$F = \frac{2T}{n.\mu(r_1 + r_2)} = \frac{2 \times 151.6 \times 10^3}{4 \times 0.3(120 + 60)} = 1404\ N$$

$$Pr.r_2 = C$$

$$C = Pr.r_2 = 60\ Pr$$

$$F = 2\pi c\ (r_1 - r_2)$$

$$1404 = 2\pi \times 60 \times Pr\ (120 - 60)$$

$$1404 = 7200\pi \times Pr$$

$$Pr = \frac{1404}{7200\pi} = 0.062\ N/mm^2$$

Homework :

1- A single plate clutch with both sides of the plate effective is required to transmit 25 kw at 1600 r.p.m. the outer diameter of the plate is Limited to 300 mm and the intensity of pressure between the plates not to exceed 0.07 N/mm^2 . Assuming uniform wear and coefficient of friction 0.3 , find the inner diameter of the plates and the axial force necessary to engage the clutch .

2- A multiple disc clutch has three discs on the driving shaft and two on the driven shaft , providing four pairs of contact surface . The outer diameter of the contact surface is 25 cm and the inner diameter is 15 cm . Determine the Max . axial intensity of pressure between the discs for transmitting 25 h.p power at 500 r.p.m. Assuming uniform wear and coefficient of friction as 0.3

3- A single plate clutch with both sides of the plate effective . The axial . Pressure being limited to 0.13 N/mm^2 . the outer diameter of the contact surface is (250mm) and the inner diameter (150 mm) . the M = 0.3 find the power transmitted by clutch at (500r.p.m) . Assuming uniform wear

Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

Learning package
In
Gears Design
For

Second year students

By

Mr. Saleh Kudhair Jebur
Assistant Lecturer
Dep. Of Mechanical Techniques
2025

1. Overview

1 / A –Target population:-

For students of Second year
Technological institute of Basra
Dep. Of Mechanical Techniques

/ B –Rationale:-
Gears provide precise, efficient power transmission with exact speed ratios and torque multiplication, making them essential for applications requiring accurate motion control, high efficiency, and reliable performance under varying load conditions

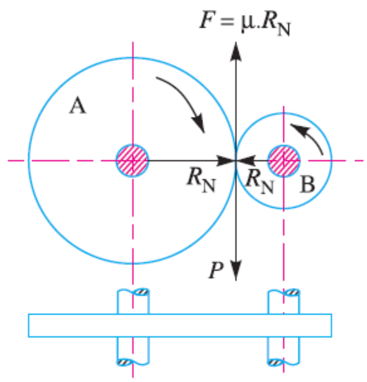
1/ C –Central Idea:-

Toothed mechanical elements that transmit motion and power between rotating shafts through positive engagement, providing precise speed ratios and torque multiplication or reduction.
1 / D – Performance Objectives

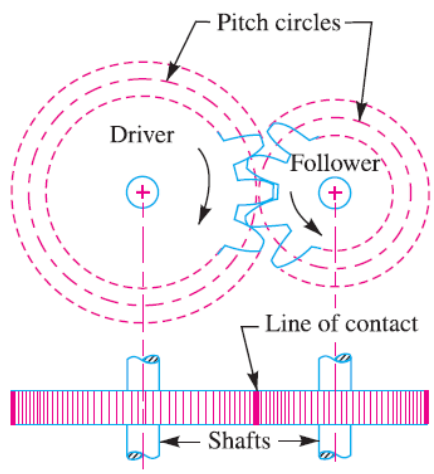
Transmit power with high efficiency and accuracy, maintain constant speed ratios, handle specified loads without tooth failure, operate smoothly with minimal noise and vibration, and provide long service life with proper lubrication.

2. Gears Design

To avoid the slipping, and increase the efficiency in transmitting power in the belts, or sometimes the distance between the two center of shafts in too small a gear wheel or toothed wheel will be used as shown below :

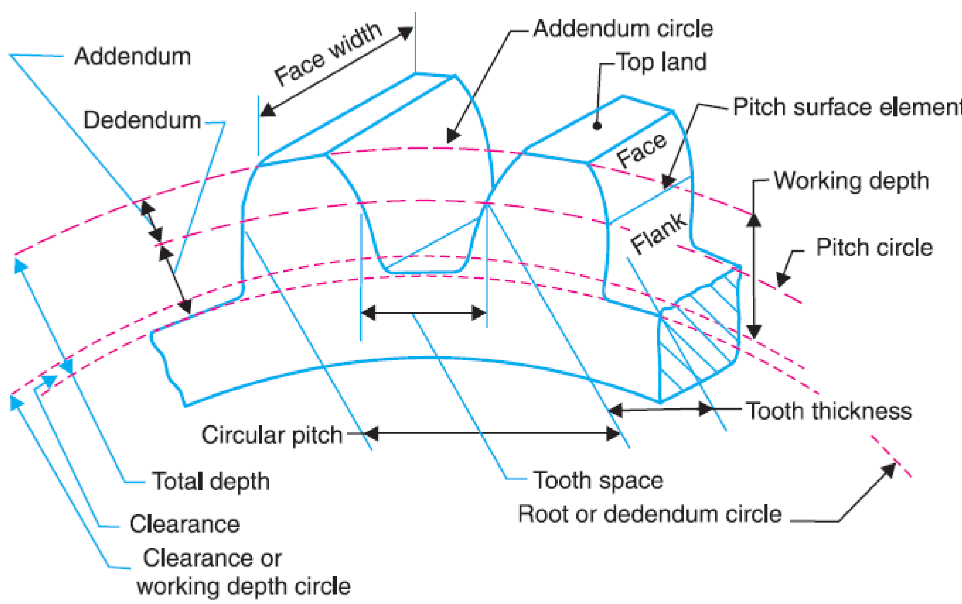


Friction wheels.



Gear or toothed wheel.

<u>Terms used in gears :</u>



Terms used in gears.

1. ***Pitch circle***. It is an imaginary circle which by pure rolling action, would give the same motion as the actual gear.

2. *Pitch circle diameter*. It is the diameter of the pitch circle. The size of the gear is usually specified by the pitch circle diameter. It is also known as ***pitch diameter***.

3. *Pitch point*. It is a common point of contact between two pitch circles.

4. *Pitch surface*. It is the surface of the rolling discs which the meshing gears have replaced at the pitch circle.

5. *Pressure angle or angle of obliquity*. It is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point. It is usually denoted by φ. The standard pressure angles are 14 1/2 ° and 20°.

6. *Addendum.* It is the radial distance of a tooth from the pitch circle to the top of the tooth.

7. *Dedendum.* It is the radial distance of a tooth from the pitch circle to the bottom of the tooth.

8. *Addendum circle*. It is the circle drawn through the top of the teeth and is concentric with the pitch circle.

9. *Dedendum circle*. It is the circle drawn through the bottom of the teeth. It is also called root circle.

Note : Root circle diameter = Pitch circle diameter × cos φ, where φ is the pressure angle.

10. *Circular pitch*. It is the distance measured on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth. It is usually denoted by *pc*.

Mathematically,

Circular pitch, $p_c = \pi D/T$

where D = Diameter of the pitch circle, and

T = Number of teeth on the wheel.

A little consideration will show that the two gears will mesh together correctly, if the two wheels have the same circular pitch.

Note : If D_1 and D_2 are the diameters of the two meshing gears having the teeth T_1 and T_2 respectively, then for them to mesh correctly,

$$p_c = \frac{\pi D_1}{T_1} = \frac{\pi D_2}{T_2} \quad \text{or} \quad \frac{D_1}{D_2} = \frac{T_1}{T_2}$$

11. *Diametral pitch.* It is the ratio of number of teeth to the pitch circle diameter in millimetres.

It is denoted by *pd* .

Mathematically,

Diametral pitch, $p_d = \frac{T}{D} = \frac{\pi}{p_c}$ $\quad \ldots \left(\because\ p_c = \frac{\pi D}{T}\right)$

where T = Number of teeth, and

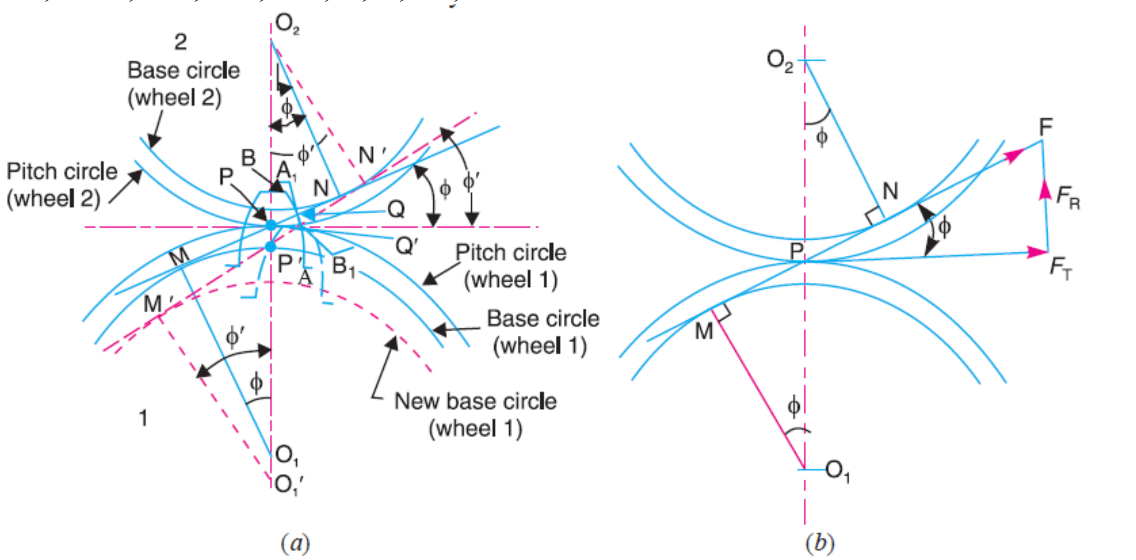
D = Pitch circle diameter.

12. *Module.* It is the ratio of the pitch circle diameter in millimeters to the number of teeth. It is usually denoted by *m*.

Mathematically,

Module, $m = D / T$

Note : The recommended series of modules in Indian Standard are 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, and 20. The modules 1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14 and 18 are of second choice.



Involute teeth.

If *F* is the maximum tooth pressure as shown in Fig. 12.10 (*b*), then

Tangential force, $F_T = F \cos \phi$

and radial or normal force, $F_R = F \sin \phi$.

$\therefore$ Torque exerted on the gear shaft

$= F_T \times r$, where *r* is the pitch circle radius of the gear.

Note : The tangential force provides the driving torque and the radial or normal force produces radial deflection of the rim and bending of the shafts.

Example 1. *A single reduction gear of 120 kW with a pinion 250 mm pitch circle diameter and speed 650 r.p.m. is supported in bearings on either side. Calculate the total load due to the power transmitted, the pressure angle being 20°.*

Solution. Given : $P = 120$ kW $= 120 \times 10^3$ W ; $d = 250$ mm or $r = 125$ mm $= 0.125$ m ; $N = 650$ r.p.m. or $\omega = 2\pi \times 650/60 = 68$ rad/s ; $\phi = 20°$

Let T = Torque transmitted in N-m.

We know that power transmitted (P),

$$120 \times 10^3 = T.\omega = T \times 68 \quad \text{or} \quad T = 120 \times 10^3/68 = 1765 \text{ N-m}$$

and tangential load on the pinion,

$$F_T = T/r = 1765 / 0.125 = 14\ 120 \text{ N}$$

∴ Total load due to power transmitted,

$$F = F_T / \cos\phi = 14\ 120 / \cos 20° = 15\ 026 \text{ N} = 15.026 \text{ kN}$$ **Ans.**

Gear trains

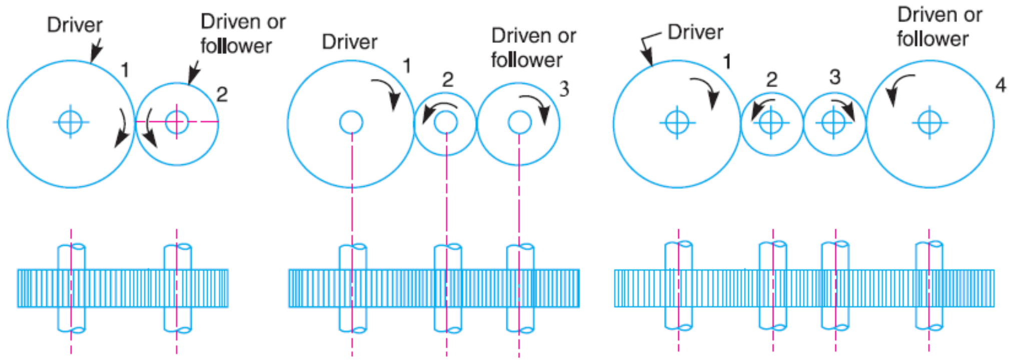


Fig. 13.1. Simple gear train.

Let N_1 = Speed of gear 1(or driver) in r.p.m.,

N_2 = Speed of gear 2 (or driven or follower) in r.p.m.,

T_1 = Number of teeth on gear 1, and

T_2 = Number of teeth on gear 2.

Since the speed ratio (or velocity ratio) of gear train is the ratio of the speed of the driver to the speed of the driven or follower and ratio of speeds of any pair of gears in mesh is the inverse of their number of teeth, therefore

$$\text{Speed ratio} = \frac{N_1}{N_2} = \frac{T_2}{T_1}$$

It may be noted that ratio of the speed of the driven or follower to the speed of the driver is known as ***train value*** of the gear train. Mathematically,

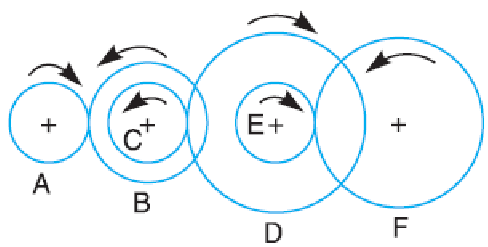
$$\text{Train value} = \frac{N_2}{N_1} = \frac{T_1}{T_2}$$

$$\text{Speed ratio} = \frac{\text{Speed of driver}}{\text{Speed of driven}} = \frac{\text{No. of teeth on driven}}{\text{No. of teeth on driver}}$$

$$\text{Train value} = \frac{\text{Speed of driven}}{\text{Speed of driver}} = \frac{\text{No. of teeth on driver}}{\text{No. of teeth on driven}}$$

Example 2. *The gearing of a machine tool is shown in Fig.* 13.3. *The motor shaft is connected to gear A and rotates at* 975 *r.p.m. The gear wheels B, C, D and E are fixed to parallel shafts rotating together. The final gear F is fixed on the output shaft. What is the speed of gear F ? The number of teeth on each gear are as given below :*

Gear	*A*	*B*	*C*	*D*	*E*	*F*
No. of teeth	*20*	*50*	*25*	*75*	*26*	*65*



Let N_F = Speed of gear *F*, *i.e.* last driven or follower.

We know that

$$\frac{\text{Speed of the first driver}}{\text{Speed of the last driven}} = \frac{\text{Product of no. of teeth on drivens}}{\text{Product of no. of teeth on drivers}}$$

$$\frac{N_A}{N_F} = \frac{T_B \times T_D \times T_F}{T_A \times T_C \times T_E} = \frac{50 \times 75 \times 65}{20 \times 25 \times 26} = 18.75$$

$$\therefore \quad N_F = \frac{N_A}{18.75} = \frac{975}{18.75} = 52 \text{ r. p. m.} \quad \textbf{Ans.}$$

Example 3. *Two parallel shafts, about 600 mm apart are to be connected by spur gears. One shaft is to run at 360 r.p.m. and the other at 120 r.p.m. Design the gears, if the circular pitch is to be 25 mm.*

Solution. Given : $x = 600$ mm ; $N_1 = 360$ r.p.m. ; $N_2 = 120$ r.p.m. ; $p_c = 25$ mm

Let d_1 = Pitch circle diameter of the first gear, and

d_2 = Pitch circle diameter of the second gear.

We know that speed ratio,

$$\frac{N_1}{N_2} = \frac{d_2}{d_1} = \frac{360}{120} = 3 \quad \text{or} \quad d_2 = 3d_1 \qquad ...(i)$$

and centre distance between the shafts (x),

$$600 = \frac{1}{2}(d_1 + d_2) \quad \text{or} \quad d_1 + d_2 = 1200 \qquad ...(ii)$$

From equations (*i*) and (*ii*), we find that

$$d_1 = 300 \text{ mm, and } d_2 = 900 \text{ mm}$$

∴ Number of teeth on the first gear,

$$T_1 = \frac{\pi d_2}{p_c} = \frac{\pi \times 300}{25} = 37.7$$

and number of teeth on the second gear,

$$T_2 = \frac{\pi d_2}{p_c} = \frac{\pi \times 900}{25} = 113.1$$

Since the number of teeth on both the gears are to be in complete numbers, therefore let us make the number of teeth on the first gear as 38. Therefore for a speed ratio of 3, the number of teeth on the second gear should be $38 \times 3 = 114$.