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SIMPLE STRESSES IN MACHINES PARTS

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

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

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

Load :

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

Types of loads :

1 – dead or steady load
2 – live or varying load
3 – suddenly applied or shock load

Stress (σ) :

The internal force per unit area at any section of the body .

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

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

Strain (Є) :

The deformation per unit length .

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

Є : Strain	**(without units)**
δl : change in length	**(mm)**
L : original length	**(mm)**

Young Modulus :(E)

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

$$\sigma \; \alpha \; \epsilon$$

$$\sigma = E\,\epsilon$$

$$E = \sigma / \epsilon$$

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

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

Shear stress : (σ_s)

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

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

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

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

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

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

t = thickness of the plate in riveted joints

n = number of rivets in crushing

Safety factor : (S.F)

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

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

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

Working stress : (σ_w)

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

Torsional shear stress : (τ)

τ **: Torque or twisting moment**

$\tau = (\pi / 16) . \sigma_s d^3$ **for solid shafts**

$\tau = (\pi / 16) . \sigma_s . (1 - k^4)$ **when $k = di / do$ for hollow shafts**

di : inner diameter of the hollow shaft (cylinder)
do : outer diameter of the hollow shaft (cylinder)

Bending stress : (σ_b)

$M = (\pi / 16) . \sigma_b . d^3$

M : bending moment

Example (1) :

A rod 100 cm long and of 2cm X 2cm cross section is subjected to a pull of 1000 kg force . if the modulus of elasticity of the material is $2*10^6$ kg/cm^2 , Determine the elongation of the rod ?

Solution :

$l = 100$ cm , $A = 2 * 2 = 4$ cm^2 , $P = 1000$ kg , $E = 2*10^6$

$$\delta l = \frac{P.l}{A.E} = \frac{1000*100}{4*2*10^6} = 0.0125 \quad cm$$

Example (2) :

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

Solution :

$P = 5$ KN $= 5000$ N , $\sigma = 100$ MN / m^2 $= 100$ N / mm^2

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

HOME WORK :

1 – Determine the elongation of the steel bar 1 m long and 1.5 cm^2 cross-sectional area , when subjected to a pull of 1500 kg . Take $E = 2*10^6$ kg / cm^2 ?

2 – A brass rod 2 cm diameter and 1.5 m long is subjected to an axial pull of 4 tonnes . Find the stress , strain and elongation of the rod , if the modulus of elasticity for the brass is $1.0*10^6$ kg / cm^2 ?

3 – A cast iron column has internal diameter of 200 mm , What should be the minimum external diameter so that it may carry a load of 1.6 MN , without the stress exceeding 90 N / mm^2 ?

ELEMENTS OF CUTTING OPERATIONS

Cutting speed : (V_c)

It is the travel of a point in the cutting edge relative to the surface of the cut in unit time .

In lathe works :

$$\boxed{V_c = \frac{\pi DN}{1000}} \quad \textbf{m/min}$$

V_c : Cutting speed

D : Diameter of workpiece

N : Revolution per minut (R.P.M)

Feed "Rate of feed" (S)

It is the travel of the cutting edge in the direction of the feed motion relative to the machined surface in unit time .

Or:The mount of tool advance for each revolution of work (mm/rev)

Deph of cut :(t)

It is thickness of the layer of metal removed in one cut or pass measured in a direction perpendicular to the machined surface

t : Depth of cut

$$\boxed{t = \frac{D_1 - D_2}{2}} \quad \textbf{(mm)}$$

In general,speed and feed depend upon the following factors :

- **i- Type of meterial of workpiece**
- **ii- Type of meterial of cutting tool**
- **iii- Quality if finish desired**
- **iv- Type of coolant used**
- **v- Rigidity of the michine**

Example (1) :

Find the cutting speed for mild steel workpiece with 50 mm diameter and 3000 r.p.m

Solution :

$$V_c = \frac{\pi DN}{1000} = \frac{3.14 * 50 * 3000}{1000} = 47.1 \quad \textbf{m / min}$$

Example (2) :

Find the rev/min for a 50 mm bar to cut at 25 m/min .

Solution :

$$V_c = \frac{\pi DN}{1000} \quad , \quad N = \frac{1000 V_c}{\pi D} = \frac{1000 * 25}{3.14 * 50} = 159 \quad \textbf{r.p.m}$$

Cutting time : (T)

The interval time to complete cutting operation .

$$\boxed{T = \frac{L}{S\ .\ N} * I}$$

T : cutting time

L : Length of cut = length of surface to be cut + Nose raduis [1.5 – 6 min]

I : number of cutting stroke

$$\boxed{I = \frac{Total\ \ depth\ \ of\ \ cut}{depth\ \ of\ \ cut}}$$

$$\boxed{Total\ \ depth\ \ of\ \ cut = \frac{D_1 - D_2}{2}}$$

Example (3) :

Calculate the cutting time required to run workpiece from 50 mm to 26 mm diameter , length of workpiece is 100 mm , cutting speed 25 m / min , feed 2 mm/rev, depth of cut 2 mm ?

Solution :

L = L + Nose raduis = 100 + 4 = 104 mm

$$N = \frac{1000 V_c}{\pi\ D} = \frac{1000 * 25}{3.14 * 50} = 159 \quad r.p.m$$

$$Total\ \ depth\ \ of\ \ cut\ = \frac{D_1 - D_2}{2} = \frac{50 - 26}{2} = 12 \qquad \text{mm}$$

$$I = \frac{Total\ \ depth\ \ of\ \ cut}{depth\ \ of\ \ cut} = \frac{12}{2} = 6$$

$$T = \frac{L}{S\ .\ N} * I = \frac{104}{159 * 2} * 6 = 1.96 \quad \text{min}$$

Example (4) :

Determine the time required to turn a brass bar 50 mm diameter and 100 mm long at a cutting speed of 36 m / min , the feed is 0.4 mm / rev and only one cut is taken ?

Solution :

$$N = \frac{1000 V_c}{\pi\ D} = \frac{1000 * 36}{3.14 * 50} = 229 \quad \textbf{r.p.m}$$

$$T = \frac{L}{S\ .\ N} * I = \frac{104 * 60}{229 * 0.4} * 1 = 68.12 \quad \textbf{seconds}$$

Example (5) :

Determine the cutting speed and machining time per cut when the workpiece having 35 mm diameter is rotated at 200 r.p.m , the feed given is 0.2 mm / rev , and the length of cut is 60 mm ?

Solution :

$$V_c = \frac{\pi DN}{1000} = \frac{3.14 * 35 * 200}{1000} = 21.98 \quad \textbf{m / min}$$

$$T = \frac{L}{S \ . \ N} * I = \frac{64}{0.2 * 200} * 1 = 1.6 \quad \textbf{min}$$

HOME WORK :

1 – A brass bar of 20 mm diameter and 150 mm long is turned with 480 r.p.m and feed of 10.25 mm / rev , Determine the cutting time for this operation .

2 – Find out the rotation speed required to machine workpiece of 200 mm with cutting speed 45 m / min ?

3 –Workpiece of mild steel was turned on central lathe machine under the following conditions :

Depth of cut = 1.5 mm , feed rate = 5 mm / rev , No. of strock = 6
length of w.p. = 70 mm , Cutting speed = 10 m / min ,
Determine the cutting time

Cutting force in turning : (Fz)

$$\boxed{Fz = K \ . \ A}$$

Fz **: Cutting force**

K : coefficient of cutting force (kg/cm^2) or (N/mm^2)
K : coefficient of metal strength for cutting .
K : coefficient of specific pressure .

$$\boxed{A = t \ . \ S}$$

A : cross- sectinal area of chip (mm^2)
t : depth of cut (mm)
S : feed or feed rate (mm / rev)

Twisting moment in turning : (M_T)

$$\boxed{M_T = Fz \ . \ \frac{D}{2}}$$

M_T : twisting moment

Fz **: cutting force**

$$\boxed{D = \frac{D_1 + D_2}{2}}$$

D_1 : Diameter of workpiece before turning .
D_2 : Diameter of workpiece after turning .

Power cutting : (P_{cut})

$$P_{cut} = \frac{Fz \ . \ Vc}{60 * 1000}$$

when (Fz) in (N) , (Vc) in m/min , (P_{cut}) in (KW)

$$P_{cut} = \frac{Fz \ . \ Vc}{6120}$$

when (Fz) in (kg) , (Vc) in m/min , (P_{cut}) in (KW)

$$Motor \ \ power = \frac{P_{cut}}{\eta}$$

η : efficiency of motor

$$P_{h.p} = \frac{2\pi NT}{45 * 10^6} \qquad \Rightarrow \qquad T = \frac{P_{h.p} * 45 * 10^6}{2\pi N}$$

$$P_{kw} = \frac{2\pi NT}{60 * 10^6} \qquad \Rightarrow \qquad T = \frac{P_{kw} * 60 * 10^6}{2\pi N}$$

Heat generation during machining : (Q)

$$Q = \frac{Fz \ . \ Vc}{J}$$

Q **: amount of heat generated in cutting (W)**
Fz **: cutting force (N)**
Vc **: cutting speed (m / sec)**
J **: Joul constant = 1019 Nm/J [mechanical equivelant]**

$$T = C_1 \ . \ Vc^z$$

T : temperture generated
C_1 : coefficient depending upon the machining conditions
Z : exponent [0.26 – 0.72]

Example (6) :
Calculate the cutting power which is required to turn workpiece from 50 mm diameter to 44 mm diameter under the following conditions :
The feed = 0.2 mm/rev , cutting force = 200 kg , η = 0.70 , number of revolution for machin 200 r.p.m , then find :

i- Motor power .
ii-Amount of heat generated in cutting .
iii- Twisting moment in turning .

Solution :

$$V_c = \frac{\pi\, DN}{1000} = \frac{3.14 * 50 * 200}{1000} = 31.4 \quad \textbf{m / min}$$

$$P_{cut} = \frac{Fz\ .\ Vc}{6120} = \frac{200 * 31.4}{6120} = 1.02 \quad \textbf{Kw}$$

$$Motor\ \ power = \frac{P_{cut}}{\eta} = \frac{1.02}{0.70} = 1.46 \quad \textbf{Kw}$$

$$Q = \frac{Fz\ \ .\ Vc}{J} = \frac{200 * 31.4}{1019} = 1.02 \quad \textbf{Kw}$$

$$D = \frac{D_1 + D_2}{2} = \frac{50 + 44}{2} = 47 \quad mm$$

$$M_T = Fz\ .\ \frac{D}{2} = 200 * \frac{47}{2 * 1000} = 47 \quad N.m$$

Example (7) :

Calculate the cutting power (Kw) which is required to turn a shaft made of structural steel under the following conditions :
Depth of cut = 5 mm , the feed = 0.4 mm/rev , number of revolution for machine = 190 r.p.m , the metal strength for cutting =160 kg/mm^2 , diameter of the shaft = 50 mm , Then find :
i- Electrical motor power if you know that the efficiency of machine 70% .
ii-Amount of heat generated in cutting (Kw) .

Solution :

$$V_c = \frac{\pi\, DN}{1000} = \frac{3.14 * 50 * 190}{1000} = 29.845 \quad m/\min = 0.497 \quad m/\sec$$

A = t . S = 5 * 0.4 = 2 mm^2

$$Fz = K\ .\ A = 160 * 2 = 320\ \ kg = 3200 \quad N$$

$$Q = \frac{Fz\ \ .\ Vc}{J} = \frac{3200 * 0.497}{1019} = 1.56 \quad \text{W}$$

$$P_{cut} = \frac{Fz\ .\ Vc}{60 * 1000} = \frac{3200 * 29.845}{60 * 1000} = 1.56 \quad Kw$$

$$Motor\ \ power = P_{elec} = \frac{P_{cut}}{\eta} = \frac{1.59}{0.70} = 2.27 \quad Kw$$

Example (8) :

In machining operation , to turn a w.p. from 50 mm diameter to 40 mm diameter with length 70 mm by using feed of 0.2 mm/rev , rotational speed 200 r.p.m , if the strength of the material is 600 kg/mm^2 and the efficiency of machine is 0.70 .

Find :

1 – the depth of cut (t) .

2 – the cross-sectional area of chip removal (A) .

3 – Cutting power (P_{cut}) .

4 – twisting moment in turning operation (M_T) .

5 – Cutting time (T) .

6 – motor power (P_{elec}) .

7 – heat generated (Q) .

Solution :

$$t = \frac{D_1 - D_2}{2} = \frac{50-40}{2} = 5 \quad \text{mm}$$

A = t . S = 5 * 0.2 = 1 mm^2

$$Fz = K \ . \ A = 600 * 1 = 600 \quad N$$

$$V_c = \frac{\pi\, DN}{1000} = \frac{3.14*50*200}{1000} = 31.4 \quad mm/\min$$

$$P_{cut} = \frac{Fz \ . \ Vc}{6120} = \frac{600*31.4}{6120} = 3.07 \quad Kw$$

$$D = \frac{D_1 + D_2}{2} = \frac{50+40}{2} = 45$$

$$M_T = Fz \ . \ \frac{D}{2} = 600 * \frac{45}{2} = 13500 \quad N.m$$

L = L + Nose raduis = 70 + 4 = 74 mm

$$I = \frac{Total \ \ depth \ \ of \ \ cut}{depth \ \ of \ \ cut} = \frac{12}{2} = 6$$

$$I = \frac{\frac{D_1 - D_2}{2}}{t} = \frac{\frac{50-40}{2}}{5} = 1$$

$$T = \frac{L}{S \ . \ N} \ * \ I = \frac{74}{0.2*200} * 1 = 1.85 \quad \min$$

$$Motor \ \ power = P_{elec} = \frac{P_{cut}}{\eta} = \frac{3.07}{0.70} = 2.149 \quad Kw$$

$$Q = \frac{Fz \ \ . \ Vc}{J} = \frac{600*31.4}{1019} = 18.48 \quad \text{W}$$

HOME WORK :

1 – For the following conditions to machining a w.p. :

$V_c = 31.4$ m / min , depth of cut = 1.5 mm , diameter = 50 mm , K= 6000 N / mm^2

Cross-sectional area = 0.3 mm^2 , $\eta = 0.7$,

Find :

a – feed rate (s)   b – motor power (Pm)

c – heat generated during machining (Q)   d – cutting time (T) .

2 – A workpiece was machined under the following conditions :

Depth of cut (t) = 1.5 mm , feed rate (s) = 5 mm / min ,

initial diameter (D_1) = 68 mm , Final diameter (D_2) = 50 mm ,

length of w.p. = 70 mm , cutting speed (Vc) = 10 m / min

Find :

a – revolution per minut (N)   b - cutting time (T)

c – cross-sectional area of chip removal (A) .

3 – Do as required in the table below :

Serial of operation	*Given machining conditions data*	*Required data*
1	*S=0.2 mm/rev, F= 200 kg , d=44 mm, N= 200 r.p.m , M_T = 47 N.m*	*Cutting speed , cutting power , heat generated*
2	*D_1=50 mm , D_2 = 40 mm , feed rate=0.4 mm/min , Vc= 31.4 m/min , F= 200kg , K=600 kg/mm^2 , L=100mm*	*Feed (s) , depth of cut (t) , cutting time , cutting power , twisting moment in turning , cross-sectional area of chip removal*
3	*Cutting speed = 31.4 mm/min , depth of cut = 1.5 mm , d = 50 mm , K= 6000 N/mm^2 , A=0.3 mm^2 , η =0.7*	*Feed rate , motor power , heat generated , total depth of cut , twisting moment in turning*

STUDY CASE

Assuming that a machining conditions are constants as in below :

Intial diameter = 70 mm , final diameter = 30 mm , length of workpiece = 96 mm

Depth of cut = 2 mm , feed = 0.2 mm/rev .

If the rotational speed (N) of the lathe central machine has the values :

(100 , 200 , 400 , 800 , 1600 r.p.m)

Required :

Draw a chart to explain the relationship between the cutting speed and the cutting time.

FASTENING

Different parts of structures , machines or other engineering products are joined together by means of fastenings .

Types of fastenings :

1 – Temporary fastenings : such as : bolts , nuts , screws , keys and pins .

2 – Permanent fastenings : such as : rivets , weldings …. etc .

Screwed joints :

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

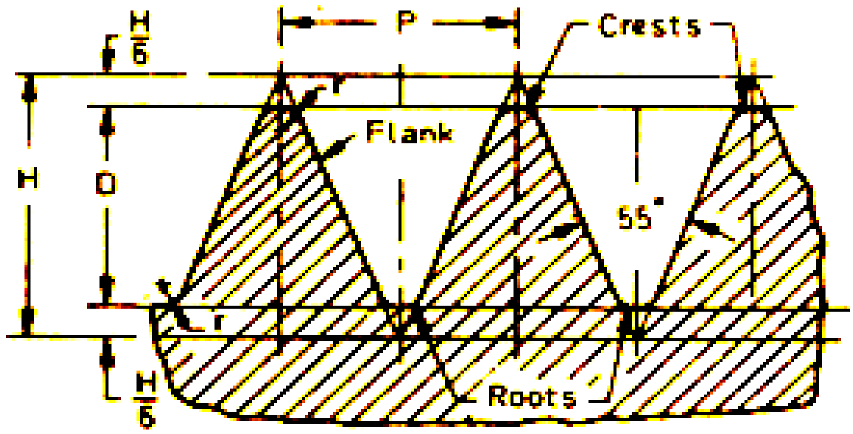
Screwed joints are widely used where the machine parts are required to be readily connected or disconnected without damage to the machine or the fastening .

Forms of screw threads :

The following are the various forms of screw threads:

1. ***British standard whiteworth (B . S . W .) thread :***

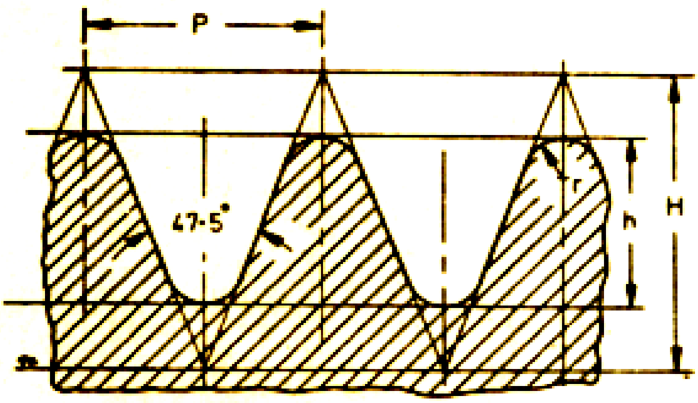
   These threads are found on bolts and screwed fastenings for several purpose , and used for steel and iron pipes and tubes carrying fluids, in external pipe threading , the threads are specified by the bore of the pipe .



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

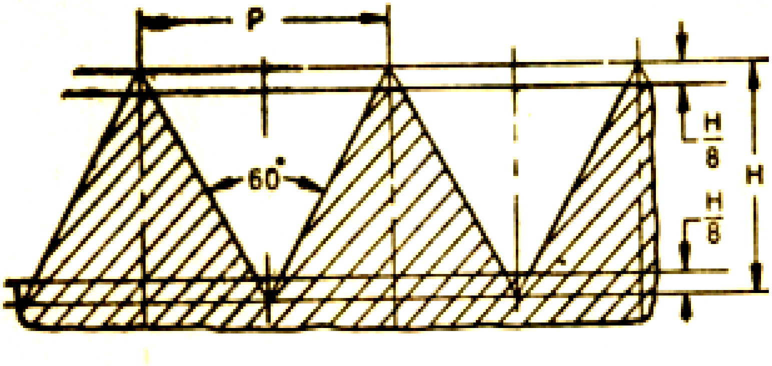
2. ***British Association (B.A.) thread :***

   These threads are used on screws for precision work .



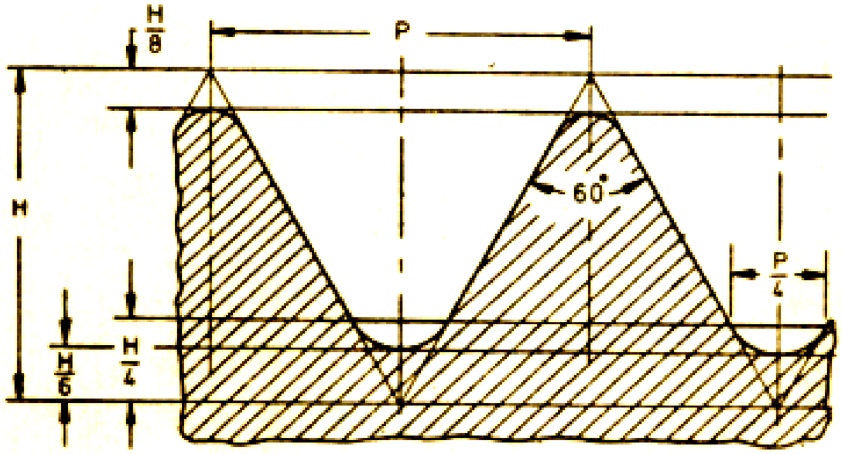
$H = 1{\cdot}13634P$; $h = 0{\cdot}6P$; $r = 0{\cdot}18083P$

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



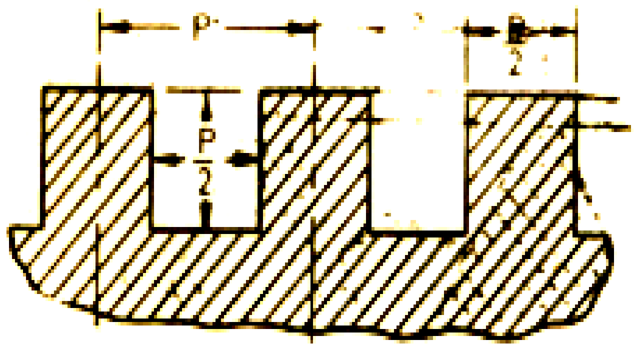
$H = 0{\cdot}866P$

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



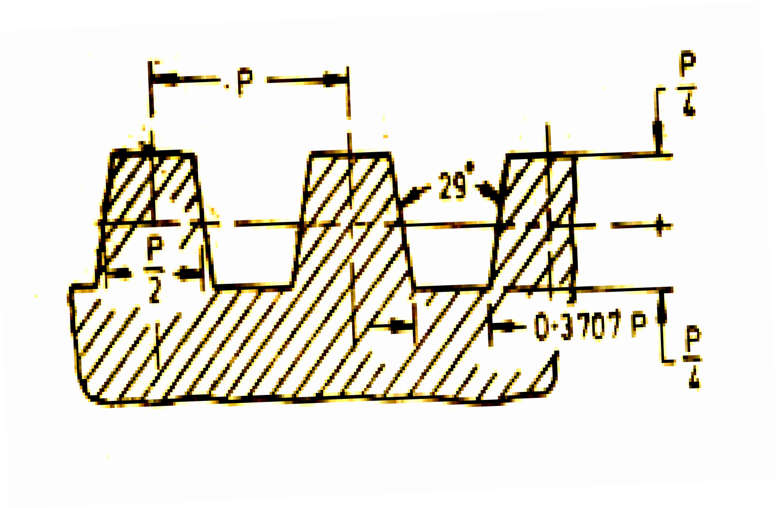
$H = 0{\cdot}866\ P$

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



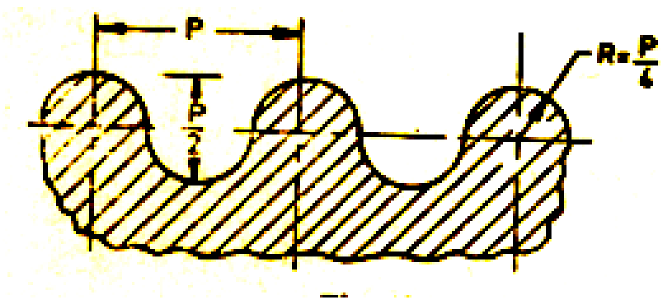
6. Acme thread :

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



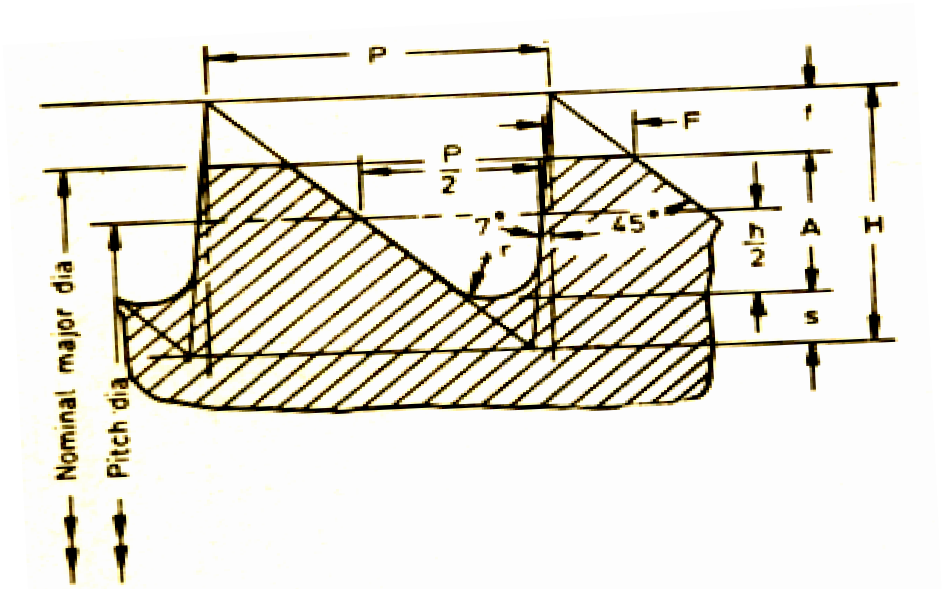
7. Knuckle thread :

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



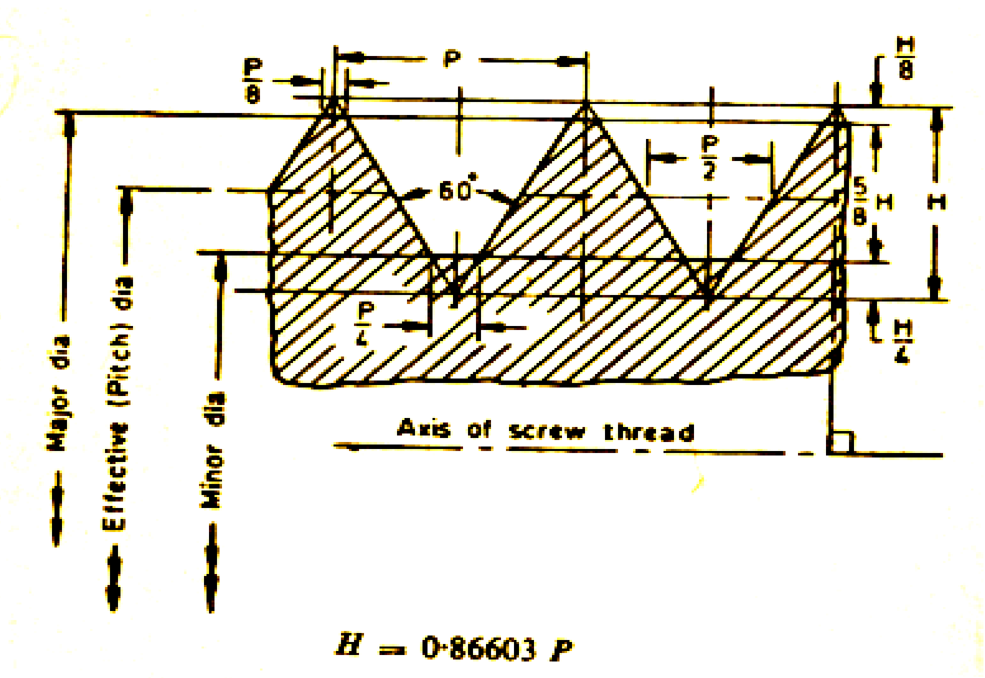
8. Buttress thread :

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

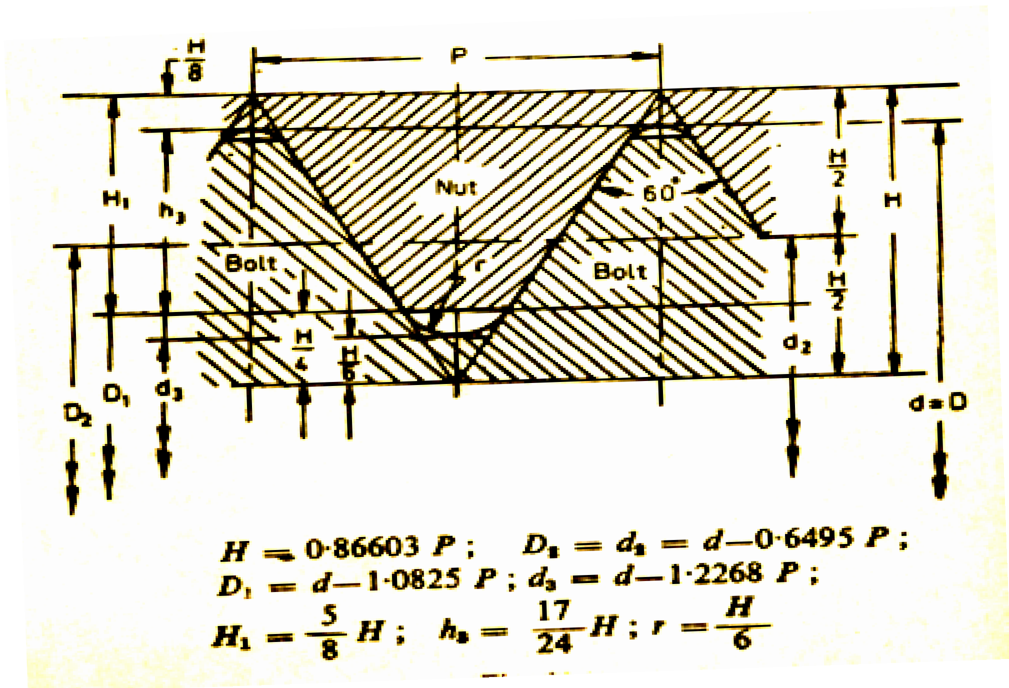


$H = 0{\cdot}89064\ P$;
$A = 0{\cdot}50286\ P$;
$f = 0{\cdot}24532\ P$;
$s = 0{\cdot}13946\ P$;
$F = 0{\cdot}27544\ P$;
$r = 0{\cdot}12055\ P$.

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

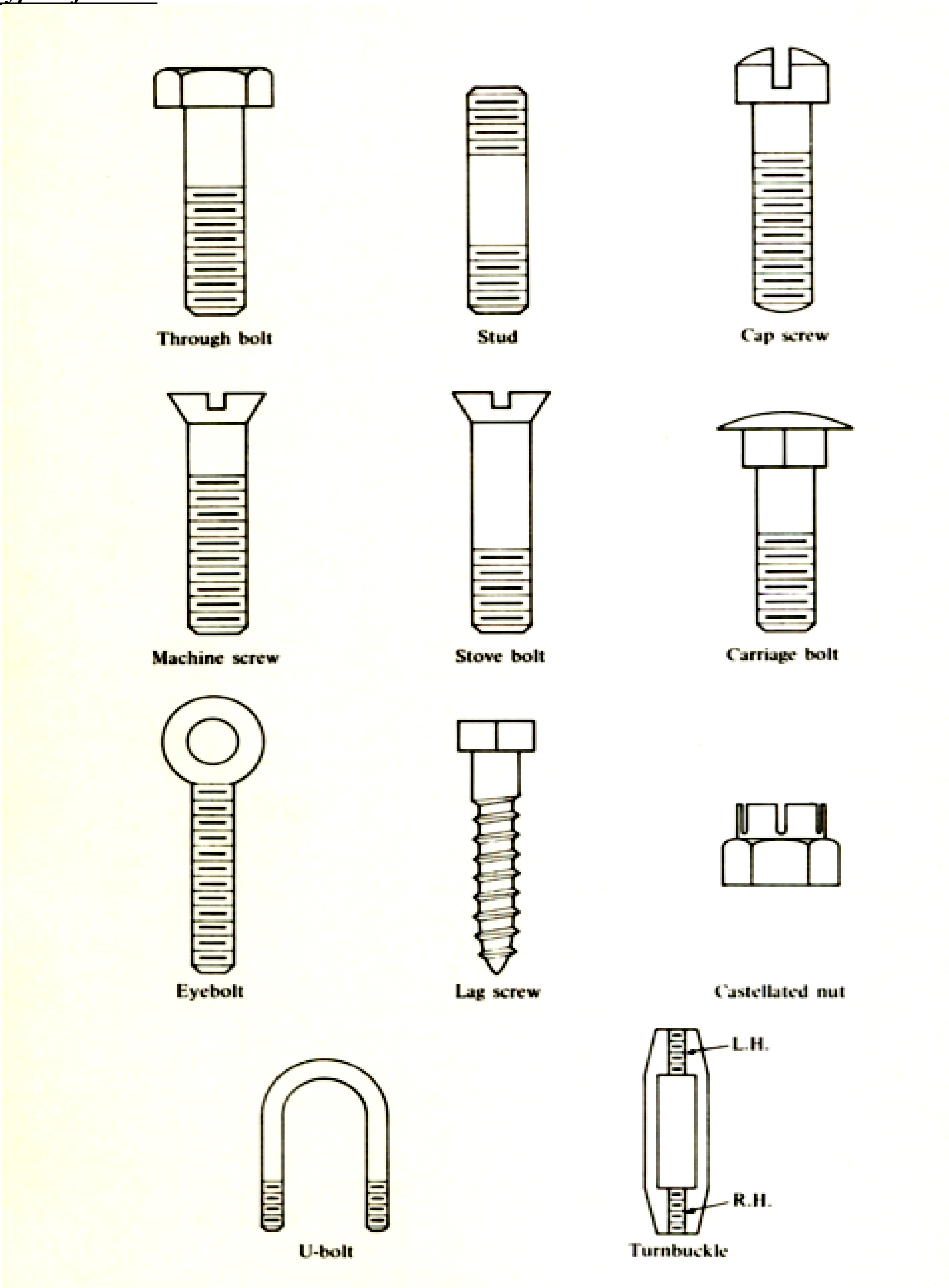


The basic profile of the thread



The design profile of the nut and bolt

Types of bolts :



Stresses in screwed fastening due to static loading

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

1. ***Intrnal stresses due to screwing up forces.***

a -Tensile stress due to stretching of the bolt.

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

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

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

d = Nominal diameter of bolt , in (mm)

$P_i = 2840\ d$ ***(kg)*** ***If (d) in (cm)***

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

P = Permissible stress X Cross-sectional area at bottom of the thread (i .e. stress area)

The stresses area may be obtained from Table 10-1 or it may be found by using the relation,

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

d_p ***= Pitch diameter.***

dc = Core or minor diameter.

b -Torsional shear stress caused by the frictional resistance of the threads during its tightening

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

We know that :

$$\frac{T}{J} = \frac{\sigma s}{r}$$

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

where:

σs ***= Torsional shear stress .***

T = Torque applied .

dc ***= Minor diameter of the thread .***

(c) Shear stress across the threads

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

$$\sigma_s = \frac{P}{\pi \cdot dc \cdot b \cdot n}$$

b = Width of the thread section at the root.

The average thread shearing stress for the nut is :

$$\sigma_s = \frac{P}{\pi \cdot d \cdot b \cdot n}$$

d = Major diameter.

d –Crushing stress on thread :

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

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

where:

d = Major diameter

d_c = Minor diameter

n = No. of threads in engagement.

e - Bending stress :

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

$$\sigma = \frac{x \cdot E}{2 \cdot l}$$

where:

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

l = Length of the shank of the bolt

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

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

Designation	Pitch(mm)	Major or Nominal diameter Nut &Bolt (d=D)mm	Effective or pitch diameter Net & bolt (dp) mm	Minor or core diameter (dc)mm		Depth of Thread (bolt) mm	Stress area mm^2
				bolt	nut		
M1	0.25	1.000	0.838	0.693	0.729	0.153	0.460
M2	0.40	2.000	1.740	1.509	1.567	0.245	2.07
M4	0.70	4.000	3.545	3.141	3.242	0.429	8.78
M5	0.80	5.000	4.480	4.019	4.134	0.491	14.2
M6	1.00	6.000	5.350	4.773	4.918	0.613	20.1
M7	1.00	7.000	6.350	5.773	5.918	0.613	28.9
M8	1.25	8.000	7.188	6.466	6.647	0.707	36.6
M10	1.50	10.00	9.026	8.160	8.876	0.920	58.3
M12	1.75	12.00	10.863	9.858	10.106	1.074	84.0
M14	2.00	14.00	12.701	11.546	11.835	1.227	115
M20	2.50	20.00	18.376	16.933	17.294	1.534	245
M30	3.50	30.00	27.727	25.706	26.211	2.147	561
M42	4.50	42.00	39.077	36.416	37.129	2.760	1.104
M52	5.00	52.00	48.752	45.795	49.587	3.067	1.755
M60	5.50	60.00	56.428	53.177	54.046	3.374	2.360

Example (1) :

Determine the safe tensile load for a bolt of M 30, assuming a safe tensile stress of 420 kg/cm*.

Solution :

Given : Size of bolt = M 30

$\therefore$ ***Major diameter of bolt :*** $d = 30$ ***mm***

Safe ternile stress, $\sigma_s = 420\ kg/cm^2$

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

$= 561\ mm^2 = 5.61\ cm^2$

$\therefore$ ***Safe tensile load = Stress area*** $\times\ \sigma_t$

$= 5.61 \times 420 = 2356.2$ ***kg***

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

Example (2) :

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

Solution :

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

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

$d_e = 11.456\ mm = 1.1456\ cm$

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

We know that, initial tension in the bolt,

$P = 284\ d = 284 \times 14 = 3976$ ***kg***

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

$$3976 = \frac{\pi}{4} \times 1.1456^2 \times \sigma_t$$

$$\sigma_t = \frac{3976 \times 4}{\pi \times 1.1456^2} = 3859.31 \quad kg/cm^2$$

2. *stress due to external forces.*

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

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

We shall now discuss these stresses in detail as below:

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.

Let P = External load applied

d_e = Root or core diameter of the thread

σ_t = Permissible tensile stress for the bolt material.

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

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

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

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

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

d_c = 0.84 d   d = Nominal diameter of bolt

2. Shear stress

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

Let : *P_s = Shearing load carried by the bolt*
d = Major diameter of the bolt,
n = Number of bolts.

Then :

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

3. Combined tension and shear stress

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

Maximum principal shear stress,

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

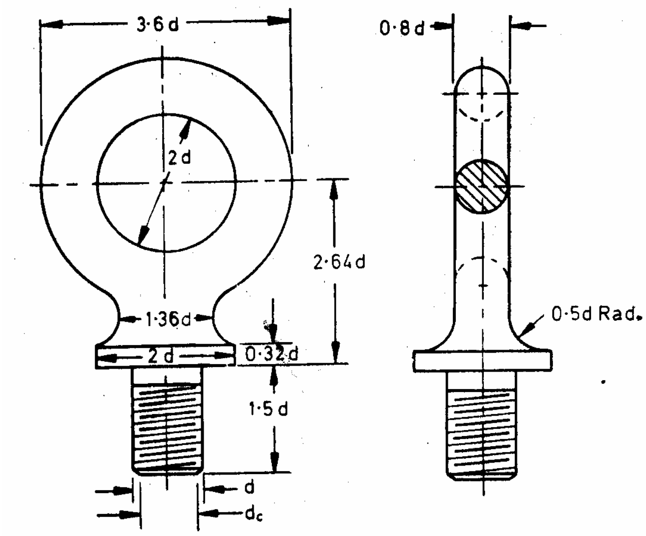
and minimum principal tensile stress.

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

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

Example (3) :

An eye bolt is to be used for lifting a load of 60 kN. Find the nominal diameter of the bolt if the tensile stress is not to exceed 100 N/mm^2. Assume coarse threads.



Solution :

Given: Load to be lifted , P = 60 kN
= 60 X 10^3 N

Permissible tensile stress , σ_t = 100 N/mm^2

Let d = Nominal diameter of the bolt
d_c = Core diameter of the bolt,
Using the relation

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

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

Example (4) :

Two shafts are connected by means of a flange coupling to transmit torque of 250 kg.cm. The flanges of thecouplaig are fastened by four bolts of the same material at a radius of 3 cm.

Find the size of the bolts if the allowable shear stress for the bolt material is 300 kg/cm^2.

Solution:

Given: Torque transmitted by a flange coupling , T= 250 kg.cm , Number of bolts, n = 4
Pitch radius , R_P = 3 cm , Allowable shear stress , σ_s = 300 kg/cm^2
We know that the shearing load carried by flange coupling,

$$P_s = \frac{T}{R_p} = \frac{250}{3} = 83.33 kg$$

Size of the bolt , Let d_c= Core diameter of the bolt

$$P_s = \frac{\pi}{4} d_c{}^2 \sigma_s \times n$$

$$\therefore d_c = \sqrt{\frac{4P}{\pi\,\sigma_s \times n}} = \sqrt{\frac{483}{\pi \times 300 \times 4}} = \sqrt{0.0883} = 0.298\,cm$$

From Table (1) , we find that the standard core diameter of the bolt is 3.141 mm and the corresponding size of the bolt is M 4.
These stresses should not exceed the safe permissible values of the stresses.

Homework :

1 –Determine the safe tensile load for bolts of M20 and M42 . Assume that the bolts are not initially stressed and take the safe tensile stress as 2000 kg/cm^2 .

2 - An eye bolt carries a tensile load of 2000 kg . Find the size of the bolt , if the tensile stress is not to exceed 1000 kg/cm^2 . Also draw a neat proportioned figure for the bolt

3 –An cylinder is 30 cm in diameter and the steam pressure is 7 kg/cm^2 . If the cylinder head is held by 12 studs , find the size , assume safe tensile stress as 280 N/cm^2 .

4 – Find the size of 14 bolts required for C . I .steam engine cylinder head . The diameter of the cylinder is 400 mm and the steam pressure is 0.12 N / mm^2.

KEYS

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

Uses of keys :

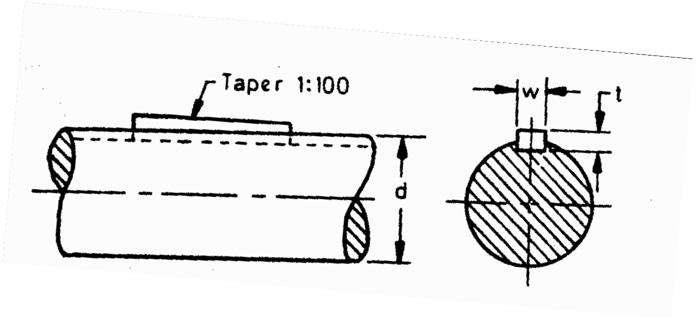
Keys are used as temporary fastening and are subjected to considerable crushing and shearing stresses .

A key way :

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

Types of keys :

1 – Sunk keys :



a – Rectangular sunk key

width of key , w = d / 4

thickness of key , t = ⅔ . w

d = diameter of the shaft

= diameter of the hole in hub

b –Squar sunk key :

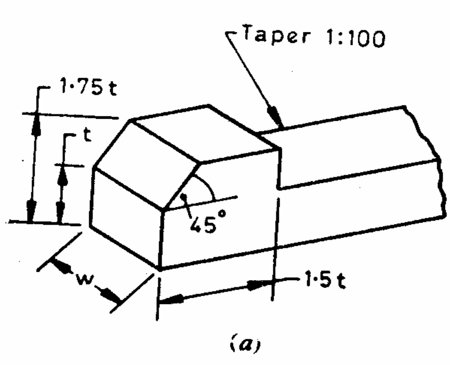
w = t = d / 4

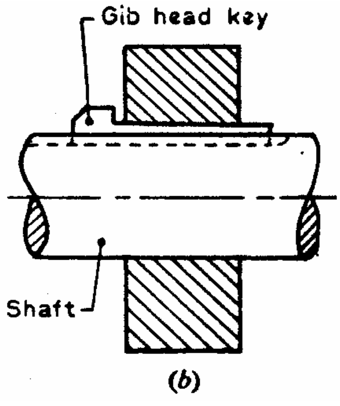
c –Parallel sunk key :

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

d – Gib-head key :

it is rectangular sunk key with a head at one end .





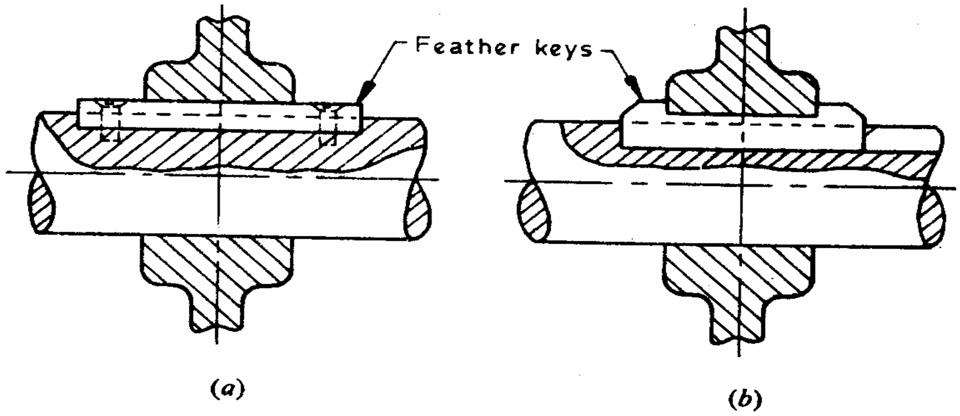
a - Gib-head key ***b - Gib-head key use***

width , w = d / 4

thickness of key , t = ⅔ . w = d / 6

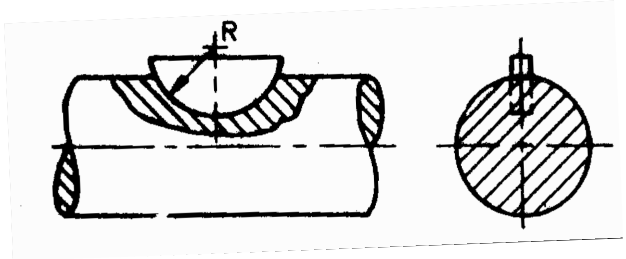
e-Feather key :

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



f – Wood ruff key :

this key is largly used in machine tool and automobile construction .

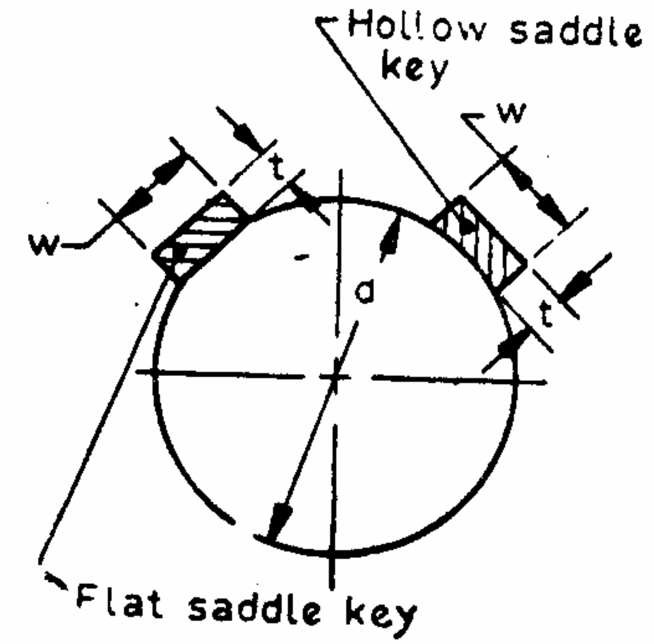


2 – Saddle keys :

the saddle keys are of two parts :

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

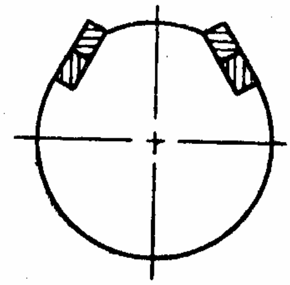
b – hollow saddle key : used as a temporary fastening in fixing and setting eccentrics cams … etc .



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

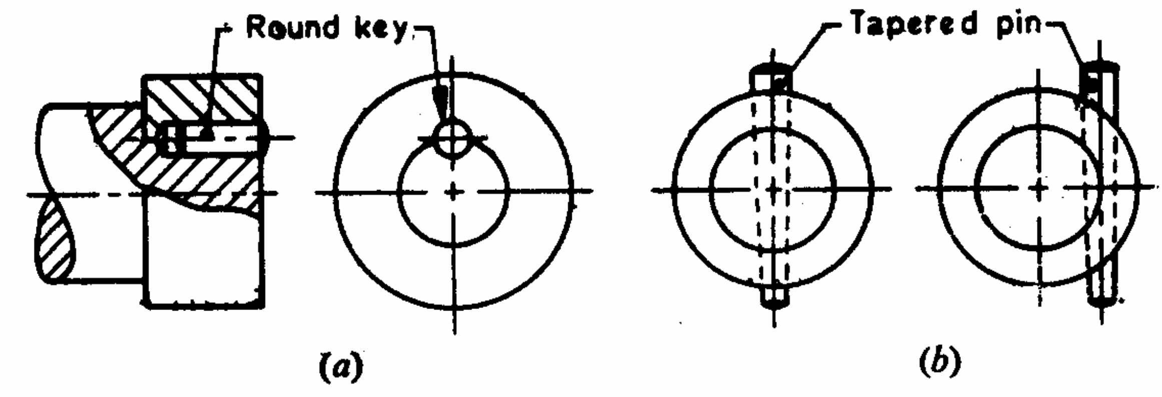
3 – Tangent keys :

these are used in large heavy duty shafts .



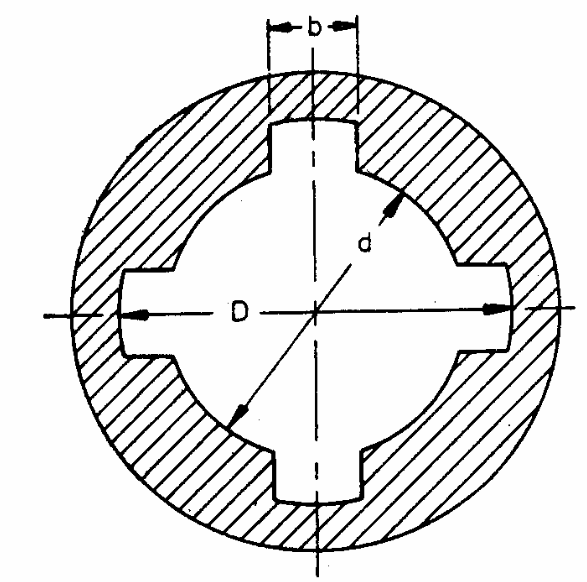
4 – Round keys :

these are usually used for low power drives .



5 – Splines keys :

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



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

STRENGTH OF *SUNK KEYS :*

T : Torque transmitted by the shaft

F: Tangential force acting at the circumference of the shaft .

d : diameter of the shaft

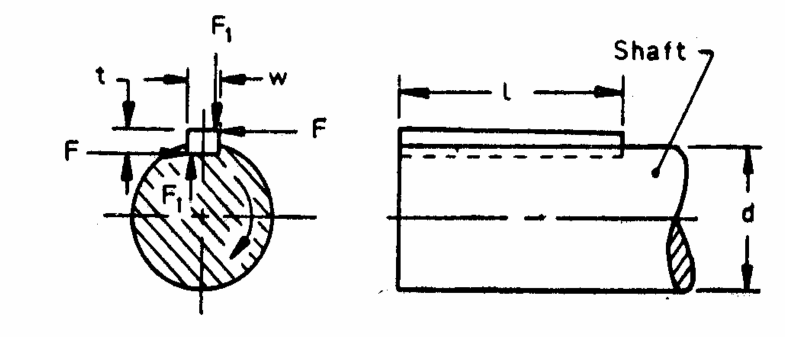
l : length of the key

w : width of the key

t : thickness of the key

σ_s : shear stress for the material of key

σ_c : crushing stress for the material of key



In shearing :

F = area resisting shearing X shear stress

$$= l \,.\, w \,.\, \sigma_s$$

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

In crushing :

F = area resisting crushing X crushing stress

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

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

The key is equally strong in shearing and crushing , if :

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

when w = t in square key , then :

$$\sigma_s \,.\, l \,.\, w \,.\, (d/2) = \sigma_c \,.\, l \,.\, (w/2) \,.\, (d/2)$$

$$\boxed{\therefore\ \sigma_s = 2 \,.\, \sigma_c}$$

The length of key :

To find the length of the key to transmit full power of the shaft , the shearing strength of the shaft is equal to the torsional shear strength of the shaft :

we know that the shearing strength of the key :

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

and the torsional shear strength of the shaft :

$$T = (\pi/16) \,.\, \sigma_{s1} \,.\, d^3 \qquad (\text{taking } \sigma_{s1} = \text{shear stress for the shaft material})$$

$$l \,.\, w \,.\, \sigma_s \,.\, (d/2) = (\pi/16) \,.\, \sigma_{s1} \,.\, d^3$$

$$l = \frac{\pi}{8} \times \frac{\sigma_{s1} \times d^3}{w \times \sigma_s}$$

$$= \frac{\pi \,.\, d}{2} \times \frac{\sigma_{s1}}{\sigma_s} \qquad (\text{taking} \quad w = \frac{d}{4})$$

$$= 1.571\ d \times \frac{\sigma_{s1}}{\sigma_s}$$

***Special case** : when the key material is same as that of the shaft , then :* $\sigma_s = \sigma_{s1}$

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

Example (1) :

Design the rectangular key for a shaft of 50 mm diameter , the shearing and crushing stresses in the key are limited to 420 kg/cm^2 and 700 kg/cm^2 .if the width of key is 16 mm and thickness of the key is 10mm

Solution :

Diameter of the shaft , d=50 mm = 5 cm

Shear stress in the key, σ_s= 420 kg/cm^2 , Crushing stress in the key, σ_c =700 kg/cm^2

shearing of the key :

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

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

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

$$l = \frac{\pi . \ d^2}{8w} = \frac{\pi \times 5^2}{8 \times 1.6} = 6.14 \quad cm$$

crushing of the key :

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

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

$$l = \frac{\pi}{4} \times \frac{\sigma_s \times d^2}{t \times \sigma_c} = \frac{\pi \times 420 \times 5^2}{4 \times 1 \times 700} = 11.8 \, cm \approx 12 \ cm$$

Example (2) :

A 20 h.p. , 960 r.p.m motor has a mild steel shaft of 4 cm diameter and the extension being 7.5 cm . The permissible shear and crushing stresses for the mild steel key are 560 kg/cm^2 and 1120 kg/cm^2 . Design the keyway in the motor shaft extension .

Solution :

$$T = \frac{P \times 4500}{2 \pi N} = \frac{20 \times 4500}{2 \times 3.14 \times 960} = 14.92 \quad kg.m = 1492 \quad kg.cm$$

Design of keyway :

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

$$1492 = 7.5 \times w \times 560 \times \frac{4}{2}$$

$$w = \frac{1492 \times 2}{7.5 \times 560 \times 4} = 0.17 \ cm \quad or \quad 1.7 \ mm$$

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

w = (d / 4) = 4 / 4 = 1 cm or 10 mm , since $\sigma_c = 2 \sigma_s$

therefore , a square key is adopted , then

t = w …….. t = 10 mm .

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

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

<u>Solution :</u>

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

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

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

$$l = 88.81 \quad mm$$

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

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

$$l = 121.8 \quad mm$$

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

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

A flat key 10 mm wide , 8 mm thickness and 75 mm long is required to transmit a torque of 10^6 N .mm from a 50 mm diameter shaft , investigate to determine whether the is sufficient , use a design stress in shear of 50 N / mm² and in crushing of 130 N / mm², if the length is not sufficient redesign the length of the key ?

<u>Solution :</u>

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

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

$$l = 80 \quad mm$$

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

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

$$l = 76.9 \quad mm$$

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

Example (5) :

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

Solution :

$$\because F_s = F_c$$

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

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

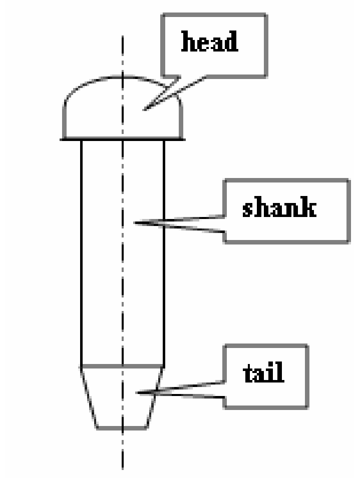
$$\because w = t$$

$$\therefore 2\sigma_s = \sigma_c$$

Homework :

1- ***Design a key to transmite 40 kw power by a shaft of 50 mm diameter which rotates 1200 r.p.m , if the permissible shear and crushing stresses are 35 N/mm^2 and 65 N/mm^2 respectively , take the key length 30 mm .***
2- ***A belt pulley is fastened to 80 mm diameter shaft transmitting 75 kw at 200 r.p.m by means of key 22 mm width and 14 mm thichness.***
   Determine the length of the key
   Take : σ_s = 40 N/mm^2 , : σ_c = 100 N/mm^2
3- ***A shaft of 50 mm diameter is used to transmite power 20 kw at 200 r.p.m , if the width of key is 10 mm , and the thickness of key is equal to width .***
   Findout the length of the key .
   Take σ_s = 50 N/mm^2 , : σ_c = 130 N/mm^2
4- ***A shaft of 80 mm diameter transmites power at maximum shear stress of 630 kg/cm^2 Find the length of a 20 mm wide key required to mount a pulley on the shaft so that the stress in the key doesnot exceed 420 kg/cm^2 .***
5- ***A shaft 30 mm diameter is transmitting power at a maximum shear stress of 800 kg/cm^2 if a pulley is connected to the shaft by means of a key.***
   Find the dimensions of the key so that the stress in the key is not to exceed 500 kg/cm^2 and the length of the key is (4) times the width .

RIVETED JOINTS



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

Uses of rivets :

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

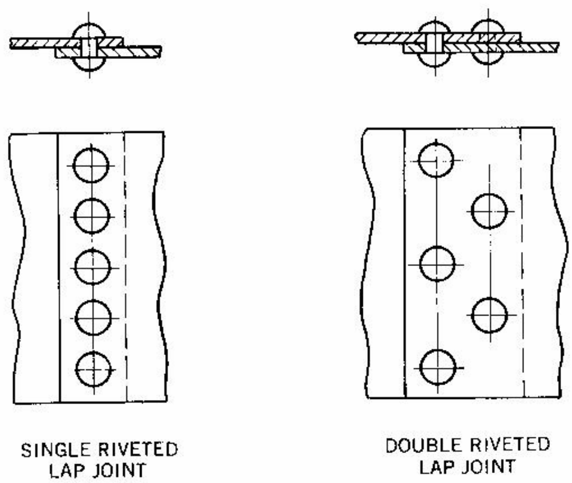
Types of riveted joints :

A – Lap joint :

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

1 - Single riveted joint

2 – Double riveted joint



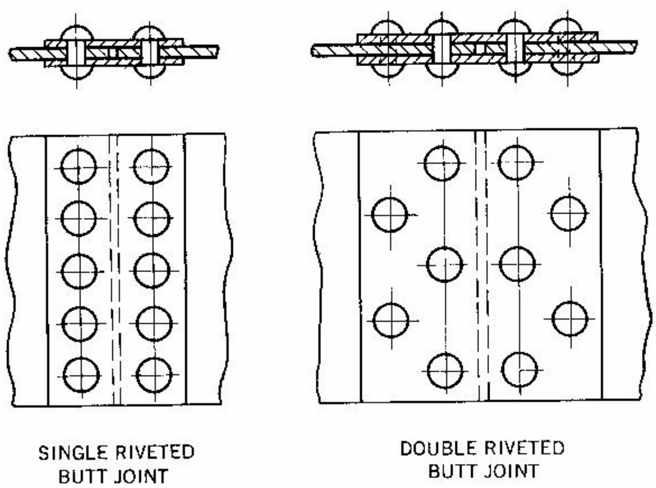
(A) LAP JOINTS

B – Butt joint :

A butt joint is that in which the main plates are kept in alignment butting (i.e touching)each other and a cover plate (i.e strap) is placed either on one side or on both sidesof the main plates , the cover plate is then riveted together with the main plate .

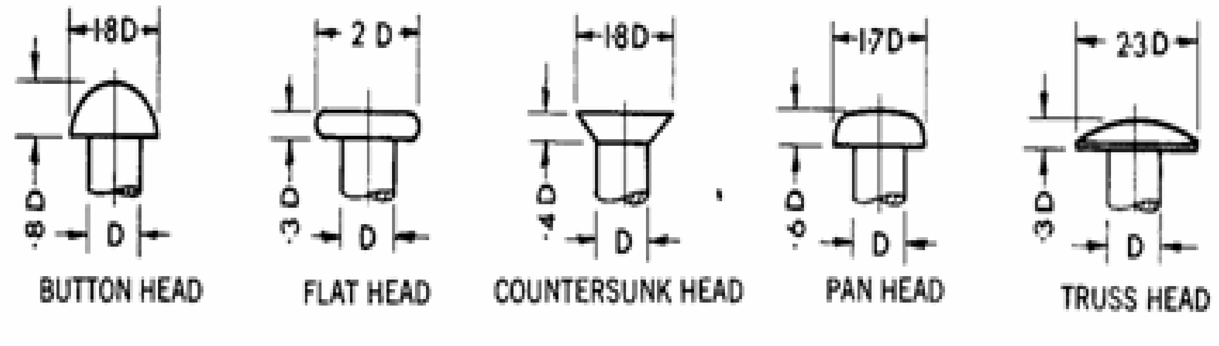
1 - Single riveted joint

2 – Double riveted joint



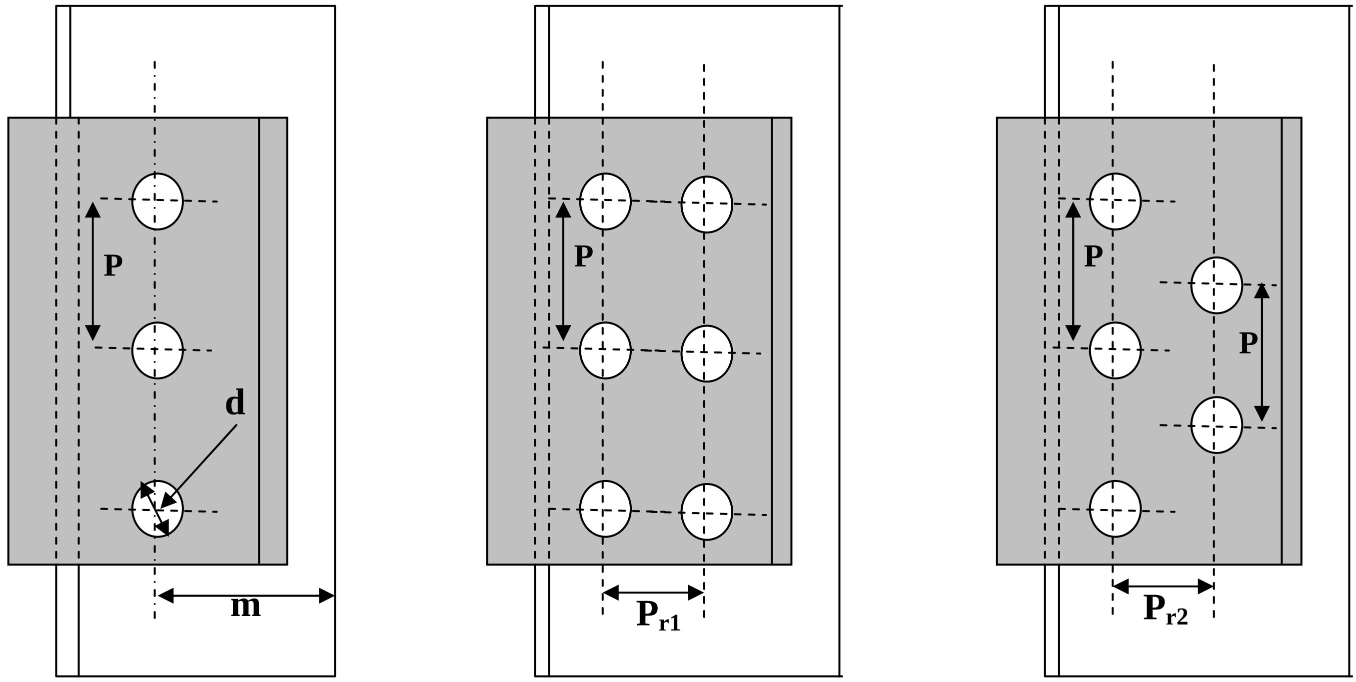
(B) BUTT JOINTS

Types of Standard rivet heads:



(A) STANDARD HEAD TYPES AND APPROXIMATE PROPORTIONS

The main dimenssions of riveted joints :



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

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

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

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

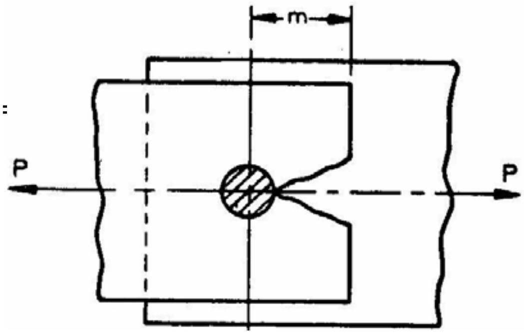
$\mathrm{Pr}_2 = 2d$

Failer of riveted joints :

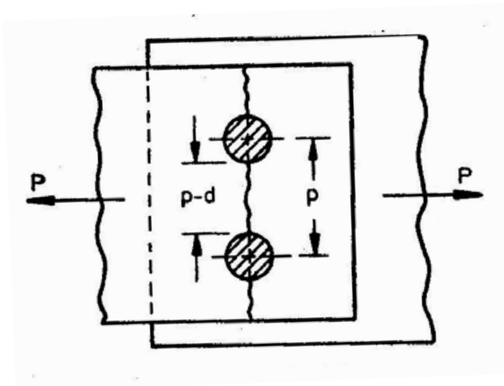
A riveted joints may fail in the following ways :

1 – Tearing of the plate at an adge :

This case can be avoided by keeping the margin , m



2 - Tearing of the plate across a row of rivets :



P : pitch of the rivets

d : diameter of the rivets

t :thickness of the plate

σ_t :tensile stress for the plate material

A_t : tearing area per pitch length

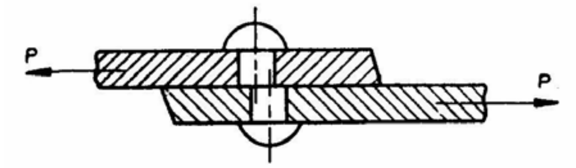
$A_t = (p - d)\ t$

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

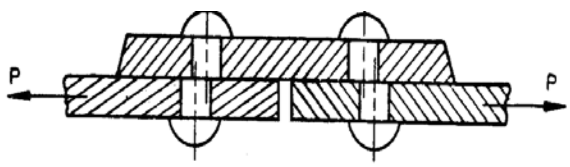
$$P_t = \sigma_t \, . \, A_t = \sigma_t \, (p - d) \, . \, t$$

3 – Shearing of the rivets :

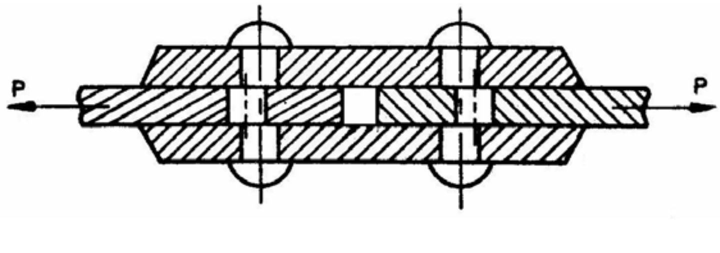
a – shearing of a rivet in a lap joint .



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



c - shearing of a rivet in double cover butt joint



d : diameter of the rivet

σ_s : safe permissible shear stress for the rivet material

n : number of rivets per pitch length

As : shearing area

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

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

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

$$Ps = \frac{\pi}{4}\times d^2\times\sigma_S\times n \qquad \textit{(in single shear)}$$

$$Ps = 2\times\frac{\pi}{4}\times d^2\times\sigma_S\times n \qquad \textit{(in double shear)}$$

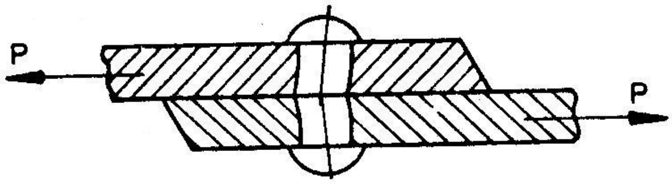
4 – Crushing of the rivets :

d : diameter of the rivet

t : thickness of the plate

σ_c : safe permissible crushing stress for the rivet material

n : number of rivets per pitch length under crushing .



$$Ac = d\times t$$

$$\therefore\ total\ \ crushing\ \ area,\ \ Ac = n\times d\times t$$

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

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

Efficiency of a riveted joint :

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

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

F : strength of the un-riveted or solid plate

F = pitch of the rivets X thickness of the plate X tensile stress

F = p . t . σ_t

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

Hints :

1 – single rivet : no changes
2 – single cover : no changes
3 – double rivet : shearing *2 , crushing * 2
4 – double cover : shearing * 2
5 – double rivet double cover : shearing * 4 , crushing * 2

Example (1) :

Find the efficiency of a single lap joint of 6 mm plates with 2 mm diameter rivets having a pitch of 5 cm , the permissible tensile ,shearing and crushing are 1200 kg/cm^2,900 kg/cm^2 and 1800 kg/cm^2 respectively .

Solution :

$$Pt = (p-d)\times t\times\sigma_t = (5-2)\times 0.6\times 1200 = 2160 \quad kg$$

$$Ps = \frac{\pi}{4}\times d^2\times\sigma_s = \frac{\pi}{4}\times 2^2\times 900 = 2827 \quad kg$$

$$Pc = d\times t\times\sigma_c = 2\times 0.6\times 1800 = 2160 \quad kg$$

The strength of the riveted joint = Least of P_t , Ps , Pc =2160 kg

F = p . t . σ_t = 5 X 0.6 X 1200 = 3600 kg

$$\eta = \frac{2160}{3600} = 0.60 \quad \textit{or } 60\ \%$$

Example (2) :

Find the efficiency of double lap joint of 6 mm plates with 2 mm diameter rivets having a pitch of 6.5 cm , the permissible tensile ,shearing and crushing are 1200 kg/cm^2,900 kg/cm^2 and 1800 kg/cm^2 respectively .

Solution :

$$Pt = (p-d)\times t\times\sigma_t = (6.5-2)\times 0.6\times 1200 = 3240 \quad kg$$

$$Ps = \frac{\pi}{4}\times d^2\times\sigma_s\times n = \frac{\pi}{4}\times 2^2\times 900\times 2 = 5654 \quad kg$$

$$Pc = n\times d\times t\times\sigma_c = 2\times 2\times 0.6\times 1800 = 4320 \quad kg$$

The strength of the riveted joint = Least of P_t , Ps , Pc =3240 kg

F = p . t . σ_t = 6.5 X 0.6 X 1200 = 4680 kg

$$\eta = \frac{3240}{4680} = 0.69 \quad \textit{or } 69\ \%$$

Example (3) :

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

σ_t =120 N/mm^2 , σ_s = 100 N/mm^2 , σ_c =150 N/mm^2 , Find the efficiecy of the joint taking the strength of the rivet in double shear as twice than that of single shear.

Solution :

$$Pt = (p-d)\times t\times\sigma_t = (100-25)\times 20\times 120 = 180000 \quad N$$

$$Ps = n\times 2\times\frac{\pi}{4}\times d^2\times\sigma_s = 2\times 2\times\frac{\pi}{4}\times 25^2\times 100 = 196300 \quad N$$

$$Pc = n\times d\times t\times\sigma_c = 2\times 25\times 20\times 150 = 150000 \quad N$$

The strength of the riveted joint = Least of P_t , Ps , Pc =150 000 N

F = p . t . σ_t = 100 X 20 X 120 = 240 000 N

$$\eta = \frac{150000}{240000} = 0.625 \quad \textit{or 62.5 \%}$$

Homework :

1- A single riveted lap joint is made in 1.5 cm thick plates with 2 cm diameter rivets . Determine the strength of the joint , if the pitch of rivets is 6 cm .
Take σ_t = 1200 kg/cm^2 ; σ_s = 900 kg/cm^2 ; σ_c = 1600 kg/cm^2

2 -Two plates 16 mm thick are joined by a double riveted lap joint. The pitch of each row of rivets is 9 cm . The rivets are 2.5 cm in diameter. The permissible stresses are as follows : σ_t = 1400 kg/cm^2 ; σ_s = 1100 kg/cm^2 ; σ_c = 2400 kg/cm^2
Find the efficiency of the joint .

3 – A double riveted double cover butt joint is made in 1.2 cm thich plates with 18 mm diameter rivets . find the efficiency of the joint for a pitch of 8 cm .
Take : σ_t = 1150 kg/cm^2 ; σ_s = 800 kg/cm^2 ; σ_c = 1600 kg/cm^2

4 – A double riveted lap joint with chain riveting is to be made for joining two plates 10 mm thick . the allowable stresses are : σ_t = 600 kg/cm^2 ; σ_s = 500 kg/cm^2 ; σ_c = 800 kg/cm^2 , find the rivet diameter , pitch of rivets , and distance between rows of rivets , then find the efficiency of the joint .

5 – 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 :
σ_t = 100 N/cm^2 ; σ_s = 80 N/cm^2 ; σ_c = 160 N/cm^2

WELDED JOINTS

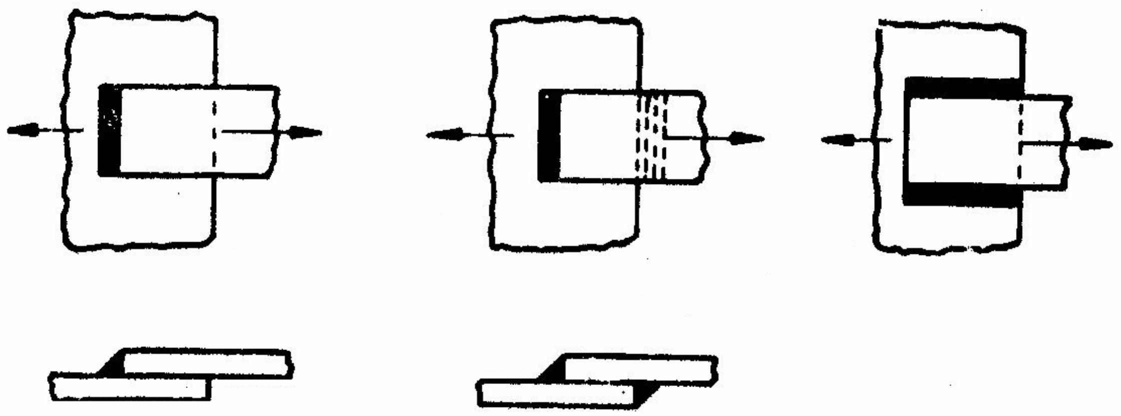
A welded joint : ***is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together , with or without the application of pressure and a fillet material .***

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

Types of welding joints :

1 – Lap joint :

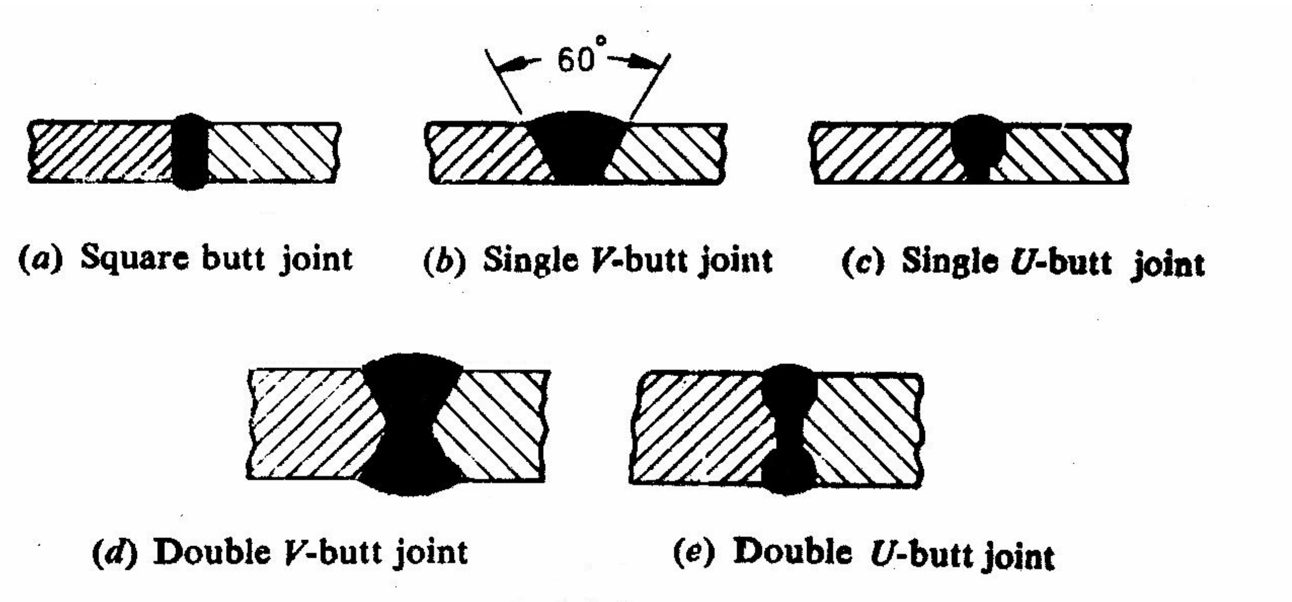
The lap joint of the fillet joint is obtained by over lapping the plates and then welding the edges of the plates , the cross – section of the fillet is approximately triangular .
There are three types of lap joint :
a – single transverse fillet
b – double transverse fillet
c – parallel fillet joint



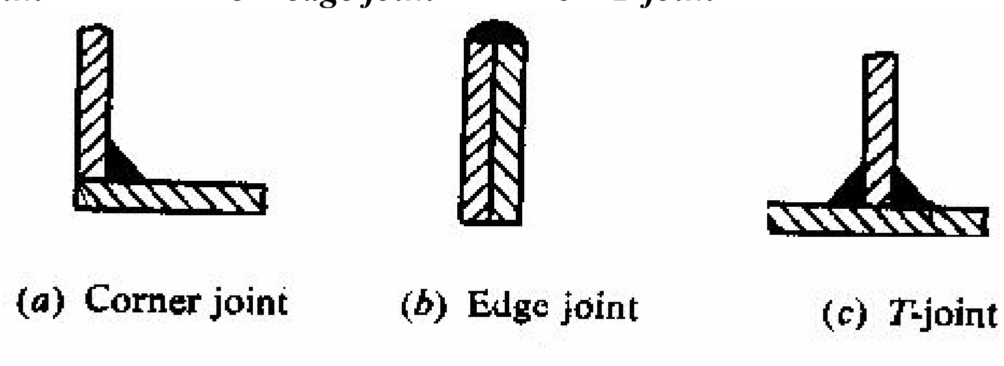
(*a*) Single transverse (*b*) Double transverse (*c*) Parallel

2 – Butt joint :

The butt joint is obtained by placing the plates edge to edge .
There are many types of butt joint :
a – square butt joint
b – single V-butt joint
c – single U-butt joint
d – double V-butt joint
e – double U-butt joint

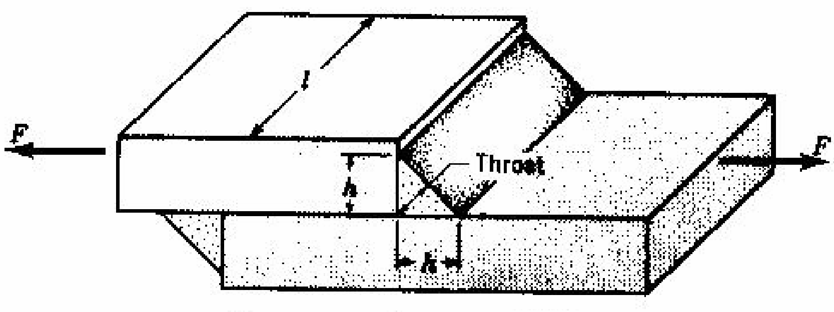


The other types of welded joints are :
a- corner joint *b – edge joint* *c – T-joint*



Strength of transverse fillet welded joints :

To determine the strength of transverse fillet welded joint , we assume that :

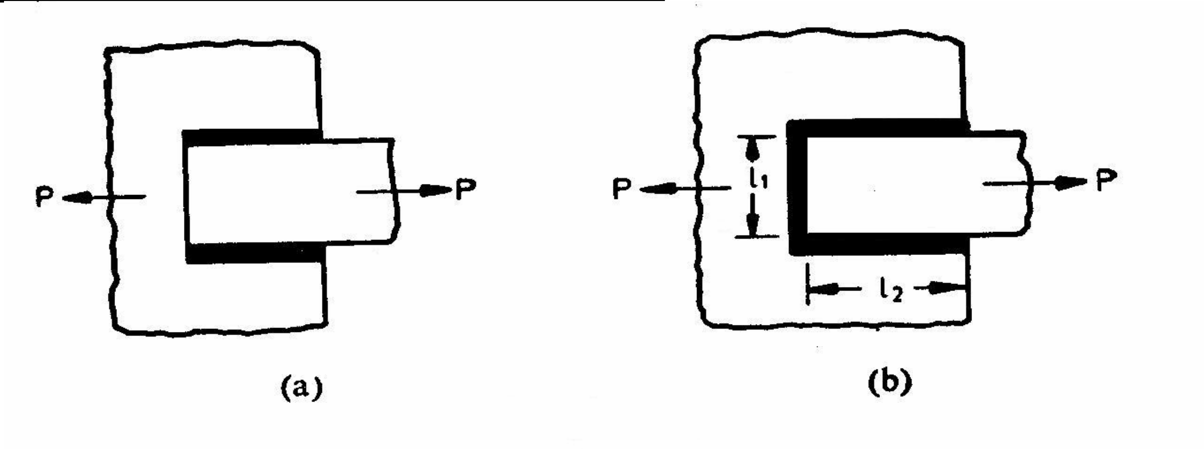


A double-filleted lap joint.

t : thickness of the plate, or
: size of the weld .
l : length of the weld .
w : width of the plate .
h : depth of the weld
σ_t : tensile stress (N/mm^2)
P : acting force (N)
In general case , t = h

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

Strength of parallel fillet welded joints :

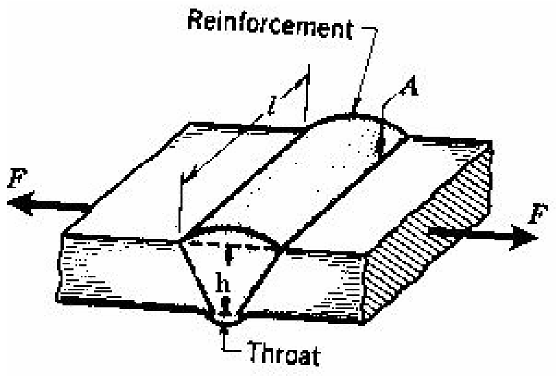


$$Ps = 1.414 \times \sigma s \times t \times l$$

Where σ S is shear stress (N / mm^2)

Strength of butt fillet welded joints :

$$Pt = \sigma t \times t \times l$$



σ_t : tensile stress (N / mm^2).
t : thickness of plate (mm) .
l : length of the weld (mm) .

Example (1) :

Two steel plates 10 cm wide and 1.25 cm thick are to be joined by double transverse fillet weld , the maximum tensile stress is not exceed 700 kg / cm^2. Find the length of the weld .

Solution :

w = 10 cm , t = 1.25 cm , σ_t = 700 kg / cm^2

Pt = Area X stress = w . t . σ_t = 10 * 1.25 * 700 = 8750 kg

$Pt = 1.414 \times \sigma_t \times t \times l$ for double transverse fillet

8750 = 1.414 * 700 * 1.25 * l

l = 7.07 cm

adding 1.25 cm in order to allow for starting and stopping of weld run .

l = 7.07 + 1.25 = 8.32 cm

Example (2) :

A plate 100 mm wide and 12.5 mm thick is to be welded to another plate by means of parallel fillet welds . The plates are subjected to a load of 50 KN , Find the length of the weld so that the maximum stress does not exceed 56 N / mm^2 .

Solution :

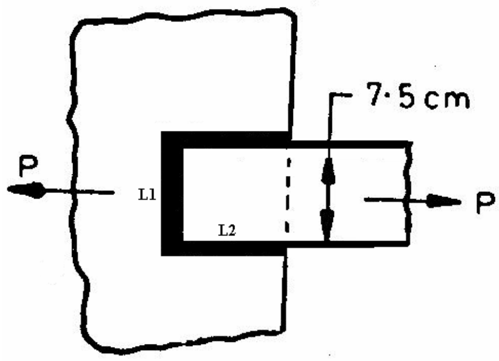
$$Ps = 1.414 \times \sigma s \times t \times l$$

50* 1000 = 1.414 * 56 * 12.5 * l

l = 50.5 mm + 12.5 = 63 mm

Example (3) :

A plate 7.5 cm wide and 1.25 cm thick is joined with another plate by a single transverse weld and a double parallel fillet welds as shown in figure , the maximum tensile and shear stresses are 700 kg / cm^2 and 560 kg / cm^2 respectively . Find the length of each parallel fillet .



Solution :

w = 7.5 cm , t = 1.25 cm , σ_t = 700 kg / cm^2 , σ_s = 560 kg / cm^2

length of weld for transverse weld , l_1 = 7.5 – 1.25= 6.25 cm

l_2 = length of each parallel fillet ,

Pt = w . t . σ_t = 7.5 * 1.25 * 700 = 6562.5 kg

Pt = Load carried by single transverse weld

= 0.707 . σ_t . t . l_1 = 0.707 * 700 * 1.25 * l_1= 3867.5 kg

Ps = Load carried by double parallel fillet weld

=1.414 . t . l_2 . σ_s = 1.414 * 1.25 * l_2 * 560 = 989 * l_2

Load carried by the joint ,

P = Pt + Ps

6562.5 = 3867.5 + 989 l_2

l_2 = 2.73 cm

adding 1.25 cm , we have l_2 = 2.73 + 1.25 = 3.98 or 4 cm

Homework :

1 – A plate 10 cm wide and 10 mm thick is to be welded with another plate by means of transverse welds at the ends , if the plates are subjected to a load of 7000 kg , Find the size of the weld , the permissible tensile stress should not exceed 700 kg / cm^2 .

(Ans. 8.32 cm)

2 – if the plates in question 1 are joined by double parallel fillets and the shear stress is not to exceed 560 kg / cm^2 , Find the length of the weld .

(Ans. 9.1 cm)

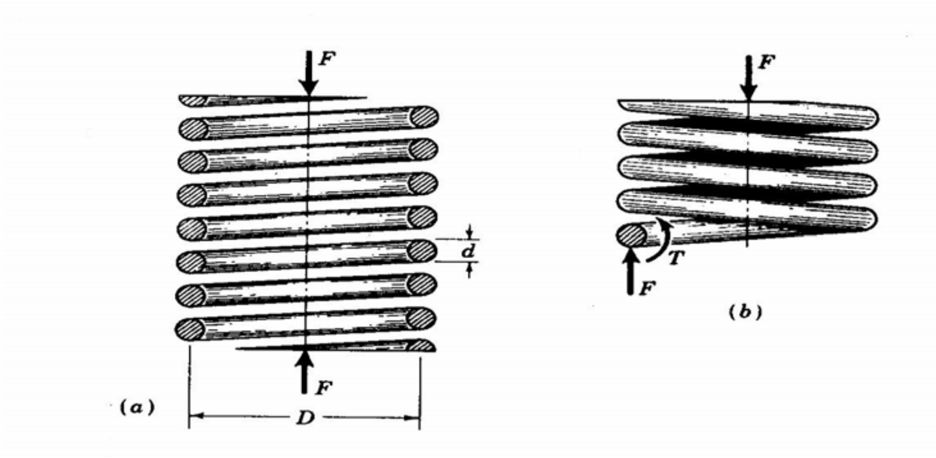
SPRINGS

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

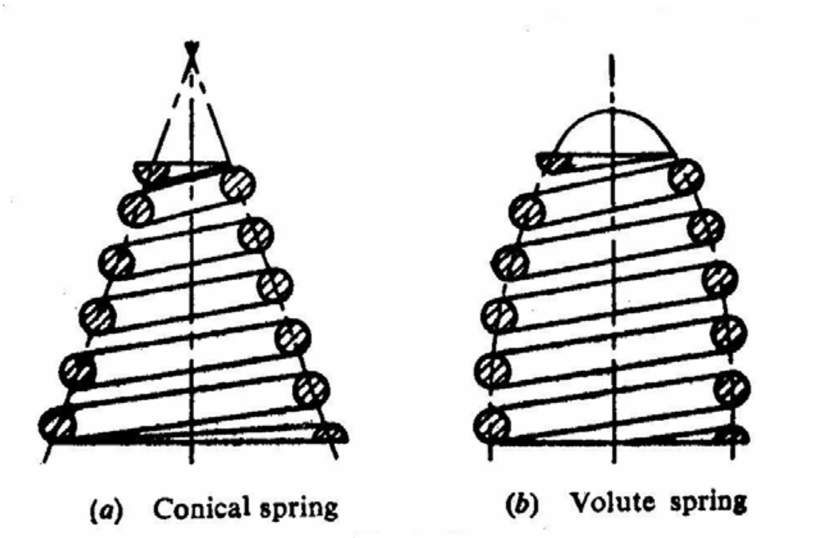
<u>The various important applications of springs are :</u>

1 – To apply forces , as in brakes and clutches and spring loaded valves .
2 – To measure forces , as in spring balances .
3 – To store energy , as in watch springs .
4 – To absorb shock and vibrations as in car springs and railway buffer .

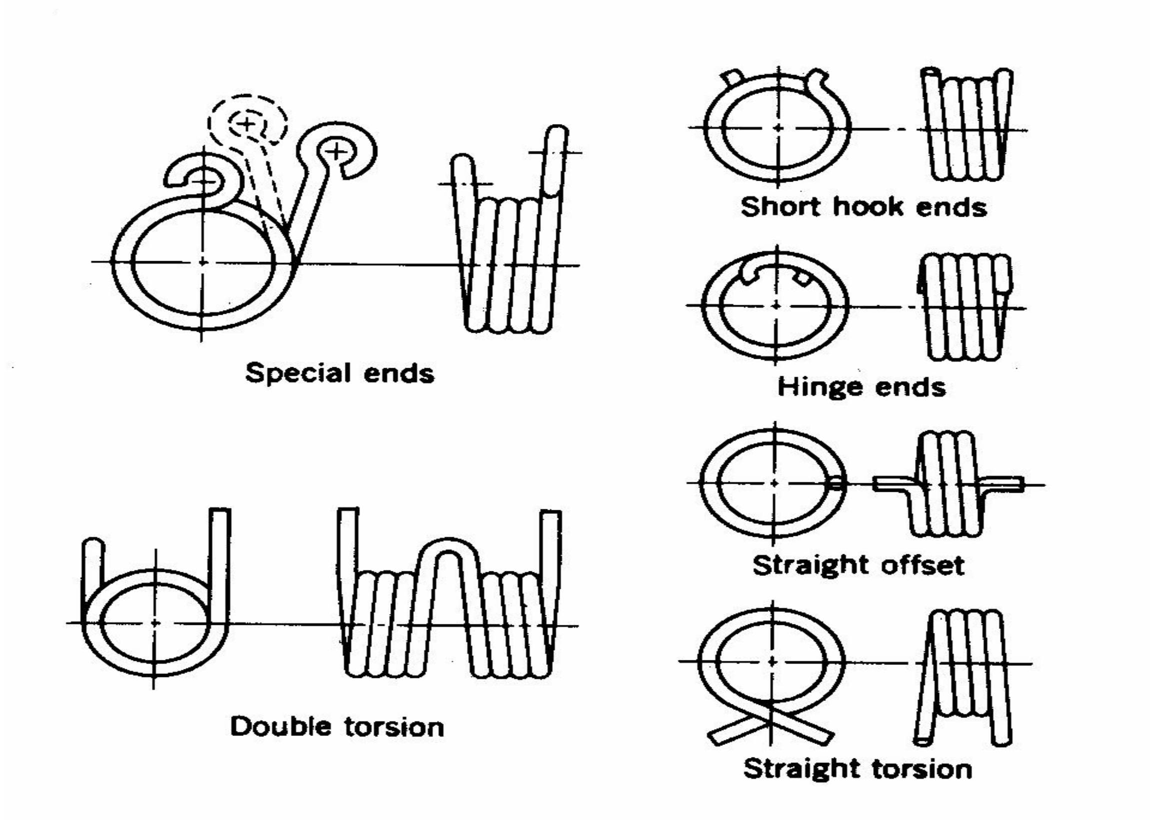
Types of springs :



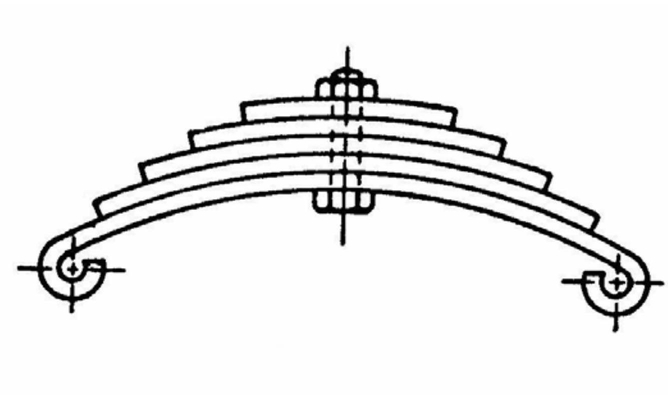
1 – Helical springs .



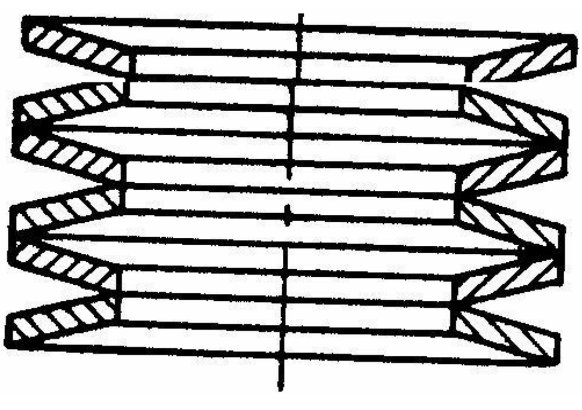
2 – Conical and volute springs .



3 – Torsion springs .

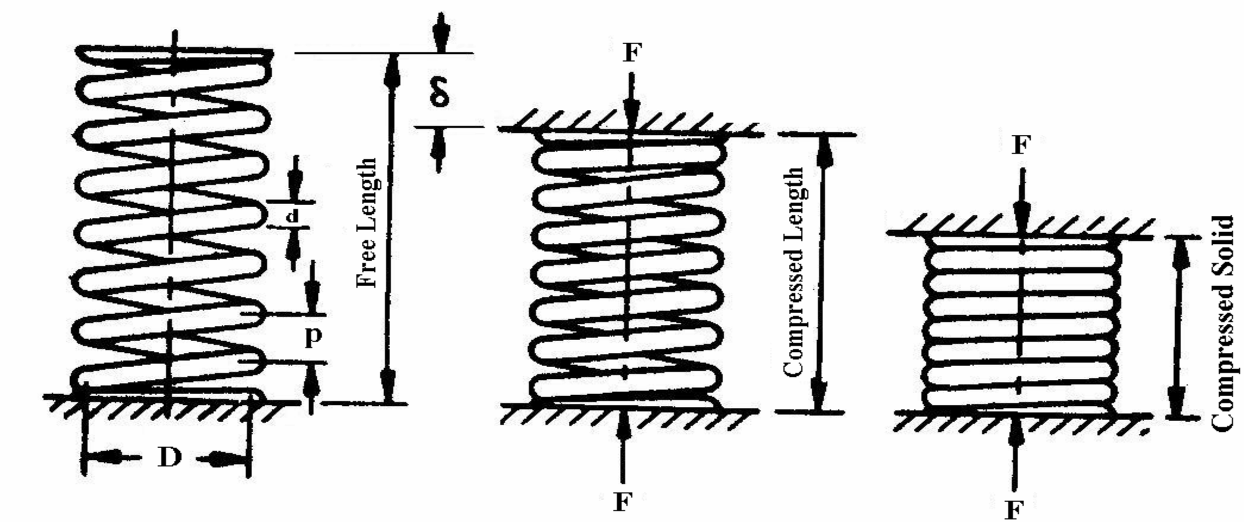


4 – Laminated or leaf springs .



5 – Disc springs .

Terms used in connection with compression spring :



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

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

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

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

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

K : spring rate
F : applied load
δ :defiection of the spring

Total active length of spring wire (L)

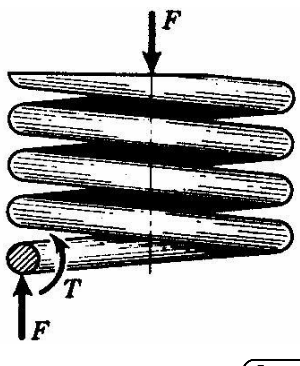
$$L = \pi . D . n$$

n : number of active turns

Stresses in helical spring of circular wire :

1 – shear stress induced in the wire due to the twisting moment :

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



2 – direct shear stress due to the load (F) :

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

3 – Stress due to curvature of spring :

a – neglecting the curvature effect :

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

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

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

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

b – considering the curvature effect :

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

When K : Wahl's correction factor

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

Deflection of helical spring of circular wire :

G : modulus of rigidity

δ : deflection of the spring

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

ملاحظات :

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

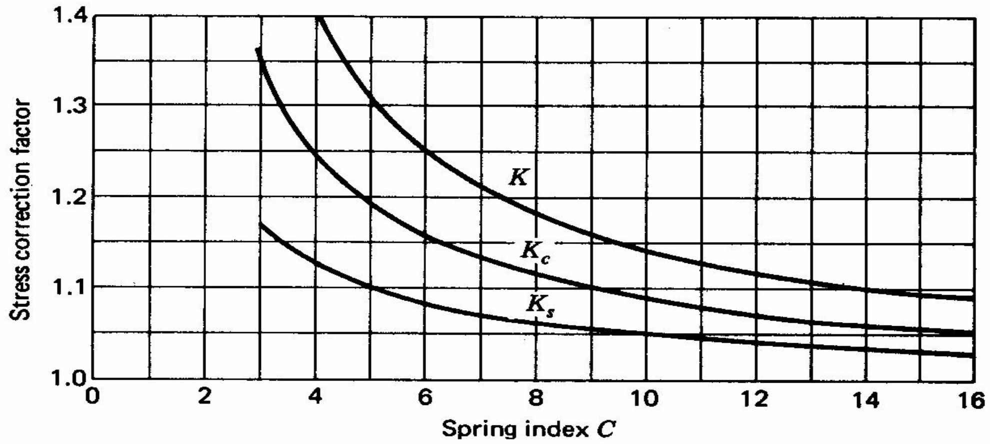
$$\sigma_s = \frac{8.F.D}{\pi.d^3} = \frac{8.F.C}{\pi.d^2}$$

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

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

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

The relationship between spring index and stress correction factor :



Example (1) :

Design a helical compression spring for maximum load of 1000 N , with a deflection of 25 mm using the value of spring index as 5 , G = modulus of rigidity = 84000 N/mm² , maximum permissible shear stress for spring wire is 420 N / mm² .

Solution :

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

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

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

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

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

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

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

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

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

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

Example (2) :

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

a – neglecting the effect of curvature .

b – considering the effect of curvature .

Solution :

d = 6 mm = 0.6 cm , D_o = 7.5 cm , σ_S 3500 ***kg / cm^2 , G = 8.4 * 10^5 kg / cm^2 ,***

D = D_o – d = 7.5 – 0.6 = 6.9 cm .

a - neglecting the effect of curvature :

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

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

b – considering the effect of curvature :

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

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

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

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

Example (3) :

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

Permissible shear stress = 3500 kg / cm^2

Modulus of rigidity = 8.4 X 10^5 kg / cm^2

Solution :

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

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

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

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

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

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

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

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

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

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

Example (4) :

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

Solution :

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

Example (5) :

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

Solution :

Spring rate (K) :

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

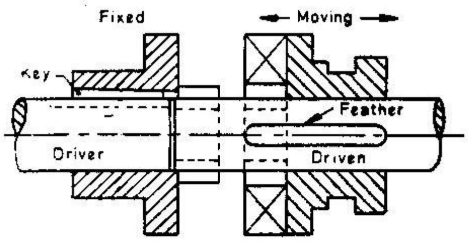
$$\delta = \frac{8\,.\,F.\;D^3.\;n}{G\,.\,d^4} \quad \Rightarrow \quad n = \frac{\delta\;.\;G.\;d^4}{8\,.\,F.\;D^3} = \frac{26.457\times 84000\times (12.6)^2}{8\times 500\times (126)^3} = 7 \;\; turns$$

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

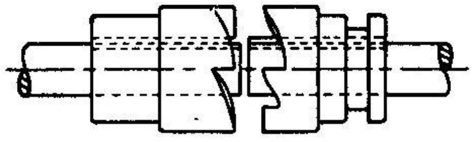
serial	Given data	Required
1	D = 300 mm , d = 30 mm F = 5 KN , n = 6 turns , G = 80 GPa	σ_S , δ , Lf
2	D = 14 cm , d = 12 mm , $\sigma_S = 100 KN$	F
3	F = 2750 N , δ = 6 mm , C= 5 , σ_S =420 N / mm^2 , G = 84 KN / mm^2	Design the spring : D , d , n , Lf , p
4	D = 40 mm , C = 5 , σ_S = 80 N / mm ^{2}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

CLUTCHES

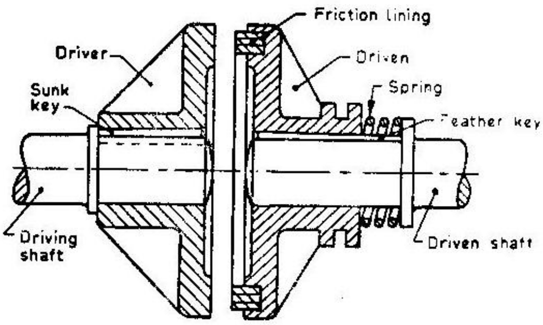
A clutch is a machine member used to connect a driving shaft to a driven shaft so that the driven shaft may be started OR stopped at without stopping the driving shaft .



(a)



(b)



(C)

The uses of the clutch is mostly found in automobiles

Types of clutches :

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

The friction clutches are :

a. **Plate (Disc) clutches (figure (C))**

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

n = number of pairs of contact surface

for single disc clutch $n = 2$

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

n_1 n_2 = number of disc on drining and driven shaft

- Considering uniform pressure

Pr : intensity of Pressure

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

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

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

T : Torque transmitted [Total friction torque]

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

μ : Coefficient of friction

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

- Considering uniform wear

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

x : distance from the axis of the clutch

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

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

$r = \dfrac{r_1 - r_2}{2}$

Notes

a. In case of a new clutch the intensity of pressure approximately uniform .
   In case of an old clutch the uniform wear theory is more approximate.
b. The uniform pressure theory gives a higher friction torque than wear theory .

Example (1) :

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

Solution :

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

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

$Pr.r_2 = C$

$C = 0.9\, r_2$

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

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

$$358 = 2\ \times 0.3\ \times 2.2\, r_2^2 \left(\frac{1.4\, r_2 + r_2}{2}\right) = 1.356\, r_2^2 \left(\frac{2.4\, r_2}{2}\right) = 1.627\, r_2^3$$

(n=2 for both sides of plate effective)

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

Example (2) :

A single disc clutch with both sides of the disc effective is used to transmit 12 horse power at 900 r.p.m. The axial pressure is limited to 0.85 kg/cm^2 . If the external diameter of the friction lining is 1.25 times to the internal diameter,Find the required dimensions of the friction lining. Assume uniform wear conditions . The coefficient of friction may be taken as 0.3 .

Solution :

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

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

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

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

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

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

$r_2^3 = 1061.11$ ……….. $r_2 = 10.195\ cm$

$$r_1 = 12.74\ cm$$

$$F = 2\pi C(r_1 - r_2) = 2 * 3.14 * 0.85 * 10.195(12.74 - 10.195) = 138.788\ kg$$

Example (3) :

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

Solution :

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

N=n1 + n2 – 1 = 3 + 2 – 1 = 4

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

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

$$Pr.r_2 = C$$

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

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

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

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

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

Homework :

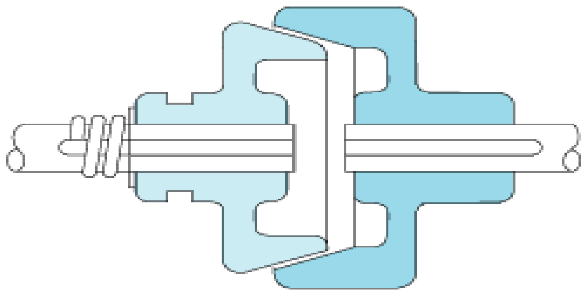
1- A single plate clutch with both sides of the plate effective is required to transmit 25 kw at 1600 r.p.m. the outer diameter of the plate is Limited to 300 mm and the intensity of pressure between the plates not to exceed 0.07 N/mm^2 . Assuming uniform wear and coefficient of friction 0.3 , find the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

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

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

b. cone clutch: (figure (D))

is used in automobile and has a conical friction surface as shown is Fig . It consists of one pair friction surface only .



area of the contact surface = 2 πrw

Normal force between contact surface

Fn = Pn x 2 πrw

Pn : normal pressure

α = angle of the friction surface with the axis of the clutch

w : width of the contact surface

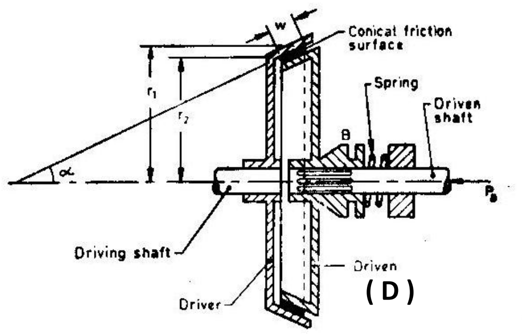
The axial spring force required to produce this normal force

Fa = Fn sin α

The Friction force or tangential force $F = \mu Fn$

Friction torque produced by the cone clutch

$T = F\,.r = Fnr = \ \mu\, Pn.\, 2\pi r^2 w$



(D)

Example (4) :

A cone clutch is to be designed to transmit 7.5 kw at 900 r.p.m the cone has a face of 12° . The width of the face is half of mean radius and the normal Pressure between the contact faces is not to exceed 0.09 N/mm^2 . Assuming uniform wear and μ = 0.2

Find the main dimension of the clutch and the axial force required to engage the clutch

$$T = \frac{P \times 60}{2\pi N} = \frac{7.5 \times 10^3 \times 60}{2\pi\,.900} = 79.56\ N.m = 79.56\ \times\ 10^3\ N/mm$$

r = mean radius of the clutch

w = face width of the clutch

$F = \frac{r}{2}$

$T = \mu . Pn . 2\pi r^2 w = \mu Pn . 2\pi r^2 . \frac{r}{2} = \mu Pn . \pi r^3$

$r^3 = \frac{T}{MP_n . \pi} = \frac{79.56 \times 10^3}{0.2 . 0.09 . \pi} = 112.44\ mm$

$F = \frac{r}{2} = 56.2\ mm$

$r_1 = r + \frac{w}{2} \sin \alpha = 112.4 + \frac{56.2}{2} \sin 12^\circ = 112.4 + 5.43 = 117.83$ **or 118 mm**

$r_2 = r - \frac{w}{2} \sin \alpha = 112.4 - \frac{56.2}{2} \sin 12 = 112.4 - 5.43 = 106.97$ **or 107 mm**

Axial force required to engage the clutch

$P_a = Pn \sin \alpha = (Pn . 2\pi r w) \sin \alpha = (0.09 \times 2\pi \times 112.4 \times 56.2) = 741.5\ N$

Shafts :

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

Types of shafts :

1- Transmission shafts

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

2- Machine shafts

These shafts form an integral part of machine itself Grank shaft . cam shaft

Design of shafts

The shafts may be designed on the basis of :

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

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

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

a - shafts subjected to twisting moment

$$\frac{T}{J} = \frac{\sigma s}{r}$$

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

T : Torque

d : diameter of shaft

σs: allowable shear stress .

$$\sigma s_{all} = \frac{\sigma s_{ult}}{s.f}$$

$$d = \sqrt[3]{\frac{16\,T}{\pi\ \sigma s}}$$ for solid shaft

$$d_O = \sqrt[3]{\frac{16\,T}{\pi \times \sigma s \times (1-k^4)}}$$ for hollow shaft $k = \frac{di}{do}$

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

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

If P in h.p then $T = \frac{P \times 4500}{2\pi N}$ kg.m

If P in kw then $T = \frac{P * 60}{2\pi N}$ kg.m

Example (1) :

A Line shaft rotating at 200 r.p.m is to transmit 25 h.p. The shaft may be assumed to be made of mild steel with an allowable shear stress of 42 N/mm^2 . Determine the diameter of the shaft . neglecting the bending moment on the shaft .

$$T = \frac{P \times 4500}{2\pi N} = \frac{25 \times 4500}{2\pi \times 200} = 89.5\ kg.m = 895 \times 10^3\ N.mm$$

$$d = \sqrt[3]{\frac{16\,T}{\pi \times \sigma s}} = \sqrt[3]{\frac{16 \times 895 \times 10^3}{\pi \times 42}} = 47.7\ mm$$

$d_{standard} = 50$ mm

Example (2) :

Find the diameter of a solid steel shaft to transmit 25 horse power at 200 r.p.m. The ultimate shear stress for steel may be taken as 360 N/mm^2 and a factor of safely as 8 .

If a hollow shaft is to be used in place of the solid shaft find the inside and outside diameter . When the ratio of in to out diameter is 0.5 .

$$\sigma s_{allowable} = \frac{360}{8} = 45\ N/mm^2$$

$$T = \frac{P\times4500}{2\pi N} = \frac{25\times4500}{2\pi\times200} = 89.5\ kg.m = 895\ N.mm\ \times\ 10^3$$

$$d = \sqrt[3]{\frac{16\times895\times10^3}{\pi\times45}} = 46.6 \approx 50mm$$

For hollow shaft :

$$d_o = \sqrt[3]{\frac{16\times895\times10^3}{\pi\times450\,(\,1-0.5^4\,)}} = 47.6 \approx 50mm$$

d_i = 0.5 d_o =25 mm

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

A solid shaft is transmitting 1×10^3 kw at 240 r.p.m. Determine the diameter of the shaft if the maximum torque transmitted exceeds the mean torque by 20% , Take the maximum allowable shear stress 60 N/mm^2

$P\ =\ 1\ \times\ 10^3\ k.w$, $N\ =\ 240\ r.p.m$, $d = ?$

$T_{max}\ =\ 1.2\ T_{mean}$, $\sigma s\ =\ 60\ N/mm^2$

$$T_{mean} = \frac{P\times60}{2\pi N} = \frac{1\times10^3\times60}{2\pi\times240} = 39.808\ \ N.m =\ 39808\ N.mm$$

$T_{max}\ =\ 1.2\ \times\ 39808\ =\ 47770.7\ N.mm$

$$d = \sqrt[3]{\frac{16T}{\pi\times\sigma s}} = \sqrt[3]{\frac{16\times47770.7\times10^3}{\pi\times760}} =\ 159.43\ mm\ =\ 160\ mm$$

<u>b. shafts subjected to bending moment</u>

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

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

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

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

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

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

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

Example (4) :

A pair of wheels of a railway wagon carries a load of 5100kg (5 tonnes) on each axle box , acting at a distance of 100 mm out side the wheel base . The gauge of the rails is 1400 mm . find the diameter of the axle between the wheels , if the stress in not exceed 100 N/mm^2 .

max . bending moment

$$M = W.L. = 5000 \times 100 = 5 \times 10^6 \ N.mm$$

$$d = \sqrt[3]{\frac{32 \times 5 \times 10^6}{\pi \times 100}}$$

$$d = 79.84 \ mm \quad , \quad d = 80 \ mm$$

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

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

The two theories .

1- **Max shear stress theory or Guests theory . it used for ductile material such as mild steel .**
2- **Max , normal stress theory or Rankine's theory . it is used for brittle materials such as cast Iron .**

Let σ_s : shear stress induced due to twisting moment

σ_b : Bending stress induced due to bending moment

According to theory (1) :

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

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

Te = equiralent twisting moment

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

According to theory (2) :

maximum normal stress theory

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

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

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

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

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

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

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

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

Example (5) :

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

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

$$\sigma_{s_{all}} = \frac{500}{6} = 83.3\,N/mm^2$$

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

Te = **10.44 × 10^6 N.mm**

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

Example (6) :

Design a shaft to transmit power from an electric motor to Lathe head stock through a pulley by means of a belt drive . The pulley weighs 20 kg and is Located at 100 mm from the centre of the bearing . The diameter of the pulley is 200 mm and the maximum power transmitted is 1.5 h.p at 120 r.p.m. The angle of Lap of the belt is 180° and coefficient of friction between the belt and the pulley is 0.3 . The allowable shear stress in the shaft may be taken 35 N/mm^2 .

$$T = \frac{P * 4500}{2\,\pi N} = \frac{1.5\,\times\,4500}{2\pi\,\times\,120} = 8.95\,kg.m = 8.95\,x\,10^4\,N.mm$$

T_1 , T_2 = Tension on the tight and slack sides of the belt

$T = (T_1 - T_2)\ R$

$8.95 \times 10^4 = (T_1 - T_2)\ 100$

$T_1 - T_2 = 8.95 \times 10^2$ N

$2.3 \text{Log}\ \frac{T_1}{T_2} = Mo = 0.3\ \pi$

$\text{Log}\ \frac{T_1}{T_2} = \frac{0.3\pi}{2.3} = 0.4098$

$\frac{T_1}{T_2} = 2.57$

$T_1 = 2.57\ T_2$

$2.57\ T_2 - T_1 = 8.95 \times 10^2$

$1.57\ T_2 = 8.95 \times 10^2$

$T_2 = 570$ N

$T_1 = 2.57 \times 570 = 1465$ N

Now total Vertical Load acting on the pulley

$= T_1 + T_2 + W = 1465 + 570 + 200 = 2235$ N

$M = (T_1 + T_2 + W)\ L = 2235 \times 100 = 2235 \times 10^2$ N.mm

$Te = \sqrt{M^2 + T^2} = \sqrt{(2235 \times 10^2)^2 + (895 \times 10^2)^2} = 3800 \times 10^2$ N.mm

$Te = \frac{\pi}{16}\ \sigma_s\ d^3$

$d = \sqrt[3]{\frac{16 \times 3800 \times 10^2}{\pi \times 35}}$

d = 38 – 40 mm .

Example (7) :

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

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

$$T = \frac{120 \times 45 \times 10^6}{2\pi \times 300} = 2.86\ \times 10^6\ N.mm$$

Max . bending moment at C or D

$M = 1500 \times 1000 = 1.5 \times 10^6$ N.mm

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

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

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

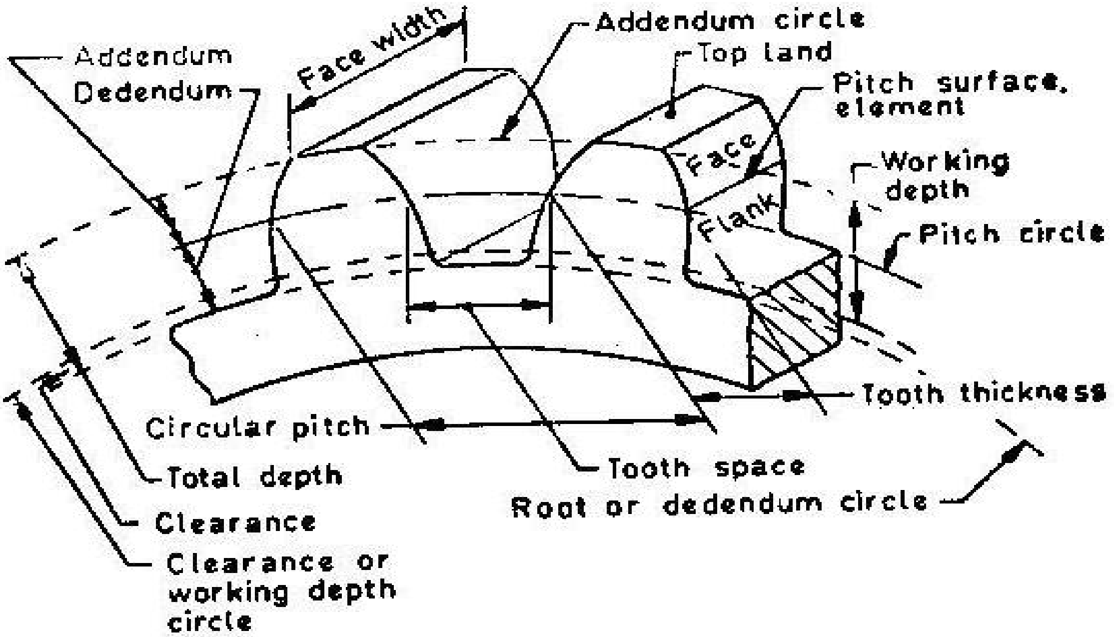
$d = \sqrt[3]{\frac{16 \times 3.2 \times 10^6}{\pi \times 60}}$ = 64.9 mm ≈ 65 mm

Gears

Technical terms :

$P_c = \frac{\pi D}{T}$ **P_c : circular pitch , D : diameter of pitch circle , T : No.of Teeth**

$m = \frac{D}{T}$ **, Model may be taken as (1.25,1.5)** , $\frac{D_1}{D_2} = \frac{T_1}{T_2}$



Initial data in kinematic calculation

1- **the number of spindle steps = Z**

2- **the Progressior ratio =Φ**

3- **Spindle speeds = N_1 , N_2 , …. , N_z**

4- **motor speed = N**

IF Z is given :

- **First we put Z in the form**

$Z = Z_1 * Z_2 * Z_3$ or $Z = Z_g * Z_{r1} * Z_{r2}$

= the product of transmission numbers (Z_1 , Z_2) in each consecutive group

- **and , the number of groups , the number of transmission within each group , and the group arrangement may be different as :**

Z = 4 = 2*2

Z = 6 = 2*3 = 3*2

Z = 8 = 2*2*2 = 2*4 = 4*2

Z= 12 = 3*2*2 = 2*3*2 = 2*2*3 = 3*4 = 4*3

For example

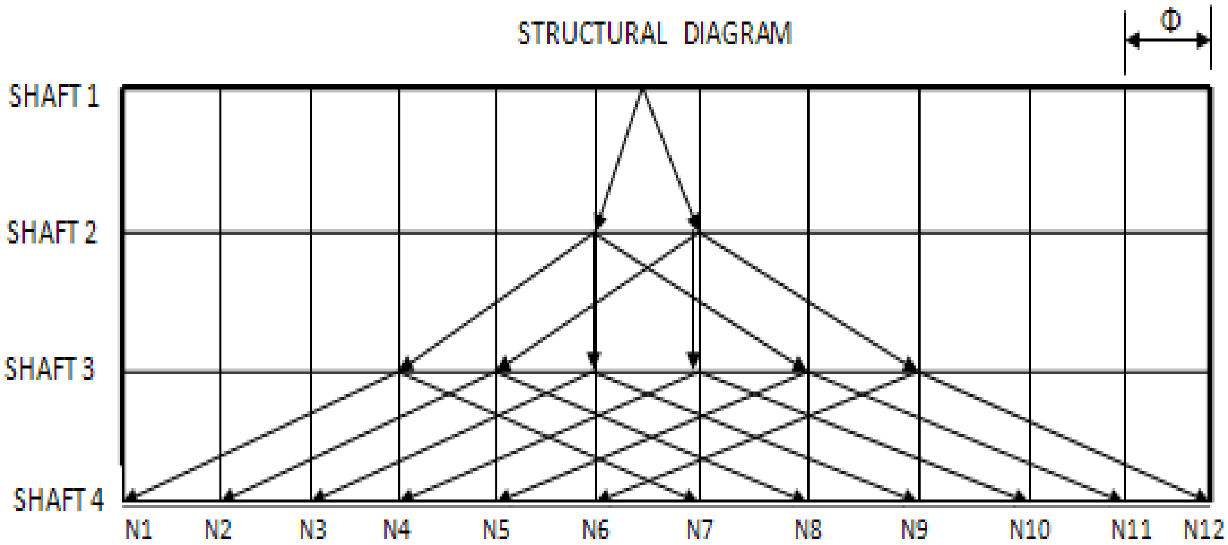
If Z = 12 = 2*3*2 → Three groups

No.of shafts = No.of groups +1 = 3+1 = 4 shaft

No.of gears = (2+3+2) *2 = 7*2 = 14 gear

To constructe the structural diagram :

- **No.of horizontal Lines = Noof shafts = 4**
- **No.of vertical Lines = Noof speeds = 12**
- **Point O in symertical position on shaft I**



For main group

- **on shaft II , put No.of points symetrically to point O = No.of steps in this group**

$Z_g = 2$

Speed chart

- **it gives absolute values of transmission ratios and speeds of all shafts**
- **To avoid excessively Large diameters of gear wheel and soon an increase in the overall radial dimension of speed gear box we put Limited values for spur gear transmissions :**

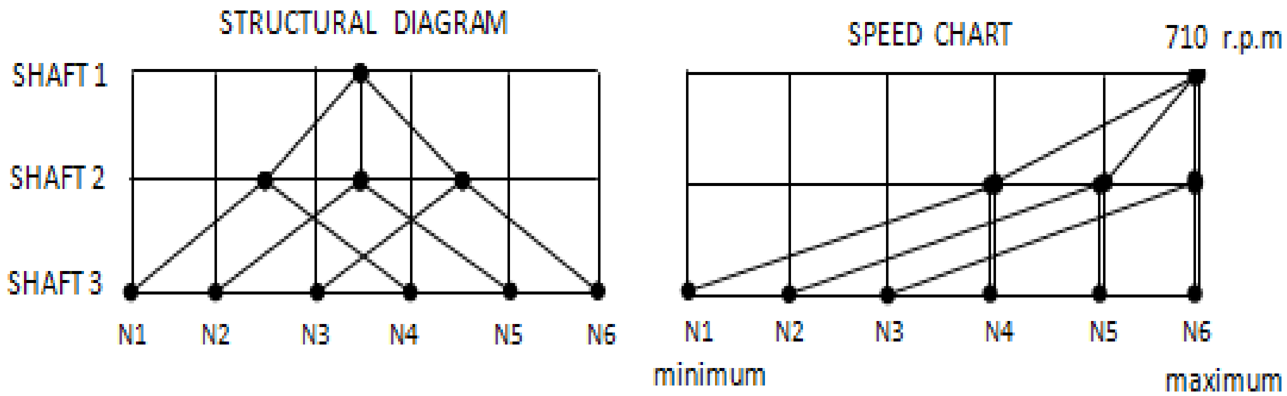
$$i_{max} = \frac{N_7}{N_5} \leq 2 \ \& \ i_{min} = \frac{N_1}{N_5} \geq \frac{1}{4}$$

Example :

6 speed gear box having two transmission groups , assume that :

its minimum speed = 233 r.p.m. , maximum speed = 710 r.p.m.

Solution :



Shaft I :

Has the following speed :

N_o : N_{max} = 710 r.p.m which coupled to the motor

shaft II :

has the following speed .

$N_6 = \frac{N_o}{\emptyset^0}$ $\quad$ $N_5 = \frac{N_o}{\emptyset^1}$ $\quad$ $N_4 = \frac{N_o}{\emptyset^2}$

$$\emptyset = \sqrt[Z-1]{\frac{N_{max}}{N_{min}}} = \sqrt[5]{\frac{710}{232}} \approx 1.25$$

$$N_6 = \frac{710}{1} = 710 \ r.p.m \qquad N_5 = \frac{710}{1.25} = 568 \ r.p.m$$

$$N_4 = \frac{710}{(1.25)^2} = 455 \ r.p.m$$

Shaft III :

$$N_6 = \frac{710}{\emptyset^0} = 710\, r.p.m \ , \ N_5 = \frac{568}{\emptyset^0} = 568\, r.p.m \ , \ N_4 = \frac{455}{\emptyset^0} = 455\, r.p.m$$

$$N_3 = \frac{710}{(1.25)^3} = 364 \ r.p.m \ , \ N_2 = \frac{568}{(1.25)^3} = 291\, r.p.m$$

$$N_1 = \frac{450}{(1.25)^3} = 233\, r.p.m$$

POWER SCREWS

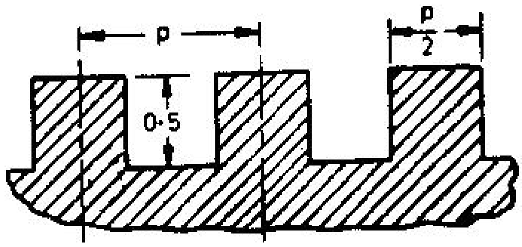
The power screws (known as translation screws) are used to convert rotary motion into translatory motion .

For example : in the case of a lead screw of lathe , the rotary motion is available , but the tool has to be advanced in the direction of the cut .

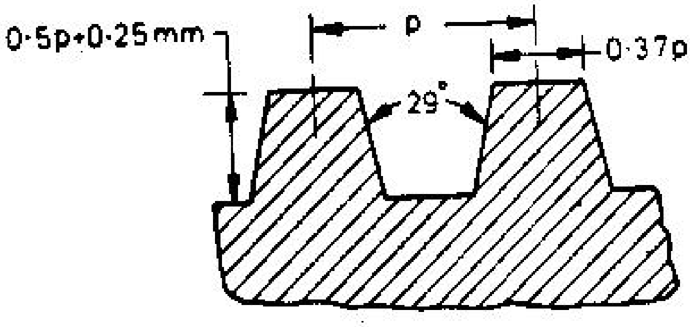
In case of a screw Jack , a small force applied in the horizontal plane is used to raise or lower a larg Load .

Power screw are also used in Vices, testing machines , presses

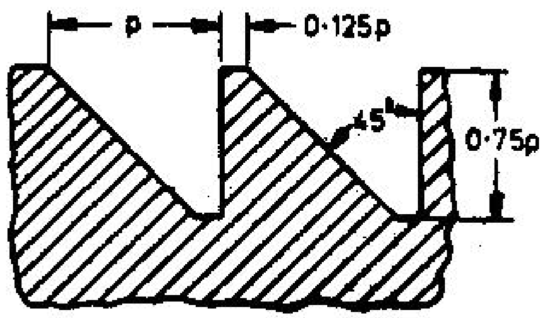
Types of screw threads used for power screw :



1- Square thread



2- Acme thread



3- Buttress thread

Torque required to raise load on squar threaded screws :

p : Pitch of the screw

d : Mean diameter of the screw

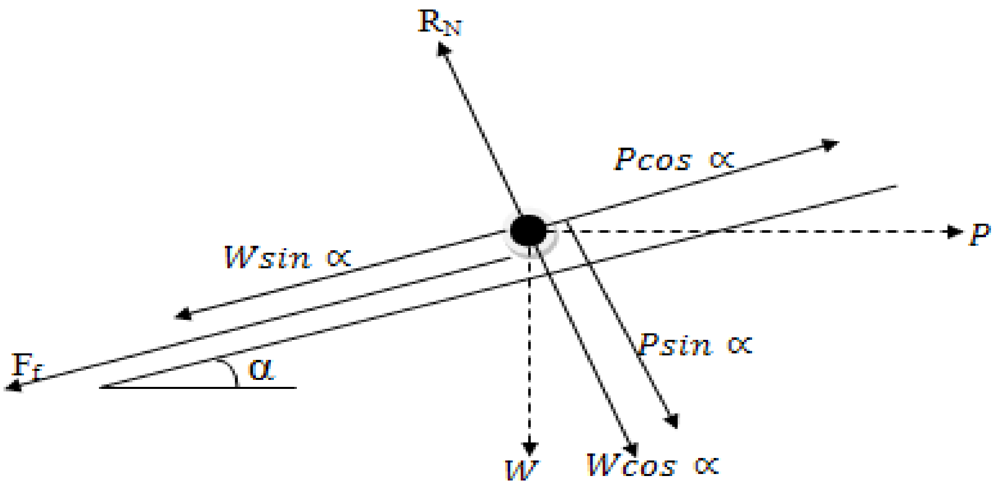
$\propto = Helix\ angle$

P : Effort applied at the circumference of screw to lift the load .

W : Weight of the body to be lifted

μ : Coefficient of friction between the screw and nut

$F_f = \mu \, . \, R_N$



$$Pcos \propto = Wsin \propto \; + \mu \, . R_N \; = Wsin \propto + \mu \, (P . sin \propto + Pcos \propto)$$

$$= Wsin \propto + \mu \, . P . sin \propto + \mu \, . Wcos \propto$$

$$Pcos \propto - \; \mu \, . P . sin \propto = Wsin \propto + \mu \, . Wcos \propto$$

$$P(cos \propto - \; \mu \, . sin \propto) = W \, (sin \propto + \mu \, . cos \propto)$$

$$P = W * \frac{sin\propto + \mu \, .cos\propto}{cos\propto - \mu \, .sin\propto}$$

$\mu = tan\emptyset$

$\emptyset : friction\ angle$

$$P = W * \frac{sin\propto + tan\emptyset\ .cos\propto}{cos\propto - tan\emptyset sin\propto}$$

$$P = W * \frac{sin\propto cos\emptyset + sin\emptyset cos\propto}{cos\propto cos\emptyset - sin\propto sin\emptyset}$$

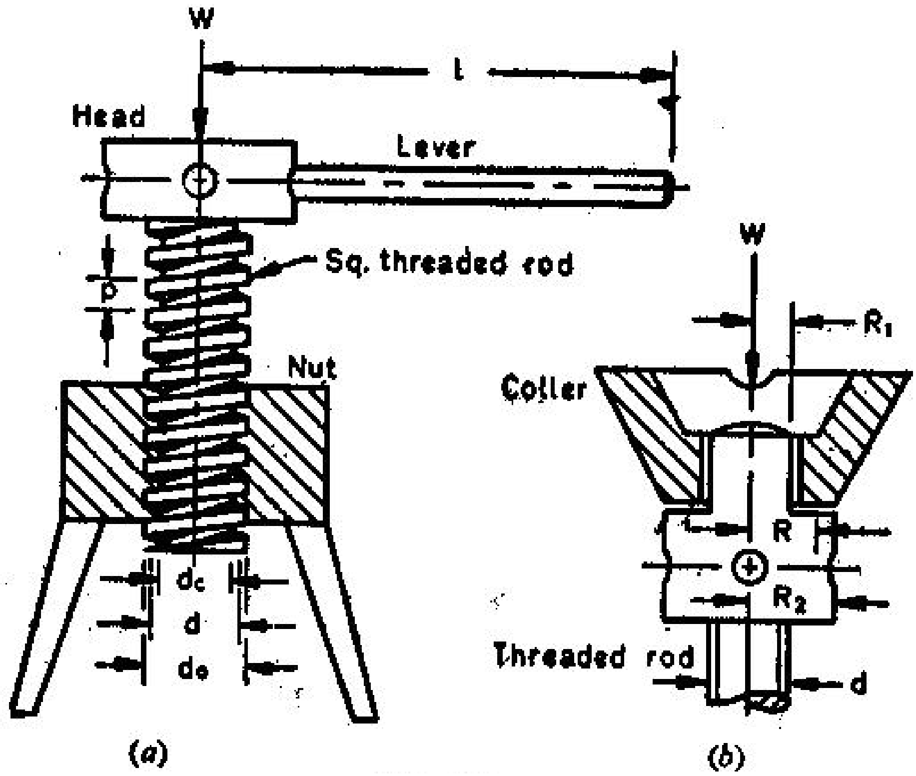
$$= W * \frac{\sin(\propto + \emptyset)}{\cos\ (\propto + \emptyset)}$$

$$= W * \tan\ (\propto + \emptyset)$$

$$T = P * \frac{d}{2} = W.tan\ (\propto + \emptyset) * \frac{d}{2}$$

Example (1) :

A vertical screw with single start square threads 5 cm mean diameter and 1 cm pitch , is raise against a load of 550 kg . by means of hand wheel , the boss of which is threaded to act as a nut , The axial load is taken up by a thrust collar which supports the wheel boss and has a mean diameter of 6.5 cm , $\mu = 0.15$ for screw , $\mu = 0.18$ for the collar , and the tangential force applied by each hand to the wheel is 14 kg . Find the suitable diameter of the hand wheel .



Solution :

d = dia . of screw = 5 cm , p = 1 cm , W=550 , D : mean dia . of collar = 6.5 cm

R : mean radius = 6.5 / 2 = 3.25 cm , P = 14 kg

coff. friction for screw $\mu = tan\emptyset = 0.15$, coff. friction for collar $\mu_1 = 0.18$

$$tan \propto = \frac{P}{\pi . d} = \frac{1}{\pi * 5} = 0.064$$

$$P = w. \tan(\propto + \emptyset) = W[\frac{tan\propto + tan\emptyset}{1 - tan\propto . tan\emptyset}]$$

$$= 550 * \left[\frac{0.064 + 0.15}{1 - 0.064 * 0.15} \right] = 119kg$$

$$T = P * \frac{d}{2} + \mu_1 W . R$$

$$= 119 * \frac{5}{2} + 0.18 * 550 * 3.25$$

$$= 619.25 kg . cm$$

Diameter of hand wheel .

$$T = 2P * \frac{D_1}{2} = 2 * 14 * \frac{D_1}{2}$$

$$= 14D_1$$

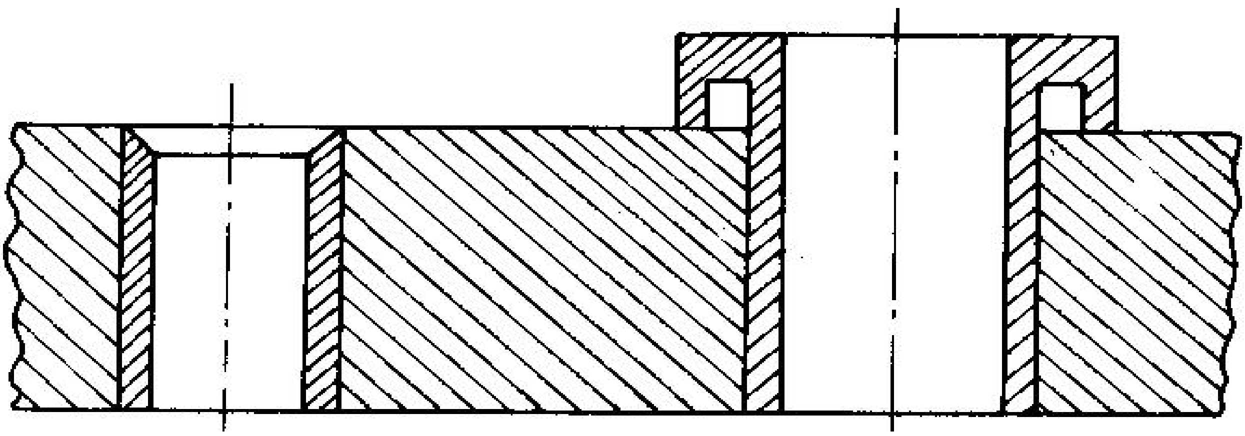
$$14D_1 = 619.25$$

$$D_1 = 44.23\ cm$$

JIGS AND FIXTURES

The Jigs and fixtures have a special importance in cutting mechanism because they help to operate the machining processes and make the feeding movements easy to use on work table .

The Jig may be one metal piece working on the diminsions of limited part , or to limit the path of other parts ,



Bush without head   Bush with head

Fig (1) : Two ways to fix the block by bushes

Determination of workpiece position

It is necessary to process machining operation in order to gaurant exact production according to its design and drawings .Fig (2) : Showsan arrangement to fixe a drilling workpiece with wrong style

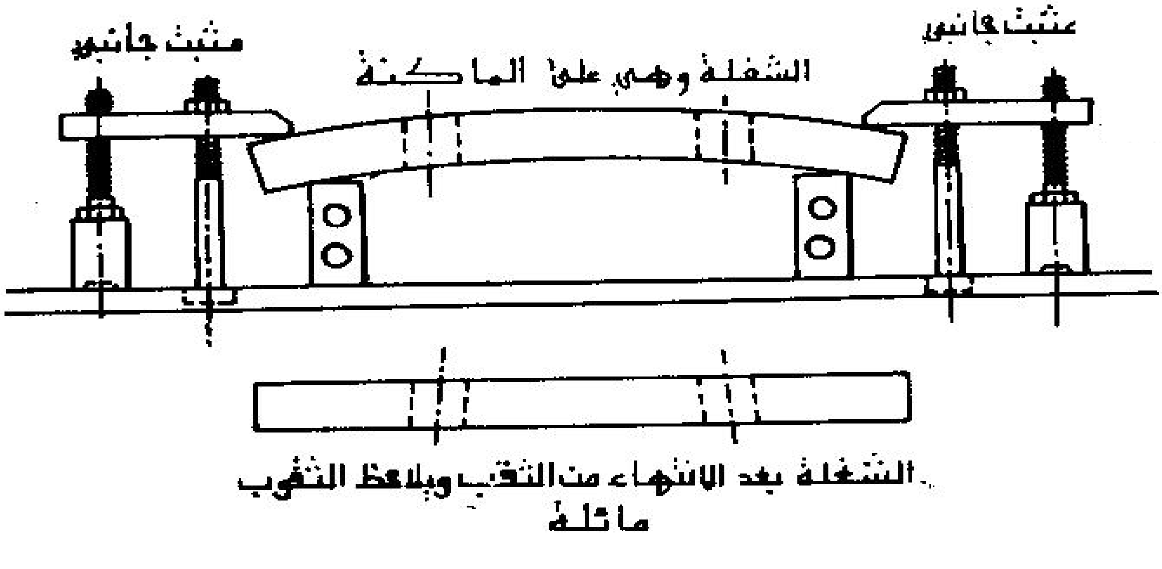


Fig (2) Wrong fixture in drilling w.p.

Fig (3) Shows an arrangement to fixe duble acting arrangement , which press the workpiece on the working table and at the same time work to determine its position horizontally a head the cutting tool .

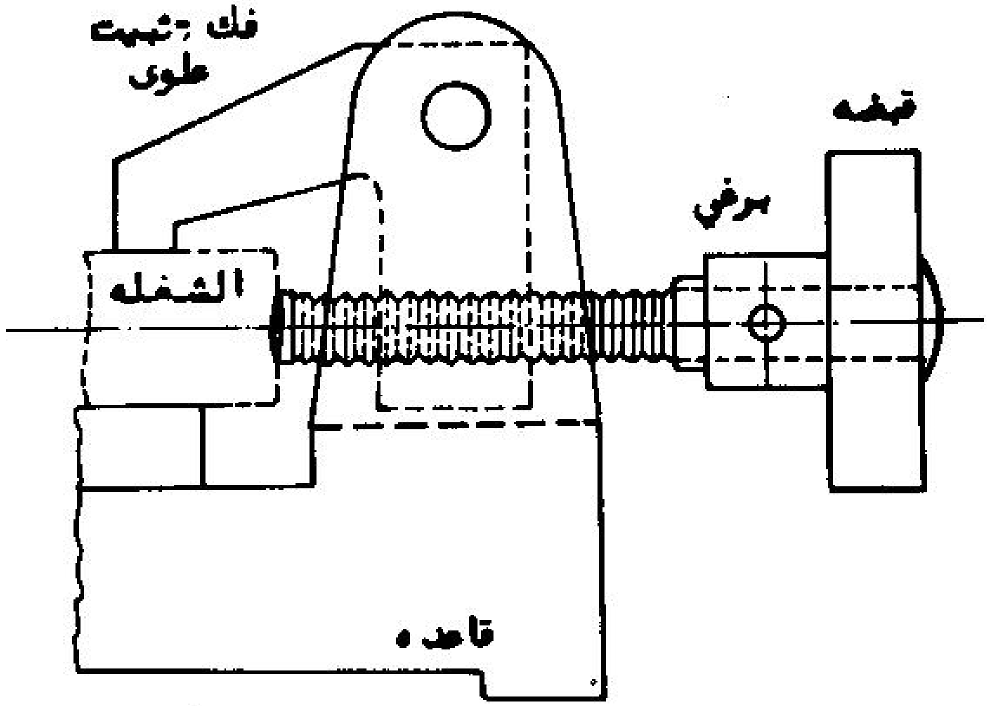
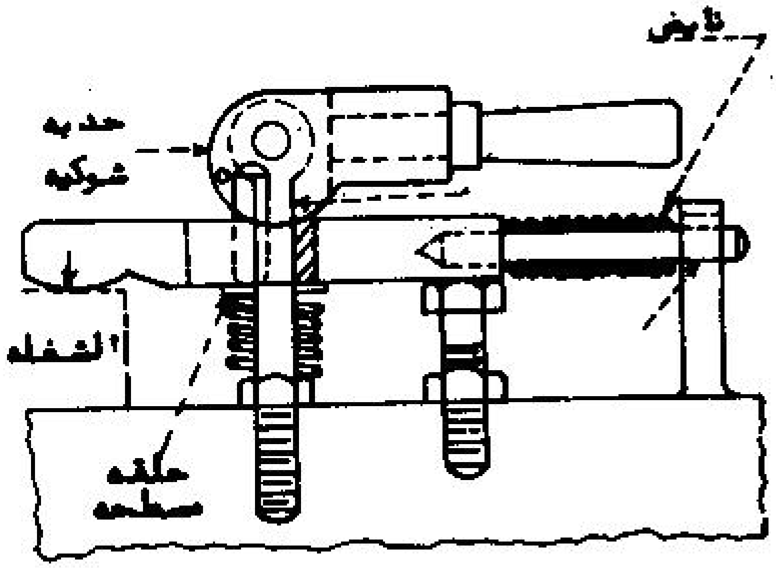
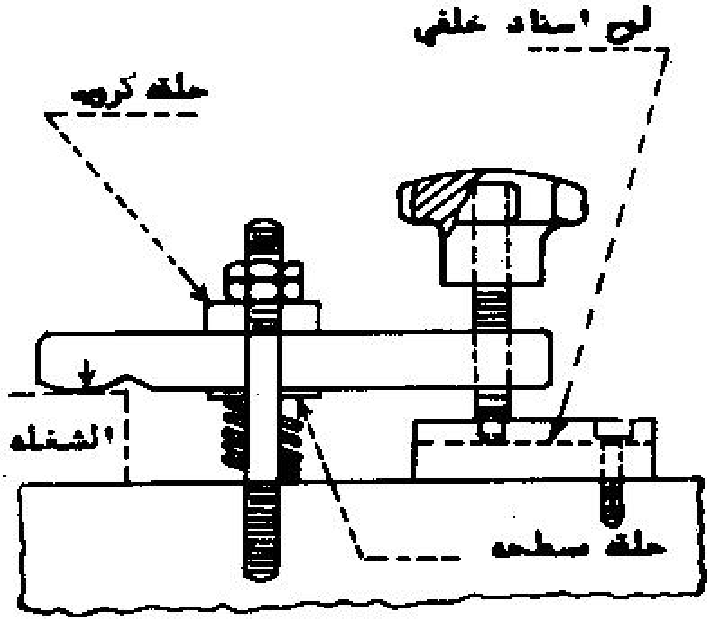


Fig (3) Double acting arrangement

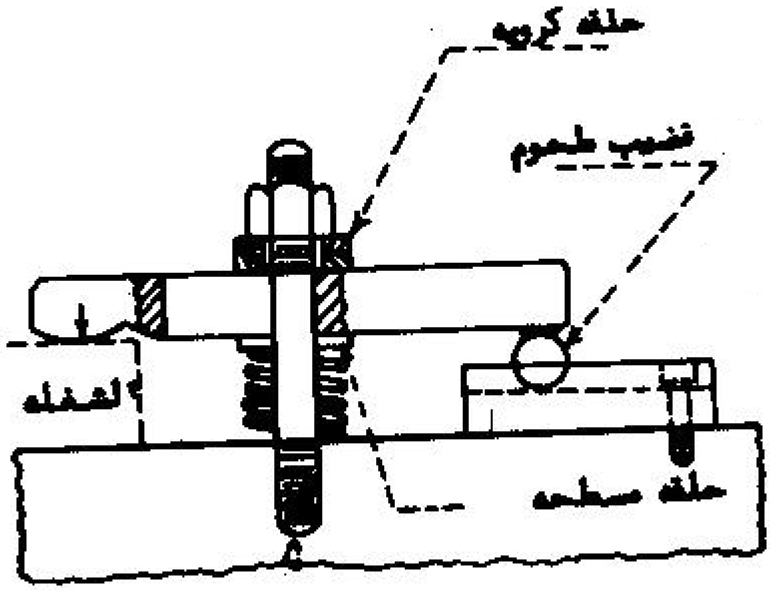
<u>Other types of fixtures & guides :</u>



Screw with ear



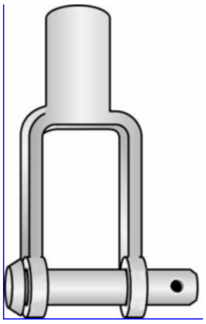
Back supported plate



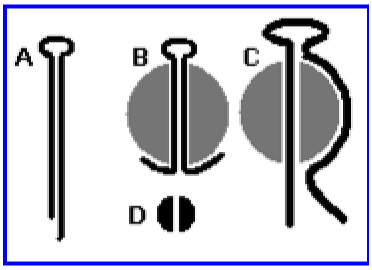
Welded bar & spherical ring

The elements used in determination of workpiece position :

1- bolts
2- kays
3- bushes
4- studs
5- cams
6- screws
7- pins : some pins shown in the following figures :



Clives pin



groove pins

push pin

bobby pin

The elements used in fixing processes

1- Mechanical elements
2- Hydroulic elements
3- Air elements
4- Air & hydroulic elements

1^{st} semester 2006 / 2007

Time : 2 hrs **Date : 11 / 2 / 2007**

NOTE : ANSWER ONLY FIVE QUESTIONS .

Q 1 : Find the length of belt necessary to drive a pulley of 80 cm diameter running parallel at a distance of 12 m from the driving pulley of diameter 480 cm .

Q 2 : ***A cast iron column has internal diameter of 100 mm , What should be the minimum external diameter so that it may carry a load of 160 N , without the stress exceeding 50 N / mm^2 ?***

Q 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 :***

$\sigma_t = 100\ N/cm^2$; $\sigma_s = 80\ N/cm^2$; $\sigma_c = 160\ N/cm^2$

Q 4 : ***Calculate the cutting power which is required to turn work piece from 50 mm diameter to 44 mm diameter under the following conditions :***

The feed = 0.2 mm/rev , cutting force = 200 kg , η = 0.70 , number of revolution for machine 200 r.p.m , then find :

i- Motor power .

ii- Amount of heat generated in cutting .

iii- Twisting moment in turning .

Q 5 : Design a compression helical spring to carry a load of 2750 N with a deflection of 6 mm , the spring index may be taken as (5) , Assume the following data for the spring material
Permissible shear stress = 420 N / mm^2 , Modulus of rigidity = 84 KN / mm^2

Q 6 : Prove that $\sigma_c = 3\ \sigma_s$, for Gib-head key when it is equally strong in shearing and crushing .

Q 7 : Explain (FIVE) of the following elements by free-hand sketches :

1. ***Acme thread***
2. **Eye bolt**
3. **T-welded joint**
4. **Splines key**
5. **Leaf spring**
6. ***Open flat belt***

Q 8 : Write three uses for the following :

1. ***Bolts***
2. **Riveted joint**
3. **Welded joint**

GOOD LUCK

FOUNDATION OF TECHNICAL EDUCATION
NASSIRIYA TECHNICAL INSTITUTE
MECHANICAL DEPARTEMENT

Machines Elements Exam For 2^{nd} Year Students

2nd semester 2006 / 2007

Time : 2 hrs **Date : 29 / 4 / 2007**

Q 1 : A single disc clutch with both sides of the disc effective is used to transmit 12 horse power at 900 r.p.m. The axial pressure is limited to 0.85 kg/cm^2 . If the external diameter of the friction lining is 1.25 times to the internal diameter, Find the required dimensions of the friction lining. Assume uniform wear conditions . The coefficient of friction may be taken as 0.3 .

Q 2 : ***A solid shaft is transmitting 1×10^3 kw at 240 r.p.m., Determine the diameter of the shaft if the maximum torque transmitted exceeds the mean torque by 20% , Take the maximum allowable shear stress 60 N/mm^2 .***

Q 3 : For maximum normal stress theory or Rankine's theory :

$$\sigma_{b_{max}} = \frac{1}{2}\,\sigma_b + \sqrt{(\frac{1}{2}\,\sigma_b)^2 + (\sigma_s)^2}$$

Prove that : $d = \sqrt[3]{\dfrac{32Me}{\pi.\sigma_b}}$ ***when (d) is the diameter of solid shaft***

Q 4 : ***A (8) speeds gear box , the maximum speed is 800 r.p.m , and the minimum speed is 47 r.p.m***

1- ***Draw structural diagram***
2- ***Draw speed chart***
3- ***Find the speeds between maximum speed and minimum speed***

$take = \sqrt[7]{17.085} \approx 1.5$

Q 5 : draw a section in gear to show the following :

a- ***Adendum circle***
b- ***Dedendum circle***
c- ***Pitch circle***
d- ***Clearance circle***
e- ***Face width***
f- ***Teeth thickness***

GOOD LUCK

1st Attempt 2006 / 2007

Time : 3 hrs **Date : / 6/ 2007**

NOTE : ANSWER ONLY SIX QUESTIONS .

Q 1 : Explain (Five) the following elements by free-hand sketches only :

1 - U-bolt , 2 - double riveted butt joint , 3 - cross flate belt , 4 - Square thread ,

5 - Double torsion spring , 6 - Clives pin

Q 2 : (A) : What are the elements used in determination of workpiece position ?

(B) : Derive an equation to calculate the force required to raise load on squar threaded screws .

Q3: A (12) speeds gear box , Z = 12 = $2 \times 3 \times 2$, maximum speed = 1600 r.p.m , minimum speed = 50 r.p.m ,

A - Constructe the structural diagram and speed chart.

B – find the speeds between maximum and minimum speed . Take : $\sqrt[11]{32}$ = 1.37

Q4: (A) : Show by sketch a double acting arrangement used in fixing processes ?

(B) : Write an equation or rule to estimate (FIVE) of the following:

1 - Depth of cut in cutting operation , 2 - Wahl's correction factor for spring design

3 – Shear stress across the threads , 4 – The transmitted torque when the power in KW

5 - Core diameter of the bolt , 6 - Strength of parallel fillet welded joints

Q 5 : Assuming that a machining conditions are constants as in below :

Intial diameter = 70 mm , final diameter = 30 mm , length of workpiece = 96 mm

Depth of cut = 2 mm , feed = 0.2 mm/rev .If the rotational speed (N) of the lathe central machine has the values : (100 , 200 , 400 , 800 , 1600 r.p.m) , Draw a chart to explain the relationship between the cutting speed and the cutting time.

Q6: A flat key 10 mm wide , 8 mm thickness and 75 mm long is required to transmit a torque of 10^6 N .mm from a 50 mm diameter shaft , investigate to determine whether the length is sufficient , use a design stress in shear of 50 N / mm^2 and in crushing of 130 N / mm^2, if the length is not sufficient redesign the length of the key ?

Q7 : Choose (A) or ((B) :

(A) : Find the efficiency of double lap joint of 6 mm plates with 2 cm diameter rivets having a pitch of 6.5 cm , the permissible tensile ,shearing and crushing are 1200 kg/cm^2,900 kg/cm^2 and 1800 kg/cm^2 .

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

Permissible shear stress = 3500 kg / cm^2 , Modulus of rigidity = 8.4 X 10^5 kg / cm^2

Q8 : Choose (A) or ((B) :

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

(B) : A pair of wheels of a railway wagon carries a load of (5 tonnes) on each axle box , acting at a distance of 100 mm out side the wheel base . The gauge of the rails is 1400 mm . find the diameter of the axle between the wheels , if the stress in not exceed 100 N/mm^2 .

GOOD LUCK

FOUNDATION OF TECHNICAL EDUCATION

NASSIRIYA TECHNICAL INSTITUTE / MECHANICAL DEPARTEMENT

Machines Elements -Final Exam -For 2nd Year Students

Time : 3 hrs **2nd Attempt 2006 / 2007** **Date : 6 / 9 / 2007**

NOTE : ANSWER ONLY SIX QUESTIONS (10 Marks for each question).

Q 1 : Explain (Five) the following elements by free-hand sketches only :

1 - Square thread , 2 – through bolt , 3 – Pan head rivet , 4 – double filleted lap weld joint ,
5 – cone clutch , 6 – Wood ruff key

Q 2 : Explain by drawing one of the following type of fixture & guides :

1- Back supported plate 2- Welded bar & spherical ring

Q3 A vertical screw with single start square threads 5 cm mean diameter and 1 cm pitch , is raise against a load of 550 kg . by means of hand wheel , the boss of which is threaded to act as a nut , The axial load is taken up by a thrust collar which supports the wheel boss and has a mean diameter of 6.5 cm , $\mu = 0.15$ for screw , $\mu = 0.18$ for the collar , and the tangential force applied by each hand to the wheel is 14 kg . Find the suitable diameter of the hand wheel .

Q4: A (8) speeds gear box , Z = 8 = $2 \times 2 \times 2$, maximum speed = 800 r.p.m , minimum speed = 40 r.p.m ,

A - Constructe the structural diagram and speed chart.

B – find the speeds between maximum and minimum speed . Take : $\sqrt[7]{20}$ = 1.53

Q 5 : Calculate the cutting power (Kw) which is required to turn a shaft made of structural steel under the following conditions :

Depth of cut = 5 mm , the feed = 0.4 mm/rev , number of revolution for machine = 190 r.p.m , the metal strength for cutting =160 kg/mm^2 , diameter of the shaft = 50 mm , Then find :

1- Electrical motor power if you know that the efficiency of machine 70% .

2- Amount of heat generated in cutting (Kw) .

Q6: A 20 h.p. , 960 r.p.m motor has a mild steel shaft of 4 cm diameter and the extension being 7.5 cm . The permissible shear and crushing stresses for the mild steel key are 560 kg/cm^2 and 1120 kg/cm^2 . Design the keyway in the motor shaft extension .

Q7 : Choose (A) or ((B) :

(A) : 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 :
σ_t = 100 N/cm^2 ; σ_s = 80 N/cm^2 ; σ_c = 160 N/cm^2

(B) : Design a helical compression spring for maximum load of 1000 N , with a deflection of 25 mm using the value of spring index as 5 , G = modulus of rigidity = 84000 N/mm2 , maximum permissible shear stress for spring wire is 420 N / mm2 .

Q8 : Choose (A) or ((B) :

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

(B) : Find the diameter of a solid steel shaft to transmit 25 horse power at 200 r.p.m. The ultimate shear stress for steel may be taken as 360 N/mm^2 and a factor of safely as 8 .
If a hollow shaft is to be used in place of the solid shaft find the inside and outside diameter . When the ratio of in to out diameter is 0.5 .

GOOD LUCK