وزارة التعليم العالي والبحث العلمي
الجامعة التقنية الجنوبية
المعهد التقني العمارة
قسم التقنيات الميكانيكية

الحقيبة التدريسية لمادة

تقنية أجزاء المكائن

الصف الثاني

تدريسي المادة
م. مروه علي حرب

٢٠٢٥

The aims *:-*

machine parts aims to explain the role of mechanical parts through machine System , the relation links them , how to conduct some calculations to design these parts and to specify all factors that are affected

الهدف من دراسة المادة:

الهدف الرئيسي لدراسة حقيبة أجزاء المكائن هي تزويد الطلاب بالمعرفةوالمهارات اللازمة لفهم وتصميم وتحليل وتصنيع المكونات الميكانيكية المختلفة يهدف هذا التخصص إلى إعداد كوادر قادرة على العمل في مجموعة واسعة من الصناعات، بما في ذلك تصميم الآلات، وتصنيعها، وصيانتها.

الفئة المستهدفة:

طلبة الصف الثاني / قسم التقنيات الميكانيكية

التقنيات التربوية المستخدمة:

1. سبورة واقلام
2. السبورة التفاعلية
3. عارض البيانات Data Show
4. جهاز حاسوب محمول Laptop

Theoretical Subjects	
Week No.	Subject Topics
1	Review of Strength of Materials
2-3	Riveted Joints .Types of Riveted Joints ,Design of Riveted Joints, Efficiency of Riveted Joints .
4-5	Welded Joints Types of welding Joints ,Design of welding Joints

6-7	*Screwed Joints, Design of Bolts for Fastening , Design of Bolts for Power Transition .*
8-9	*Keyed Joints , Types of Key , Design of Sunk Key .*
10-11	*Frictional Clutches, Type of Frictional Clutches , Design of Frictional Clutches.*
12-13	*Types of Springs , Design of Springs*
Week No.	*Subject Topics*
14-15	*Types of Belts , Design of Belts.*
16-17	*Design of Shafts*
18-19	*Design of Journal Bearings*
20	*Selection of Ball Bearings*
21-22	*Design of Gears by Lewis Equation*
23-24	*Gears Trains*
25-26	*Design of Simple Gears Box*
27-28	*Worm Gears*
29-30	*Cams*

References:-

1- Strength of Material by Ferdinal L .Singer

2- Strength of Materials by R.S.Khurmi. 3-

Machine Design by R.S. Khurmi, J.K. Gupta

4- Machine Design by Paul H.Black .

5- Schaums Outline Series of Machine Design by Hall ,Holowenko , Laughin

Machine design element :-

Is an interdisciplinary subject embracing broadly strength of materials , engineering Mechanics, manufacturing processes, and material science.

SI Units

Table 1.2. Derived units.

S.No.	Quantity	Symbol	Units
1.	Linear velocity	V	m/s
2.	Linear acceleration	a	m/s^2
3.	Angular velocity	ω	rad/s
4.	Angular acceleration	α	rad/s^2
5.	Mass density	ρ	kg/m^3
6.	Force, Weight	F, W	N ; 1N = 1kg-m/s^2
7.	Pressure	P	N/m^2
8.	Work, Energy, Enthalpy	W, E, H	J ; 1J = 1N-m
9.	Power	P	W ; 1W = 1J/s
10.	Absolute or dynamic viscosity	μ	N-s/m^2
11.	Kinematic viscosity	v	m^2/s
12.	Frequency	f	Hz ; 1Hz = 1cycle/s
13.	Gas constant	R	J/kg K
14.	Thermal conductance	h	W/m^2 K
15.	Thermal conductivity	k	W/m K
16.	Specific heat	c	J/kg K
17.	Molar mass or Molecular mass	M	kg/mol

Table 1.3. Prefixes used in basic units.

Factor by which the unit is multiplied	*Standard form*	*Prefix*	*Abbreviation*
1 000 000 000 000	10^{12}	tera	T
1 000 000 000	10^{9}	giga	G
1 000 000	10^{6}	mega	M
1000	10^{3}	kilo	K
100	10^{2}	hecto*	h
10	10^{1}	deca*	da
0.1	10^{-1}	deci*	d
0.01	10^{-2}	centi*	c
0.001	10^{-3}	milli	m
0.000 001	10^{-6}	micro	μ
0.000 000 001	10^{-9}	nano	n
0.000 000 000 001	10^{-12}	pico	p

Strength of materials

Deals with the relations between externally applied load and their internal effects on body.

Load

It is defined as any external force acting upon amachine part .

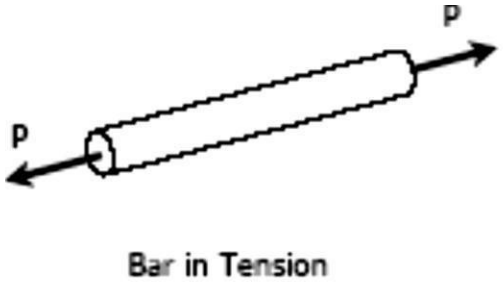
Stress

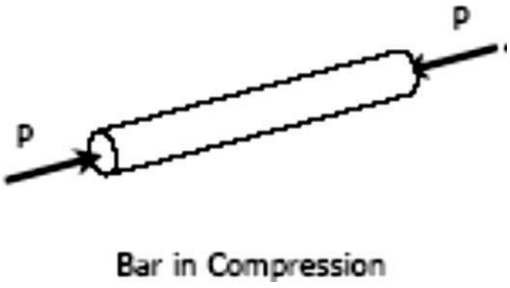
Simple stress

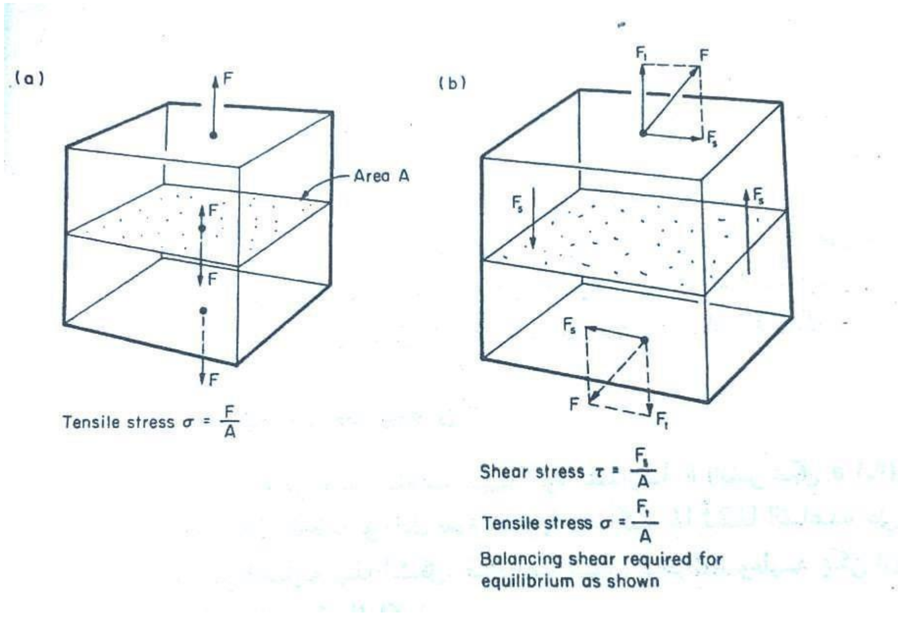
Simple stress can be classified as ***normal stress, shear stress, and bearing stress***

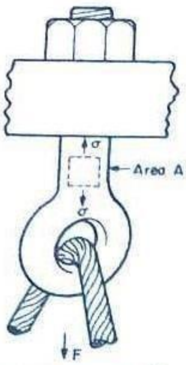
1- Normal stress

develops when a force is applied perpendicular to the cross-sectional area of the material (tensile stress and compressive stress)

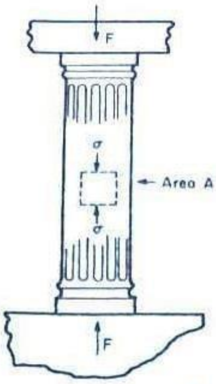




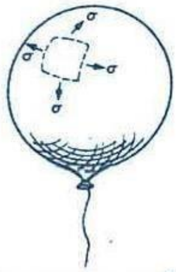

(a)
F
Area A
F
F
F
Tensile stress σ = F/A
(b)
F
F
F
F
F
F
F
Shear stress τ = Fs/A
Tensile stress σ = Ft/A
Balancing shear required for
equilibrium as shown



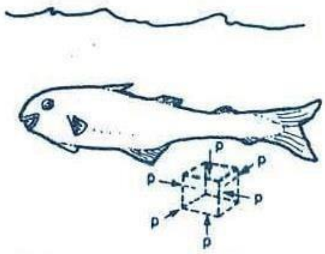
Simple tension, $\sigma = \frac{F}{A}$



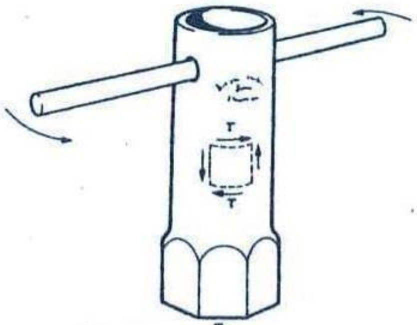
Simple compression, $\sigma = \frac{F}{A}$



Biaxial tension, $\sigma = \frac{F}{A}$



Hydrostatic pressure, $p = -\frac{F}{A}$



Pure shear, $\tau = \frac{F_s}{A}$

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

Where P = Force or load acting on a body, and A

= Cross-sectional area of the body.

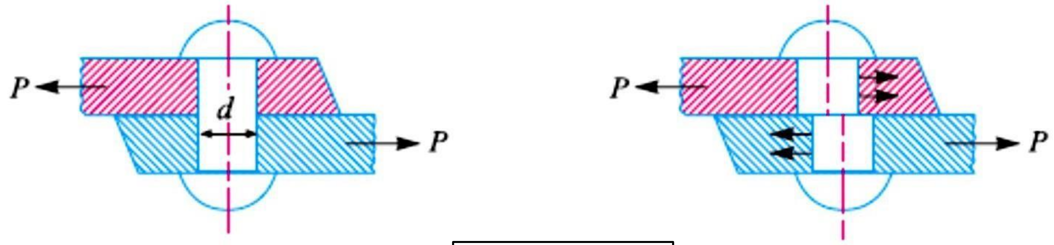
In S.I. units, the stress is usually expressed in Pascal (Pa) such that

1 MPa = 1 × 106 N/m^2 = 1 N/mm^2 and

1 GPa = 1 × 109 N/m^2 = 1 kN/mm^2

2- Shearing Stress

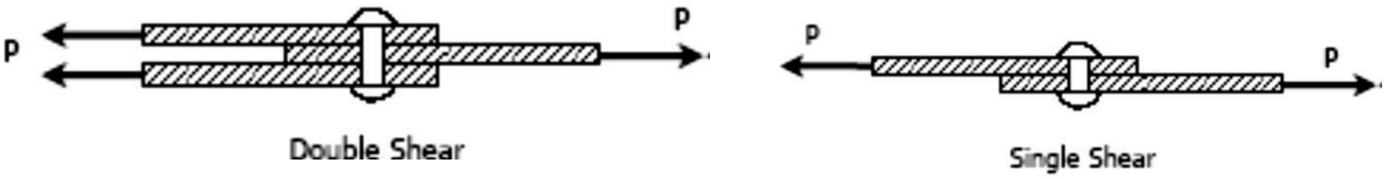
Forces parallel to the area resisting the force cause shearing stress. Shearing stress is also known as tangential stress



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

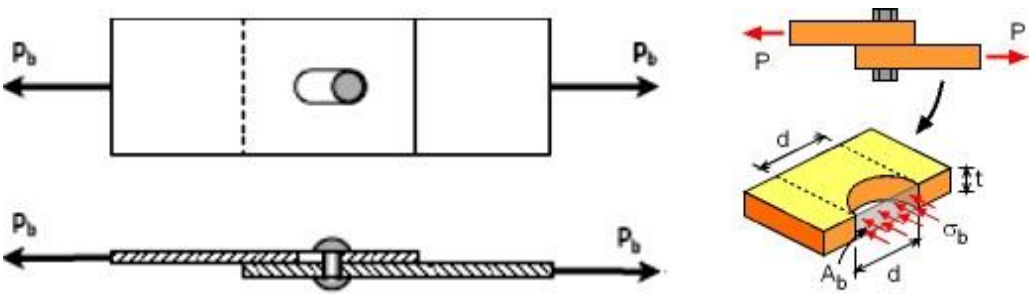
Where P = Force or load acting on a body, and

A = Cross-sectional area of the body , η = Shear stress



3- Bearing Stress

Bearing stress is the contact pressure between the separate bodies.



$$\sigma_b \text{ (or } \sigma_c) = \frac{P}{d.t.n}$$

d = Diameter of the rivet,

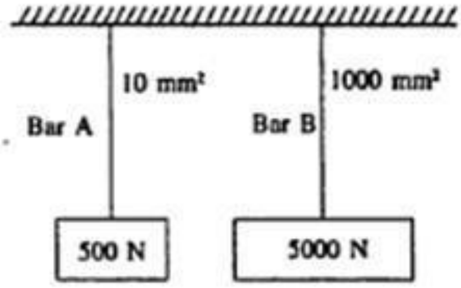
t = Thickness of the plate,

$d.t$ = Projected area of the rivet, and

n = Number of rivets per pitch length in bearing or crushing.

Example:

1. Two bars of equal lengths but different materials are suspended from a common support. Bar A supports 500 N and bar B supports 5000 N. If the cross-sectional area of bar A is 10 mm^2 and bar B is 1000 mm^2, compare the strength of bars A and B.



Solution:

$$S_A = \frac{P_A}{A_A} = \frac{500\ N}{10\ mm^2 \times 10^{-6}\ m^2/mm^2}$$

$$S_B = \frac{P_B}{A_B} = \frac{5000\ N}{1000\ mm^2 \times 10^{-6}\ m^2/mm^2}$$

$$= 50 \times 10^6\ N/m^2$$

Comparing, $S_A = 10\ S_B$, Bar A is 10 times stronger than Bar B.

The unit strength of a material is usually defined as the stress in the material. Hence,

$S = \frac{P}{A}$ where S, P and A have the same meaning as before.

$1\ N/m^2 = 1\ Pa$

$MPa = 10^6\ Pa$

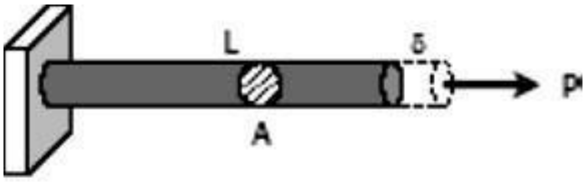
The stress in Bar A maybe expressed as

$50 \times 10^6\ N/m^2 = 50\ MN/m^2 = 50\ MPa$

Strain

Simple Strain

A strain is the ratio of the change in length caused by the applied force, to the original length.

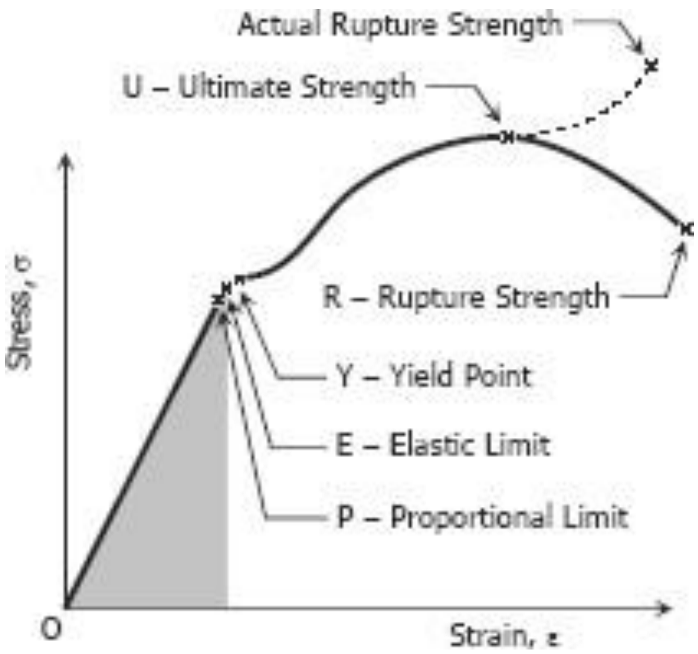


$$\varepsilon = \frac{\delta}{L}$$

where δ is the deformation and L is the original length, thus (ε) is dimensionless.

Stress-Strain Diagram

Metallic engineering materials are classified as either ductile or brittle materials. A ductile material is one having relatively large tensile strains up to the point of rupture like structural steel and aluminum, whereas brittle materials has a relatively small strain up to the point of rupture like cast iron and concrete. An arbitrary strain of 0.05 mm/mm is frequency taken as the dividing line between these two classes.



Young's Modulus or Modulus of Elasticity

Hooke's law states that when a material is loaded within elastic limit, the stress is directly proportional to strain, i.e.*

$$\sigma \propto \varepsilon \qquad or \qquad \sigma = E.\varepsilon$$

$$\therefore E = \frac{\sigma}{\varepsilon} \Longrightarrow \boxed{E = \frac{P.L}{A.\delta}}$$

E = a constant of proportionality known as Young's modulus or modulus of elasticity

P = Axial compressive force acting on the body,

A = Cross-sectional area of the body,

L = Original length, and δ = Decrease in length

Table 4.1. Values of E for the commonly used engineering materials.

Material	Modulus of elasticity (E) in GPa i.e. GN/m^2 or kN/mm^2
Steel and Nickel	200 to 220
Wrought iron	190 to 200
Cast iron	100 to 160
Copper	90 to 110
Brass	80 to 90
Aluminium	60 to 80
Timber	10

Shear Modulus or Modulus of Rigidity

The shear stress is directly proportional to shear strain

$$\tau \propto \varphi \qquad or \qquad G = \frac{\tau}{\varphi}$$

φ = shear strain

τ = shear stress

G = modulus of rigidity

Table 4.2. Values of C for the commonly used materials.

Material	Modulus of rigidity (C) in GPa i.e. GN/m^2 or kN/mm^2
Steel	80 to 100
Wrought iron	80 to 90
Cast iron	40 to 50
Copper	30 to 50
Brass	30 to 50
Timber	10

Factor of Safety

It is defined, in general, as the ratio of the maximum stress to the working stress.

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

In case of ductile materials e.g. mild steel, where the yield point is clearly defined,

$$Factor\ of\ safety = \frac{Field\ point\ stress\ stress}{Working\ or\ design\ stress}$$

In case of brittle materials e.g. cast iron, the yield point is not well defined as for ductile materials.

Therefore, the factor of safety for brittle materials is based on ultimate stress.

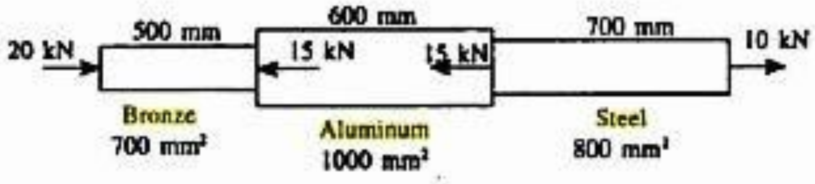
$$\therefore Factor\ of\ safety = \frac{Ultimate\ \ stress}{Working\ or\ design\ stress}$$

Table 4.3. Values of factor of safety.

Material	Steady load	Live load	Shock load
Cast iron	5 to 6	8 to 12	16 to 20
Wrought iron	4	7	10 to 15
Steel	4	8	12 to 16
Soft materials and alloys	6	9	15
Leather	9	12	15
Timber	7	10 to 15	20

Ex(1)

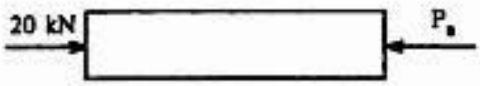
An Aluminum tube is rigidly fastened between a bronze and steel rods as shown. Axial loads are applied at the positions indicated. Determine the stress in each material.



Solution:

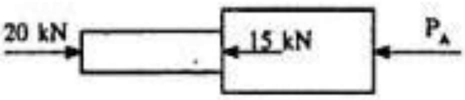
Take the bronze as a Free Body

$$\Sigma F_H = 0$$
$$20 - P_B = 0$$
$$P_B = 20 \text{ kN}$$



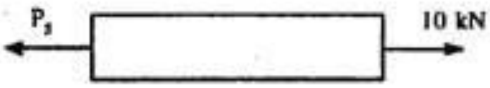
Pass section 1-1 and take the left portion as a Free Body.

$$\Sigma F_H = 0$$
$$20 - 15 - P_A = 0$$
$$P_A = 5 \text{ kN}$$



Take the steel rod as a Free Body

$$\Sigma F_H = 0$$
$$-P_S + 10 = 0$$
$$P_S = 10 \text{ kN}$$



Compute for the stresses for each materials

(a) Bronze:

$$S_B = \frac{P_B}{A_B} = \frac{20 \text{ kN} \times 10^3 \text{ N/kN}}{700 \text{ mm}^2 \times 10^{-6} \text{ m}^2/\text{mm}^2} = 286 \times 10^6 \text{ N/m}^2$$
$$= 286 \text{ MPa}$$

(b) Aluminum:

$$S_A = \frac{P_A}{A_A} = \frac{5 \text{ kN} \times 10^3 \text{ N/kN}}{1000 \text{ mm}^2 \times 10^{-6} \text{ m}^2/\text{mm}^2}$$
$$= 5 \times 10^6 \text{ N/m}^2$$
$$= 5 \text{ MPa}$$

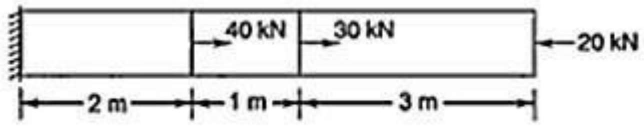
(c) Steel:

$$S_S = \frac{P_S}{A_S} = \frac{10 \text{ kN} \times 10^3 \text{ N/kN}}{800 \text{ mm}^2 \times 10^{-6} \text{ m}^2/\text{mm}^2}$$
$$= 12.5 \times 10^6 \text{ N/m}^2$$
$$= 12.5 \text{ MPa}$$

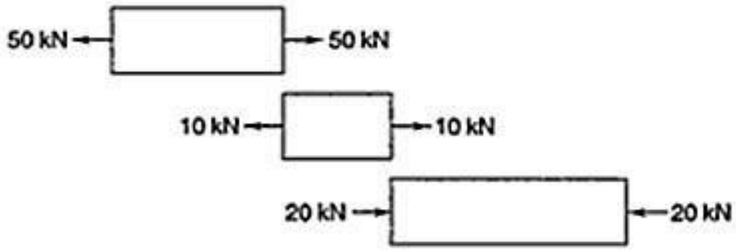
Note: The stresses in the bronze and aluminum are compressive stresses while the stress in the steel is tensile stress.

Ex(2)

Determine the stress in all the three sections and total deformation of the steel rod shown in the figure. Cross-sectional area = 10 cm^2, E = 200 GN/m^2.



Draw the free body diagram of three sections:



Using the principal of superposition,

$$\delta l = \frac{1}{AE}(P_1 l_1 + P_2 l_2 - P_3 l_3)$$

$$= \frac{1}{10\times10^{-4}\times200\times10^{9}}\ (50 \times 10^3 \times 2 + 10 \times 10^3 \times 1 - 20 \times 10^3 \times 3)$$

$$= \frac{10^3}{2\times10^8}\ (100 + 10 - 60)$$

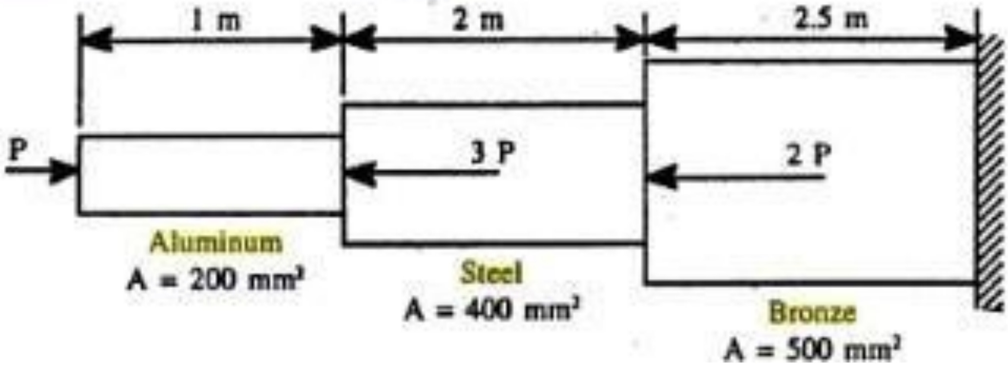
$$= 25 \times 10^{-5}\ \text{m}$$

$$= 0.25\ \text{mm}$$

Home work

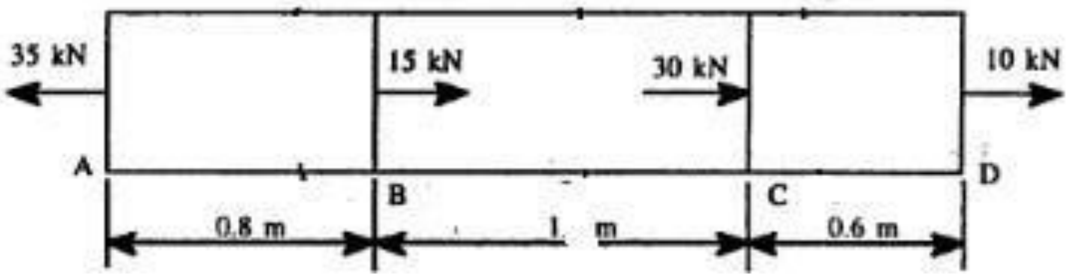
1.

A steel tube is rigidly attached between an aluminum rod and a bronze rod as shown. Axial loads are applied at the positions indicated. Find the maximum value of P that will not exceed a stress in aluminum of 80 MPa, in steel of 150 MPa, or in bronze of 100 MPa.



2.

An aluminum bar having a cross-sectional area of 160 mm² carries an axial loads as shown. If E = 70 G Pa, compute the total deformation of the bar. Assume that the bar is suitably braced to prevent buckling.



= .00315 m = 3.75 mm

The fastenings

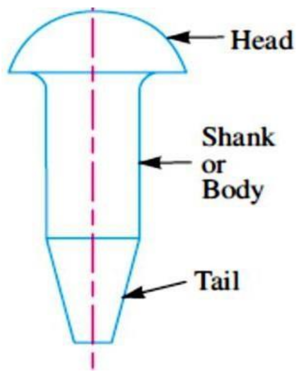
May be classified into the following two groups :

1. *Permanent fastenings, are those fastenings which can not be disassembled without destroying the connecting components. The examples of permanent fastenings in order of strength are soldered, brazed, welded and riveted joints*
2. *Temporary or detachable fastenings.*

are those fastenings which can be disassembled without destroying the connecting components. The examples of temporary fastenings are crewed, keys, cotters, pins and splined joints.

Rivet Joint

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

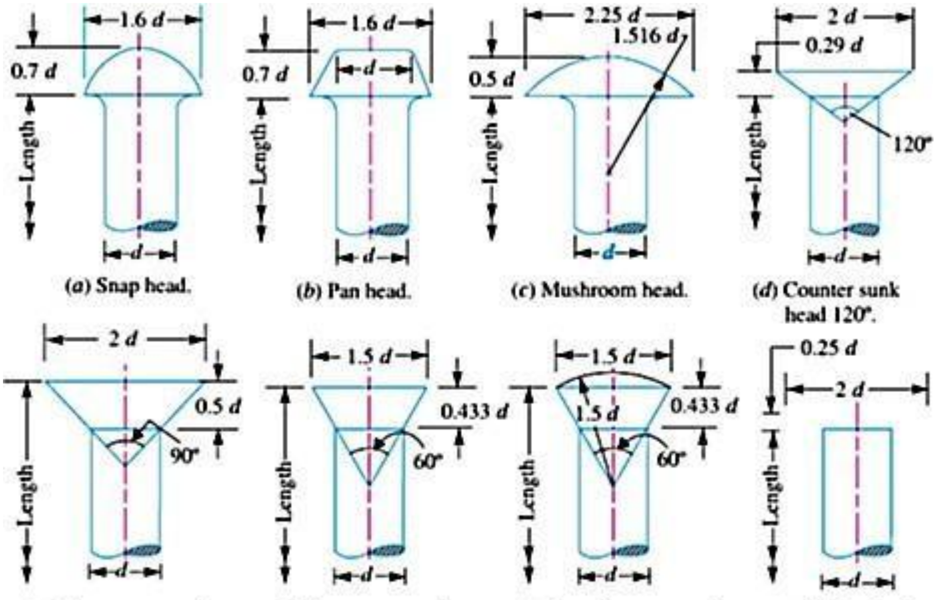


Material of Rivets

The material of the rivets must be tough and ductile. They are usually made of steel (low carbon steel or nickel steel), brass, aluminum or copper, but when strength and a fluid tight joint is the main consideration, then the steel rivets are used.

Types of Rivet Heads

According to Indian standard specifications, the rivet heads are classified into the following :

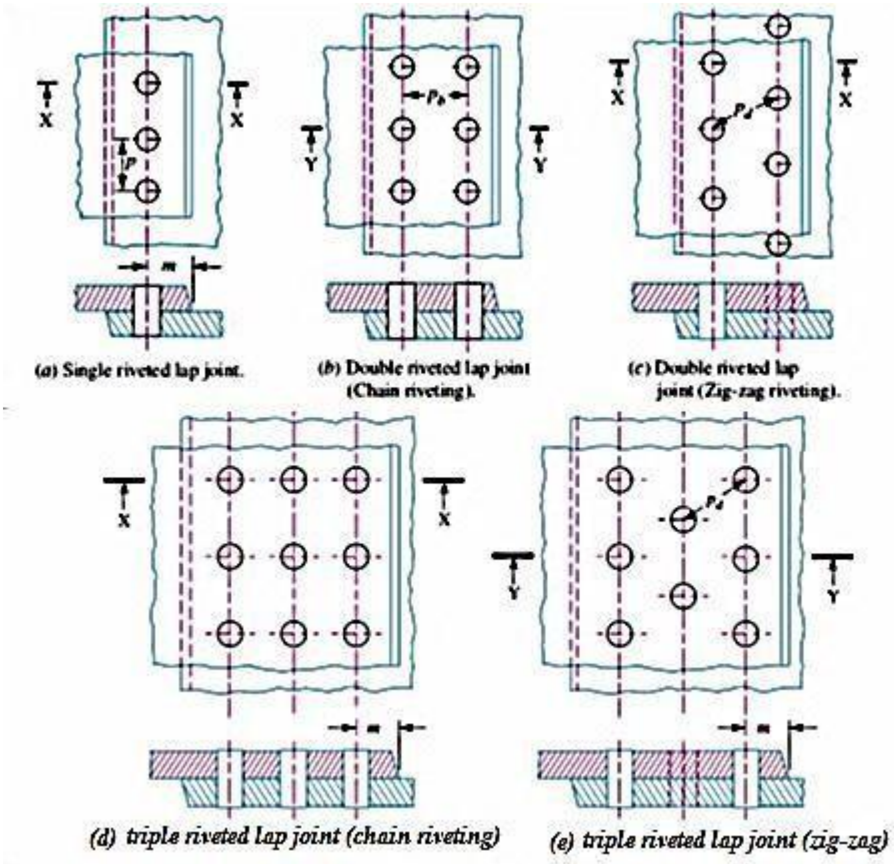


(a) Snap head. (b) Pan head. (c) Mushroom head. (d) Counter sunk head 120°.
(e) Flat counter sunk head 90°. (f) Flat counter sunk head 60°. (g) Round counter sunk head 60°. (h) Flat head.

Types of Riveted Joints

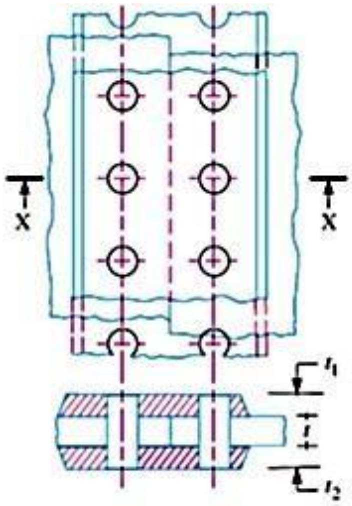
Following are the two types of riveted joints, depending upon the way in which the plates are connected.

1- Lap Joint *A lap joint is that in which one plate overlaps the other and the two plates are then riveted together.*

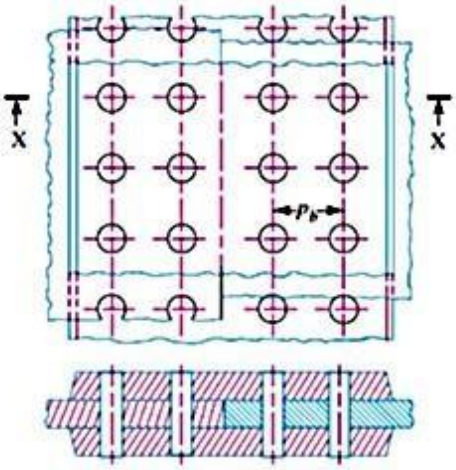


Butt Joint

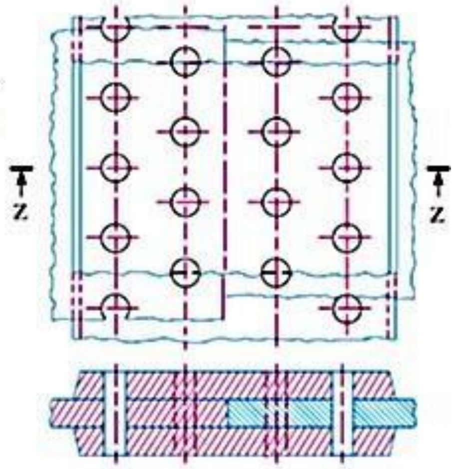
A butt joint is that in which the main plates are kept in alignment butting (i.e. touching) each other and a cover plate (i.e. strap) is placed either on



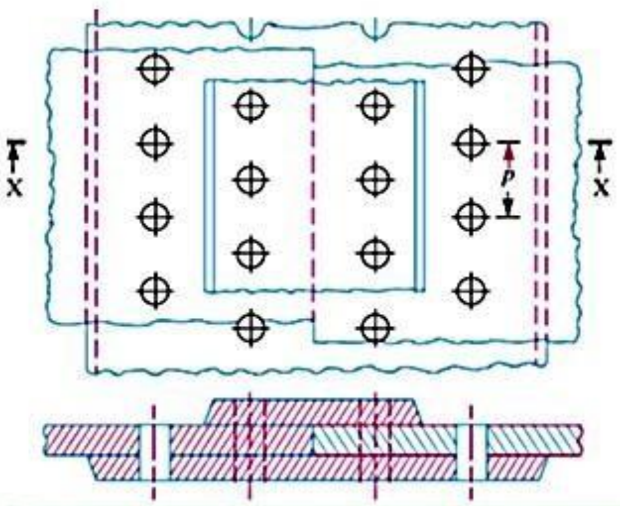
(a) single riveted double strips butt joints



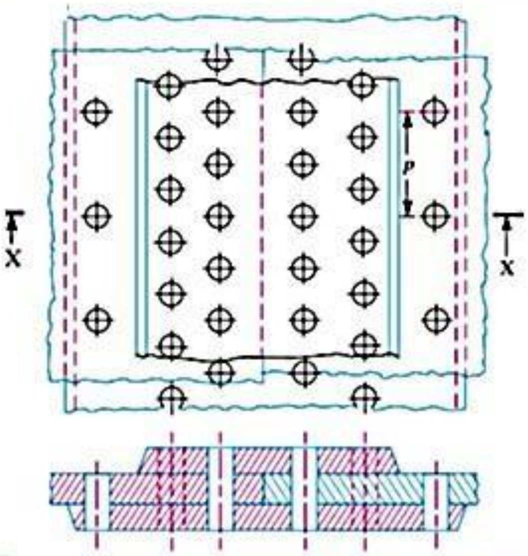
(b) double riveted double strips butt joints (chain riveting)



(c) double riveted double strips butt joints (zig -zag riveting)



. Double riveted double strap (unequal) butt joint with zig-zag riveting



Triple riveted double strap (unequal) butt joint.

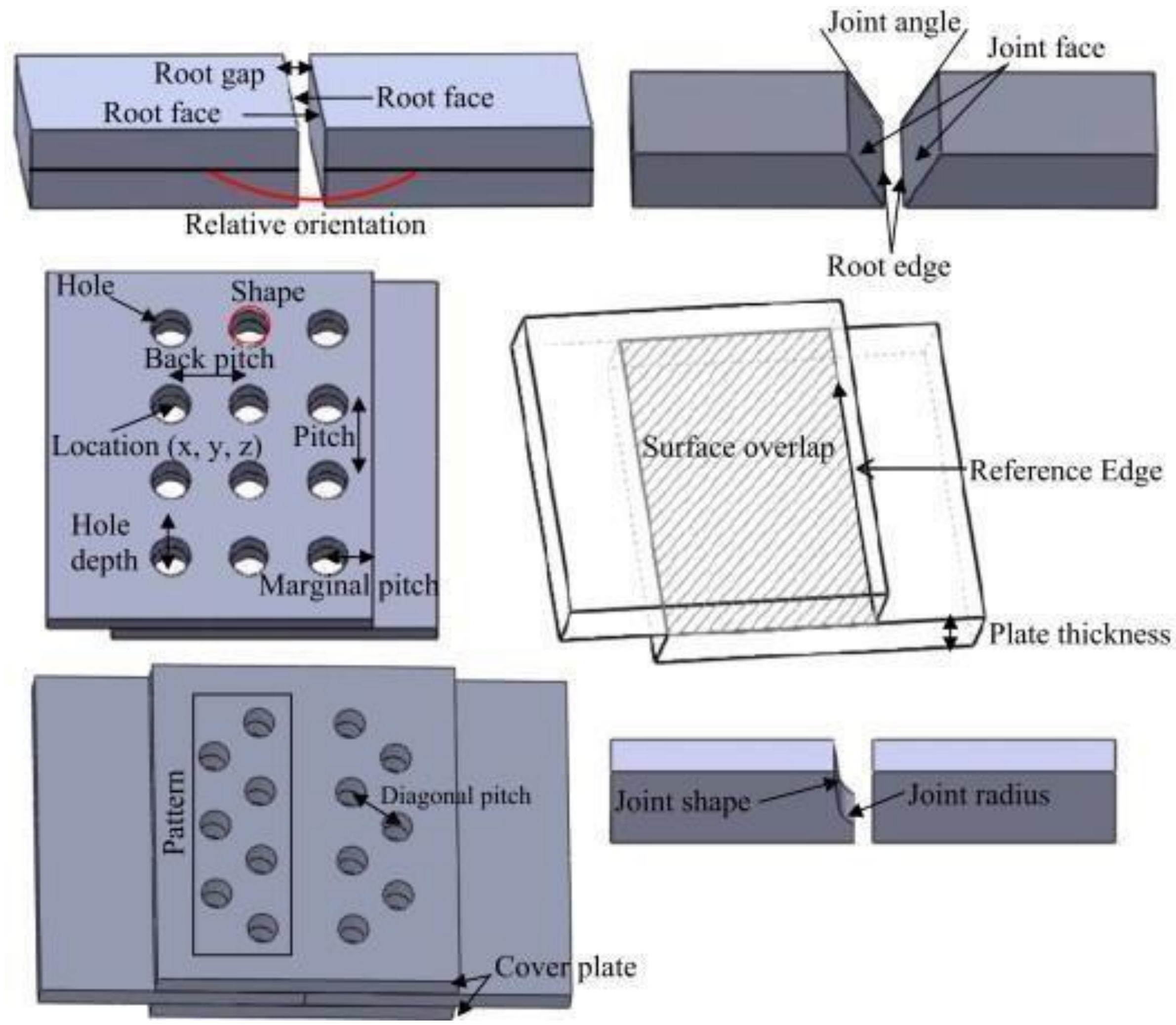
one side or on both sides of the main plates.

Important Terms Used in Riveted Joints

The following terms in connection with the riveted joints are important from the subject point of view :

1. *Pitch. It is the distance from the centre of one rivet to the centre of the next rivet measured parallel to the seam as shown in Fig. It is usually denoted by p.*
2. *Back pitch. It is the perpendicular distance between the centre lines of the successive rows as shown in Figure bellow. It is usually denoted by p_b.*
3. *Diagonal pitch. It is the distance between the centers of the rivets in adjacent rows of zig-zag riveted joint as shown in Fig. It is usually denoted by p_d.*

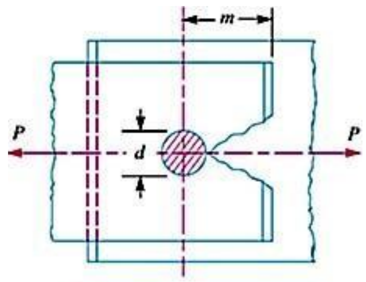
4.Margin or marginal pitch. It is the distance between the centre of rivet hole to the nearest edge of the plate as shown in Figure bellow. It is usually denoted by m.



Failures of a Riveted Joint

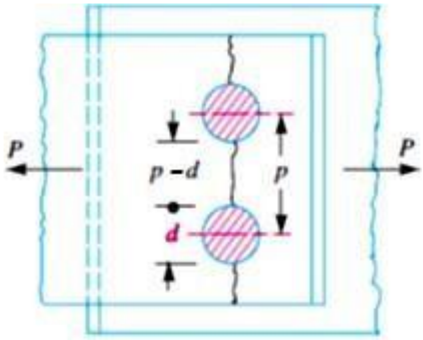
A riveted joint may fail in the following ways :

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



Tearing of the plate at an edge

__2. Tearing of the plate across a row of rivets__. Due to the tensile stresses in the main plates, the main plate or cover plates may tear off across a row of rivets as shown in Fig.



Tearing of the plate across the rows of rivets.

$$A_t = (p - d)\,t$$

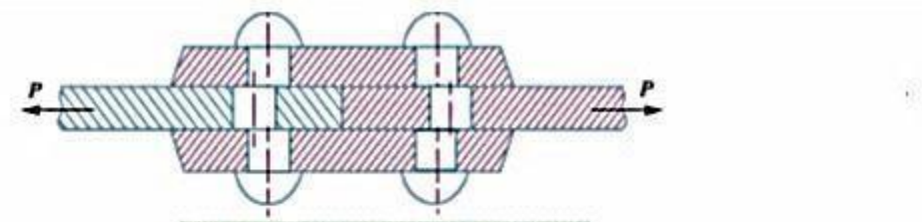
Let p = Pitch of the rivets, d = Diameter of the rivet hole, t = Thickness of the plate, and

ζ_t = Permissible tensile stress for the plate material.

∴ Tearing resistance (P_t)or pull required to tear off the plate per pitch length,

$$P_t = A_t\,.\zeta_t = (p - d)\,t\,.\zeta_t$$

__3. Shearing of the rivets__. The plates which are connected by the rivets exert tensile stress on the rivets, and if the rivets are unable to resist the stress, they are sheared off as shown in Fig.



Shearing off a rivet in double cover butt joint

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

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

d = Diameter of the rivet hole, η = Safe permissible shear stress for the rivet material, and n = Number of rivets per pitch length.

We know that shearing area,

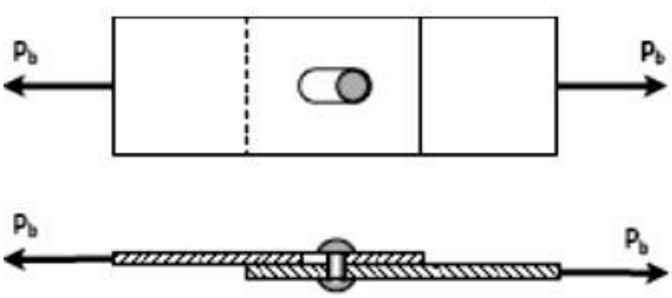
$$As = \frac{\pi}{4} \times d^2 \quad \text{...(In single shear)}$$

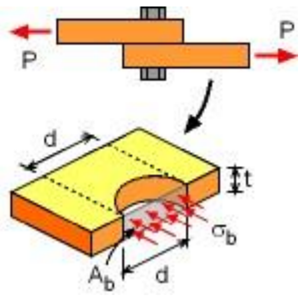
$$= 2 \times \frac{\pi}{4} \times d^2 \quad \text{...(Theoretically, in double shear)}$$

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

P_s =shearing resistance

***4. Crushing of the plate or rivets**. Sometimes, the rivets do not actually shear off under the tensile stress, but are crushed as shown in Fig.*





Let d = Diameter of the rivet hole,

t = Thickness of the plate,

ζ_c *= permissible crushing stress for the rive, and n =*

Number of rivets per pitch length under crushing.

We know that crushing area per rivet (i.e. projected area per rivet),

$$A_c = d.t$$

$\therefore$ *Total crushing area = n.d.t*

$$P_c = d\ .t\ .\zeta_c\ .n$$

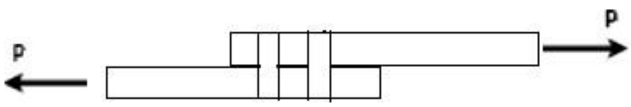
P_c *= crushing resistance*

Efficiency of a Riveted Joint

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

Take the Least of P_t *,* P_s *and* P_c *and Strength of the un-riveted or solid plate per pitch length,*

$$P_e = p \times t \times \zeta_t$$



Efficiency of the riveted joint, $\eta = \dfrac{\text{least of } Pt, Ps \text{ and } P}{Pe}\ \%$

Where p = Pitch of the rivets,

t = Thickness of the plate, and

ζ_t *= Permissible tensile stress of the plate material*

P_e *= equivalent resistance*

<u>Notes</u>

❖ *If the thickness of plate (t) ≥ (8 mm) then:*

$d = 6\sqrt{t}$ .... *Unwins formula*

- *If the thickness of plate (t) ≤ (8 mm) then the diameter (d) result from :*

$$P_s = P_c$$

- *To find pitch (p) from the relation :-*

$$P_s = P_t$$

Example (1). *Find the efficiency of the Single riveted lap joint of (6 mm) thick plates with(20 mm) diameter rivets having a pitch of(50 mm). Permissible tensile stress in plate (120MPa) ,Permissible shearing stress in rivets (90 MPa) and Permissible crushing stress in rivets (180 MPa).*

Solution .Given *:*

$t = 6$ mm ; $d = 20$ mm ; $\zeta_t = 120$ MPa $= 120$ N/mm^2 ;
$\eta = 90$ MPa $= 90$ N/mm^2 ; $\zeta_c = 180$ MPa $= 180$ N/mm^2
Pitch, $p = 50$ mm

(i) Tearing resistance of the plate

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

(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

$$P_s = \frac{\pi}{4} \cdot d^2 \cdot \eta \,.n$$

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

(iii) Crushing resistance of the rivet
Since the joint is a single riveted, therefore strength of one rivet is taken.
$P_c = d\,.t\,.\zeta_c = 20 \times 6 \times 180 = 21\,600$ N
∴ *Strength of the joint= Least of* P_t, P_s *and* $P_c = 21\,600$ N
We know that strength of the unriveted or solid plate, P
$= p \times t \times \zeta_t = 50 \times 6 \times 120 = 36\,000$ N

∴*Efficiency of the joint,*

$l \quad f\,P \quad P \quad d\,P$

$$\eta = \frac{east\ o \quad t, \quad s\ an \quad c}{Pe}\ \%$$

$$\eta = \frac{21600}{36000}\ \% = 0.60\ or\ 60\ \%$$

Example (2). *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 (ζ_t = 120 MPa); (η = 100 MPa;) and (ζ_c = 150 MPa) .Find the efficiency of joint.*

Solution .Given *: t = 20 mm ; d = 25 mm ; p = 100 mm ; ζ_t = 120 MPa = 120 N/mm^2 ; η = 100 MPa = 100 N/mm^2 ; ζ_c = 150 MPa = 150 N/mm^2*

(i) Tearing resistance of the plate

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

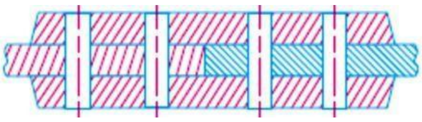
$P_t = (p - d)\ t \times \zeta_t = (100 - 25)\ 20 \times 120 = 180\ 000\ N$

(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.

$$P_s = \frac{\pi}{4} \cdot d^2 \cdot \eta \cdot n$$

$$= 2 \times \frac{\pi}{4} \times (25)^2 \times 100 \times 2 = 196\ 250\ N$$



(iii) Crushing resistance of the rivets

Since the joint is double riveted, therefore the strength of two rivets is taken.

$P_c = d \times t \times \zeta_c \times n = 25 \times 20 \times 150 \times 2 = 150\ 000\ N$

∴ *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,

$P = p \times t \times \zeta_t = 100 \times 20 \times 120 = 240\ 000\ N$ ∴

Efficiency of the joint=

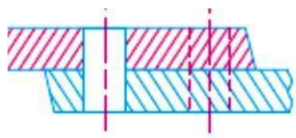
$$\eta = \frac{least\ of\ Pt, Ps\ and\ P\ c}{P}\ \% \qquad e$$

$$\eta = \frac{150000}{240000}\ \% = 0.62\ or\ 62.5\%$$

Example (3). *A double riveted lap joint with zig-zag riveting is to be designed for (13 mm) thick plates. Assume (ζ_t = 80 MPa) ;(η = 60 MPa); and (ζ_c = 120 MPa) State how the joint will fail and find the efficiency of the joint.*

Solution .Given *: t = 13 mm ; ζ_t = 80 MPa = 80 N/mm^2 ;*

η = 60 MPa = 60 N/mm^2 ;ζ_c = 120 MPa = 120 N/mm^2



Double riveted lap joint (Zig-zag riveting).

Since the thickness of plate is greater than 8 mm, therefore diameter of rivet hole,

$$d = 6\sqrt{t} \quad d = 6\sqrt{13} = 21.6mm$$

Let p = Pitch of the rivets. To find it From equations $P_t = P_s$

$$(p - d)t \times \zeta_t = \frac{\pi}{4} . d^2 . \eta .n$$

Since the joint is a double riveted lap joint with zig-zag riveting therefore there are two rivets per pitch length, i.e. n = 2. Also, in a lap joint, the rivets are in single shear .We know that tearing resistance of the plate,

$$(p - 21.6)\ 13 \times 80 = \frac{\pi}{4} \times (21.6)^2 \times 60 \times 2$$

$$(p - 21.6)\ 1040 = 43\,949.9\ N \quad \text{or } 1040\,p = 43\,949.9 + 66413.9$$

$$p = \frac{66413.9}{1040} = 63.85\ mm$$

Now let us find the tearing resistance of the plate, shearing resistance and crushing resistance of the rivets .We know that tearing resistance of the plate,

$$P_t = (p - d)\ t \times \zeta_t = (63.85 - 21.6)13 \times 80 = 43940\ N$$

Shearing resistance of the rivets,

$$P_s = \frac{4}{\pi} . d^2 . \eta . n = \frac{4}{\pi} \times (21.6)^2 \times 60 \times 2 = 43940\ N$$

and crushing resistance of the rivets,

$P_c = d \times t \times \zeta_c \times n = 21.6 \times 13 \times 120 \times 2 = 67392$ N The least of P_t, P_s and P_c is $P_t = 43940$ N.

Efficiency of the joint

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

$= p \times t \times \zeta_t = 63.85 \times 13 \times 80 = 66404$ N

$\therefore$ Efficiency of the joint= $\eta = \frac{\text{least of } Pt, Ps \text{ and } P\,c}{P}\ \%$

e

$$\eta = \frac{43940}{66404}\ \% = 0.66 \text{ or } 66\ \%$$

Home work

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 ζt = 120 MPa; η = 90 MPa and ζc = 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 :*
ζt = 140 MPa ; η = 110 MPa and ζc = 240 MPa .Find the efficiency of the joint. *[Ans. 53.5%]*
3. *A single riveted double cover butt joint is made in(10 mm) thick plates with (20 mm) diameter rivets with a pitch of (60 mm). Calculate the efficiency of the joint, if ζ_t = 100 MPa ; η = 80 MPa and ζc = 160 MPa.*
[Ans. 53.8%]

The welding

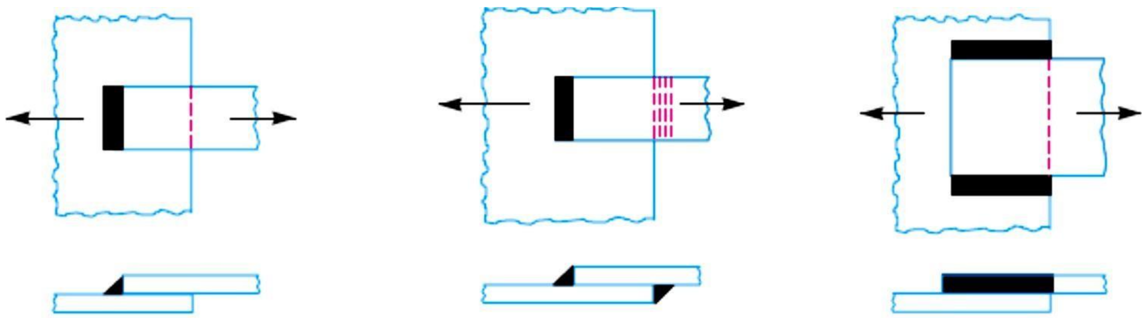
A welded joint is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together, with or without the application of pressure and a filler material

Types of Welded Joints

Following two types of welded joints are important from the subject point of view:
1. Lap joint or fillet joint, and 2. Butt joint

1- Lap Joint

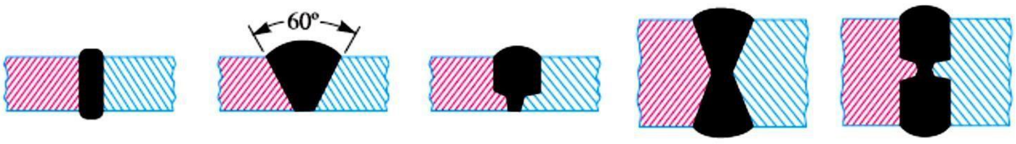
The lap joint or the fillet joint is obtained by overlapping the plates and then welding the edges of the plates. The cross-section of the fillet is approximately triangular. The fillet joints may be
1. Single transverse fillet, 2. Double transverse fillet, and 3. Parallel fillet joints.



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

2- Butt Joint

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



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

The other type of welded joints are corner joint, edge joint and T-joint

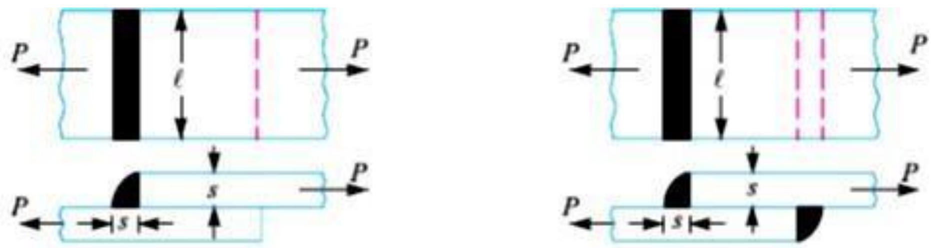
(a) Corner joint. (b) Edge joint. (c) T-joint.

Design of welding joint

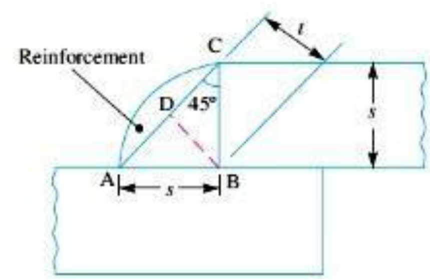
1- Lap or Fillet Welded Joints

a. Single Transverse fillet weld

The transverse fillet welds are designed for tensile strength.



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



t = Throat thickness s = Leg or size of weld,=

Thickness of plate, and l = Length of weld,

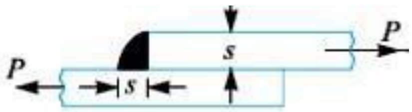
we find that the throat thickness, t = s × sin 45° = 0.707 s Minimum

area of the weld or throat area,

A = Throat thickness × Length of weld

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

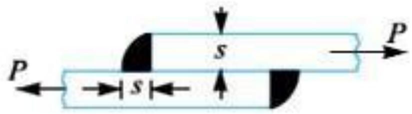
If (ζ_t *) is the allowable tensile stress for the weld metal, then the tensile strength of the joint for single fillet weld, P = Throat area × Allowable tensile stress*



(*a*) Single transverse fillet weld.

$$P = 0.707\ s \times l \times \zeta_t$$

b- double Transverse fillet welds joint

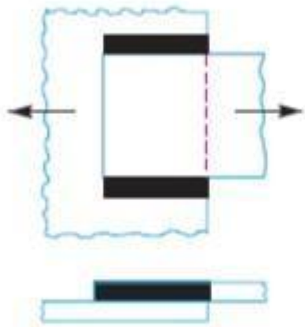


(*b*) Double transverse fillet weld.

and tensile strength of the joint for double fillet weld,

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

c- Parallel Fillet Welded Joints



(*c*) Parallel fillet.

The parallel fillet welded joints are designed for shear strength

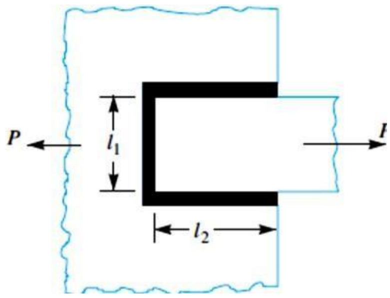
P = Throat area × Allowable shear stress

$$P = 0.707\ s \times l \times \eta$$ *for single*

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

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

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

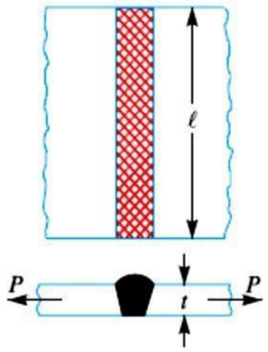


$$P = 0.707s \times l_1 \times \zeta_t + 2 \times 0.707 \times s \times l_2 \times \eta$$

2- Strength of Butt Joints

The butt joints are designed for tension or compression .Consider a single 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),



(a) Single V-butt joint.

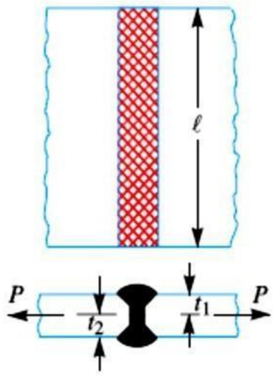
$$P = t \times l \times \zeta_t$$

where l = Length of weld. It is generally equal to the width of plateand tensile strength for double-V butt join is given by

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

Where t_1 = Throat thickness at the top, and t_2 =

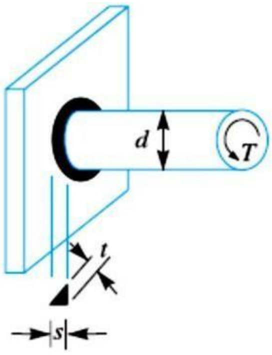
Throat thickness at the bottom.



(b) Double V-butt joint.

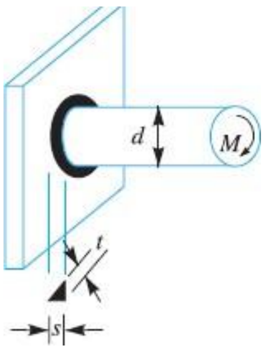
ملاحظات:-

- ارا كان الحام على شكل دائزة ومضع حجج حاثيز عزم الخىءا كما مبيه بالشكل الخالي وسخخذم العلاقت الخاليت:



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

- وارا كان اللحام على شكل دائزة ومضع حجج حاثيز عزم اوحاء كما مبيه بالشكل الخالي وسخخذم العلاقت الخاليت:



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

<u>Example(1)</u>. *A plate (100 mm) wide and (10 mm) thick is to be welded to another plate by means of double parallel fillets. The plates are subjected to a static load of (80 kN). Find the length of weld if the permissible shear stress in the weld does not exceed (55 MPa).*

Solution. Given: *Width = 100 mm ;*

Thickness = 10 mm ; $P = 80$ *kN* $= 80 \times 10^3$ *N ;* $\eta =$

55 MPa = 55 N/mm^2 *Let l =Length of weld, and s =*

Size of weld = Plate thickness = 10 mm

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

$$P = 2 \times 0.707\, s \times l \times \eta$$

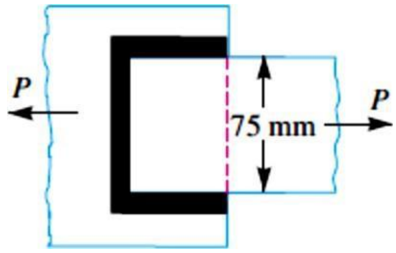
$$80 \times 10^3 = 1.414 \times 10 \times l \times 55 = 778\, l$$

$$\therefore l = 80 \times 10^3 / 778 = 103 \text{ mm}$$

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

$l = 103 + 12.5 = 115.5$ mm

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 weld as shown in Fig. The maximum tensile and shear stresses are(70MPa) and(56 MPa) respectively .Find the length of each parallel fillet weld,*



Given *: Width = 75 mm ; s = 12.5 mm* $\zeta_t = 70$ *MPa*

$= 70$ N/mm^2 ; $\eta = 56$ 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.

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

Let l_2 = Length of each parallel fillet .We know that the maximum load which the plate can carry is

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

Load carried by single transverse weld,

$P_1 = 0.707\ s \times l_1 \times \zeta_t = 0.707 \times 12.5 \times 62.5 \times 70 = 38\ 664$ N

and the load carried by double parallel fillet weld, $P_2 = 1.414$

$s \times l_2 \times \eta = 1.414 \times 12.5 \times l_2 \times 56 = 990\ l_2$ N

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

$65\ 625 = P_1 + P_2 = 38\ 664 + 990\ l_2$ or $l_2 = 27.2$ mm

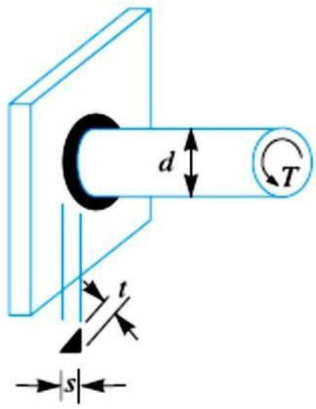
Adding 12.5 mm for starting and stopping of weld run, we have

$l_2 = 27.2 + 12.5 = 39.7$ say 40 mm

Example (3) *A(50 mm) diameter solid shaft is welded to a flat plate by (10 mm)fillet weld as shown in Fig.. Find the maximum torque that the welded joint can sustain if the maximum shear stress intensity in the weld material is not to exceed (80 MPa.).*

Solution .Given : d = 50 mm ; s = 10 mm ;

$\eta_{max} = 80\ MPa = 80\ N/mm^2$



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

$$80 = \frac{2T}{\pi \times 0.707\ (50)^2}$$

$$80 = \frac{2T}{7855}$$

$$T = 80 \times 78\,550/2.83$$
$$= 2.22 \times 10^6\ N\text{-}mm$$

Home work

1. *A plate(100 mm) wide and (10 mm) thick is to be welded with another plate by means of double transverse welds at the ends. If the plates are subjected to a load of (70 kN), find the size of weld for static as well as fatigue load. The permissible tensile stress should not exceed (70 MPa)*

[Ans. 83.2 mm; 118.5 mm]

2. *A circular steel bar (50 mm) diameter and (200 mm) long is welded perpendicularly to a steel plate to form a cantilever to be loaded with (5 kN)at the free end. Determine the size of the weld, assuming the allowable stress in the weld as (100 MPa).*

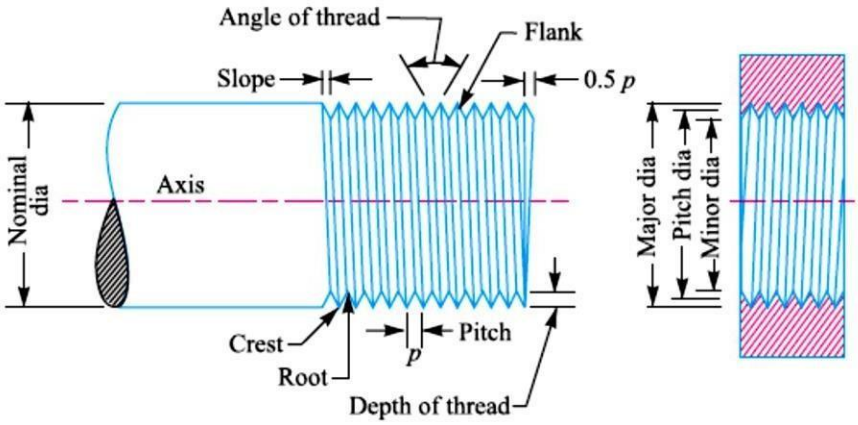
[Ans. 7.2 mm]

Screwed Joint

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

Important Terms Used in Screw Threads

The following terms used in screw threads, as shown in Fig , are important from the subject point of view :



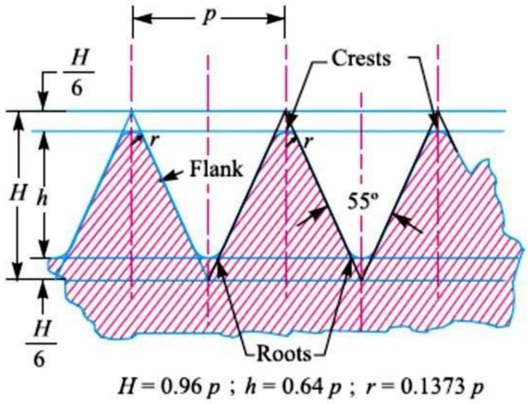
1. **Major diameter**. It is the largest diameter of an external or internal screw thread. The screw is specified by this diameter. It is also known as outside or nominal diameter.
2. **Minor diameter**. It is the smallest diameter of an external or internal screw thread. It is also known as core or root diameter.
3. **Pitch diameter**. It is the diameter of an imaginary cylinder, on a cylindrical screw thread,
4. **Pitch**. It is the distance from a point on one thread to the corresponding point on the next.

Forms of Screw Threads

The following are the various forms of screw threads.

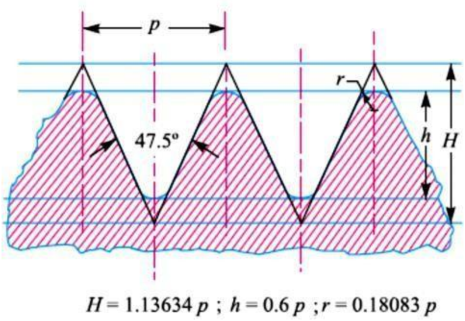
1. British standard whitworth (B.S.W.) thread

These threads are found on bolts and screwed fastenings for special purposes.



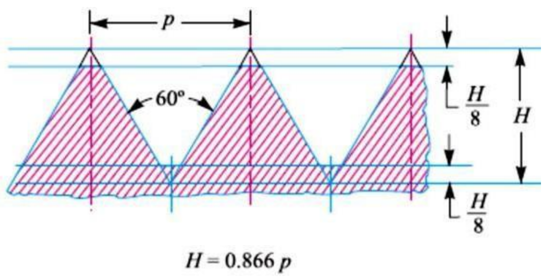
2. British association (B.A.) thread

These threads are used for instruments and other precision works

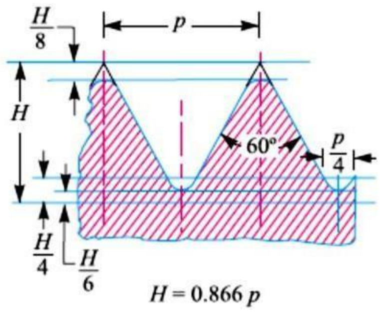


3. American national standard thread.

These threads are used for general purposes e.g. on bolts, nuts, screws and tapped holes

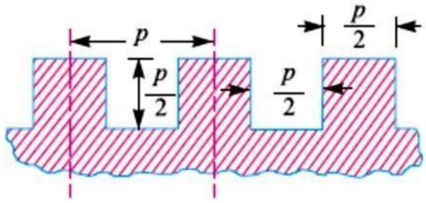


4. Unified standard thread



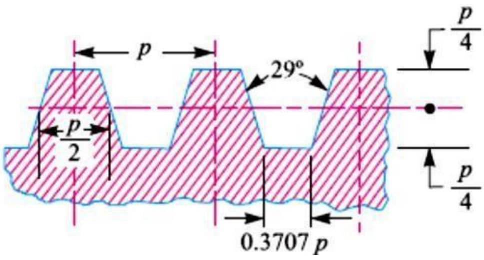
5. Square thread

Are widely 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 etc.



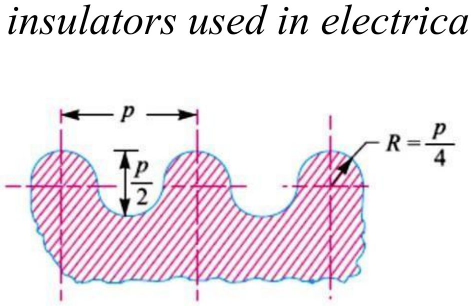
6. Acme thread

These threads are frequently used on screw cutting lathes, brass valves, cocksand bench vices



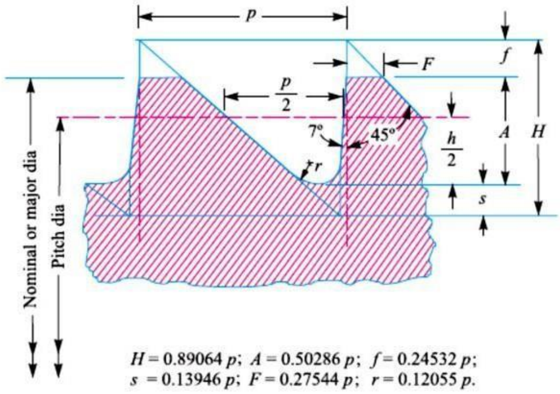
7. Knuckle thread.

They are usually found on railway carriage couplings, hydrants, necks of glass bottles and large molded insulators used in electrical trade

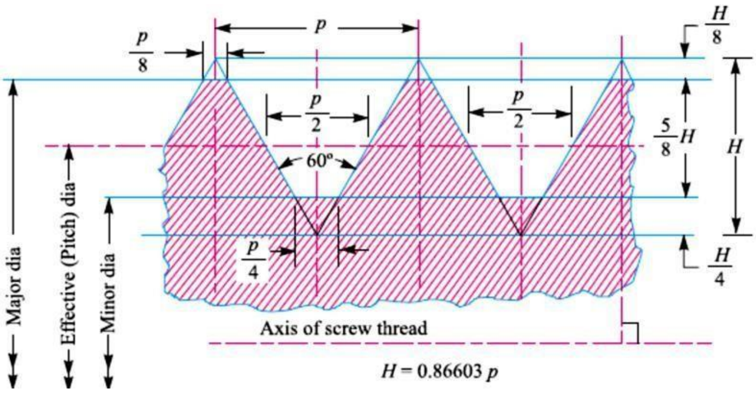


8. Buttress thread

It is used for transmission of power in one direction only.



9. Metric thread.

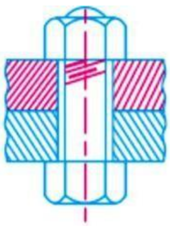


Common Types of Screw Fastenings

Following are the common types of screw fastenings :

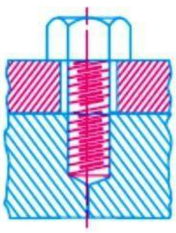
1. Through bolts

Their usage may be known as machine bolts, carriage bolts, automobile bolts, eye bolts



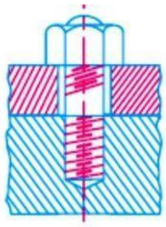
2. Tap bolts

It is screwed into a tapped hole of one of the parts to be fastened without the nut

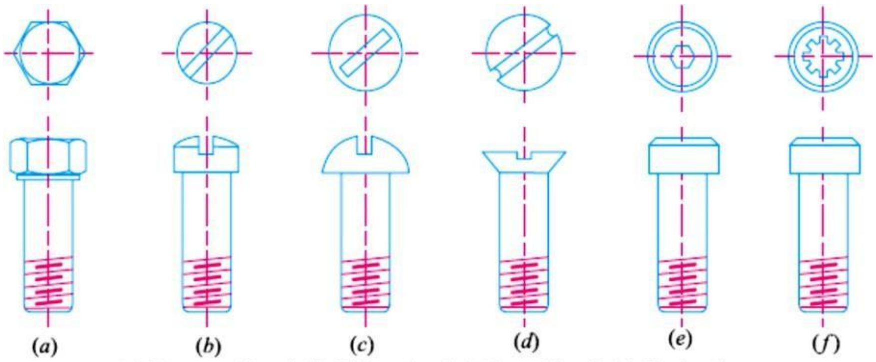


3. *Studs*

Studs are chiefly used instead of tap bolts for securing various kinds of covers e.g. covers of engine and pump cylinders, valves, chests etc.



4. *Cap screws.*



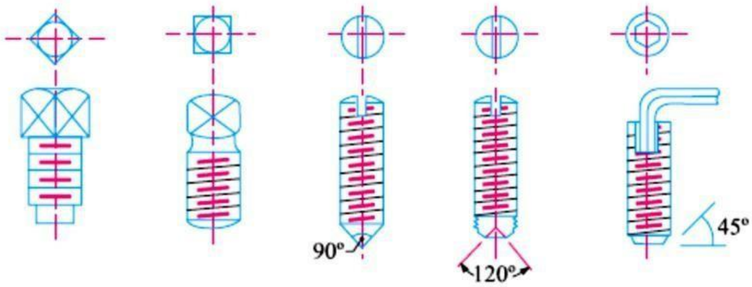
(*a*) Hexagonal head; (*b*) Fillister head; (*c*) Round head; (*d*) Flat head; (*e*) Hexagonal socket; (*f*) Fluted socket.

5. *Machine screws*.

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

6. *Set screws*

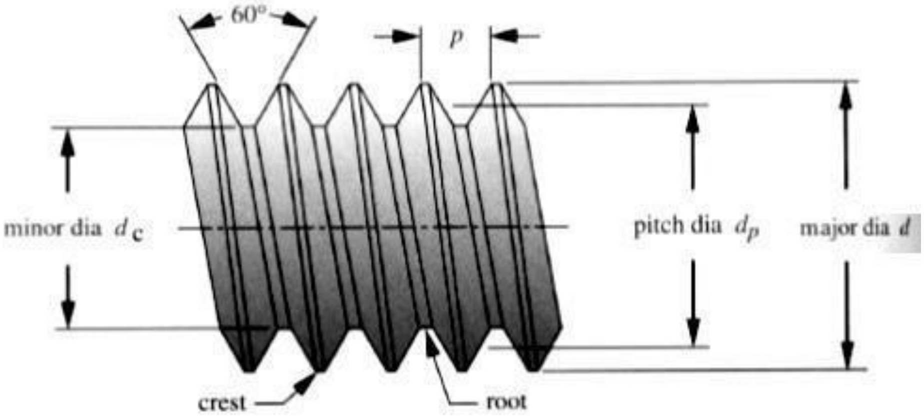
These are used to prevent relative motion between the two parts



Initial Stresses due to Screwing up Forces

The following stresses are induced in a bolt, screw or stud when it is screwed up tightly.

1. Tensile stress due to stretching of bolt



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

P_i = 2840 d .... N

P_i = Initial tension in a bolt, and , d = Nominal diameter of bolt, in mm

A=Cross-sectional area at bottom of the thread $= \frac{\pi}{4}\left(\frac{dc+dp}{2}\right)^2$ d_c=minor diameter or core dia. , d_p= pitch diameter

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

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

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

$$\tau = \frac{16\,T}{\pi\,(d_c)^3}$$

τ = Torsional shear stress, T = Torque applied, and

d_c = Minor or core diameter of the thread.

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

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

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

The average thread shearing stress for the nut is

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

where d = Major diameter.

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

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

d = Major diameter,

d_c = Minor diameter, and

n = Number of threads in engagement.

<u>Stresses due to External Forces</u>

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

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

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

d_c = Root or core diameter of the thread, and ζ_t =

Permissible tensile stress for the bolt material *n=a number of bolts P=external*

load applied

In case the standard table is not available, then for coarse threads, $d_c = 0.84\, d$

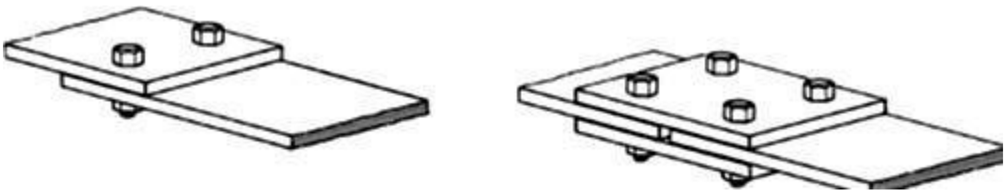
2. Shear stress

Let d = Major diameter of the bolt, and

n = Number of bolts.

∴ Shearing load carried by the bolts,

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



3. Combined tension and shear stress

Maximum principal shear stress,

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

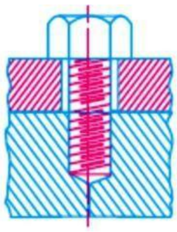
and maximum principal tensile stress,

$$\sigma_{t(max)} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{(\sigma_2)^2 + 4\tau^2}$$

Example (1)

Two machine parts are fastened together tightly by means of a (24 mm) tap bolt. If the load tending to separate these parts is neglected, find the stress that is set up in the bolt by the initial tightening.

Solution .Given : d = 24 mm



Let ζ_t = Stress set up in the bolt.

$P = 2840\,d = 2840 \times 24 = 68\,160$ N $d_c = 0.84\,d = 0.84 \times 24 = 20.15$ mm

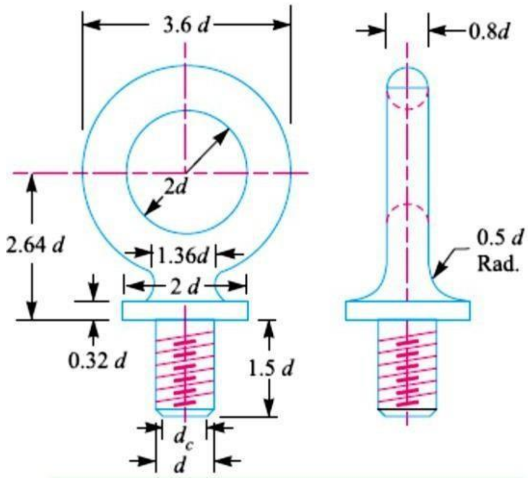
$$\zeta = \frac{Pi}{A}$$

$$\sigma_t = \frac{68\,160}{\frac{\pi}{4}(d_c)^2} = \frac{68\,160}{\frac{\pi}{4}(20.15)^2} = \frac{68\,160}{324} = 215\ \text{N/mm}^2$$

Example (2).

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 MPa).

Solution .Given : $P = 60$ kN $= 60 \times 10^3$ N ; $\zeta_t = 100$ MPa $= 100$ N/mm^2



$$60 \times 10^3 = \frac{\pi}{4}(d_c)^2\sigma_t = \frac{\pi}{4}(d_c)^2\,100 = 78.55\,(d_c)^2$$

$$60 \times 10^3 = 78.55\,(d_c)^2$$

$$(d_c)^2 = 600 \times 10^3 / 78.55 = 764 \quad \text{or} \quad d_c = 27.6 \text{ mm}$$

$$d_c = 0.84\,d \implies d = \frac{d_c}{0.84} = \frac{27.6}{0.84} = 32.85 \text{ mm}$$

Example (3)

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 MPa).

Solution *.Given : T = 25 N-m = 25 × 10^3 N-mm ; n = 4 ; R_p = 30 mm ;*

η = 30 MPa = 30 N/mm^2

$$P_s = \frac{T}{R_p} = \frac{25 \times 10^3}{30} = 833.3 \text{ N}$$

∴ Resisting load on the bolts $P_s = \frac{\pi}{4} \times d^2 \times \tau \times n$

$$= \frac{\pi}{4}(d\ )^2 \tau \times n = \frac{\pi}{4}(d\ )^2\ 30 \times 4 = 94.26\ (d\ )^2$$

$(d\ )^2 = 833.3 / 94.26 = 8.84$ or $d = 2.97$ mm

<u>Home work</u>

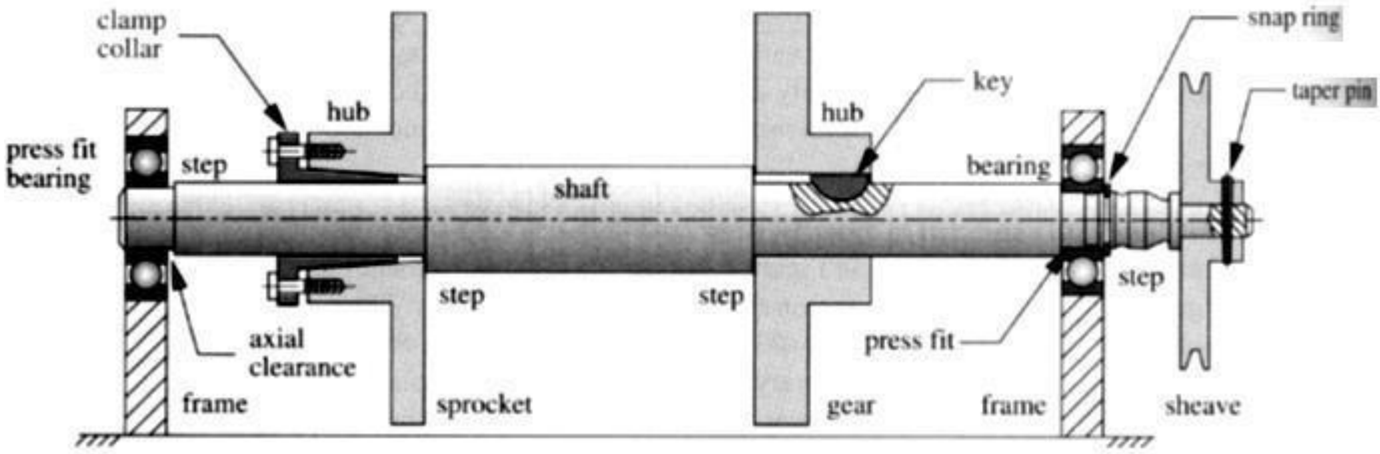
1. *Determine the safe tensile load for bolts of(M 20) and (M 36). Assume that the bolts are not initially stressed and take the safe tensile stress as (200 MPa).*

   [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]*

<u>هلاحظث</u>: *d= M* القطش الكبيش للبشغي بالبلن

<u>***keys***</u>

A key is a piece of mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them It is always inserted parallel to the axis of the shaft. Keys are used as temporary fastenings and are subjected to considerable crushing and shearing stresses



Types of Keys

The following types of keys are important from the subject point of view :

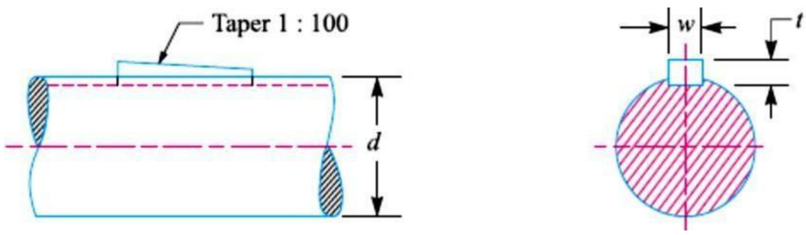
1. Sunk keys,

The sunk keys are of the following types :

***a. Rectangular sunk key**. A rectangular sunk key. The usual proportions of this key are :*

Width of key, $w = d / 4$ and thickness of key, $t = 2w / 3 = d / 6$

where d = Diameter of the shaft or diameter of the hole in the hub



***b. Square sunk key**.*

The only difference between a rectangular sunk key and a square sunk key is that its width and thickness are equal, i.e.

$$w = t = d / 4$$

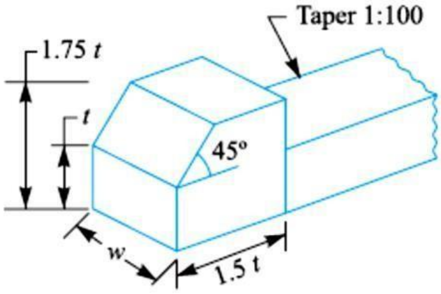
***c. Parallel sunk key**. The parallel sunk keys may be of rectangular or square section uniform in width and thickness.*

d. Gib-head key

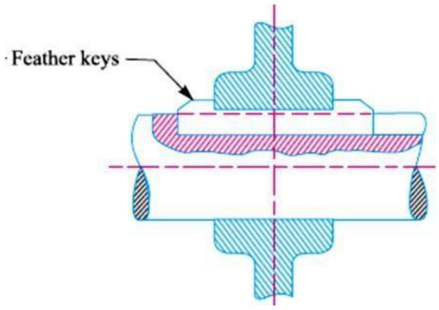
The usual proportions of the gib head key are :

Width, $w = d / 4$;

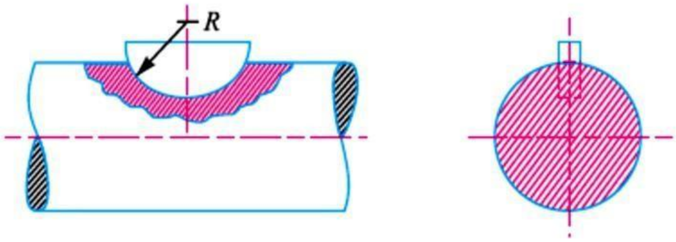
and thickness at large end, $t = 2w / 3 = d / 6$



e. Feather key



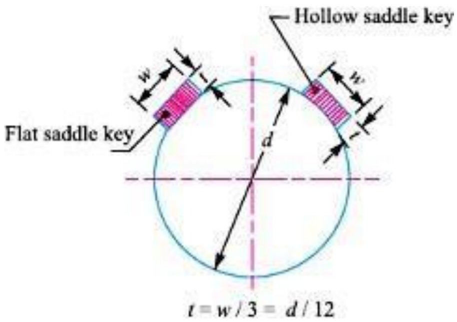
f. Woodruff key



2. Saddle keys

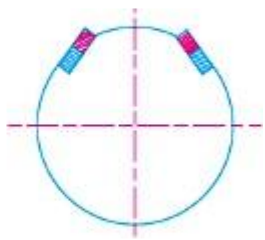
a.flat saddle key *is a taper key which fits in a keyway in the hub and is flat on the shaft*

b.hollow saddle *key is a taper key which fits in a keyway in the hub and the bottom of the key is shaped to fit the curved surface of the shaft.*



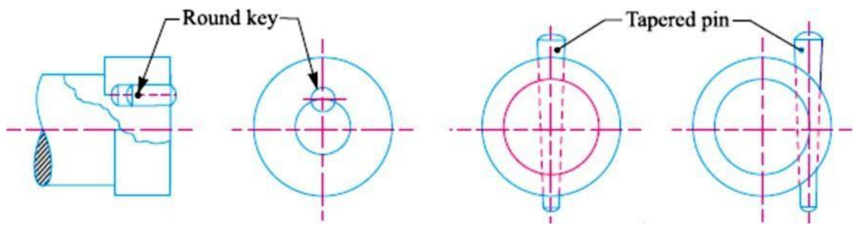
3. Tangent Keys

The tangent keys are fitted in pair at right .



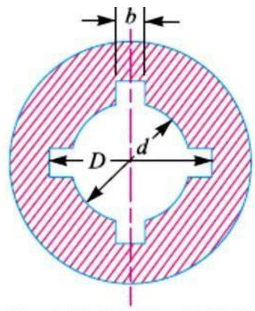
These are used in large heavy duty shafts

4. Round Keys



5. Splines

Sometimes, keys are made integral with the shaft which fits in the keyways broached in the hub



$D = 1.25\ d$ and $b = 0.25\ D$

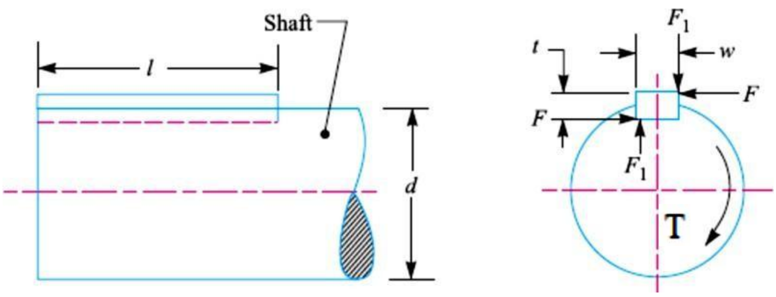
Forces acting on a Sunk Key

When a key is used in transmitting torque from a shaft to a rotor or hub, the following two types of forces act on the key :

***1. Forces (F_1)** due to fit of the key in its keyway,. These forces produce compressive stresses in the key which are difficult to determine in magnitude.*

***2. Forces (F)** due to the torque transmitted by the shaft. These forces produce shearing and compressive (or crushing) stresses in the key.*

8



T = Torque transmitted by the shaft,

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

l = Length of key,

w = *Width of key.*

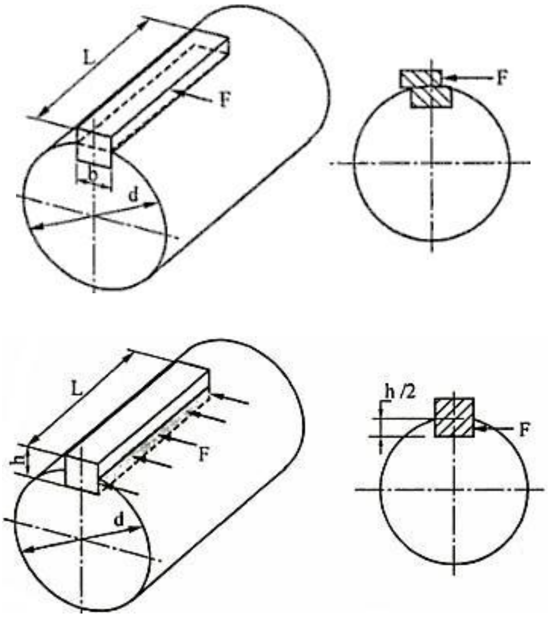
t = *Thickness of key, and* η *and* ζ_c = *Shear and crushing stresses for the material of key*

1. Shear stresses

F = Area resisting shearing × Shear stress

Area resisting shearing = $l \times w$

$F = l \times w \times \eta$



Torque transmitted by the shaft, $T = F \times \frac{d}{2}$

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

2. crushing stresses

Area resisting crushing = $l \times \frac{t}{2}$

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

$\therefore$ Torque transmitted by the shaft, $T = F \times \frac{d}{2}$

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

torsional shear strength of the shaft,

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

To find width of key by **equally strong in shearing and crushing, if**

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

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

And to find the length of key by equally shearing strength of the key to the torsional shear strength of the shaft

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

When the key material is same as that of the shaft, then $\tau = \tau_1$.

$$\therefore \qquad l = 1.571\ d$$

Example (1).

Design the rectangular key for a shaft of(50 mm) diameter. The shearing and crushing stresses for the key material are (42 MPa) and(70 MPa) with cross section of key (16 x 10) mm.

Solution *.Given : d = 50 mm ; η = 42 MPa = 42 N/mm^2 ;*

ζ_c = 70 MPa = 70 N/mm^2 , w = 16 mm. t = 10 mm.

Considering shearing of the key

$$T = l \times w \times \tau \times \frac{d}{2} = l \times 16 \times 42 \times \frac{50}{2} = 16\,800\ l \ \text{N-mm}$$

The torque transmitted of the shaft

$$T = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 42\ (50)^3 = 1.03 \times 10^6 \ \text{N-mm}$$

The torsional shearing strength(or torque transmitted) of the key is equal to the torsional shear strength of the shaft

$$l = 1.03 \times 10^6 / 16\,800 = 61.31 \ \text{mm}$$

Now considering crushing torque of the key

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2} = l \times \frac{10}{2} \times 70 \times \frac{50}{2} = 8750\ l \text{ N-mm}$$

The torsional crushing strength(or torque transmitted) of the key is equal to the torsional shear strength of the shaft

$$l = 1.03 \times 10^6 / 8750 = 117.7 \text{ mm}$$

Taking larger of the two values, we have length of key,

$$l = 117.7 \text{ say } 120 \text{ mm} \quad \textbf{Ans.}$$

Example (2)

Determine the dimensions of the rectangular sunk key made up of mild steel for (80mm) diameter of mild steel shaft to transmit a torque of (135 N.m) .Assume Shear stress ($50 N/mm^2$) and crushing stress ($120 N/mm^2$) assume cross section of key (22 x 14)mm.

Solution *.Given : $d = 80$ mm ; $\eta = 50\ N/mm^2$*

$\zeta_c = 120\ N/mm^2$

$w = 22$ mm. $t = 14$ mm

torque $T=135\ N.m=135\times10^3\ N.mm$

$$L=1.5\ d =1.5 \times 80 =120\ mm$$

1. *Check for shear strength of key*

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

$$\tau = \frac{2T}{l\,w\,d} = \frac{2 \times 135\times10^3}{120\times22\times80} = 1.278\ N/mm^2$$

Which is less than the allowable strength .Hence the dimensions are in the safe limit.

2. *Check for crushing strength of key*

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

$$\sigma_c = \frac{4T}{l\,t\,d} = \frac{4\times135\times10^3}{120\times14\times80} = 4.02N/mm^2$$

Which is less than the allowable strength .Hence the dimensions are in the safe limit

Example (3)

A square key of (10×10×75mm) dimension is require to transmit (1100 N.m) torque from a (60 mm)diameter solid shaft .Determine whether the length is sufficient or not if the permissible shear stress(60 Mpa) and crushing stress (170 Mpa) .

Solution *.Given : d = 60 mm ; η = 60 Mpa= 60 N/mm²*

2 ζc

= 170 Mpa= 170 N/mm

w = 10 mm.

t = 10 mm

L=75mm torque T=1100

N.m=1100×10³ N.mm

1. *Check for shear strength of key*

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

$$\tau = \frac{2T}{l\,\mathrm{w}\,d} = \frac{2\times1100\times10^3}{75\times10\times60} = 48.89 \ N/mm^2$$

Which is less than the permissible value (60 Mpa). Hence the length is sufficient .

2.check for crushing strength of key

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

$$\sigma_c = \frac{4T}{l\,t\,d} = \frac{4\times1100\times10^3}{75\times10\times60} = 97.78 \ N/mm^2$$

Which is also less than the permissible. Hence the length is sufficient .

Home work

1. *A shaft 80 mm diameter transmits power at maximum shear stress of (63 MPa). Find the length of a(20mm) wide key required to mount a pulley on the shaft so that the stress in the key does not exceed (42MPa).*
   [Ans. 152 mm]
2. *A shaft (30 mm)diameter is transmitting power at a maximum shear stress of (80 MPa). 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 MPa) and length of the key is (4) times the width.*
   [Ans. l = 126 mm]

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 will, without stopping the driving shaft. The use of a clutch is mostly found in automobiles

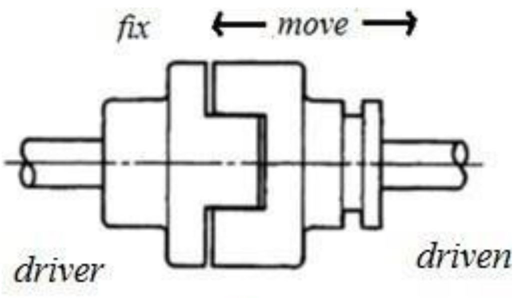
Types of Clutches

Following are the two main types of clutches commonly used in engineering practice : 1. Positive clutches, and 2. Friction clutches.

1. Positive clutches

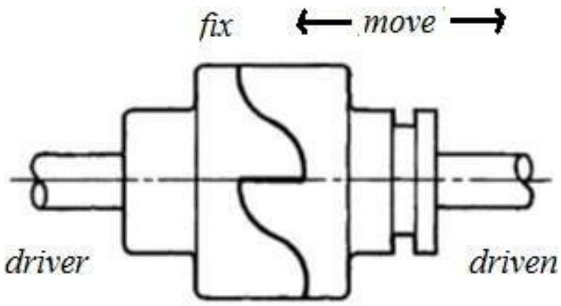
a. square jaw clutch :-

A square jaw type is used where engagement and disengagement in motion and under load is not necessary



a

b. Spiral jaw clutch

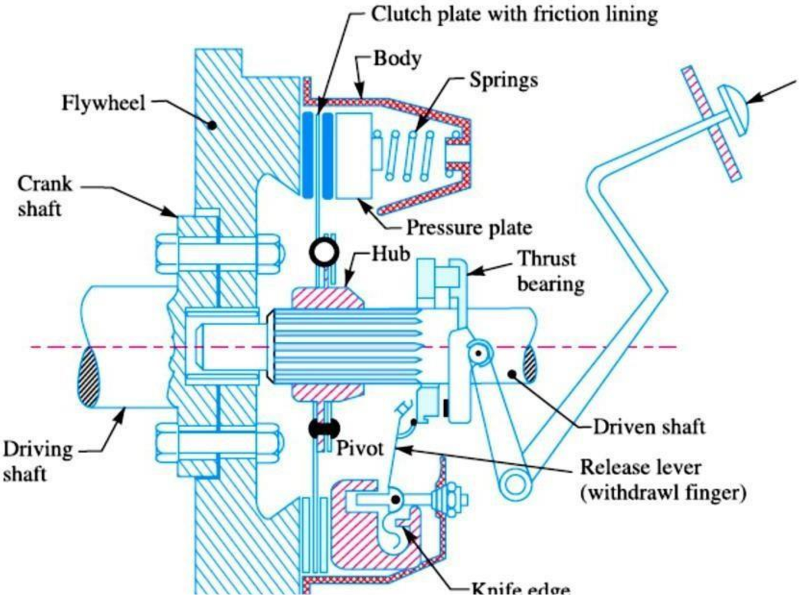


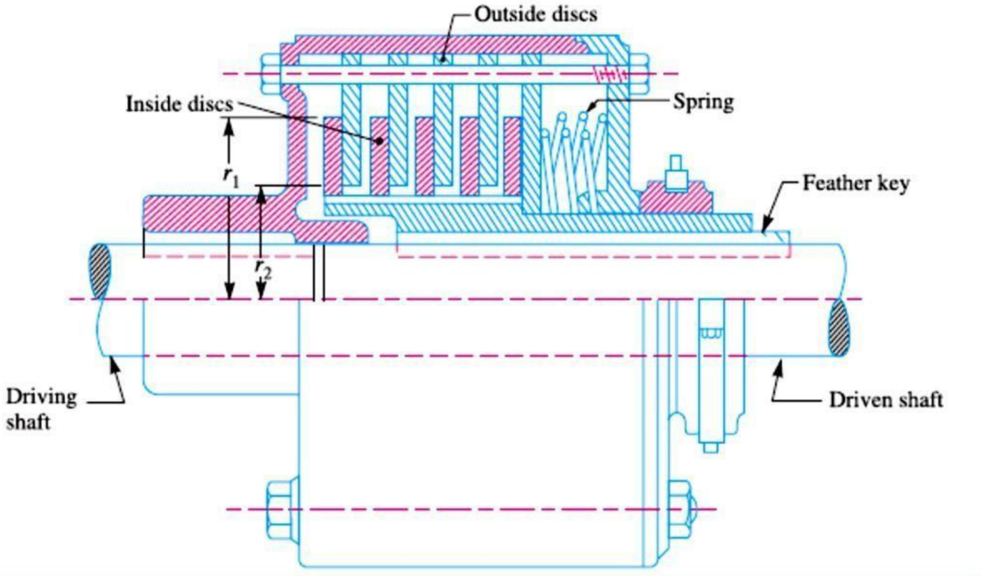
b

2. Friction clutches

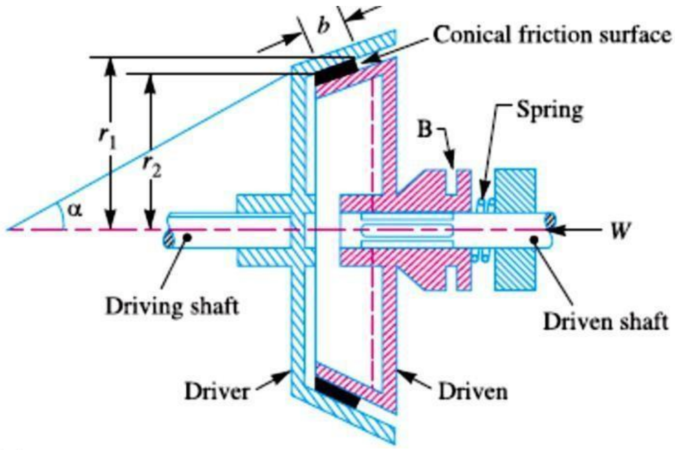
There are many types of friction clutches, Disc or plate clutches (single disc or multiple disc clutch), Cone clutches, and Centrifugal clutches

1. Disc or plate clutches (single disc or multiple disc clutch),

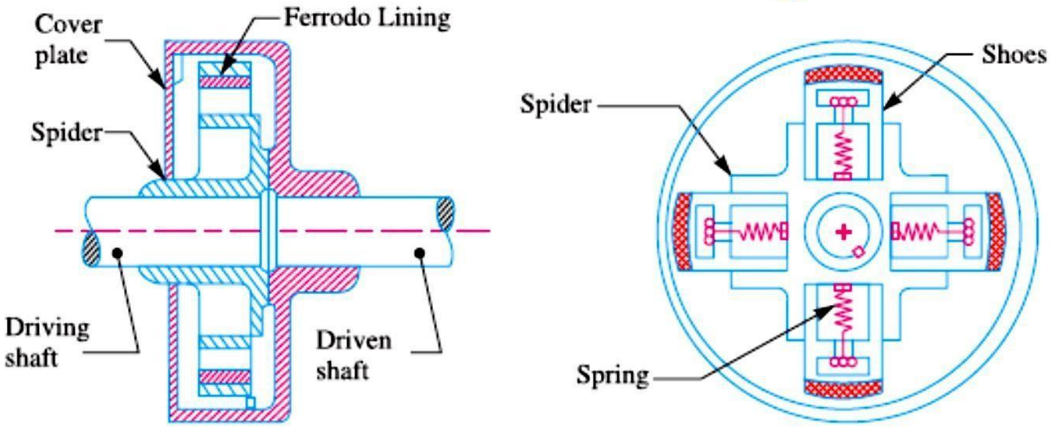




2. Cone clutches

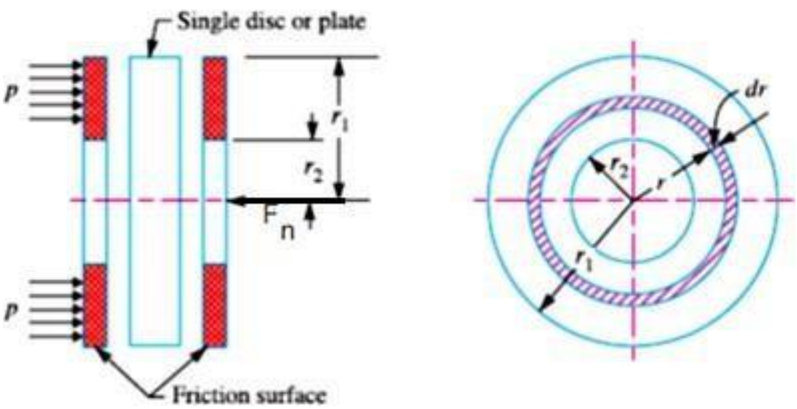


3. Centrifugal clutches



Design of a Disc or Plate Clutch

Consider two friction surfaces maintained in contact by an axial thrust (F_n) as shown in Fig



F_n= Normal or axial force on the ring T = Torque transmitted by the clutch, p = Intensity of axial pressure with which the contact surfaces are held together, r_1 and r_2 = External and internal radius of friction

faces, r = Mean radius of the friction face, and μ = Coefficient of friction

We know that area of the contact surface or friction surface$= 2\pi\, r.dr$

$\therefore$ *Normal or axial force on the ring,* F_n = *Pressure* × *Area* $= p \times 2\pi\, r.dr$
and the frictional force (F_r) *on the ring acting tangentially at radius r,*

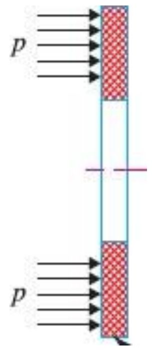
$$F_r = \mu \times F_n = \mu.p \times 2\pi\, r.dr$$

$\therefore$ *Frictional torque acting on the ring,* $T_r = F_r \times r = \mu.p \times 2\pi\, r.dr \times r$

$$= 2\,\pi\,\mu\, p.\, r^2.dr$$

1. Considering uniform pressure

When the pressure is uniformly distributed over the entire area of the friction face , then the intensity of pressure, Assumption:-



- *New clutch*
- *Using more number of spring*

Integrating equations within the limits from r_2 *to* r_1 *.*

$$F_n = p\,2\pi r\,.dr$$

اولا - تكامل معادلة القوة لايجاد قانون الضغط

$$F_n = \int_{r_2}^{r_1} p\,2\pi r\,.dr = \pi\,p\left[r^2\right]_{r_2}^{r_1} = \pi\,p\left[(r_1)^2 - (r_2)^2\right]$$

$$p = \frac{F_n}{\pi\left[(r_1)^2 - (r_2)^2\right]}$$

ثانيا - تكامل معادلة العزم لايجاد قانون العزم المنقول

$$T = \int_{r_2}^{r_1} 2\pi\,\mu.p.r^2.dr = 2\pi\mu.p\left[\frac{r^3}{3}\right]_{r_2}^{r_1}$$

$$= 2\pi\,\mu.p\left[\frac{(r_1)^3 - (r_2)^3}{3}\right] = 2\pi\,\mu \times \frac{F_n}{\pi\left[(r_1)^2 - (r_2)^2\right]}\left[\frac{(r_1)^3 - (r_2)^3}{3}\right]$$

... (Substituting the value of p)

$$= \frac{2}{3}\mu.F_n\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right] = \mu\,F_n R$$

$$R = \frac{2}{3}\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right]$$ = Mean radius of the friction surface.

2. Considering uniform axial wear

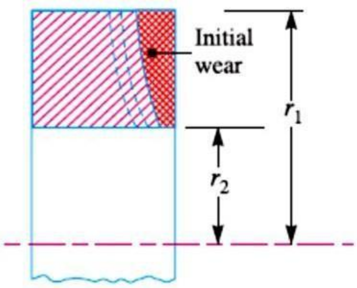
The work of friction is proportional to the product of normal pressure (p) and the sliding velocity (V=2πrN). Therefore,

Normal wear ∝ Work of friction ∝p.V

Or p.V = K (a constant) or p = K /V

Since the intensity of pressure varies inversely with the distance, therefore

p.r = C (a constant) or p = C / r Assumption:-



- *old clutch*
- *The work of friction is proportional to the product of normal pressure (p) and the sliding velocity (V)*

Total force acing on the friction surface,

$$F_n = p.2\pi r.dr = \frac{C}{r} \times 2\pi r.dr = 2\pi C.dr$$

اولا - تكامل معادلة القوة لايجاد قيمة الثابت C

$$F_n = \int_{r_2}^{r_1} 2\pi\, C\, dr = 2\pi\, C\, [r]_{r_2}^{r_1} = 2\pi\, C\,(r_1 - r_2)$$

$$p.r = C = \frac{F_n}{2\pi\,(r_1 - r_2)}$$

We know that the frictional torque acting on the ring,

$$T_r = 2\pi\,\mu.p r^2\,dr = 2\pi\,\mu \times \frac{C}{r} \times r^2\,dr = 2\pi\mu.C r.dr \qquad ...(\because p = C/r)$$

∴ Total frictional torque acting on the friction surface (or on the clutch),

ثانيا - تكامل العزم لايجاد قانون العزم

$$T = \int_{r_2}^{r_1} 2\pi\,\mu\, C r.dr = 2\pi\,\mu C\left[\frac{r^2}{2}\right]_{r_2}^{r_1}$$

$$= 2\pi\mu\,.C\left[\frac{(r_1)^2 - (r_2)^2}{2}\right] = \pi\,\mu.C\,[(r_1)^2 - (r_2)^2]$$

$$= \pi\mu \times \frac{F_n}{2\pi\,(r_1 - r_2)}[(r_1)^2 - (r_2)^2] = \frac{1}{2} \times \mu F_n (r_1 + r_2)$$

notes

- *For a single disc or plate clutch, normally both sides of the disc are effective. Therefore a single disc clutch has two pairs of surfaces in contact (i.e. n = 2).*
- *To find torque using the*

$$\text{Torque (T)} = \frac{P \times 60}{2\pi\, n}$$

- *For multiple disc clutch*

  Let n_1 = Number of discs on the driving shaft, and

  n_2 = Number of discs on the driven shaft.

  ∴ Number of pairs of contact surfaces, $n = n_1 + n_2 - 1$

$$T = \frac{2}{3}\mu . F_n\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right]\times n \text{ .... for uniform pressure}$$

$$T = \frac{1}{2}\times \mu . F_n(r_1 + r_2)\times n \text{ ...... for uniform wear}$$

$$P_{max}\, r_2 = c$$

$$P_{min}\, r_1 = c$$

Example(1)

A plate clutch having a single driving plate with contact surfaces on each side is required to transmit(110 kW) at (1250 r.p.m). The outer diameter of the contact surfaces is to be (300 mm). The coefficient of friction is(0.4).

(a) Assuming a uniform pressure of (0.17 N/mm^2); determine the inner diameter of the friction surfaces.

(b) Assuming the same dimensions and the same total axial thrust, determine the maximum torque that can be transmitted and the maximum intensity of pressure when uniform wear conditions have been reached.

Solution *.Given : P = 110 kW = 110 × 10^{3}W ; N = 1250 r.p.m. ; d_1 = 300 mm or r_1 = 150 mm ;μ = 0.4 ; p = 0.17 N/mm^2*

(a) Inner diameter of the friction surfaces

We know that the torque transmitted by the clutch

$$T = \frac{P \times 60}{2\,\pi\,N} = \frac{110 \times 10^3 \times 60}{2\,\pi \times 1250} = 840 \text{ N-m}$$

$$= 840 \times 10^3 \text{ N-mm}$$

$$p = \frac{Fn}{\pi\left[(r_1)^2 - (r_2)^2\right]}$$

$$F_n = p\,\pi[(r_1)^2 - (r_2)^2]$$

$$= 0.17 \times \pi\,[(150)^2 - (r_2)^2] = 0.534\,[(150)^2 - (r_2)^2]$$

$$T = \frac{2}{3}\mu\,F_n\left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right) \times n \quad \ldots\ldots for\ uniform\ pressure$$

$$840 \times 10^3 = \frac{2}{3} \times 0.4 \times 0.534\,[\cancel{(150)^2 - (r_2)^2}] \times 2\left[\frac{(150)^3 - (r_2)^3}{\cancel{(150)^2 - (r_2)^2}}\right]$$

$$840 \times 10^3 = 0.285\,[(150)^3 - (r_2)^3]$$

$$(150)^3 - (r_2)^3 = 840 \times 10^3 / 0.285 = 2.95 \times 10^6$$

$$(r_2)^3 = (150)^3 - 2.95 \times 10^6 = 0.425 \times 10^6 \text{ or } r_2 = 75 \text{ mm}$$

$$d_2 = 2r_2 = 2 \times 75 = 150 \text{ mm}$$

(b) Maximum torque transmitted

We know that the axial thrust,

$$p = \frac{Fn}{\pi\left[(r_1)^2 - (r_2)^2\right]}$$

$$F_n = p\,\pi[(r_1)^2 - (r_2)^2]$$

$$F_n = 0.534\,[(150)^2 - (r_2)^2]$$
$$= 0.534\,[(150)^2 - (75)^2] = 9011\text{ N}$$

$$T = \frac{1}{2}\,\mu F_n\left(r_1 + r_2\right) \times n$$

$$T = 0.4 \times 9011 \times 2\left(\frac{150 + 75}{2}\right) = 811 \times 10^3\ N.mm$$

Maximum intensity of pressure

For uniform wear conditions, p.r = C (a constant). Since the intensity of pressure is maximum at the inner radius (r_2), therefore

$$p_{max} \times r_2 = C$$

$$p_{max} \times r_2 = \frac{F_n}{\pi \times 2(r_1 - r_2)}$$

$$p_{max} = \frac{F_n}{r_2 \times \pi \times 2(r_1 - r_2)} = \frac{9011}{75 \times \pi \times 2 \times (150 - 75)}$$
$$= \frac{9011}{35347} = 0.255\ N/mm^2$$

Example(2)

A multi-disc clutch has three discs on the driving shaft and two on the driven shaft. The inside diameter of the contact surface is (120 mm). The maximum pressure between the surface is limited to(0.1 N/mm^2). Design the clutch for transmitting(25 kW) at(1575 r.p.m). Assume uniform wear condition and coefficient of friction as(0.3).

Solution .Given *: $n_1 = 3$; $n_2 = 2$; $d_2 = 120$ mm or $r_2 = 60$ mm ; $p_{max} = 0.1$ N/mm2 ; $P = 25$ kW $= 25 \times 10^3$ W ; $N = 1575$ r.p.m. ; $\mu = 0.3$*

We know that the torque transmitted

$$T = \frac{P \times 60}{2\pi N} = \frac{25 \times 10^3 \times 60}{2\pi \times 1575} = 151.6 \text{ N-m} = 151\,600 \text{ N-mm}$$

For uniform wear, we know that $p.r = C$. Since the intensity of pressure is maximum at the inner radius (r_2), therefore,

$$p_{max} \times r_2 = C \quad \text{or} \quad C = 0.1 \times 60 = 6 \text{ N/mm}$$

$$F_n = p_{max} \times r_2 \times \pi \times 2(r_1 - r_2)$$

$$F_n = 60 \times 0.1 \times \pi \times 2(r_1 - 60) = 37.7(r_1 - 60)$$

We know that number of pairs of contact surfaces,

$$n = n_1 + n_2 - 1 = 3 + 2 - 1 = 4$$

∴ Torque transmitted (T),

$$T = \frac{1}{2}\mu\, F_n\,(r_1 + r_2)\, n$$

$$151600 = \frac{1}{2} \times 0.4 \times 37.7(r_1 - 60)(r_1 + 60) \times 4 =$$

$$151600 = \frac{1}{2} \times 0.3 \times 37.7(r_1^2 + 60r_1 - 60r_1 - 3600) \times 4$$

$$151600 = 22.62\, r_1^2 - 81432$$

$$r_1^2 = \frac{151600 + 81432}{22.62} = 1030$$

$$r_1 = 101.5 \text{ mm}$$

<u>Design of a Cone Clutch</u>

Consider a pair of friction surfaces of a cone clutch as shown in Fig. A little consideration will show that the area of contact of a pair of friction surface .

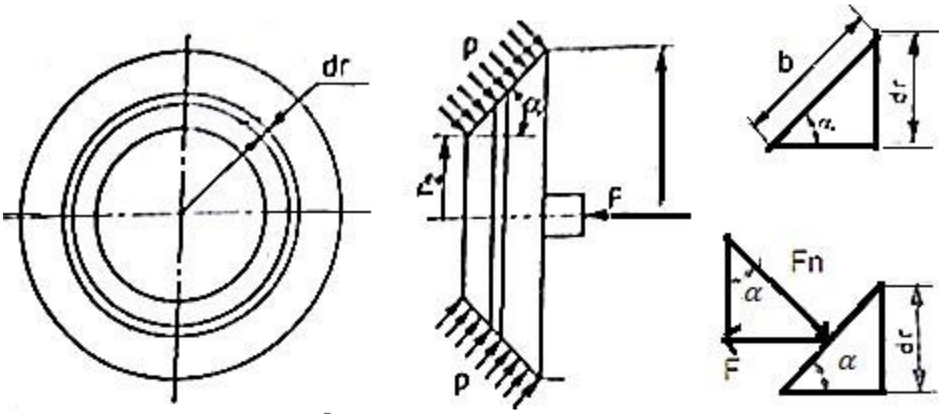
Let p = Intensity of pressure with which the conical friction surface r_1 = Outer radius of friction surface, r_2 = Inner radius of friction surface, R = Mean radius of friction surface = r α = Semi-angle of the cone (also called face angle of the cone) or angle of the friction surface with the axis of the clutch, μ = Coefficient of friction between the contact surfaces,

F_n =normal forces ,

F= axial forces

b= face width

T_f =torque transmitted



$$face\ width\ \ b = \frac{dr}{sin\alpha} = \frac{r_1 - r_2}{sin\alpha}$$

$$area\ \ A = 2\pi\, r\, \frac{dr}{sin\alpha}$$

$$Normal\ force\ on\ the\ ring\ \ F_n = A \times p$$

$$F_n = 2\pi\, r \frac{dr}{sin\alpha} \times p$$

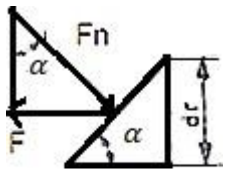
$$The\ frictional\ force \quad F_f = 2\pi\, r \frac{dr}{sin\alpha} \times p \times \mu$$

dr

Axial force $F = 2\pi \int_{r1}^{r2} prdr$ $F=F_n$

$sin\alpha = 2\pi\, r \frac{}{sin\alpha} \times p \times sin\alpha = 2\pi\, rdr\, p$

F= 2π r(r₁- r₂)p



According uniform pressure theory

$$Tf = \frac{2}{3} \times \frac{\mu F}{sin\alpha} \left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right)$$

According uniform wear theory

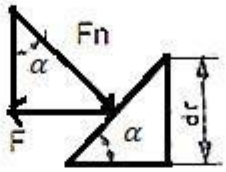
$$Tf = \frac{1}{2} \frac{\mu F}{sin\alpha} (r_1 - r_2)$$

Example(1)

A cone clutch transmits(5 kW) power at (240 r.p.m). The smaller radius of the cone is (200mm)and the face width is (50mm).The cone has a face angle of (15^0).Determine the axial force necessary to engage the clutch if the coefficient of friction at the contact surface is (0.25). Also determine the maximum pressure on the contact surfaces .Assuming uniform wear.

Solution .Given *: ;* $P = 5\ kW = 5 \times 10^3\ W$ *,* $n = 240$ *r.p.m ;* $r_2 = 200$ *mm* $b =$ *50 mm ;* $\alpha = 15^0$ *;* $\mu = 0.25$

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



$$\sin 15^0 = \frac{r_1 - 200}{50}$$

$$\therefore \quad r_1 = 213 \text{ mm}$$

Torque to be transmitted by the clutch,

$$T = \frac{P \times 60}{2\pi N} = \frac{5 \times 1000 \times 60}{2\pi \times 240} \times 1000 = 198.94 \times 10^3 \text{ N-mm}$$

Friction torque, $T_f = \mu F \times \frac{(r_1 + r_2)}{2} \times \frac{1}{\sin 15^0}$

$$198.94 \times 10^3 = 0.25 \times F \times \frac{(213 + 200)}{2} \times \frac{1}{\sin 15^0}$$

$$198.94 \times 10^3 = 199.5 \times F$$

$$F = 997.2 \text{ N}$$

(ii) Maximum pressure on the contact surfaces

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

$$p = \frac{F}{2\pi r_1 (r_1 - r_2)} = \frac{997.2}{2\pi \times 213\,(213 - 200)} = 0.06 \text{ N/mm}^2$$

<u>Home work</u>

1. *(10 kW)power at(900 r.p.m).The axial pressure is limited to(0.085 N/mm^2). If the external diameter of the friction lining is(1.25 times) the internal diameter, find the required dimensions of the friction lining and the axial force exerted by the springs. Assume uniform wear conditions. The coefficient of friction may be taken as(0.3).*
[Ans. 132.5 mm ; 106 mm ; 1500 N]

2. *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.* *[Ans. 90 mm ; 2375 N]*

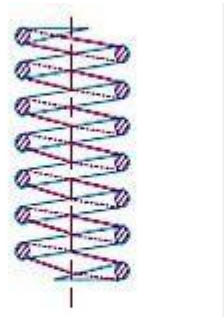
3. *A multiple disc clutch has three discs on the driving shaft and two on the driven shaft, providing four pairs of contact surfaces. The outer diameter of the contact surfaces is(250 mm) and the inner diameter is (150 mm). Determine the maximum axial intensity of pressure between the discs for transmitting (18.75 kW)at (500 r.p.m). Assume uniform wear and coefficient of friction as(0.3).*

Spring

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. The various important applications of springs are as follows :

1. *To cushion, absorb or control energy due to either shock or vibration as in car springs, railway buffers, air-craft landing gears, shock absorbers and vibration dampers.*
2. *To apply forces, as in brakes, clutches and spring loaded valves.*
3. *To control motion by maintaining contact between two elements as in cams and followers.*
4. *To measure forces, as in spring balances and engine indicators.*
5. *To store energy, as in watches, toys, etc.* ***Types of Springs***

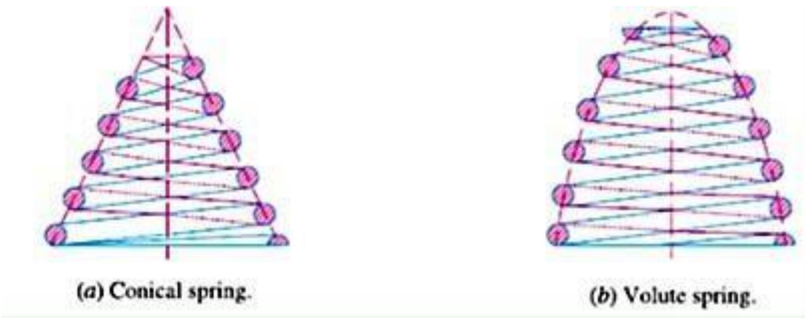
1. *Helical springs: is primarily intended for compressive or tensile loads.*



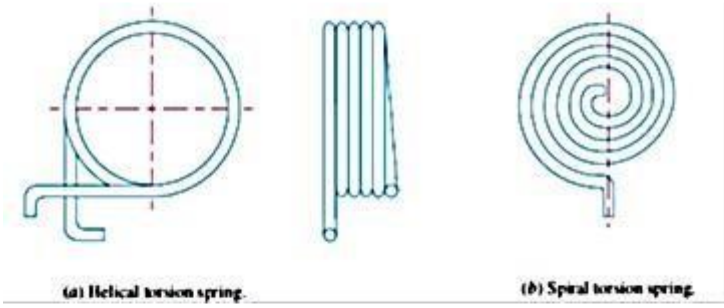
(*a*) Compression helical spring.

(*b*) Tension helical spring.

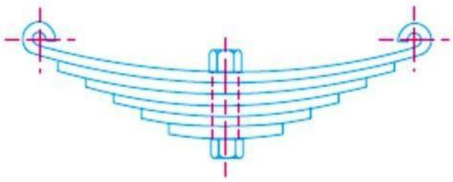
2. *Conical and volute spring :These springs may be of conical spring and volute springs*



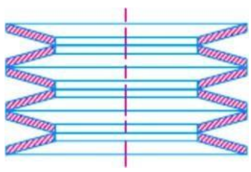
3. Torsion springs; These springs may be of helical or spiral type



4. Laminated or leaf springs: The laminated or leaf spring (also known as flat spring or carriage spring) consists of a number of flat plates (known as leaves) of varying lengths held together by means of clamps and bolts,. These are mostly used in automobiles. The major stresses produced in leaf springs are tensile and compressive stresses.



5. Disc or Belleville springs: These springs consist of a number of conical discs held together against slipping by a central bolt or tube. The major stresses produced in disc or Belleville springs are tensile and compressive stresses.



6. *Special purpose springs. These springs are air or liquid springs, rubber springs, ring springs etc. The fluids (air or liquid) can behave as a compression spring. These springs are used for special types of application only.*

Table 23.1. Values of allowable shear stress, Modulus of elasticity and Modulus of rigidity for various spring materials.

Material	Allowable shear stress (τ) MPa			Modulus of rigidity (G) kN/m²	Modulus of elasticity (E) kN/mm²
	Severe service	Average service	Light service		
1. Carbon steel					
(a) Upto to 2.125 mm dia.	420	525	651		
(b) 2.125 to 4.625 mm	385	483	595		
(c) 4.625 to 8.00 mm	336	420	525		
(d) 8.00 to 13.25 mm	294	364	455		
(e) 13.25 to 24.25 mm	252	315	392	80	210
(f) 24.25 to 38.00 mm	224	280	350		
2. Music wire	392	490	612		
3. Oil tempered wire	336	420	525		
4. Hard-drawn spring wire	280	350	437.5		
5. Stainless-steel wire	280	350	437.5	70	196
6. Monel metal	196	245	306	44	105
7. Phosphor bronze	196	245	306	44	105
8. Brass	140	175	219	35	100

Terms used in Compression Springs

The following terms used in connection with compression springs are important from the subject point of view.

1. Solid length*. When the compression spring is compressed until the coils come in contact with each other, then the spring is said to be solid. The solid length of a spring is the product of total number of coils and the diameter of the wire. Mathematically, Solid length of the spring,*

$$L_S = n'.d$$

where n' = Total number of coils, and d
= Diameter of the wire.

2. Free length*. The free length of a compression spring,, is the length of the spring in the free or unloaded condition. It is equal to the solid length plus the maximum deflection or compression of the spring and the clearance between the adjacent coils (when fully compressed).Mathematically Free length of the spring,*

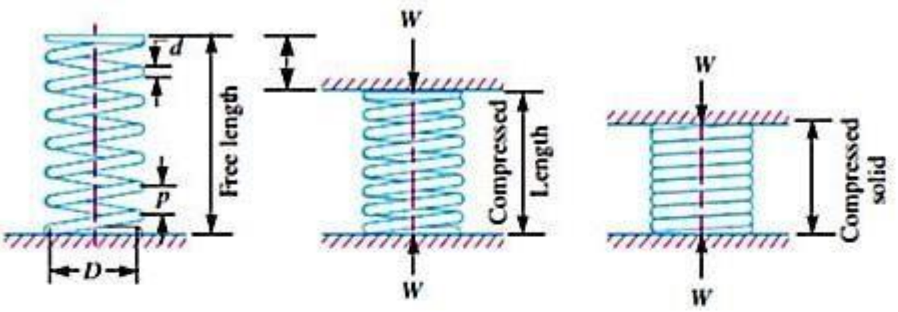
L_F = *Solid length + Maximum compression + *Clearance between adjacent coils (or clash allowance)*

$L_F = n'.d + \delta_{max} + 0.15\ \delta_{max}$

The following relation may also be used to find the free length of the spring, i.e.

$L_F = n'.d + \delta_{max} + (n' - 1) \times 1\ mm$

In this expression, the clearance between the two adjacent coils is taken as (1 mm).



3. Spring index. *The spring index is defined as the ratio of the mean diameter of the coil to the diameter of the wire. Mathematically,*

Spring index, $C = D / d$

where D = Mean diameter of the coil, and d = Diameter of the wire.

4. Spring rate. *The spring rate (or stiffness or spring constant) is defined as the load required per unit deflection of the spring. Mathematically,*

Spring rate, $k = W / \delta$

Where W = Load, and
δ *= Deflection of the spring*

5. Pitch. *The pitch of the coil is defined as the axial distance between adjacent coils in uncompressed state. Mathematically,*

$$p = \frac{\text{Free length}}{n' - 1}$$

The pitch of the coil may also be obtained by using the following relation, i.e. Pitch of the coil

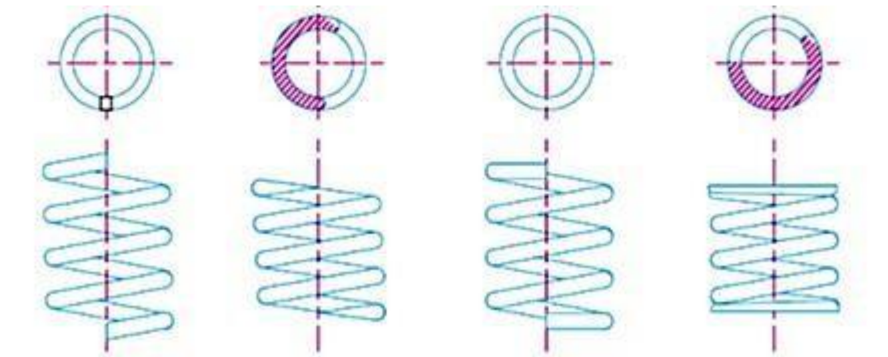
$$p = \frac{L_F - L_S}{n'} + d$$

where L_F *= Free length of the spring,*
L_S *= Solid length of the spring,*

n' = Total number of coils, and
d = Diameter of the wire

End Connections for Compression Helical Springs

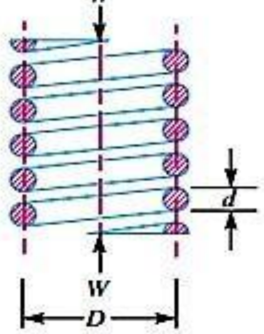
The end connections for compression helical springs are suitably formed in order to apply the load.



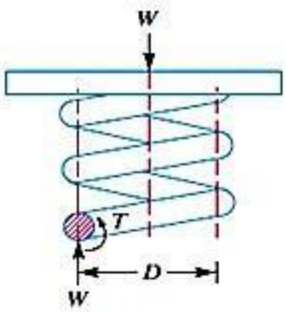
Stresses in Helical Springs of Circular Wire

Consider a helical compression spring made of circular wire and subjected to an axial load W, as.

Let D = Mean diameter of the spring coil,
D_o=outer diameter of spring =D+d
d = Diameter of the spring wire, n
= Number of active coils,
G = Modulus of rigidity for the spring material,
W = Axial load on the spring, η = Maximum
shear stress induced in the wire,
C = Spring index = D/d, p = Pitch of the coils, and δ =
Deflection of the spring, as a result of an axial load W.



(a) Axially loaded helical spring.



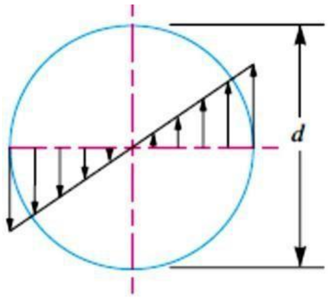
(b) Free body diagram showing that wire is subjected to torsional shear and a direct shear.

A little consideration will show that part of the spring in equilibrium under the action of two forces W and the twisting moment T.

$$T = W \times \frac{D}{2} \qquad T = \frac{\pi}{16} \times \tau_1 \times d^3$$

$$T = W \times \frac{D}{2} = \frac{\pi}{16} \times \tau_1 \times d^3$$

$$\tau_1 = \frac{8W.D}{\pi d^3}$$



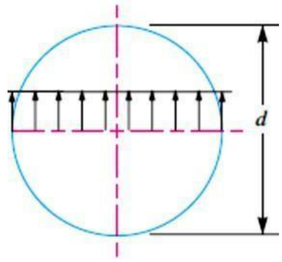
Torsional shear stress diagram.

Where η_1= the torsional shear stress

In addition to the torsional shear stress (η_1) induced in the wire, the following stresses also act on the wire :

1. Direct shear stress due to the load W, and

2. Stress due to curvature of wire.



Direct shear stress diagram.

We know that direct shear stress due to the load W,

$$\tau_2 = \frac{\text{Load}}{\text{Cross-sectional area of the wire}}$$

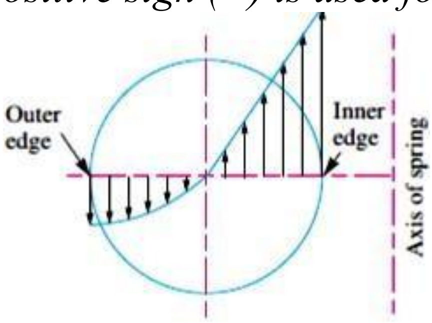
$$= \frac{W}{\frac{\pi}{4} \times d^2} = \frac{4W}{\pi d^2}$$

1. The resultant of torsional shear stress and direct shear stress

η = Torsional shear stress + Direct shear stress

The positive sign (+) is used for the

$$\tau = \tau_1 \pm \tau_2 = \frac{8W.D}{\pi d^3} \pm \frac{4W}{\pi d^2}$$



Resultant torsional shear and direct shear stress diagram.

inner edge of the wire and negative sign(-) is used for the outer edge of the wire. Since the stress is maximum at the inner edge of the wire

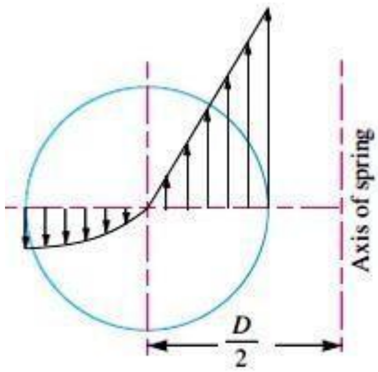
$$\tau_{max} = \frac{8W.D}{\pi d^3} + \frac{4W}{\pi d^2} = \frac{8W.D}{\pi d^3}\left(1 + \frac{d}{2D}\right)$$

$$D/d = C \qquad K_S = \text{Shear stress factor} = 1 + \frac{1}{2C}$$

$$\tau_{max} = \frac{8W.D}{\pi d^3}\left(1 + \frac{1}{2C}\right) = K_S \times \frac{8W.D}{\pi d^3}$$

2.Stress due to curvature of wire

The resultant diagram of torsional shear, direct shear and curvature shear stress



Resultant torsional shear, direct shear and curvature shear stress diagram.

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

K = Wahl's Stress Factor

$$K = \frac{4C - 1}{4C - 4} + \frac{0.615}{C}$$

Deflection of Helical Springs of Circular Wire *l =*

Length of one coil × No. of active coils = π D × n

Let θ = Angular deflection of the wire when acted upon by the torque T. ∴ Axial deflection of the spring,

$$\delta = \theta \times D/2$$

$$\frac{T}{J} = \frac{\tau}{D/2} = \frac{G.\theta}{l}$$

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

J = Polar moment of inertia of the spring wire

$= \frac{\pi}{32} \times d^4$, d being the diameter of spring wire.

G = Modulus of rigidity for the material of the spring wire.

Now substituting the values of l and J in the above equation, we have

$$\theta = \frac{T.l}{J.G} = \frac{\left(W \times \frac{D}{2}\right)\pi D.n}{\frac{\pi}{32} \times d^4 G} = \frac{16W.D^2.n}{G.d^4}$$

Substituting this value of θ in equation

$$\delta = \frac{16W.D^2.n}{G.d^4} \times \frac{D}{2} = \frac{8\,W.D^3.n}{G.d^4} = \frac{8\,W.C^3.n}{G.d} \quad \ldots (\because C = D/d)$$

and the stiffness of the spring or spring rate

$$\frac{W}{\delta} = \frac{G.d^4}{8\,D^3.n} = \frac{G.d}{8\,C^3.n} = \text{constant}$$

Example (1).

A compression coil spring made of an alloy steel is having the following specifications :Mean diameter of coil = 50 mm ; Wire diameter = 5 mm ; Number of active coils = 20. If this spring is subjected to an axial load of 500 N ; calculate the maximum shear stress (neglect the curvature effect) to which the spring material is subjected.

Solution *.Given : D = 50 mm ; d = 5 mm ; n = 20 ; W = 500 N We know that the spring index*

$$C = \frac{D}{d} = \frac{50}{5} = 10$$

∴ Shear stress factor

$$K_S = 1 + \frac{1}{2C} = 1 + \frac{1}{2 \times 10} = 1.05$$

and maximum shear stress (neglecting the effect of wire curvature),

$$\tau = K_S \times \frac{8W.D}{\pi d^3} = 1.05 \times \frac{8 \times 500 \times 50}{\pi \times 5^3} = 534.7 \text{ N/mm}^2$$

$= 534.7$ MPa **Ans.**

Example (2).

Design a helical compression spring for a maximum load of(1000 N)for a deflection of (25 mm) using the value of spring index as(5). The maximum permissible shear stress for spring wire is(420 MPa) and modulus of rigidity is (84 kN/mm^2).

Solution *.Given : $W = 1000$ N ; $\delta = 25$ mm ; $C = D/d = 5$; $\eta = 420$ MPa $= 420$ N/mm^2 ; $G = 84$ kN/mm^2 $= 84 \times 10^3$ N/mm^2*

1. Mean diameter of the spring coil

We know that Wahl's stress factor

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

and maximum shear stress (τ)

$$420 = K \times \frac{8\,W.C}{\pi\,d^2} = 1.31 \times \frac{8\times1000\times5}{\pi\,d^2} = \frac{16\,677}{d^2}$$

$$\therefore \quad d^2 = 16\,677/420 = 39.7 \quad \text{or} \quad d = 6.3 \text{ mm}$$

$$D = c\,d = 5x6.3 = 32 \text{ mm}$$

and outer diameter of the spring coil,

$$Do = D + d = 32.005 + 6.401 = 38.406 \text{ mm Ans.}$$

2. Number of turns of the coils

Let n = Number of active turns of the coils

We know that compression of the spring (δ),

$$\delta = \frac{8W.C^3.n}{G.d} \Rightarrow \quad 25 = \frac{8\times1000\,(5)^3 n}{84\times10^3\times6.3} = 1.86\,n$$

$$\therefore \quad n = 25/1.86 = 13.44 \text{ say } 14 \text{ Ans.}$$

For squared and ground ends, the total number of turns,

$$n' = n + 2 = 14 + 2 = 16 \text{ Ans.}$$

3. ***Free length of the spring***

We know that free length of the spring

$$= n'.d + \delta + 0.15\,\delta = 16\times6.401 + 25 + 0.15\times25$$

$$= 131.2 \text{ mm Ans.}$$

4. ***Pitch of the coil***

We know that pitch of the coil

$$= \frac{\text{Free length}}{n'-1} = \frac{131.2}{16-1} = 8.75 \text{ mm Ans.}$$

Example (3).

A helical spring is made from a wire of (6 mm) diameter and has outside diameter of(75 mm). If the permissible shear stress is (350 MPa) and *modulus of rigidity(84 kN/mm²), find the axial load which the spring can carry and the deflection per active turn.*

Solution *.Given : d = 6 mm ; Do = 75 mm ; η = 350 MPa = 350 N/mm² ;*

$$G = 84\ kN/mm^2 = 84 \times 10^3\ N/mm^2$$

We know that mean diameter of the spring,

$$D = Do - d = 75 - 6 = 69\ mm$$

∴ Spring index, $C = \frac{D}{d} = \frac{69}{6} = 11.5$

Let W = Axial load, and

1. *Neglecting the effect of curvature*

We know that the shear stress factor,

$$K_S = 1 + \frac{1}{2C} = 1 + \frac{1}{2 \times 11.5} = 1.043$$

and maximum shear stress induced in the wire (τ),

$$350 = K_S \times \frac{8\,W.D}{\pi\,d^3} = 1.043 \times \frac{8\,W \times 69}{\pi \times 6^3} = 0.848\,W$$

∴ $W = 350 / 0.848 = 412.7$ N **Ans.**

We know that deflection of the spring,

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

∴ Deflection per active turn,

$$\frac{\delta}{n} = \frac{8\,W.D^3}{G.d^4} = \frac{8 \times 412.7\,(69)^3}{84 \times 10^3 \times 6^4} = 9.96\ \text{mm}$$ **Ans.**

2. *Considering the effect of curvature*

We know that Wahl's stress factor,

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

We also know that the maximum shear stress induced in the wire (τ),

$$350 = K \times \frac{8W.C}{\pi\, d^2} = 1.123 \times \frac{8 \times W \times 11.5}{\pi \times 6^2} = 0.913\, W$$

$$\therefore \quad W = 350 / 0.913 = 383.4 \text{ N} \quad \textbf{Ans.}$$

and deflection of the spring,

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

∴ Deflection per active turn,

$$\frac{\delta}{n} = \frac{8\,W.D^3}{G.d^4} = \frac{8 \times 383.4\,(69)^3}{84 \times 10^3 \times 6^4} = 9.26 \text{ mm} \quad \textbf{Ans.}$$

Home work

1. Design a compression helical spring to carry a load of 500 N with a deflection of 25 mm. The spring index may be taken as 8. Assume the following values for the spring material:

   Permissible shear stress = 350 MPa

   Modulus of rigidity = 84 kN/mm²

   Wahl's factor $= \dfrac{4C-1}{4C-4} + \dfrac{0.615}{C}$, where C = spring index.

   [Ans. d = 5.893 mm ; D = 47.144 mm ; n = 6]

2. A helical valve spring is to be designed for an operating load range of approximately 90 to 135 N. The deflection of the spring for the load range is 7.5 mm. Assume a spring index of 10. Permissible shear stress for the material of the spring = 480 MPa and its modulus of rigidity = 80 kN/mm². Design the spring.

   Take Wahl's factor $= \dfrac{4C-1}{4C-4} + \dfrac{0.615}{C}$, C being the spring index.

   [Ans. d = 2.74 mm ; D = 27.4 mm ; n = 6]

Belt drive

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 Belt Drives

The belt drives are usually classified into the following three groups:

1. Light drives. *These are used to transmit small powers at belt speeds up to about (10 m/s)as in agricultural machines and small machine tools.*

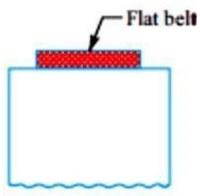
2. Medium drives. *These are used to transmit medium powers at belt speeds over 10 m/s but up to (22 m/s), as in machine tools.*

3. Heavy drives. *These are used to transmit large powers at belt speeds above (22 m/s) as in compressors and generators.*

Types of Belts

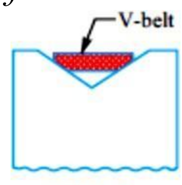
Though there are many types of belts used the following:

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



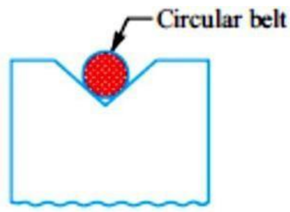
(a) Flat belt.

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.

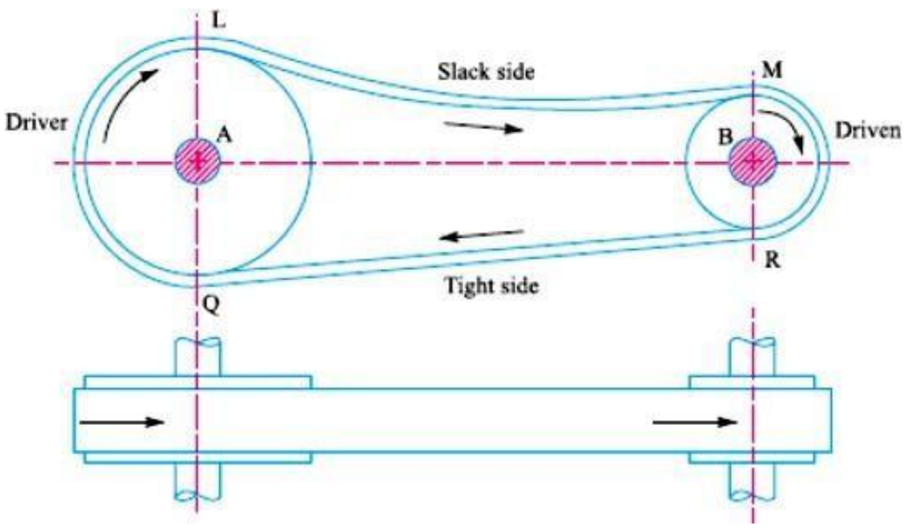
Material used for Belts

1. Leather belts. The most important material for flat belt is leather.
2. Cotton or fabric belts. Most of the fabric belts are made by folding canvass or cotton duck to three or more layers (depending upon the thickness desired) and stitching together
3. Rubber belt. The rubber belts are made of layers of fabric impregnated with rubber composition and have a thin layer of rubber on the faces
4. Balata belts. These belts are similar to rubber belts except that balata gum is used in place of rubber.

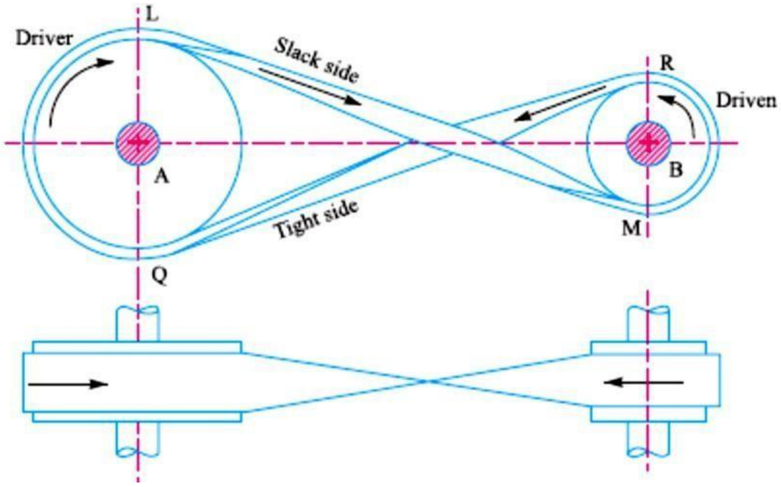
Types of Flat Belt Drives

The power from one pulley to another may be transmitted by any of the following types of belt

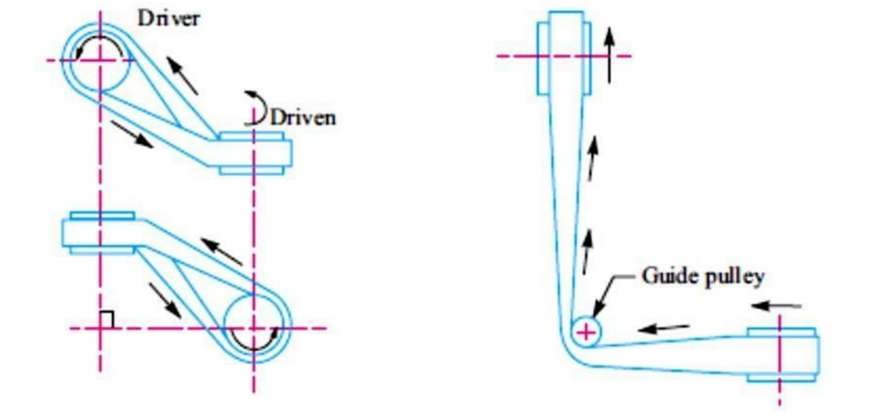
1. Open belt drive. The open belt drive, as shown in Fig., is used with shafts arranged parallel and rotating in the same direction



2. *Crossed or twist belt drive. The crossed or twist belt drive, as shown in Fig., is used with shafts arranged parallel and rotating in the opposite directions*



3. *Quarter turn belt drive. The quarter turn belt drive (also known as right angle belt drive) as shown in Fig. (a), is used with shafts arranged at right angles and rotating in one definite direction. or when the reversible motion is desired, then a quarter turn belt drive with a guide pulley, as shown in Fig. (b), may be used*



(*a*) Quarter turn belt drive. (*b*) Quarter turn belt drive with guide pulley.

4. *Belt drive with idler pulleys. A belt drive with an idler pulley (also known as jockey pulley drive) as shown in Fig. , is used with shafts arranged parallel*

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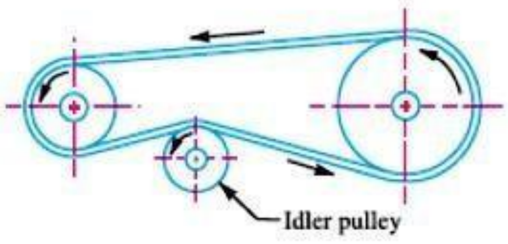
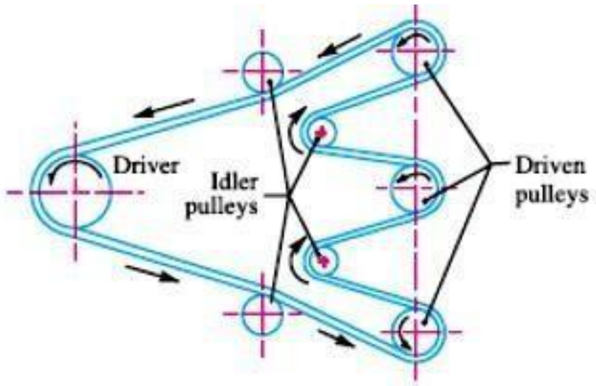
and when an open belt drive can not be used due to small angle of contact on the



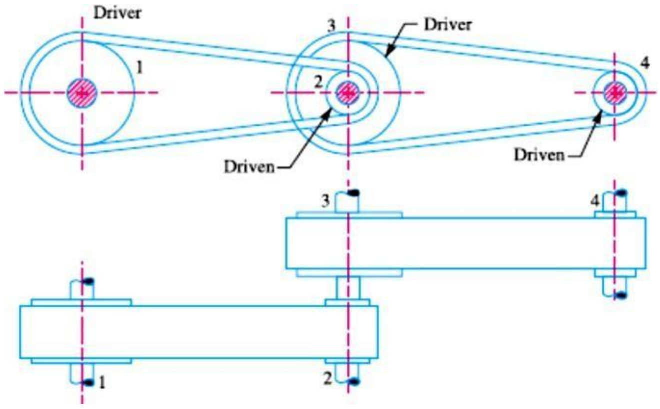
Belt drive with single idler pulley



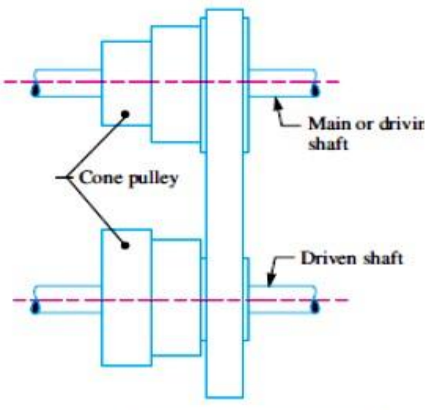
Belt drive with many idler pulleys.

smaller pulley

5. *Compound belt drive. A compound belt drive as shown in Fig , is used when power is transmitted from one shaft to another through a number of pulleys.*



6. *Stepped or cone pulley drive. A stepped or cone pulley drive, as shown in Fig., is used for changing the speed of the driven shaft while the main or driving shaft runs at constant speed. This is accomplished by shifting the belt from one part of the steps to the other*



Stepped or cone pulley drive.

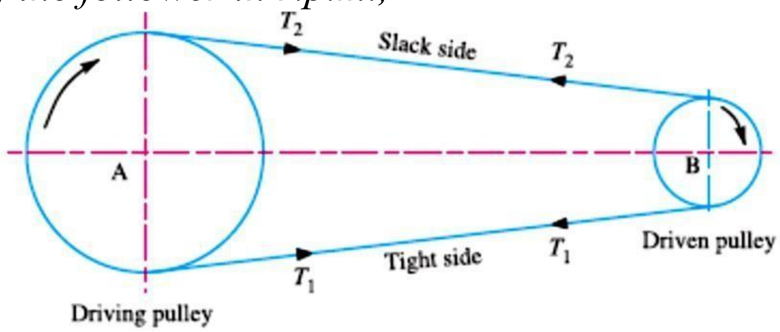
Design of flat belt drive

1. Velocity Ratio of a 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.,

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



d_1, N_1, S_1 d_2, N_2, S_2

∴ *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

$$\therefore \pi d_1 N_1 = \pi d_2 N_2$$

and velocity ratio, $\frac{N_2}{N_1} = \frac{d_1}{d_2}$

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

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

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

and peripheral velocity of the belt on the driven pulley,

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

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

$\pi d_1 N_1 \quad \pi d_2 N_2 \quad N_2 \quad d_1$

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

In case of a compound belt drive

$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4} \quad \text{or} \quad \frac{\text{Speed of last driven}}{\text{Speed of first driver}} = \frac{\text{Product of diameters of drivers}}{\text{Product of diameters of drivens}}$$

2.Slip of the Belt (s)

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}\left(1 - \frac{s}{100}\right)$$

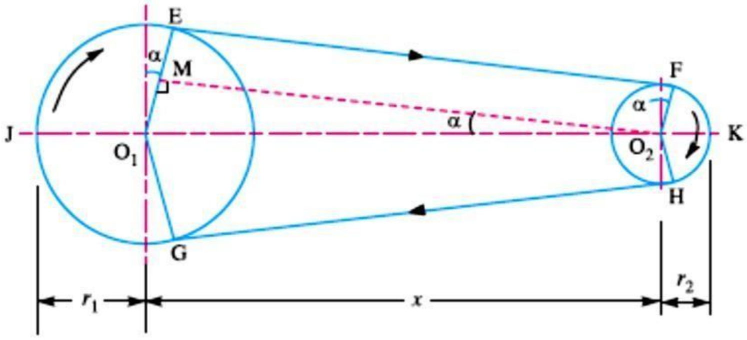
$S = s_1 + s_2$

s_1 *% = Slip between the driver and the belt, and*

s_2 *% = Slip between the belt and follower*

t= thickness of belt

3. Length of an Open Belt Drive (Lo)

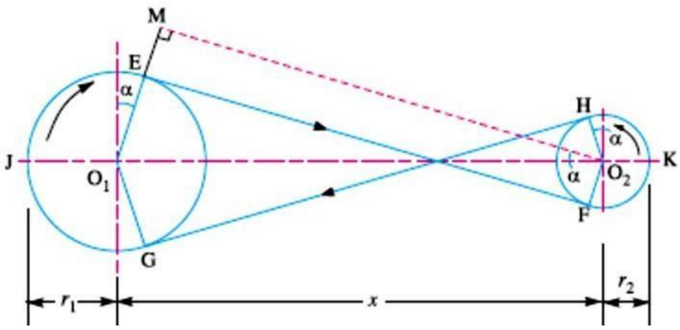


x = Distance between of two centres Lo = Total length of the open belt

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

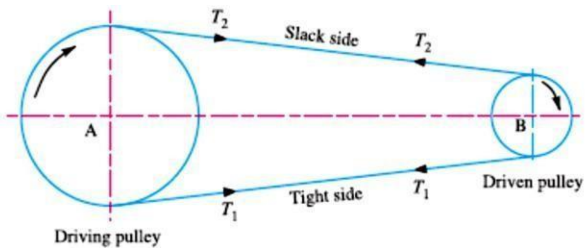
4.Length of an cross Belt Drive (Lc)

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



5.Power Transmitted by a Belt

T_1 and T_2 = Tensions in the tight side and slack side of the belt respectively in Newtons ,r_1 and r_2 = Radii of the driving and driven pulleys respectively in metres ,and v = Velocity of the belt in m/s.



The power transmitted

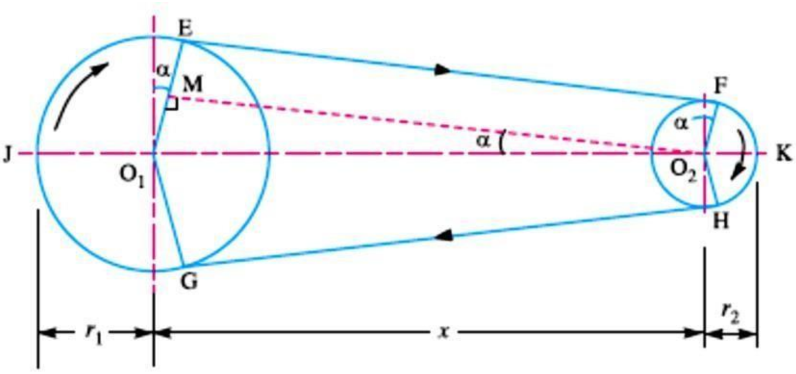
$P = (T_1 - T_2)\, v$ W ... (∵1 N-m/s = 1W)
Torque exerted on the driving pulley is $(T_1 - T_2)\, r_1$.
Torque exerted on the driven pulley is $(T_1 - T_2)\, r_2$.

6. Ratio of Driving Tensions for Flat Belt Drive

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

μ =coefficient of friction α = the angle of contact at the smaller pulley MO_1O_2

,



$$\frac{T_1}{T_2} = e^{\mu.\theta}$$

∴ Angle of contact or lap,

$$\theta = (180° - 2\alpha)\frac{\pi}{180} \text{ rad} \qquad \text{...(for open belt drive)}$$

$$= (180° + 2\alpha)\frac{\pi}{180} \text{ rad} \qquad \text{... (for cross-belt drive)}$$

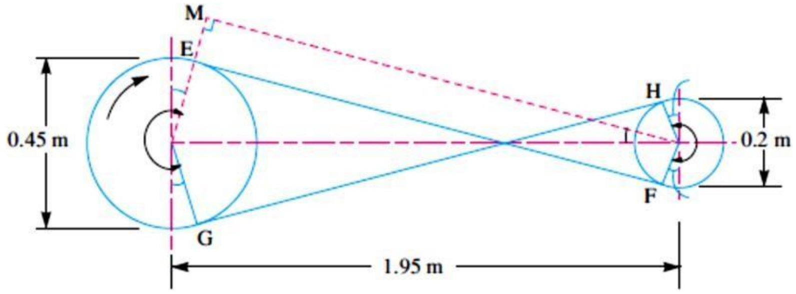
x

$$= \frac{r_1 + r_2}{x} \qquad \text{... (for cross-belt drive)}$$

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

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

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

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

Angle of contact between the belt and each pulley

Let θ = Angle of contact between the belt and each pulley.

We know that for a crossed belt drive,

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

$$\alpha = 9.6°$$

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

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

Power transmitted

$$\frac{T_1}{T_2} = e^{\mu\theta} \Rightarrow \frac{1000}{T_2} = e^{0.25\times3.47} \Rightarrow T_2 = \frac{1000}{2.387} = 419\ N$$

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,

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

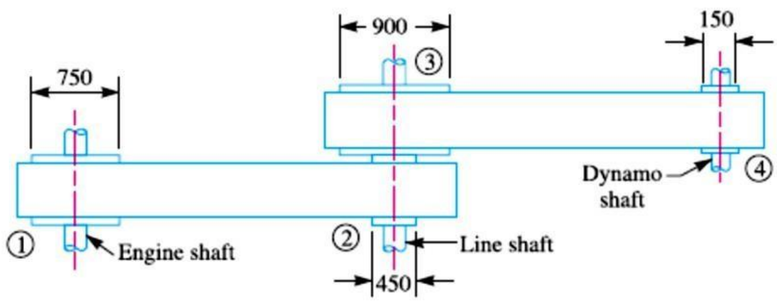
Example (2)

An engine running at (150 r.p.m).drives a line shaft by means of a belt. The engine pulley is(750 mm) diameter and the pulley on the line shaft is(450 mm). A (900 mm) diameter pulley on the line shaft drives a(150 mm) diameter pulley keyed to a dynamo shaft. Fine the speed of dynamo shaft, when

1. there is no slip, and

2. there is a slip of 2% at each drive.

Solution *.Given : N_1 = 150 r.p.m. ; d_1 = 750 mm ; d_2 = 450 mm ; d_3 = 900 mm d_4 = 150 mm ; $s_1 = s_2$ = 2%*



1.When there is no slip

$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4} \text{ or } \frac{N_4}{150} = \frac{750 \times 900}{450 \times 150} = 10$$

$$N_4 = 150 \times 10 = 1500 \text{ r.p.m. } \textbf{Ans.}$$

2. When there is a slip of 2% at each drive

$$S = s_1 + s_2$$

$$S = 2\% + 2\% = 4\%$$

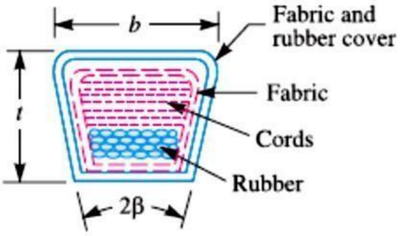
$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4}\left(1 - \frac{S}{100}\right)$$

$$\frac{N_4}{150} = \frac{750 \times 900}{450 \times 150}\left(1 - \frac{4}{100}\right) = 9.6$$

$$N_4 = 150 \times 9.6 = 1440 \text{ r.p.m. } \textbf{Ans.}$$

V-belt

belts are molded to a trapezoidal shape and are made endless V-belt is mostly used in 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



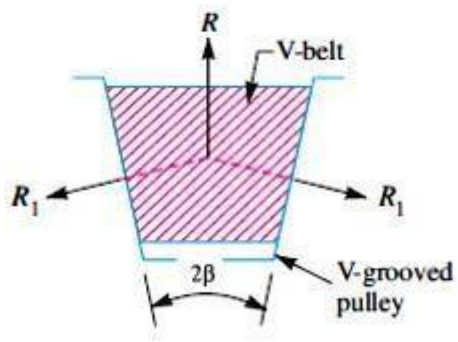
Advantages and Disadvantages of V-belt Drive over Flat Belt Drive

Following are the advantages and disadvantages of the V-belt drive over flat belt drive :Advantages

1. *The V-belt drive gives compactness due to the small distance between centres of pulleys.*
2. *The drive is positive, because the slip between the belt and the pulley groove is negligible.*
3. *Since the V-belts are made endless and there is no joint trouble, therefore the drive is smooth.*
4. *It provides longer life, 3 to 5 years.*
5. *It can be easily installed and removed.*
6. *The operation of the belt and pulley is quiet.*
7. *The belts have the ability to cushion the shock when machines are started.*
8. *The high velocity ratio (maximum 10) may be obtained.*

Ratio of Driving Tensions for V-belt

A V-belt with a grooved pulley is shown in Fig.



Let R_1 = Normal reactions between belts and sides of the groove.

R = Total reaction in the plane of the groove .

μ = Coefficient of friction between the belt and sides of the groove .

Resolving the reactions vertically to the groove, we have $R = R_1 \sin\beta + R_1 \sin\beta = 2R_1 \sin\beta$

or $$R_1 = \frac{R}{2\sin\beta}$$

We know that the frictional force

$$= 2\,\mu.R_1 = 2\,\mu \times \frac{R}{2\sin\beta} = \frac{\mu.R}{\sin\beta} = \mu.R.\text{cosec}\,\beta$$

- *The relation between T_1 and T_2 for the V-belt drive will be*

$$\frac{T_1}{T_2} = e^{\mu\theta \times \text{cosec}\,\beta}$$

- *Centrifugal tension, $T_C = m.v^2$*
- *The mass of the belt per metre length, m = Area × length × density*
  $m = A \times \rho$
- *Maximum tension in the belt, $T = \sigma \times A$*
- *Tension in the tight side of the belt, $T_1 = T - T_C$*

$$\text{Number of } V\text{-belts} = \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}}$$

Example (1)

A compressor, requiring(90 kW), is to run at about(250 r.p.m). The drive is by V-belts from an electric motor running at(750 r.p.m). The diameter of the pulley on the compressor shaft must not be greater than (1 metre) while the centre distance between the pulleys is limited to (1.75metre). The belt speed should not exceed (1600 m / min).Determine the number of V-belts required to transmit the power if each belt has a cross sectional area of(375 mm^2), density (1000 kg / m^3) and an allowable tensile stress of (2.5 MPa). The groove angle of the pulleys is (35°). The coefficient of friction between the belt and the pulley is(0.25).Calculate also the length required of each belt.

***Solution**. Given : $P = 90$ kW $= 90 \times 10^3$ W ; $N_2 = 250$ r.p.m. ; $N_1 = 750$ r.p.m. ; $d_2 = 1$ m ;$x = 1.75$ m ; $v = 1600$ m/min $= 26.67$ m/s ;$A = 375$ mm^2 = 375 ×*

$10^{-6}\, m^2$; ρ = 1000 kg / m^3 ; ζ =

$\beta = 17.5°$; $\mu = 0.25$

First of all, let us find the diameter of pulley on the motor shaft (d_1). We know that

$$\frac{N_1}{N_2} = \frac{d_1}{d_2} \quad \text{or} \quad d_1 = \frac{d_2 N_2}{N_1} = \frac{1 \times 250}{750} = 0.33 \text{ m}$$

$$\sin\alpha = \frac{r_2 - r_1}{x} = \frac{d_2 - d_1}{2x} = \frac{1 - 0.33}{2 \times 1.75} = 0.1914$$

$$\alpha = \sin^{-1} 0.1914 = 11.04°$$

$$\theta = 180° - 2\alpha = 180 - 2 \times 11.04 = 157.92°$$

$$= 157.92 \times \frac{\pi}{180} = 2.76 \text{ rad}$$

.

2.5MPa = 2.5 N/mm^2 ; $2\beta = 35°$ or

We know that mass of the belt per metre length,

$m = Area \times length \times density = 375 \times 10^{-6} \times 1 \times 1000 = 0.375\ kg / m$

∴ *Centrifugal tension,*

$T_C = m.v^2 = 0.375\,(26.67)^2 = 267\ N$

and maximum tension in the belt,

$T = \sigma \times a = 2.5 \times 375 = 937.5\ N$

∴ *Tension in the tight side of the belt,*

$T_1 = T - T_C = 937.5 - 267 = 670.5\ N$

Let T_2 = *Tension in the slack side of the belt*

$$\frac{T_1}{T_2} = e^{\mu\theta \times cosec\ \beta}$$

$$\frac{670}{T_2} = e^{0,25 \times 2.76 \times cosec\ 17.5}$$

$$\frac{670}{T_2} = 9.95$$

$$T_2 = \frac{670}{9.95} = 67.4\ N$$

Number of V-belts

We know that the power transmitted per belt,

$= (T_1 - T_2)\,v = (670.5 - 67.4)\,26.67 = 16\,085\ W = 16.085\ kW$

$$Number\ of\ V\text{-}belts = \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}} = \frac{90}{16.085} = 5.6 \text{ say } 6 \text{ Ans.}$$

Ho me work

1. *An engine shaft running at(120 r.p.m.) is required to drive a machine shaft by means of a belt. The pulley on the engine shaft is of (2 m)diameter and that of the machine shaft is(1 m)diameter. If the belt thickness is(5 mm);*

determine the speed of the machine shaft, when 1. there is no slip; and 2. there is a slip of (3%.)

[Ans. 239.4 r.p.m. ; 232.3 r.p.m.]

2. *A pulley is driven by a flat belt running at a speed of (600 m/min). The coefficient of friction between the pulley and the belt is (0.3) and the angle of*
   lap is (160°). If the maximum tension in the belt is(700 N);find the power transmitted by a belt. *[Ans. 3.974 kW]*
3. *Find the width of the belt necessary to transmit(10 kW)to a pulley (300 mm) diameter, if the pulley makes(1600 r.p.m). and the coefficient of friction between the belt and the pulley is(0.22). Assume the angle of contact as(210°) and the maximum tension in the belt is not to exceed (8N/mm) width.* *[Ans. 90 mm]*
4. *An open belt (100 mm)wide connects two pulleys mounted on parallel shafts with their centres(2.4 m) apart. The diameter of the larger pulley is(450 mm) and that of the smaller pulley (300 mm). The coefficient of friction between the belt and the pulley is(0.3)and the maximum stress in the belt is limited to (14 N/mm)width. If the larger pulley rotates at (120 r.p.m)., find the maximum power that can be transmitted.*

[Ans. 2.387 Kw]

shaft

A shaft is a rotating machine element which is used to transmit power from one place to another. The power is delivered to the shaft by some tangential force and the resultant torque (or twisting moment) set up within *the shaft permits the power to be transferred to various machines linked up to the shaft. In order to transfer the power from one shaft to another*

Material Used for Shafts

The material used for shafts should have the following properties :

1. It should have high strength.

2. It should have good machinability.

3. It should have low notch sensitivity factor.

4. It should have good heat treatment properties.

5. It should have high wear resistant properties.

The material used for ordinary shafts is carbon steel of grades 40 C 8, 45 C 8, 50 C 4 and 50 C 12.

Types of Shafts

The following two types of shafts are important from the subject point of view :

1. *Transmission shafts. These shafts transmit power between the source and the machines absorbing power. The counter shafts, line shafts, over head shafts and all factory shafts are transmission shafts. Since these shafts carry machine parts such as pulleys, gears etc., therefore they are subjected to bending in addition to twisting.*
2. *Machine shafts. These shafts form an integral part of the machine itself. The crank shaft is an example of machine shaft*

Stresses in Shafts

The following stresses are induced in the shafts :

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

Design of Shafts

The shafts may be designed on the basis of

1. Strength, and 2.Rigidity and stiffness.

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

(1) Shafts subjected to twisting moment or torque only,

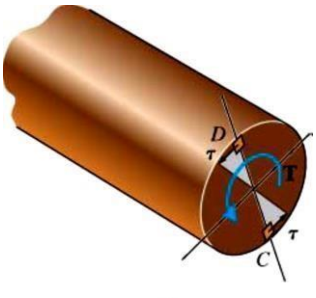
(2) Shafts subjected to bending moment only,

(3) Shafts subjected to combined twisting and bending moments

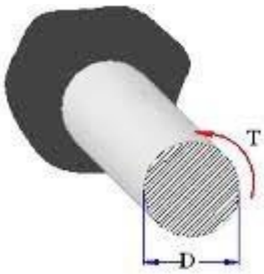
1. Shafts Subjected to Twisting Moment Only

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

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



(f)



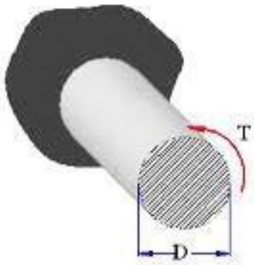
T = Twisting moment (or torque) acting upon the shaft,

J = Polar moment of inertia of the shaft about the axis of rotation, η = Torsional shear stress, and r = Distance from neutral axis to the outer most fiber= d / 2; where d is the diameter of the shaft.

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

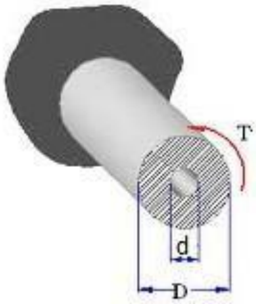
$$J = \frac{\pi}{32} \times d^4$$

$$\frac{T}{\frac{\pi}{32} \times d^4} = \frac{\tau}{\frac{d}{2}} \quad \text{or} \quad T = \frac{\pi}{16} \times \tau \times d^3$$



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

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



do and di = Outside and inside diameter of the shaft, and r = do / 2. Substituting these values in equation, we have

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

k = Ratio of inside diameter and outside diameter of the shaft = di / do

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

power transmitted (P) by the shaft,

T = Twisting moment in N-m, and

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

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

Example*(1).*

A line shaft rotating at(200r.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 × 10^3 W;*

η = 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),

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

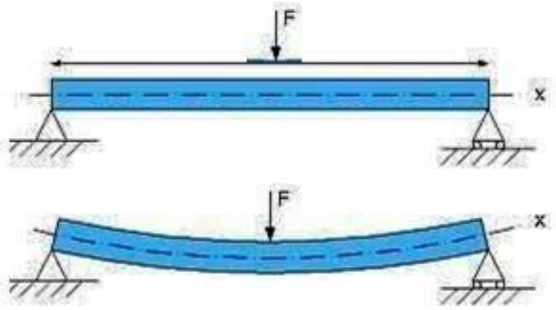
$$d^3 = 955 \times 10^3 / 8.25$$

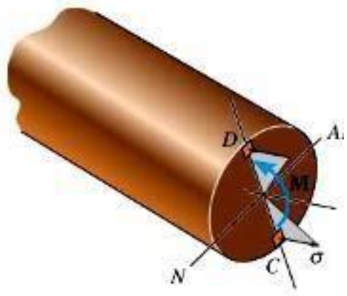
$$d^3 = 115\ 733$$

$$d = \sqrt[3]{115733} = 48.7\ mm$$

2. Shafts Subjected to Bending Moment Only

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





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

M = Bending moment,

I = Moment of inertia of cross-sectional area of the shaft about the axis of rotation

ζ_b = Bending stress, and y = Distance from neutral axis to the outer-most fiber.

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

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

Substituting these values in equation

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

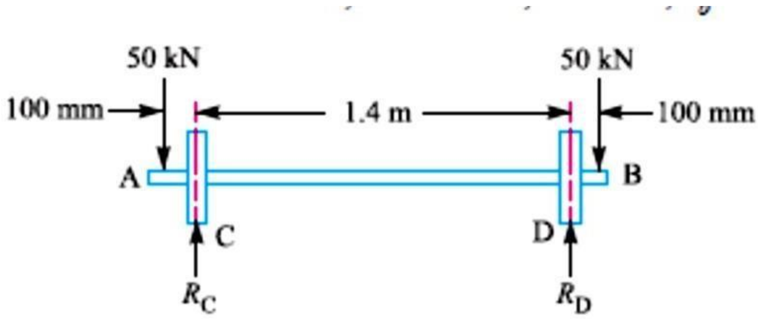
From this equation, diameter of the solid shaft (d) may be obtained .We also know that for a hollow shaft, moment of inertia

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

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

Example (1)

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 × 10^3 N ; L = 100 mm ; x = 1.4 m ; ζ_b = 100 MPa = 100 N/mm^2*

The axle with wheels is shown in Fig. .

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

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

Let d = Diameter of the axle.

We know that the maximum bending moment (M),

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

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

3. Shafts Subjected to Combined Twisting Moment and Bending Moment

When the shaft is subjected to combined twisting moment and bending moment, then the shaft must be designed on the basis of the two moments *simultaneously. The following two theories are important from the subject point of view :*

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

Let η = Shear stress induced due to twisting moment, and ζ$_b$ = Bending stress (tensile or compressive) induced due to bending moment. According to maximum shear stress theory, the maximum shear

stress in the shaft

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

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

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

The expression $\sqrt{M^2 + T^2}$ is known as ***equivalent twisting moment*** is denoted by T_e.

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times \tau \times d^3 \quad \text{for round solid shaft.}$$

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times \tau\,(d_o)^3\,(1 - k^4) \quad \text{In case of a hollow shaft.}$$

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

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

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

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

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

The expression $\frac{1}{2}\left[(M + \sqrt{M^2 + T^2})\right]$ is known as ***equivalent bending moment*** and is denoted by M_e.

$$M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = \frac{\pi}{32} \times \sigma_b \times d^3 \qquad \text{for round solid shaft.}$$

$$M_e = \frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right) = \frac{\pi}{32} \times \sigma_b\,(d_o)^3\,(1 - k^4) \qquad \text{In case of a hollow shaft.}$$

Example (1).

A solid circular shaft is subjected to a bending moment of (3000 N-m) and a torque of (10 000 N-m). The shaft is made of 45 C 8 steel having ultimate tensile stress of (700 MPa) and a ultimate shear stress of (500 MPa). Assuming a factor of safety as (6), determine the diameter of the shaft.

Solution *.Given : M = 3000 N-m = 3 × 10^6 N-mm ;*

T = 10 000 N-m = 10 × 10^6 N-mm ;ζt_u = 700 MPa = 700 N/mm^2 ; η_u = 500 MPa = 500 N/mm^2

We know that the allowable tensile stress

$$\sigma_t \text{ or } \sigma_b = \frac{\sigma_{tu}}{F.S.} = \frac{700}{6} = 116.7 \text{ N/mm}^2$$

and allowable shear stress,

$$\tau = \frac{\sigma_u}{F.S.} = \frac{500}{6} = 83.3 \text{ N/mm}^2$$

d = Diameter of the shaft in mm.

According to maximum shear stress theory, equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(3 \times 10^6)^2 + (10 \times 10^6)^2} = 10.44 \times 10^6 \text{ N-mm}$$

We also know that equivalent twisting moment (Te),

$$10.44 \times 10^6 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 83.3 \times d^3 = 16.36\, d^3$$

$$d^3 = 10.44 \times 10^6 / 16.36 = 0.636 \times 10^6 \text{ or } d = 86 \text{ mm}$$

According to maximum normal stress theory, equivalent bending moment,

$$M_e = \tfrac{1}{2}\left(M + \sqrt{M^2 + T^2}\right) = \tfrac{1}{2}(M + T_e)$$

$$= \tfrac{1}{2}(3 \times 10^6 + 10.44 \times 10^6) = 6.72 \times 10^6 \text{ N-mm}$$

We also know that the equivalent bending moment (M_e),

$$6.72 \times 10^6 = \frac{\pi}{32} \times \sigma_b \times d^3 = \frac{\pi}{32} \times 116.7 \times d^3 = 11.46\, d^3$$

$$\therefore \quad d^3 = 6.72 \times 10^6 / 11.46 = 0.586 \times 10^6 \text{ or } d = 83.7 \text{ mm}$$

Taking the larger of the two values, we have

d = 86 say 90 mm **Ans.**

Home work

1.A shaft running at(400 r.p.m). transmits (10 kW). Assuming allowable shear stress in shaft as (40 MPa), find the diameter of the shaft.

[Ans. 35 mm]

2.A hollow steel shaft transmits (600 kW) at (500 r.p.m). The maximum shear stress is (62.4 MPa). Find the outside and inside diameter of the shaft, if the outer diameter is twice of inside diameter, assuming that the maximum torque is(20%) greater than the mean torque. [Ans. 100 mm ; 50 mm]

3.A hollow shaft for a rotary compressor is to be designed to transmit a maximum torque of (4750 N-m)The shear stress in the shaft is limited to (50 MPa). Determine the inside and outside diameters of the shaft, if the ratio of the inside to the outside diameter is (0.4). [Ans. 35 mm ; 90 mm]

4.A motor car shaft consists of a steel tube (30 mm) internal diameter and (4mm)thick. The engine develops (10 kW) at (2000 r.p.m). Find the maximum shear stress in the tube when the power is transmitted through a 4 : 1 gearing. [Ans. 30 MPa]

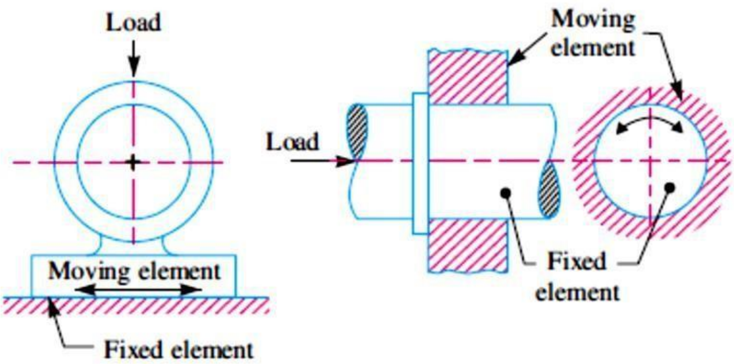
The bearing

A bearing is a machine element which support another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load

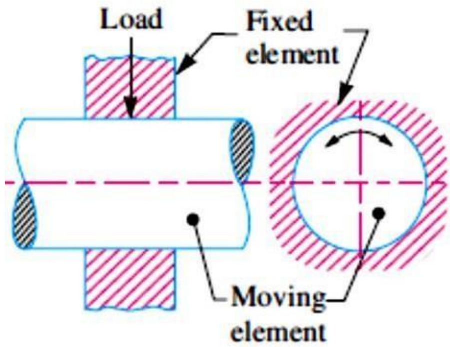
Classification of Bearings

1. Depending upon the direction of load to be supported. The bearings under this group are classified as:

(a) Radial bearings, the load acts perpendicular to the direction of motion of the moving element

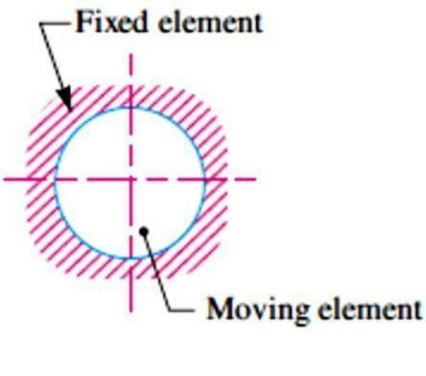


(b) Thrust bearings .In thrust bearings, the load acts along the axis of rotation

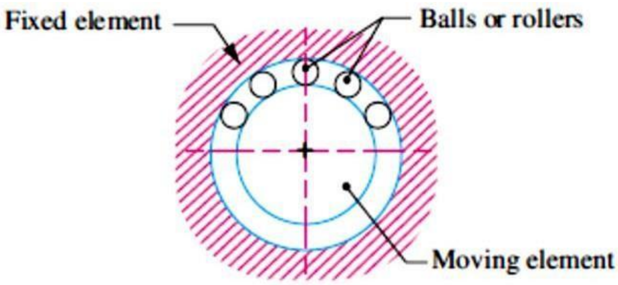


2. *Depending upon the nature of contact. The bearings under this group are classified as :*

(a) Sliding contact bearings ,the sliding takes place along the surfaces of contact between the moving element and the fixed element. The sliding contact bearings are also known as plain bearings

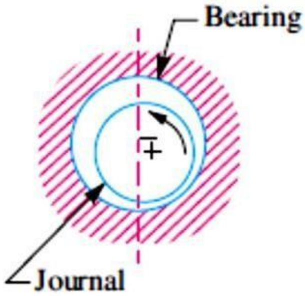


(b) Rolling contact bearings .the steel balls or rollers, are interposed between the moving and fixed elements. The balls offer rolling friction at two points for each ball or roller



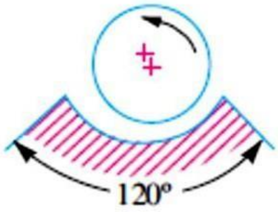
Types of Sliding Contact Bearings

1. ***full journal bearing*** *:The sliding contact bearings in which the sliding action is along the circumference of a circle or an arc of a circle and carrying radial loads are known as journal or sleeve bearings. When the angle of contact of the bearing with the journal is 360° as shown in Fig.(a), then the bearing is called a full journal bearing. This type of bearing is commonly used in industrial machinery to accommodate bearing loads in any radial direction*



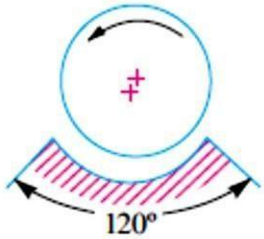
(*a*) Full journal bearing.

2. ***partial journal bearing****: When the angle of contact of the bearing with the journal is 120°, as shown in Fig (b), then the bearing is said to be partial journal bearing. This type of bearing has less friction than full journal bearing, but it can be used only where the load is always in one direction. The most common application of the partial journal bearings is found in rail road car axles. The full and partial journal bearings may be called as clearance bearings because the diameter of the journal is less than that of bearing*



(*b*) Partial journal bearing.

3. ***fitted bearing****: When a partial journal bearing has no clearance i.e. the diameters of the journal and bearing are equal, then the bearing is called a fitted bearing, as shown in Fig. (c).*



(*c*) Fitted journal bearing.

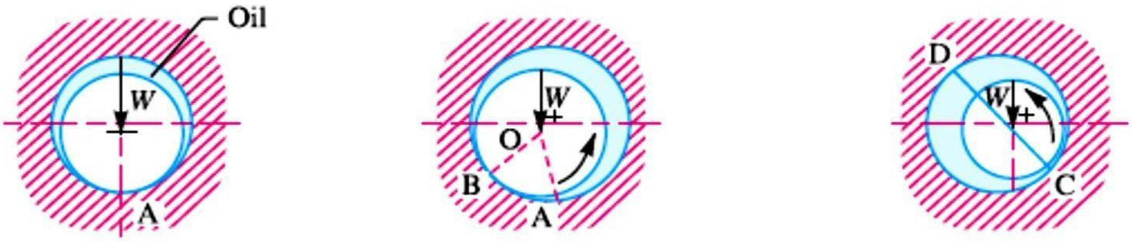
The sliding contact bearings, according to the thickness of layer of the lubricant between the bearing and the journal, may also be classified as follows :

1. ***Thick film bearings****. The thick film bearings are those in which the working surfaces are completely separated from each other by the lubricant. Such type of bearings are also called as hydrodynamic lubricated bearings.*
2. ***Thin film bearings****. The thin film bearings are those in which, although lubricant is present ,the working surfaces partially contact each other at least part of the time. Such type of bearings are also called boundary lubricated bearings.*
3. ***Zero film bearings****. The zero film bearings are those which operate without any lubricant present.*
4. ***Hydrostatic or externally pressurized lubricated bearings****. The hydrostatic bearings are those which can support steady loads without any relative*

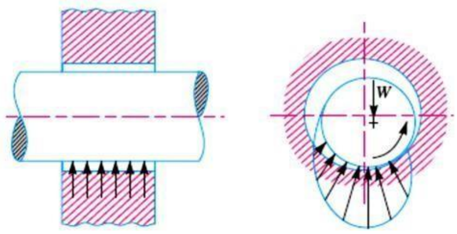
motion between the journal and the bearing .This is achieved by forcing externally pressurized lubricant between the members.

Wedge Film Journal Bearings

. Fig. (a) shows a journal at rest with metal to metal contact at A on the line of action of the supported load. When the journal rotates slowly in the anticlockwise direction, as shown in Fig (b),When the speed of the journal is increased, a continuous fluid film is established as in Fig(c)



(a) At rest. (b) Slow speed. (c) High speed.



Bearing Characteristic Number and Bearing Modulus for Journal Bearings

The coefficient of friction in design of bearings is of great importance, because it affords a means for determining the loss of power due to bearing friction. It has been shown by experiments that the coefficient of friction for a full lubricated journal bearing is a function of three variables, i.e.

$$(i)\ \frac{ZN}{p};\qquad (ii)\ \frac{d}{c};\qquad \text{and}\qquad (iii)\ \frac{l}{d}$$

$$\mu = \phi\left(\frac{ZN}{p}, \frac{d}{c}, \frac{l}{d}\right)$$

μ = Coefficient of friction, θ
= A functional relationship,
Z = Absolute viscosity of the lubricant, in kg / m-s,
N = Speed of the journal in r.p.m.,
p = Bearing pressure on the projected bearing area in N/mm^2,
= Load on the journal = l × d
d = Diameter of the journal, l
= Length of the bearing, and c
= Diametral clearance.
The factor ZN / p is termed as bearing characteristic number and is a dimensionless number

Coefficient of Friction for Journal Bearings

Coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k$$ ... (when Z is in kg / m-s and p is in N / mm^2)

k = Factor to correct for end leakage. It depends upon the ratio of length to the diameter of the bearing (i.e. l / d).
= 0.002 for l / d ratios of 0.75 to 2.8

Table 26.3. Design values for journal bearings.

Machinery	Bearing	Maximum bearing pressure (p) in N/mm^2	Operating values			
			Absolute Viscosity (Z) in kg/m-s	ZN/p Z in kg/m-s p in N/mm^2	$\frac{c}{d}$	$\frac{l}{d}$
Automobile and air-craft engines	Main	5.6 – 12	0.007	2.1	—	0.8 – 1.8
	Crank pin	10.5 – 24.5	0.008	1.4		0.7 – 1.4
	Wrist pin	16 – 35	0.008	1.12		1.5 – 2.2
Four stroke-Gas and oil engines	Main	5 – 8.5	0.02	2.8	0.001	0.6 – 2
	Crank pin	9.8 – 12.6	0.04	1.4		0.6 – 1.5
	Wrist pin	12.6 – 15.4	0.065	0.7		1.5 – 2
Two sroke-Gas and oil engines	Main	3.5 – 5.6	0.02	3.5	0.001	0.6 – 2
	Crank pin	7 – 10.5	0.04	1.8		0.6 – 1.5
	Wrist pin	8.4 – 12.6	0.065	1.4		1.5 – 2
Marine steam engines	Main	3.5	0.03	2.8	0.001	0.7 – 1.5
	Crank pin	4.2	0.04	2.1		0.7 – 1.2
	Wrist pin	10.5	0.05	1.4		1.2 – 1.7
Stationary, slow speed steam engines	Main	2.8	0.06	2.8	0.001	1 – 2
	Crank pin	10.5	0.08	0.84		0.9 – 1.3
	Wrist pin	12.6	0.06	0.7		1.2 – 1.5
Stationary, high speed steam engine	Main	1.75	0.015	3.5	0.001	1.5 – 3
	Crank pin	4.2	0.030	0.84		0.9 – 1.5
	Wrist pin	12.6	0.025	0.7		13 – 1.7
Reciprocating pumps and compressors	Main	1.75	0.03	4.2	0.001	1 – 2.2
	Crank pin	4.2	0.05	2.8		0.9 – 1.7
	Wrist pin	7.0	0.08	1.4		1.5 – 2.0
Steam locomotives	Driving axle	3.85	0.10	4.2	0.001	1.6 – 1.8
	Crank pin	14	0.04	0.7		0.7 – 1.1
	Wrist pin	28	0.03	0.7		0.8 – 1.3

Critical Pressure of the Journal Bearing

The pressure at which the oil film breaks down so that metal to metal contact begins, is known as critical pressure or the minimum operating pressure of the bearing. It may be obtained by the following empirical relation, i.e .Critical pressure or minimum operating pressure,

$$p = \frac{ZN}{4.75 \times 10^6}\left(\frac{d}{c}\right)^2\left(\frac{l}{d+l}\right) \text{N/mm}^2 \qquad \text{...(when Z is in kg / m-s)}$$

Sommerfeld Number

The Sommerfeld number is also a dimensionless parameter used extensively in the design of journal bearings. Mathematically

$$\text{Sommerfeld number} = \frac{ZN}{p}\left(\frac{d}{c}\right)^2$$

For design purposes, its value is taken as follows :

$$\frac{ZN}{p}\left(\frac{d}{c}\right)^2 = 14.3 \times 10^6 \qquad \text{... (when } Z \text{ is in kg / m-s and } p \text{ is in N / mm}^2\text{)}$$

Heat Generated in a Journal Bearing

The heat generated in a bearing is due to the fluid friction and friction of the parts having relative motion. Mathematically, heat generated in a bearing,

$Qg = \mu.W.V$ N-m/s or J/s or watts

μ = Coefficient of friction,

W = Load on the bearing in N,

= Pressure on the bearing in N/mm^2 × Projected area of the bearing in mm^2 = p (l × d),

V = Rubbing velocity in $\text{m/s} = \frac{\pi d.N}{60}$, *d* is in metres, :

, and

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

$$Q_d = C.A\,(t_b - t_a) \text{ J/s or W} \qquad \text{... (}\because\ 1 \text{ J/s} = 1 \text{ W)}$$

The value of C have been determined experimentally by O. Lasche. The values depend upon thetype of bearing, its ventilation and the temperature difference. The average values of C (in W/m^2/°C),for journal bearings may be taken as follows :

For unventilated bearings (Still air)= 140 to 420 W/m²/°C
For well ventilated bearings= 490 to 1400 W/m²/°C

It has been shown by experiments that the temperature of the bearing (t_b) is approximately mid-way between the temperature of the oil film (t_0) and the temperature of the outside air (ta). In other words

$$t_b - t_a = \frac{1}{2}(t_0 - t_a)$$

Example (1)

A(150 mm) diameter shaft supporting a load of (10 kN)has a speed of (1500r.p.m). The shaft runs in a bearing whose length is (1.5 times) the shaft diameter. If the diametral clearance of the bearing is(0.15 mm) and the absolute viscosity of the oil at the operating temperature is(0.011 kg/m-s), find the power wasted in friction.

Solution *.Given : d = 150 mm = 0.15 m ; W = 10 kN = 10 000 N ; N = 1500 r.p.m. ; l = 1.5 d ;c = 0.15 mm ; Z = 0.011 kg/m-s*

We know that length of bearing, l
= 1.5 d = 1.5 × 150 = 225 mm
∴ Bearing pressure,

$$p = \frac{W}{A} = \frac{W}{l.d} = \frac{10000}{225 \times 150} = 0.296 \text{ N/mm}^2$$

We know that coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.011 \times 1500}{0.296}\right)\left(\frac{150}{0.15}\right) + 0.002$$
$$= 0.018 + 0.002 = 0.02$$

and rubbing velocity, $V = \frac{\pi d.N}{60} = \frac{\pi \times 0.15 \times 1500}{60} = 11.78 \text{ m/s}$

We know that heat generated due to friction,

$$Q_g = \mu.W.V = 0.02 \times 10\,000 \times 11.78 = 2356 \text{ W}$$

∴ Power wasted in friction

$$= Q_g = 2356 \text{ W} = 2.356 \text{ kW} \textbf{ Ans.}$$

Example(2)

A journal bearing (60 mm)is diameter and(90 mm) long runs at (450 r.p.m). The oil used for hydrodynamic lubrication has absolute viscosity of (0.06 kg / m-s). If the diametral clearance is(0.1 mm), find the safe load on the bearing.

***Solution** .Given : d = 60 mm = 0.06 m ; l = 90 mm = 0.09 m ; N = 450 r.p.m. ;Z = 0.06 kg / m-s ; c = 0.1 mm*

First of all, let us find the bearing pressure (p) by using Sommerfeld number. We know that

$$\frac{ZN}{p}\left(\frac{d}{c}\right)^2 = 14.3 \times 10^6$$

$$\frac{0.06 \times 450}{p}\left(\frac{60}{0.1}\right)^2 = 14.3 \times 10^6 \quad \text{or} \quad \frac{9.72 \times 10^6}{p} = 14.3 \times 10^6$$

$$\therefore \quad p = 9.72 \times 10^6 / 14.3 \times 10^6 = 0.68 \text{ N/mm}^2$$

We know that safe load on the bearing,

$$W = p.A = p.l.d = 0.68 \times 90 \times 60 = 3672 \text{ N} \textbf{ Ans.}$$

Example(3)

. The load on the journal bearing is(150 kN)due to turbine shaft of(300 mm)diameter running at(1800 r.p.m). Determine the following :

1. Length of the bearing if the allowable bearing pressure is (1.6 N/mm^2), 2. Amount of heat to be removed by the lubricant per minute if the bearing temperature is (60°C)and viscosity of the oil at(60°C) is(0.02kg/m-s) and the bearing clearance is (0.25 mm).

***Solution** .Given : W = 150 kN = 150 × 10^3 N ;d = 300 mm = 0.3 m ; N = 1800 r.p.m. ;p = 1.6 N/mm2 ; Z = 0.02 kg / m-s ; c = 0.25 mm*

1. Length of the bearing
Let l = Length of the bearing in mm.
We know that projected bearing area,
$A = l \times d = l \times 300 = 300\, l\ mm^2$ and
allowable bearing pressure (p),

$$1.6 = \frac{W}{A} = \frac{150 \times 10^3}{300\, l} = \frac{500}{l}$$

$l = 500 / 1.6 = 312.5$ mm **Ans.**

2. Amount of heat to be removed by the lubricant
We know that coefficient of friction for the bearing,

$$\mu = \frac{33}{10^8}\left(\frac{Z.N}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.02 \times 1800}{1.6}\right)\left(\frac{300}{0.25}\right) + 0.002$$

$$= 0.009 + 0.002 = 0.011$$

$$V = \frac{\pi d.N}{60} = \frac{\pi \times 0.3 \times 1800}{60} = 28.3 \text{ m/s}$$

∴ Amount of heat to be removed by the lubricant,

$Q_g = \mu.W.V = 0.011 \times 150 \times 103 \times 28.3 = 46\,695$ J/s or W

$= 46.695$ kW **Ans.** ... (1 J/s = 1 W)

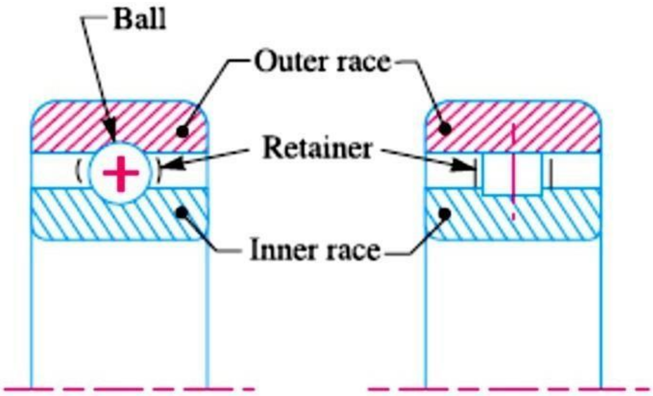
Home work

1. *The main bearing of a steam engine is (100 mm) in diameter and (175 mm)long. The bearing supports a load of(28 kN) at (250 r.p.m). If the ratio of the diametral clearance to the diameter is(0.001) and the absolute viscosity of the lubricating oil is(0.015 kg/m-s), find :*
   1. The coefficient of friction ; and 2.The heat generated at the bearing due to friction.
   [Ans. 0.002 77 ; 101.5 J/s]
2. *A journal bearing is proposed for a steam engine. The load on the journal is (3 kN), diameter (50 mm)length (75 mm), speed(1600 r.p.m)., diametral clearance(0.001 mm), ambient temperature (15.5°C). Oil SAE (10) is used and the film temperature is (60°C). Determine the heat generated and heat dissipated .Take absolute viscosity of SAE10 at (60°C = 0.014 kg/m-s).*
   [Ans. 141.3 J/s ; 25 J/s]

3. *A (100 mm) long and(60 mm) diameter journal bearing supports a load of(2500 N) at (600 r.p.m). If the room temperature is(20°C), what should be the viscosity of oil to limit the bearing surface temperature to(60°C)? The diametral clearance is(0.06 mm) and the energy dissipation coefficient based on projected area of bearing is*
   (210 W/m2/°C). [Ans. 0.0183 kg/m-s]

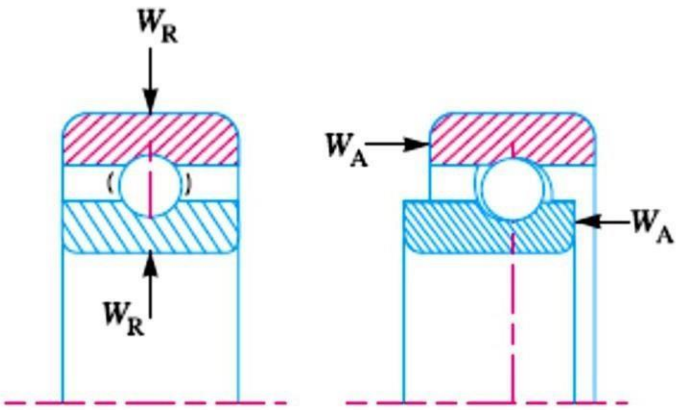
Ball and Roller Bearings

The ball and roller bearings consist of an inner race which is mounted on the shaft or journal and an outer race which is carried by the housing or casing. In between the inner and outer race .The following are the two types of rolling contact bearings:
1. Ball bearings; and 2. Roller bearings.



(a) Ball bearing. (b) Roller bearing.

1. Ball bearings are classified as :
(a) Radial bearings,
(b) Thrust bearings.



(a) Radial ball bearing. (b) Thrust ball bearing.

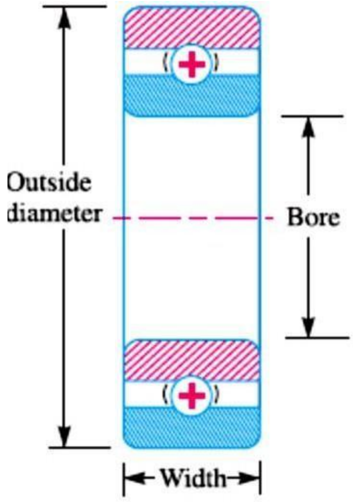
1. Radial Ball Bearings

Following are the various types of radial ball bearings:

(*a*) Single row deep groove. (*b*) Filling notch. (*c*) Angular contact. (*d*) Double row. (*e*) Self-aligning.

The most common ball bearings are available in four series as follows :

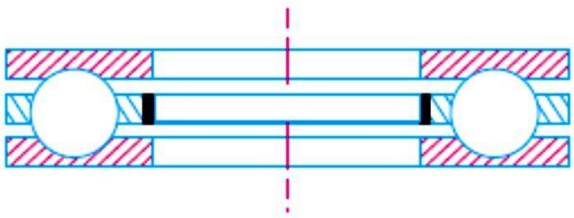
1. Extra light (100),
2. Light (200),
3. Medium (300),
4. Heavy (400)



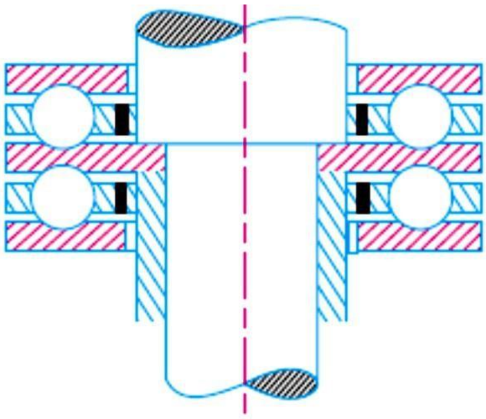
Standard designations of ball bearings

2. Thrust Ball Bearings

The thrust ball bearings are used for carrying thrust loads exclusively and at speeds below 2000r.p.m.



(*a*) Single direction thrust ball bearing.

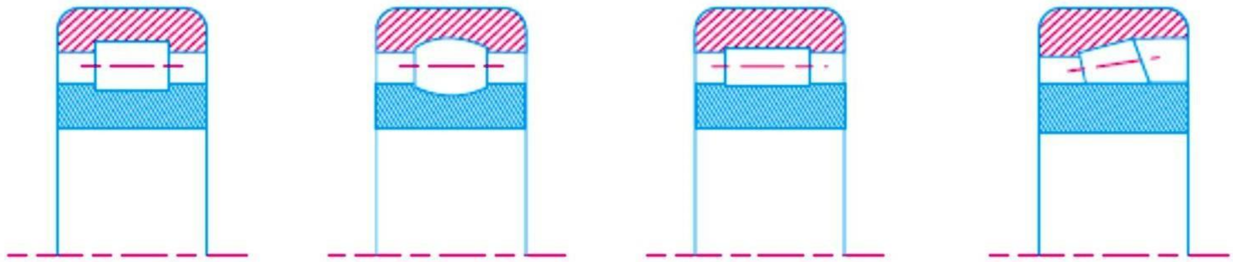


(*b*) Double direction thrust ball bearing.

o

ller Bearings

Following are the principal types of roller bearings :



(*a*) Cylindrical roller. (*b*) Spherical roller. (*c*) Needle roller. (*d*) Tapered roller.

Basic Static Load Rating of Contact Bearings

The load carried by a non-rotating bearing is called a static load. The basic static load rating is defined as the static radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings) .

1. For radial ball bearings, the basic static radial load rating (C_0) is given by

$C_0 = f_0.i.Z.D^2 \cos \alpha$

Where i = Number of rows of balls in any one bearing,

Z = Number of ball per row, D = Diameter of balls, in mm, α = Nominal angle of contact i.e. the nominal angle between the line of action of the ball load and a plane perpendicular to the axis of bearing ,and

f_0 = A factor depending upon the type of bearing.

The value of factor (f_0) for bearings made of hardened steel are taken as follows :

f_0 = 3.33, for self-aligning ball bearings

= 12.3, for radial contact and angular contact groove ball bearings.

2. For radial roller bearings, the basic static radial load rating is given by

$C_0 = f_0.i.Z.le.D \cos \alpha$

Where i = Number of rows of rollers in the bearing, Z = Number of rollers per row, le = Effective length of contact between one roller and that ring (or washer) where the contact is the shortest (in mm).

D = Diameter of roller in mm. It is the mean diameter in case of tapered rollers, α = Nominal angle of contact. It is the angle between the line of action of the roller resultant load and a plane perpendicular to the axis of the bearing, and $f_0 = 21.6$*, for bearings made of hardened steel.*

3. For thrust ball bearings, the basic static axial load rating is given by

$C_0 = f_0.Z.D^2 \sin \alpha$ *where Z = Number of balls carrying thrust in one direction, and* $f_0 = 49$*, for bearings made of hardened steel.*

4. For thrust roller bearings, the basic static axial load rating is given by

$C_0 = f_0.Z.le.D.\sin \alpha$ *where Z = Number of rollers carrying thrust in one direction, and* $f_0 = 98.1$*, for bearings made of hardened steel*

Static Equivalent Load Contact Bearings

The static equivalent load may be defined as the static radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings) or thrust loads is given by the greater magnitude of those obtained by the following two equations, i.e.

1. $W_{0R} = X_0.W_R + Y_0.W_A$;

2. $W_{0R} = W_R$ *where* W_R = *Radial load,*

W_A = *Axial or thrust load,*

X_0 = *Radial load factor, and*

Y_0 = *Axial or thrust load factor*

According to IS : 3824 – 1984, the values of X_0 and Y_0 for different bearings are given in the following table :

Table 27.2. Values of X_0 and Y_0 for radial bearings.

S.No.	Type of bearing	Single row bearing		Double row bearing	
		X_0	Y_0	X_0	Y_0
1.	Radial contact groove ball bearings	0.60	0.50	0.60	0.50
2.	Self aligning ball or roller bearings and tapered roller bearing	0.50	0.22 cot θ	1	0.44 cot θ
3.	Angular contact groove bearings :				
	α = 15°	0.50	0.46	1	0.92
	α = 20°	0.50	0.42	1	0.84
	α = 25°	0.50	0.38	1	0.76
	α = 30°	0.50	0.33	1	0.66
	α = 35°	0.50	0.29	1	0.58
	α = 40°	0.50	0.26	1	0.52
	α = 45°	0.50	0.22	1	0.44

Notes :

1. *The static equivalent radial load (W0R) is always greater than or equal to the radial load (WR).*

2. *For two similar single row angular contact ball bearings, mounted 'face-to-face' or back-to-back', use the values of X_0 and Y_0 which apply to a double row angular contact ball bearings. For two or more similar single row angular contact ball bearings mounted 'in tandem', use the values of X_0 and Y_0 which apply to a single row angular contact ball bearings.*
3. *The static equivalent radial load (W_{0R}) for all cylindrical roller bearings is equal to the radial load (W_R).*
4. *The static equivalent axial or thrust load (W_{0A}) for thrust ball or roller bearings with angle of contactα ≠ 90^o, under combined radial and axial loads is given by*

$W_{0A} = 2.3\, W_R.\tan\alpha + W_A$

This formula is valid for all ratios of radial to axial load in the case of direction bearings. For single direction bearings, it is valid where $W_R / W_A \leq 0.44 \cot\alpha$.The *thrust ball or roller bearings with $\alpha = 90^o$ can support axial loads only. The static equivalent axial load for this type of bearing is given by $W_{0A} = W_A$*

Basic Dynamic Load Rating of Contact Bearings

The basic dynamic load rating is defined as the constant stationary radial load (in case of radial ball or roller bearings) or constant axial load (in case of thrust ball or roller bearings) .

The basic dynamic load rating (C) in newtons for ball and roller bearings may be obtained as below :

1. *According to IS: 3824 (Part 1)– 1983, the basic dynamic radial load rating for radial and angular contact ball bearings, except the filling slot type, with balls not larger than 25.4 mm in diameter ,is given by $C = fc\ (i\cos\alpha)^{0.7}\ Z^{2/3}\ .\ D^{1.8}$*

and for balls larger than 25.4 mm in diameter,
$C = 3.647\ fc\ (i\cos\alpha)^{0.7}\ Z^{2/3}\ .\ D^{1.4}$

where fc = A factor, depending upon the geometry of the bearing components, the accuracy of manufacture and the material used.

2. *According to IS: 3824 (Part 2)–1983, the basic dynamic radial load rating for radial roller bearings is given by $C = fc\ (i.le\cos\alpha)^{7/9}\ Z^{3/4}.\ D^{29/27}$*

2. *According to IS: 3824 (Part 3)–1983, the basic dynamic axial load rating for single row, single or double direction thrust ball bearings is given as follows :*
   (a) For balls not larger than 25.4 mm in diameter and $\alpha = 90^o$, $C = fc\ .\ Z^{2/3}\ .\ D^{1.8}$

   (b) For balls not larger than 25.4 mm in diameter and $\alpha \neq 90^o$, $C = fc\ (\cos\alpha)^{0.7}\ \tan\alpha.\ Z^{2/3}\ .\ D^{1.8}$

   (c) For balls larger than 25.4 mm in diameter and $\alpha = 90^o$
   $C = 3.647\ fc\ .\ Z^{2/3}\ .\ D^{1.4}$

   (d) For balls larger than 25.4 mm in diameter and $\alpha \neq 90^o$,
   $D = 3.647\ fc\ (\cos\alpha)^{0.7}\ \tan\alpha\ .\ Z^{2/3}\ .\ D^{1.4}$

3. *According to IS: 3824 (Part 4)–1983, the basic dynamic axial load rating for single row ,single or double direction thrust roller bearings is given by $C = fc\ .le^{7/9}\ .\ Z^{3/4}\ .\ D^{29/27}$ ... (when $\alpha = 90^o$)*
   $= fc\ (le\cos\alpha)^{7/9}\ \tan\alpha.Z^{3/4}.\ D^{29/27}$ ... (when $\alpha \neq 90^o$)

Life of a Bearing

The life of an individual ball (or roller) bearing may be defined as the number of revolutions (or hours at some given constant speed) which the bearing runs before the *first evidence of fatigue develops in the material of one of the rings or any of the rolling elements.*

Dynamic Equivalent Load for Rolling Contact Bearings

The dynamic equivalent load may be defined as the constant stationary radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings)

The dynamic equivalent radial load (W) for radial and angular contact bearings, except the filling slot types, under combined constant radial load (WR) and constant axial or thrust load (WA) is given by

$W = X \cdot V \cdot W_R + Y \cdot W_A$ *where*

V = A rotation factor,

= 1, for all types of bearings when the inner race is rotating,

= 1, for self-aligning bearings when inner race is stationary,

= 1.2, for all types of bearings except self-aligning, when inner race is stationary.

The values of radial load factor (X) and axial or thrust load factor (Y) for the dynamically loaded bearings may be taken from the following table

Table 27.4. Values of *X* and *Y* for dynamically loaded bearings.

Type of bearing	Specifications	$\frac{W_A}{W_R} \leq e$		$\frac{W_A}{W_R} > e$		e
		X	Y	X	Y	
Deep groove	$\frac{W_A}{C_0} = 0.025$				2.0	0.22
ball bearing	= 0.04				1.8	0.24
	= 0.07				1.6	0.27
	= 0.13	1	0	0.56	1.4	0.31
	= 0.25				1.2	0.37
	= 0.50				1.0	0.44
Angular contact	Single row		0	0.35	0.57	1.14
ball bearings	Two rows in tandem		0	0.35	0.57	1.14
	Two rows back to back	1	0.55	0.57	0.93	1.14
	Double row		0.73	0.62	1.17	0.86
Self-aligning bearings	Light series : for bores					
	10 – 20 mm		1.3		2.0	0.50
	25 – 35	1	1.7	6.5	2.6	0.37
	40 – 45		2.0		3.1	0.31
	50 – 65		2.3		3.5	0.28
	70 – 100		2.4		3.8	0.26
	105 – 110		2.3		3.5	0.28
	Medium series : for bores					
	12 mm		1.0	0.65	1.6	0.63
	15 – 20		1.2		1.9	0.52
	25 – 50		1.5		2.3	0.43
	55 – 90		1.6		2.5	0.39
Spherical roller	For bores :					
bearings	25 – 35 mm		2.1		3.1	0.32
	40 – 45	1	2.5	0.67	3.7	0.27
	50 – 100		2.9		4.4	0.23
	100 – 200		2.6		3.9	0.26
Taper roller	For bores :					
bearings	30 – 40 mm				1.60	0.37
	45 – 110	1	0	0.4	1.45	0.44
	120 – 150				1.35	0.41

Dynamic Load Rating for Contact Bearings under Variable Loads

The approximate rating (or service) life of ball or roller bearings is based on the fundamental equation,

$$L = \left(\frac{C}{W}\right)^k \times 10^6 \text{ revolutions}$$

$$C = W\left(\frac{L}{10^6}\right)^{1/k}$$

Where L = Rating life,

C = Basic dynamic load rating,

W = Equivalent dynamic load,

k = 3, for ball bearings,

= 10/3, for roller bearings.

The relationship between the life in revolutions (L) and the life in working hours (L_H) is given by

$L = 60\ N\ .\ L_H$ *revolutions*

Where N is the speed in r.p.m

Selection of Radial Ball

Bearings In order to select a most suitable ball bearing, first of all, the basic dynamic radial load is calculated. It is then multiplied by the service factor (KS) to get the design basic dynamic radial load capacity. The service factor for the ball bearings is shown in the following table.

Table 27.5. Values of service factor (K_S).

S.No.	Type of service	Service factor (K_S) for radial ball bearings
1.	Uniform and steady load	1.0
2.	Light shock load	1.5
3.	Moderate shock load	2.0
4.	Heavy shock load	2.5
5.	Extreme shock load	3.0

Table 27.6. Basic static and dynamic capacities of various types of radial ball bearings.

Bearing No.	Basic capacities in kN							
	Single row deep groove ball bearing		Single row angular contact ball bearing		Double row angular contact ball bearing		Self-aligning ball bearing	
	Static (C_0)	Dynamic (C)	Stetic (C_0)	Dynamic (C)	Static (C_0)	Dynamic (C)	Static (C_0)	Dynamic (C)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
200	2.24	4	—	—	4.55	7.35	1.80	5.70
300	3.60	6.3	—	—	—	—	—	—
201	3	5.4	—	—	5.6	8.3	2.0	5.85
301	4.3	7.65	—	—	—	—	3.0	9.15
202	3.55	6.10	3.75	6.30	5.6	8.3	2.16	6
302	5.20	8.80	—	—	9.3	14	3.35	9.3
203	4.4	7.5	4.75	7.8	8.15	11.6	2.8	7.65
303	6.3	10.6	7.2	11.6	12.9	19.3	4.15	11.2
403	11	18	—	—	—	—	—	—
204	6.55	10	6.55	10.4	11	16	3.9	9.8
304	7.65	12.5	8.3	13.7	14	19.3	5.5	14
404	15.6	24	—	—	—	—	—	—
205	7.1	11	7.8	11.6	13.7	17.3	4.25	9.8
305	10.4	16.6	12.5	19.3	20	26.5	7.65	19
405	19	28	—	—	—	—	—	—
206	10	15.3	11.2	16	20.4	25	5.6	12
306	14.6	22	17	24.5	27.5	35.5	10.2	24.5
406	23.2	33.5	—	—	—	—	—	—
207	13.7	20	15.3	21.2	28	34	8	17
307	17.6	26	20.4	28.5	36	45	13.2	30.5
407	30.5	43	—	—	—	—	—	—
208	16	22.8	19	25	32.5	39	9.15	17.6
308	22	32	25.5	35.5	45.5	55	16	35.5
408	37.5	50	—	—	—	—	—	—
209	18.3	25.5	21.6	28	37.5	41.5	10.2	18
309	30	41.5	34	45.5	56	67	19.6	42.5
409	44	60	—	—	—	—	—	—
210	21.2	27.5	23.6	29	43	47.5	10.8	18
310	35.5	48	40.5	53	73.5	81.5	24	50
410	50	68	—	—	—	—	—	—

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
211	26	34	30	36.5	49	53	12.7	20.8
311	42.5	56	47.5	62	80	88	28.5	58.5
411	60	78	—	—	—	—	—	—
212	32	40.5	36.5	44	63	65.5	16	26.5
312	48	64	55	71	96.5	102	33.5	68
412	67	85	—	—	—	—	—	—
213	35.5	44	43	50	69.5	69.5	20.4	34
313	55	72	63	80	112	118	39	75
413	76.5	93	—	—	—	—	—	—
214	39	48	47.5	54	71	69.5	21.6	34.5
314	63	81.5	73.5	90	129	137	45	85
414	102	112	—	—	—	—	—	—
215	42.5	52	50	56	80	76.5	22.4	34.5
315	72	90	81.5	98	140	143	52	95
415	110	120	—	—	—	—	—	—
216	45.5	57	57	63	96.5	93	25	38
316	80	96.5	91.5	106	160	163	58.5	106
416	120	127	—	—	—	—	—	—
217	55	65.5	65.5	71	100	106	30	45.5
317	88	104	102	114	180	180	62	110
417	132	134	—	—	—	—	—	—
218	63	75	76.5	83	127	118	36	55
318	98	112	114	122	—	—	69.5	118
418	146	146	—	—	—	—	—	—
219	72	85	88	95	150	137	43	65.5
319	112	120	125	132	—	—	—	—
220	81.5	96.5	93	102	160	146	51	76.5
320	132	137	153	150	—	—	—	—
221	93	104	104	110	—	—	56	85
321	143	143	166	160	—	—	—	—
222	104	112	116	120	—	—	64	98
322	166	160	193	176	—	—	—	—

Example (1)

Select a single row deep groove ball bearing for a radial load of 4000 N and an axial load of 5000 N, operating at a speed of 1600 r.p.m. for an average life of 5 years at 10 hours per day. Assume uniform and steady load.

Solution *.Given :* $W_R = 4000$ N ; $W_A = 5000$ N ; $N = 1600$ r.p.m.

Since the average life of the bearing is 5 years at 10 hours per day, therefore life of the bearing in hours,

$L_H = 5 \times 300 \times 10 = 15\,000$ *hours ... (Assuming 300 working days per year)*

and life of the bearing in revolutions,

$L = 60\ N \times L_H = 60 \times 1600 \times 15\,000 = 1440 \times 10^6$ *rev*

We know that the basic dynamic equivalent radial load,

$W = X.V.W_R + Y.W_A$ *...(i)*

In order to determine the radial load factor (X) and axial load factor (Y), we require W_A / W_R *and* W_A / C_0*. Since the value of basic static load capacity (C_0) is not known, therefore let us take*

$W_A / C_0 = 0.5$*. Now from Table 27.4, we find that the values of X and Y corresponding to WA / C0*

= 0.5 and $W_A/W_R = 5000 / 4000 = 1.25$ *(which is greater than e = 0.44) are*

$X = 0.56$ *and* $Y = 1$

Since the rotational factor (V) for most of the bearings is 1, therefore basic dynamic equivalent radial load,

$W = 0.56 \times 1 \times 4000 + 1 \times 5000 = 7240$ *N*

From Table 27.5, we find that for uniform and steady load, the service factor (K_S) for ball bearings is 1. Therefore the bearing should be selected for W = 7240 N.

We know that basic dynamic load rating

$$C = W\left(\frac{L}{10^6}\right)^{1/k} = 7240\left(\frac{1440 \times 10^6}{10^6}\right)^{1/3} = 81\,760 \text{ N}$$

$$= 81.76 \text{ kN} \quad \ldots (\because k = 3, \text{ for ball bearings})$$

From Table 27.6, let us select the bearing No. 315 which has the following basic capacities,

$C_0 = 72$ *kN = 72 000 N and C = 90 kN = 90 000 N*

Now $W_A / C_0 = 5000 / 72\,000 = 0.07$

∴ *From Table 27.4, the values of X and Y are*

$X = 0.56$ *and* $Y = 1.6$

Substituting these values in equation (i), we have dynamic equivalent load,

$W = 0.56 \times 1 \times 4000 + 1.6 \times 5000 = 10\,240$ N

∴ *Basic dynamic load rating*

$$C = 10\,240\left(\frac{1440\times10^6}{10^6}\right)^{1/3} = 115\,635 \text{ N} = 115.635 \text{ kN}$$

From Table 27.6, the bearing number 319 having C = 120 kN, may be selected. Ans.

Example(2).

A single row angular contact ball bearing number (310)is used for an axial flow compressor. The bearing is to carry a radial load of (2500 N) and an axial or thrust load of (1500 N).Assuming light shock load, determine the rating life of the bearing.

Solution. *Given :* $W_R = 2500$ N ; $W_A = 1500$ N

From Table 27.4, we find that for single row angular contact ball bearing, the values of Radial factor (X) and thrust factor (Y) for $W_A / W_R = 1500 / 2500 = 0.6$ *are*

$X = 1$ *and* $Y = 0$

Since the rotational factor (V) for most of the bearings is 1, therefore dynamic equivalent load,

$W = X.V.W_R + Y.W_A = 1 \times 1 \times 2500 + 0 \times 1500 = 2500$ N

From Table 27.5, we find that for light shock load, the service factor (K_S) *is 1.5. Therefore the design dynamic equivalent load should be taken as*

$W = 2500 \times 1.5 = 3750$ N

From Table 27.6, we find that for a single row angular contact ball bearing number 310, the basic dynamic capacity,

$C = 53$ kN $= 53\,000$ N

We know that rating life of the bearing in revolutions

$$L = \left(\frac{C}{W}\right)^k \times 10^6 = \left(\frac{53\,000}{3750}\right)^3 \times 10^6 = 2823 \times 10^6 \text{ rev} \quad \textbf{Ans.}$$

... ($\because$ $k = 3$, for ball bearings)

Example(3)

Design a self-aligning ball bearing for a radial load of 7000 N and a thrust load of (2100 N). The desired life of the bearing is (160 millions)of revolutions at(300 r.p.m). Assume uniform and steady load,

Solution .*Given :* $W_R = 7000$ N ; $W_A = 2100$ N ; $L = 160 \times 10^6$ rev ; $N = 300$ r.p.m.

From Table 27.4, we find that for a self-aligning ball bearing, the values of radial factor (X) and thrust factor (Y) for $W_A / W_R = 2100 / 7000 = 0.3$*, are as follows :*

$X = 0.65$ *and* $Y = 3.5$

Since the rotational factor (V) for most of the bearings is 1, therefore dynamic equivalent load $W = X.V.W_R + Y.W_A = 0.65 \times 1 \times 7000 + 3.5 \times 2100 = 11\,900$ N

From Table 27.5, we find that for uniform and steady load, the service factor KS for ball bearings is 1. Therefore the bearing should be selected for W = 11 900 N. We know that the basic dynamic load rating

$$C = W\left(\frac{L}{10^6}\right)^{1/k} = 11\,900\left(\frac{160 \times 10^6}{10^6}\right)^{1/3} = 64\,600 \text{ N} = 64.6 \text{ kN}$$

... ($\because$ $k = 3$, for ball bearings)

From Table 27.6, let us select bearing number 219 having $C = 65.5$ kN Ans.

Home work

A single row deep groove ball bearing operating at(2000 r.p.m). is acted by a (10 kN) radial load and (8 kN) thrust load. The bearing is subjected to a light shock load and the outer ring is rotating. Determine the rating life of the bearing.

[Ans. 15.52 × 106 rev]

Design Of Gears

In engineering and technology the term gear is defined as a machine element used to transmit motion and power between rotating shaft by means of progressive engagement of projections call teeth . A gear drive is also provided, when the distance between the driver and the follower is very small.

Advantages and Disadvantages of Gear Drives The following are the advantages and disadvantages of the gear drive as compared to other drives, i.e. belt, rope and chain drives : ***Advantages***

1. It transmits exact velocity ratio.

2. It may be used to transmit large power.

3. It may be used for small centre distances of shafts.

4. It has high efficiency.

5. It has reliable service.

6. It has compact layout.

Disadvantages

1. Since the manufacture of gears require special tools and equipment, therefore it is costlier than other drives

2. The error in cutting teeth may cause vibrations and noise during operation.

3. It requires suitable lubricant and reliable method of applying it, for the proper operation of gear drives

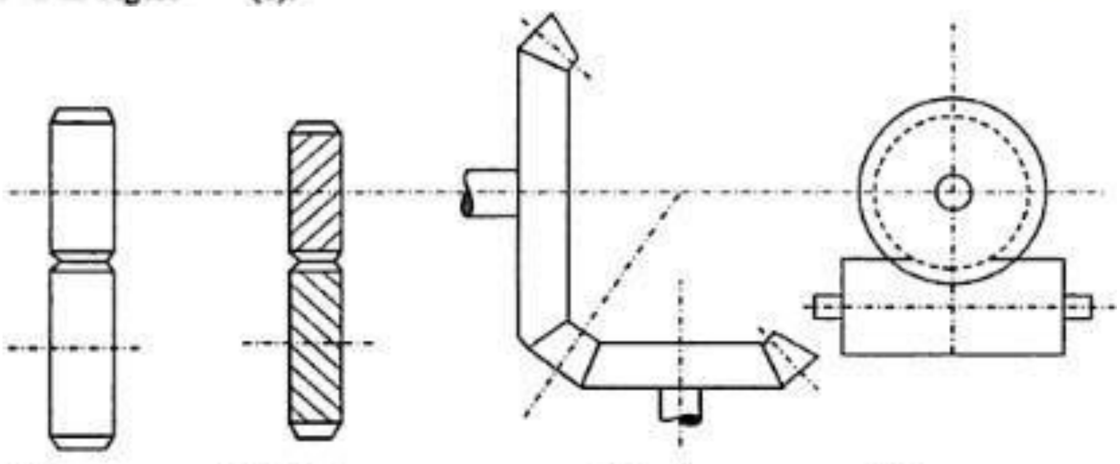
Classification of Gears

Spur gears. A gear having straight teeth along the axis is called the spur gear. Spur gears are used to transmit power between two parallel shafts as shown in Figure (a). A rack is a straight tooth gear which can be thought of as a segment of spur gear of infinite diameter.

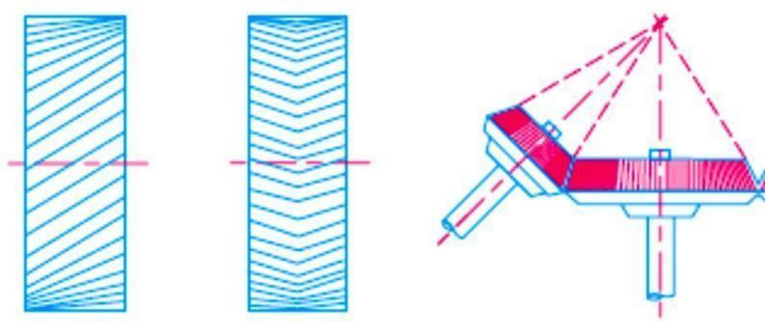
Helical gears. They are also used to transmit power between two parallel shafts and teeth are cut on the cylindrical disc. The tooth faces of these gears have a certain degree of helix angle of opposite hand on pinion and gear as shown in Figure (b). These gears are smooth in operation and therefore can transmit power at a high pitch line velocity.

Bevel gear. When power is to be transmitted between two intersecting shafts, bevel gears are used. The angle of intersection of shafts is called the *shaft angle*. The gear blank is a frustum of cone on which teeth are generated. The teeth are straight but their sides are tapered so that all lines, when extended, meet at a common point called the *apex* of the cone, as shown in Figure (c).

Worm and worm gears. In this system of gearing, the axes of the power transmitting shafts are neither parallel nor intersecting but the planes containing the axes are generally at right angles to each other. The teeth used are helical. The schematic diagram of a worm gear set is shown in Figure (d).



(a) Spur gears (b) Helical gears (c) Bevel gears (d) Worm gears

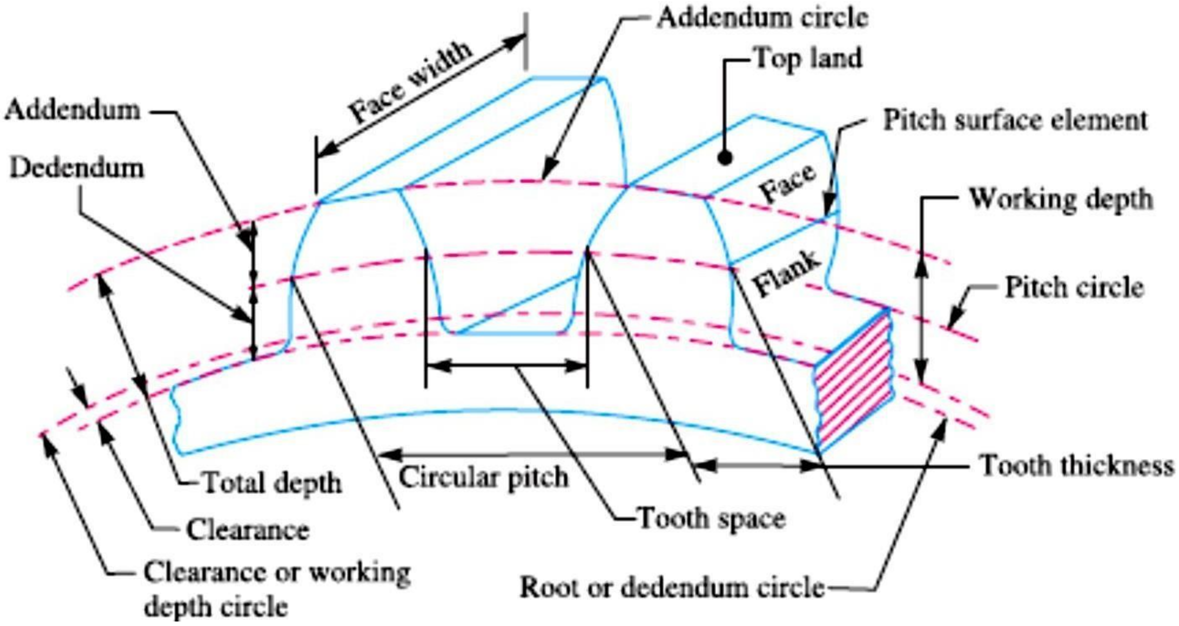


(*a*) Single helical gear. (*b*) Double helical gear. (*c*) Bevel gear.

Terms used in Gears

The following terms, which will be mostly used in this chapter, should be clearly understood atthis stage. These terms are illustrated in Fig.

1. *Pitch circle.*



2. *Pitch circle diameter.*
3. *Pitch point.*
4. *Pitch surface.*
5. *Pressure angle or angle of obliquity*
6. *Addendum.*
7. *Dedendum.*
8. *Addendum circle.*
9. *Dedendum circle.*
10. *Circular pit Circular pitch, pc = πD/T where D = Diameter of the pitch circle, and T = Number of teeth on the wheel.*
11. *Diametral pitch. It denoted by*

Pd. Mathematically ,Diametral pitch, pd=T / D=π / pc where

T = Number of teeth, and

D = Pitch circle diameter.

12. Module. 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, 16,20, 25, 32, 40 and 50.

The modules 1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5,5.5, 7, 9, 11, 14, 18, 22, 28, 36 and 45 are of second choice.

13. *Clearance.*
14. *Total depth. It is the radial distance between the addendum and the dedendum circle of a gear.*
15. *Working depth. It is radial distance from the addendum circle to the clearance circle.*
16. *Tooth thickness. It is the width of the tooth measured along the pitch circle.*
17. *Tooth space. It is the width of space between the two adjacent teeth measured along the pitch circle.*
18. *Backlash. It is the difference between the tooth space and the tooth thickness, as Measured on the pitch circle.*
19. *Face of the tooth. It is surface of the tooth above the pitch surface.*
20. *Top land. It is the surface of the top of the tooth.*
21. *Flank of the tooth. It is the surface of the tooth below the pitch surface.*
22. *Face width. It is the width of the gear tooth measured parallel to its axis.*
23. *Profile. It is the curve formed by the face and flank of the tooth.*
24. *Fillet radius. It is the radius that connects the root circle to the profile of the tooth.*
25. *Path of contact. It is the path traced by the point of contact of two teeth from the Beginning to the end of engagement.*
26. *Length of the path of contact. It is the length of the common normal cut-off by the Addendum circles of the wheel and pinion.*
27. *Arc of contact. It is the path traced by a point on the pitch circle from the beginning to the end of engagement of a given pair of teeth. The arc of contact consists of two parts,*

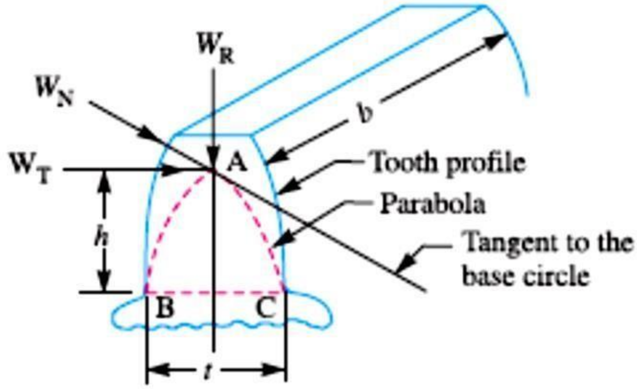
Gear Materials

The cast iron is widely used for the manufacture of gears due to its good wearing properties .The steel is used for high strength gears and steel may be plain carbon steel or alloy steel .The steel gears are usually heat treated in order to combine properly the toughness and tooth hardness.

The phosphor bronze is widely used for worm gears in order to reduce wear of the worms which will be excessive with cast iron or steel. The following table shows the properties of commonly used gear materials.

Strength of Gear Teeth – Lewis Equation

Lewis assumed that as the load is being transmitted from one gear to another, it is all given and taken by one tooth, because it is not always safe to assume that the load is distributed among several teeth. When contact begins, the load is assumed to be at the end of the driven teeth and as contact ceases, it is at the end of the driving teeth. This may not be true when the number of teeth in a pair of mating gears is large, because the load may be distributed among several teeth.



Consider each tooth as a cantilever beam loaded by a normal load (W_N) as shown in Fig.. It is resolved into two components i.e .tangential component (W_T) and radial component(W_R) acting perpendicular and parallel to the centre line of the tooth respectively. The tangential component (W_T) induces a bending stress which tends to break the tooth. The radial component (W_R) induces a compressive stress of relatively small magnitude ,therefore its effect on the tooth may be neglected .

Hence, the bending stress is used as the basis for design calculations. The critical section or the section of maximum bending stress may be obtained by drawing a parabola through A and tangential to the tooth curves at B and C. But the tooth is larger than the parabola at every section except BC. We therefore, conclude that the section BC is the section of maximum stressor the critical section. The maximum value of the bending stress (or the permissible working stress),at the section BC is given by

$\sigma_w = M.y / I$ ...(i)

where M = Maximum bending moment at the critical section BC = $W_T \times h$,
W_T = Tangential load acting at the tooth, h = Length of the tooth,
y = Half the thickness of the tooth (t) at critical section BC = t/2,
I = Moment of inertia about the centre line of the tooth = b.t3/12,
b = Width of gear face

Substituting the values for M, y and I in equation (i), we get

$$\sigma_w = \frac{(W_T \times h)\ t/2}{b t^3/12} = \frac{(W_T \times h) \times 6}{b.t^2}$$

$$W_T = \sigma_w \times b \times t^2 / 6\ h$$

Let $t = x \times p_c$, and $h = k \times p_c$; where x and k are constants.

$$\therefore \quad W_T = \sigma_w \times b \times \frac{x^2 . p_c^2}{6k.p_c} = \sigma_w \times b \times p_c \times \frac{x^2}{6k}$$

Substituting $x^2 / 6k = y$, another constant, we have

$$W_T = \sigma_w . b . p_c . y = \sigma_w . b . \pi m . y \qquad ...(\because p_c = \pi m)$$

The quantity y is known as ***Lewis form factor*** or ***tooth form factor*** and W_T (which is the tangential load acting at the tooth) is called the ***beam strength of the tooth.***

$$\text{Since } y = \frac{x^2}{6k} = \frac{t^2}{(p_c)^2} \times \frac{p_c}{6h} = \frac{t^2}{6h . p_c}$$

therefore in order to find the value of y, the quantities t, h and pc may be determined analytically or measured from the drawing similar to Fig. 28.12. It may be noted that if the gear is enlarged, the distances t, h and pc will each increase proportionately. Therefore the value of y will remain unchanged. A little consideration will show that the value of y is independent of the size of the tooth and depends only on the number of teeth on a gear and the system of teeth. The value of y in terms of the number of teeth may be expressed as follows

$$y = 0.124 - \frac{0.684}{T}, \text{ for } 14\tfrac{1}{2}^\circ \text{ composite and full depth involute system.}$$

$$= 0.154 - \frac{0.912}{T}, \text{ for } 20^\circ \text{ full depth involute system.}$$

$$= 0.175 - \frac{0.841}{T}, \text{ for } 20^\circ \text{ stub system.}$$

Permissible Working Stress for Gear Teeth in the Lewis Equation

The permissible working stress (ζ_w) in the Lewis equation depends upon the material for which an allowable static stress (ζ_o) may be determined. The allowable static stress is the stress at the elastic limit of the material. It is also called the basic stress. In order to account for

the dynamic effects which become more severe as the pitch line velocity increases, the value of ζ_w is reduced .According to the Barth formula, the permissible working stress,
$\zeta_w = \zeta_o \times C_v$
where ζ_o = Allowable static stress, and C_v=
Velocity factor.
The values of the velocity factor (C_v) are given as follows :

$$C_v = \frac{3}{3+v}, \text{ for ordinary cut gears operating at velocities upto 12.5 m / s.}$$

$$= \frac{4.5}{4.5+v}, \text{ for carefully cut gears operating at velocities upto 12.5 m/s.}$$

$$= \frac{6}{6+v}, \text{ for very accurately cut and ground metallic gears operating at velocities upto 20 m / s.}$$

$$= \frac{0.75}{0.75+\sqrt{v}}, \text{ for precision gears cut with high accuracy and operating at velocities upto 20 m / s.}$$

$$= \left(\frac{0.75}{1+v}\right) + 0.25, \text{ for non-metallic gears.}$$

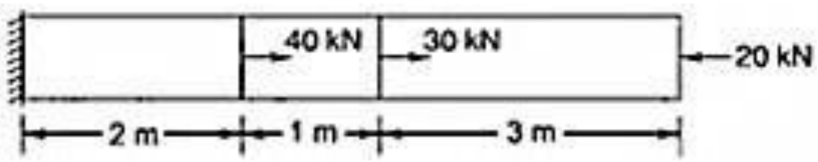
In the above expressions, v is the pitch line velocity in meters per second .The following table shows the values of allowable static stresses for the different gear materials.

FOUNDATION OF TECHNICAL EDUCATION
AMARA TECHNICAL INSTITUTE
MECHANICAL DEPARTEMENT
Machines Elements Exam For 2nd Year Students
1st semester Time : 2 hrs

NOTE : ANSWER ONLY FOUR QUESTIONS .

Q1

Determine the stress in all the three sections and total deformation of steel rod shown in the figure cross- sectional area (10 cm^2) and modulus of elasticity (200 GN/m^2) (5 marks)



Q2

Find the efficiency of the Single riveted lap joint of (6 mm) thick plates with (20 mm) diameter rivets having a pitch of(50 mm). Permissible tensile stress in plate (120MPa) ,Permissible shearing stress in rivets (90 MPa) and Permissible crushing stress in rivets (180 MPa). (5 marks)

Q3

A plate (100 mm) wide and (10 mm) thick is to be welded to another plate by means of double parallel fillets. The plates are subjected to a static load of (80 kN). Find the length of weld if the permissible shear stress in the weld does not exceed (55 MPa). (5 marks)

Q4

Two machine parts are fastened together tightly by means of a (24 mm) tap bolt. If the load tending to separate these parts is neglected, find the stress that is set up in the bolt by the initial tightening. (5 marks)

Q5

Design the rectangular key for a shaft of(50 mm) diameter. The shearing and crushing stresses for the key material are (42 MPa) and(70 MPa) with cross section of key (16 x 10) mm. (5 marks)

--------------------------------------***Good luck*** ------------------------------------

AMARA TECHNICAL INSTITUTE

MECHANICAL DEPARTEMENT

Machines Elements Exam For 2nd Year Students

2st semester Time : 2 hrs NOTE : ANSWER ALL QUESTIONS

Q1

Design the rectangular key for a shaft of(50 mm) diameter. The shearing and crushing stresses for the key material are (42 MPa) and(70 MPa) with cross section of key (16 x 10) mm. (5 marks)

Q2

A plate clutch having a single driving plate with contact surfaces on each side is required to transmit (110 kW) at (1250 r.p.m). The outer diameter of the contact surfaces is to be (300 mm). The coefficient of friction is(0.4). Assuming a uniform pressure of (0.17 N/mm^2); determine the inner diameter of the friction surfaces. (5 marks)

Q3

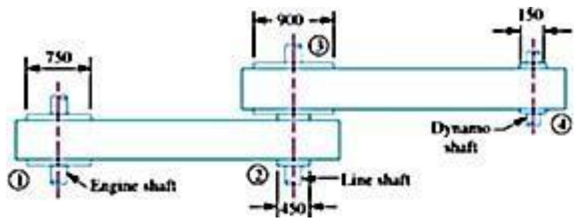
Design a helical compression spring for a maximum load of(1000 N)for a deflection of (25 mm) using the value of spring index as(5). The maximum permissible shear stress for spring wire is(420 MPa) and modulus of rigidity is (84 kN/mm^2). (5 marks)

Q4

An engine running at (150r.p.m).drives a line shaft by means of a belt.

The engine pulley is(750 mm) diameter and the pulley on the line shaft is (

450 mm). A (900 mm) diameter pulley on the line shaft drives a(150 mm) diameter pulley keyed to a dynamo shaft. Fine the speed of dynamo shaft, when 1. there is no slip, And 2. there a slip of (2%) at each drive. *(5 marks)*



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FOUNDATION OF TECHNICAL EDUCATION

AMARA TECHNICAL INSTITUTE

MECHANICAL DEPARTEMENT

Machines Elements Exam For 2nd Year Students

2st semester

Time : 2 hrs

NOTE : ANSWER ALL QUESTIONS .

Q1

A plate clutch having a single driving plate with contact surface on each side is required to transmit (110 Kw) at (1250 r.p.m) .The outer radius of contact surface is to be (150 mm) . The coefficient of friction (0.4) . Assuming uniform pressure of (0.17 N/mm^2).Determine the inner diameter of the friction surface .

Q2

A helical spring is made of wire (6 mm) diameter and has spring index (11.5 mm).If the permissible shear stress is (350 MPa) and modulus of rigidity (84 KN/mm^2) .Find the axial load which the spring can carry and the deflection per active turn .

Q3

A two pulleys one (450 mm) diameter and the other (200mm) diameter on parallel shaft (1.95 m) apart are connected by a crossed belt .Find the length of belt required and the angle of connect between the belt and each pulley . What power can be transmitted by the belt when the large pulley rotates at (200 rev/min) .If the maximum permissible tension in the belt is (1000 N) and the coefficient of friction between the belt and pulley is (0.25) .Find the number of belt required if the total power (90 KW)

---------------------------------------Good luck ---------------------------------------

FOUNDATION OF TECHNICAL EDUCATION \ AMARA TECHNICAL INSTITUTE
MECHANICAL DEPARTEMENT
Machines Elements Exam For 2nd Year Students

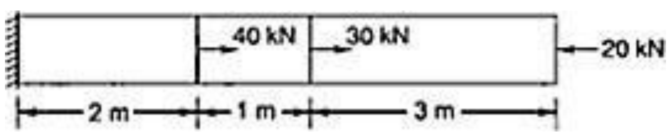
Final Exam 1st Attempt

Time : 3 hrs

NOTE : ANSWER ONLY FIVE QUESTIONS .

Q1

Determine the stress in all the three sections and total deformation of steel rod shown in the figure cross-sectional area (10 cm^2) and modulus of elasticity (20 GN/m^2). (10 marks)



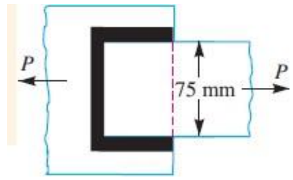
Q2

A double riveted lap joint with zig-zag riveting is to be design for (13 mm) thickness of plate if the permissible tensile stress(80 MPa) , permissible shearing stress in rivets (60 MPa)and permissible crushing stress in rivets (120MPa) . Find the efficiency . (10 marks)

Q3

A plate(75 mm)wide and (10 mm) thick is joined with another plate steel by means of a single transverse and a double parallel fillet welds as shown in Fig. The joint is subject to a maximum tensile force of (55 KN) .The permissible tensile stress (70 N/mm^2) and permissible shearing stress (50 N/mm^2).

Determine the required length of each parallel fillet weld. (10 marks)



Q4

A plate clutch having a single driving plate with contact surface on each side is required to transmit (110 Kw) at (1250 r.p.m) .The outer diameter of contact surface is to be (300 mm) . The coefficient of friction (0.4) . Assuming uniform pressure of (0.17 N/mm^2).Determine the inner radius of the friction surface . (10 marks)

Q5

A helical spring is made of wire (6 mm) diameter and has outside diameter of (75 mm) .If the permissible shear stress is (350 MPa) and modulus of rigidity (84 KN/mm^2) .Find the axial load which the spring can carry and the deflection per active turn . (10 marks)

Q6

A two pulleys one (450 mm) diameter and the other (200mm) diameter on parallel shaft (1.95 m) apart are connected by a crossed belt .Find the length of belt required and the angle of connect between the belt and each pulley . What power can be transmitted by the belt when the large pulley rotates at (200 r.p.m) .If the maximum permissible tension in the belt is (1000 N) and the coefficient of friction between the belt and pulley is (0.25) .Find the number of belt required if the total power (90 KW).

(10 marks)

--Good luck

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MECHANICAL DEPARTEMENT

Machines Elements Exam For 2nd Year Students

Final Exam 1st Attempt

Time : 3 hrs

NOTE : ANSWER ONLY FIVE QUESTIONS .

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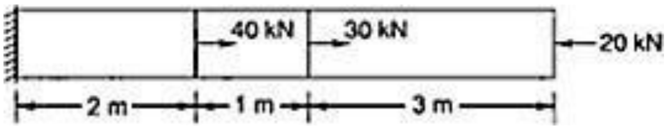
Q1

Determine the stress in all the three sections and total deformation of steel rod shown in the figure cross-sectional area (10 cm^2) and modulus of elasticity (20 GN/m^2). (10 marks)

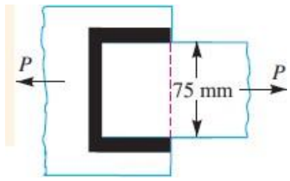


Q2

Find the efficiency of double riveted double cover butt joint in plates (20 mm) thick is made with (25 mm) diameter rivets at (100 mm) pitch. The permissible tensile stress(120 MPa) , permissible shearing stress in rivets (100 MPa)and permissible crushing stress in rivets (150MPa). (10 marks)

Q3

A plate(75 mm)wide and (10 mm) thick is joined with another plate steel by means of a single transverse and a double parallel fillet welds as shown in Fig. The joint is subject to a maximum tensile force of (55 KN) .The permissible tensile stress (70 N/mm^2) and permissible shearing stress (50 N/mm^2). Determine the required length of each parallel fillet weld (10 marks)



Q4

A plate clutch having a single driving plate with contact surface on each side is required to transmit (110 Kw) at (1250 r.p.m) .The outer radius of contact surface is to be (150 mm) . The coefficient of friction (0.4) . Assuming uniform pressure of (0.17 N/mm^2).Determine the inner diameter of the friction surface . (10 marks)

Q5

A helical spring is made of wire (6 mm) diameter and has spring index (11.5 mm).If the permissible shear stress is (350 MPa) and modulus of rigidity (84 KN/mm^2) .Find the axial load which the spring can carry and the deflection per active turn . (10 marks)

Q6

A two pulleys one (450 mm) diameter and the other (200mm) diameter on parallel shaft (1.95 m) apart are connected by a crossed belt .Find the length of belt required and the angle of connect between the belt and each pulley . What power can be transmitted by the belt when the large pulley rotates at (200 rev/min) .If the maximum permissible tension in the belt is (1000 N) and the coefficient of friction between the belt and pulley is (0.25) .Find the number of belt required if the total power (90 KW).

(10 marks)

--Good luck

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Machines Elements Exam For 2nd Year Students

Final Exam 1st Attempt

Time : 3 hrs

NOTE : ANSWER ONLY FIVE QUESTIONS .

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Q1

Two plates are fastened by means of two bolt as shown in the figure .The bolt are made of plain carbon steel with yield stress (400 N/mm2) and factor of safety is (5).Determine the size of the bolt if the load applied is (5kN). (10 marks)

Q2

Find the efficiency of double riveted double cover butt joint in plates (20 mm) thick is made with (25 mm) diameter rivets at (100 mm) pitch. The permissible tensile stress(120 MPa) , permissible shearing stress in rivets (100 MPa)and permissible crushing stress in rivets (150MPa). (10 marks)

Q3

A(45 mm)diameter shaft is made of steel with a yield strength of(400 MPa).A parallel key of size (14 mm) wide and(9 mm) thick made of steel with a yield strength of (340 MPa) is to be used. Find the required length of key, assume a factor of safety of(2)

(10 marks)

Q4

A plate clutch having a single driving plate with contact surface on each side is required to transmit (110 Kw) at (1250 r.p.m) .The outer radius of contact surface is to be (150 mm) . The coefficient of friction (0.4) . Assuming uniform pressure of (0.17 N/mm^2).Determine the inner diameter of the friction surface .

(10 marks)

Q5

A helical spring is made of wire (6 mm) diameter and has spring index (11.5 mm).If the permissible shear stress is (350 MPa) and modulus of rigidity (84 KN/mm^2) .Find the axial load which the spring can carry and the deflection per active turn (10 marks).

Q6

A two pulleys one (450 mm) diameter and the other (200mm) diameter on parallel shaft (1.95 m) apart are connected by a crossed belt .Find the length of belt required and the angle of connect between the belt and each pulley . What power can be transmitted by the belt when the large pulley rotates at (200 rev/min) .If the maximum permissible tension in the belt is (1000 N) and the coefficient of friction between the belt and pulley is (0.25) .Find the number of belt required if the total power (90 KW)

(10 marks)

--Good luck --