Al-Dour Technical Institute

Mechanical Department

2nd Stage

Training Package

In

Types of Belts , Design of Belts

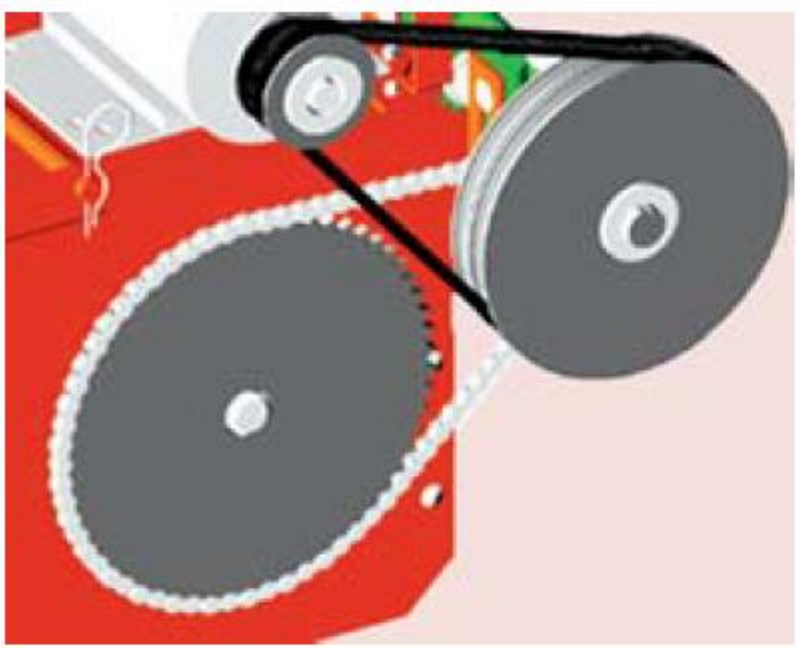
For

Students of second class

Mechanical Department/ Production

By

Nadum I. Naser



Overview

The belts or ropes are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or at different speeds.

Objectives :-

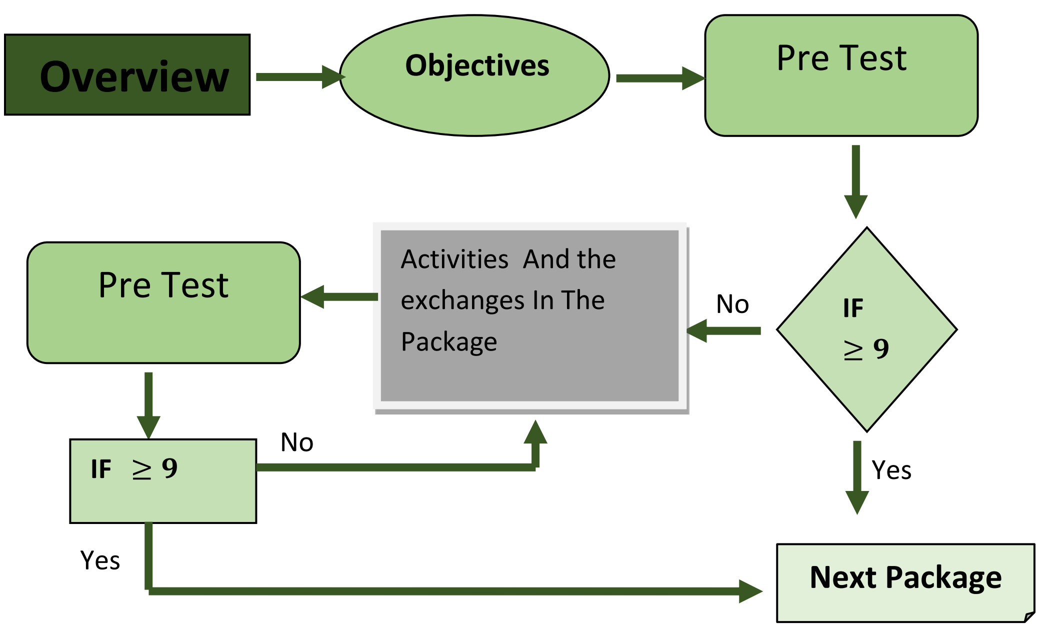
After studying the first modular unit , the student will be able to:-

1-Define Types of belt drives.

2-calculate Power transmitted by a belt .

3-Calculate the length of belt.

Flow Chart:-



Pre Test

Find the length of belt which is necessary to drive a pulley of 80 cm dia. is running parallel at a distance of 12 meters from the driving pulley of 480 cm dia.

Notes

-10 degrees for the above question.

- Check your answers in key answer.

The Text

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

1.the velocity of the belt

2.the tension under which the belt is placed on the pulleys.

3.the arc of contact between the belt and the smaller pulley.

4.the conditions under which the belt is used.

Types of belt drives

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

1.light drives: these are used to transmit small powers at belt speeds up to 10 m / s as in agricultural machines.

2.medium drives: these are used to transmit medium powers at belt speeds over 10 m / s to 22 m/ s as in machine tools.

3. heavy drives: these are used to transmit large powers at belt speeds over 22 m / s as in compressors and generators.

Types of belts

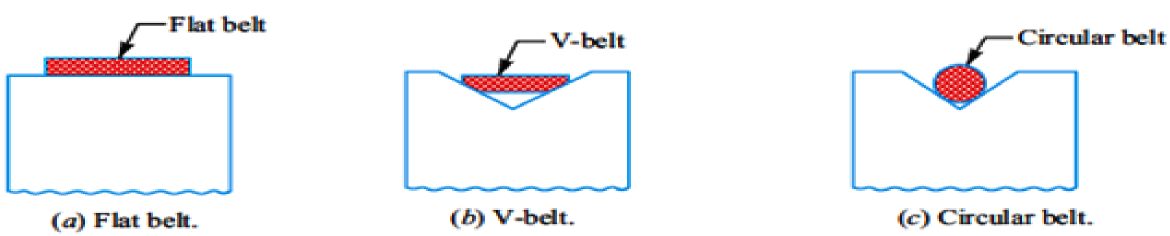
1.flat belt 2.V-belt 3.circular belt

Material used for belts

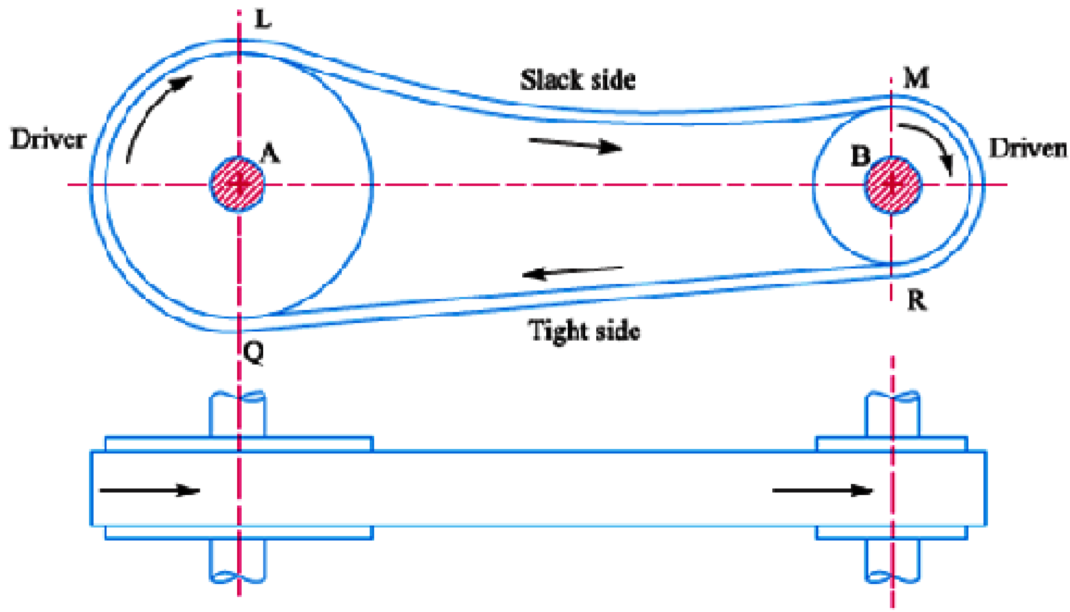
1.lather belts 2. Cotton or fabric belts 3.rubber belts 4.balata belts

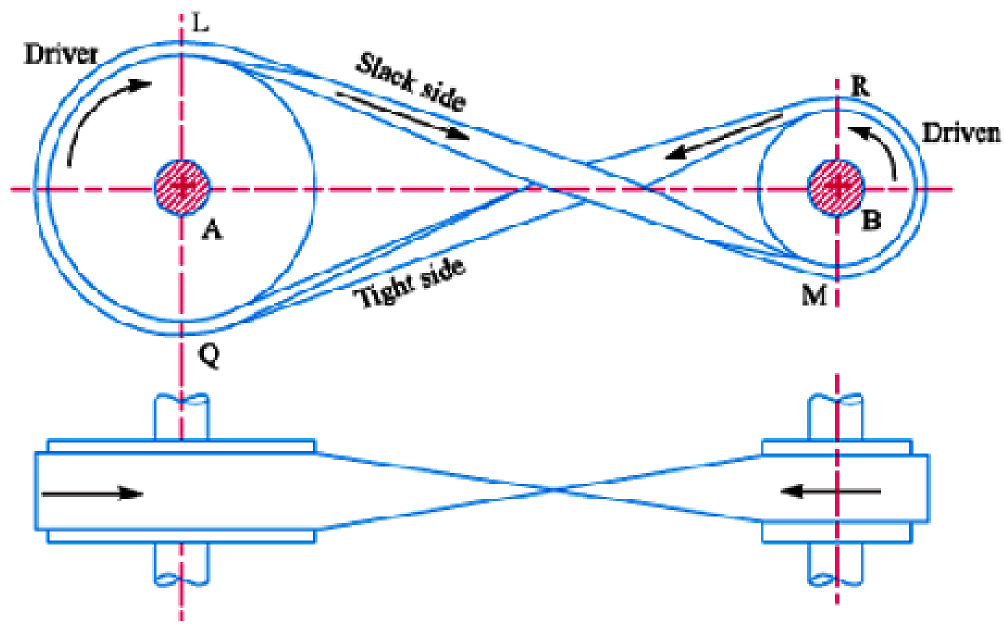
Types of flat belt drives

1.open belt drive 2.crossed or twist belt drive



Types of belts





Velocity ratio of belt drive

$$v_1 = \frac{\pi d_1 N_1}{60} \quad m/s \qquad v_2 = \frac{\pi d_2 N_2}{60} \quad m/s$$

$$v_1 = v_2$$

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

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

$d_1 - diameter of\ the\ driver$

$d_2 - diameter of\ the\ follower$

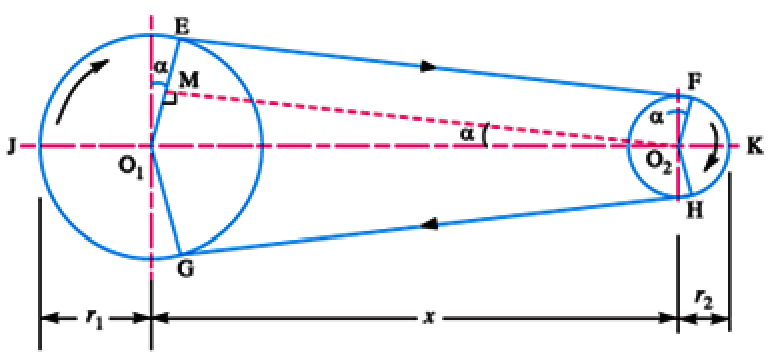
$N_1 - speed of\ the driver in\ R.P.M$

$N_2 - speed of\ the\ follower in\ R.P.M$

$v_1 - velocity of\ the belt on the drive\ pulley$

$v_2 - velocity of\ the belt on the driven pulley$

Length of an open belt drive



open belt drive

Let the belt leaves the larger pulley at E and G and the smaller pulley at F and H
Through O_2 draw O_2M parallel to FE.
From the geometry of the figure, we find that O_2M will be perpendicular to O_1E.
Let the angle $MO_2O_1 = \alpha$ radians.
We know that the length of the belt,

$$L = \text{Arc } GJE + EF + \text{Arc } FKH + HG$$

$$= 2\,(\text{Arc } JE + EF + \text{Arc } FK) \qquad \ldots(i)$$

From the geometry of the figure, we also find that

$$\sin\alpha = \frac{O_1M}{O_1O_2} = \frac{O_1E - EM}{O_1O_2} = \frac{r_1 - r_2}{x}$$

Since the angle α is very small, therefore putting

$$\sin\alpha = \alpha \text{ (in radians)} = \frac{r_1 - r_2}{x} \qquad \ldots(ii)$$

$$\therefore \qquad \text{Arc } JE = r_1\left(\frac{\pi}{2} + \alpha\right) \qquad \ldots(iii)$$

Similarly,

$$\text{arc } FK = r_2\left(\frac{\pi}{2} - \alpha\right) \qquad \ldots(iv)$$

$$EF = MO_2 = \sqrt{(O_1O_2)^2 - (O_1M)^2} = \sqrt{x^2 - (r_1 - r_2)^2}$$

$$= x\sqrt{1 - \left(\frac{r_1 - r_2}{x}\right)^2}$$

Expanding this equation by binomial theorem, we have

$$EF = x\left[1 - \frac{1}{2}\left(\frac{r_1 - r_2}{x}\right)^2 + \ldots\right] = x - \frac{(r_1 - r_2)^2}{2x} \qquad \ldots(v)$$

Substituting the values of arc JE from equation *(iii)*, arc FK from equation *(iv)* and EF from ation *(v)* in equation *(i)*, we get

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

$$= 2\left[r_1 \times \frac{\pi}{2} + r_1\alpha + x - \frac{(r_1 - r_2)^2}{2x} + r_2 \times \frac{\pi}{2} - r_2\alpha\right]$$

$$= 2\left[\frac{\pi}{2}(r_1 + r_2) + \alpha\,(r_1 - r_2) + x - \frac{(r_1 - r_2)^2}{2x}\right]$$

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

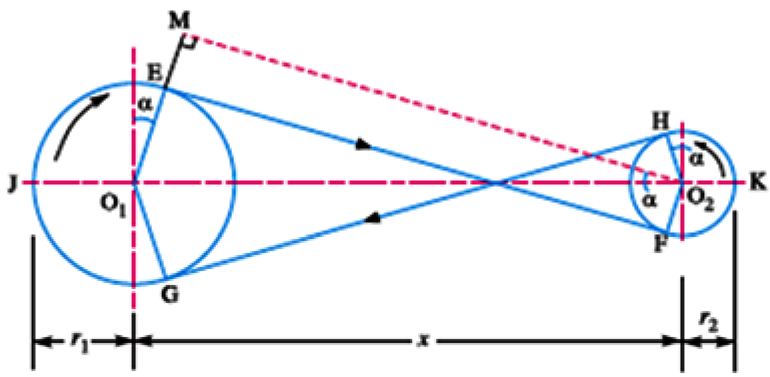
Substituting the value of $\alpha = \dfrac{(r_1 - r_2)}{x}$ from equation *(ii)*, we get

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

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

$$= \pi\,(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x} \qquad \ldots \text{(in terms of pulley radii)}$$

Length of a cross belt drive



crossed belt drive

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

x = Distance between the centres of two pulleys (*i.e.* O_1O_2), and

L = Total length of the belt.

Let the belt leaves the larger pulley at E and G and the smaller pulley at F and H

Through O_2 draw O_2M parallel to FE.

From the geometry of the figure, we find that O_2M will be perpendicular to O_1E.

Let the angle $MO_2O_1 = \alpha$ radians.

We know that the length of the belt,

$$L = \text{Arc } GJE + EF + \text{Arc } FKH + HG$$

$$= 2\,(\text{Arc } JE + FE + \text{Arc } FK) \qquad ...(i)$$

From the geometry of the figure, we find that

$$\sin\alpha = \frac{O_1M}{O_1O_2} = \frac{O_1E + EM}{O_1O_2} = \frac{r_1 + r_2}{x}$$

Since the angle α is very small, therefore putting

$$\sin\alpha = \alpha \text{ (in radians)} = \frac{r_1 + r_2}{x} \qquad ...(ii)$$

$$\therefore \qquad \text{Arc } JE = r_1\left(\frac{\pi}{2} + \alpha\right) \qquad ...(iii)$$

Similarly,

$$\text{arc } FK = r_2\left(\frac{\pi}{2} + \alpha\right) \qquad ...(iv)$$

and

$$EF = MO_2 = \sqrt{(O_1O_2)^2 - (O_1M)^2} = \sqrt{x^2 - (r_1 + r_2)^2}$$

$$= x\sqrt{1 - \left(\frac{r_1 + r_2}{x}\right)^2}$$

Expanding this equation by binomial theorem, we have

$$EF = x\left[1 - \frac{1}{2}\left(\frac{r_1 + r_2}{x}\right)^2 + ...\right] = x - \frac{(r_1 + r_2)^2}{2x} \qquad ...(v)$$

Substituting the values of arc JE from equation (*iii*), arc FK from equation (*iv*) and EF from equation (*v*) in equation (*i*), we get,

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

$$= 2\left[r_1\times\frac{\pi}{2}+r_1.\alpha+x-\frac{(r_1+r_2)^2}{2x}+r_2\times\frac{\pi}{2}+r_2.\alpha\right]$$

$$= 2\left[\frac{\pi}{2}(r_1+r_2)+\alpha\,(r_1+r_2)+x-\frac{(r_1+r_2)^2}{2x}\right]$$

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

Substituting the value of $\alpha = \frac{(r_1+r_2)}{x}$ from equation (*ii*), we get

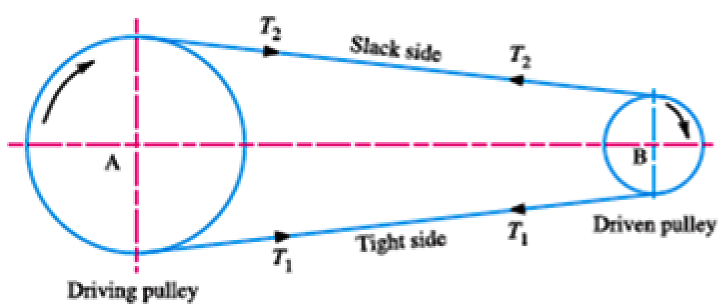
$$L = \pi\,(r_1+r_2)+2\times\frac{(r_1+r_2)}{x}(r_1+r_2)+2x-\frac{(r_1+r_2)^2}{x}$$

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

$$= \pi\,(r_1+r_2)+2x+\frac{(r_1+r_2)^2}{x} \qquad \text{... (in terms of pulley radii)}$$

$$= \frac{\pi}{2}(d_1+d_2)+2x+\frac{(d_1+d_2)^2}{4x} \qquad \text{... (in terms of pulley diameters)}$$

Power transmitted by a belt



power transmitted by a belt

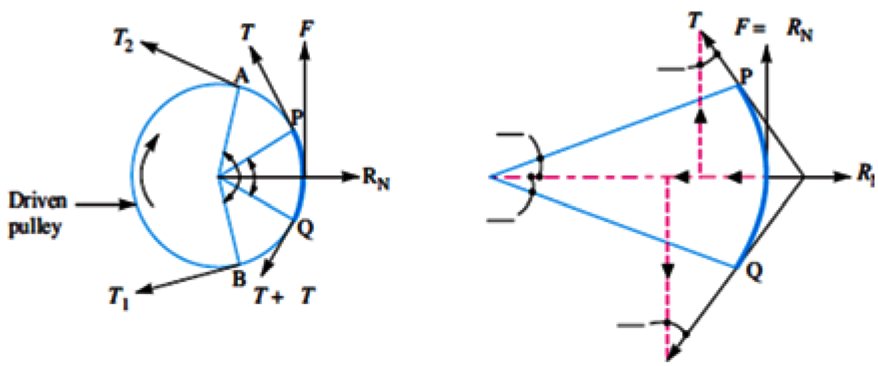
$P = (T_1 - T_2) \times v$

$v - velocity\,of\,the\,belt\ \ m/s$

$T_1 - tension\,in\,the\,tight\,side$

$T_2 - tension\,in\,the\,slack\,side$

Ratio of driving tensions for flat belt drive



Ratio of driving tensions for flat belt

$F = \mu \times R_N$

$$F - frictional\ force$$

$$\mu - coeficient\ of\ friction$$

$$R_N - normal\ reaction\ force$$

$$\log_e\left(\frac{T_1}{T_2}\right) = \mu \times \theta$$

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

$$2.3\log\left(\frac{T_1}{T_2}\right) = \mu \times \theta$$

$$\theta - angle\ of\ contact$$

$$\theta = \left(180^0 - 2\alpha\right)\frac{\pi}{180}\ rad \qquad (for\ open\ belt\ drive)$$

$$\theta = \left(180^0 + 2\alpha\right)\frac{\pi}{180}\ rad \qquad (for\ cross-belt\ drive)$$

Post Test

Find the length of the flat belt under the following conditions; diameter of the small pulley 630 mm, rotate at 740 R.P.M , and diameter of large pulley 2000 mm, center distance is 6 meters. Also find the power , if the friction factor is 0.3 and the tension in the slack side is 300 N.

Key Answer

Pre Test

1.length of open belt drive

$$L = \frac{\pi}{2}(d_1 + d_2) + 2X + \frac{(d_1 - d_2)^2}{4X}$$

$$L = \frac{\pi}{2}(480 + 80) + 2 \times 1200 + \frac{(480 - 80)^2}{4 \times 1200}$$

$$L = 33.13\ m$$

2.length of cross-belt drive

$$L = \frac{\pi}{2}(d_1 + d_2) + 2X + \frac{(d_1 + d_2)^2}{4X}$$

$$L = \frac{\pi}{2}(480 + 80) + 2 \times 1200 + \frac{(480 + 80)^2}{4 \times 1200}$$

$$L = 33.45\ m$$

Post Test

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi \times 0.63 \times 740}{60} = 24.5\ m/s$$

$$L = \frac{\pi}{2}(d_1 + d_2) + 2X + \frac{(d_1 - d_2)^2}{4X}$$

$$L = \frac{\pi}{2}(2000 + 630) + 2 \times 6000 + \frac{(2000 - 630)^2}{4 \times 6000}$$

$$L = 16209\, mm\ = 16.21 m$$

$$\sin\alpha = \frac{r_1 - r_2}{X} = \frac{1000 - 315}{6000}$$

$$\alpha = 6.5^0$$

$$\theta = 180 - 2\alpha = 180 - 2 \times 6.5 = 166.8 \times \frac{\pi}{180} = 2.9\ rad$$

$$2.3\log\left(\frac{T_1}{T_2}\right) = \mu \times \theta = 0.3 \times 2.9 = 0.87$$

$$\log\left(\frac{T_1}{T_2}\right) = \frac{0.87}{2.3} = 0.378$$

$$\frac{T_1}{T_2} = 2.387$$

$$T_1 = 2.387 \times T_2 = 2.387 \times 300 = 716\ N$$

$$power\ P = (T_1 - T_2) \times v = (716 - 300) \times 24.5 = 10192\, W$$

Reference

R. S. Khurmi, J. K. Gupta, "Theory of machine"

Foundation of Technical Education

Al-Dour Technical Institute

Mechanical Department

2nd Stage

Training Package

In

Design of Shafts

For

Students of second class

Mechanical Department/ Production

By

Nadum I. Naser

Overview

A shaft is a rotating machine element which is used to transmit power from one place to another. in order to transfer the power from one shaft to another , the various members such as pulleys , gears, etc.. are mounted on it by means of keys or splines.

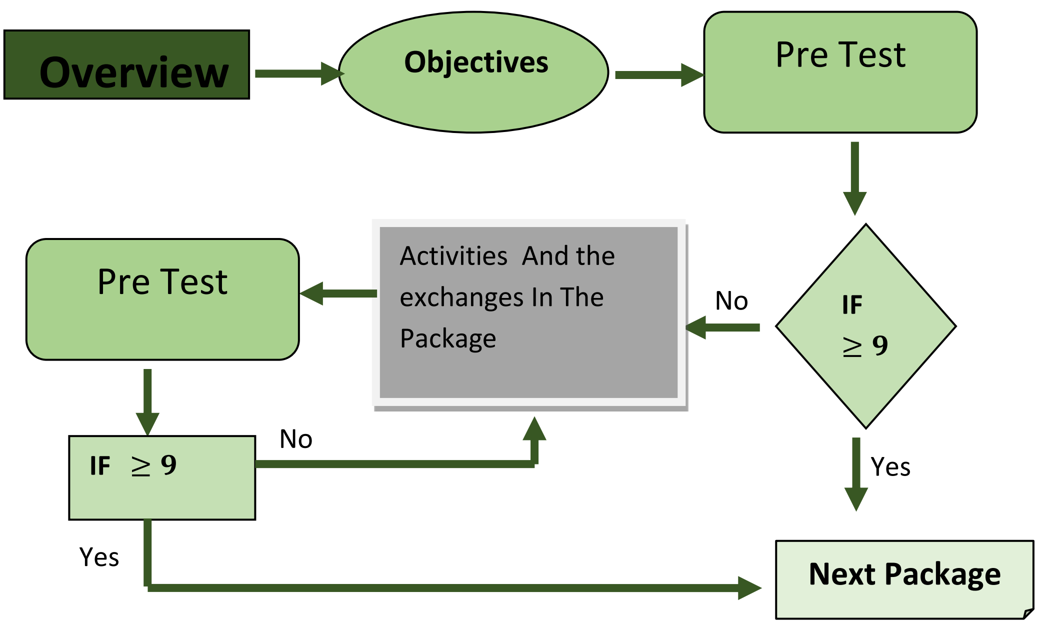
Objectives :-

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

1-Calculate the stresses on shafts.

2-Design of shaft.

Flow Chart:-



Pre Test

Ex 1: A line shaft rotating at 200 R.P.M. is to transmit 20 KW. The shaft may be assumed to be made of mild steel with an allowable shear stress of 42 MPs. determine the diameter of the shaft, neglecting the bending moment?

Notes

-10 degrees for the above question.

- Check your answers in key answer.

The Text

A shaft is a rotating machine element which is used to transmit power from one place to another. In order to transfer the power from one shaft to another, the various members such as pulleys, gears, etc...are mounted on it by means of keys or splines.

An axle is a similar in shape to the shaft, it is a stationary machine element and is used for transmission of bending moment only, and it acts as a support for some rotating body such as car wheel.

A spindle is a short shaft that imparts motion either to a cutting tool (drill press spindles) or to a work piece (lathe spindles)

Standard sizes of transmission shafts

25 mm to 60 mm with 5 mm steps

60 mm to 110 mm with 10 mm steps

110 mm to 140 mm with 15 mm steps

140 mm to 500 mm with 20 mm steps

The standard lengths of shafts are 5 m, 6 m and 7 m.

Stresses in shafts

1. Shear stresses due to the transmission of torque (torsional load)

2. Bending stresses (tensile or compressive) due to the forces acting upon machine elements like gears, pulleys as well as due to the weight of the shaft itself.

3. Stresses due to combined torsional and bending loads.

Design of shafts:

(a) Shafts subjected to twisting moment only

(b) Shafts subjected to bending moment only

(c) Shafts subjected to combined twisting and bending moments.

1. Shafts subjected to twisting moment (torque):

The diameter of the shaft may be obtained by using the torsion equation:

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

$T - twisting\, moment\, (torque)$

$J - polar\, moment\, of\, inertia$

$\tau_s - torsional\, shear\, stress$

$r - radius\, of\, the\, shaft$

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

$$\frac{T}{\frac{\pi}{32} \times d^4} = \frac{\tau_s}{\frac{d}{2}}$$

$$T = \frac{\pi}{16} \times \tau_s \times d^3 \quad for\, solid\, shafts$$

And for hollow shafts the polar moment of inertia

$$J = \frac{\pi}{32}\left[(d_o)^4 - (d_i)^4\right] \qquad r = \frac{d_o}{2}$$

$$\frac{T}{\frac{\pi}{32}\left[(d_o)^4 - (d_i)^4\right]} = \frac{\tau_s}{\frac{d_o}{2}} \qquad or$$

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

$$k = \frac{d_i}{d_o} \quad ratio\, of\, inside\, and\, outside\, diameter\, of\, the\, shaft$$

$$T = \frac{\pi}{16} \times \tau_s \times (d_o)^3 \times (1 - k^4)$$

The power transmitted by the shaft (in watts)

$$P = T \times w \qquad w - angular\ velocity\ in\ rad/s$$

$$w = \frac{2\pi N}{60}$$

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

$T -$ twisting moment

$N -$ speed of the shaft in R.P.M

In case of belt drives, the twisting moment is given by

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

$T_1, T_2 -$ tensions in tight and slack side of the belt

$R -$ radius of the pulley

2. Shafts subjected to bending moment:

When the shaft is subjected to a bending moment only then the maximum stress (tensile or compressive) is given by the bending equation:

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

$M -$ bending moment

$I -$ moment of inertia of the shaft about the axis of rotation

$S_b -$ bending stress

$y -$ distance from nutral axis to the outer surface

Moment of inertia for the solid shaft is

$$I = \frac{\pi}{64} \times d^4 \qquad y = \frac{d}{2}$$

$$\therefore \frac{M}{\frac{\pi}{64} \times d^4} = \frac{S_b}{\frac{d}{2}} \qquad or$$

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

And moment of inertia for hollow shaft is

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

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

$$\frac{M}{\frac{\pi}{64} \times (d_o)^4 \times (1 - k^4)} = \frac{S_b}{\frac{d_o}{2}} \qquad or$$

$$M = \frac{\pi}{32} \times S_b \times (d_o)^3 \times (1 - k^4)$$

3. Shafts subjected to combined twisting and bending moments:

(A) Maximum shear stress theory or Guest's theory; it is used for ductile materials such as mild steel

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

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

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

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

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

T_e -Equivalent twisting moment: it is the twisting moment which when acting alone, produces the same shear stress as the actual twisting moment.

(b) Maximum normal stress theory or Rankin's theory it is used for brittle materials such as cast iron.

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

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

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

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

$$\frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = M_e$$

$$\therefore\ M_e = \frac{\pi}{32} \times S_b \times d^3$$

M_e - Equivalent bending moment: it is the moment which when acting alone produces the same tensile or compressive stress as the actual bending moment.

Note: in case of hollow shafts the equivalent equations are written as follows:

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

$$M_e = \frac{\pi}{32} \times S_b \times (d_o)^3 \times (1 - k^4)$$

Post Test

Ex 2: A solid shaft is transmitting 1 MW at 240 R.P.M. determine the diameter of the shaft if the maximum torque transmitted exceeds the mean torque by 20 %. Take the maximum shear stress as 60 MPs.

Key Answer

Pre Test

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

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

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

$$d^{3}=\frac{955\times 10^{3}}{8.25}=115733 \qquad or \quad d=48.7\,mm$$

$$or\ \ d\cong 50mm$$

Post Test

$$T_{mean} = \frac{P \times 60}{2\pi N} = \frac{1 \times 10^6 \times 60}{2\pi \times 240} = 39784 N.m = 39784 \times 10^3\ N.mm$$

$$T_{\max} = 1.2 T_{mean} = 1.2 \times 39784 \times 10^3 = 47741 \times 10^3\ N.mm$$

$$47741 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 60 \times d^3 = 11.78 d^3$$

$$d^3 = \frac{47741 \times 10^3}{11.78} = 4053 \times 10^3 \qquad or \quad d = 159.4\, mm$$

$$or \quad d \cong 160\ mm$$

Reference

R. S. Khurmi, J. K. Gupta, "Theory of machine"

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Design of Journal Bearings

For

Students of second class

Mechanical Department/ Production

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Overview

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

Objectives :-

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

1-Knew the Classification of the sliding bearings.

2-Define the Types of sliding contact bearing.

3-Calculate the Heat dissipated by the bearing.

Flow Chart:-

IF
≥ 9

Next Package

Pre Test

The load on the journal bearing is 150 KN, due to turbine shaft of 300 mm diameter running at 1800 R.P.M. determine the following:

1.length of the bearing if the bearing pressure is 1.6 N/mm ^{2}

2.amount of heat to be removed by lubricant per minute if the bearing temperature is $60\,^{0}C$ and viscosity of the oil is 0.02 kg/ m . s . the bearing clearance is 0.25 mm.

Notes

-10 degrees for the above question.

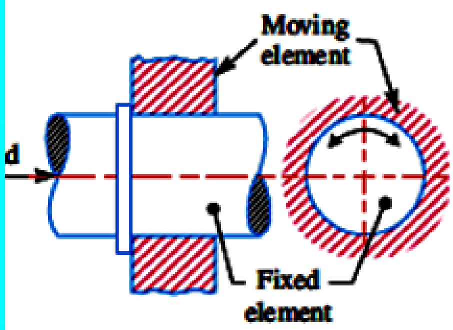
- Check your answers in key answer.

The Text

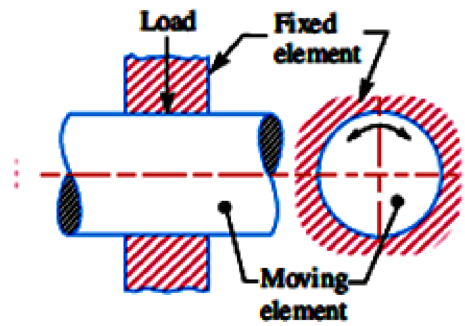
a. depending upon the direction of load

1.radial bearings

2.thrust bearings



Radial bearing.

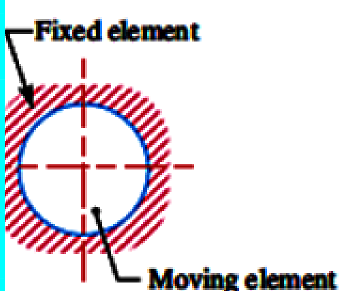


Thrust bearing.

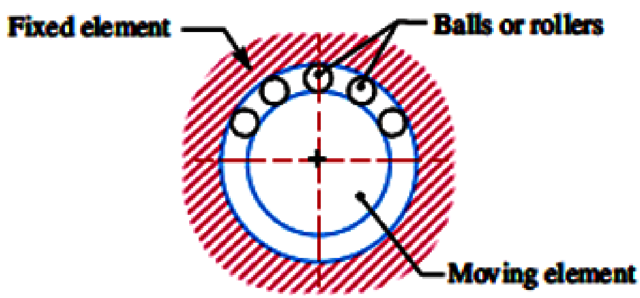
b. depending upon the nature of contact

1.sliding bearings

2.rolling bearings

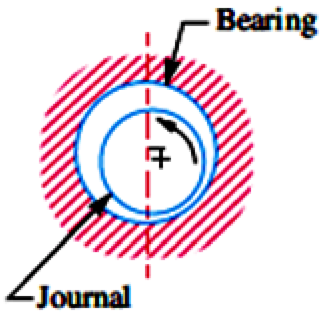


Sliding contact bearing.

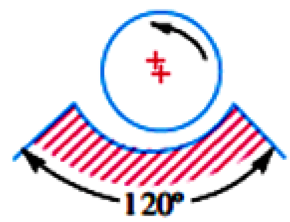


(b) Rolling contact bearings.

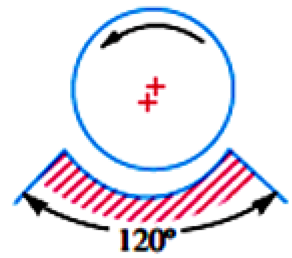
Types of sliding contact bearing



(*a*) Full journal bearing.



(*b*) Partial journal bearing.



(*c*) Fitted journal bearing.

journal or sleeve bearings

lubricant

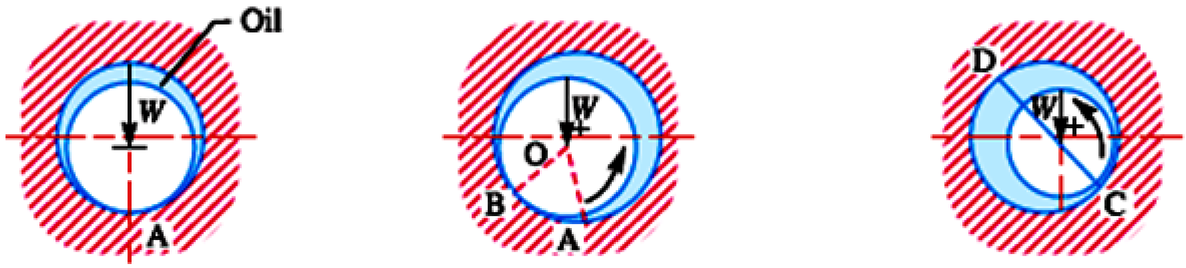
The lubricants are used in bearings to reduce friction between the rubbing surfaces and to carry away the heat generated by friction. It also protect the bearing against corrosion.

Types of lubricant:

1.liquid as : mineral oils

2.semi liquid as : grease

3.solid as : graphite either alone or mixed with oil or grease

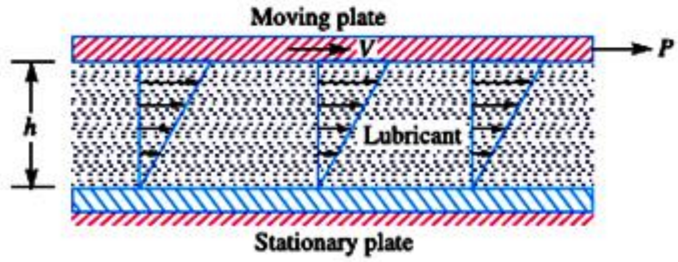


(a) At rest. (b) Slow speed. (c) High speed.

Wedge film journal bearing

Properties of lubricant

1.viscosity Z



Viscosity

$$\tau = Z \times \frac{V}{h}$$

$$Z = \tau \times \frac{h}{V} \qquad N.s/m^2 \ or \ kg/m.s$$

$Z -$ absolute viscosity

$\tau -$ shear stress (lubricant and plate)

$V -$ velocity of lubricant

$h -$ space between two surfaces

2.density ρ

The density of most oils at $15.5^0 C$ is about 900 kg / m^3

Absolute viscosity of commonly used lubricating oils

o.	Type of oil	Absolute viscosity in kg/ m-s at temperature in °C											
		30	35	40	45	50	55	60	65	70	75	80	90
	SAE 10	0.05	0.036	0.027	0.0245	0.021	0.017	0.014	0.012	0.011	0.009	0.008	0.005
	SAE 20	0.069	0.055	0.042	0.034	0.027	0.023	0.020	0.017	0.014	0.011	0.010	0.0075
	SAE 30	0.13	0.10	0.078	0.057	0.048	0.040	0.034	0.027	0.022	0.019	0.016	0.010
	SAE 40	0.21	0.17	0.12	0.096	0.78	0.06	0.046	0.04	0.034	0.027	0.022	0.013
	SAE 50	0.30	0.25	0.20	0.17	0.12	0.09	0.076	0.06	0.05	0.038	0.034	0.020
	SAE 60	0.45	0.32	0.27	0.20	0.16	0.12	0.09	0.072	0.057	0.046	0.040	0.025
	SAE 70	1.0	0.69	0.45	0.31	0.21	0.165	0.12	0.087	0.067	0.052	0.043	0.033

Coefficient of friction of journal

$$\mu=\frac{33}{10^8}\left(\frac{Z\,N}{p}\right)\left(\frac{d}{c}\right)+k$$

$Z - absolute\,viscosity$

$N - speed\ of\ the\ journal\,in\ R.P.M$

$p - bearing\ pressure$

$$p=\frac{W}{A}=\frac{W}{l\times d}$$

$W - load\ on\,the\ journal$

$l\,,d - length\,and\,diameter\,of\ the\ journal$

$c = D-d \qquad\qquad clearance$

$D - bearing\,diameter$

$$k=0.002 \quad for \ \frac{l}{d} \ ratios \ \ is\ about\,0.75 \ to \ \ 2.8$$

Heat generated in a journal bearing:

$$Q_g = \mu . W . V \qquad N.m/s \quad or \quad J/s \quad or \quad Watt$$

$$W = p \times (l \times d)$$

$$V = \frac{\pi d \ N}{60} \qquad m/s$$

$$Q_g = \mu \times p \times l \times d \times \frac{\pi d \ N}{60} \qquad Watt$$

$$V - velocity$$

Heat dissipated by the bearing

$Q_d = C.A(t_b - t_a)$ Watt

$C -$ heat dissipation coeficient Watt $/ m^2 /^0 C$

$C = 140$ to 420 $W / m^2 /^0 C$ for unventilated bearings

$C = 490$ to 1400 $W / m^2 /^0 C$ for well ventilated bearings

$A -$ bearing area

$A = l \times d$

$t_b -$ temperature of the bearing surface in ^0C

$t_a -$ temperature of the air in ^0C

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

$t_o -$ temperature of the lubricant oil not more than $60\,^0C$

Design values for journal bearings.

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

Machinery	Bearing	Maximum bearing pressure (p) in N/mm^2	Operating values			
			Absolute Viscosity (Z) in kg/m-s	ZN/p Z in kg/m-s p in N/mm^2	$\frac{c}{d}$	$\frac{l}{d}$
way cars	Axle	3.5	0.1	7	0.001	1.8 – 2
n turbines	Main	0.7 – 2	0.002 – 0.016	14	0.001	1 – 2
erators, motors, ifugal pumps	Rotor	0.7 – 1.4	0.025	28	0.0013	1 – 2
smission shafts	Light, fixed Self -aligning Heavy	0.175 1.05 1.05	0.025- 0.060	7 2.1 2.1	0.001	2 – 3 2.5 – 4 2– 3
hine tools	Main	2.1	0.04	0.14	0.001	1–4
hing and shearing ines	Main Crank pin	28 56	0.10	—	0.001	1–2
ing Mills	Main	21	0.05	1.4	0.0015	1–1.5

Post Test

A 150 mm diameter shaft supporting a load of 10 KN, has a speed of 1500 R.P.M . the shaft runs in a bearing whose length is 1.5 times the shaft diameter. If the clearance of the bearing is 0.15 mm and the absolute viscosity of the oil is 0.011 kg/m. s . find the power wasted in friction.

Key Answer

Pre Test

$$p = \frac{W}{l \times d}$$

$$l = \frac{W}{p \times d} = \frac{150 \times 10^3}{1.6 \times 300} = 312.5\ mm$$

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

$$= 0.009 + 0.002 = 0.011$$

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

$$Q_g = \mu .W .V = 0.011 \times 150 \times 10^3 \times 28.3 = 46695\ J/s \quad or \quad Watt$$

Post Test

$$l = 1.5d = 1.5 \times 150 = 225\ mm$$

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

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

$$= 0.018 + 0.002 = 0.02$$

$$V = \frac{\pi d\ N}{60} = \frac{\pi \times 0.15 \times 1500}{60} = 11.78\ m/s$$

$$Q_g = \mu.W.V = 0.02 \times 10000 \times 11.78 = 2365\ Watt = 2.365 KW$$

Reference

R. S. Khurmi, J. K. Gupta, "Theory of machine"

Foundation of Technical Education

Al-Dour Technical Institute

Mechanical Department

2^{nd} Stage

Training Package

In

Selection of Ball Bearings

For

Students of second class

Mechanical Department/ Production

By

Nadum I. Naser

Overview

A machine element which supports another moving element(known as rolling) , it permits a relative motion between the contact surface of the members while carrying the load

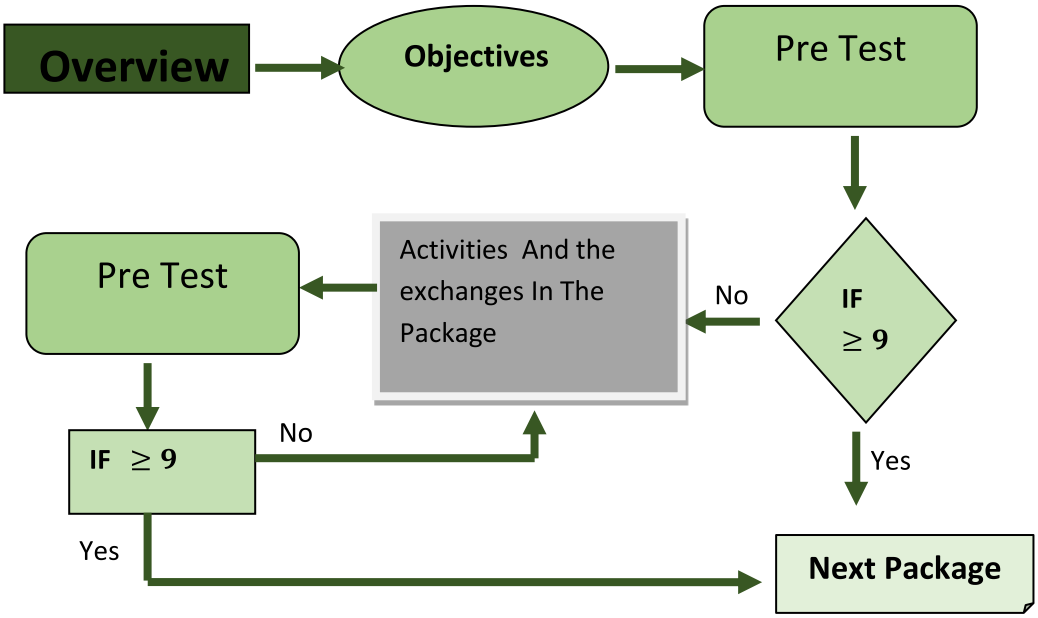
Objectives :-

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

1-Define Types of bearings.

2-Select a ball bearing.

Flow Chart:-



Pre Test

Select a single row deep groove ball bearing for a radial load of 4000 N and an axial load of 5000 N, operating at a speed of 1600 R.P.M , for an average life of 5 years at 10 hours per day .assume uniform and steady load?

Notes

-10 degrees for the above question.

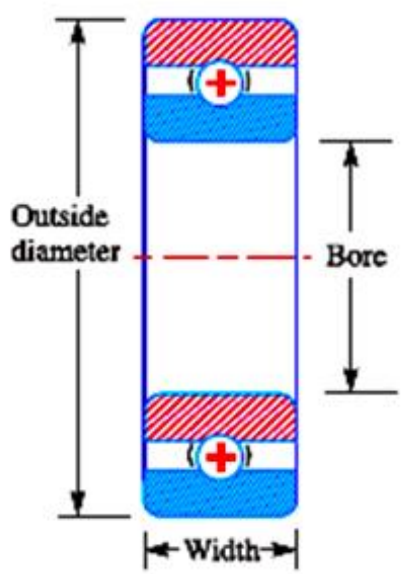
- Check your answers in key answer.

The Text

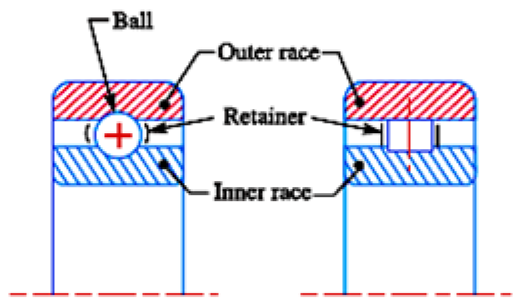
A machine element which supports another moving element(known as rolling) , it permits a relative motion between the contact surface of the members while carrying the load.

Types of bearings

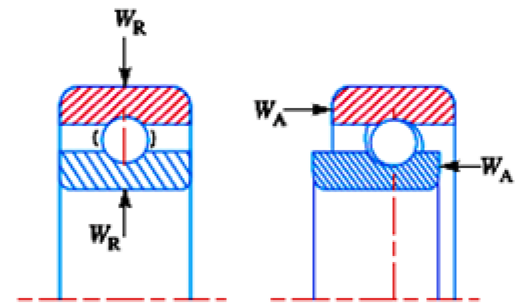
1.ball bearings 2.roller bearings



The ball bearings are usually used for light loads and the roller bearings are used for heavier loads.



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

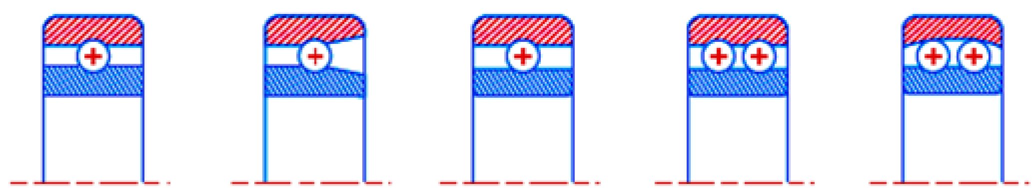


(a) Radial ball bearing. (b) Thrust ball bearing.

the rolling contact bearing, depending upon the load to be carried, are classified as :

a. radial bearings b. thrust bearings

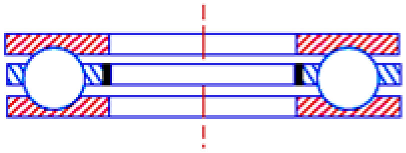
types of radial ball bearings



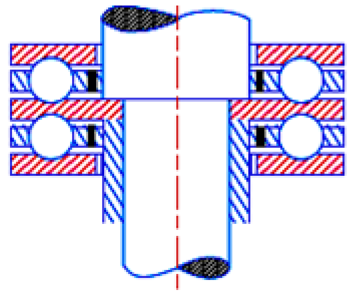
(a) Single row deep groove. (b) Filling notch. (c) Angular contact. (d) Double row. (e) Self-aligning.

Thrust ball bearings

The thrust ball bearings are used for carrying loads at speeds below 2000 R.P.M



(a) Single direction thrust ball bearing.



(b) Double direction thrust ball bearing.

Basic static load rating (C_0) :

it is the static radial load(in case of radial ball or roller bearings) or axial load(in case of thrust ball or roller bearings)which acts on non rotating bearings.

1.for radial ball bearings

$$C_0 = f_0 \times i \times Z \times D^2 \times \cos\alpha$$

f_0 – factor depending on the type of bearing

i – No. of rows of balls in any bearing

Z – No. of balls per row

D – diameter of ball in mm

α – Nomi al angle of contact

2.for radial roller bearings

$$C_0 = f_0 \times i \times Z \times l_e \times D \times \cos\alpha$$

l_e – length of roller

3.for thrust ball bearings

$$C_0 = f_0 \times Z \times D^2 \times \sin\alpha$$

4.for thrust roller bearings

$$C_0 = f_0 \times Z \times l_e \times D \times \sin\alpha$$

Note:

$f_0 -$ May takes the following values: 3.33 , 12.3 , 21.6 , 49 , 98.1

Static equivalent load(W_{0R}):

$$W_{0R} = X_0 W_R + Y_0 W_A$$

$X_0 - radialload\ factor$

$Y_0 - axialor\,thrustload\ factor$

$W_R - radial\ load$

$W_A - axial\ load$

Values of X_0 and Y_0 for radial bearings.

S.No.	Type of bearing	Single row bearing		Double row bearing	
		X_0	Y_0	X_0	Y_0
1.	Radial contact groove ball bearings	0.60	0.50	0.60	0.50
2.	Self aligning ball or roller bearings and tapered roller bearing	0.50	0.22 cot θ	1	0.44 cot θ
3.	Angular contact groove bearings :				
	α = 15°	0.50	0.46	1	0.92
	α = 20°	0.50	0.42	1	0.84
	α = 25°	0.50	0.38	1	0.76
	α = 30°	0.50	0.33	1	0.66
	α = 35°	0.50	0.29	1	0.58
	α = 40°	0.50	0.26	1	0.52
	α = 45°	0.50	0.22	1	0.44

Basic dynamic load rating (C):

$$C = f_c\,(i\ \cos\alpha)^{0.7} \times Z^{\frac{2}{3}} \times D^{1.8}$$

$$f_c - dynamic\ factor\ depending\ on\ the\ geomerty\ of\ bearing\ component$$

Dynamic equivalent load (W):

$$W = X.V.W_R + Y.W_A$$

$$V - rotation\ factor$$

$$\cong 1\ \ for\ all\ types\ of\ bearings\ when\ the\ inner\ race\ is\ rotating$$

Values of *X* and *Y* for dynamically loaded bearings.

Type of bearing	*Specifications*	$\frac{W_A}{W_R} \le e$		$\frac{W_A}{W_R} > e$		*e*
		X	*Y*	*X*	*Y*	
Deep groove	$\frac{W_A}{C_0}$ = 0.025				2.0	0.22
ball bearing	= 0.04				1.8	0.24
	= 0.07				1.6	0.27
	= 0.13	1	0	0.56	1.4	0.31
	= 0.25				1.2	0.37
	= 0.50				1.0	0.44
Angular contact	Single row		0	0.35	0.57	1.14
ball bearings	Two rows in tandem		0	0.35	0.57	1.14
	Two rows back to back	1	0.55	0.57	0.93	1.14
	Double row		0.73	0.62	1.17	0.86
Self-aligning bearings	Light series : for bores					
	10 – 20 mm		1.3		2.0	0.50
	25 – 35	1	1.7	6.5	2.6	0.37
	40 – 45		2.0		3.1	0.31
	50 – 65		2.3		3.5	0.28
	70 – 100		2.4		3.8	0.26
	105 – 110		2.3		3.5	0.28
	Medium series : for bores					
	12 mm		1.0	0.65	1.6	0.63
	15 – 20		1.2		1.9	0.52
	25 – 50		1.5		2.3	0.43
	55 – 90		1.6		2.5	0.39
Spherical roller	For bores :					
bearings	25 – 35 mm		2.1		3.1	0.32
	40 – 45	1	2.5	0.67	3.7	0.27
	50 – 100		2.9		4.4	0.23
	100 – 200		2.6		3.9	0.26
Taper roller	For bores :					
bearings	30 – 40 mm				1.60	0.37
	45 – 110	1	0	0.4	1.45	0.44
	120 – 150				1.35	0.41

Values of service factor (Ks)

S.No.	Type of service	Service factor (K_S) for radial ball bearings
1.	Uniform and steady load	1.0
2.	Light shock load	1.5
3.	Moderate shock load	2.0
4.	Heavy shock load	2.5
5.	Extreme shock load	3.0

the design dynamic load = basic dynamic load × service factor

Life of a bearing(L):

It is the number of revolutions or hours which the bearing runs before the first evidence of fatigue develops in the material.

$$L = \left(\frac{C}{W}\right)^k \times 10^6 \ revolutions$$

$$k = 3 \ for\,ball\,bearings$$

$$k = \frac{10}{3} \ for\,roller\,bearings$$

The relationship between the life in revolutions(L) and the life in working hours ( )is given:

$$L = 60\,N \times L_H$$

$$N - speed\ in\ R.P.M$$

The rating life of a group of bearings is defined as the number of revolutions or hours that 90 percent of a group of bearings will exceed before the first evidence of fatigue develops

The average life of bearing is 5 times the rating life.

The longest life of a single bearing is about 30 –50 times the rating life.

Basic static and dynamic capacities of various types of radial ball bearings

Bearing No.	Basic capacities in kN							
	Single row deep groove ball bearing		Single row angular contact ball bearing		Double row angular contact ball bearing		Self-aligning ball bearing	
	Static (C_0)	Dynamic (C)	Stctic (C_0)	Dynamic (C)	Static (C_0)	Dynamic (C)	Static (C_0)	Dynamic (C)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
200	2.24	4	—	—	4.55	7.35	1.80	5.70
300	3.60	6.3	—	—	—	—	—	—
201	3	5.4	—	—	5.6	8.3	2.0	5.85
301	4.3	7.65	—	—	—	—	3.0	9.15
202	3.55	6.10	3.75	6.30	5.6	8.3	2.16	6
302	5.20	8.80	—	—	9.3	14	3.35	9.3
203	4.4	7.5	4.75	7.8	8.15	11.6	2.8	7.65
303	6.3	10.6	7.2	11.6	12.9	19.3	4.15	11.2
403	11	18	—	—	—	—	—	—
204	6.55	10	6.55	10.4	11	16	3.9	9.8
304	7.65	12.5	8.3	13.7	14	19.3	5.5	14
404	15.6	24	—	—	—	—	—	—
205	7.1	11	7.8	11.6	13.7	17.3	4.25	9.8
305	10.4	16.6	12.5	19.3	20	26.5	7.65	19
405	19	28	—	—	—	—	—	—
206	10	15.3	11.2	16	20.4	25	5.6	12
306	14.6	22	17	24.5	27.5	35.5	10.2	24.5
406	23.2	33.5	—	—	—	—	—	—
207	13.7	20	15.3	21.2	28	34	8	17
307	17.6	26	20.4	28.5	36	45	13.2	30.5
407	30.5	43	—	—	—	—	—	—
208	16	22.8	19	25	32.5	39	9.15	17.6
308	22	32	25.5	35.5	45.5	55	16	35.5
408	37.5	50	—	—	—	—	—	—
209	18.3	25.5	21.6	28	37.5	41.5	10.2	18
309	30	41.5	34	45.5	56	67	19.6	42.5
409	44	60	—	—	—	—	—	—
210	21.2	27.5	23.6	29	43	47.5	10.8	18
310	35.5	48	40.5	53	73.5	81.5	24	50
410	50	68	—	—	—	—	—	—

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
211	26	34	30	36.5	49	53	12.7	20.8
311	42.5	56	47.5	62	80	88	28.5	58.5
411	60	78	—	—	—	—	—	—
212	32	40.5	36.5	44	63	65.5	16	26.5
312	48	64	55	71	96.5	102	33.5	68
412	67	85	—	—	—	—	—	—
213	35.5	44	43	50	69.5	69.5	20.4	34
313	55	72	63	80	112	118	39	75
413	76.5	93	—	—	—	—	—	—
214	39	48	47.5	54	71	69.5	21.6	34.5
314	63	81.5	73.5	90	129	137	45	85
414	102	112	—	—	—	—	—	—
215	42.5	52	50	56	80	76.5	22.4	34.5
315	72	90	81.5	98	140	143	52	95
415	110	120	—	—	—	—	—	—
216	45.5	57	57	63	96.5	93	25	38
316	80	96.5	91.5	106	160	163	58.5	106
416	120	127	—	—	—	—	—	—
217	55	65.5	65.5	71	100	106	30	45.5
317	88	104	102	114	180	180	62	110
417	132	134	—	—	—	—	—	—
218	63	75	76.5	83	127	118	36	55
318	98	112	114	122	—	—	69.5	118
418	146	146	—	—	—	—	—	—
219	72	85	88	95	150	137	43	65.5
319	112	120	125	132	—	—	—	—
220	81.5	96.5	93	102	160	146	51	76.5
320	132	137	153	150	—	—	—	—
221	93	104	104	110	—	—	56	85
321	143	143	166	160	—	—	—	—
222	104	112	116	120	—	—	64	98
322	166	160	193	176	—	—	—	—

Post Test

Design a self aligning ball bearing for a radial load of 7000 N and a thrust load of 2100 N the desired life of bearing is 160 millions of revolutions at 300 R.P.M . Assume uniform and steady load.

Key Answer

Pre Test

$$L_H = 5 \times 300 \times 10 = 15000 \ hours \qquad (300 \ working \ day \ per \ year)$$

$$L = 60\, N \times L_H = 60 \times 1600 \times 15000 = 1440 \times 10^6 \ revolutions$$

From the table for dynamically loaded bearings let us take the value $\dfrac{W_A}{C_0} = 0.5$

$$\frac{W_A}{W_R} = \frac{5000}{4000} = 1.25 \qquad which\ is\ greater\ than \quad e = 0.44$$

$$then \quad X = 0.56 \qquad Y = 1$$

$$W = X\,V\,W_R + Y\,W_A \qquad V = 1$$

$$= 0.56 \times 1 \times 4000 + 1 \times 5000 = 7240\ N$$

$$L = \left(\frac{C}{W}\right)^k \times 10^6 \quad rev.$$

$$C = W\left(\frac{L}{10^6}\right)^{\frac{1}{k}} = 7240 \times \left(\frac{1440 \times 10^6}{10^6}\right)^{\frac{1}{3}} = 81760 N = 81.76\ KN$$

$$from\ the\ table\ we\ select\ the\ bearing\ No.315$$

Post Test

$$\frac{W_A}{W_R}=\frac{2100}{7000}=0.3$$

$From\,table\; we\; find\,that\; for\,a\,self\; aligning\,ball\,bearing$

$X=0.65 \qquad\qquad\qquad Y=3.5$

$W=X\,V\,W_R+Y\,W_A=0.65\times1\times7000+3.5\times2100=11900\;N$

$from\,table\,we\; find\,that\,the\,service\,factor\;K_s\;=1$

$\therefore the\,bearing\,should\,be\,selected\; for\;W=11900\;N$

$$L=\left(\frac{C}{W}\right)^k\times10^6\quad rev.$$

$$C=W\left(\frac{L}{10^6}\right)^{1/k}=11900\times\left(\frac{160\times10^6}{10^6}\right)^{1/3}=64600\;N=64.6\;KN$$

$from\;table\,we\,select\;bearing\,number\,219\;having\,C=65.5\,KN$

Reference

R. S. Khurmi, J. K. Gupta, "Theory of machine"

Foundation of Technical Education

Al-Dour Technical Institute

Mechanical Department

2nd Stage

Training Package

In

Design of Gears by Lewis Equation

For

Students of second class

Mechanical Department/ Production

By

Nadum I. Naser

Overview

Gears are an adaptation of rolling cylinders and cones, designed to ensure positive motion. Gears are used to transmit power between shafts rotating usually at different speeds.

Objectives :-

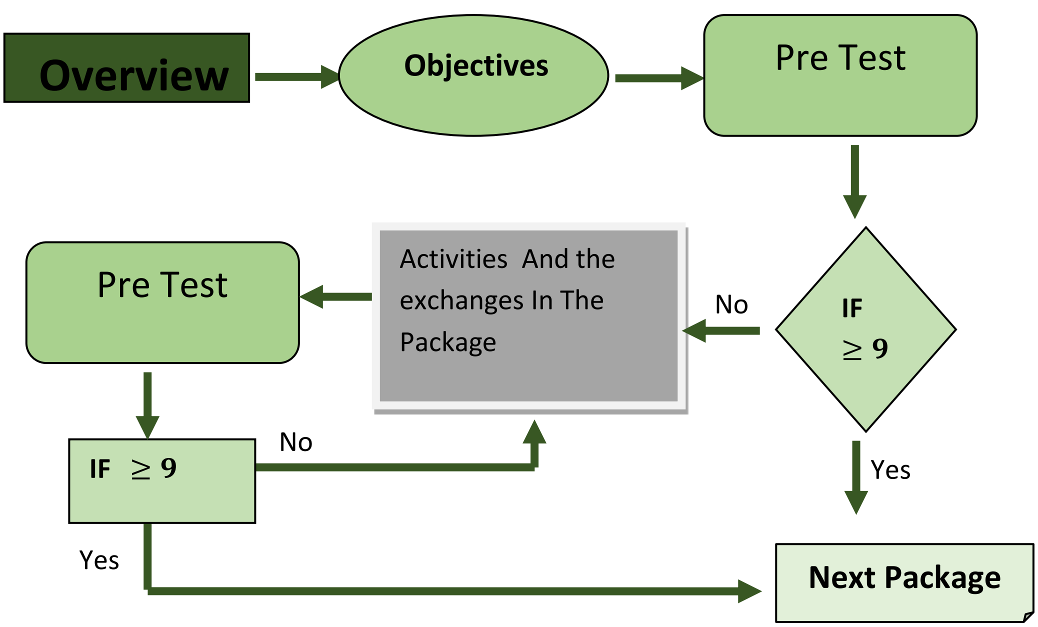
After studying the first modular unit , the student will be able to:-

1-Define Types of Gears.

2- Design spur gears.

3-Calculate Strength of gear teeth –Lewis Equation.

Flow Chart:-



Pre Test

Two parallel shafts, about 600 mm apart are to be connected by spur gears, one shaft is to run at 360 R.P.M , and the other at 120 R.P.M , design the gears if the circular pitch is to be 25 mm?

Notes

-10 degrees for the above question.

- Check your answers in key answer.

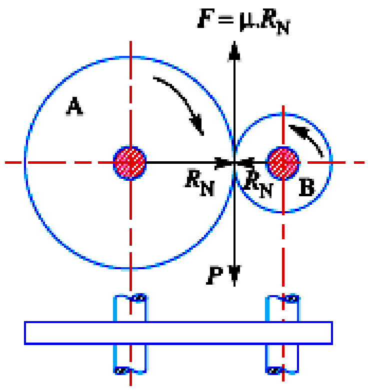
The Text

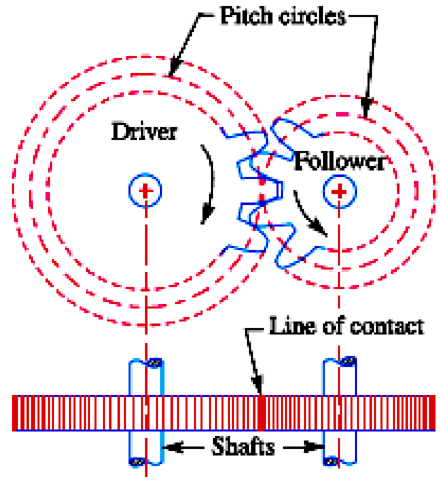
Gears are an adaptation of rolling cylinders and cones, designed to ensure positive motion.

Gears are used to transmit power between shafts rotating usually at different speeds.

Types of Gears

Spur gears, have teeth parallel to the axis of rotation and are used to transmit motion
from one shaft to another, parallel, shaft. Of all types, the spur gear is the simplest .

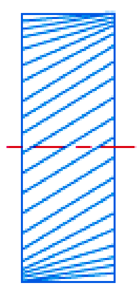




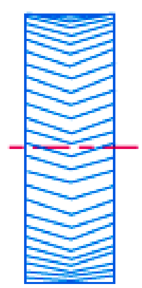
Friction wheels.

Gear or toothed wheel.

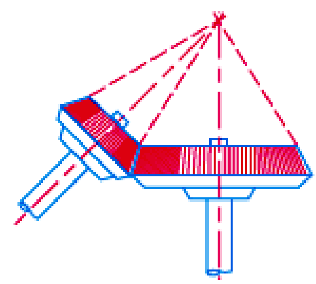
Helical gears have teeth inclined to the axis of rotation. Helical gears can be used for
the same applications as spur gears and, when so used, are not as noisy, because of the
more gradual engagement of the teeth during meshing. The inclined tooth also develops thrust loads and bending couples, which are not present with spur gearing.
Sometimes helical gears are used to transmit motion between nonparallel shafts.



Single helical gear.



Double helical gear.

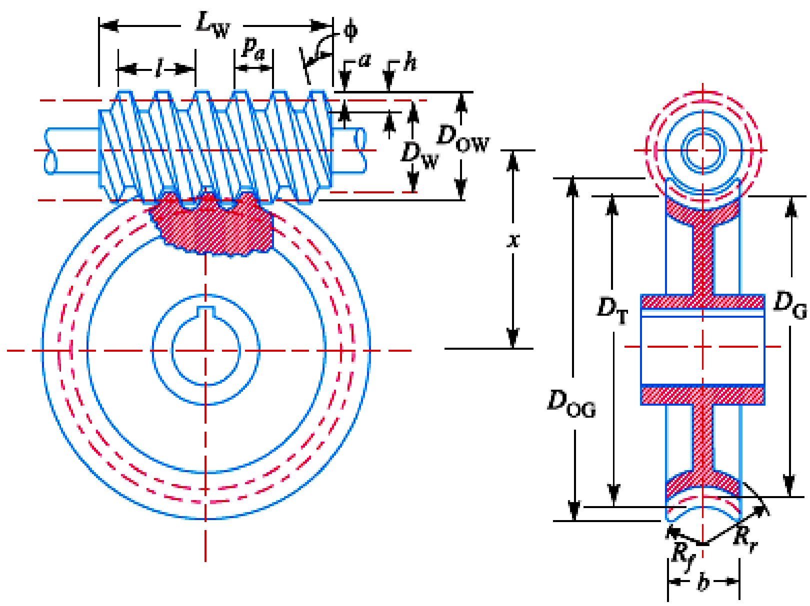


Bevel gear.

Bevel gears, have teeth formed on conical surfaces and are used mostly for transmitting motion between intersecting shafts.

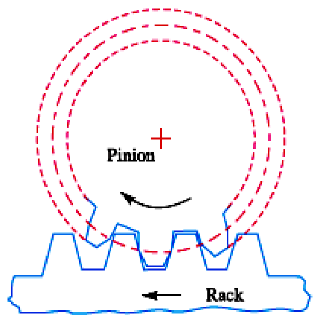
Worms and *worm gears*, represent the fourth basic gear type. the worm resembles a screw.
The direction of rotation of the worm gear, also called the worm wheel, depends upon
the direction of rotation of the worm and upon whether the worm teeth are cut
right-hand or left-hand. Worm-gear sets are also made so that the teeth of one or both wrap
partly around the other. Worm-gear sets are mostly used when the speed ratios of the two

shafts are quite high, say, 3 or more.

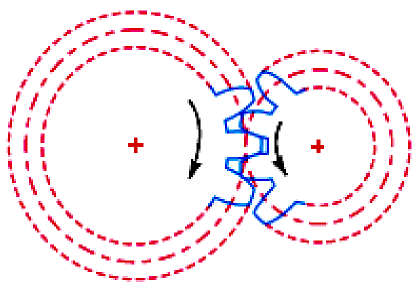


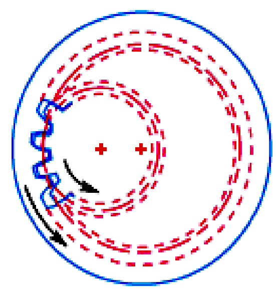
The gears are classified according to the type of gearing as :

a)external gearing b)internal gearing c)rack and pinion



rack and pinion





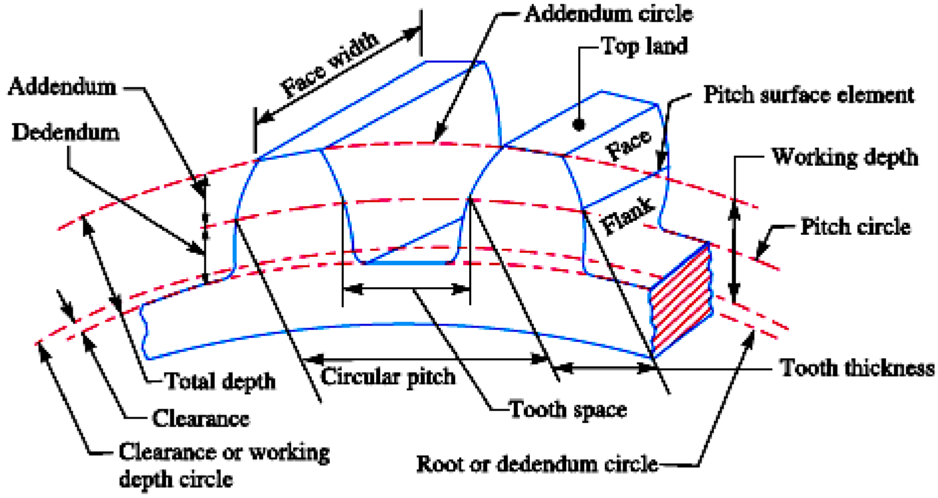
External gearing

Internal gearing

Terms used in gears

1.pitch circle: it is an imaginary circle which by pure rolling action , would give the same motion as the actual gear.

2.pitch circle diameter: it is the diameter of pitch circle.



3.pressure angle: it is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point. It is usually denoted by Φ. The standard pressure angles are 14 $1/2^0$ and 20^0.

4.addendum: it is the radial distance of a tooth from the pitch circle to the top of the tooth.

5.dedendum: it is the radial distance of a tooth from the pitch circle to the bottom of the tooth.

6.circular pitch: it is the distance measured on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth .

$$P_c = \frac{\pi D}{T}$$

D – diameter of pitch circle

T – number of teeth on the wheel.

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

$$P_c = \frac{\pi D_1}{T_1} = \frac{\pi D_2}{T_2} \qquad or \qquad \frac{D_1}{D_2} = \frac{T_1}{T_2}$$

7.diametral pitch: it is the ratio of number of teeth of the pitch circle diameter in millimeters.

$$P_d = \frac{T}{D} = \frac{\pi}{P_c}$$

8.module: it is the ratio of the pitch circle diameter in millimeters to the number of teeth.

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

Note1: the standard modules are 1 , 1.25 , 1.5 , 2 , 2.5 , 3 , 4 , 5, 6 , 8 , 10 , 12 , 16 , 20 , 25 , 32 , 40 and 50

The modules 1.125,1.375, 1.75 , 2.25, 2.75, 3.5 , 4.5 ,5.5 , 7, 9, 11 , 14 , 18 , 22 , 28 , 36 and 45 are of second choice.

Note2:if D_1 and D_2 are pitch circle diameters of gears 1 and 2 then the velocity ratio is

$$\frac{w1}{w1} = \frac{D_2}{D_1} = \frac{T_2}{T_1}$$

Design of spur gears:

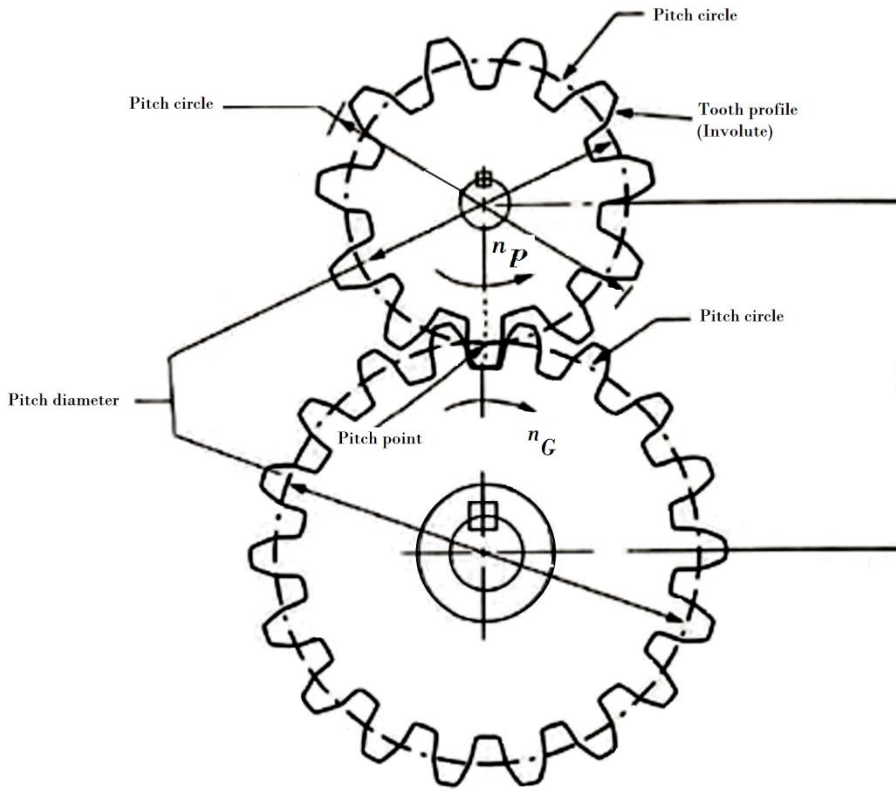
Sometimes, the spur gears (driver and driven) are to be designed for the given velocity ratio and the distance between the centers of their shafts.

Let x – distance between the centers of two shafts

N_1, N_2 - speed of the driver and driven

T_1, T_2 – number of teeth of the driver and driven

d_1 , d_2 – pitch circle diameter of the driver and the driven



$$x = \frac{d_1 + d_2}{2}$$

And speed ratio or velocity ratio

$$\frac{N_1}{N_2} = \frac{d_2}{d_1} = \frac{T_2}{T_1}$$

Systems of gear teeth:

1. 14 1/2^0 composite system

2. 14 1/2^0 full depth involute system

3. 20^0 full depth involute system

4. 20^0 stub involute system (has a strong tooth to take a heavy loads)

Standard proportions of gear systems:

The following table shows the standard proportions in module (m) for the four gear systems

Standard proportions of gear systems.

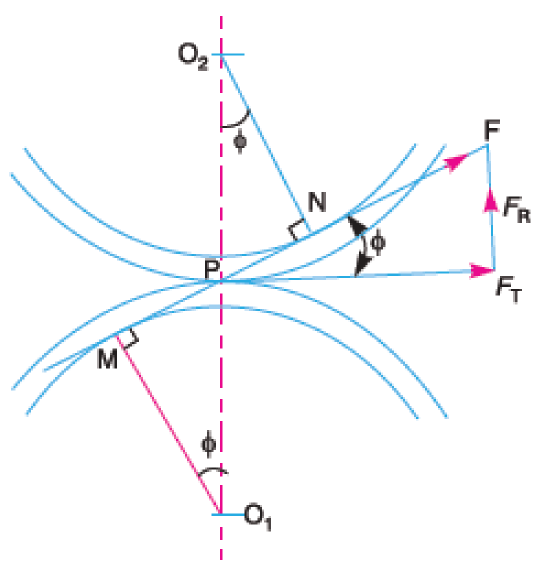
S. No.	Particulars	$14\frac{1}{2}^\circ$ composite or full depth involute system	20° full depth involute system	20° stub involute system
1.	Addendum	1*m*	1*m*	0.8 *m*
2.	Dedendum	1.25 *m*	1.25 *m*	1 *m*
3.	Working depth	2 *m*	2 *m*	1.60 *m*
4.	Minimum total depth	2.25 *m*	2.25 *m*	1.80 *m*
5.	Tooth thickness	1.5708 *m*	1.5708 *m*	1.5708 *m*
6.	Minimum clearance	0.25 *m*	0.25 *m*	0.2 *m*
7.	Fillet radius at root	0.4 *m*	0.4 *m*	0.4 *m*

If *F* is the maximum tooth pressure, then tangential force $F_T = F\cos\phi$

and radial or normal force $F_N = F\sin\phi$

Torque exerted on the gear shaft $T = F_T \times r$

r – radius of the pitch circle of the gear



Minimum number of teeth on the pinion

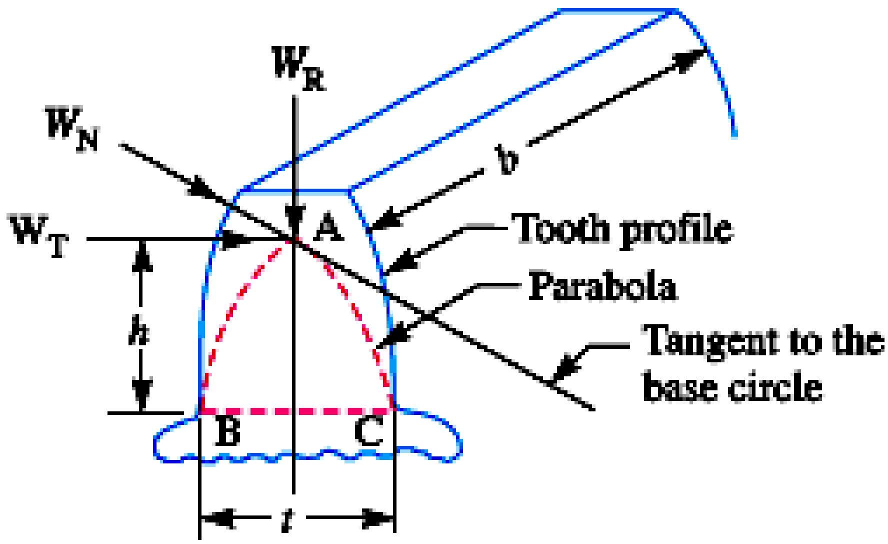
Minimum number of teeth on the pinion

S. No.	*Systems of gear teeth*	*Minimum number of teeth on the pinion*
1.	$14\frac{1}{2}°$ Composite	12
2.	$14\frac{1}{2}°$ Full depth involute	32
3.	20° Full depth involute	18
4.	20° Stub involute	14

Spur gears

Strength of gear teeth –Lewis Equation

The maximum bending stress(working stress) at the section BC is :



W_R
W_N
W_T
A
b
Tooth profile
Parabola
h
Tangent to the
base circle
B
C
t

$$S_W = \frac{M \cdot y}{I}$$

$\therefore M -$ *maximum bending moment at the critical se tion*

$$M = W_T \times h$$

$W_T -$ *tangential load acting at the tooth*

$h -$ *length of the tooth*

$y -$ *half the thickness of the tooth*

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

$I -$ *moment of inertia about the centre line of the tooth*

$$I = \frac{b \cdot t^3}{12}$$

$b -$ *width of gear face*

$$\therefore \quad S_W = \frac{(W_T \times h) \times \frac{t}{2}}{\frac{b \times t^3}{12}} = \frac{(W_T \times h) \times 6}{b \times t^2}$$

$$W_T = \frac{S_W \times b \times t^2}{6\,h} \qquad let \qquad t = x \times p_c \qquad h = k \times p_c$$

$$W_T = S_W \times b \times \frac{x^2 \times p_c{}^2}{6 \times k \times p_c} = S_W \times b \times p_c \times \frac{x^2}{6 \times k}$$

$$let \quad y = \frac{x^2}{6 \times k}$$

$$W_T = S_W \times b \times p_c \times y = S_W \times b \times \pi m \times y$$

$S_W = S_0 \times C_v$

$S_0 - allowable\ static\ stress$

$C_v\ - velocity\ \ factor$

$C_v = \frac{3}{3+v}$ $for\ ordinary\ cut\ \ gears operating\ at\ velocities\ upto\ 12.5\ m/s$

$= \frac{4.5}{4.5+v}$ $for\ carefully\ cut\ \ gears operating at\ velocities\ upto\ 12.5\ \ m/s$

$= \frac{6}{6+v}$ $for\ very\ accurately\ cut\ \ and\ \ ground\ metallic\ gears\ operating$

$at\ velocities\ upto\ \ 20\ m/s$

$= \left(\frac{0.75}{1+v}\right) + 0.25 \quad for\ non-metallic\ gears$

$v - pitch\ line\ velocity\ in \quad m/s$

$$v = \frac{\pi\ D\ N}{60}$$

Design procedure for spur gears:

1.tangential tooth load is obtained from the power transmitted and the pitch line velocity:

$$W_T = \frac{P}{v} \times C_S$$

$W_T - tangential\ tooth\ load\ \ in\ \ newtons$

$P - power\ transmitted\ \ \ in\ \ watts$

$C_S - service\ \ factor$

Values of service factor.

Type of load	Type of service		
	Intermittent or 3 hours per day	8-10 hours per day	Continuous 24 hours per day
Steady	0.8	1.00	1.25
Light shock	1.00	1.25	1.54
Medium shock	1.25	1.54	1.80
Heavy shock	1.54	1.80	2.00

2.apply the Lewis equation as :

$$W_T = S_W \times b \times p_c \times y = S_W \times b \times \pi\, m \times y = (S_0 \times C_v) \times b \times \pi\, m \times y$$

$$y = 0124 - \frac{0.684}{T} \quad for \;\; 14\tfrac{1}{2}^0 \quad coposite\ and\ \ full\ \ depth\ involute\ system$$

$$= 0.154 - \frac{0.912}{T} \quad for \;\; 20^0 \quad full\ \ depth\ \ involute\ system$$

$$= 0.175 - \frac{0.841}{T} \quad for\ 20^0 \quad stub\ system$$

$$y - tooth\ \ form\ \ factor$$

3.find the static load W_S:

$$W_S = S_0 \times b \times p_c \times y = S_0 \times b \times \pi\, m \times y$$

W_S should be greater than W_D

4. for steady loads $W_S \geq 1.25\ W_D$

For pulsating loads $W_S \geq 1.35\ W_D$

For shock loads $W_S \geq 1.5\ W_D$

Post Test

A bronze spur pinion rotating at 600 R.P.M. drives a cast iron spur gear at a transmission ratio of 4 : 1 the allowable static stresses for the bronze pinion is 48 MPs and for cast iron gear is 105 MPs.

The pinion has 16 standard 20^0 full depth involute teeth of module 8 mm. the face width of both the gears is 90 mm. find the power that can be transmitted.

Key Answer

Pre Test

$$\frac{N_1}{N_2}=\frac{d_2}{d_1}=\frac{360}{120}=3$$

$$d_2=3d_1$$

$$x=\frac{d_1+d_2}{2}$$

$$600=\frac{d_1+d_2}{2}$$

$$d_1+d_2=1200$$

$$d_1+3d_1=1200$$

$$d_1=\frac{1200}{4}=300mm \qquad \Rightarrow d_2=3d_1=3\times 300=900mm$$

$$T_1=\frac{\pi d_1}{P_c}=\frac{\pi\times 300}{25}=37.7$$

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

The number of teeth on both gears should be in complete numbers, then for the first gear we take T_1 = 38

And the number of the second gear should be T2 = 3 × 38 = 114

Now the exact pitch circle diameter of the first gear is

$$d_1{}'=\frac{T_1\times P_c}{\pi}=\frac{38\times 25}{\pi}=302.36mm$$

And the exact pitch circle diameter of the second gear is

$$d_2{}'=\frac{T_2\times P_c}{\pi}=\frac{114\times 25}{\pi}=907.1mm$$

Exact distance between the two shafts is

$$x'=\frac{d_1{}'+d_2{}'}{2}=\frac{302.26+907.1}{2}=604.73mm$$

Post Test

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

$$D_P=m.T_P=8\times16=128mm=0.128\ m$$

$$\therefore v=\frac{\pi\,D_P\,.N_P}{60}=\frac{\pi\times0.128\times600}{60}=4.02\,m/s$$

v *is lessthan* $12.5\ m/s$

$$\therefore C_v=\frac{3}{3+v}=\frac{3}{3+4.02}=0.427$$

for 20^0 *full depth involute tooth,*

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

$$W_T=S_{WP}.b.\pi\ m.y_P=(S_{0P}\times C_v)\,b.\pi\,m.y_P$$

$$=84\times0.427\times90\times\pi\times8\times0.097=7870N$$

$$W_T=\frac{P}{v}\times C_S$$

for steady load $8-10$ *hours per day* $C_S=1$

$$P=W_T\times v=7870\times4.02=31640Watt=31.64\ KW$$

Reference

R. S. Khurmi, J. K. Gupta, "Theory of machine"

Foundation of Technical Education

Al-Dour Technical Institute

Mechanical Department

2nd Stage

Training Package

In

Gears Trains

For

Students of second class

Mechanical Department/ Production

By

Nadum I. Naser

Overview

Sometimes, two or more gears are made to mesh with each other to transmit power from one shaft to another. Such a combination is called ***gear train*** or ***train of toothed wheels.*** The nature of the train used depends upon the velocity ratio required and the relative position of the axes of shafts. A gear train may consist of spur, bevel or spiral gears.

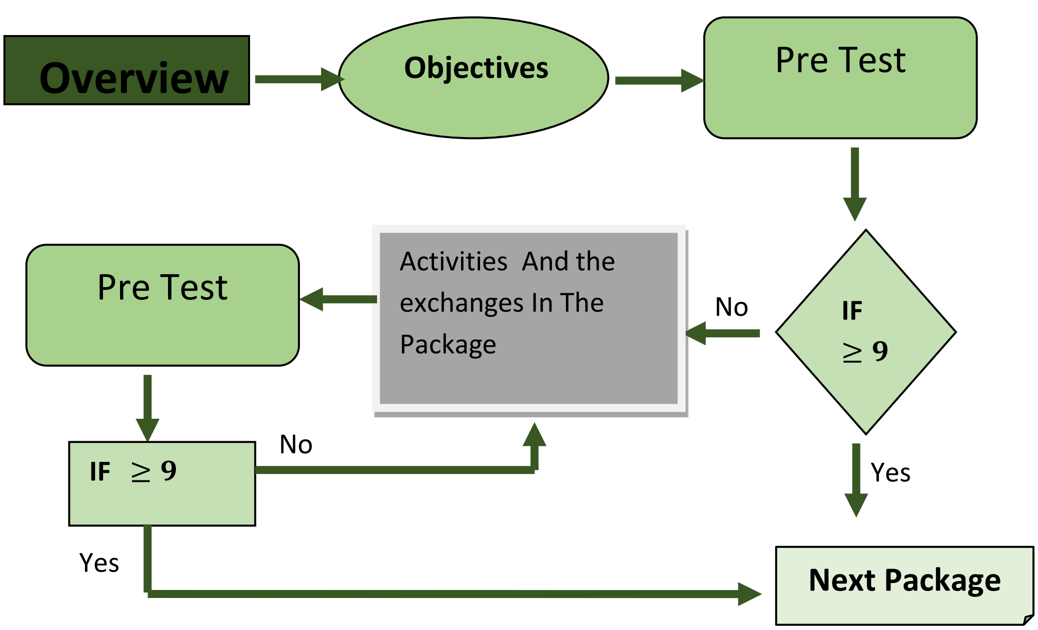
Objectives :-

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

1-Define Types of gear trains.

2-Calculate the speed ratio of gears.

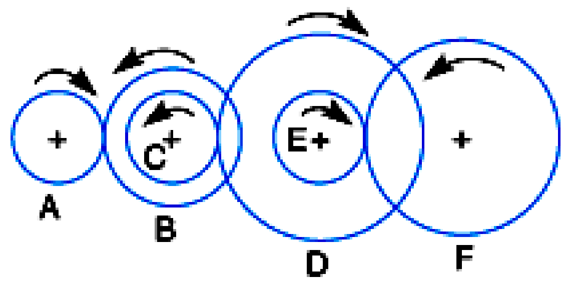
Flow Chart:-



Pre Test

The motor shaft is connected to gear A and rotates at 975 R.P.M . the gear wheels B,C,D and E are fixed to parallel shafts rotating together. The final gear F is fixed on the output shaft. What is the speed of gear F. the number of teeth on each gear are as in below:

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



Notes

-10 degrees for the above question.

- Check your answers in key answer.

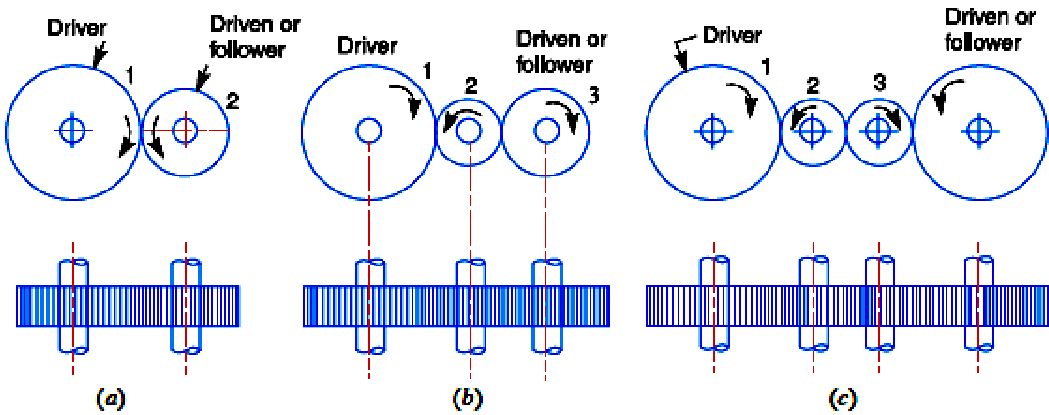
The Text

It is a combination of two or more gears are made to mesh with each other to transmit power from one shaft to another.

Types of gear trains

1.simple gear train 2. Compound gear train 3.reverted gear 4. Epicyclic gear train

Simple gear train: it is a train in which each shaft carries one gear only.



Simple gear train.

Gear 1 drives the gear 2, gear 1 is called driver , gear 2 is called driven or follower

The motion of the driven gear is opposite to the motion of the driving gear.

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

N_1-speed of gear 1 in R.P.M

N_2- speed of gear 2 in R.P.M

T_1- number of teeth on gear 1

T_2- number of teeth on gear 2

As shown in figure (a)

In figure (b) an intermediate gear 2 is in mesh with the drive gear 1 and the driven gear 3

The speed ratio of gear 1 and gear 2 $= \frac{N_1}{N_2} = \frac{T_2}{T_1}$

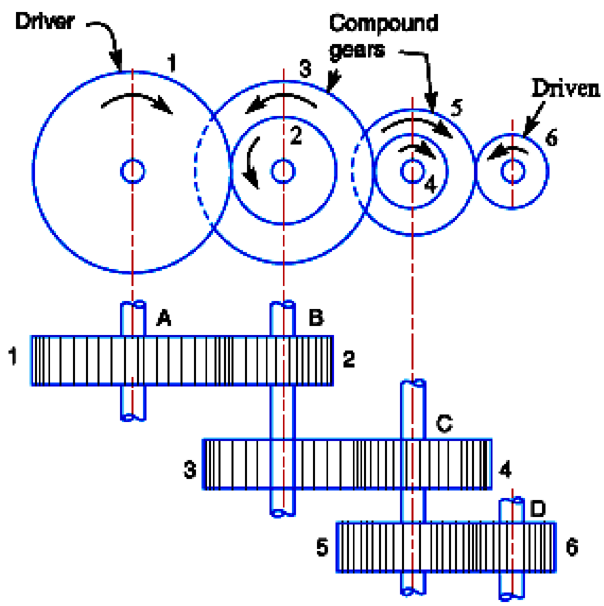
The speed ratio of gear 2 and gear 3 $= \frac{N_2}{N_3} = \frac{T_3}{T_2}$

The total speed ratio of the gear train is $\frac{N_1}{N_2} \times \frac{N_2}{N_3} = \frac{T_2}{T_1} \times \frac{T_3}{T_2} \quad or \quad \frac{N_1}{N_3} = \frac{T_3}{T_1}$

$$\therefore \; speed\,ratio = \frac{speed\,of\;driver}{speed\,of\;driven} = \frac{No.of\;teeth\,of\;driven}{No.of\;teeth\,of\;driver}$$

Compound gear train: it is a train in which each shaft except the first and last carries two gears.

The two gears which are keyed to the same shaft must revolve at the same speed



Compound gear train.

The speed ratio for gear 1 and gear 2 is $= \frac{N_1}{N_2} = \frac{T_2}{T_1}$

And for gears 3 and 4 speed ratio is $= \frac{N_3}{N_4} = \frac{T_4}{T_3}$

And for gears 5 and 6 speed ratio is $= \frac{N_5}{N_6} = \frac{T_6}{T_5}$

The total speed ratio for the above gear train is

$$\frac{N_1}{N_2} \times \frac{N_3}{N_4} \times \frac{N_5}{N_6} = \frac{T_2}{T_1} \times \frac{T_4}{T_3} \times \frac{T_6}{T_5}$$

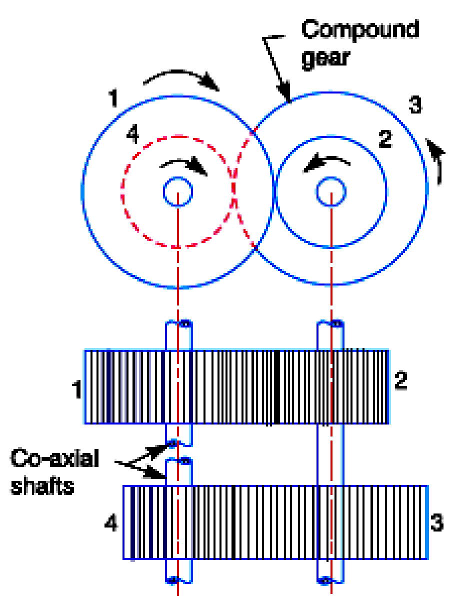
$N_2 = N_3$ and $N_4 = N_5$

$$\therefore \frac{N_1}{N_6} = \frac{T_2 \times T_4 \times T_6}{T_1 \times T_3 \times T_5}$$

Speed ratio for compound gear train

$$= \frac{speed\, of\ the\ first\, driver}{speed\, of\ the\, last\, driven} = \frac{product\, of\ No.\, of\ teeth\ on\ the\ drivens}{product\, of\ No.\, of\ teeth\ on\ the\ drivers}$$

Reverted gear train: it is a train in which the first and the last gear are co-axial.



Reverted gear train.

Gear 1 drives gear 2 in the opposite direction ,gear 2 and gear 3 are mounted on the same shaft(compound gears) , gear 3 as a driver rotates gear 4 (the last follower).

$r_1 + r_2 = r_3 + r_4$

r_1, r_2, r_3, r_4 – radii of respective gears

Note: number of teeth on each gear is directly proportional to its circumference or radius.

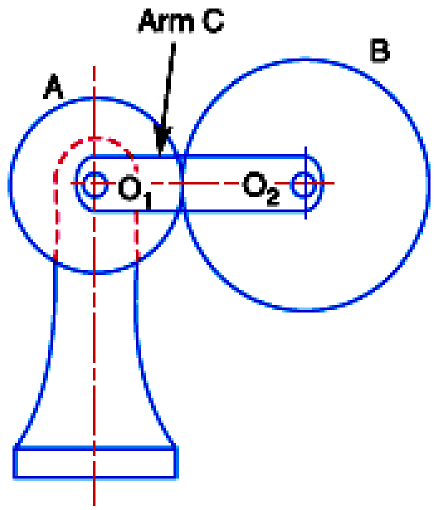
$T_1 + T_2 = T_3 + T_4$

$$speed\ ratio = \frac{product\ of\ No.of\ teeth\ on\ drivens}{product\ of\ No.\ of\ teeth\ on\ drivers}$$

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

The reverted gear trains are used in automotive transmissions , lathe back gears, industrial speed reducers and in clocks.

Epicyclic gear train: it is a train in which the axes of the shafts over which the gears are mounted may move relative to a fixed axis.



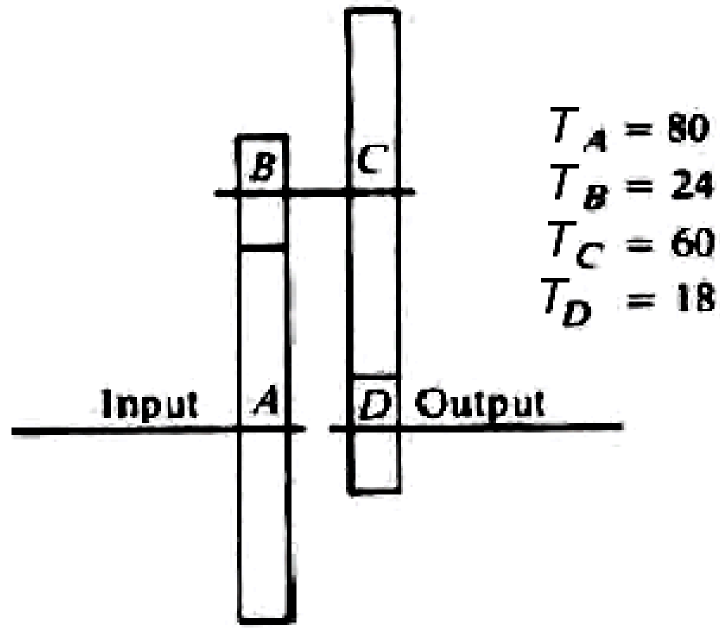
Epicyclic gear train.

gear A and the arm C have a common axis at O_1, about which they can rotate. The gear B meshes with gear A and has its axis on the arm at O_2, about which the gear B can rotate.

The epicyclic gear train may be simple or compound, so they are useful for transmitting high velocity ratios as in back gear of lathe, differential gears of automobiles, hoists, pulley blocks, wrist watches, etc ..

Post Test

Compute the output speed and the direction of rotation of the output shaft if the input shaft rotates at 1750 R.P.M clockwise.



Key Answer

Pre Test

$$\frac{speed\ of\ the\ first\ driver}{speed\ of\ the\ last\ driven} = \frac{product\ of\ No.\ of\ teeth\ on\ the\ drivens}{product\ of\ No.\ of\ teeth\ on\ the\ drivers}$$

$$or \qquad \frac{N_A}{N_F} = \frac{T_B \times T_D \times T_F}{T_A \times T_C \times T_E}$$

$$= \frac{50 \times 75 \times 65}{20 \times 25 \times 26} = 18.75$$

$$\therefore \qquad N_F = \frac{N_A}{18.75} = \frac{975}{18.75} = 52\ R.P.M$$

Post Test

$$\frac{speed\,of\ the\ first\,driver}{speed\,of\ the\,last\,driven}=\frac{product\,of\ No.of\ teeth\ on\ the\ drivens}{product\,of\ No.of\ teeth\ on\ the\ drivers}$$

$$or \qquad \frac{N_A}{N_D}=\frac{T_B\times T_D}{T_A\times T_C}$$

$$=\frac{24\times 18}{80\times 60}=0.09$$

$$\therefore \qquad N_D=\frac{N_A}{0.09}=\frac{1750}{0.09}=19444\ R.P.M$$

The direction of rotation is the same as the input shaft(clockwise)

Reference

R. S. Khurmi, J. K. Gupta, "Theory of machine"

Foundation of Technical Education

Al-Dour Technical Institute

Mechanical Department

2nd Stage

Training Package

In

Worm Gears

For

Students of second class

Mechanical Department/ Production

By

Nadum I. Naser

Overview

Worm Gears are right angle drives providing large speed ratios on comparatively short center distances from 1/4” to 11”. When properly mounted and lubricated they function as the quietist and smoothest running type of gearing. Because of the high ratios possible with worm gearing,

maximum speed reduction can be accomplished in less space than many other types of gearing. Worm and worm gears operate on non-intersecting shafts at 90 angles.

The worm gears are widely used for transmitting power at high velocity ratios between non-intersecting shafts that are generally, but not necessarily, at right angles. It can give velocity ratios as high as 300 : 1 or more in a single step in a minimum of space, but it has a lower efficiency. The worm gearing is mostly used as a speed reducer, which consists of worm and a worm wheel or gear. The worm (which is the driving member) is usually of a cylindrical form having threads of the same shape as that of an involute rack. The threads of the worm may be left handed or right handed and single or multiple threads. The worm wheel or gear (which is the driven member) is similar to a helical gear with a face curved to conform to the shape of the worm. The worm is generally made of steel while the worm gear is made of bronze or cast iron for light service.

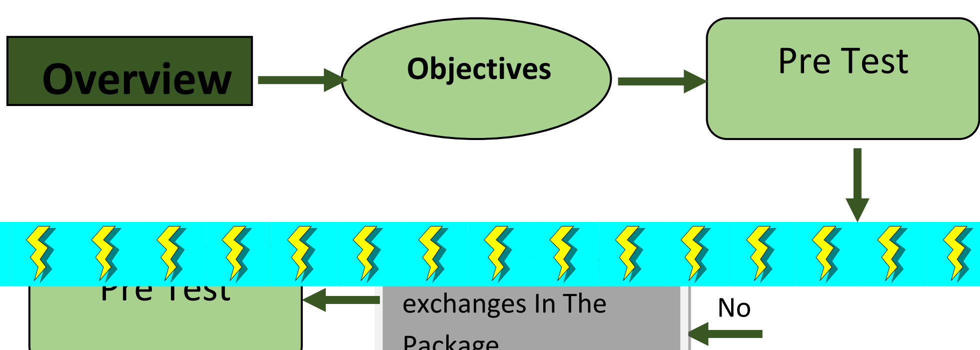
Objectives :-

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

1-Define Types of worm gear.

2-Calculate the EFFICIENCY of worm gear drives.

Flow Chart:-



IF

≥ 9

Next Package

Pre Test

1. Discuss, with neat sketches, the various types of worms and worm gears.
2. Define the following terms used in worm gearing :

   (*a*) Lead; (*b*) Lead angle; (*c*) Normal pitch; and (*d*) Helix angle.
3. What are the various forces acting on worm and worm gears ?
4. Write the expression for centre distance in terms of axial lead, lead angle and velocity ratio.

Notes

-10 degrees for the above question.

- Check your answers in key answer.

The Text

The worm gearing is classified as non-interchangeable, because a worm wheel cut with a hob of one diameter will not operate satisfactorily with a worm of different diameter, even if the thread pitch is same.

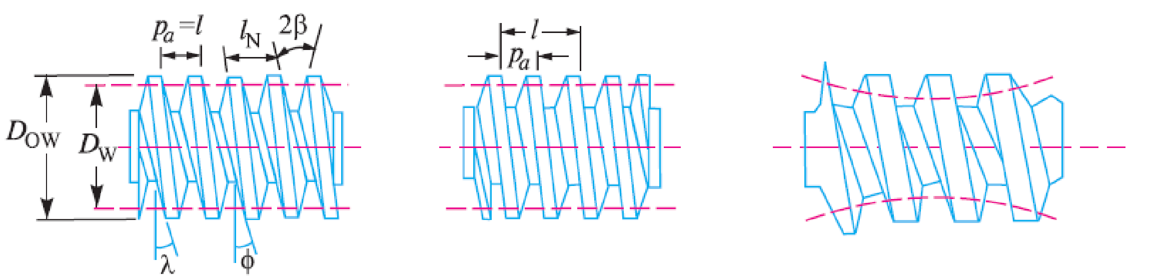
Types of Worms

The following are the two types of worms :

1. Cylindrical or straight worm, and
2. Cone or double enveloping worm.

The ***cylindrical*** or ***straight worm***, as shown in Fig. 31.1 (*a*), is most commonly used. The shape of the thread is involute helicoid of pressure angle 14 ½° for single and double threaded worms and 20° for triple and quadruple threaded worms. The worm threads are cut by a straight sided milling cutter having its diameter not less than the outside diameter of worm or greater than 1.25 times the outside diameter of worm.

The ***cone*** or ***double enveloping worm***, as shown in Fig. 31.1 (*b*), is used to some extent, but it requires extremely accurate alignment.



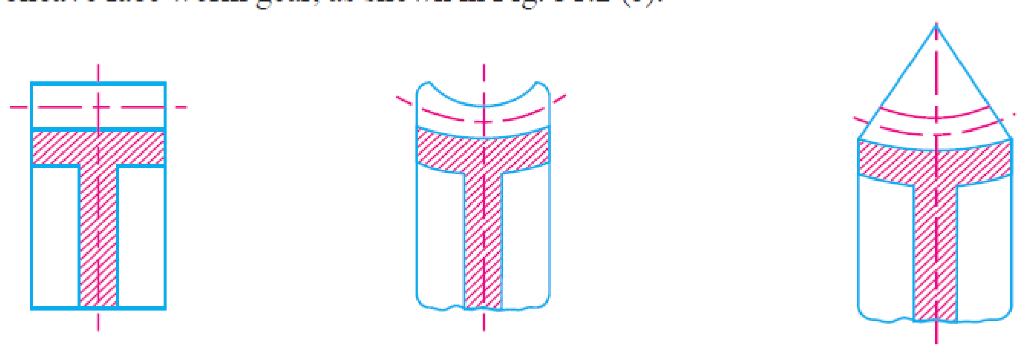
Single threaded. Double threaded.

(*a*) Cylindrical or straight worm. (*b*) Cone or double enveloping worm.

Types of Worm Gears

The following three types of worm gears are important from the subject point of view :

1. Straight face worm gear, as shown in Fig. 31.2 (*a*),
2. Hobbed straight face worm gear, as shown in Fig. 31.2 (*b*), and
3. Concave face worm gear, as shown in Fig. 31.2 (*c*).



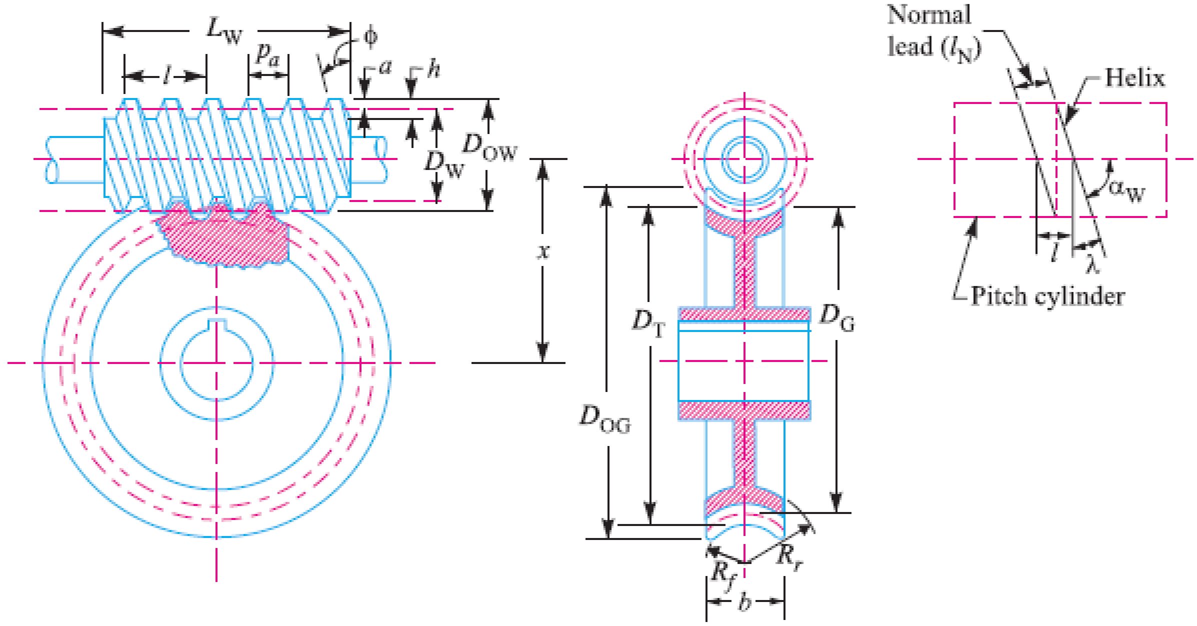
(*a*) Straight face. (*b*) Hobbed straight face. (*c*) Concave face.

Terms used in Worm Gearing

The worm and worm gear in mesh is shown in Fig. 31.3.

The following terms, in connection with the worm gearing, are important from the subject point of view :

1. *Axial pitch.* It is also known as ***linear pitch*** of a worm. It is the distance measured axially (*i.e.* parallel to the axis of worm) from a point on one thread to the corresponding point on the adjacent thread on the worm, as shown in Fig. 31.3. It may be noted that the axial pitch (p_a) of a worm is equal to the circular pitch (p_c) of the mating worm gear, when the shafts are at right angles.



2. *Lead.* It is the linear distance through which a point on a thread moves ahead in one revolution of the worm. For single start threads, lead is equal to the axial pitch, but for multiple start threads, lead is equal to the product of axial pitch and number of starts. Mathematically,

$$\text{Lead, } \quad l = p_a \,.\, n$$

where p_a = Axial pitch ; and n = Number of starts.

3. *Lead angle.* It is the angle between the tangent to the thread helix on the pitch cylinder and the plane normal to the axis of the worm. It is denoted by λ.

A little consideration will show that if one complete turn of a worm thread be imagined to be unwound from the body of the worm, it will form an inclined plane whose base is equal to the pitch circumference of the worm and altitude equal to lead of the worm, as shown in Fig. 31.4.

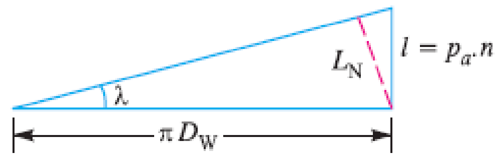


Fig. 31.4. Development of a helix thread.

From the geometry of the figure, we find that

$$\tan \lambda = \frac{\text{Lead of the worm}}{\text{Pitch circumference of the worm}}$$

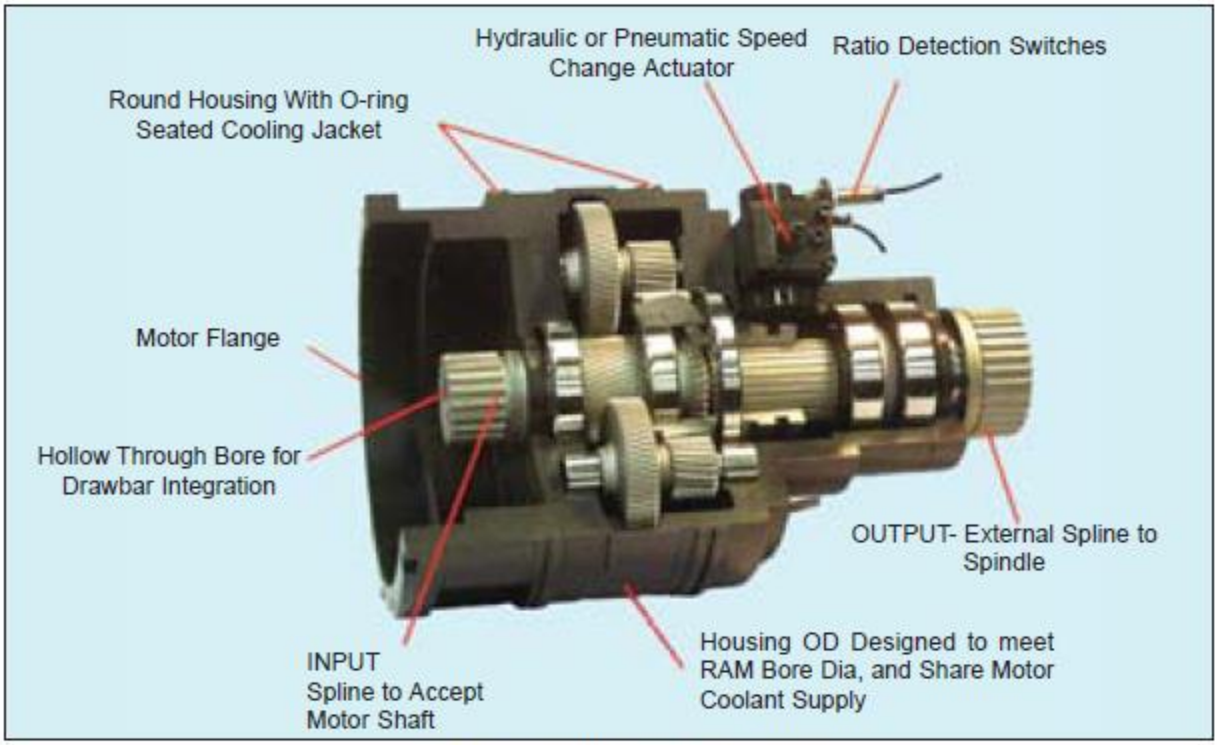
$$= \frac{l}{\pi\, D_W} = \frac{p_a \,.n}{\pi\, D_W} \qquad \ldots (\because\ l = p_a \,.\, n)$$

$$= \frac{p_c \,.n}{\pi\, D_W} = \frac{\pi\, m \,.n}{\pi\, D_W} = \frac{m\,.n}{D_W} \qquad \ldots (\because\ p_a = p_c\ ;\ \text{and}\ p_c = \pi\, m)$$

where m = Module, and

D_W = Pitch circle diameter of worm.

The lead angle (λ) may vary from 9° to 45°. It has been shown by F.A. Halsey that a lead angle less than 9° results in rapid wear and the safe value of λ is 12½°.



Model of sun and planet gears.

For a compact design, the lead angle may be determined by the following relation, *i.e.*

$$\tan \lambda = \left(\frac{N_G}{N_W} \right)^{1/3}$$

where N_G is the speed of the worm gear and N_W is the speed of the worm.

4. *Tooth pressure angle.* It is measured in a plane containing the axis of the worm and is equal to one-half the thread profile angle as shown in Fig. 31.3.

The following table shows the recommended values of lead angle (λ) and tooth pressure angle (φ).

Table 31.1. Recommended values of lead angle and pressure angle.

Lead angle (λ) in degrees	0 – 16	16 – 25	25 – 35	35 – 45
Pressure angle(φ) in degrees	14½	20	25	30

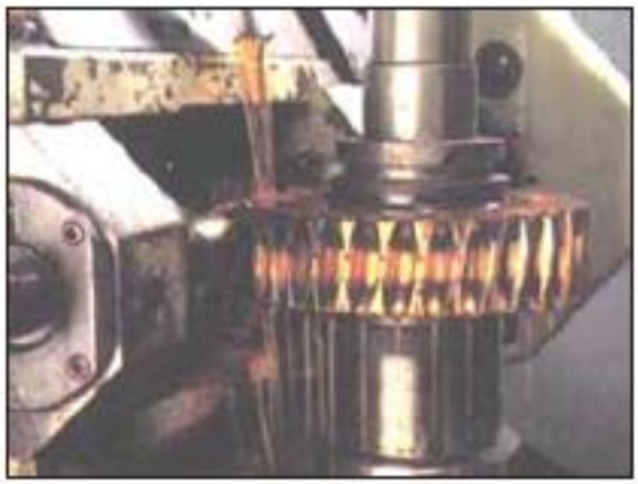
For automotive applications, the pressure angle of 30° is recommended to obtain a high efficiency and to permit overhauling.

5. *Normal pitch.* It is the distance measured along the normal to the threads between two corresponding points on two adjacent threads of the worm. Mathematically,

Normal pitch, $p_N = p_a . \cos \lambda$

Note. The term normal pitch is used for a worm having single start threads. In case of a worm having multiple start threads, the term normal lead (l_N) is used, such that

$$l_N = l \, . \cos \lambda$$



Worm gear teeth generation on gear hobbing machine.

6. *Helix angle.* It is the angle between the tangent to the thread helix on the pitch cylinder and the axis of the worm. It is denoted by α_W, in Fig. 31.3. The worm helix angle is the complement of worm lead angle, *i.e.*

$$\alpha_W + \lambda = 90°$$

It may be noted that the helix angle on the worm is generally quite large and that on the worm gear is very small. Thus, it is usual to specify the lead angle (λ) on the worm and helix angle (α_G) on the worm gear. These two angles are equal for a 90° shaft angle.

7. *Velocity ratio.* It is the ratio of the speed of worm (N_W) in r.p.m. to the speed of the worm gear (N_G) in r.p.m. Mathematically, velocity ratio,

$$V.R. = \frac{N_W}{N_G}$$

Let l = Lead of the worm, and

D_G = Pitch circle diameter of the worm gear.

We know that linear velocity of the worm,

$$v_W = \frac{l . N_W}{60}$$

and linear velocity of the worm gear,

$$v_G = \frac{\pi D_G N_G}{60}$$

Since the linear velocity of the worm and worm gear are equal, therefore

$$\frac{l . N_W}{60} = \frac{\pi D_G . N_G}{60} \quad \text{or} \quad \frac{N_W}{N_G} = \frac{\pi D_G}{l}$$

We know that pitch circle diameter of the worm gear,

$$D_G = m . T_G$$

where m is the module and T_G is the number of teeth on the worm gear.

$$\therefore \qquad V.R. = \frac{N_W}{N_G} = \frac{\pi D_G}{l} = \frac{\pi m . T_G}{l}$$

$$= \frac{p_c . T_G}{l} = \frac{p_a . T_G}{p_a . n} = \frac{T_G}{n} \qquad \text{... } (\because p_c = \pi m = p_a \text{; and } l = p_a . n)$$

where n = Number of starts of the worm.

From above, we see that velocity ratio may also be defined as the ratio of number of teeth on the worm gear to the number of starts of the worm.

The following table shows the number of starts to be used on the worm for the different velocity ratios :

Table 31.2. Number of starts to be used on the worm for different velocity ratios.

Velocity ratio (*V.R.*)	36 and above	12 to 36	8 to 12	6 to 12	4 to 10
Number of starts or threads on the worm ($n = T_w$)	Single	Double	Triple	Quadruple	Sextuple

31.5 Proportions for Worms

The following table shows the various porportions for worms in terms of the axial or circular pitch (p_c) in mm.

Table 31.3. Proportions for worm.

S. No.	*Particulars*	*Single and double threaded worms*	*Triple and quadruple threaded worms*
1.	Normal pressure angle (ϕ)	14½°	20°
2.	Pitch circle diameter for worms integral with the shaft	$2.35\ p_c + 10$ mm	$2.35\ p_c + 10$ mm
3.	Pitch circle diameter for worms bored to fit over the shaft	$2.4\ p_c + 28$ mm	$2.4\ p_c + 28$ mm
4.	Maximum bore for shaft	$p_c + 13.5$ mm	$p_c + 13.5$ mm
5.	Hub diameter	$1.66\ p_c + 25$ mm	$1.726\ p_c + 25$ mm
6.	Face length (L_W)	$p_c\ (4.5 + 0.02\ T_W)$	$p_c\ (4.5 + 0.02\ T_W)$
7.	Depth of tooth (h)	$0.686\ p_c$	$0.623\ p_c$
8.	Addendum (a)	$0.318\ p_c$	$0.286\ p_c$

Notes: 1. The pitch circle diameter of the worm (D_W) in terms of the centre distance between the shafts (x) may be taken as follows :

$$D_W = \frac{(x)^{0.875}}{\ldots} \qquad \text{... (when } x \text{ is in mm)}$$

2. The pitch circle diameter of the worm (D_W) may also be taken as

$$D_W = 3\,p_c, \text{ where } p_c \text{ is the axial or circular pitch.}$$

3. The face length (or length of the threaded portion) of the worm should be increased by 25 to 30 mm for the feed marks produced by the vibrating grinding wheel as it leaves the thread root.

31.6 Proportions for Worm Gear

The following table shows the various proportions for worm gears in terms of circular pitch (p_c) in mm.

Table 31.4. Proportions for worm gear.

S. No.	*Particulars*	*Single and double threads*	*Triple and quadruple threads*
1.	Normal pressure angle (ϕ)	14½°	20°
2.	Outside diameter (D_{OG})	$D_G + 1.0135\,p_c$	$D_G + 0.8903\,p_c$
3.	Throat diameter (D_T)	$D_G + 0.636\,p_c$	$D_G + 0.572\,p_c$
4.	Face width (b)	$2.38\,p_c + 6.5$ mm	$2.15\,p_c + 5$ mm
5.	Radius of gear face (R_f)	$0.882\,p_c + 14$ mm	$0.914\,p_c + 14$ mm
6.	Radius of gear rim (R_r)	$2.2\,p_c + 14$ mm	$2.1\,p_c + 14$ mm

31.7 Efficiency of Worm Gearing

The efficiency of worm gearing may be defined as the ratio of work done by the worm gear to the work done by the worm.

Mathematically, the efficiency of worm gearing is given by

$$\eta = \frac{\tan\lambda\,(\cos\phi - \mu\tan\lambda)}{\cos\phi\tan\lambda + \mu} \qquad \ldots(i)$$

where ϕ = Normal pressure angle,

μ = Coefficient of friction, and

λ = Lead angle.

The efficiency is maximum, when

$$\tan\lambda = \sqrt{1+\mu^2} - \mu$$

In order to find the approximate value of the efficiency, assuming square threads, the following relation may be used :

$$\text{Efficiency, } \eta = \frac{\tan\lambda\,(1-\mu\tan\lambda)}{\tan\lambda + \mu}$$

$$= \frac{1-\mu\tan\lambda}{1+\mu/\tan\lambda}$$

$$= \frac{\tan\lambda}{\tan(\lambda+\phi_1)}$$

...(Substituting in equation (*i*), $\phi = 0$, for square threads)

where ϕ_1 = Angle of friction, such that $\tan\phi_1 = \mu$.

A gear-cutting machine is used to cut gears.

The coefficient of friction varies with the speed, reaching a minimum value of 0.015 at a rubbing speed $\left(v_r = \dfrac{\pi D_W . N_W}{\cos \lambda} \right)$ between 100 and 165 m/min. For a speed below 10 m/min, take $\mu = 0.015$. The following empirical relations may be used to find the value of μ, *i.e.*

$$\mu = \frac{0.275}{(v_r)^{0.25}}, \text{ for rubbing speeds between 12 and 180 m/min}$$

$$= 0.025 + \frac{v_r}{18000} \quad \text{for rubbing speed more than 180 m/min}$$

Note : If the efficiency of worm gearing is less than 50%, then the worm gearing is said to be ***self locking***, *i.e.* it cannot be driven by applying a torque to the wheel. This property of self locking is desirable in some applications such as hoisting machinery.

Hardened and ground worm shaft and worm wheel pair

Example 31.1. *A triple threaded worm has teeth of 6 mm module and pitch circle diameter of 50 mm. If the worm gear has 30 teeth of 14½° and the coefficient of friction of the worm gearing is 0.05, find 1. the lead angle of the worm, 2. velocity ratio, 3. centre distance, and 4. efficiency of the worm gearing.*

Solution. Given : $n = 3$; $m = 6$; $D_W = 50$ mm ; $T_G = 30$; $\phi = 14.5°$; $\mu = 0.05$.

1. Lead angle of the worm

Let λ = Lead angle of the worm.

We know that $\tan \lambda = \dfrac{m.n}{D_W} = \dfrac{6 \times 3}{50} = 0.36$

$\therefore \quad \lambda = \tan^{-1} (0.36) = 19.8°$ **Ans.**

2. Velocity ratio

We know that velocity ratio,

$$V.R. = T_G / n = 30 / 3 = 10 \text{ Ans.}$$

3. Centre distance

We know that pitch circle diameter of the worm gear

$$D_G = m.T_G = 6 \times 30 = 180 \text{ mm}$$

$\therefore$ Centre distance,

$$x = \frac{D_W + D_G}{2} = \frac{50 + 180}{2} = 115 \text{ mm Ans.}$$

4. Efficiency of the worm gearing

We know that efficiency of the worm gearing,

$$\eta = \frac{\tan \lambda (\cos \phi - \mu \tan \lambda)}{\cos \phi . \tan \lambda + \mu}$$

$$= \frac{\tan 19.8° (\cos 14.5° - 0.05 \times \tan 19.8°)}{\cos 14.5° \times \tan 19.8° + 0.05}$$

Note : The approximate value of the efficiency assuming square threads is

$$\eta = \frac{1 - \mu \tan \lambda}{1 + \mu / \tan \lambda} = \frac{1 - 0.05 \times 0.36}{1 + 0.05/0.36} = \frac{0.982}{1.139} = 0.86 \text{ or } 86\% \quad \textbf{Ans.}$$

31.8 Strength of Worm Gear Teeth

In finding the tooth size and strength, it is safe to assume that the teeth of worm gear are always weaker than the threads of the worm. In worm gearing, two or more teeth are usually in contact, but due to uncertainty of load distribution among themselves it is assumed that the load is transmitted by one tooth only. We know that according to Lewis equation,

$$W_T = (\sigma_o \,.\, C_v)\, b \,.\, \pi\, m \,.\, y$$

where
W_T = Permissible tangential tooth load or beam strength of gear tooth,
σ_o = Allowable static stress,
C_v = Velocity factor,
b = Face width,
m = Module, and
y = Tooth form factor or Lewis factor.

Notes : 1. The velocity factor is given by

$$C_v = \frac{6}{6 + v}, \text{ where } v \text{ is the peripheral velocity of the worm gear in m/s.}$$

2. The tooth form factor or Lewis factor (y) may be obtained in the similar manner as discussed in spur gears (Art. 28.17), *i.e.*

$$y = 0.124 - \frac{0.684}{T_G}, \text{ for } 14\tfrac{1}{2}° \text{ involute teeth.}$$

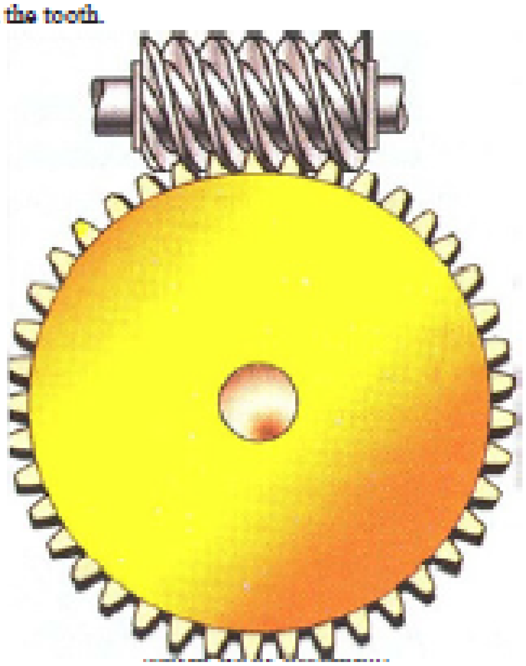
$$= 0.154 - \frac{0.912}{T_G}, \text{ for } 20° \text{ involute teeth.}$$

3. The dynamic tooth load on the worm gear is given by

$$W_D = \frac{W_T}{C_v} = W_T\left(\frac{6 + v}{6}\right)$$

where W_T = Actual tangential load on the tooth.

The dynamic load need not to be calculated because it is not so severe due to the sliding action between the worm and worm gear.



4. The static tooth load or endurance strength of the tooth (W_S) may also be obtained in the similar manner as discussed in spur gears (Art. 28.20), *i.e.*

$$W_S = \sigma_e . b\, \pi\, m . y$$

where σ_e = Flexural endurance limit. Its value may be taken as 84 MPa for cast iron and 168 MPa for phosphor bronze gears.

31.9 Wear Tooth Load for Worm Gear

The limiting or maximum load for wear (W_W) is given by

$$W_W = D_G \,.\, b \,.\, K$$

where D_G = Pitch circle diameter of the worm gear,

b = Face width of the worm gear, and

K = Load stress factor (also known as material combination factor).

The load stress factor depends upon the combination of materials used for the worm and worm gear. The following table shows the values of load stress factor for different combination of worm and worm gear materials.

Table 31.5. Values of load stress factor (*K*).

S.No.	*Material*		*Load stress factor (K)* N/mm^2
	Worm	*Worm gear*	
1.	Steel (B.H.N. 250)	Phosphor bronze	0.415
2.	Hardened steel	Cast iron	0.345
3.	Hardened steel	Phosphor bronze	0.550
4.	Hardened steel	Chilled phosphor bronze	0.830
5.	Hardened steel	Antimony bronze	0.830
6.	Cast iron	Phosphor bronze	1.035

Note : The value of K given in the above table are suitable for lead angles upto 10°. For lead angles between 10° and 25°, the values of K should be increased by 25 per cent and for lead angles greater than 25°, increase the value of K by 50 per cent.

31.10 Thermal Rating of Worm Gearing

In the worm gearing, the heat generated due to the work lost in friction must be dissipated in order to avoid over heating of the drive and lubricating oil. The quantity of heat generated (Q_g) is given by

$$Q_g = \text{Power lost in friction in watts} = P(1 - \eta) \qquad ...(i)$$

where P = Power transmitted in watts, and

η = Efficiency of the worm gearing.

The heat generated must be dissipated through the lubricating oil to the gear box housing and then to the atmosphere. The heat dissipating capacity depends upon the following factors :

1. Area of the housing (A),
2. Temperature difference between the housing surface and surrounding air ($t_2 - t_1$), and
3. Conductivity of the material (K).

Mathematically, the heat dissipating capacity,

$$Q_d = A(t_2 - t_1)K \qquad ...(ii)$$

From equations (*i*) and (*ii*), we can find the temperature difference ($t_2 - t_1$). The average value of K may be taken as 378 W/m²/°C.

Notes : 1. The maximum temperature ($t_2 - t_1$) should not exceed 27 to 38°C.

2. The maximum temperature of the lubricant should not exceed 60°C.

3. According to AGMA recommendations, the limiting input power of a plain worm gear unit from the standpoint of heat dissipation, for worm gear speeds upto 2000 r.p.m., may be checked from the following relation, *i.e.*

$$P = \frac{3650\, x^{1.7}}{V.R. + 5}$$

where P = Permissible input power in kW,

x = Centre distance in metres, and

31.11 Forces Acting on Worm Gears

When the worm gearing is transmitting power, the forces acting on the worm are similar to those on a power screw. Fig. 31.5 shows the forces acting on the worm. It may be noted that the forces on a worm gear are equal in magnitude to that of worm, but opposite in direction to those shown in Fig. 31.5.

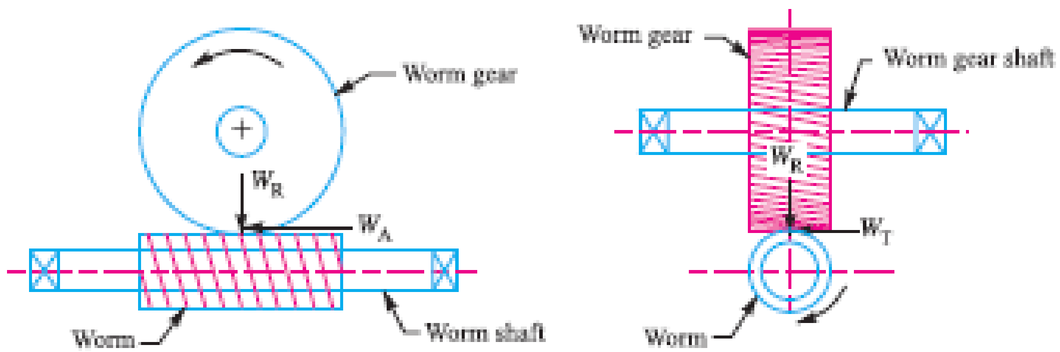


Fig. 31.5. Forces acting on worm teeth.

The various forces acting on the worm may be determined as follows :

1. Tangential force on the worm,

$$W_T = \frac{2 \times \text{Torque on worm}}{\text{Pitch circle diameter of worm } (D_W)}$$

$$= \text{Axial force or thrust on the worm gear}$$

The tangential force (W_T) on the worm produces a twisting moment of magnitude ($W_T \times D_W / 2$) and bends the worm in the horizontal plane.

2. Axial force or thrust on the worm,

$$W_A = W_T \,/\, \tan \lambda = \text{Tangential force on the worm gear}$$

$$= \frac{2 \times \text{Torque on the worm gear}}{\text{Pitch circle diameter of worm gear} (D_G)}$$

The axial force on the worm tends to move the worm axially, induces an axial load on the bearings and bends the worm in a vertical plane with a bending moment of magnitude ($W_A \times D_W / 2$).

3. Radial or separating force on the worm,

$$W_R = W_A \,.\, \tan \phi = \text{Radial or separating force on the worm gear}$$

The radial or separating force tends to force the worm and worm gear out of mesh. This force also bends the worm in the vertical plane.

Example 31.2. *A worm drive transmits 15 kW at 2000 r.p.m. to a machine carriage at 75 r.p.m. The worm is triple threaded and has 65 mm pitch diameter. The worm gear has 90 teeth of 6 mm module. The tooth form is to be 20° full depth involute. The coefficient of friction between the mating teeth may be taken as 0.10. Calculate : 1. tangential force acting on the worm ; 2. axial thrust and separating force on worm; and 3. efficiency of the worm drive.*

Solution. Given : $P = 15$ kW $= 15 \times 10^3$ W ; $N_W = 2000$ r.p.m. ; $N_G = 75$ r.p.m. ; $n = 3$; $D_W = 65$ mm ; $T_G = 90$; $m = 6$ mm ; $\phi = 20°$; $\mu = 0.10$

1. *Tangential force acting on the worm*

We know that the torque transmitted by the worm

$$= \frac{P \times 60}{2 \pi N_W} = \frac{15 \times 10^3 \times 60}{2 \pi \times 2000} = 71.6 \text{ N-m} = 71\,600 \text{ N-mm}$$

The centre distance may be expressed in terms of the axial lead (l), lead angle (λ) and velocity ratio ($V.R.$), as follows :

$$x = \frac{l}{2\pi}(\cot\lambda + V.R.)$$

In terms of normal lead ($l_N = l\cos\lambda$), the above expression may be written as :

$$x = \frac{l_N}{2\pi}\left(\frac{1}{\sin\lambda} + \frac{V.R.}{\cos\lambda}\right)$$

or
$$\frac{x}{l_N} = \frac{1}{2\pi}\left(\frac{1}{\sin\lambda} + \frac{V.R.}{\cos\lambda}\right) \qquad ...(i)$$

Since the velocity ratio ($V.R.$) is usually given, therefore the equation (*i*) contains three variables *i.e.* x, l_N and λ. The right hand side of the above expression may be calculated for various values of velocity ratios and the curves are plotted as shown in Fig. 31.7. The lowest point on each of the curves gives the lead angle which corresponds to the minimum value of x / l_N. This minimum value represents the minimum centre distance that can be used with a given lead or inversely the maximum lead that can be used with a given centre distance. Now by using Table 31.2 and standard modules, we can determine the combination of lead angle, lead, centre distance and diameters for the given design specifications.

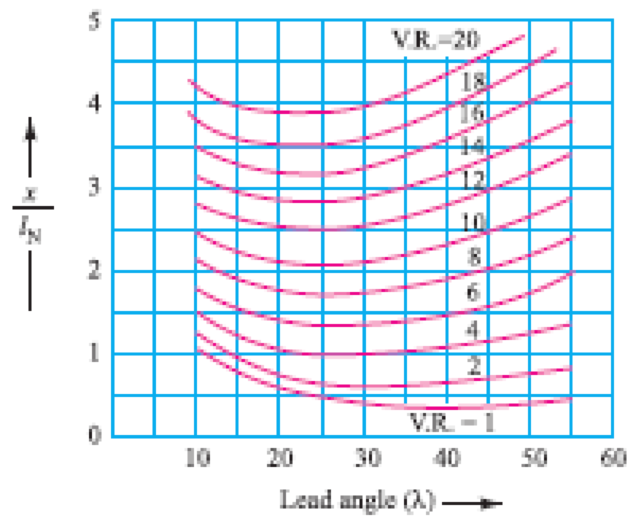


Fig. 31.7. Worm gear design curves.

Note : The lowest point on the curve may be determined mathematically by differentiating the equation (*i*) with respect to λ and equating to zero, *i.e.*

$$\frac{(V.R.)\sin^3\lambda - \cos^3\lambda}{\sin^2\lambda\cos^2\lambda} = 0 \quad \text{or} \quad V.R. = \cot^3\lambda$$

Example 31.3. *Design 20° involute worm and gear to transmit 10 kW with worm rotating at 1400 r.p.m. and to obtain a speed reduction of 12 : 1. The distance between the shafts is 225 mm.*

Solution. Given : $\phi = 20°$; $P = 10$ kW $= 10\,000$ W ; $N_W = 1400$ r.p.m. ; $V.R. = 12$; $x = 225$ mm

The worm and gear is designed as discussed below :

1. Design of worm

Let $\quad l_N$ = Normal lead, and

λ = Lead angle.

Worm gear of a steering mechanism in an automobile.

We have discussed in Art. 31.12 that the value of x / l_N will be minimum corresponding to

$$\cot^3 \lambda = V.R. = 12 \quad \text{or} \quad \cot \lambda = 2.29$$

$$\therefore \qquad \lambda = 23.6°$$

We know that
$$\frac{x}{l_N} = \frac{1}{2\pi}\left(\frac{1}{\sin \lambda} + \frac{V.R.}{\cos \lambda}\right)$$

$$\frac{225}{l_N} = \frac{1}{2\pi}\left(\frac{1}{\sin 23.6°} + \frac{12}{\cos 23.6°}\right) = \frac{1}{2\pi}(2.5 + 13.1) = 25$$

$$\therefore \qquad l_N = 225 / 2.5 = 90 \text{ mm}$$

and axial lead, $l = l_N / \cos \lambda = 90 / \cos 23.6° = 98.2$ mm

From Table 31.2, we find that for a velocity ratio of 12, the number of starts or threads on the worm,

$$n = T_W = 4$$

$\therefore$ Axial pitch of the threads on the worm,

$$p_a = l / 4 = 98.2 / 4 = 24.55 \text{ mm}$$

$$\therefore \qquad m = p_a / \pi = 24.55 / \pi = 7.8 \text{ mm}$$

Let us take the standard value of module, $m = 8$ mm

$\therefore$ Axial pitch of the threads on the worm,

$$p_a = \pi \, m = p \times 8 = 25.136 \text{ mm} \quad \textbf{Ans.}$$

Axial lead of the threads on the worm,

$$l = p_a \, . \, n = 25.136 \times 4 = 100.544 \text{ mm} \quad \textbf{Ans.}$$

and normal lead of the threads on the worm,

$$l_N = l \cos \lambda = 100.544 \cos 23.6° = 92 \text{ mm} \quad \textbf{Ans.}$$

We know that the centre distance,

$$x = \frac{l_N}{2\pi}\left(\frac{1}{\sin \lambda} + \frac{V.R.}{\cos \lambda}\right) = \frac{92}{2\pi}\left(\frac{1}{\sin 23.6°} + \frac{12}{\cos 23.6°}\right)$$

$$= 14.64 (2.5 + 13.1) = 230 \text{ mm} \quad \textbf{Ans.}$$

Let D_W = Pitch circle diameter of the worm.

We know that
$$\tan \lambda = \frac{l}{\pi D_W}$$

$$\therefore \qquad D_W = \frac{l}{\pi \tan \lambda} = \frac{100.544}{\pi \tan 23.6°} = 73.24 \text{ mm} \quad \textbf{Ans.}$$

Since the velocity ratio is 12 and the worm has quadruple threads (*i.e.* $n = T_W = 4$), therefore number of teeth on the worm gear,

$$T_G = 12 \times 4 = 48$$

From Table 31.3, we find that the face length of the worm or the length of threaded portion is

$$L_W = p_c (4.5 + 0.02 \, T_W)$$
$$= 25.136 (4.5 + 0.02 \times 4) = 115 \text{ mm} \quad ...(\because p_c = p_a)$$

This length should be increased by 25 to 30 mm for the feed marks produced by the vibrating grinding wheel as it leaves the thread root. Therefore let us take

$$L_W = 140 \text{ mm} \quad \textbf{Ans.}$$

We know that depth of tooth,

$$h = 0.623 \, p_c = 0.623 \times 25.136 = 15.66 \text{ mm} \quad \textbf{Ans.}$$

...(From Table 31.3)

and addendum, $a = 0.286 \, p_c = 0.286 \times 25.136 = 7.2$ mm **Ans.**

$\therefore$ Outside diameter of worm,

$$D_{OW} = D_W + 2a = 73.24 + 2 \times 7.2 = 87.64 \text{ mm} \quad \textbf{Ans.}$$

2. Design of worm gear

We know that pitch circle diameter of the worm gear,

$$D_G = m \, . \, T_G = 8 \times 48 = 384 \text{ mm} = 0.384 \text{ m} \quad \textbf{Ans.}$$

From Table 31.4, we find that outside diameter of worm gear,

$$D_{OG} = D_G + 0.8903 \, p_c = 384 + 0.8903 \times 25.136 = 406.4 \text{ mm} \quad \textbf{Ans.}$$

Throat diameter,

$$D_T = D_G + 0.572 \, p_c = 384 + 0.572 \times 25.136 = 398.4 \text{ mm} \quad \textbf{Ans.}$$

and face width, $b = 2.15 \, p_c + 5 \text{ mm} = 2.15 \times 25.136 + 5 = 59$ mm **Ans.**

Let us now check the designed worm gearing from the standpoint of tangential load, dynamic load, static load or endurance strength, wear load and heat dissipation.

(a) Check for the tangential load

Let N_G = Speed of the worm gear in r.p.m.

We know that velocity ratio of the drive,

$$V.R. = \frac{N_W}{N_G} \quad \text{or} \quad N_G = \frac{N_W}{V.R.} = \frac{1400}{12} = 116.7 \text{ r.p.m}$$

$\therefore$ Torque transmitted,

$$T = \frac{P \times 60}{2 \, \pi \, N_G} = \frac{10\,000 \times 60}{2 \, \pi \times 116.7} = 818.2 \text{ N-m}$$

and tangential load acting on the gear,

$$W_T = \frac{2 \times \text{Torque}}{D_G} = \frac{2 \times 818.2}{0.384} = 4260 \text{ N}$$

We know that pitch line or peripheral velocity of the worm gear,

$$v = \frac{\pi \, . \, D_G \, . \, N_G}{60} = \frac{\pi \times 0.384 \times 116.7}{60} = 2.35 \text{ m/s}$$

$\therefore$ Velocity factor,

$$C_v = \frac{6}{6 + v} = \frac{6}{6 + 2.35} = 0.72$$

Gears are usually enclosed in boxes to protect them from environmental pollution and provide them proper lubrication.

and tooth form factor for 20° involute teeth,

$$y = 0.154 - \frac{0.912}{T_G} = 0.154 - \frac{0.912}{48} = 0.135$$

Since the worm gear is generally made of phosphor bronze, therefore taking the allowable static stress for phosphor bronze, $\sigma_o = 84$ MPa or N/mm².

We know that the designed tangential load,

$$W_T = (\sigma_o \cdot C_v)\, b \cdot \pi\, m \cdot y = (84 \times 0.72)\, 59 \times \pi \times 8 \times 0.135 \text{ N}$$
$$= 12\ 110 \text{ N}$$

Since this is more than the tangential load acting on the gear (*i.e.* 4260 N), therefore the design is safe from the standpoint of tangential load.

(b) Check for dynamic load

We know that the dynamic load,

$$W_D = W_T / v = 12\ 110 / 0.72 = 16\ 820 \text{ N}$$

Since this is more than $W_T = 4260$ N, therefore the design is safe from the standpoint of dynamic load.

(c) Check for static load or endurance strength

We know that the flexural endurance limit for phosphor bronze is

$$\sigma_e = 168 \text{ MPa or N/mm}^2$$

∴ Static load or endurance strength,

$$W_S = \sigma_e \cdot b \cdot \pi\, m \cdot y = 168 \times 59 \times \pi \times 8 \times 0.135 = 33\ 635 \text{ N}$$

Since this is much more than $W_T = 4260$ N, therefore the design is safe from the standpoint of static load or endurance strength.

(d) Check for wear

Assuming the material for worm as hardened steel, therefore from Table 31.5, we find that for hardened steel worm and phosphor bronze worm gear, the value of load stress factor,

$$K = 0.55 \text{ N/mm}^2$$

$\therefore$ Limiting or maximum load for wear,

$$W_W = D_G \cdot b \cdot K = 384 \times 59 \times 0.55 = 12\,461 \text{ N}$$

Since this is more than $W_T = 4260$ N, therefore the design is safe from the standpoint of wear.

(e) Check for heat dissipation

First of all, let us find the efficiency of the worm gearing (η).

We know that rubbing velocity,

$$v_r = \frac{\pi D_W \cdot N_W}{\cos \lambda} = \frac{\pi \times 0.073\,24 \times 1400}{\cos 23.6°} = 351.6 \text{ m / min}$$

...(D_W is taken in metres)

$\therefore$ Coefficient of friction,

$$\mu = 0.025 + \frac{v_r}{18\,000} = 0.025 + \frac{351.6}{18\,000} = 0.0445$$

...($\because$ v_r is greater than 180 m/min)

and angle of friction, $\phi_1 = \tan^{-1} \mu = \tan^{-1} (0.0445) = 2.548°$

We know that efficiency,

$$\eta = \frac{\tan \lambda}{\tan (\lambda + \phi_1)} = \frac{\tan 23.6°}{\tan (23.6 + 2.548)} = \frac{0.4369}{0.4909} = 0.89 \text{ or } 89\%$$

Assuming 25 per cent overload, heat generated,

$$Q_g = 1.25\, P\, (1 - \eta) = 1.25 \times 10\,000\, (1 - 0.89) = 1375 \text{ W}$$

We know that projected area of the worm,

$$A_W = \frac{\pi}{4} (D_W)^2 = \frac{\pi}{4} (73.24)^2 = 4214 \text{ mm}^2$$

and projected area of the worm gear,

$$A_G = \frac{\pi}{4} (D_G)^2 = \frac{\pi}{4} (384)^2 = 115\,827 \text{ mm}^2$$

$\therefore$ Total projected area of worm and worm gear,

$$A = A_W + A_G = 4214 + 115\,827 = 120\,041 \text{ mm}^2$$
$$= 120\,041 \times 10^{-6} \text{ m}^2$$

We know that heat dissipating capacity,

$$Q_d = A\,(t_2 - t_1)\,K = 120\,041 \times 10^{-6}\,(t_2 - t_1)\,378 = 45.4\,(t_2 - t_1)$$

The heat generated must be dissipated in order to avoid over heating of the drive, therefore equating $Q_g = Q_d$, we have

$$t_2 - t_1 = 1375 / 45.4 = 30.3 \text{°C}$$

Since this temperature difference $(t_2 - t_1)$ is within safe limits of 27 to 38°C, therefore the design is safe from the standpoint of heat.

3. Design of worm shaft

Let d_W = Diameter of worm shaft.

We know that torque acting on the worm gear shaft,

$$T_{gear} = \frac{1.25\, P \times 60}{2\,\pi\, N_G} = \frac{1.25 \times 10\,000 \times 60}{2\,\pi \times 116.7} = 1023 \text{ N-m}$$
$$= 1023 \times 10^3 \text{ N-mm}$$

...(Taking 25% overload)

$\therefore$ Torque acting on the worm shaft,

$$T_{worm} = \frac{T_{gear}}{V.R. \times \eta} = \frac{1023}{12 \times 0.89} = 96 \text{ N-m} = 96 \times 10^3 \text{ N-mm}$$

Differential inside an automobile.

We know that tangential force on the worm,

$$W_T = \text{Axial force on the worm gear}$$
$$= \frac{2 \times T_{worm}}{D_W} = \frac{2 \times 96 \times 10^3}{73.24} = 2622 \text{ N}$$

Axial force on the worm,

$$W_A = \text{Tangential force on the worm gear}$$
$$= \frac{2 \times T_{gear}}{D_G} = \frac{2 \times 1023 \times 10^3}{384} = 5328 \text{ N}$$

and radial or separating force on the worm

$$W_R = \text{Radial or separating force on the worm gear}$$
$$= W_A \cdot \tan \phi = 5328 \times \tan 20° = 1940 \text{ N}$$

Let us take the distance between the bearings of the worm shaft (x_1) equal to the diameter of the worm gear (D_G), *i.e.*

$$x_1 = D_G = 384 \text{ mm}$$

∴ Bending moment due to the radial force (W_R) in the vertical plane

$$= \frac{W_R \times x_1}{4} = \frac{1940 \times 384}{4} = 186\,240 \text{ N-mm}$$

and bending moment due to axial force (W_A) in the vertical plane

$$= \frac{W_A \times D_W}{4} = \frac{5328 \times 73.24}{4} = 97\,556 \text{ N-mm}$$

∴ Total bending moment in the vertical plane,

$$M_1 = 186\,240 + 97\,556 = 283\,796 \text{ N-mm}$$

We know that bending moment due to tangential force (W_T) in the horizontal plane,

$$M_2 = \frac{W_T \times D_G}{4} = \frac{2622 \times 384}{4} = 251\,712 \text{ N-mm}$$

∴ Resultant bending moment on the worm shaft,

$$M_{worm} = \sqrt{(M_1)^2 + (M_2)^2} = \sqrt{(283\,796)^2 + (251\,712)^2} = 379\,340 \text{ N-mm}$$

We know that equivalent twisting moment on the worm shaft,

$$T_{ew} = \sqrt{(T_{worm})^2 + (M_{worm})^2} = \sqrt{(96 \times 10^3)^2 + (379\ 340)^2} \text{ N-mm}$$
$$= 391\ 300 \text{ N-mm}$$

We also know that equivalent twisting moment (T_{ew}),

$$391\ 300 = \frac{\pi}{16} \times \tau\,(d_W)^3 = \frac{\pi}{16} \times 50\,(d_W)^3 = 9.82\,(d_W)^3$$

...(Taking τ = 50 MPa or N/mm²)

$\therefore$ $(d_W)^3 = 391\ 300 / 9.82 = 39\ 850$ or $d_W = 34.2$ say 35 mm **Ans.**

Let us now check the maximum shear stress induced.

We know that the actual shear stress,

$$\tau = \frac{16\,T_{ew}}{\pi\,(d_W)^3} = \frac{16 \times 391\ 300}{\pi\,(35)^3} = 46.5 \text{ N/mm}^2$$

and direct compressive stress on the shaft due to the axial force,

$$\sigma_c = \frac{W_A}{\frac{\pi}{4}\,(d_W)^2} = \frac{5328}{\frac{\pi}{4}\,(35)^2} = 5.54 \text{ N/mm}^2$$

$\therefore$ Maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\,\tau^2} = \frac{1}{2}\sqrt{(5.54)^2 + 4\,(46.5)^2} = 46.6 \text{ MPa}$$

Since the maximum shear stress induced is less than 50 MPa (assumed), therefore the design of worm shaft is satisfactory.

4. Design of worm gear shaft

Let d_G = Diameter of worm gear shaft.

We have calculated above that the axial force on the worm gear

$$= 2622 \text{ N}$$

Tangential force on the worm gear

$$= 5328 \text{ N}$$

and radial or separating force on the worm gear

$$= 1940 \text{ N}$$

We know that bending moment due to the axial force on the worm gear

$$= \frac{\text{Axial force} \times D_G}{4} = \frac{2622 \times 384}{4} = 251\ 712 \text{ N-mm}$$

The bending moment due to the axial force will be in the vertical plane.

Let us take the distance between the bearings of the worm gear shaft (x_2) as 250 mm.

$\therefore$ Bending moment due to the radial force on the worm gear

$$= \frac{\text{Radial force} \times x_2}{4} = \frac{1940 \times 250}{4} = 121\ 250 \text{ N-mm}$$

The bending moment due to the radial force will also be in the vertical plane.

$\therefore$ Total bending moment in the vertical plane

$$M_3 = 251\ 712 + 121\ 250 = 372962 \text{ N-mm}$$

We know that the bending moment due to the tangential force in the horizontal plane

$$M_4 = \frac{\text{Tangential force} \times x_2}{4} = \frac{5328 \times 250}{4} = 333\ 000 \text{ N-mm}$$

$\therefore$ Resultant bending moment on the worm gear shaft,

$$M_{gear} = \sqrt{(M_3)^2 + (M_4)^2} = \sqrt{(372\ 962)^2 + (333\ 000)^2} \text{ N-mm}$$
$$= 500 \times 10^3 \text{ N-mm}$$

We have already calculated that the torque acting on the worm gear shaft,

$$T_{gear} = 1023 \times 10^3 \text{ N-mm}$$

$\therefore$ Equivalent twisting moment on the worm gear shaft,

$$T_{eg} = \sqrt{(T_{gear})^2 + (M_{gear})^2} = \sqrt{(1023 \times 10^3)^2 + (500 \times 10^3)^2} \text{ N-mm}$$
$$= 1.14 \times 10^6 \text{ N-mm}$$

We know that equivalent twisting moment (T_{eg}),

$$1.14 \times 10^6 = \frac{\pi}{16} \times \tau\, (d_G)^3 = \frac{\pi}{16} \times 50\, (d_G)^3 = 9.82\, (d_G)^3$$

$\therefore$ $$(d_G)^3 = 1.14 \times 10^6 / 9.82 = 109 \times 10^3$$

or $$d_G = 48.8 \text{ say } 50 \text{ mm}$$ **Ans.**

Let us now check the maximum shear stress induced.

We know that actual shear stress,

$$\tau = \frac{16\, T_{eg}}{\pi\, (d_G)^3} = \frac{16 \times 1.14 \times 10^6}{\pi\, (50)^3} = 46.4 \text{ N/mm}^2 = 46.4 \text{ MPa}$$

and direct compressive stress on the shaft due to the axial force,

$$\sigma_c = \frac{\text{Axial force}}{\frac{\pi}{4} (d_G)^2} = \frac{2622}{\frac{\pi}{4} (50)^2} = 1.33 \text{ N/mm}^2 = 1.33 \text{ MPa}$$

$\therefore$ Maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\,\tau^2} = \frac{1}{2}\sqrt{(1.33)^2 + 4\,(46.4)^2} = 46.4 \text{ MPa}$$

Post Test

Example 31.4. *A speed reducer unit is to be designed for an input of 1.1 kW with a transmission ratio 27. The speed of the hardened steel worm is 1440 r.p.m. The worm wheel is to be made*

Key answer

Pre Test

Return to the text

Post test

Solution. Given : $P = 1.1$ kW $= 1100$ W ; $V.R. = 27$; $N_W = 1440$ r.p.m. ; $\phi = 20°$

A speed reducer unit (*i.e.*, worm and worm gear) may be designed as discussed below.

Since the centre distance between the shafts is not known, therefore let us assume that for this size unit, the centre distance (x) = 100 mm.

We know that pitch circle diameter of the worm,

$$D_W = \frac{(x)^{0.875}}{1.416} = \frac{(100)^{0.875}}{1.416} = 39.7 \text{ say } 40 \text{ mm}$$

∴ Pitch circle diameter of the worm gear,

$$D_G = 2x - D_W = 2 \times 100 - 40 = 160 \text{ mm}$$

From Table 31.2, we find that for the transmission ratio of 27, we shall use double start worms.

∴ Number of teeth on the worm gear,

$$T_G = 2 \times 27 = 54$$

We know that the axial pitch of the threads on the worm (p_a) is equal to circular pitch of teeth on the worm gear (p_c).

∴ $$p_a = p_c = \frac{\pi D_G}{T_G} = \frac{\pi \times 160}{54} = 9.3 \text{ mm}$$

and module, $$m = \frac{p_c}{\pi} = \frac{9.3}{\pi} = 2.963 \text{ say } 3 \text{ mm}$$

∴ Actual circular pitch,

$$p_c = \pi m = \pi \times 3 = 9.426 \text{ mm}$$

Actual pitch circle diameter of the worm gear,

$$D_G = \frac{p_c T_G}{\pi} = \frac{9.426 \times 54}{\pi} = 162 \text{ mm} \quad \textbf{Ans.}$$

and actual pitch circle diameter of the worm,

$$D_W = 2x - D_G = 2 \times 100 - 162 = 38 \text{ mm} \quad \textbf{Ans.}$$

The face width of the worm gear (b) may be taken as 0.73 times the pitch circle diameter of worm (D_W).

∴ $$b = 0.73\, D_W = 0.73 \times 38 = 27.7 \text{ say } 28 \text{ mm}$$

Let us now check the design from the standpoint of tangential load, dynamic load, static load or endurance strength, wear load and heat dissipation.

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1. *Check for the tangential load*

Let N_G = Speed of the worm gear in r.p.m.

We know that velocity ratio of the drive,

$$V.R. = \frac{N_W}{N_G} \text{ or } N_G = \frac{N_W}{V.R.} = \frac{1440}{27} = 53.3 \text{ r.p.m}$$

$\therefore$ Peripheral velocity of the worm gear,

$$v = \frac{\pi D_G \,.\, N_G}{60} = \frac{\pi \times 0.162 \times 53.3}{60} = 0.452 \text{ m/s}$$

... (D_G is taken in metres)

and velocity factor,

$$C_v = \frac{6}{6+v} = \frac{6}{6+0.452} = 0.93$$

We know that for 20° involute teeth, the tooth form factor,

$$y = 0.154 - \frac{0.912}{T_G} = 0.154 - \frac{0.912}{54} = 0.137$$

From Table 31.4, we find that allowable static stress for phosphor bronze is

$$\sigma_o = 84 \text{ MPa or N/mm}^2$$

$\therefore$ Tangential load transmitted,

$$W_T = (\sigma_o \,.\, C_v)\, b \,.\, \pi\, m \,.\, y = (84 \times 0.93)\ 28 \times \pi \times 3 \times 0.137 \text{ N}$$
$$= 2825 \text{ N}$$

and power transmitted due to the tangential load,

$$P = W_T \times v = 2825 \times 0.452 = 1277 \text{ W} = 1.277 \text{ kW}$$

Since this power is more than the given power to be transmitted (1.1 kW), therefore the design is safe from the standpoint of tangential load.

2. *Check for the dynamic load*

We know that the dynamic load,

$$W_D = W_T / C_v = 2825 / 0.93 = 3038 \text{ N}$$

and power transmitted due to the dynamic laod,

$$P = W_D \times v = 3038 \times 0.452 = 1373 \text{ W} = 1.373 \text{ kW}$$

Since this power is more than the given power to be transmitted, therefore the design is safe from the standpoint of dynamic load.

---

3. Check for the static load or endurance strength

From Table 31.8, we find that the flexural endurance limit for phosphor bronze is

$$\sigma_e = 168 \text{ MPa or N/mm}^2$$

∴ Static load or endurance strength,

$$W_S = \sigma_e \,.\, b \,.\, \pi\, m \,.\, y = 168 \times 28 \times \pi \times 3 \times 0.137 = 6075 \text{ N}$$

and power transmitted due to the static load,

$$P = W_S \times v = 6075 \times 0.452 = 2746 \text{ W} = 2.746 \text{ kW}$$

Since this power is more than the power to be transmitted (1.1 kW), therefore the design is safe from the standpoint of static load.

4. Check for the wear load

From Table 31.5, we find that the load stress factor for hardened steel worm and phosphor bronze worm gear is

$$K = 0.55 \text{ N/mm}^2$$

∴ Limiting or maximum load for wear,

$$W_W = D_G \,.\, b \,.\, K = 162 \times 28 \times 0.55 = 2495 \text{ N}$$

and power transmitted due to the wear load,

$$P = W_W \times v = 2495 \times 0.452 = 1128 \text{ W} = 1.128 \text{ kW}$$

Since this power is more than the given power to be transmitted (1.1 kW), therefore the design is safe from the standpoint of wear.

5. Check for the heat dissipation

We know that permissible input power,

$$P = \frac{3650\,(x)^{1.7}}{V.R + 5} = \frac{3650\,(0.1)^{1.7}}{27 + 5} = 2.27 \text{ kW} \quad \text{... (} x \text{ is taken in metres)}$$

Since this power is more than the given power to be transmitted (1.1 kW), therefore the design is safe from the standpoint of heat dissipation.

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Mechanical Department/ Production

By

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Overview

A *cam* is a rotating machine element which gives reciprocating or oscillating motion to another element known as *follower*. The cam and the follower have a line contact and constitute a higher pair. The cams are usually rotated at uniform speed by a shaft, but the follower motion is pre-determined and will be according to the shape of the cam. The cam and follower is one of the simplest as well as one of the most important mechanisms found in modern machinery today. The cams are widely used for operating the inlet and exhaust valves of internal combustion engines, automatic attachment of machineries, paper cutting machines, spinning and weaving textile machineries, feed mechanism of automatic lathes etc.

Objectives :-

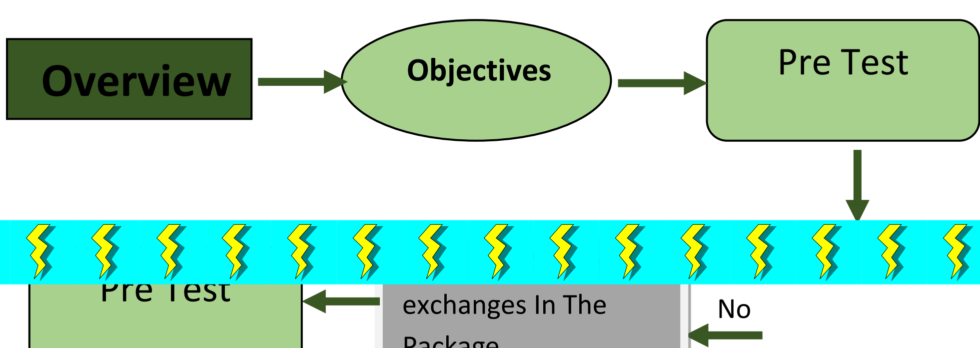
After studying the first modular unit , the student will be able to:-

1-

2-

3-

Flow Chart:-



IF
≥ 9

Next Package

Pre Test

a cam is to give the following motion to a knife-edged follower :

1.out stroke during 60^{o} of cam rotation

2.dwell for the next 30^{o} of cam rotation

3.return stroke during next 60^{o} of cam rotation, and

4.dwell for the remaining 210^{o} of cam rotation

The stroke of the follower is 40 mm and the minimum radius of the cam is 50 mm. the follower moves with uniform velocity during both the outstroke and return strokes. Draw the profile of the cam when(a) the axis of the follower passes through the axis of the cam shaft. (b) the axes of the follower is offset by 20 mm from the axis of cam shaft.

Notes

-10 degrees for the above question.

- Check your answers in key answer.

The Text

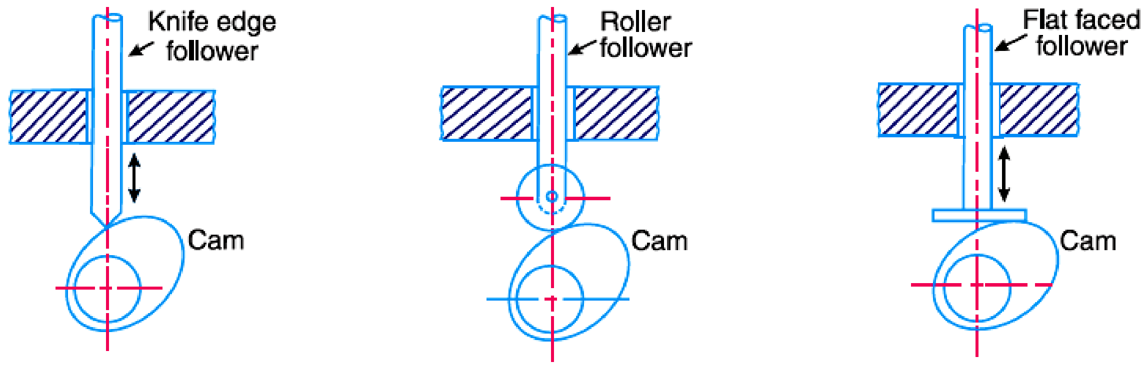
A cam is a rotating machine element which gives reciprocating or oscillating motion to another element known as follower .

the cams are usually rotated at uniform speed by a shaft , but the follower motion is predetermined and will be according to the shape of the cam .

the cams are widely used for operating the inlet and exhaust valves for internal combustion engines, automatic attachment of machineries, paper cutting machines, feed mechanism of automatic lathes etc..

classification of cams

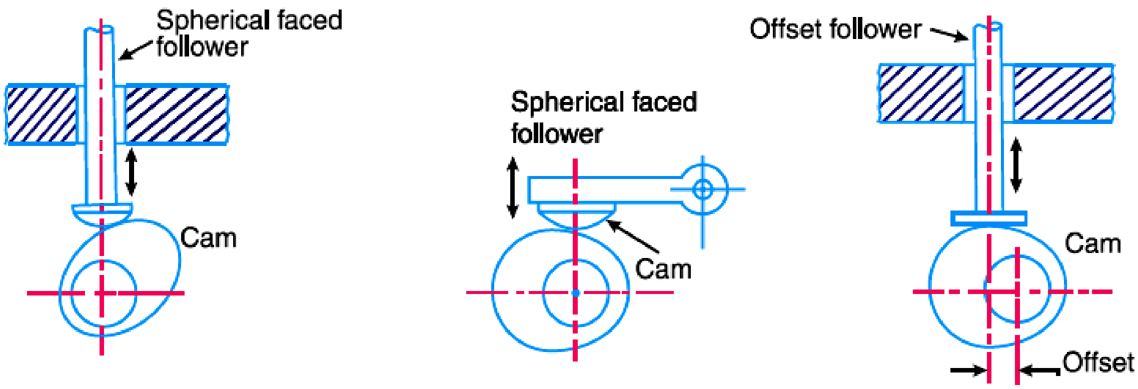
1.radial or disc cam



(*a*) Cam with knife edge follower.

(*b*) Cam with roller follower.

(*c*) Cam with flat faced follower.

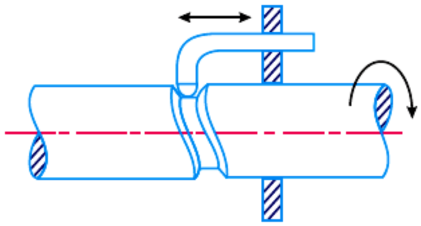


(*d*) Cam with spherical faced follower.

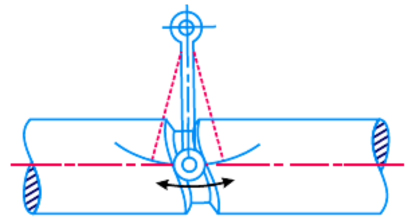
(*e*) Cam with spherical faced follower.

(*f*) Cam with offset follower.

2.cylindrical cam



(*a*) Cylindrical cam with reciprocating follower.



(*b*) Cylindrical cam with oscillating follower.

Cylindrical cam.

Classification of the followers

1.according to the surface in contact

a. knife edge follower

b. roller follower

c. flat face follower

d. spherical face follower

2.according to the motion of the follower

a. translating follower

b. oscillating follower

3.according to the path of motion of the follower

a. radial follower

b. off-set follower

terms used in radial cams

1. base circle: it’s the smallest circle that can be drawn to the cam profile.

2. trace point: it’s a reference point on the follower and it used to generate the pitch curve.

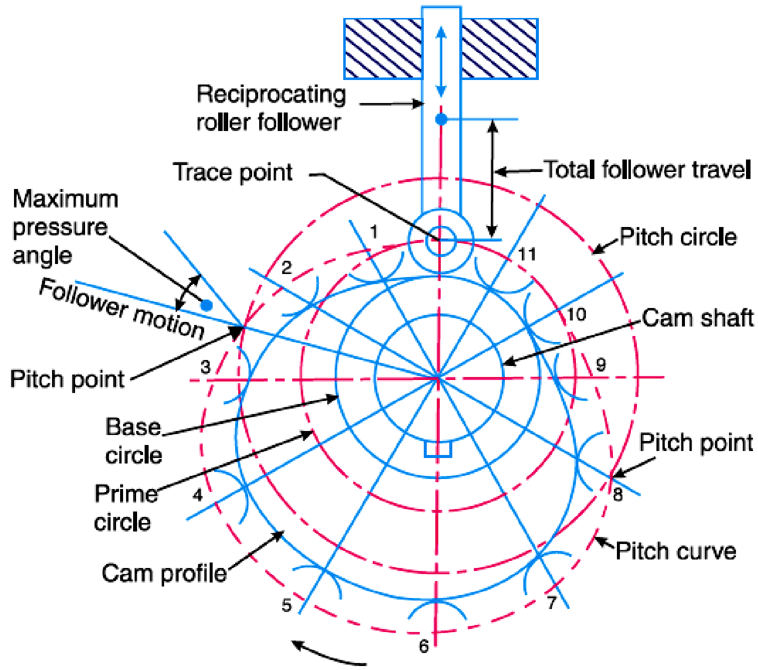
3. pitch curve : it’s the curve generated by trace point as the follower moves relative to the cam.

4. pressure angle : it’s the angle between the direction of the follower motion and a normal to the pitch curve.

5. pitch point :it’s a point on the pitch curve having the maximum pressure angle.

6. prime circle : it’s the smallest circle that can be drawn from the center of the cam and tangent to the pitch curve.

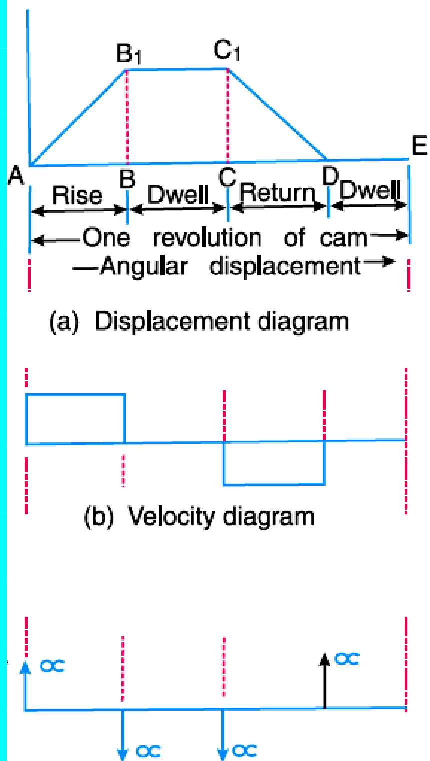
7. stroke : it’s the maximum travel of the follower from its lowest position to the topmost position.



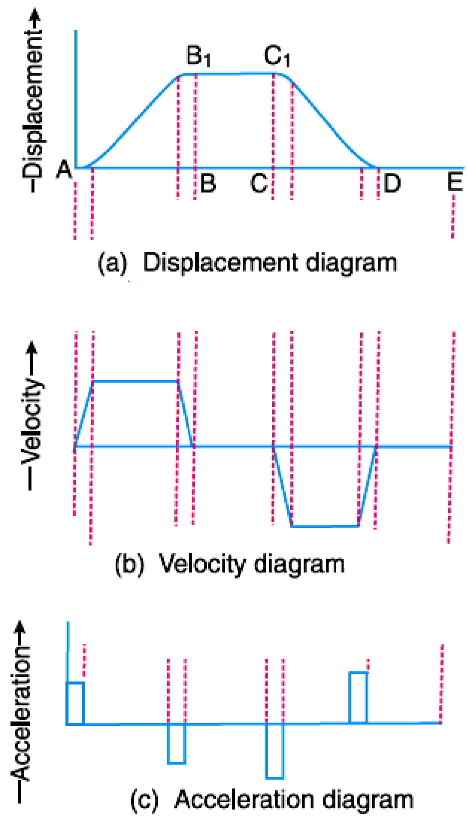
Motion of the follower

1. uniform velocity 2. simple harmonic motion 3. Uniform acceleration and retardation and 4. Cycloidal motion

Displacement, velocity and acceleration diagrams when the follower moves with uniform velocity



(a) Displacement diagram
(b) Velocity diagram
(c) Acceleration diagram



(a) Displacement diagram
(b) Velocity diagram
(c) Acceleration diagram

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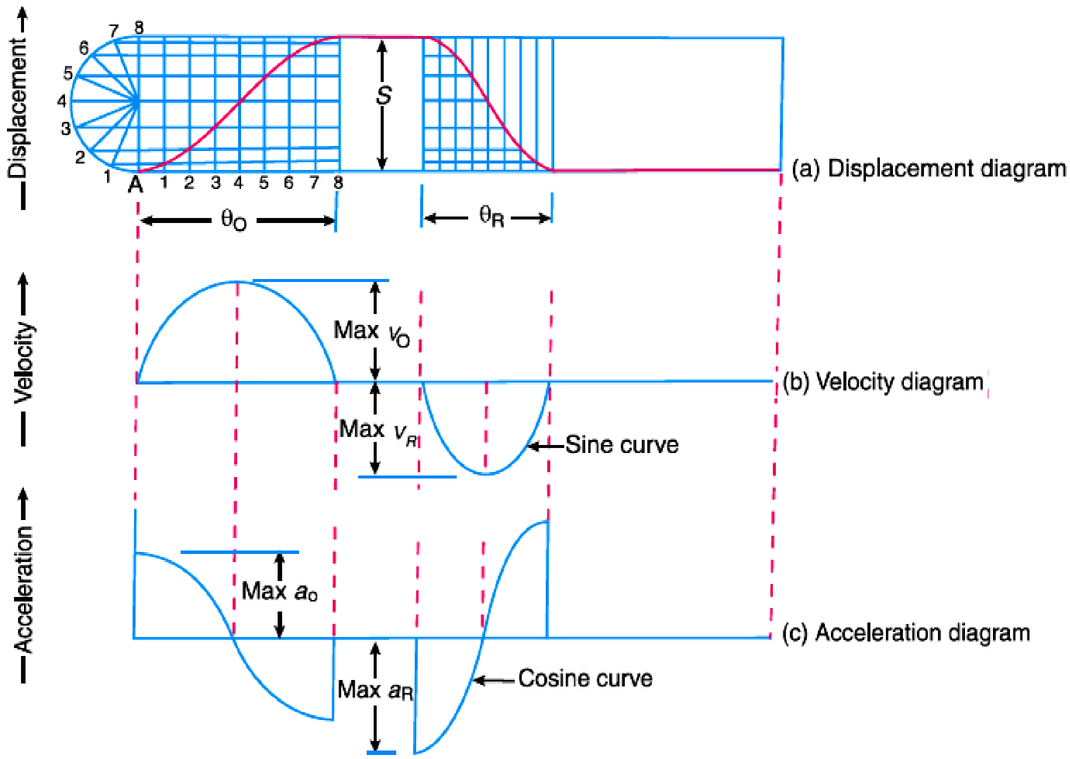
eration diagrams when the
wer moves with uniform velocity.

Modified displacement, velocity and acceleration diagrams when the follower moves with uniform velocity.

Displacement, velocity and acceleration diagrams when the follower moves with simple harmonic motion



Displacement, velocity and acceleration diagrams when the follower moves with simple harmonic motion.

Let S – stroke of the follower

$\theta_o - \theta_R$ - angular displacement of the cam during out stroke and return stroke of the follower in Radians

ω - angular velocity of the cam in *Rad/s*

t_o - time required for the out stroke of the follower in seconds

$$\omega = \frac{\theta_o}{t_o}$$

$$t_o = \frac{\theta_o}{\omega}$$

The point which defines the motion of the follower on the cam moves at uniform speed round the circumference of a circle of diameter *S* in time *2tₒ*

$$\therefore \quad v_o = \frac{\pi\, S}{2\, t_o} = \frac{\pi\, S}{2 \times \dfrac{\theta_o}{\omega}} = \frac{\pi\, S}{2} \times \frac{\omega}{\theta_o} = \frac{\pi\, \omega\, S}{2\theta_o}$$

$$v_o = v_p$$

Maximum acceleration of the follower on the outstroke

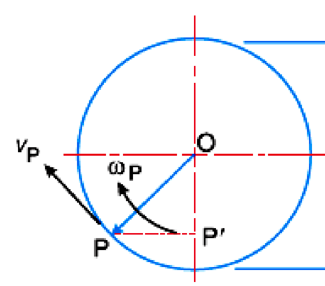
$$a_o = a_p = \frac{(v_p)^2}{OP} = \frac{\left(\dfrac{\pi\, \omega\, S}{2\, \theta_o}\right)^2}{\dfrac{S}{2}} = \frac{\pi^2 \omega^2\, S}{2\, \theta_o^{\,2}}$$

Similarly, maximum velocity of the follower on the return stroke

$$v_R = \frac{\pi\, \omega\, S}{2\theta_R}$$

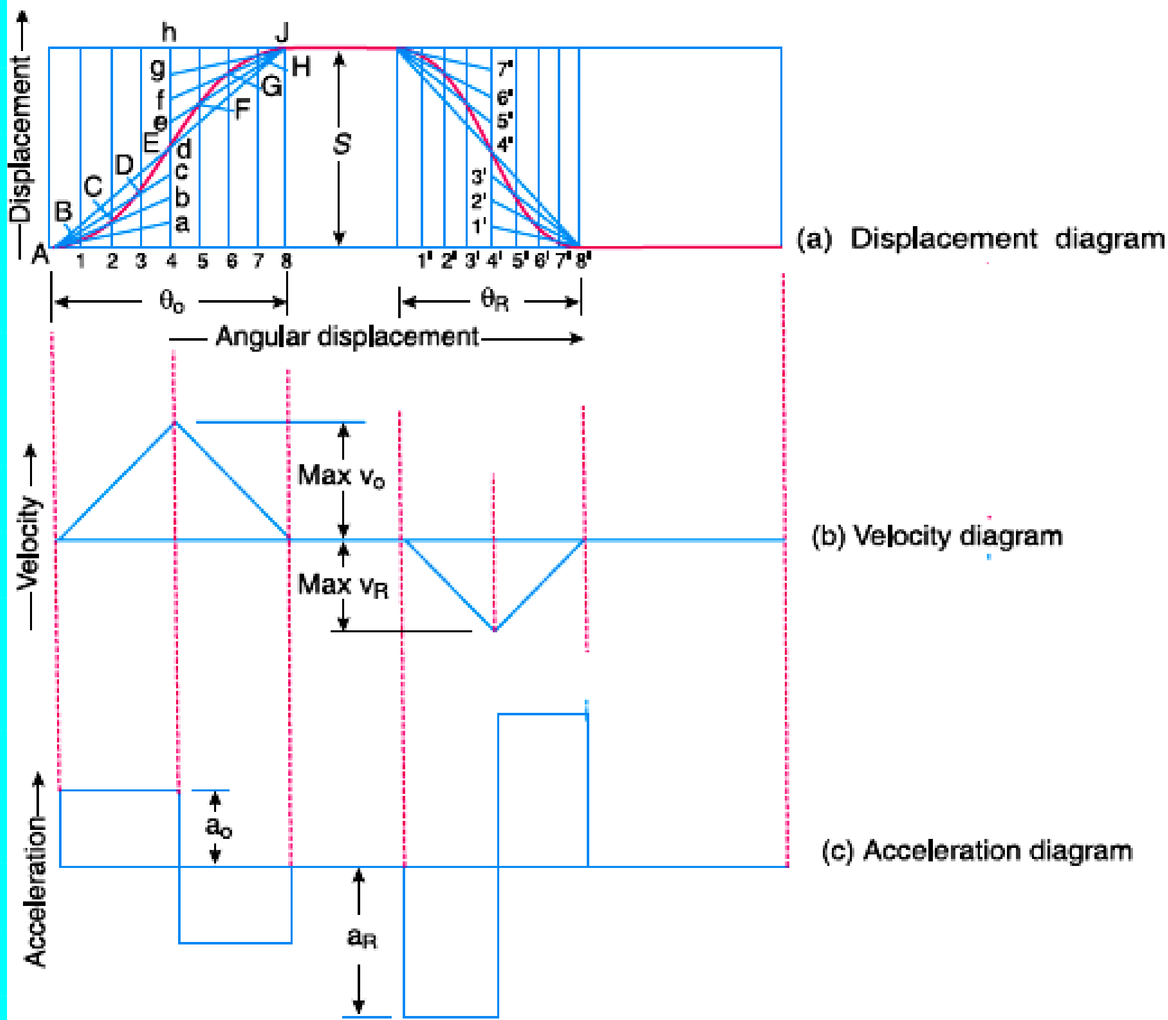
Maximum acceleration of the follower on the return stroke

$$a_R = \frac{\pi^2 \omega^2\, S}{2\, \theta_R^{\,2}}$$



Motion of a point.

Displacement, velocity and acceleration diagrams when the follower moves with uniform acceleration and retardation



isplacement, velocity and acceleration diagrams when the follower moves with uniform acceleration and retardation.

Mean velocity of the follower during out stroke $v_o = \frac{S}{t_o}$

Mean velocity of the follower during return stroke $v_R = \frac{S}{t_R}$

The maximum velocity of the follower is equal to twice mean velocity

$$v_o = \frac{2\ S}{t_o} = \frac{2\ \omega\ S}{\theta_o}$$

$$v_R = \frac{2\ \omega\ S}{\theta_R}$$

The maximum velocity of the follower is reached after the time $\frac{t_o}{2}$ during the out stroke and $\frac{t_R}{2}$ during the return stroke

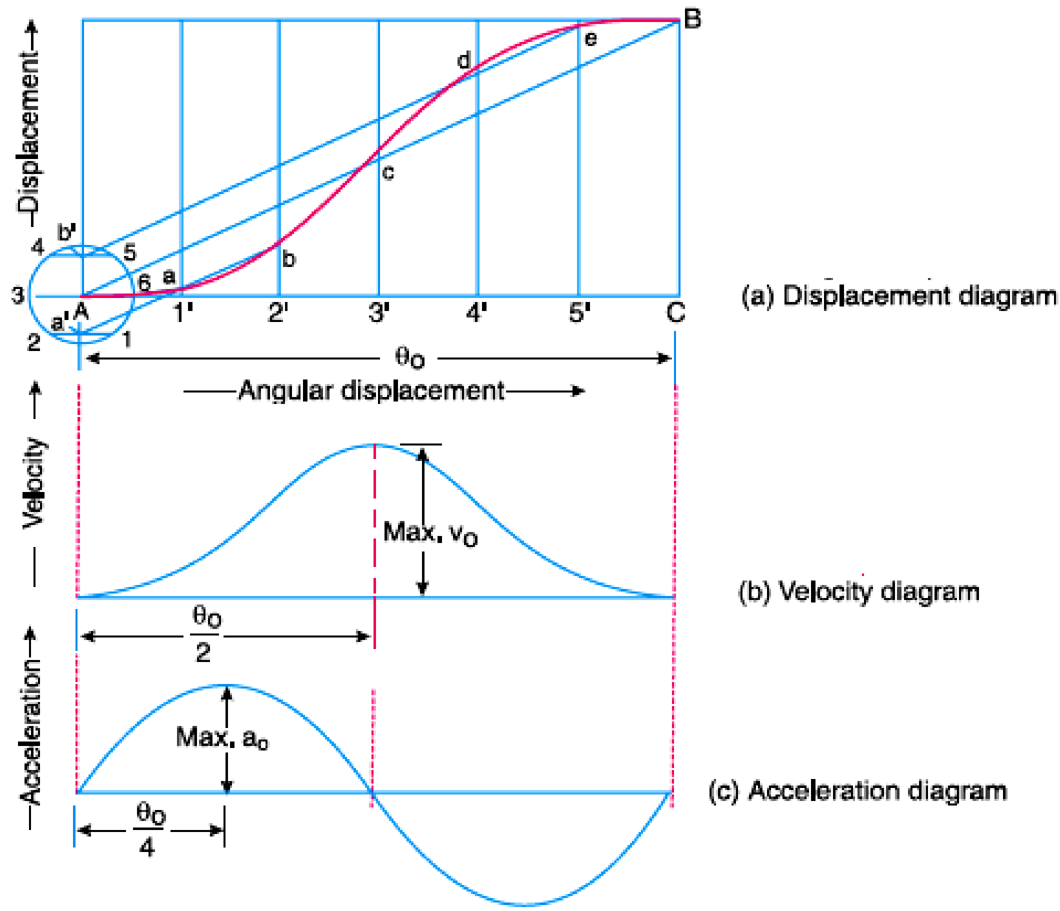
Maximum acceleration of the follower on the outstroke

$$a_o = \frac{v_o}{\frac{t_o}{2}} = \frac{2\ v_o}{t_o} = \frac{2 \times 2 \omega\ S}{t_o\ \theta_o} = \frac{4\ \omega^2\ S}{\theta_o{}^2}$$

Similarly, maximum acceleration of the follower on the return stroke

$$a_R = \frac{4\ \omega^2\ S}{\theta_R{}^2}$$

Displacement, velocity and acceleration diagrams when the follower moves with cycloidal motion



Displacement, velocity and acceleration diagrams when the follower moves with cycloidal motion.

Maximum velocity of the follower during out stroke is : $v_o = \dfrac{2\omega S}{\theta_o}$

Similarly, maximum velocity of the follower during return stroke is :

$$v_R = \frac{2\,\omega\,S}{\theta_R}$$

Maximum acceleration of the follower during out stroke is :

$$a_o = \frac{2\,\pi\,\omega^2\,S}{(\theta_o)^2}$$

Similarly, maximum acceleration of the follower during

return stroke is : $$a_R = \frac{2\,\pi\,\omega^2\ S}{(\theta_R)^2}$$

Post Test

A cam is to be designed for a knife edge follower with following data:

1.cam lift = 40 mm during 90^o of cam rotation with simple harmonic motion.

2.dwell for the next 30^o.

3.during the next 60^o of cam rotation, the follower return to its original position with simple harmonic motion.

4.dwell during the remaining 180^o .

Draw the profile of the cam when:

(a)the line of stroke of the follower passes through the axis of cam shaft, and

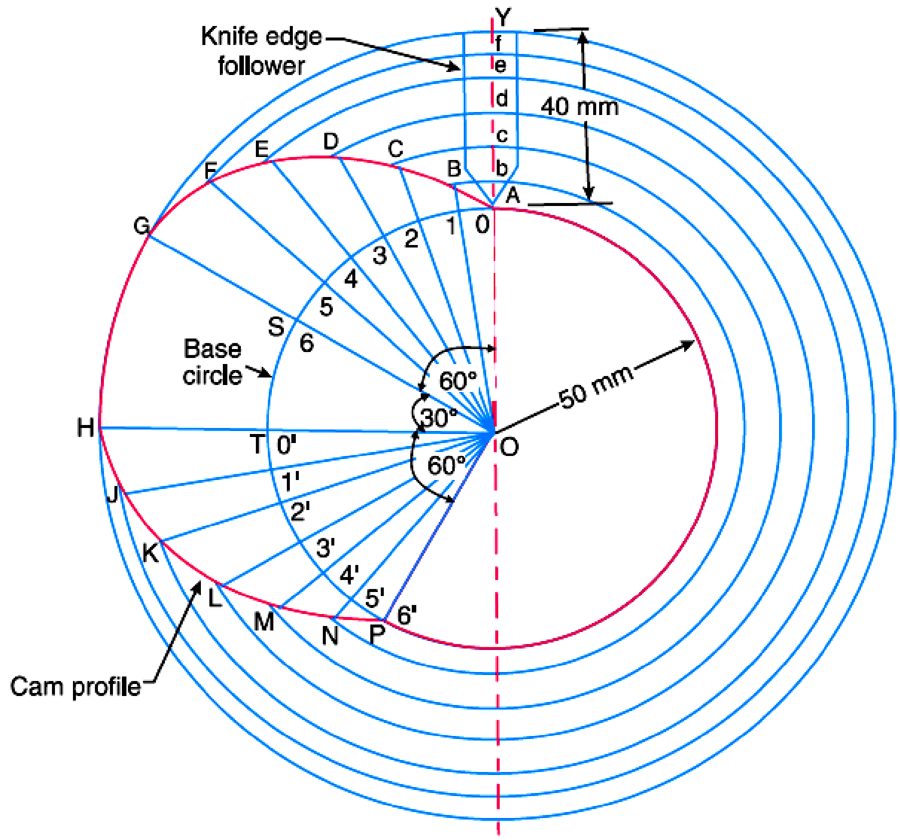
(b)the line of stroke is offset 20 mm from the axis of the cam shaft.

The radius of the base circle of the cam is 40 mm. determine the maximum velocity and acceleration of the follower during its ascent and descent, if the cam rotates at 240 R.P.M.

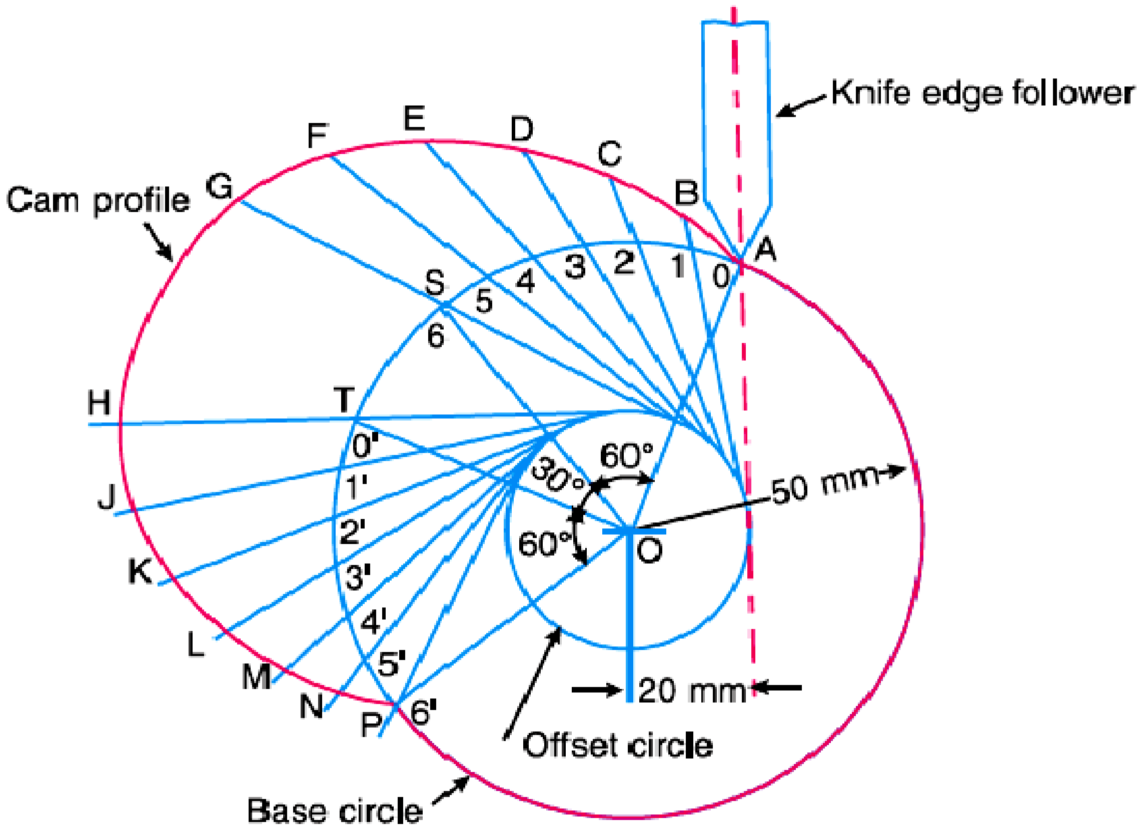
Key Answer

Pre Test

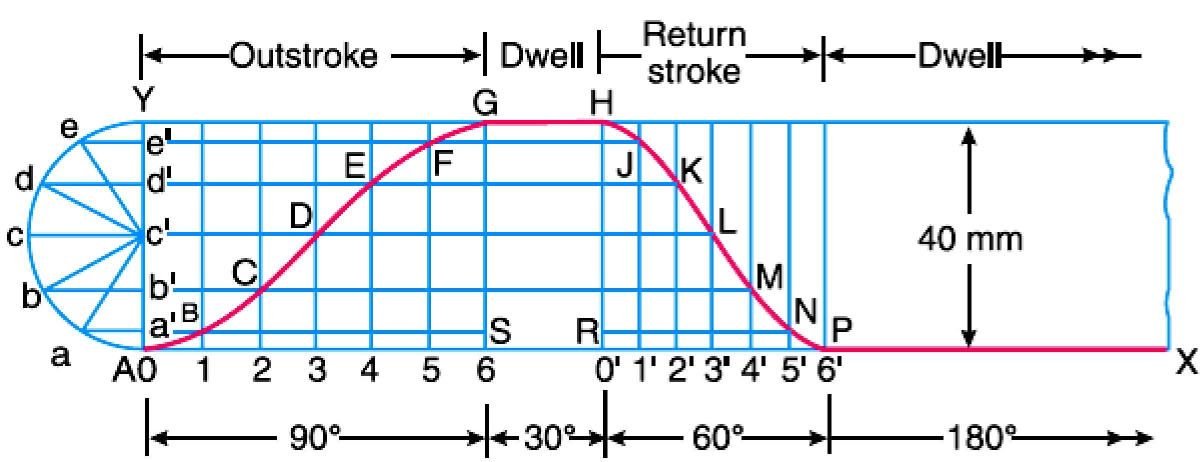
(a)profile of the cam when the axis of follower passes through the axis of cam shaft



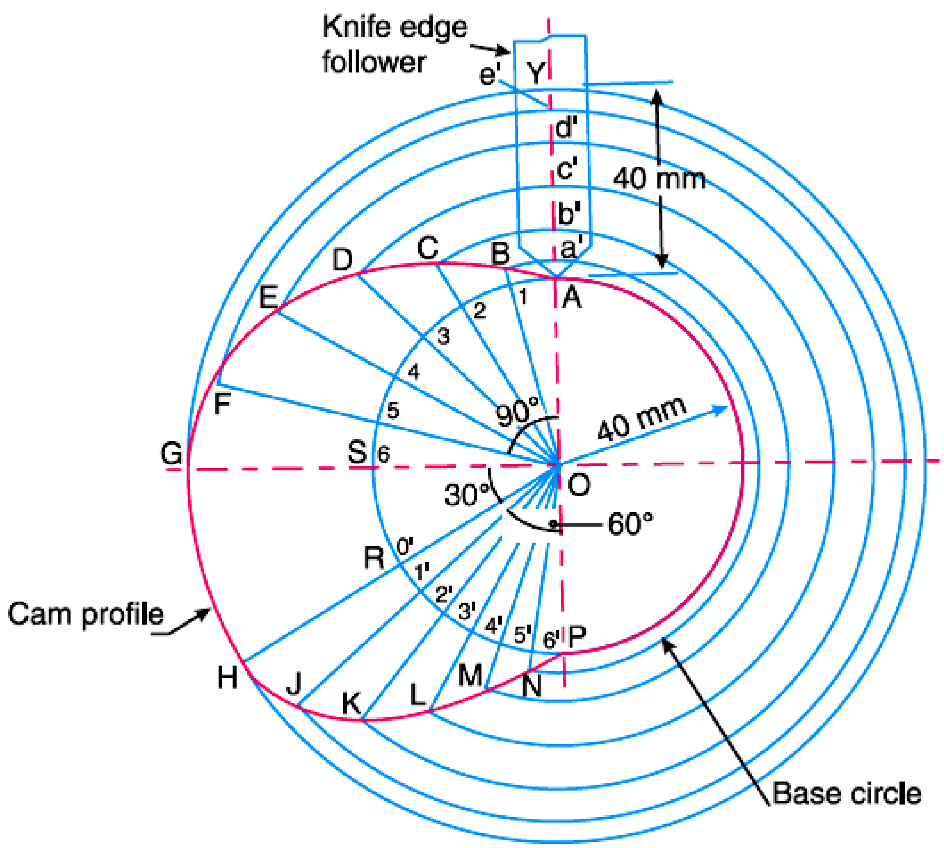
(b)profile of the cam when the axis of follower is offset by 20 mm from the axis of the cam shaft


Knife edge follower
Cam profile
A
B
C
D
E
F
G
H
J
K
L
M
N
P
S
T
0
1
2
3
4
5
6
0'
1'
2'
3'
4'
5'
6'
60°
30°
60°
O
50 mm
20 mm
Offset circle
Base circle

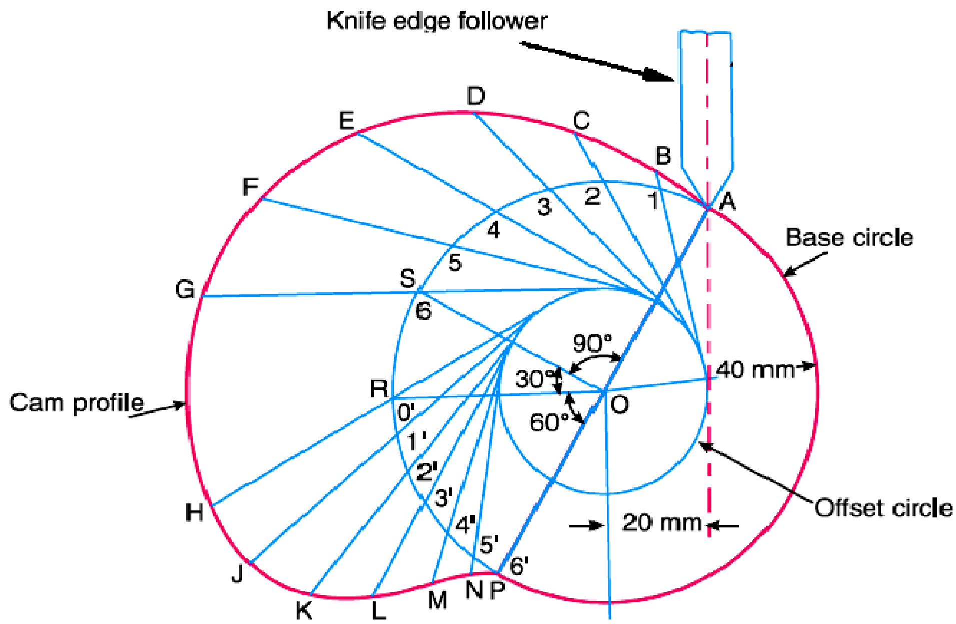
Post Test


Outstroke
Dwell
Return stroke
Dwell
Y
G
H
e
d
c
b
a
e'
d'
c'
b'
a'
B
C
D
E
F
J
K
L
M
N
P
S
R
A0
1
2
3
4
5
6
0'
1'
2'
3'
4'
5'
6'
X
40 mm
90°
30°
60°
180°

(a) profile of cam:



(b) profile of the cam with offset:



MAXIMUM VELOCITY OF THE FOLLOWER(during ascent and descent)

$$\omega = \frac{2\,\pi\,N}{60} = \frac{2\,\pi \times 240}{60} = 25.14\ rad/s$$

$$\theta_o = 90^o = \frac{\pi}{2}\ rad = 1.571\,rad$$

$$v_o = \frac{\pi\ \omega\ S}{2\ \theta_o} = \frac{\pi \times 25.14 \times 0.04}{2 \times 1.571} = 1\ m/s$$

$$\theta_R = 60^o = \frac{\pi}{3}\ rad = 1.047\ rad$$

$$v_R = \frac{\pi\ \omega\ S}{2\ \theta_R} = \frac{\pi \times 25.14 \times 0.04}{2 \times 1.047} = 1.51\ m/s$$

MAXIMUM ACCELERATION OF THE FOLLOWER(during ascent and descent)

$$a_o = \frac{\pi^2 \ \omega^2 \ S}{2\,(\theta_O)^2} = \frac{\pi^2 \times (25.14)^2 \times 0.04}{2 \times (1.571)^2} = 50.6 \ m/s^2$$

$$a_R = \frac{\pi^2 \ \omega^2 \ S}{2\,(\theta_R)^2} = \frac{\pi^2 \times (25.14)^2 \times 0.04}{2 \times (1.047)^2} = 113.8 \ m/s^2$$

Reference

R. S. Khurmi, J. K. Gupta, "Theory of machine"

وزارة التعليم العالي والبحث العلمي

هيئة التعليم التقني

التخصصات / التكنولوجية

القسم / الميكانيك　　　　　　　　الفرع / الإنتاج (مستمر)

اسم المادة	السنة الدراسية	الساعات الأسبوعية		
تقنية اجزاء المكائن (Machine Parts)	الثانية	نظري	عملي	المجموع
		3	---	3

هدف المادة : تهدف أجزاء المكائن الى توضيح دور الاجزاء الميكانيكية في نظام الماكنة والعلاقة

التي تربط هذه الاجزاء ببعضها وكيفية أجراء بعض الحسابات لتصميم هذه الأجزاء وتحديد كـــل

العوامل المؤثرة عليها .

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

References:-

1-Strength of Material by Ferdinal L .Singer

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

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

4-Machine Design by Paul H.Black .

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

Theoretical Subjects	
Week No.	Subject Topics
1	Review of Strength of Materials
2-3	Riveted Joints. Types of Riveted Joints, Design of Riveted Joints, Efficiency of Riveted Joints .
4-5	Welded Joints Types of welding Joints , Design of welding Joints
6-7	Screwed Joints, Design of Bolts for Fastening , Design of Bolts for Pow Transition .
8-9	Keyed Joints , Types of Key , Design of Sunk Key .
10-11	Frictional Clutches, Type of Frictional Clutches , Design of Frictional Clutches.
12-13	Types of Springs , Design of Springs
Week No.	Subject Topics

14-15	Types of Belts , Design of Belts.
16-17	Design of Shafts
18-19	Design of Journal Bearings
20	Selection of Ball Bearings
21-22	Design of Gears by Lewis Equation
23-24	Gears Trains
25-26	Design of Simple Gears Box
27-28	Worm Gears
29-30	Cams

تضم مادة تقنية الاجزاء للمرحلة الثانية عدد من المواضيع موزعة في المنهاج على 30 اسبوع (انظر ملف المحتويات) ولكنها موزعة على 15 محاضرة في هذه الحقيبة.