وزارة التعليم العالي و البحث العلمي
هيئة التعليم التقني
التخصصات التكنولوجية
القسم/ تـاينقت الميكانيك

أسم المادة	السنة الدراسية	الساعات الأسبوعية		
		نظري	عملي	المجموع
أجزاء المكائن(Machine Parts)	الثانية	3	---------	3

هدف المادة: الممام الطالب بوظيفة أجزاء المكائن و علاقتها مع بعضها و أجراء بعض الحسابات التصميمية لهذه الأجزاء وتحديد تأثير كافة العوامل المؤشر عليها

Week No.	Theoretical Subjects
	Subject Topics
1	Review of Strength of Materials
2-3	Riveted Joints
4-5	Welded Joints
6-7	Screwed Joints
8-9	Keyed Joints
10-11	Frictional Clutches
12-13	Design of Springs
14-15	Design of Belts
16-17	Design of Shafts
18-19	Design of Journal Bearings
20	Selection of Ball Bearings
21-22	Design of Gears by Lewis Equation
23-24	Gears Trains
25-26	Design of Simple Gears Box
27-28	Worm Gears
29-30	Cams

References:-

1- Strength of Materials by Ferdinal L. Singer.
2- Strength of Materials by R.S. Khurmi.
3- Machine Design by R.S. Khurmi, J.K. Gupta.
4- Machine Design by Paul H. Black.
5- Schaum's Outline Series of Machine Design by Hall, Holowenko, Laughin .

Riveted Joints

Rivet is a short cylinder bar with a head integral to it, as shown in fig (1). The rivets are used to make permanent fasting between the plates such structural work , ship building ,tanks and boiler shell .

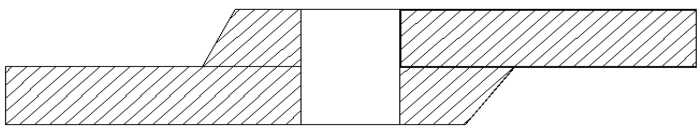
*The function of rivets in a joint is to make a connection that has strength and tightness.

*The material of the rivets must be tough and ductile. They are made of low carbon steel ,brass , aluminum or copper.

Types of Riveted Joints :-

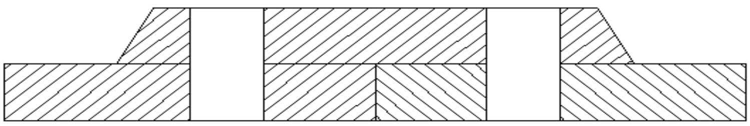
1- Lap joint:- a lap joint is that in which one plate overlaps the other and the two plates are then riveted together.

Single riveted Lap Joint

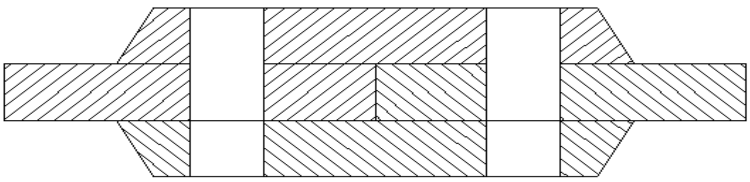


2- Butt Joint:- a butt joint is that in which the main plates are kept in alignment butting touching each other and a cover plate (strap) is placed either on one side or on both sides of the main plates.
The cover plate is then riveted together with the main plates. There are two types of the butt joint:-
   1- Single strap butt joint.
   2- Double strap butt joint.

Single Strap Double Riveted Butt Joint



Double Strap Double Riveted Butt Joint



Design of Riveted Joint :-

1- Tearing of the plate

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

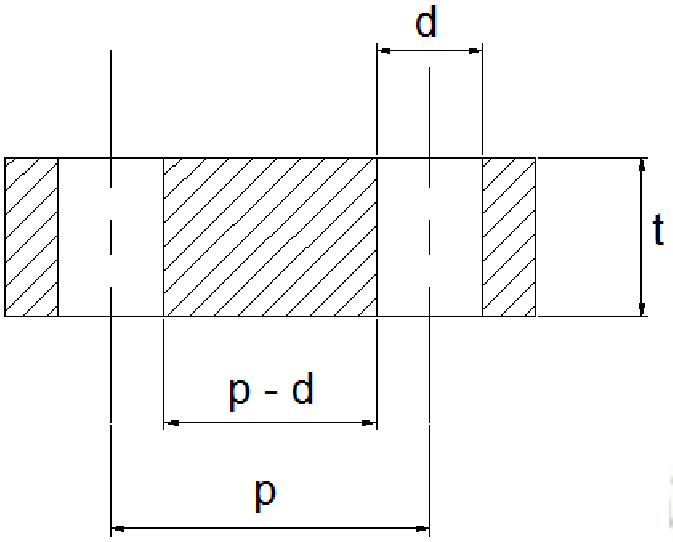
$$F_t = \sigma_t \times A_t$$

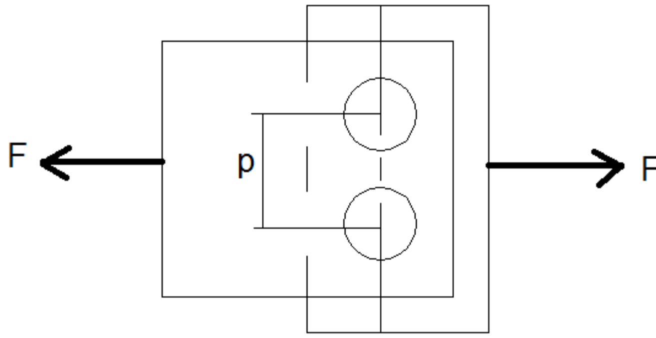
$$F_t = \sigma_t \times (p - d) \times t$$

σ_t= safe permissible tensile for the plate.

F_t=tensile force ,p=pitch of the rivets,A_t=tearing force,

d=rivet diameter hole, t=plate thickness , n= number of rivets per pitch length.





2- Shearing of the rivets

F_s = Shear force

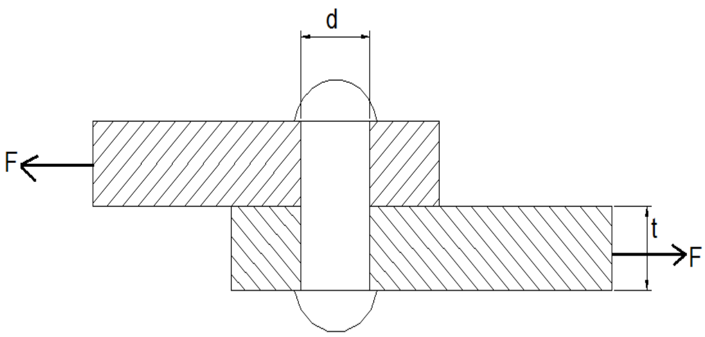
τ= Safe permissible shear stress

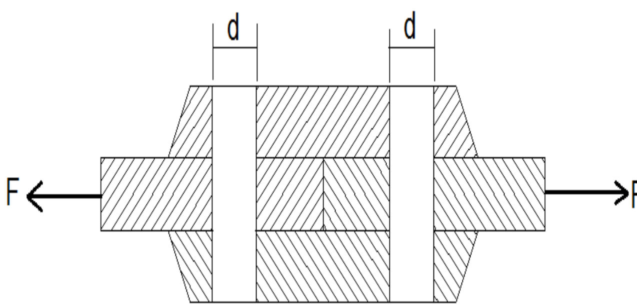
$$F_s = \tau \times A \times n$$

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

$$F_s = \tau \times A \times 2 \times n$$

$$= \tau \times \frac{\pi d^2}{4} \times 2 \times n \text{ (in double strap)}$$





3- Crushing of the rivets

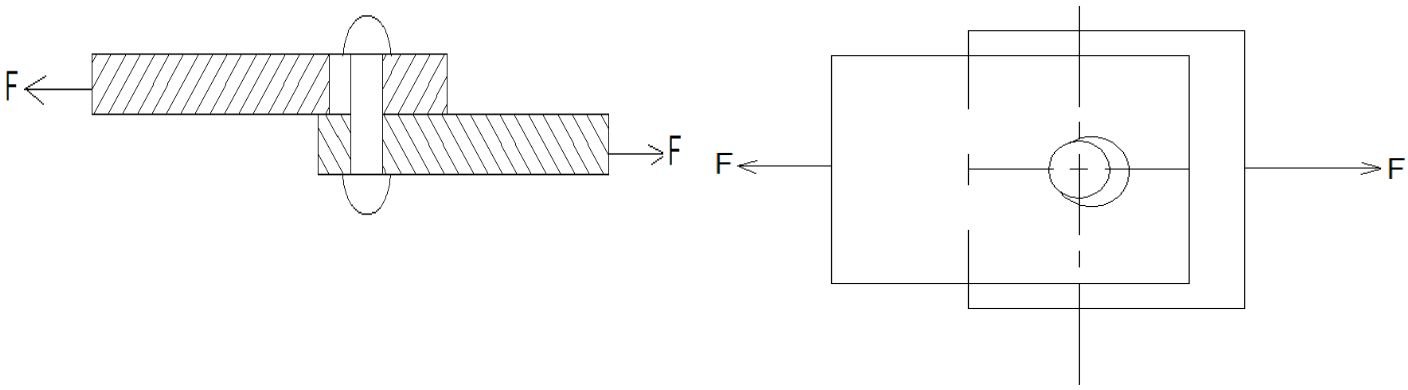
$$F_c = \sigma_c \times A_c \times n$$

$A_c = d \times t$

$F_c =$ Crushing force

$\sigma_c =$ Safe permissible crushing stress for the rivet.

$A_c =$Crushing Area



Efficiency of a riveted joint:-

$$\zeta = \frac{Least\ of\ F_t\ ,F_s\ and\ F_c}{F} \times 100\%$$

Where:-

$F = p \times t \times \sigma_t$

P=pitch of the rivet.

σ_t=permissible tensile stress of the plate material.

Design Procedure of Rivets Joints:-

To calculate the diameter of the rivets:-

a-IF t ≥ 8 mm then $d = 6\sqrt{t}$

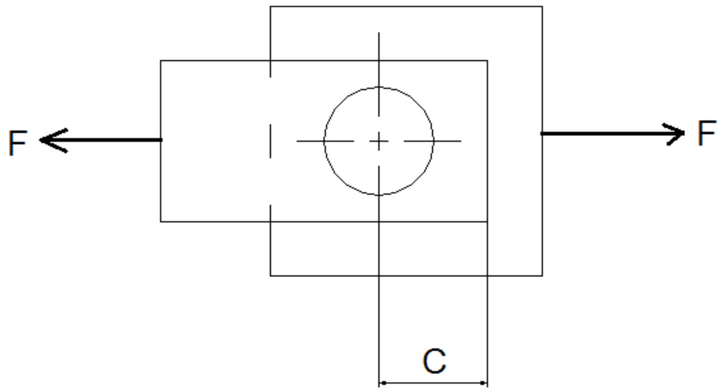
b- IF t < 8 mm then $F_s = F_c$

To calculate pitch (p)

$F_s = F_t$

To Calculate the distance between the edge of the plate and center of rivet (c).

$C = 1.5d$

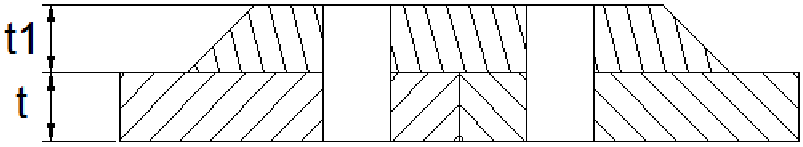


To calculate the cover of butt joints.

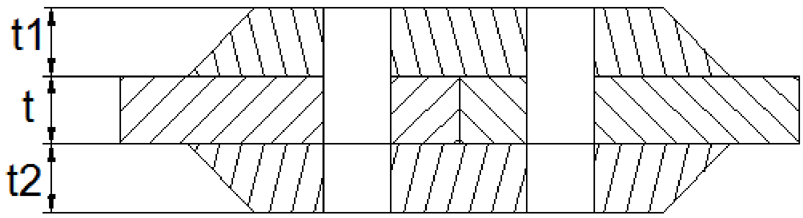
$t_1 = 1.25t$ (for single butt strap)

$t_2 = (0.6 - 0.8)$(for double butt strap when $t_1 = t_2$)

a



b



Exp1:- A double riveted lap joint is made from 15mm thickness .The rivet diameter and pitch are 25mm and 75mm respectively. If the ultimate stresses are 400 N/mm^2 in tensile and 320N/mm^2 in shear and 640N/mm^2 in crushing, Find the minimum force per pitch which will rupture the joint also find the efficiency of the rivet?

Solution:-

t=15mm,d=25mm,p=75mm,$\sigma_t = 400$N/mm^2,$\tau = 320$N/mm^2

σ_c=640N/mm^2

$$F_t = \sigma_t \times (p - d) \times t$$

$$= 400 \times (75 - 25) \times 15$$

$$= 300\ 000\ N$$

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

$$= 320 \times \frac{\pi(25)^2}{4} \times 2$$
$$= 314\ 200\ N$$
$$F_c = \sigma_c \times d \times t \times n$$
$$= 640 \times 25 \times 15 \times 2$$
$$= 480\ 000 N$$
$$\zeta = \frac{Least\ of\ F_t\ ,F_s\ and\ F_c}{F} \times 100\%$$
$$F = \sigma_t \times t \times p$$
$$= 400 \times 75 \times 15$$
$$= 450\ 000 N$$
$$\zeta = \frac{300\ 000}{450\ 000} \times 100\% = 66.6\%$$

Exp2:- A double riveted cover butt joint in plates 20mm thickness made with 25mm diameter rivets at 100 mm pitch. The permissible stresses are tension=120 N/mm^2,$shear = 100$N/mm^2 and crushing =150 N/mm^2.Find the efficiency of joint ,taking the strength of the rivet in double shear as twice than that of single shear?

Solution:-

t=20mm,d=25mm,p=100mm,$\sigma_t = 120$N/mm^2,$\tau = 100$N/mm^2

σ_c=150N/mm^2

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

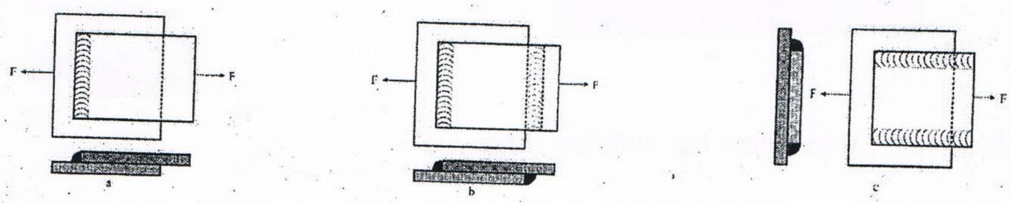
$$\mathrm{F_s = \tau \times \frac{\pi d^2}{4} \times 2 \times n}$$
$$= 100 \times \frac{\pi(25)^2}{4} \times 2 \times 2$$
$$= 196375\ N$$
$$F_c = \sigma_c \times d \times t \times n$$
$$= 150 \times 25 \times 20 \times 2$$
$$= 150\ 000 N$$
$$\zeta = \frac{Least\ of\ F_t\ ,F_s\ and\ F_c}{F} \times 100\%$$

$$F = \sigma_t \times t \times p$$
$$= 120 \times 100 \times 20$$
$$= 240\ 000N$$
$$\zeta = \frac{150\ 000}{240\ 000} \times 100\% = 62.5\%$$

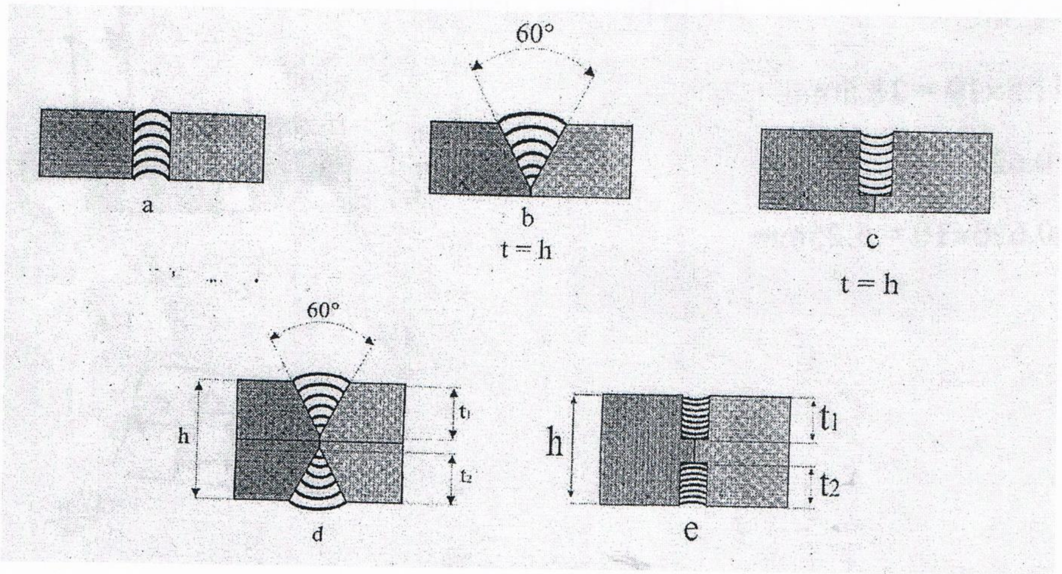
Welding Joint

Types of welding joint:-

1. Lap welding joint:
   a. Single transverse lap welding joint.
   b. Double transverse lap welding joint.
   c. Parallel lap welding joint.

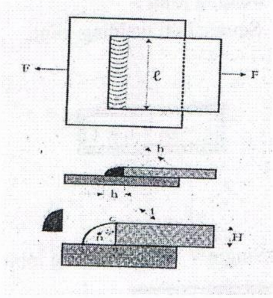


2. Butt welding joint:
   a. Square butt welding joint.
   b. Single V-butt welding joint.
   c. Single U- butt welding joint.
   d. Double V- butt welding joint.
   e. Double U - butt welding joint.



Design of welding joint:-

1. Lap welding joint:
   a. Single transverse lap welding joint.



$$t = 0.707 \times h$$

$$\boxed{F = 0.707 \times \sigma_t \times \ell \times h}$$

   b. Double transverse lap welding joint.

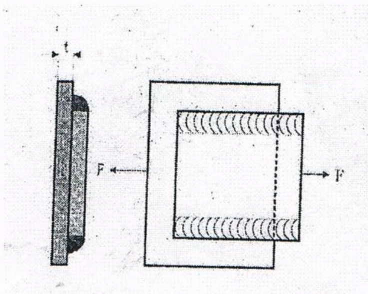
$$\boxed{F = 0.707 \times \sigma_t \times \ell \times h \times 2}$$

F	Load
t	Throat thickness
σ_t	Tensile stress
ℓ	Length of weld
h	Thickness of plate

   c. Parallel lap welding joint.

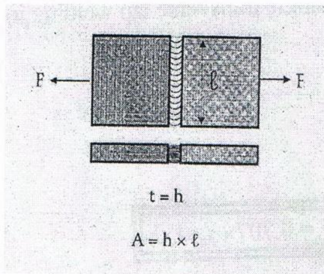
$$F = \tau \times A$$
$$F = \tau \times (t \times \ell)$$

$$\boxed{F = 0.707 \times \tau \times h \times \ell \times 2}$$



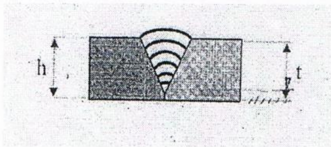
2. Butt welding joint :-

a. Square butt welding joint.



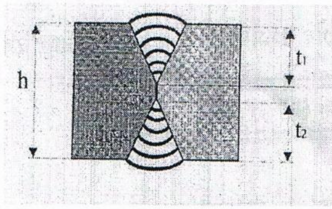
$$F = \sigma_t \times h \times \ell$$

b. Single V – butt welding joint.



$$F = \sigma_t \times h \times \ell$$

c. Double V – butt welding joint.



$$F = \sigma_t \times (t_1 + t_2) \times \ell$$

Ex. 1:-

Tow steel plate of 12.5mm thickness. Are to be welded by double transverse lap joint, Find the length of weld, assume allowable shear stress for the weld metal ($50N/mm^2$) and allowable tensile stress for the weld metal($80N/mm^2$) load =30kN.

Solution:-

$$F = 0.707 \times \sigma_t \times h \times \ell \times 2$$

$$kN = 1000N$$

$$30000 = 0.707 \times 80 \times 12.5 \times \ell \times 2$$

$$\ell = \frac{30000}{1414} = 21.21mm$$

Ex. 2:-

Tow plates of (10mm) thickness are to be joint by single transverse lap joint. Design the joint let shear stress for weld metal ($60N/mm^2$) tensile stress for weld metal ($110N/mm^2$) and external force for the weld metal (40 kN).

Solution:-

$$F = 0.707 \times \sigma_t \times h \times \ell$$

$$kN = 1000N$$

$$40000 = 0.707 \times 110 \times 10 \times \ell$$

$$\ell = \frac{40000}{777.7} = 51.43mm.$$

Ex. 3:-

Tow plates of (20mm) thickness are to be welded by double parallel lap joint take; shear stress ($60N/mm^2$), tensile stress ($105N/mm^2$) and external load (70KN).

Solution:-

$$F = \tau \times 0.707 \times h \times \ell \times 2$$

$$70000 = 60 \times 0.707 \times 20 \times \ell \times 2$$

$$\ell = \frac{70000}{1696.8}$$

$$\ell = 41.25mm$$

Ex. 4:-

Tow plates of (16mm) thickness are be welded by single transverse length and double parallel of (120mm) length lap joint. Find the external force assume; shear stress ($40N/mm^2$); tensile stress ($75N/mm^2$).

Solution:-

$$F_{total} = F_t + F_p$$

$$F_t = 0.707 \times \sigma_t \times h \times \ell$$

$$F_t = 0.707 \times 75 \times 16 \times 120$$

$$F_t = 101808N$$

$$F_p = 0.707 \times \tau \times h \times \ell \times 2$$

$$F_p = 0.707 \times 40 \times 16 \times 120 \times 2$$

$$F_p = 108595.2N$$

$$F_{total} = F_t + F_p$$

$$F_{total} = 101808 + 108595.2$$

$$F_{total} = 210403.2N$$

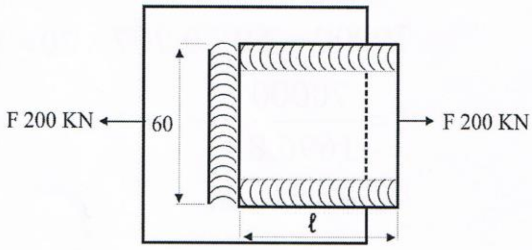
$$F_{total} = 210.4032KN$$

Ex. 5:-

مهم جدا

A plate (60mm) wide and (10mm) thick is jointed with another plate by single shown in fig. The maximum tensile and shear stresses are (100N/mm^2) and (60N/mm^2). Find the length of each parallel welds, if the joint is subjected to external load of (200KN).

Solution:-



$$F_{total} = F_t + F_p$$
$$F_t = 0.707 \times \sigma_t \times h \times \ell$$
$$F_t = 0.707 \times 100 \times 10 \times 60$$
$$F_t = 42420N$$
$$200000 = 42420 + F_p$$
$$F_p = 200000 - 42420$$
$$F_p = 157580N$$
$$F_p = 0.707 \times \tau \times h \times \ell \times 2$$
$$157580 = 0.707 \times 60 \times 10 \times \ell \times 2$$
$$\ell = \frac{157580}{848.4}$$
$$\ell = 185.73mm$$

Ex. 6:-

A spherical gas tank is made of (10mm) steel plate hemispheres butt welded together. The stress of weld material is ($50N/mm^2$). Design the welded joints if the maximum gas pressure is ($1N/mm^2$).

Solution:-

$$t = \frac{p \times d}{4 \times \sigma_t}$$

$$10 = \frac{1 \times d}{4 \times 50}$$

$$10 = \frac{d}{200}$$

$$d = 10 \times 200$$

$$d = 2000mm$$

$$\ell = \pi \times d$$

$$\ell = \pi \times 2000$$

$$\ell = 6283mm$$

Ex. 7:-

A cylinder steam boiler (1200mm) in diameter, generators steam at a gauge pressure of ($2N/mm^2$). Design the welded joints of the boiler if the tensile stress of weld material is ($60N/mm^2$).

Solution:-

$$t = \frac{p \times d}{4 \times \sigma_t}$$

$$t = \frac{2 \times 1200}{4 \times 60} = \frac{2400}{240}$$

$$t = 10mm$$

H.W

Ex. 8:-

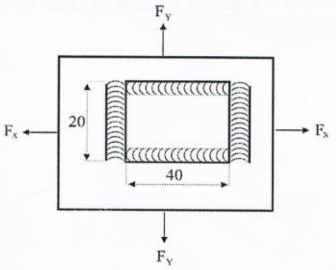
From the following fig. try to find the smallest force which destroy the weld and is direction.

$\tau = 40 N/mm^2$

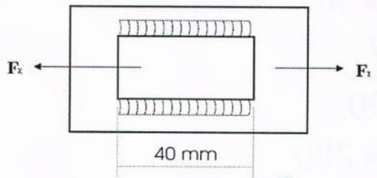
$\sigma_t = 80 N/mm^2$

$h = 10mm$

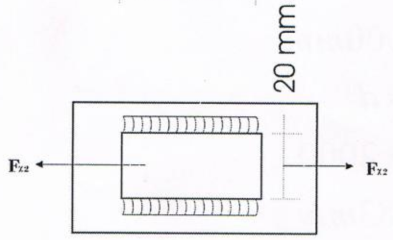
Solution:-



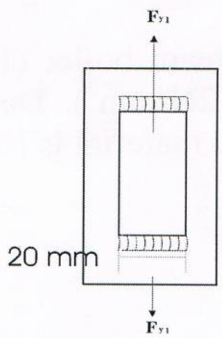
$$F_{x1} = 0.707 \times \tau \times \ell \times h \times 2$$
$$F_{x1} = 0.707 \times 40 \times 40 \times 10 \times 2$$
$$F_{x1} = 22624N$$



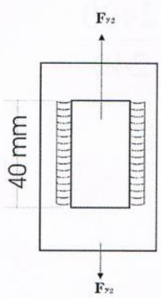
$$F_{x2} = 0.707 \times \sigma_t \times \ell \times h \times 2$$
$$F_{x2} = 0.707 \times 80 \times 20 \times 10 \times 2$$
$$F_{x2} = 22624N$$



$$F_{y1} = 0.707 \times \sigma_t \times \ell \times h \times 2$$
$$F_{y1} = 0.707 \times 80 \times 20 \times 10 \times 2$$
$$F_{y1} = 22624N$$



$$F_{y2} = 0.707 \times \tau \times \ell \times h \times 2$$
$$F_{y2} = 0.707 \times 40 \times 40 \times 10 \times 2$$
$$F_{y2} = 22624N$$



$$\therefore \Sigma F_x = F_{x1} + F_{x2}$$
$$22624 + 22624 = 45248N$$
$$\therefore \Sigma F_y = 22624 + 22624 = 45248N$$

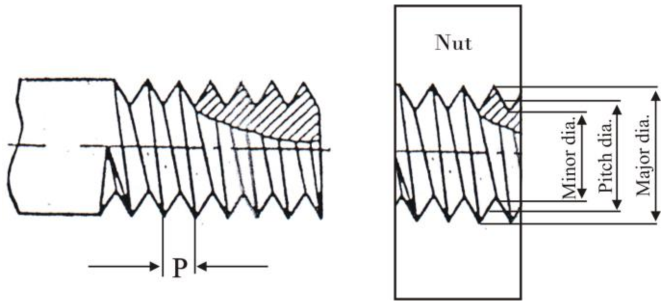
Screwed Joints

Design equation :-

$F_i = 2840 \times d$
$F = F_i + K \times F_e$
$d_c = 0.84 \times d$

M16 × 2
d = 16mm
p = 2mm

تمثيل نظام عالمي



d = Major (nominal) diameter mm.
d_p = pitch diameter mm.
d_c = Minor diameter (mm).
F_i = Initial tension force in bolt (N).
F_e =external load on bolt (N).
F = Resultant axial load on bolt (N).
K = Constant

$T = F \times r$ …… قانون العزم

Ex. 1:-

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 exceeding (100N/mm^2).

Solution :-

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

$$d_c = \sqrt{\frac{4\times F_t}{n\times\pi\times\sigma_t}}$$

d_c
d

$$d_c = \sqrt{\frac{4\times 60000}{1\times 3.14\times 100}} = \sqrt{\frac{240000}{314}} = \sqrt{764.3}$$

$$d_c = 27.6mm$$

$$d = \frac{d_c}{0.84}$$

$$d = \frac{27.6}{0.84}$$

$$d \cong 33mm$$

Ex. 2:-

Tow shaft are connect by of flange coupling to transmit torque of (280Kg.Cm). The flange of coupling is fastened by four bolt at radius of (4Cm). Design the bolt if allowable shear stress ($100Kg/Cm^2$).

Solution:-

$$T = F_s \times r$$

$$F_s = \frac{T}{r} \Rightarrow F_s = \frac{280}{4} = 70Kg$$

$$\tau = \frac{F_s}{\frac{\pi\times(dc)^2}{4}\times n}$$

$$d_c = \sqrt{\frac{4\times 70}{4\times\pi\times 100}} = \sqrt{\frac{280}{1256}} = \sqrt{0.222}$$

$$d_c = 0.47Cm$$

$$d = \frac{d_c}{0.84} \Rightarrow d = \frac{0.47}{0.84} = 0.56Cm$$

Ex. 3:-

A single plate clutch transmits (15 kW) at (1200 r.p.m). The clutch has (4 bolts) placed at (120mm), pitch circle diameter. Determine the suitable diameter of bolts if shear stress of bolt is (14 MN/m^2).

Solution :-

$$kW = 1000W$$

$$P = T \times \omega$$

$$\underbrace{15000(N.m/\sec)}_{Watt} = T\,(N.m) \times 1200 \times \frac{2\pi}{60}(rad/\sec)$$

$$15000 = T \times 1200 \times \frac{2 \times 3.14}{60}$$

$$15000 = T \times 125.6$$

$$T = \frac{15000}{125.6}$$

$$T = 119.3N.m$$

$$T = F_s \times r$$

$$119.3 = F_s \times \frac{\frac{120}{2}}{1000}$$

$$F_s = \frac{119.3}{0.06}$$

$$F_s = 1988.3N$$

$$\tau = \frac{F_s}{\frac{\pi \times (d_c)^2}{4} \times n}$$

$$d_c = \sqrt{\frac{4 \times 1988.3}{4 \times 3.14 \times 14}} = 6.72mm$$

$$d = \frac{dc}{0.84}$$

$$d = \frac{6.72}{0.84} = 8mm.$$

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.

Types of Keys:-

1- Sunk keys
2- Saddle keys
3- Tangent keys
4- Round keys
5- Splines

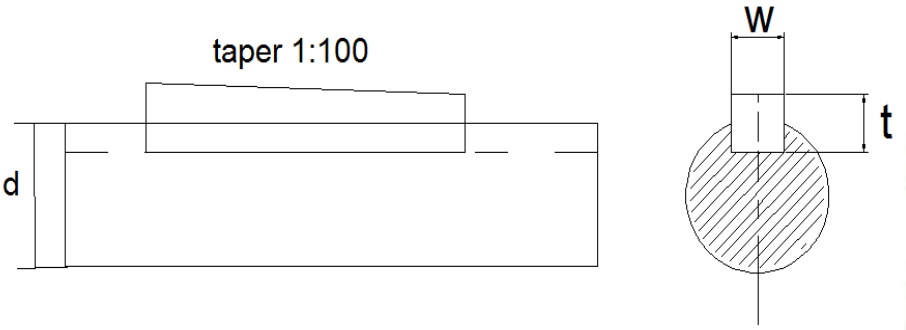
Sunk keys:

Types of sunk keys:-

a- Rectangular Sunk Key

$$W = \frac{d}{4}$$

$$t = \frac{2W}{3} = \frac{d}{6}$$



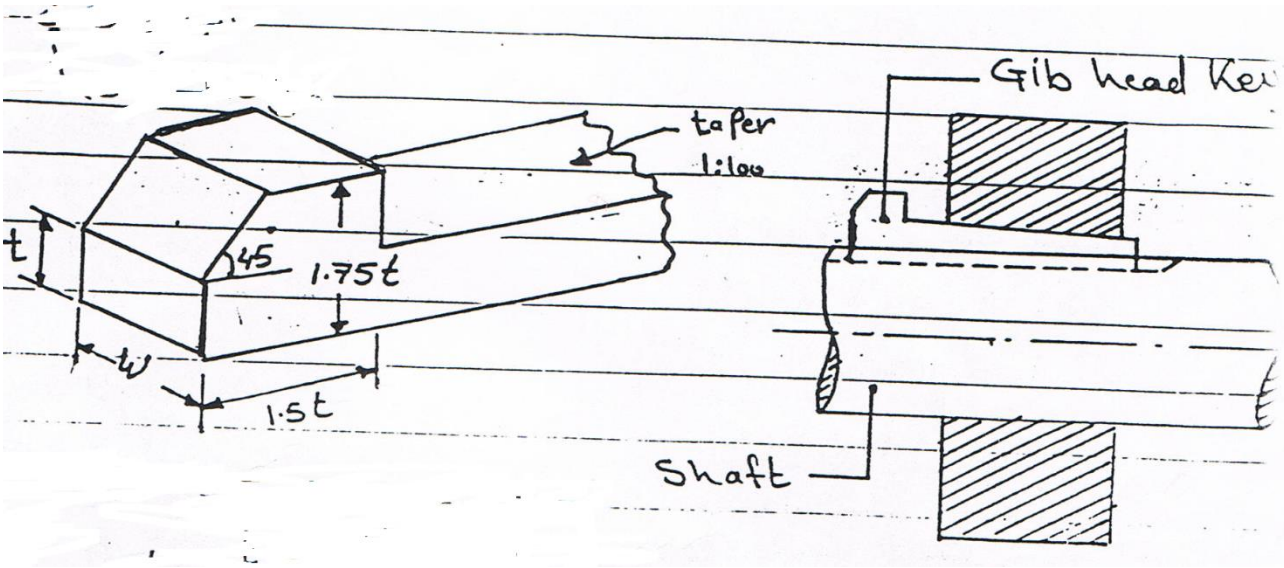
b- Square Sunk Key

$$W = t = \frac{d}{4}$$

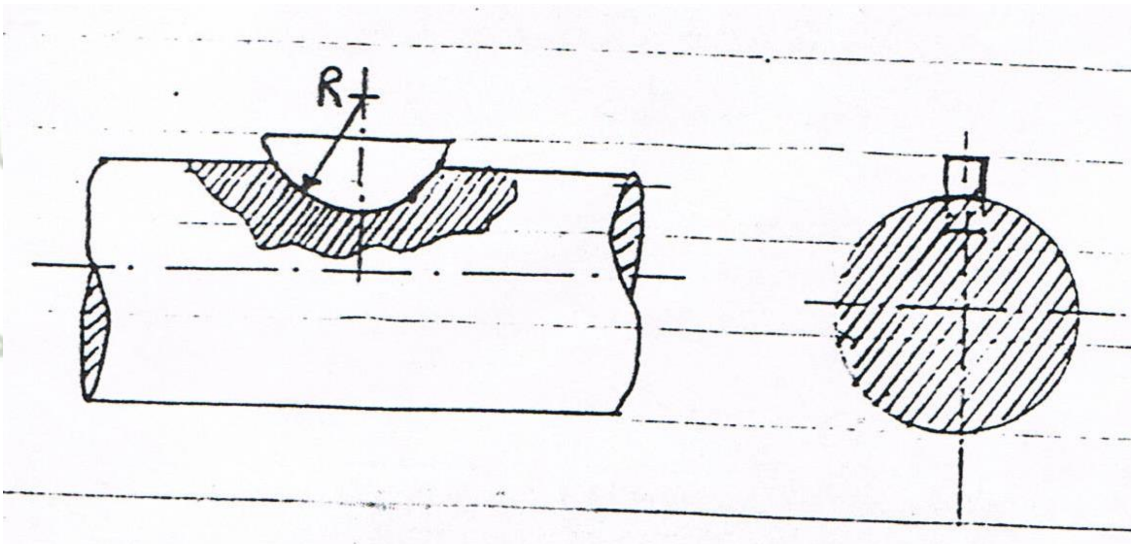
c- Parallel Sunk Key: It may be rectangular or square section but without taper (taperless)

d- Gib-head Key

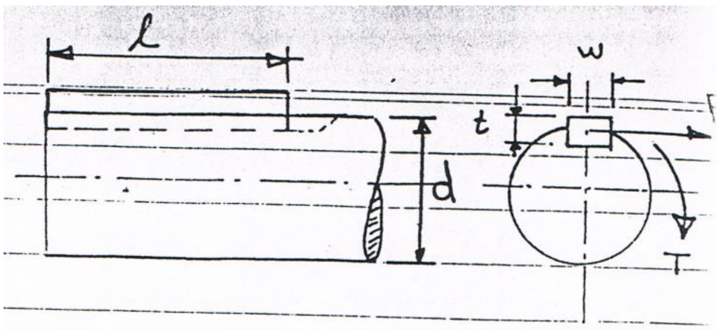
$$W = \frac{d}{4} \quad ,t = \frac{2W}{3} = \frac{d}{6}$$



e- Woodruff Key



Design of Sunk Key:-



$$T = F \times \frac{d}{2} ----------(1)$$

$$\tau = \frac{F}{W \times l}$$

$$T = \tau \times (W \times l) \times \frac{d}{2} ------(2)$$

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

$$T = \sigma_c \times \left(\frac{t}{2} \times l\right) \times \frac{d}{2} -------(3)$$

نساوي المعادلتين2 و 3 للحصول:-

$$\tau \times (W \times \cancel{l}) \times \cancel{\frac{d}{2}} = \sigma_c \times (\frac{t}{2} \times \cancel{l}) \times \cancel{\frac{d}{2}}$$

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

من علاقة عزم الالتواء مع أجهاد القص نحصل:-

$$T = \frac{\pi}{16} \times d^3 \times \tau - - - - - - - -(4)$$

$$\tau \times (W \times l) \times \frac{d}{2} = \frac{\pi}{16} \times d^3 \times \tau$$

$$l = \frac{\pi}{8} \times \frac{d^2}{W} - - - - - - - -(5)$$

نعوض في المعادلة5 العلاقة $W = \frac{d}{4}$ لنحصل على :-

$l = 1.57d$ (for Shearing Stress)

نساوي المعادلتين 3 و 4 كالاتي:-

$$\sigma_c \times \left(\frac{t}{2} \times l\right) \times \frac{d}{2} = \frac{\pi}{16} \times d^3 \times \tau$$

$$l = \frac{\pi}{4} \times \frac{d^2}{t} \times \frac{\tau}{\sigma_c} (for\ crushing\ stress)$$

Exp1:-Design the rectangular key for shaft of 50mm diameter . The shearing and crushing stresses for key materials are 42 Mpa and 70 Mpa.

Solution:- d=50mm.$\tau = 42$ N/mm^2,$\sigma_c = 70$N/mm^2

For rectangular key:

$$W = \frac{d}{4}$$

$$W = \frac{50}{4} = 12.5\ mm$$

$$t = \frac{2}{3}W = \frac{2}{3} \times 12.5 = 8.3\ mm$$

$l = 1.57d = 1.57 \times 50 = 78.9\ mm$ (for shearing stress length)

$$l = \frac{\pi}{4} \times \frac{d^2}{t} \times \frac{\tau}{\sigma_c}$$

$$l = \frac{\pi}{4} \times \frac{50^2}{8.3} \times \frac{42}{70}$$

$l = 141\ mm$

Exp2:- A 15kW ,960 rpm motor has a mild steel shaft of 40 mm diameter and the extension being 75 mm . The permissible shear and crushing stresses for mild steel key are 56 Mpa and 112 Mpa . Design the keyway in the motor shaft extension.

Solution:- P= 15$\times 10^3\ watt$, N=960 rpm ,d=40mm, l=75 mm,

$\tau = 56$ N/mm^2, $\sigma_c = 112$ N/mm^2.

$$P = T \times \omega$$

$$T = \frac{15 \times 10^3}{960 \times (^{2\pi}/_{60})} = 149\ N.m = 149\ \times 10^3 N.mm$$

$$T = \tau \times (W \times l) \times \frac{d}{2}$$

$$149\ \times 10^3 = 56 \times (W \times 75) \times \frac{40}{2}$$

$$W = \frac{149\ \times 10^3}{84 \times 10^3} = 1.77\ mm$$

$$W = \frac{d}{4} = \frac{40}{4} = 10\ mm$$

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

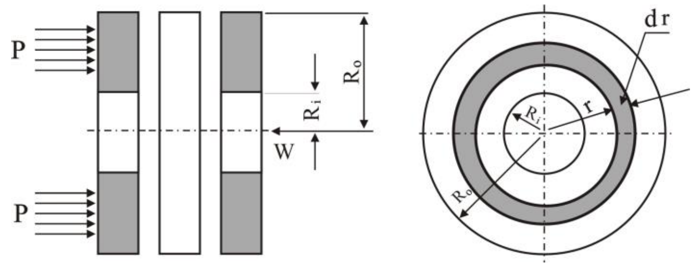
$$\frac{2 \times 56}{112} = \frac{t}{10}$$

t=10 mm

<u>Frictional Clutches</u>

<u>Types of clutches :-</u>

1. Disc clutch (single or multiple).
   a. Single disc clutch.



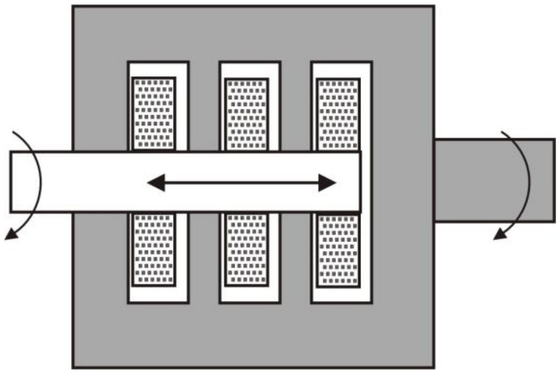
P : pressure.
W: Thrust axial load.
R_o , R_i : External & internal radius.
r : Mean radius of fraction surface.

$$P = \frac{W}{\pi(R_o{}^2 - R_i{}^2)}$$

$$r = \frac{2}{3}\left[\frac{R_o{}^3 - R_i{}^3}{R_o{}^2 - R_i{}^2}\right]$$ with uniform pressure ضغط متجانس

$$r = \frac{R_o + R_i}{2}$$ with uniform wear ضغط بلى (متماثل)

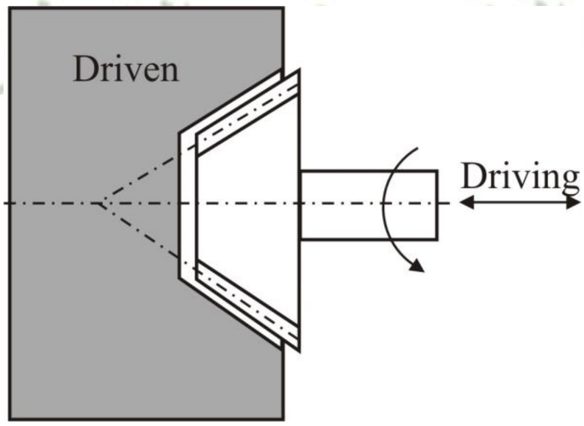
b. Multiple disc clutch.
In case of a multiple disc clutch let (n). The number of pairs of contact surface.



$$T = n \cdot \mu \cdot w \cdot r$$
$$N = n_1 + n_2 - 1$$

T : Torque transmitted be clutch.
n : number of pairs of contact surface.
n_1 : number of discs on the driving shaft.
n_2 : number of discs on the driven shaft.

2. Cone clutch.



__Ex. 1:-__

A multi-disc clutch has three disc on the driving shaft and tow on the driven shaft, The out side radius of the contact surface is 130 mm. and in side radius is 70 mm. assume the uniform ware and the coefficient of fraction is (0.1). Find the thrust force for the transmitting 5 k Watt for angular velocity ω 100 rad/sec.

__Data :-__

R_o = 130mm.	R_i = 70mm.	μ = 0.1
w = ?	P = 5×10^3 watt	ω = 100 rad/sec.

__Solution :-__

$$n = n_1 + n_2 - 1$$

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

$$r = \frac{R_o + R_i}{2}$$

$$r = \frac{130+70}{2} = 100mm.\left[for\ uniform\ ware\right]$$

$$T = n \times \mu \times w \times r$$

$$P = T \times \omega$$

$$T = \frac{P}{\omega} = \frac{5000}{100} = 50\ \text{N.M} \Rightarrow 50\times10^3\ \text{N.}mm.$$

$$T = n \times \mu \times w \times r$$

$$50\times10^3 = 4\times0.1\times w \times 100$$

$$w = 1250\ N$$

Ex. 2:-

Design a clutch to transmit 10kW at 1000 r.p.m. The ratio between the outside and inside radius of the contact surfaces is 2, The coefficient of the fraction is 0.3 and thrust force is 500N.

Solution:-

$$kW = 1000W$$

$$P = T \times \omega$$

$$T = \frac{P}{\omega} = \frac{10000 \times 60}{1000 \times 2\pi} = 95.5 N.m$$

$$T = n \times \mu \times W \times r$$

$$95.5 = 1 \times 0.3 \times 500 \times r$$

$$r = \frac{95.5}{150} = 0.636 m$$

$$R_o = 2R_i \text{ ....(1)}$$

$$r = \frac{2}{3} \times \frac{R_o{}^3 - R_i{}^3}{R_o{}^2 - R_i{}^2}$$

$$0.636 = \frac{2}{3} \times \frac{(2R_i)^3 - R_i{}^3}{(2R_i)^2 - R_i{}^2}$$

$$0.636 = \frac{2}{3} \times \frac{8R_i{}^3 - R_i{}^3}{4R_i{}^2 - R_i{}^2}$$

$$0.636 = \frac{2}{3} \times \frac{R_i{}^3 (8-1)}{R_i{}^2 (4-1)}$$

$$0.636 = \frac{2}{3} \times \frac{7}{3} R_i$$

$$0.636 = \frac{14}{9} R_i$$

$$R_i = \frac{0.636}{1.555} = 0.409 m$$

$$R_o = 2R_i$$

$$R_o = 2 \times 0.409 = 0.817 m$$

Ex. 3:-

A clutch having two pairs plate is required to transmit 110kW at 1250 r.p.m. The outer diameter of the contact surface is to be 300mm. The coefficient of fraction is (0.4). Assume a uniform pressure of (0.17 N/mm²), determine the inner diameter of the fractions surfaces.

Solution:

$$\underset{Power}{P} = T \times \omega$$

$$T = \frac{P}{\omega} = \frac{110000}{1250 \times \frac{2\pi}{60}} = \frac{110000}{130.8} = 840 N.m = 840 \times 10^3 N.mm$$

$$\underset{\text{Pressure}}{\text{P}} = \frac{W}{\pi (R_o{}^2 - R_i{}^2)}$$

$$W = P \times \pi (R_o{}^2 - R_i{}^2)$$

$$W = 0.17 \times 3.14 (150^2 - R_i{}^2) \text{..........}(1)$$

$$r = \frac{2}{3}\left[\frac{R_o{}^3 - R_i{}^3}{R_o{}^2 - R_i{}^2}\right]$$

$$r = \frac{2}{3}\left[\frac{(150)^3 - R_i{}^3}{(150)^2 - R_i{}^2}\right] \text{..........}(2)$$

$$T = n \times \mu \times W \times r$$

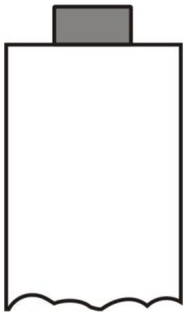
$$840000 = 2 \times 0.4 \times 0.17 \times 3.14 (150^2 - R_i{}^2) \times \frac{2}{3}\left[\frac{(150)^3 - R_i{}^3}{(150)^2 - R_i{}^2}\right]$$

$$840000 = 0.285 \times (150^3 - R_i{}^3)$$

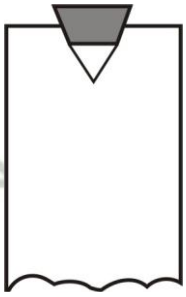
$$R_i = 70\, mm.$$

Belts Design

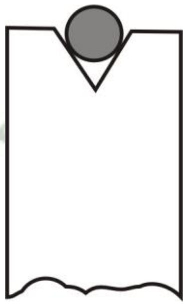
1. Types of belts :-
   a. Flat belts :- السير العدل يستخدم لنقل القدرات الواطئة لمسافات كبيرة



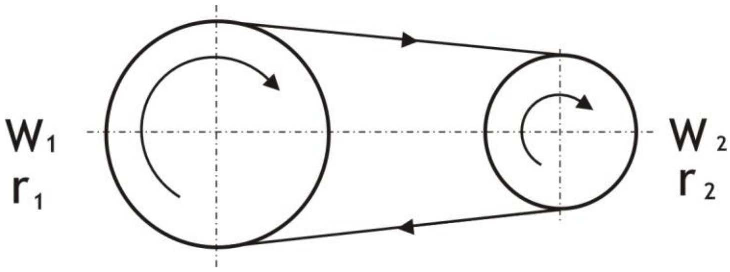
   b. V – belts :- هذا السير يستخدم لنقل القدرات العالية ولمسافات قصيرة



   c. Circle belts :- السير الدائري يستخدم لنقل القدرات الواطئة ولمسافات قليلة



2. Velocity ratio :-



$$v = w \times r \ (m / sec.)$$

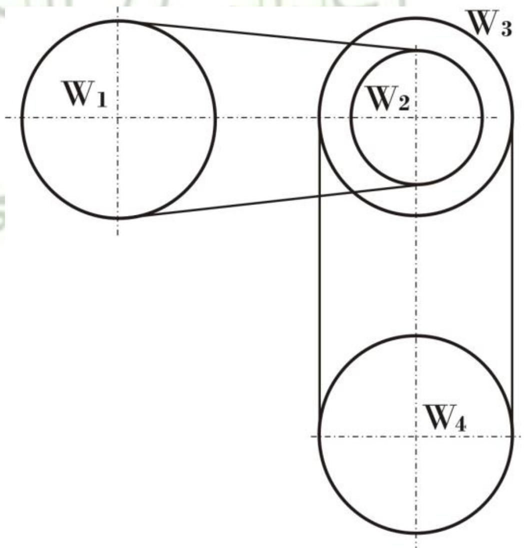
W_1	= Speed of driver
W_2	= Speed of driven
r	= radius

$$\frac{W_1}{W_2} = \frac{r_2}{r_2}$$

$$\frac{W_1}{W_2} = \frac{d_2}{d_2}$$

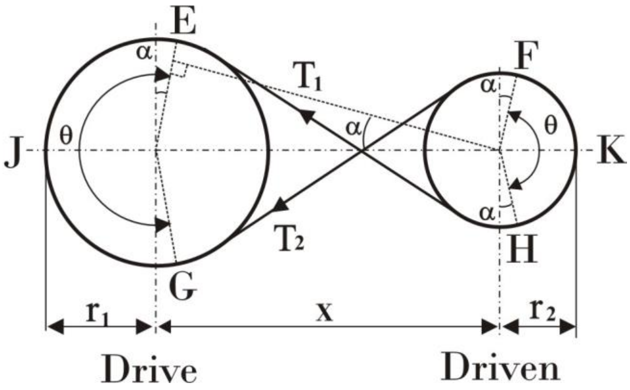
$$\frac{W_1}{W_2} \times \frac{W_2}{W_3} \times \frac{W_3}{W_4} = \frac{d_2}{d_1} \times \frac{d_3}{d_2} \times \frac{d_4}{d_3}$$

$$\frac{W_1}{W_4} = \frac{d_4}{d_1}$$



3. Length of belt :-

a. Cross

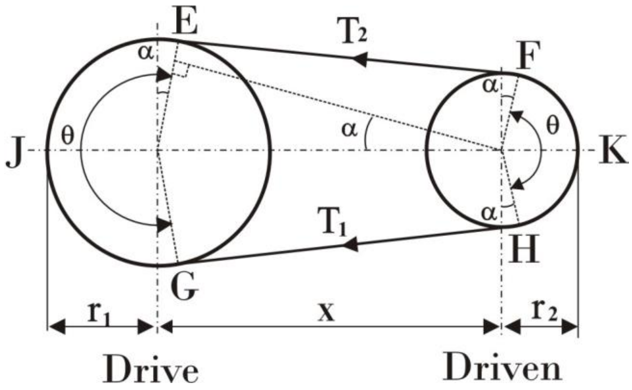


$$\sin\alpha = \frac{r_1 + r_2}{\chi}$$

$$\theta = (\pi + 2\alpha) \ldots\ldots radian\ (rad)$$

$$L = \left[\pi(r_1 + r_2) + 2\chi + \frac{(r_1 + r_2)^2}{\chi}\right]$$

b. Open



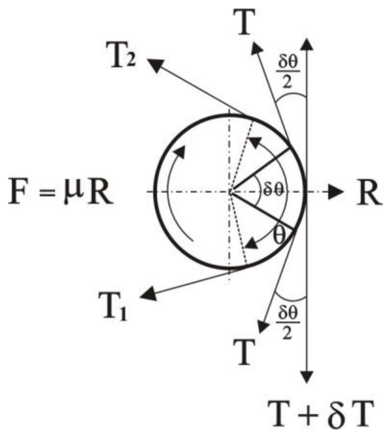
$$\sin\alpha = \frac{r_1 - r_2}{\chi}$$

$$\theta = (\pi - 2\chi) \ldots\ldots radians$$

$$L = \left[\pi\left(r_1 + r_2\right)2\chi + \frac{(r_1 - r_2)^2}{\chi}\right]$$

L = belt length.
r_1, r_2 = radius of pulleys.
α = contact angle between belt & pulley.
χ = distance between the center of tow pulleys.

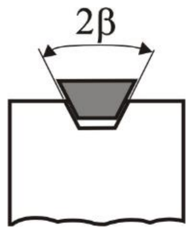
4. Ratio of tensions :-



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

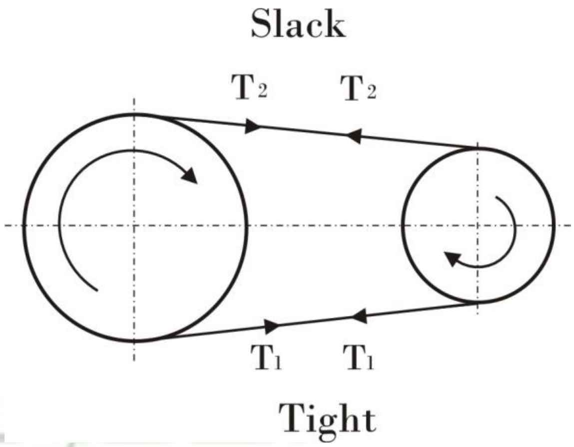
T_1 = tension in belt in tight side.
T_2 = tension in belt in slack side.

For V – belt :-



$$\frac{T_1}{T_2} = e^{\left(\frac{\mu.\theta}{\sin\beta}\right)}$$

5. Power transmitted by belt :-



$$v = \omega \times r \ (m/\text{min}t)$$

$$v = \frac{\pi \times D \times \omega}{60} \ (m/s)$$

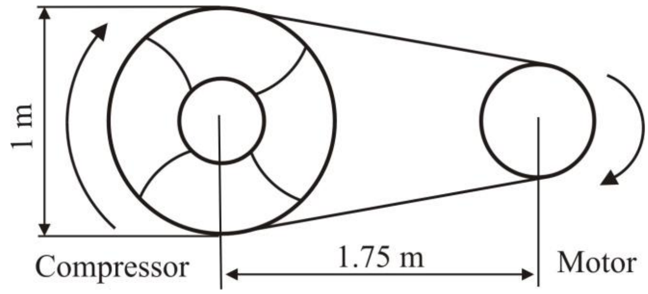
$$P = (T_1 - T_2) \times v \ (watt)$$

v = velocity m/s السرعة
P = Power القدرة بالواط

Ex. 2:-

A compressor required 25 h.p. is turn at 250 r.p.m. the drive is by motor running at 750 r.p.m. The diameter of the pulley on the compressor shaft is (1m) while the center distance between the pulleys is 1.75 m. Determine the number of V-belts required to transmit the power if the tension on tight side is (66.6 kg) the groove angle of the pulley is $\frac{35^\circ}{2\beta}$ and $\mu = 0.25$.

Solution :-



$$\frac{\omega_1}{\omega_2}=\frac{d_2}{d_1}$$

$$\frac{250}{750}=\frac{d_2}{1}$$

$$d_2=0.333\,m$$

$$\alpha=\frac{r_1-r_2}{\chi}=\frac{0.5-0.166}{1.75}=0.19\,rad$$

$$\theta=(\pi-2\times0.19)\,for\ small\ \ pulley\ [rad]$$

$$\theta=2.76\,rad$$

$$\frac{T_1}{T_2}=e^{\left[\frac{\mu\theta}{\sin\beta}\right]}$$

$$\frac{66.6}{T_2}=e^{\left[\frac{0.25\times2.76}{\sin1.75\times\frac{\pi}{180}}\right]}$$

$$T_2=9.97\,kg$$

$$P=(T_1-T_2)v$$

$$v=\omega.r\,[we\ take\ any\ \ pulley\ to\ get\ v]$$

$$For\ l\arg:$$

$$v=250\times\frac{2\pi}{60}\times0.5$$

$$v=13.08\,m/\sec.$$

$$P=(66.6-9.97)\times13.08$$

$$P=740.72\frac{kg.m}{\sec}watt\Rightarrow\frac{740.72}{75}=9.87\,h.p.$$

$$No.of\ V-belts=\frac{Total\ \ power}{Powe\ \ per\ belt}=\frac{25}{9.87}=2.53=3belts$$

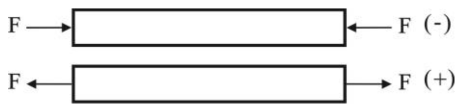
<u>Design Of Shaft</u>

1. Stresses :-
   a. Normal stresses (σ)
   b. Shear stresses (τ)

a. Normal stresses (σ)

Axial stress

Bending stress

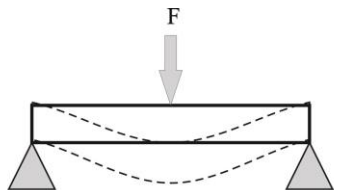


$$\sigma = \mp \frac{F}{A}$$

$$\sigma_b = \frac{BM}{Zb}$$

$$Zb = \frac{\pi}{32} \times d^3$$

$$\sigma_{Combined} = \mp \frac{F}{A} + \frac{BM}{Zb} \;...... (1)$$



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

$$\sigma_b = \frac{BM \times y}{I}$$

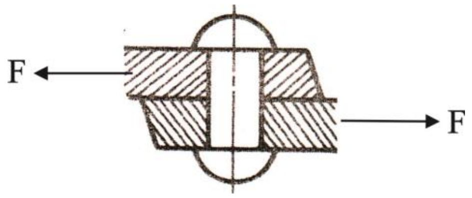
$$I = \frac{\pi}{64} \times d^4$$

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

$\sigma_{Combined}$	الإجهاد الاعتيادي الكلي
BM	Bending Moment عزم الانحناء
I	$\frac{\pi}{64} \times d^4$ عزم القصور الذاتي للدائرة
Zb	ثابت مقطع الشكل الدائري لعزم الانحناء

b. Shear stresses (τ)

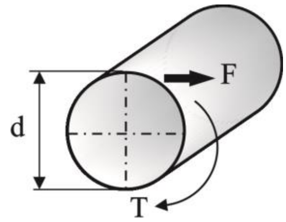
Transverse Shear Stress | Torsional Shear Stress



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

$$\tau_{Total} = \sqrt{\tau_{tr}{}^{2} + \tau_{t}{}^{2}} \;......(2)$$

$$\tau_{Comparison} = \sqrt{\frac{(\sigma_{Combined})^{2}}{2} + \tau_{Total}{}^{2}} \;......(3)$$



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

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

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

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

$$Z_{t} = \frac{\pi}{16} \times d^{3}$$

$$\tau = \frac{T}{Z_{t}}$$

τ_{Total}	إجهاد القص الكلي
τ_{tr}	إجهاد القص الاعتيادي (المستعرض)
τ_{t}	إجهاد القص المصاحب لعزم الـ Torsional shear stress
$\tau_{Comparison}$	إجهاد قص المقارنة
τ_{Total}	إجهاد القص الكلي

Ex. 1:-

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

Data:-

$P = 1\times10^{6}$ watt $\omega = 240$ r.p.m. $\tau_{max} = 60 N/mm^2$

Solution:-

$$\sigma_{Combind} = \mp\frac{F}{A}\mp\frac{BM}{Zb} \ldots\ldots (1)$$

$$\tau_{Total} = \sqrt{\tau_{tr}^{\ 2}+\tau_{t}^{\ 2}}$$

$$\tau_{Comparsion} = \sqrt{\frac{(\sigma_{Combind})^2}{2}+\tau_{Total}^{\ 2}}$$

$$\tau_{Comparsion} = \tau_{Total} = \frac{T}{Zt} = \frac{16T}{\pi\times d^3}$$

$$P = T\times\omega$$

$$T_{Mean} = \frac{P\times60}{2\pi\times\omega}$$

$$T_{Mean} = \frac{1\times10^6\times60}{2\times3.14\times240} = 39788\, N.M$$

$$T_{Max.} = T_{Mean}\times1.2$$

$$T_{Max.} = 39788\times1.2 = 47745.6\, N.M$$

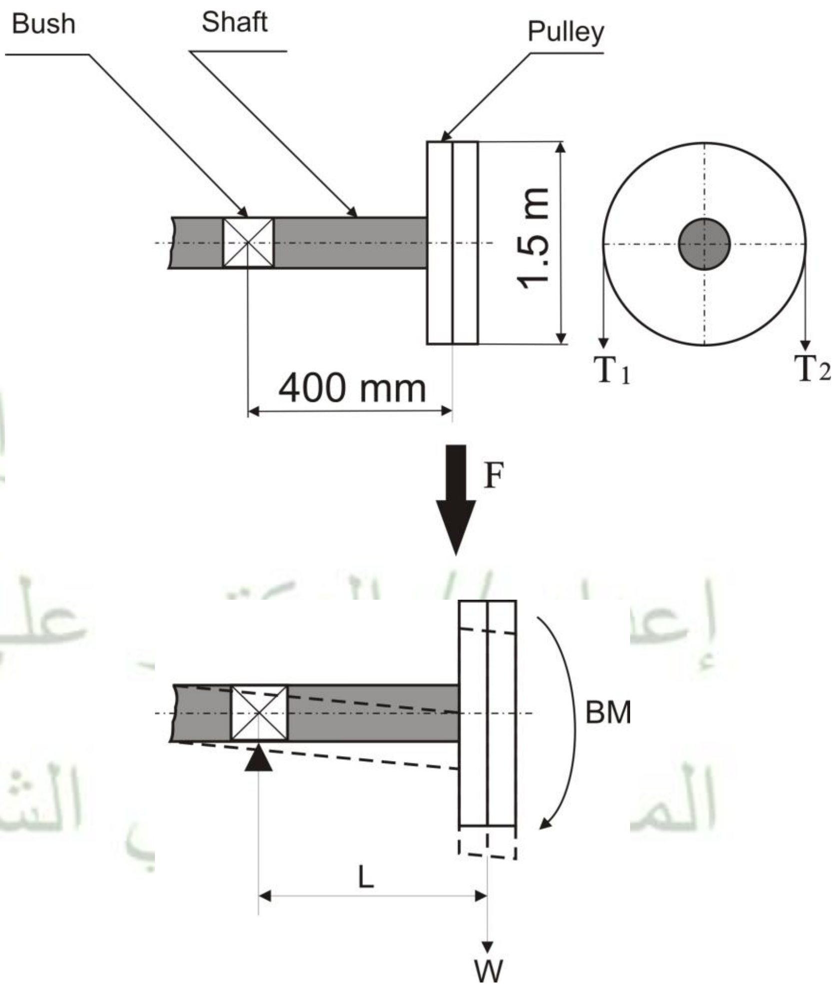
$$\tau_{Comparsion} = \frac{16\times T_{Max.}}{\pi\times d^3}$$

$$60 = \frac{16\times47745.6\times10^3}{3.14\times d^3}$$

$$d = \sqrt[3]{\frac{16\times47745.6\times10^3}{3.14\times60}} = 160mm$$

Ex. 2:-

A line shaft is driven by means of motor placed vertically below it . The pulley on the line shaft is (1.5m) in diameter and has belt tension (5.4 kN) and (1.8kN) on the tight side and slack side of the belt respectively . Both these tensions may be assumed vertical. If the pulley be overhang from the shaft , The distance of the center line of the pulley from the center line of the bearing being (400mm). Find the diameter of the shaft . Assuming maximum allowable shear stress of (42 N/mm^2).



Solution:-

Torque transmitted by the shaft

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

$$R = \frac{1.5}{2} = 0.75\, m$$

$$T = (5400 - 1800) \times 0.75 = 2700\, N.M \Rightarrow 2700.000 N.MM$$

Total vertical load (إهمال الوزن)

$$W = T_1 + T_2$$

$$W = 5400 + 1800 = 7200\ N$$

$$BM = W \times L$$

$$BM = 7200 \times 400 = 2880.000\ N.mm$$

$$\sigma_{Combined} = \mp \frac{F}{A} \mp \frac{BM}{Zb} \ldots\ldots (1)$$

$$\tau_{Total} = \sqrt{\tau_{tr}{}^2 + \tau_t{}^2} \ldots\ldots (2)$$

$$\tau_{Comparison} = \sqrt{\left(\frac{\sigma_{Combind}}{2}\right)^2 + \tau_{Total}{}^2} \ldots\ldots (3)$$

$$\sigma_{Combined} = \frac{32 \times 2880.000}{\pi \times d^3} \ldots\ldots (1)$$

$$\tau_{Total} = \sqrt{\left(\frac{F}{A}\right)^2 + \left(\frac{T}{Zt}\right)^2}$$

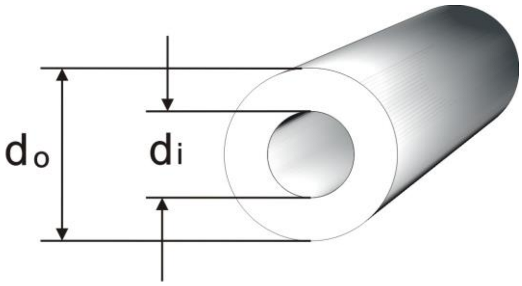
$$\tau_{Total} = \sqrt{\left(\frac{7200 \times 4}{\pi \times d^2}\right)^2 + \left(\frac{2700.000 \times 16}{\pi \times d^3}\right)^2} \ldots\ldots (2)$$

$$\tau_{Comparsion} \times 42 = \sqrt{\left(\frac{32 \times 2880.000}{2 \times \pi \times d^3}\right)^2 + \left(\frac{7200 \times 4}{\pi \times d^2}\right)^2 + \left(\frac{2700.000 \times 16}{\pi \times d^3}\right)^2}$$

By trial and error we get :
If d = 80 mm.
$42 \neq 39$
If d = 78 mm.

Ex. 3:-

Solve exp.(1) , If a hallow shaft is to be used in place of solid shaft . The ratio of inside to outside diameter is (0.5).



Solution:-

$$\tau_{Comparison} = \tau_{Total} = \tau_t = \frac{T}{Zt}$$

$$\tau_{Comparison} = \frac{T}{\frac{\pi \times d_o{}^2}{16} \times (1 - K^4)}$$

$$60 = \frac{16 \times 47745.6 \times 10^3}{\pi d_o{}^3 (1 - 0.5)^4}$$

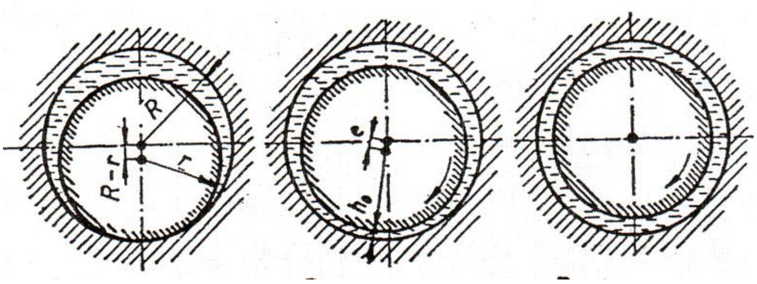
$$d_o = \sqrt[3]{\frac{763929600}{11.775}}$$

$$d_o = 401.8$$

$$d_i = 0.5 \times d_o$$

$$d_i = 200\, mm$$

Design of Journal Bearing



Data should be given :-

1. Type of machine.
   a. Shaft diameter.
   b. speed of shaft.
   c. Load.

2. Data should be selected :-
   a. Type of oil.
   b. Operating temperature of oil.

3. Outer data :-
   a. Bearing length.
   b. Bearing diameter.

❖ Design procedure :-

1. From table with the type of machine choosing (L/d).
2. From point (1) with the value of (d) find bearing length (L).
3. Check the value of bearing length (L) from the following point:-
   a. Determine the bearing pressure from the following equation :

$$\underset{Pressure}{P} = \frac{\overset{Load}{W}}{\underset{Length}{L} \times \underset{Diameter}{d}}$$

   b. If the value of bearing pressure in point (a) with in the range of values of bearing pressure in table, then the procedure is ok. If no try to choose another (L/d) in point (1).

4. From table with the lubricant oil and its operating temperature, get the value of viscosity (Z).

5. Calculate the value of $\dfrac{ZN}{P}$

6. $\left(\dfrac{ZN}{P}\right)_{Cal}$ should be larger than $\dfrac{1}{3}\left(\dfrac{ZN}{P}\right)_{tab.}$

$$\left(\frac{ZN}{P}\right)_{cal.} > K$$

where K = modulus of bearing

$$\therefore \text{ K} = \frac{1}{3}\left(\frac{ZN}{P}\right)_{tab.}$$

7. If equation in step (6) is ok, find the value of (c/d) from table, then find the clearance (c)

$$d_{bearing} = d_{shaft} + 2c$$

if equation in step (6) is not ok, then go to step (4) to change the type of oil and its operating temperature.

Ex. 1:-

Design a journal bearing for a generator from the following data :-

1. Load on the journal = 1500 Kg.
2. Diameter of the shaft = 10 Cm.
3. Speed = 1000 r.p.m.
4. Type of oil = SAE 10

Solution:-

1. $\frac{L}{d} = 1.5$

2. $L = 1.5 \times d_{shaft}$

$L = 1.5 \times 10 = 15\,Cm$

3. $P = \frac{W}{L \times d} = \frac{1500}{15 \times 10} 10\,Kg$ $Since\ the\ bearing\ pressure\ in\ table$

$(7 - 14)\,Kg / Cm^2\ therefore \ldots it\ is\ save.$

4. $SAE\ 10\ \&\ assume\ t = 50\,C^{\circ}$

$then\ Z = 21\,centi - poise$

5. $\left(\frac{ZN}{P}\right)_{cal.} = \frac{21 \times 1000}{10} = 2100$

6. $from\ table \left(\frac{ZN}{P}\right)_{tab.} = 2800$

$\therefore K = \frac{2800}{3} = 933.3$

7. $Since \left(\frac{ZN}{P}\right)_{cal.}\ 2100\ is\ more\ than\ K\ 933.3\ therefore \ldots\ldots it\ is\ save.$

$C / d = 0.0013$

$C = 0.0013 \times 10 = 0.013$

$d_{bearing} = d_{shaft} + 2C$

$= 10 + 2 \times 0.013 = 10.026\,Cm$

Ex. 2:-

Design a journal bearing for four stork, main bearing from the following :-

Data:-

1. Load on the journal = 1800 Kg
2. Diameter of the shaft = 7 Cm.
3. Speed of the journal = 3600 r.p.m.

Solution:-

1. $L/d = 1$
2. $\frac{L}{d} = 1 \Rightarrow \frac{L}{7} = 1$
3. $P = \frac{W}{L \times d} = \frac{1800}{7 \times 7} = 36.7\ Kg/Cm$ *Since the bearing pressure in table* $(50-85)\ Kg$ *...therfore it is not save.*

1. $Let\ L/d = 0.6$
2. $\frac{L}{7} = 0.6 \Rightarrow L = 4.2\,Cm.$
3. $P = \frac{W}{L \times d} = \frac{1800}{4.2 \times 7} = 61.22\ Kg/Cm^2$ *....Therefore it is save*
4. $SAE\ 30\ \&\ assume\ t = 50C^{\circ} \Rightarrow then\ Z = 48$ centipoises
5. $\left(\frac{\mathrm{ZN}}{\mathrm{P}}\right)_{cal} = \frac{48 \times 3600}{61.22} = 2822.6$
6. $From\ table\ \frac{ZN}{P} = 280$

   $\therefore K = \frac{1}{3} \times 280 = 93.3$
7. $Since \left(\frac{ZN}{p}\right)_{cal} > K \Rightarrow therefore\ it\ is\ save$

   $\frac{C}{d} = 0.001$

   $C = 0.001 \times 7 = 0.007$

   $d_{bearing} = d_{shaft} + 2C$

   $= 7 + 2 \times 0.007 = 7.014\,Cm.$

Selection of Ball Bearings

Design Procedure :-

The following data are known

1- Radial load (R).
2- Thrust load (T).
3- Speed of Shaft (N).
4- Desired life of bearing (L).
5- Departure.

1-Determine the rated radial load (C) from the following equation :-

$$C = \frac{RF}{F_s}\sqrt[4]{\frac{L}{Q}}$$

Where

F= Thrust factor.

F_s= Speed factor.

$(\frac{L}{Q})^{\frac{1}{4}}$ = Life factor

Q=2280 for NDur 300

= 4560 for NDur 600

2- By using rated radial load, the bearing number(basic number) may be selected from tables

3- By using the bearing number, the dimensions of ball bearing are known from table.

Exp1:-

Select a ball bearing with the following data:-

R=250 Ib , T = 50 Ib ,N= 3600 rpm , L= 16000 Hr, NDur 300

1-

$$C = \frac{RF}{F_s}\sqrt[4]{\frac{L}{Q}}$$

From table with T/R=50/250=0.2 get F = 1.06

From table with N=3600 get F_s = 0.726

Therefore

$$C = \frac{250 * 1.06}{0.726} \sqrt[4]{\frac{16000}{2280}} = 594\ Ib$$

2-From table use radial load 635,then bearing number is 3204.
3- From table use bearing number 3204 the
Bore=20 mm
Diameter = 47 mm
Width = 14 mm
Balls Dia.= 5/16 in
Balls No. = 8
Radius = 0.04 mm

Exp1:-
Select a ball bearing with the following data:-
R=400 Ib , T = 200 Ib ,N= 1000 rpm , L= 20000 Hr, NDur 600
1-

$$C = \frac{RF}{F_S} \sqrt[4]{\frac{L}{Q}}$$

From table with T/R=200/400=0.5 get F = 1.28
From table with N=1000 get F_s = 1
Therefore

$$C = \frac{400 * 1.28}{1} \sqrt[4]{\frac{20000}{4560}} = 740.94\ Ib$$

2-From table use radial load 1010, then bearing number is 3304.
3- From table use bearing number 3304 the
Bore=20 mm
Diameter = 52 mm
Width = 15 mm
Balls Dia.= 13/32 in
Balls No. = 7
Radius = 0.04 mm

T .-2 Thrust factor F

T/R	F	T/R	F	T/R	F	T/R	F	T/R	F
0.05	1.01	0.30	1.12	0.60	1.37	1.25	2.02	4.00	4.76
0.10	1.02	0.35	1.16	0.70	1.46	1.50	2.27	5.00	5.77
0.15	1.04	0.40	1.20	0.80	1.56	1.75	2.52	7.50	8.27
0.20	1.06	0.45	1.24	0.90	1.67	2.00	2.77	10.00	10.77
0.25	1.09	0.50	1.28	1.00	1.77	3.00	3.77		

جدول رقم (4-3) ثابت الضغط

T: .. 22-1 Speed factors F_S

Rpm	F_s	Rpm	F_s	Rpm	F_s	Rpm	F_s	Rpm	F_s	Rpm	F_s	Rpm	F_s
		270	1.387	825	1.049	1,725	0.8726	3,250	0.7448	5,100	0.6654	8,700	0.5823
10	3.162	280	1.375	850	1.041	1,750	0.8695	3,300	0.7419	5,200	0.6622	8,800	0.5806
15	2.858	290	1.363	875	1.034	1,775	0.8664	3,350	0.7392	5,300	0.6591	8,900	0.5790
20	2.659	300	1.351	900	1.027	1,800	0.8633	3,400	0.7364	5,400	0.6560	9,000	0.5774
25	2.515	310	1.340	925	1.020	1,825	0.8604	3,450	0.7337	5,500	0.6530	9,100	0.5758
30	2.403	320	1.330	950	1.013	1,850	0.8575	3,500	0.7311	5,600	0.6501	9,200	0.5742
35	2.312	330	1.320	975	1.006	1,875	0.8546	3,550	0.7285	5,700	0.6472	9,300	0.5726
40	2.236	340	1.310	1,000	1.000	1,900	0.8518	3,600	0.7260	5,800	0.6444	9,400	0.5711
45	2.171	350	1.300	1,025	0.9938	1,925	0.8490	3,650	0.7235	5,900	0.6416	9,500	0.5696
50	2.115	360	1.291	1,050	0.9878	1,950	0.8462	3,700	0.7210	6,000	0.6389	9,600	0.5681
55	2.065	370	1.282	1,075	0.9821	1,975	0.8436	3,750	0.7186	6,100	0.6363	9,700	0.5666
60	2.021	380	1.274	1,100	0.9765	2,000	0.8409	3,800	0.7162	6,200	0.6337	9,800	0.5652
65	1.981	390	1.265	1,125	0.9710	2,050	0.8357	3,850	0.7139	6,300	0.6312	9,900	0.5637
70	1.944	400	1.257	1,150	0.9657	2,100	0.8307	3,900	0.7116	6,400	0.6287	10,000	0.5624
75	1.911	410	1.250	1,175	0.9605	2,150	0.8258	3,950	0.7093	6,500	0.6263		
80	1.880	420	1.242	1,200	0.9554	2,200	0.8211	4,000	0.7071	6,600	0.6239		
85	1.852	430	1.235	1,225	0.9506	2,250	0.8165	4,050	0.7049	6,700	0.6215		
90	1.826	440	1.228	1,250	0.9457	2,300	0.8120	4,100	0.7027	6,800	0.6193		
95	1.801	450	1.221	1,275	0.9411	2,350	0.8077	4,150	0.7006	6,900	0.6170		
100	1.778	460	1.214	1,300	0.9365	2,400	0.8034	4,200	0.6985	7,000	0.6148		
110	1.736	470	1.208	1,325	0.9321	2,450	0.7993	4,250	0.6965	7,100	0.6126		
120	1.699	480	1.201	1,350	0.9277	2,500	0.7953	4,300	0.6944	7,200	0.6105		
130	1.665	490	1.195	1,375	0.9235	2,550	0.7914	4,350	0.6924	7,300	0.6084		
140	1.635	500	1.189	1,400	0.9193	2,600	0.7875	4,400	0.6905	7,400	0.6063		
150	1.607	525	1.175	1,425	0.9153	2,650	0.7838	4,450	0.6885	7,500	0.6043		
160	1.581	550	1.161	1,450	0.9113	2,700	0.7801	4,500	0.6866	7,600	0.6023		
170	1.557	575	1.149	1,475	0.9074	2,750	0.7765	4,550	0.6847	7,700	0.6003		
180	1.535	600	1.136	1,500	0.9036	2,800	0.7731	4,600	0.6828	7,800	0.5984		
190	1.515	625	1.125	1,525	0.8999	2,850	0.7696	4,650	0.6810	7,900	0.5965		
200	1.495	650	1.114	1,550	0.8962	2,900	0.7663	4,700	0.6792	8,000	0.5946		
210	1.477	675	1.103	1,575	0.8926	2,950	0.7630	4,750	0.6774	8,100	0.5928		
220	1.460	700	1.093	1,600	0.8891	3,000	0.7598	4,800	0.6756	8,200	0.5910		
230	1.444	725	1.084	1,625	0.8857	3,050	0.7567	4,850	0.6738	8,300	0.5892		
240	1.429	750	1.075	1,650	0.8823	3,100	0.7536	4,900	0.6721	8,400	0.5874		
250	1.414	775	1.066	1,675	0.8790	3,150	0.7506	4,950	0.6704	8,500	0.5856		
260	1.400	800	1.057	1,700	0.8758	3,200	0.7477	5,000	0.6687	8,600	0.5840		

جدول رقم (4-4) ثابت السرعة

2

Table 22-1* Rated radial loads for single-row, deep-groove (Type 3000 New Departure) NDur bearings adjusted to speed of the inner race at 1,000 rpm

Basic no.	Radial load	Basic no.	Radial load	Basic no.	Radial load	Basic no.	Radial load
3200†	270	3205	690	3210	1,820	3215	3,180
3300	360	3305	1,110	3310†	3,000	3315	4,810
3201	270	3206	1,020	3211	2,250	3216	3,430
3301	465	3306	1,470	3311	3,390	3316	5,260
3202	290	3207	1,590	3212	2,550	3217	4,190
3302	580	3307	1,820	3312	3,780	3218	4,070
3203	510	3208	1,590	3213	2,990	3219	5,180
3303	710	3308	2,200	3313	4,190	3220	5,710
3204	635	3209	1,710	3214	3,180	3221	5,950
3304	1,070	3309	2,780	3314	4,620	3222	6,510

* Courtesy of New Departure Division of General Motors Corporation, Bristol, Conn.

† Bearing numbers 3200 through 3210 are NDur 300; numbers 3310 through 3222 are NDur 600.

may be selected from Table 22-1:

$$C = \frac{RF}{F_s}\sqrt[4]{\frac{L}{Q}} \tag{22-1}$$

where C = rated radial load adjusted to tabular value (Table 22-1) at 1,000 rpm, lb

R = radial load on bearing, lb

T = thrust load on bearing, lb

F = thrust factor (Table 22-2)

F_s = speed factor for operating speed (Table 22-3)

$(L/Q)^{\frac{1}{4}}$ = life factor; see following paragraph

L = B-10 life of bearing, hr

Q = 2,280 for New Departure NDur 300 bearings

= 4,560 for New Departure NDur 600 bearings

Table 22-2 Thrust factor F

T/R	F	T/R	F	T/R	F	T/R	F	T/R	F
0.05	1.01	0.30	1.12	0.60	1.37	1.25	2.02	4.00	4.76
0.10	1.02	0.35	1.16	0.70	1.46	1.50	2.27	5.00	5.77
0.15	1.04	0.40	1.20	0.80	1.56	1.75	2.52	7.50	8.27
0.20	1.06	0.45	1.24	0.90	1.67	2.00	2.77	10.00	10.77
0.25	1.09	0.50	1.28	1.00	1.77	3.00	3.77		

Table 22-4 Principal dimensions for radial ball bearings

Bearing no. plain	Bore mm	Bore in.	Diameter mm	Diameter in.	Width mm	Width in.	Balls Diam.	Balls No.	Radius r
3200	10	0.3937	30	1.1811	9	0.3543	7/32	7	0.025
3300			35	1.3780	11	0.4331	1/4	7	
3201	12	0.4724	32	1.2598	10	0.3937	0.210	8	0.025
3301			37	1.4507	12	0.4724	9/32	7	0.04
3202	15	0.5906	35	1.3780	11	0.4331	0.210	9	0.025
3302			42	1.6535	13	0.5118	5/16	7	0.04
3203	17	0.6693	40	1.5748	12	0.4724	9/32	8	0.025
3303			47	1.8504	14	0.5512	11/32	7	0.04
3204	20	0.7874	47	1.8504	14	0.5512	5/16	8	0.04
3304			52	2.0472	15	0.5906	13/32	7	
3205	25	0.9843	52	2.0472	15	0.5906	5/16	9	0.04
3305			62	2.4409	17	0.6693	13/32	8	
3206	30	1.1811	62	2.4409	16	0.6299	3/8	9	0.04
3306			72	2.8346	19	0.7480	15/32	8	
3207	35	1.3780	72	2.8346	17	0.6693	7/16	9	0.04
3307			80	3.1496	21	0.8268	17/32	8	0.06
3208	40	1.5748	80	3.1496	18	0.7087	15/32	9	0.04
3308			90	3.5433	23	0.9055	19/32	8	0.06
3209	45	1.7717	85	3.3465	19	0.7480	15/32	10	0.04
3309			100	3.9370	25	0.9843	21/32	8	0.06
3210	50	1.9685	90	3.5433	20	0.7874	15/32	11	0.04
3310			110	4.3307	27	1.0630	23/32	8	0.08
3211	55	2.1654	100	3.9370	21	0.8268	17/32	11	0.06
3311			120	4.7244	29	1.1417	25/32	8	0.08
3212	60	2.3622	110	4.3307	22	0.8661	19/32	10	0.06
3312			130	5.1181	31	1.2205	27/32	8	0.03
3213	65	2.5591	120	4.7244	23	0.9055	21/32	10	0.06
3313			140	5.5118	33	1.2992	29/32	8	0.08
3214	70	2.7559	125	4.9213	24	0.9449	21/32	11	0.06
3314			150	5.9055	35	1.3780	31/32	8	0.08
3215	75	2.9528	130	5.1181	25	0.9843	21/32	11	0.06
3315			160	6.2992	37	1.4567	1	8	0.08
3216	80	3.1496	140	5.5118	26	1.0236	11/16	11	0.08
3316			170	6.6929	39	1.5354	1 1/16	8	
3217	85	3.3465	150	5.9055	28	1.1024	25/32	11	0.08
3317			180	7.0866	41	1.6142	1 1/8	8	0.10
3218	90	3.5433	160	6.2992	30	1.1811	27/32	11	0.08
3318			190	7.4803	43	1.6929	1 3/16	8	0.10

Lewis Equation

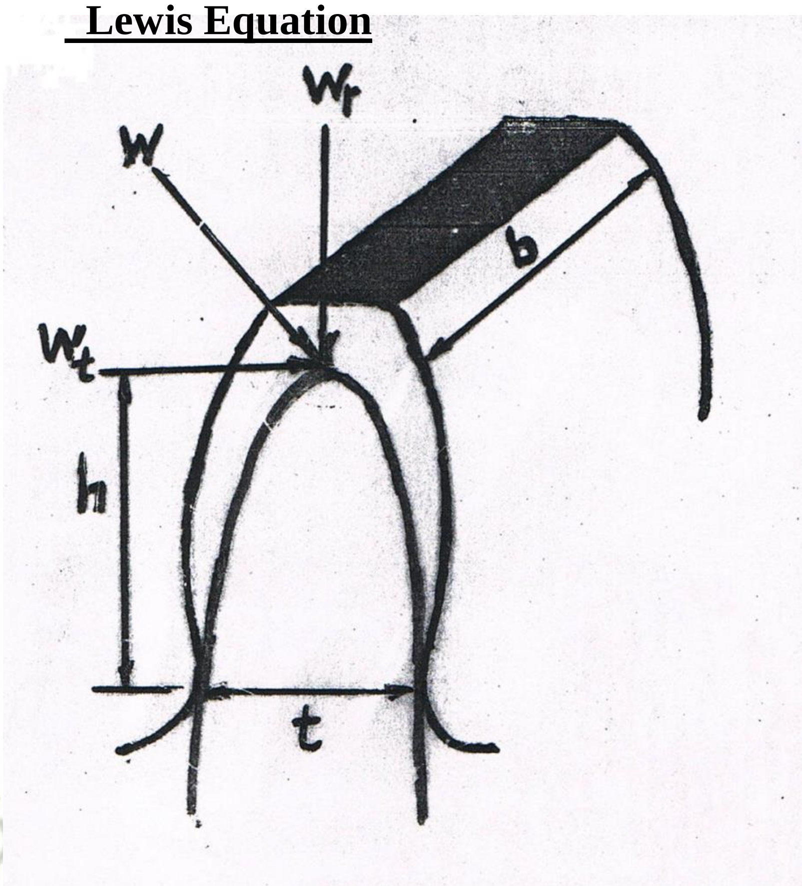
$$\sigma_w = \frac{BM}{z_b}$$

$$z_b = \frac{bt^2}{6}$$

$$BM = W_t h$$

$$\sigma_w = \frac{6W_t\, h}{bt^2}$$

$$W_t = \sigma_w b \ \frac{t^2}{6h}$$



T and h are variable depending upon the size of the tooth (i.e Circular Pitch)
$t = xp_c$
$h = Kp$
Where x and K are constants.

$$W_t = \sigma_w b \ \frac{x^2p^2}{6hp}$$

$$W_t = \sigma_w bp \ \frac{x^2}{6k}, y = \frac{x^2}{6k}$$

$$W_t = \sigma_w b \ \ p_c y$$

$$W_t = \sigma_w b \ \ \pi m y$$

Lewis Equation

$$y \ = 0.124 \ - \frac{0.684}{T} \ for\ 14.5^o$$

$$y \ = 0.154 \ - \frac{0.912}{T} \ for\ 20^o$$

<u>Notes:-</u>

$$\boxed{W_t = \sigma_w b\, \pi\, m\, y}$$

Lewis Equation

$$y = 0.154 - \frac{0.912}{T}$$

$$P=TW$$

$$T = W_t \frac{D}{2}$$

$$W_t = W_t C_s$$, $m=\frac{D}{T}$, $T = \frac{D}{m}$

$$\sigma_w = \sigma_a C_v \quad , C_v=\frac{3}{3+V}$$

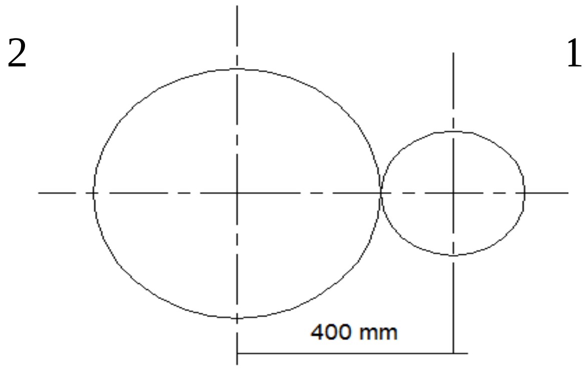
C_v =Velocity Factor (V m/sec)

b=10 m

Exp:- For reciprocating compressor is to be connected to an electric motor with the help of spur gears . The distance between the shafts is to be (400 mm) . The speed of the electric motor is (360 rpm) and the speed of the compressor shaft is desired to be (120 rpm).Using the following data to design the gear :-

1- The torque to be transmitted = 2000 N.m
2- Service Factor = 1.25
3- Allowable Stress for large gear (σ_{ag})=140 N/mm^2
4- Allowable stress for small gear (σ_{ap})= 170 N/mm^2

Gear (compressor)
N_2=120 rpm
σ_{ag}=140 N/mm^2



pinion (motor)
N_1=360rpm
σ_{ap}=170N/mm^2

$d_1 + d_2 = 800$ --------------------(1)

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

$$3 = \frac{d_2}{d_1}$$

d_1=200

d_2=600

$$W_t = \sigma_w \, b \, \pi \, m \, y$$

Since ($\sigma_{ag} < \sigma_{ap}$) ……..therefore the design should be based upon the gear.

$$\sigma_w = \sigma_a C_v$$

$$V = \omega \frac{d_2}{2} = \frac{120 \times 2\pi}{60} \times 300 \times \frac{1}{1000}$$

V=3.768 m/sec

$C_v = \frac{3}{3+V} = \frac{3}{3+3.768} = 0.443$

$$\sigma_w = 140 \times 0.443 = 62.02 \; N/mm^2$$

$$W_t = \frac{T}{d_2/_2} = \frac{2000 \times 1.25 \times 1000}{600/_2} = 8333.3 \; N$$

b=(9.5-12.5)m

let b= 10 m

$$y \;\; = 0.154 \;\; - \frac{0.912}{T} \qquad \text{m}= \frac{\mathbf{D}}{\mathbf{T}} \;\; , \;\; T = \frac{D}{m}$$

$$y \;\; = 0.154 \;\; - \frac{0.912m}{d_2}$$

$$y \;\; = 0.154 \;\; - \frac{0.912m}{600}$$

$$W_t = \sigma_w b \, \pi \, m \, y$$

8333.3=62.02$\times$ $10m \times \pi m(0.154 \;\; - \frac{0.912\text{m}}{600})$

By trial and error we get:-

m$\cong$ $5 \; mm$

b=10m=50mm

m= $\frac{\mathbf{D}}{\mathbf{T}}$, $T_1 = \frac{200}{5} = 40$, $T_2 = \frac{600}{5} = 120$

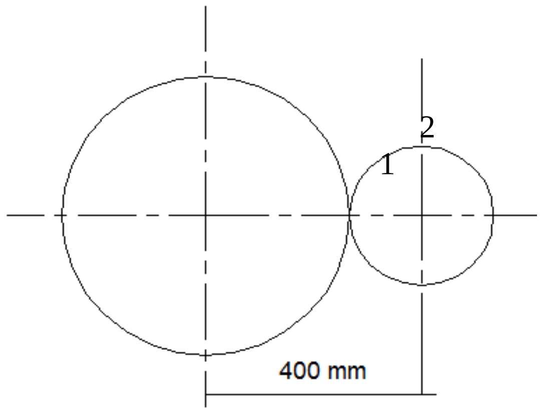
Exp2:- A bronze spur pinion (S_a=83Mn/m^2)rotating at 600 rpm drives a cast steel spur gear (S_a=103 Mn/m^2) at a transmission ratio of 4 to 1 .

The pinion has 16 standard 20^o full depth involute teeth of 8 module. The face width of both gears is 90 mm . How much power can be transmitted from the standpoint of strength?

Machine Design Schaum's Series P.229

$\frac{N_1}{N_2} = \frac{4}{1}$

P?



Solution :- S_{ag}=103,S_{ap}=83,N_1=600,T_1=16,$\Phi=20^o$,m=8,b=90 .
Since $S_{ap} < S_{ag}$ therefore the the design should be based upon the pinion.

$$W_t = \sigma_w \, b \, \pi \, m \, y$$

$$\sigma_w = \sigma_a C_v$$

$$m = {d_1}/{T_1}$$

$$d_1 = m \times T_1$$

$$= 8 \times 167$$

$$= 128 \, mm$$

$$V = \omega_1 r_1$$

$$= 600 \times \frac{2\pi}{60} \times \frac{128}{2} \times \frac{1}{1000}$$

$$= 4.02 m/sec$$

$$C_v = \frac{3}{3+V} = \frac{3}{3+4.028} = 0.427$$

$$\sigma_w = 83 \times 0.427 = 35.47 \; N/mm^2$$

$$y = 0.154 - \frac{0.912}{T}$$

$$= 0.154 - \frac{0.912}{16} = 0.097$$

$$W_t = 35.47 \times 90 \times \pi \times 8 \times 0.097$$

$$= 778204 \text{ N}$$

$$P = W_t \times V$$

$$= 7782.4 \times 4.02$$

$$= 31285.2 \text{ Watt}$$

$$= 31.28 \text{ kW}$$

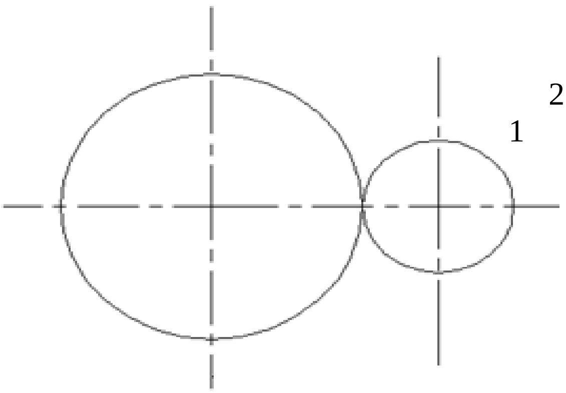
Exp3:- A spur pinion of cast steel (σ_a=140 MN/m^2) is to drive a spur gear of cast iron (σ_a= 55 MN/m^2) . The transmission ratio is to be $2\frac{1}{3}$ *to* 1 .The diameter of the pinion is to be 105 mm and 20kW will be transmitted at 900rpm of the pinion. The teeth are to be 20^o full depth involute from. Design for the greatest number of teeth. Determine the necessary module and face width of gear for strength only.

Machine Design

Schaum's Series

P.229

Solution :- σ_{ag}=55,σ_{ap}=140,d_1=105,N1=900,P=20kW,Φ=20^o.



Since $\sigma_{ag} < \sigma_{ap}$ therefore the the design should be based upon the gear.

$$W_t = \sigma_w \, b \, \pi \, m \, y$$

$$\frac{N_1}{N_2} = \frac{2.333}{1}$$

$$\frac{900}{N_2} = \frac{2.333}{1}$$

N_2=385.769

P=Tω

20 000 = T$\times$ $385.769 \times \frac{2\pi}{60}$

T=495.078 N.m

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

$$\frac{2.333}{1} = \frac{d_2}{105}$$

$$d_2 = 245\ mm$$

$$T = W_t \times \frac{D_2}{2}$$

$$495.078 = W_t \times \frac{245}{2} \times \frac{1}{1000}$$

$$W_t = 4042.03$$

$$\sigma_w = \sigma_{ag} C_v$$

$$V = \omega_2 r_2$$

$$= 385.769 \times \frac{2\pi}{60} \times \frac{245}{2} \times \frac{1}{1000}$$

$$= 4.947\ m/sec$$

C_v= $\frac{3}{3+V}$ =$\frac{3}{3+4.947}$ $= 0.3775$

$$\sigma_w = 55 \times 0.3775 = 20.76\ N/mm^2$$

$$y \quad = 0.154 \quad - \frac{0.912}{T_1}$$

$$m = {d_2}/{T_2}$$

$$y \quad = 0.154 \quad - \frac{0.912m}{245}$$

4042.03=20.76$\times$ $10m \times \pi m(0.154 \quad - \frac{0.912\text{m}}{245})$

By trial and error get

m$\cong$ 7

$m=\frac{d_1}{T_1}=\frac{d_2}{T_2}$

$T_1=\frac{105}{7}=15$

$T_2=\frac{245}{7}=33$

b=10m

= 10×7

=70 mm

<u>Type of gear train :-</u>

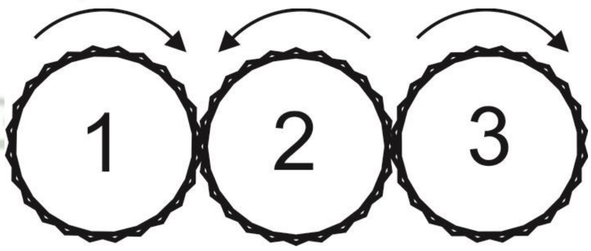
1. Simple gear train. البسيط

$$\frac{N_1}{N_2}=\frac{T_2}{T_1}\text{..........}(1)$$

$$\frac{N_2}{N_3}=\frac{T_2}{T_3}\text{..........}(2)$$

$$\frac{N_1}{\cancel{N_2}}\times\frac{\cancel{N_2}}{N_3}=\frac{\cancel{T_2}}{T_1}\times\frac{T_3}{\cancel{T_2}}$$

$$\frac{N_1}{N_3}=\frac{T_3}{T_1}\text{..........}(3)$$



2. Compound gear train. المركب

$$\frac{N_1}{N_2} = \frac{T_2}{T_1} \text{..........} (1)$$

$$\frac{N_3}{N_4} = \frac{T_4}{T_3} \text{..........} (2)$$

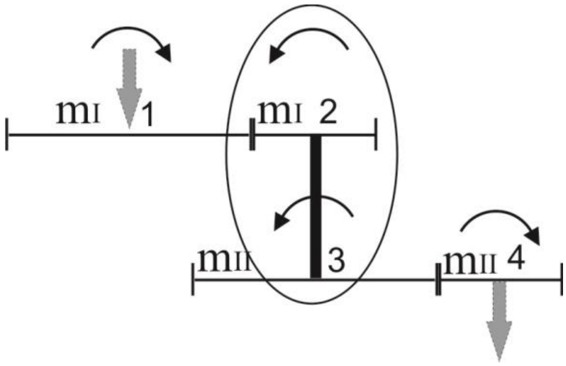
$$N_2 = N_3$$

$$\frac{N_1}{\cancel{N_2}} \times \frac{\cancel{N_3}}{N_4} = \frac{T_2}{T_1} \times \frac{T_4}{T_3}$$

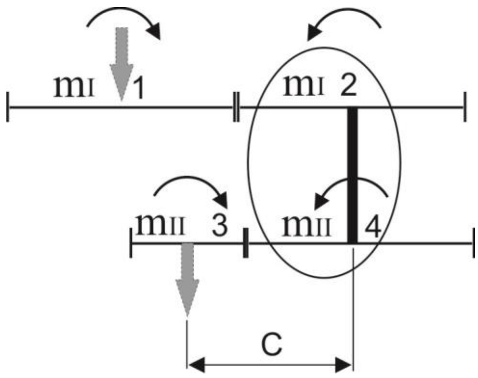
$$\frac{N_1}{N_4} = \frac{T_2}{T_1} \times \frac{T_4}{T_3}$$

$$\frac{N_4}{N_1} = \frac{T_1}{T_2} \times \frac{T_3}{T_4}$$

$$N_4 = N_1 \times \frac{T_1}{T_2} \times \frac{T_3}{T_4} \text{..........} (3)$$



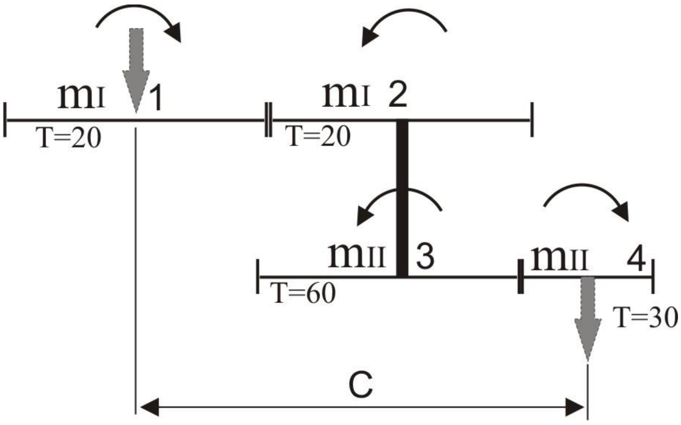
3. Reverted gear train المعكوس



4. Epicycle train. شمسي وقمري

Ex.1 :-

T = 20
N = 1000
m = 6
fine = C



1.

$$\frac{N_1}{N_4} = \frac{T_2 \times T_4}{T_1 \times T_3}$$

$$\frac{1000}{N_4} = \frac{\overset{1}{\cancel{20}} \times \overset{1}{\cancel{30}}}{\underset{1}{\cancel{20}} \times \underset{2}{\cancel{60}}}$$

$$N_4 = 2000\, r.p.m$$

2.

a same direction

3.

$$C = r_1 + r_2 + r_3 + r_4$$

$$D_1 = m \times T_1 \Rightarrow 6 \times 20 = 120mm$$

$$D_2 = m \times T_2 \Rightarrow 6 \times 20 = 120mm$$

$$D_3 = m \times T_3 \Rightarrow 6 \times 60 = 360mm$$

$$D_4 = m \times T_4 \Rightarrow 6 \times 30 = 180mm$$

$$C = r_1 + r_2 + r_3 + r_4$$

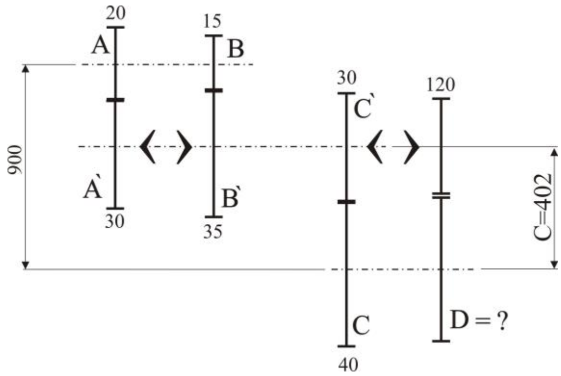
$$C = 60 + 60 + 180 + 90 = 390mm$$

__Ex.2 :-__

Design Gear

N = 1000

m = 6



$$m = \frac{D}{T}$$

$$m = \frac{D_D}{T_D}$$

$$m = \frac{D_{D'}}{T_{D'}}$$

$$C = r_{D'} + r_D$$

$$D = m \times r_{D'}$$

$$D = 6 \times 40 = 240$$

$$402 = 120 + r_D$$

$$402 - 120 = r_D$$

$$r_D = 282$$

$$D_D = r_D \times 2$$

$$D_D = 564$$

$$T = \frac{D}{m}$$

$$T_D = \frac{564}{6} = 94$$

$$m = \frac{D_D}{T_D}$$

$$T_D = \frac{D_D}{m}$$

$$94 = \frac{D_D}{6}$$

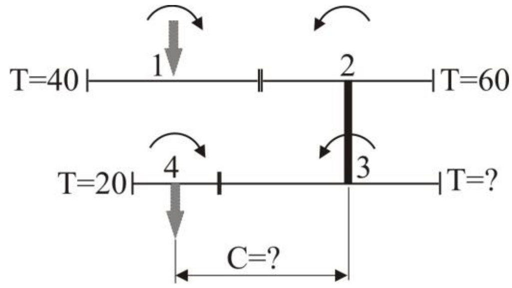
$$D_D = 564$$

<u>Ex.3 :-</u>

m = 5

N = 1000

find T_3 , C



1.

$$T_1 + T_2 = T_3 + T_4$$

$$40 + 60 = T_3 + 20$$

$$T_3 = 100 - 20 = 80$$

$$\frac{N_1}{N_4} = \frac{T_2 \times T_4}{T_1 \times T_3}$$

$$\frac{1000}{N_4} = \frac{\overset{3}{\cancel{60}} \times \overset{1}{\cancel{20}}}{\underset{2}{\cancel{40}} \times \underset{4}{\cancel{80}}} = \frac{3}{8}$$

$$N_4 = 2666.6\, r.p.m$$

2. *a same direction*

3.

$$C = r_1 + r_2$$

$$D_r = m \times T_1$$

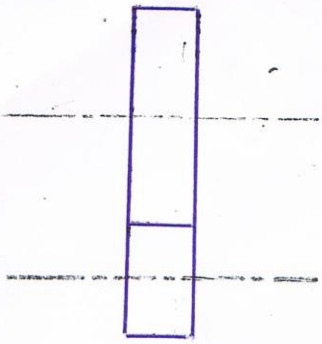
$$= 5 \times 40 = 200mm$$

$$D_2 = m \times T_2$$

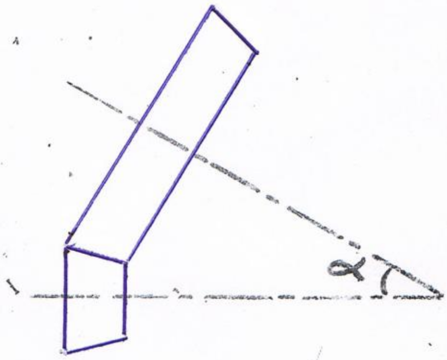
$$= 5 \times 60 = 300mm$$

$$C = 100 + 150 = 250mm$$

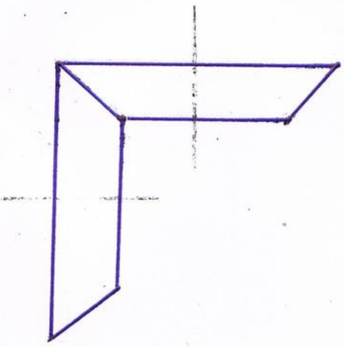
Worm gearing



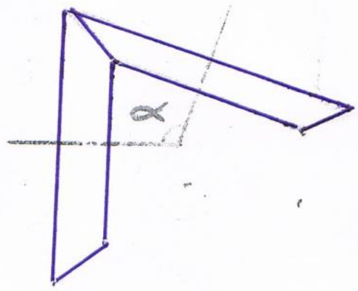
$\alpha = 0$



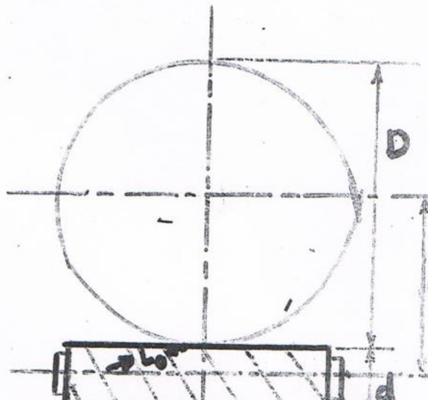
$\alpha < 90$



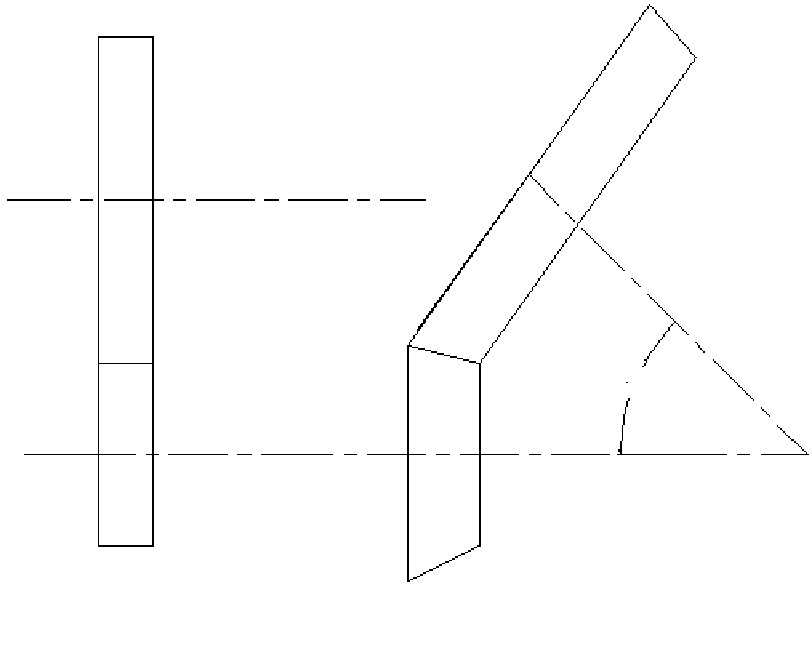
$\alpha = 90$

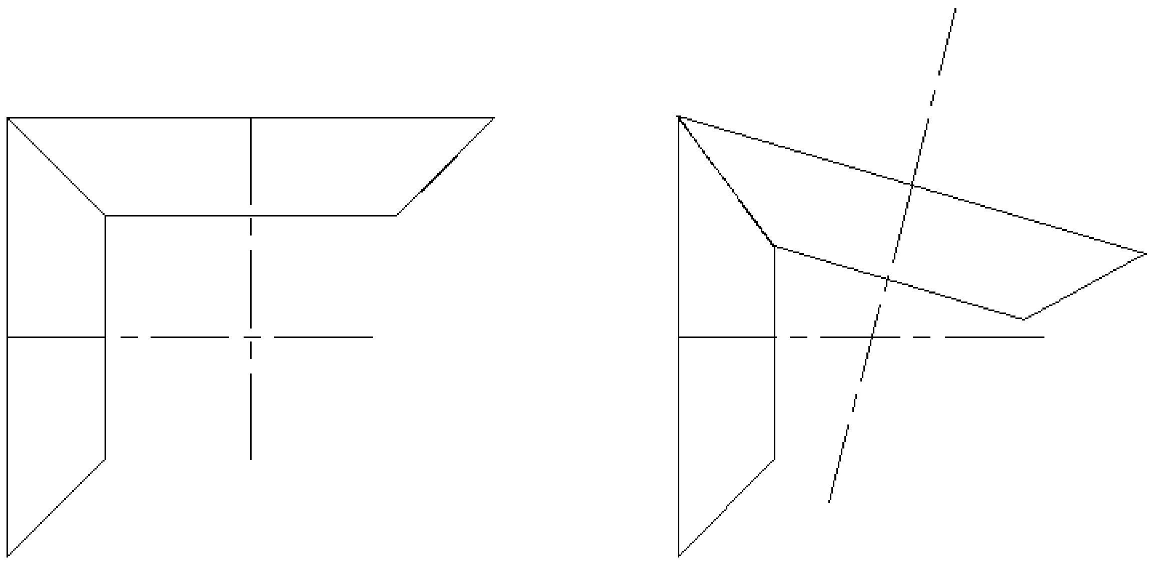


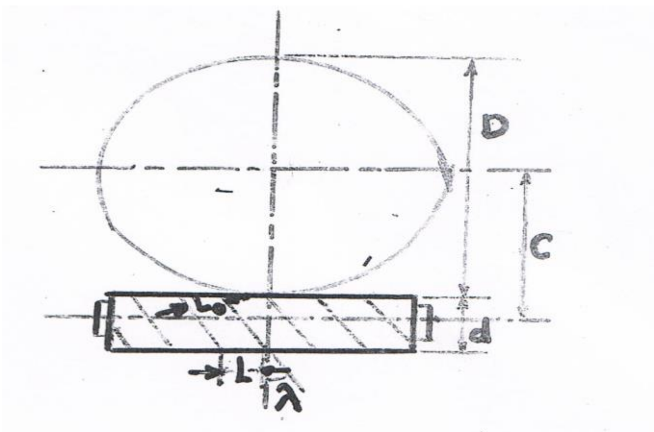
$\alpha > 90$



λ







Exp:-

Design a worm gear from the following data:-

1- Velocity ratio (r) = 14
2- Approximately center distance = 6 in

Solution :-
1- From worm-gear design curves with r= 14 get :-
$\lambda=22^{\circ}$, C/L= 2.83
2- Therefore

$\frac{C}{L_o}=2.83$

$\frac{6}{L_o}=2.83$

L_o= 2.12 in

3- L $=\frac{L_o}{Cos\lambda}=\frac{2.12}{0.923}=2.29\ in$

4- p=L/n
Where n= No. of threads
Let n=3

$p=\frac{2.29}{3}=0.763\ in$

The nearest standard pitch is 0.75 in , then
5-
p= L/n
L= 3 *0.75= 2.25

$L=\frac{L_o}{cos\lambda}$

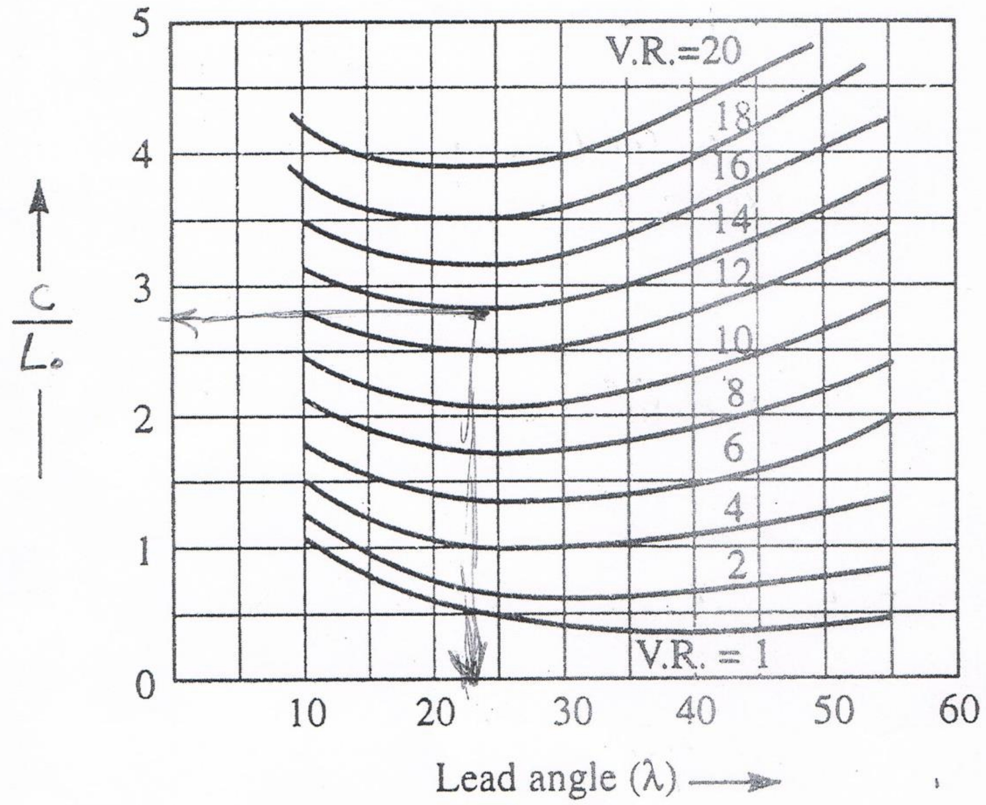
$L_o=2.25*0.927=2.09$ in

$C/L_o=2.83$
C=2.83* 2.09 = 5.91 in

$d=\frac{L}{\pi\ tan\lambda}=\frac{2.25}{\pi*0.404}=1.774\ in$

$C=\frac{D+d}{2}$,

$D=2C-d=2*5.91-1.774=10.05\ in$

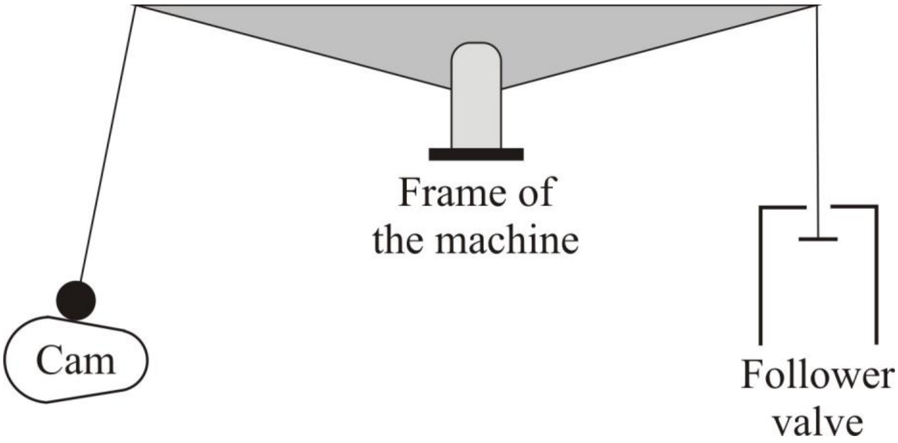


$\lambda = 22$

$\frac{C}{L_o} = 2.89$

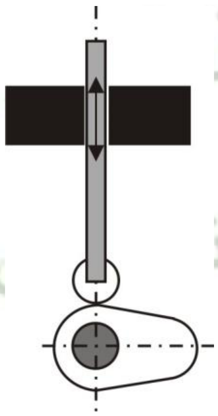
<u>Cams</u>

A cam is a rotating machine element which gives reciprocating or oscillating motion to another element known as follower.



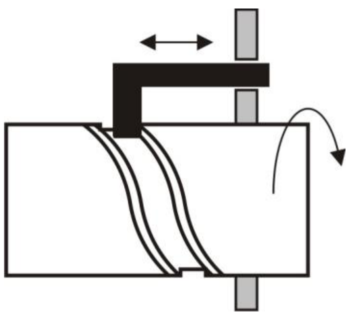
<u>Type of cams:-</u>

1. Disc cams :
   The follower moves in the direction perpendicular to the cam axis.



Cylindrical cams :

The follower moves in the direction parallel to the cam axis.



__Ex.1:-__

For S = 3mm. $\theta = 150°$ speed = 100 rad/sec. Find d , v and f $\beta = 50°$ and $90°$.

Solution:-

$$For\ \beta = 50^{\circ} \Rightarrow 0 \le \beta \le \frac{\theta}{2}$$

$$d = \frac{2S\,\beta^2}{\theta^2} = \frac{2\times 30\times(50)^2}{(150)^2} = 6.66mm.$$

$$v = \frac{4S\,\omega\beta}{\theta^2} = \frac{4\times 30\times 100\times\frac{180}{\pi}\times 50}{(150)^2} = 1527.8mm.$$

$$f = \frac{4S\,\omega^2}{\theta^2} = \frac{4\times 30\times(100\times\frac{180}{\pi})^2}{(150)^2} 175260.6mm.$$

$$For\ \beta = 90^{\circ} \Rightarrow \frac{\theta}{2} \le \beta \le \theta$$

$$d = S\left[\frac{4\beta}{\theta} - 1 - \frac{2\beta^2}{\theta^2}\right]$$

$$d = 30\left[\frac{4\times 90}{150} - 1 - \frac{2\times(90)^2}{(150)^2}\right] = 0.72$$

$$v = \frac{4S\,\omega}{\theta}\left[1 - \frac{\beta}{\theta}\right]$$

$$v = \frac{4\times 30\times 100\frac{180}{\pi}}{150}\left[1 - \frac{90}{150}\right] = 183mm.$$

$$t = -175260.6mm./\sec^2$$

__Ex.2 :-__

The exhaust valve of a four stroke engine is operated by a cam designed to give a uniform acceleration and retardation motion (UARM). If the valve stroke 22mm. during 54° of cam rotation.

Find distance, velocity and acceleration of the valve for the angle of cam (β) of 0° , 20° and 54°. Using speed of cam s