UE18ME101: MECHANICAL ENGINEERING SCIENCES (4-0-0-4-4)

UNIT -3

CCI:	**AJN**		**NO. OF HOURS : 09**		
Class No.	**Reference**	**Topic**	**Page No. (Reference Material)**	**Reference Section**	**Remark**
22	T1 - Chapter 15 R4 - Chapter 8	**Unit 3: Power Transmission** Power transmission systems - types: belt, chain, rope and gear, belt drive - velocity ratio, description of slip and creep in belt drives	1, 2, 3, 10, 11, 13	15.1, 15.2 15.3, 15.4, 15.5, 15.2.2, 15.2.3	
23	T1 - Chapter 15 R4 - Chapter 8	types of belt sections; open and cross belt drives; tight and slack sides; pulleys - stepped cone pulleys	2, 4, 5,	15.2.1 15.2.4 15.2.5	
24	T1 - Chapter 15 R4 - Chapter 8	ratio of tensions in open belt drives (flat and v-belt), power transmitted; **simple numericals** based on the same. **(no derivation)**	6, 7,	15.2.6 R4 - 8.6	Not only ones shown in the reference material
25	T1 - Chapter 15 R4 - Chapter 8	Decription of chain drives, length of chain drive, simple numericals	11	15.4.1 15.4.2 R4 – 8.6 R3 – 4.4	
26	T1 - Chapter 15 R4 - Chapter 8	Gear Drives: description, velocity ratio, train value	13, 14, 19	15.5.1 15.7.2	
27	T1 - Chapter 15 R4 - Chapter 8	gear classification - based on shafts and teeth meshing, applications	16, 17, 18	15.6.1	
28	T1 - Chapter 15 R4 - Chapter 8	simple gear train - **numericals**	16		Not only one shown in the reference material
29	T1 - Chapter 15 R4 - Chapter 8	compound gear train - **numericals**	20		Not only one shown in the reference material
30	T1 - Chapter 15 R4 - Chapter 8	Introduction to reverted gear train, epicyclic gear train, sun and planetary gear train	20, 21	15.7.3 15.7.4 15.7.5	

REFERENCES:

T1	“Basic Mechanical Engineering”, Pravin Kumar, Pearson, 2013
R1	"Non Conventional Energy Resources", B H Khan, Tata Mc Graw Hill, Second Edition, New Delhi, 2009
R2	“An Introduction to Mechanical Engineering – Part I”, Hodder Education, 2009
R3	“An Introduction to Mechanical Engineering – Part II”, Hodder Education, 2010
R4	“An Introduction to Mechanical Engineering”, J Wickert and K Lewis, Cengage Learning, 3rd Edition, 2013

2. A steam engine of 100 kW runs at 100 rpm. The speed of the engine is to be maintained within 1% variation in mean speed. The flywheel has mass of 2,000 kg and radius of gyration of 1 m. Determine the coefficient of fluctuation of energy.
3. An electric motor drives a punching press. A flywheel fitted to the press has a radius of gyration 0.4 m and runs at 240 rpm. The press is capable of punching 600 holes/h with each punching operation taking 1.5 s and requiring 10,000 N m of work. Determine the rating of the machine in kW and the mass of the flywheel if the speed of the flywheel does not drop below 220 rpm.
4. A porter governor has equal arms length of 220 mm long and pivoted at the axis of rotation. Each ball has a mass of 5 kg and the mass of central load on the sleeve is 20 kg. When the governor begins to lift, the radius of rotation of the ball is 100 and 120 mm when the governor is at maximum speed. Find the minimum and maximum speeds and range of speed of the governor.
5. A Proell governor has equal arms length of 420 mm. The upper and lower ends of the arms are pivoted on the axis of the governor. The extension arms of the lower links are each 80 mm long and parallel to the axis when the radii of rotation of the balls are 160 and 220 mm. The mass of each ball is 12 kg and mass of the central load is 60 kg. Determine the range of the speed of the governor.

15 Power Transmission Devices

Chapter Objectives

In this chapter, you will learn about:

- Belt drive
- Chain drive
- Gear drive
- Gear train

15.1 INTRODUCTION

Power transmission is a process to transmit motion from one shaft to another by using some connection between them like belt, rope, chain, and gears. To connect the shafts, mainly two types of connectors are used, one is flexible and other is rigid. In flexible types of connection, there is relative velocity between shaft and connectors due to slip and strain produced in the connectors. But in case of rigid connection, there is no relative velocity between the connector and shaft.

Belt, rope, and chain are flexible connectors where gears are rigid connectors. Generally, belt, rope, and chain drives are used when the distance between the shafts is large and gears are used when distance between the shafts is very small. Efficiency of gear drive is much more than that of belt, rope, and chain drive due to absence of slipping effect.

15.2 BELT DRIVE

In belt drive, the velocity of two shafts can be varied by variation in diameter of pulley on which belt is mounted. But in chain or gear drive, the velocity of two shafts is varied by variation in the number of teeth on sprocket and gear, respectively. If an unstretched belt is mounted on the pulleys, the outer and inner faces of belt are subjected to tension and compression, respectively. In between the sections, there is a neutral section which has no tension or compression. Usually, this is considered at the half of thickness of the belt. The effective radius of rotation of a pulley is obtained by

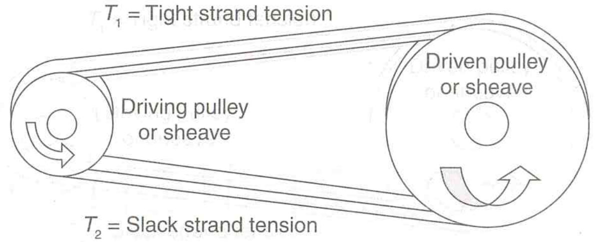


Figure 15.1 Belt Drive

15.2.1 Type of Belt Cross-sections

Figure 15.2 shows flat belt and V-belt. In the flat belt drive, rim of pulley is slightly crowned which helps to keep the belt running centrally on the pulley rim as shown in Figure 15.2(a). For the V-belt drive, grooves are made on rim of pulley for wedging action. The belt does not touch the bottom of the groove as shown in Figure 15.2(b). Owing to wedging action, V-belts need little adjustment and transmit more power, without slip, as compared to flat belts. In multiple V-belt system, more than one belt on the pulleys can be used to increase the power transmission capacity.

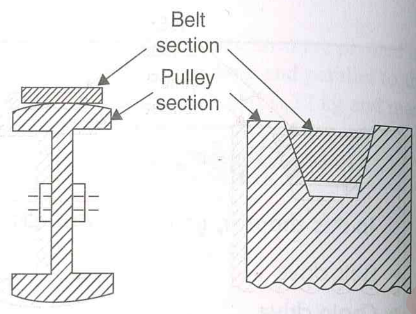


Figure 15.2 Type of Belt Cross-Sections: (a) Flat Belt and (b) V-Belt

15.2.2 Velocity Ratio

Velocity ratio is the ratio of speed of the driven pulley to that of the driving pulley.

Let N_1 is rotational speed of the driving pulley
N_2 is rotational speed of the driven pulley
D_1 is diameter of driving pulley
D_2 is diameter of driven pulley
t is thickness of the belt

$$V = \frac{\pi(D_1 + 2t)N_1}{60} = \frac{\pi(D_2 + 2t)N_2}{60}$$

i.e., $D_1N_1 = D_2N_2$, where t is very small in comparison to D, therefore it can be negelected.

$$VR = \frac{N_2}{N_1} = \frac{D_1}{D_2}$$

Slip

The effect of slip is a decrease in the speed of belt on driving shaft and then driven shaft.

Let ω_1 is angular velocity of driving pulley
ω_2 is angular velocity of driven pulley
S_1 is percentage slip between driving pulley and belt

Peripheral speed of the driving pulley $= \dfrac{\omega_1 D_1}{2}$

Speed of belt on driving pulley $= \dfrac{\omega_1 D_1}{2}\left(\dfrac{100 - S_1}{100}\right)$

This is also the speed of belt on driven pulley.

Now, peripheral speed of driven pulley $= \dfrac{\omega_1 D_1}{2}\left(\dfrac{100 - S_1}{100}\right)\left(\dfrac{100 - S_2}{100}\right)$

If S is total slip percentage, peripheral speed of driven pulley $= \dfrac{\omega_1 D_1}{2}\left(\dfrac{100 - S}{100}\right)$

or
$$\frac{\omega_1 D_1}{2}\left(\frac{100 - S_1}{100}\right)\left(\frac{100 - S_2}{100}\right) = \frac{\omega_1 D_1}{2}\left(\frac{100 - S}{100}\right)$$

or
$$S = S_1 + S_2 - 0.01\, S_1 S_2$$

Thus, velocity ratio,
$$VR = \frac{D_1}{D_2}\left(\frac{100 - S}{100}\right) = \frac{N_2}{N_1}$$

15.2.3 Creep

When belt passes from slack to tight side, a certain portion of belt extends and again contracts when belt passes through tight to slack side. Due to fluctuation in length of the belt, there is relative motion between belt and pulley surface. This relative motion is known as creep. Considering the creep, velocity ratio can be expressed by,

$$\frac{N_2}{N_1} = \frac{D_1}{D_2} \times \frac{E + \sqrt{\sigma_2}}{E + \sqrt{\sigma_1}}$$

where N_1 and N_2 are the speeds of driving and driven pulleys, respectively; D_1 and D_2 are the diameters of driver and driven pulleys, respectively; σ_1 and σ_2 are the stresses developed in tight and slack side of belt, respectively; and E is the modulus of elasticity of belt materials.

Example 15.1: The speed of a driving shaft is 100 rpm and the speed of the driven shaft is 150 rpm. Diameter of the driving pulley is given as 500 mm, find the diameter of driven pulley in the following cases:

(i) If the belt thickness is negligible.
(ii) If the belt thickness is 6 mm.
(iii) If total slip is 5% considering thickness of belt.
(iv) If a slip is 2% on each pulley considering thickness of belt.

Solution:

(i) $\dfrac{N_1}{N_2} = \dfrac{D_2}{D_1} \Rightarrow D_2 = D_1 \times \dfrac{N_1}{N_2} = 500 \times \dfrac{100}{150} = 333.33\,\text{mm}$

(iii) $$\frac{N_2}{N_1} = \frac{D_1 + t}{D_2 + t}\left(\frac{100 - S}{100}\right)$$

or $$\frac{150}{100} = \left(\frac{500+6}{D_2+6}\right)\left(\frac{100-5}{100}\right) \Rightarrow D_2 = \frac{100}{150}\left(\frac{100-5}{100}\right)(500+6) - 6 = 314.46\,\text{mm}$$

(iv) $$\frac{N_2}{N_1} = \frac{D_1 + t}{D_2 + t}\left(\frac{100 - S}{100}\right),$$

where $S = S_1 + S_2 - 0.01S_1 \times S_2 = 2 + 2 - 0.01 \times 2 \times 2 = 3.96$

$$D_2 = \frac{100}{150}\left(\frac{100-3.96}{100}\right)(500+6) - 6 = 317.97$$

15.2.4 Open-belt Drive

Open-belt drive is used to provide same direction of rotation to driven shaft as the direction of rotation of driving shaft.

Let L is length of belt for open drive
r is radius of smaller pulley
R is radius of larger pulley
C is centre distance between pulleys
β is angle subtended by each common tangent on centre of pulley (CD) or (EF)
AB is the line joining the centres of pulleys

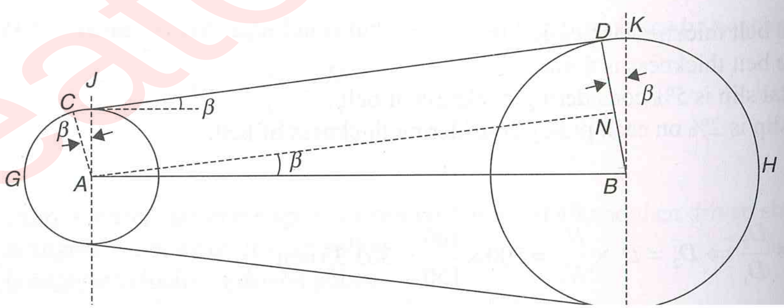
From Figure 15.3,

$$\angle CAJ = \angle NAB = \beta$$

$$L = 2(\text{Arc } GC + CD + \text{Arc } DH) = 2\left[\left(\frac{\pi}{2} - \beta\right)r + AN + \left(\frac{\pi}{2} + \beta\right)R\right]$$

$$= \pi(R + r) + 2\beta(R - r) + 2C\cos\beta; \text{ since } AN = AB\cos\beta = C\cos\beta$$

for small angle β, $\beta = \sin\beta = \dfrac{R - r}{C}$ and $\cos\beta = \sqrt{1 - \sin^2\beta} \cong 1 - \dfrac{1}{2}\sin^2\beta = 1 - \dfrac{1}{2}\left(\dfrac{R - r}{C}\right)^2$



Putting the value of β and $\cos\beta$ in equation of L, we get

$$L = \pi(R + r) + 2C + \frac{(R - r)^2}{C}$$

15.2.5 Crossed-belt Drive

Cross-belt drive is used to provide reverse direction of rotation to driven shaft as the direction of rotation of driving shaft.

Similar to open-belt drive, let A and B be the pulley centres and CD and EF be the common tangents to the two pulleys as shown in Figure 15.4.

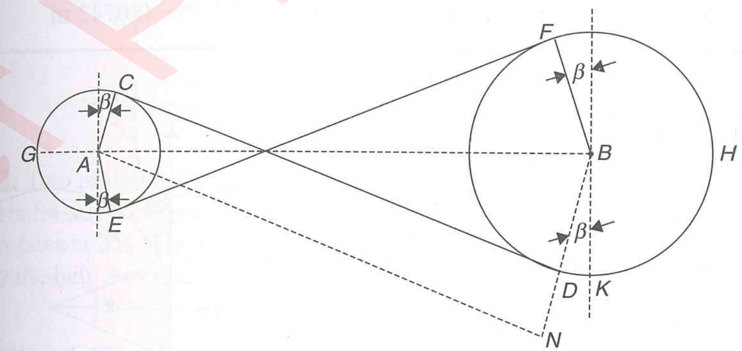


Figure 15.4 Crossed-belt Drive

In Figure 15.4, belt length, $L = 2\,(\text{Arc } GC + CD + \text{Arc } DH)$

or $$L = \left[2\left(\frac{\pi}{2} + \beta\right)r + C\cos\beta + \left(\frac{\pi}{2} + \beta\right)R\right] = [(\pi + 2\beta)(R + r) + 2C\cos\beta]$$

As $$\beta = \sin\beta = \frac{R + r}{C} \text{ as } \beta \text{ is very small.}$$

$$\cos\beta = \left(1 - \frac{1}{2}\beta^2\right) = 1 - \frac{1}{2}\left(\frac{R + R}{C}\right)^2$$

or $$L = \left[\pi + 2\left(\frac{R + r}{C}\right)\right](R + r) + 2C\left[1 - \frac{1}{2}\left(\frac{R + r}{C}\right)^2\right]$$

$$L = \pi(R + r) + 2C + \frac{(R + r)^2}{C}$$

Example 15.2: Two shafts drive are arranged parallel to each other at a distance of 5 m. If the pulley diameters mounted on the shafts are 500 mm and 750 mm. Determine the difference in length of the belts for opposite direction of rotation and same direction of rotation.

Solution:

For crossed-belt drive:

$$L=\pi(R+r)+2C+\frac{(R+r)^2}{C}=\frac{\pi\times(500+750)\times10^{-3}}{2}+2\times5+\left(\frac{500+750}{2}\right)^2\times10^{-6}\times\frac{1}{5}$$

$= 12.041$ m.

For open-belt drive:

$$L=\pi(R+r)+2C+\frac{(R-r)^2}{C}=\frac{\pi\times(500+750)\times10^{-3}}{2}+2\times5+\left(\frac{750-500}{2}\right)^2\times10^{-6}\times\frac{1}{5}$$

$= 11.966$ m

Difference in length of cross belt and open belt $= 12.041 - 11.966 = 0.0743$ m

15.2.6 Ratio of Tensions

Let T_1 is tension in tight side of the belt
T_2 is tension in slack side of the belt
θ is angle of lap of belt over the pulley
μ is coefficient of friction between the belt and pulley

Consider a short length of belt subtending an angle $\delta\theta$ at the centre of the pulley as shown in Figure 15.5.

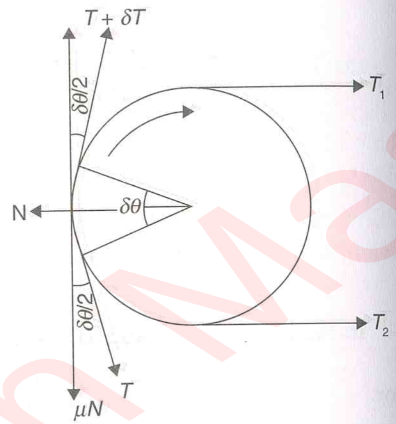


Figure 15.5 Tensions in Belt

Let N is a normal reaction between the element length of a belt and pulley
δT is increase in tension in tight side than that on slack side
$T + \delta T$ is tension on the tight side of the element

Resolving the force in tangential direction,

$$\mu N + T\cos\delta\theta/2 - (T+\delta T)\cos\delta\theta/2 = 0$$

As $\delta\theta$ is very small, $\cos\delta\theta/2 \cong 1$

$$\mu N + T - T - \delta T = 0 \quad \text{or} \quad \delta T = \mu N \tag{15.1}$$

Resolving the force in radial direction,

$$N - T\sin\delta\theta/2 - (T+\delta T)\sin\delta\theta/2 = 0$$

As $\delta\theta$ is very small, $\sin\delta\theta/2 \cong \delta\theta/2$

$$N - T\delta\theta/2 - T\delta\theta/2 - \delta T\,\delta\theta/2 = 0$$

Neglecting the product of two small quantities

$$N = T\,\delta\theta \tag{15.2}$$

δT

On integration, we get

$$\int_{T_2}^{T_1}\frac{\delta T}{T}=\int_0^{\theta}\mu\,\delta\theta \Rightarrow \frac{T_1}{T_2}=e^{\mu\theta}$$

In V-belt, $\frac{T_1}{T_2}=e^{\mu\theta/\sin\alpha}$, where α is the angle made by V-section of the belt.

Power Transmission in Belt Drive

$$P = T_{\text{resultant}}\times\text{Velocity}$$

$= (T_1 - T_2)\times V$, where P is power transmitted in watt and V is velocity of belt.

$$=T_1\left(1-\frac{T_1}{T_2}\right)\times V=T_1\left(1-\frac{1}{e^{\mu\theta}}\right)\times V$$

$$=T_1\left(\frac{e^{\mu\theta}-1}{e^{\mu\theta}}\right)\times V,\ \text{watt}$$

Example 15.3: Two pulleys of diameters 500 and 200 mm are mounted on two parallel shafts 2 m apart. These shafts are connected by a cross belt. Find the angle of contact of belt and pulley. If larger pulley rotates at 250 rpm and maximum permissible tension in the belt is 1 kN, find the power transmitted by the belt. Assume coefficient of friction between belt and pulley is 0.25.

Solution:

For crossed belt (Refer Figure 15.4)

$$\sin\beta=\frac{r_1+r_2}{C}=\frac{100+250}{2{,}000}=0.175\Rightarrow\beta=10.07^\circ$$

$$\theta=180^\circ+2\beta=180^\circ+2\times10.07^\circ=200.15^\circ=200.15^\circ\times\frac{\pi}{180^\circ}=3.493\,\text{rad}$$

$$\frac{T_1}{T_2}=e^{\mu\theta}=e^{0.25\times3.493}=2.3946$$

$$\Rightarrow\quad T_2=\frac{T_1}{2.3946}=\frac{1{,}000}{2.3946}=417.592\,\text{N}$$

$$P=(T_1-T_2)V=(T_1-T_2)\frac{\pi D_1N_1}{60}=(1{,}000-417.592)\times\frac{\pi\times500\times250}{60}=3.811\,\text{kW}$$

15.2.7 Effect of Centrifugal Force on Belt Drive

When velocity of the belt is more than 10 m/s, the centrifugal force becomes predominant. Analyse the various components of forces as shown in Figure 15.6.

Let ρ is the density of belt materials
T_c is centrifugal tension on the belt element in tight and slack side
r is radius of the pulley
t is thickness of the belt

σ is maximum allowable stress in the belt
m is mass per unit length of the belt
F_C is the centrifugal force on the element
V is the velocity of the belt
$\Delta\theta$ is the angle of lap

Now,

F_c = (Length of belt element × Mass per unit length) × Acceleration

$$F_c = rd\theta \times bt \times \rho \frac{V^2}{r} = \rho V^2 \times bt \times \delta\theta \quad (15.3)$$

Also, $F_c = 2T_c \sin \delta\theta/2 = 2T_c \times \delta\theta/2 = T_c \times \delta\theta$ (15.4)

From Eqs. (15.3) and (15.4), we get

$$T_c = \rho\,(bt)\,V^2$$

Total tension on tight side,

$$T = T_1 + T_c,$$

where T is maximum allowable tension equal to $\sigma \times b \times t$

Total tension on slack side = $T_2 + T_c$

Now, Power, $$P = T_1\left(\frac{e^{\mu\theta}-1}{e^{\mu\theta}}\right)\times V = \left(\sigma bt - \rho V^2 bt\right)\left(\frac{e^{\mu\theta}-1}{e^{\mu\theta}}\right)\times V$$

$$= bt\left(\sigma - \rho V^2\right)\left(\frac{e^{\mu\theta}-1}{e^{\mu\theta}}\right)\times V$$

For maximum power transmission, $\frac{\delta P}{\delta V} = 0$

or $$bt\left(\sigma - 3\rho V^2\right)\left(\frac{e^{\mu\theta}-1}{e^{\mu\theta}}\right) = 0$$

or $$\sigma = 3\rho V^2$$

or $$V = \sqrt{\frac{\sigma}{3\rho}} = \sqrt{\frac{T}{3m}}$$

Initial tension in the belt, $T_0 = (T_1 + T_2)/2$

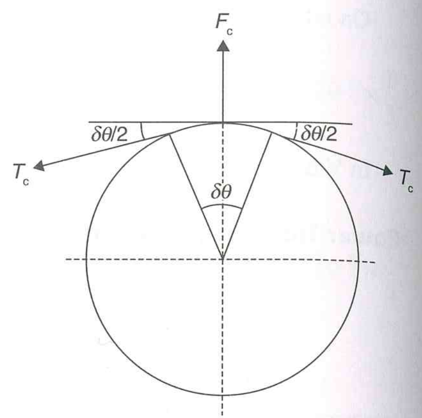


Figure 15.6 Centrifugal Force in Belt Drive

Example 15.4: A leather belt of density 1,000 kg/m³, thickness 10 mm is used to transmit a power of 8 kW from a pulley 1.5 m in diameter running at 300 rpm. Determine the width of the belt taking centrifugal tension into account. If angle of lap is 165° and coefficient of friction between belt and pulley is 0.25. Assuming allowable stress for leather belt is 1.5 MPa.

Solution:

$\pi D_1 N_1 \quad \pi\times 1.5\times 300$

$$\frac{T_1}{T_2} = e^{\mu\theta} = e^{0.25\times 2.879} = 2.054$$

$$T_1 = 2.054T_2 \quad (15.5)$$

$$P = (T_1 - T_2)\,V \quad \text{or} \quad 8\times 10^3 = (T_1 - T_2)\times 23.56$$

$$T_1 - T_2 = \frac{8\times 10^3}{23.56} = 339.558 \quad (15.6)$$

From Eqs (15.5) and (15.6), we get

$$2.054T_2 - T_2 = 339.558 \Rightarrow T_2 = \frac{339.558}{1.054} = 661.72\ \text{N}$$

Mass of the belt per unit length = Area × Density

$$m = b\times t\times \rho = b\times 0.01\times 1{,}000 = 10b\ \text{kg}$$

$$\text{Centrifugal tension} = mV^2 = 10b\,(23.56)^2 = 5550.73b$$

$$\text{Maximum tension in the belt} = \sigma\times b\times t = 1.5\times 10^6\times 0.01\times b = 15{,}000b\ \text{N}$$

$$T = T_1 + T_c \quad \text{or} \quad 15{,}000b = 661.72 + 5{,}550.73b$$

or $$b = \frac{661.72}{9{,}449.264} = 0.07002\ \text{m} = 70\ \text{mm}$$

Example 15.5: An open-belt drive transmits a power of 3.0 kW. The linear velocity of the belt is 3 m/s. Angle of lap on smaller pulley is 160°. The coefficient of friction between belt and pulley is 0.25. Determine the effect on power transmission in the following cases:

(i) Initial tension in belt is increased by 10%.
(ii) Angle of lap is increased by 10% using idler pulley for the same speed and tension in tight side.
(iii) Coefficient of friction is increased by 10% for same initial tension.

Solution:

$$P = 3\ \text{kW} = 3\times 10^3\ \text{W}$$

$$\theta = 160° = 160°\times\frac{\pi}{180°} = 2.792\ \text{rad}$$

$$\mu = 0.25,\ V = 3\ \text{m/s}$$

$$P = (T_1 - T_2)\,V \quad \text{or} \quad 3\times 10^3 = (T_1 - T_2)\,3$$

$$T_1 - T_2 = 1{,}000\ \text{N}$$

$$\frac{T_1}{T_2} = e^{\mu\theta} = e^{0.25\times 2.792} = 2 \quad \text{or} \quad T_1 = 2T_2$$

i.e., $$2T_2 - T_2 = T_2 = 1{,}000\ \text{N and } T_1 = 2T_2 = 2{,}000\ \text{N}$$

Initial tension, $$T_0 = \frac{T_1 + T_2}{2} = \frac{2{,}000 + 1{,}000}{2} = 1{,}500\ \text{N}$$

$$T_0' = \frac{T_1 + T_2}{2} = 1{,}650\text{ N} \Rightarrow T_1 + T_2 = 3{,}300\text{ N}$$

As μ and θ remain unchanged, $T_1 = 2T_2$

$$2T_2 + T_2 = 3T_2 = 3{,}300\text{ N} \Rightarrow T_2 = 1{,}100 \text{ and } T_1 = 2{,}000\text{ N}$$

$$P = (T_1 - T_2)\,V = (2{,}200 - 1{,}100) \times 3 = 3{,}300\text{ W}$$

$$\text{Increase in power} = \frac{3{,}300 - 3{,}000}{3{,}000} = 10\%$$

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

T_1 is the same as before whereas θ is increased by 10%

$$\frac{2{,}000}{T_2} = e^{0.25 \times 2.792 \times 1.1} = 2.155 \Rightarrow T_2 = 928.06$$

$$P = (T_1 - T_2)\,V = (2{,}000 - 928.06) \times 3 = 3{,}215.82\text{ W}$$

$$\text{Increase in power} = \frac{3{,}215.82 - 3{,}000}{3{,}000} = 7.1\%$$

(iii) When frictional coefficient increases by 10%

$$\frac{T_1}{T_2} = e^{\mu'\theta} = e^{0.25 \times 1.1 \times 2.792} = 2.155 \Rightarrow T_1 = 2.155T_2$$

$$2.155T_2 + T_2 = 3.155T_2 = 3{,}000\text{ N} \Rightarrow T_2 = 950.871\text{ N}$$

$$T_1 = 2.155T_2 = 2.155 \times 950.871 = 2{,}049.128\text{ N}$$

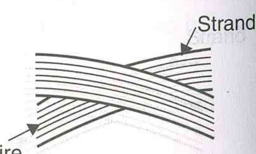
$$P = (T_1 - T_2)\,V = (2{,}049.128 - 950.871) \times 3 = 3{,}924.772\text{ W}$$

$$\text{Increase in power} = \frac{3{,}294.772 - 3{,}000}{3{,}000} = 9.82\%$$

15.3 ROPE DRIVE

Rope drive is very similar to belt drive. It is classified as:

- Fibre ropes
- Wire ropes



Fibre ropes are made of manila or cotton. Wire ropes are made of steel wires. A number of wires make a strand and strands make a rope as shown in Figure 15.7. Each strand is twisted with other strands

15.4 CHAIN DRIVE

To overcome the problem of slip in belt drive or rope drive, chain drive is used. A schematic diagram of chain drive is shown in Figure 15.8. The velocity ratio in chain drive remains constant. But, chain drive is heavier than the belt drive and there is gradual stretching in its strength. Time to time some of its links have to be removed. Lubrication of its parts is also desired. The wheel over which chains are run, corresponding to the pulleys in a belt is known as sprocket having projected teeth that fit into the recess in the chain.

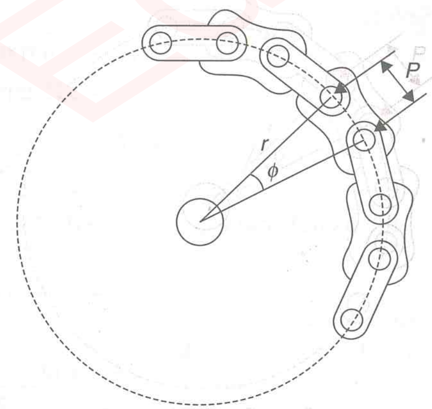


Figure 15.8 Chain Drive

Pitch: Distance between two consecutive roller centres is known as pitch, p.

Pitch Circle: The circle drawn through the roller centres of a wrapped chain round a sprocket is called the pitch circle.

Let T is number of teeth on a sprocket
ϕ is angle subtended by chord of link at the centre
r is radius of pitch circle

$$\frac{p}{2} = r\sin\phi/2;\; p = 2r\sin\frac{1}{2}\left(\frac{360^\circ}{T}\right) = 2r\sin\frac{180^\circ}{T}$$

or $$r = \frac{p}{2\sin\dfrac{180^\circ}{T}} = \frac{p}{2}\operatorname{cosec}\frac{180^\circ}{T}$$

15.4.1 Chain Length

Let R and r are the radii of the pitch circle of two sprockets having teeth T and t, respectively.
L is length of the chain
C is centre distance between sprockets $= K \cdot p$

$$L = \pi(R + r) + \frac{(R - r)^2}{C} + 2C$$

Since $$R = \frac{p}{2}\operatorname{cosec}\frac{180^\circ}{T} \text{ and } r = \frac{p}{2}\operatorname{cosec}\frac{180^\circ}{t}$$

Now, $$L = \frac{p \times (T + t)}{2} + \frac{\frac{p}{2}\left(\operatorname{cosec}\frac{180^\circ}{T} - \operatorname{cosec}\frac{180^\circ}{t}\right)^2}{K \cdot p} + 2\,K \cdot p$$

or $$L = p\left[\frac{(T + t)}{2} + \frac{\left(\operatorname{cosec}\frac{180^\circ}{T} - \operatorname{cosec}\frac{180^\circ}{t}\right)}{4K} + 2K\right]$$

diameter of the driven sprocket is 650 mm and centre distance between sprockets is 100 mm. Also, determine the pitch and length of the chain.

Solution:

$$N_1 = 250 \text{ rpm}, N_2 = 100 \text{ rpm}, T_1 = 20, T_2 = ?$$

$$r_2 = 325 \text{ mm}, C = 1{,}000 \text{ mm} = 1 \text{ m}$$

$$N_1 \cdot T_1 = N_2 \cdot T_2 \quad \text{or} \quad T_2 = \frac{N_1 \times T_1}{N_2} = \frac{250 \times 20}{100} = 50 \text{ Teeth}$$

Pitch circle radius, $$r_2 = 0.325 = \frac{p}{2}\operatorname{cosec}\frac{180°}{T_2} = \frac{p}{2}\operatorname{cosec}\frac{180°}{50}$$

$$p = 0.0408 \text{ m} = 40.8 \text{ mm}$$

$$L = p\left[\frac{(T_1 + T_2)}{2} + \frac{\left(\operatorname{cosec}\frac{180°}{T} - \operatorname{cosec}\frac{180°}{t}\right)}{4K} + 2K\right]$$

$$= 0.0408\left[\frac{(20+50)}{2} + \frac{\left(\operatorname{cosec}\frac{180°}{20} - \operatorname{cosec}\frac{180°}{50}\right)}{4 \times 1} + 2 \times 1\right] = 3.465 \text{ m}$$

15.4.2 Types of Chain

Hosting Chain: This type of chain is used for lower speed. It consists of oval links as shown in Figure 15.9.

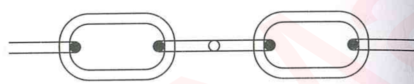


Figure 15.9 Hosting Chain

Conveyer Chain: Conveyer chain may be detachable/hook joint type/closed end pintle type as shown in Figure 15.10. The sprocket teeth are so shaped and spaced that the chain could run onto and off the sprockets smoothly and without interference. Such chains are used for low speed applications.

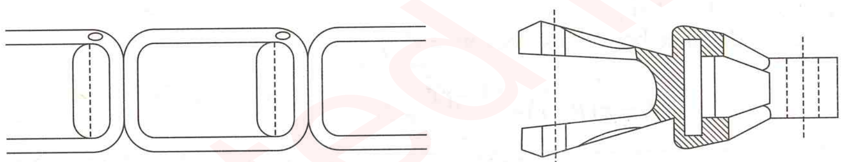


Figure 15.10 (a) Hook Joint Type Conveyer Chain and (b) Closed-end Pintle Type Conveyer Chain

Power Transmission Chains

Block Chain: This is used for power transmission at low speed such as bicycle, motor bike, etc. (Figure 15.11).

Roller Chain: A common form of roller chain is shown in Figure 15.12. A bush is fixed in inner link whereas the outer link has a pin fixed to it. There is only sliding motion between the pin and the bush. The roller is made of hardened steel and is free to turn on the bush. A good roller chain is quite and wears less in comparison to a block chain.

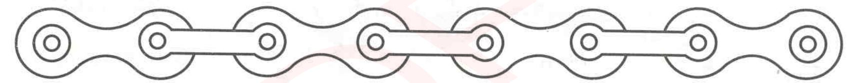


Figure 15.12 Roller Chain

Silent/Inverted Tooth Chain: Roller chains can run at very high speed. But when maximum quietness is required, inverted tooth chains are required. It has no roller; the links are so shaped as to engage directly with the sprocket teeth and included angle is either 60° or 75° (Figure 15.13).

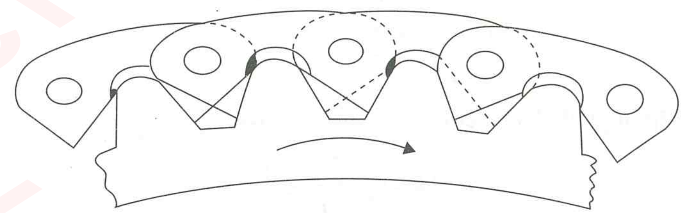


Figure 15.13 Silent or Inverted Tooth Chain

15.5 GEAR DRIVE

Gears are compact power transmission device that controls the speed, torque, and direction of rotation of driven shaft. Gears may be classified into five main categories: spur, helical, bevel, hypoid, and worm. Shaft orientation, efficiency, and speed determine the application of gear drive. Gears are toothed disc which transmit power from one shaft to other shaft by meshing with teeth of other gear.

15.5.1 Gear Terminology

All the important gear terminologies are shown in Figure 15.14.

Pitch Point: The point of contact between pitch circles of two gears is known as pitch point.

Pitch Circle: The circle passing through point of contacts of two gears is known as pitch circle.

Pitch Diameter, D: Diameter of pitch circle is known as pitch diameter.

$$D = \frac{N}{P_d} = \frac{N \times P_c}{\pi}$$

Circular Pitch, P_c: It is the distance measured along the circumference of the pitch circle from a point on one tooth of the corresponding point on the adjacent tooth.

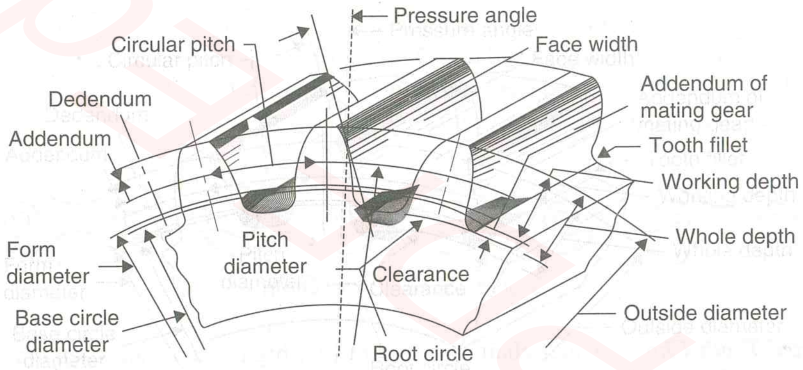


Figure 15.14 Nomenclature of Gear

Diametral Pitch, P_d: It is the number of teeth per unit length of the pitch circle diameter.

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

Module, m: It is ratio of pitch diameter to the number of teeth.

$$m = \frac{D}{T} = \frac{1}{P_d} = \frac{P_c}{\pi}$$

Gear Ratio: It is the ratio of number of teeth on gear and pinion.

$$G = \frac{T}{t}$$

Velocity Ratio: It is ratio of angular velocity of the driving gear to driven gear.

$$\text{VR} = \frac{\omega_1}{\omega_2} = \frac{N_1}{N_2} = \frac{D_2}{D_1} = \frac{T_2}{T_1}$$

Here, subscripts 1 and 2 are used for driving and driven gears, respectively.

Addendum Circle: It is a circle passing through the tips of the teeth.

Addendum: It is the radial height of tooth above the pitch circle. Its standard value is one module.

Dedendum Circle: It is a circle passing through roots of the teeth.

Dedendum: It is a radial depth of a tooth below the pitch circle. Its standard value is 1.157 m.

Full Depth of Teeth: It is the total depth of the tooth space, i.e., Full depth = Addendum + Dedendum = (1 + 1.157) × Module = 2.157 × Module.

Working Depth of Teeth: The maximum depth at which a tooth penetrates into tooth space of the mating gear is known as working depth of teeth.

Backlash: It is difference between the space width and the tooth thickness along the pitch circle.

Face Width: It is length of tooth parallel to the gear axis.

Top Land: It is the surface of the top of the tooth.

Bottom Land: The surface of the bottom of the tooth between the adjacent fillets.

Face: It is the tooth surface between the pitch circle and the top land.

Flank: It is the curved portion of the tooth flank at the root circle.

Pressure Angle, φ: The angle between the pressure line and the common tangent at the pitch point is known as the pressure angle or angle of obliquity.

Path of Contact or Contact Length: Locus of the point of contact of teeth of two mating gears from beginning of engagement to end of engagement is known as path of contact or the contact length.

Path of Approach: Portion of the path of contact from the beginning of engagement to the pitch point is known as path of approach.

Path of Recess: Portion of the path of contact from the pitch point to the end of engagement is known as path of recess.

Arc of Contact: Locus of points on the pitch circle from the beginning of engagement to the end of engagement of two mating gears is known as arc of contact.

Arc of Approach: It is the portion of arc of contact from the beginning of engagement to the pitch point of two mating gears is known as arc of contact.

Arc of Recess: It is the portion of arc of contact from the pitch point to the end of engagement of two mating gears is known as arc of recess.

Contact Ratio: It is the ratio of the length of arc of contact to circular pitch.

$$\text{Contact ratio (Number of pair of teeth in contact)} = \frac{\text{Length of arc of contact}}{\text{Pitch}}$$

15.5.2 Law of Gearing

The law of gearing gives the condition for the tooth profiles for constant angular velocity for two mating gears, which can be explained as: "If angular velocities of two mating gears remain constant, the common normal at the point of the two teeth should always pass through a fixed point P which divides the line joining the centres in the inverse ratio of angular velocities of the gears". In Figure 15.15,

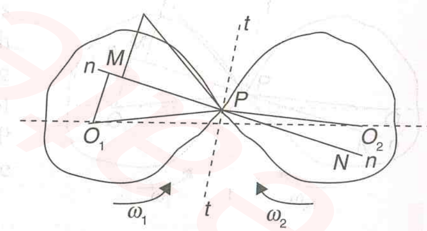


Figure 15.15 Two Teeth in Meshing

$$\frac{\omega_1}{\omega_2} = \frac{O_2N}{O_1M} = \frac{O_2P}{O_1P}$$

15.5.3 Forms of Teeth

Cycloidal Profile Teeth

A cycloid is the locus of points on the circumference of a circle that rolls without slipping on a fixed straight line. An epicycloid is the locus of points on the circumference of a circle that rolls without slipping outside the circumference of another circle. A hypocycloid is the locus of points on the circumference of a circle that rolls without slipping inside the circumference of another circle. The construction of cyloidal teeth is shown in Figure 15.16.

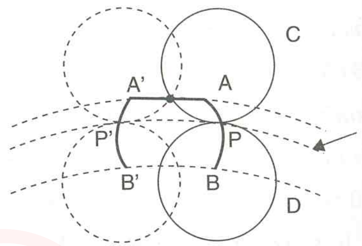


Figure 15.16 Cycloidal Profile of Gear Teeth

Advantages of Cycloidal Gears

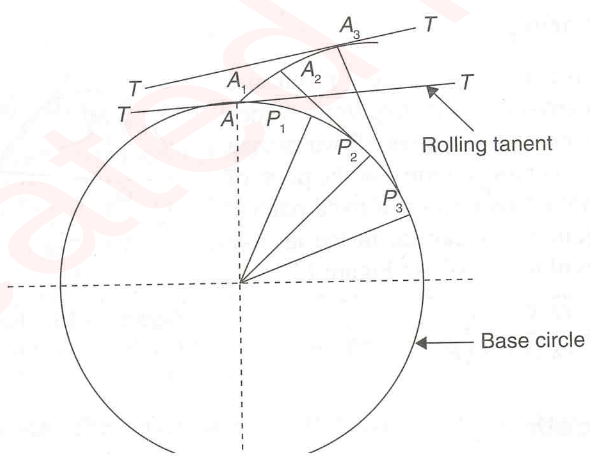
- Due to wider flank, cycloidal gear is stronger than involute gear for the same pitch.
- Less wear occurs in cycloidal teeth.
- There is no phenomenon of interference.

Involute Profile Tooth

An involutes profile is a plane curve generated by the points on tangent on a circle which rolls without slipping or by points on a tight string which is unwrapped from a reel as shown in Figure 15.17.

Advantages of Involute Gears

- Centre distance can be varied within limit without change in pressure angle which is not possible in cycloidal gears.
- Pressure angle remains constant throughout the engagement but in the case of cycloidal gears, pressure angle is maximum at the beginning and end of engagement and minimum at the pitch point.
- The face and flank of involute teeth are generated by a single curve where in cycloidal gears, epicycloids and hypo-cycloid are required for face and flank, respectively. Thus, involute teeth are easy to manufacture than the cycloidal gear.



15.6 CLASSIFICATION OF GEARS

Gears can be classified according to position of shafts as follows.

15.6.1 Parallel Shafts

Spur Gears

General: Spur gears are the most commonly used gear. They are characterized by teeth which are parallel to the axis. The basic descriptive geometry for a spur gear is shown in Figure 15.18.

Advantages: Spur gears are easy to find, inexpensive, and efficient.

Limitations: Spur gears generally cannot be used when a direction change between the two shafts is required. Also, this type of gear is used for smaller speed due to noise creation at high speed power transmission.

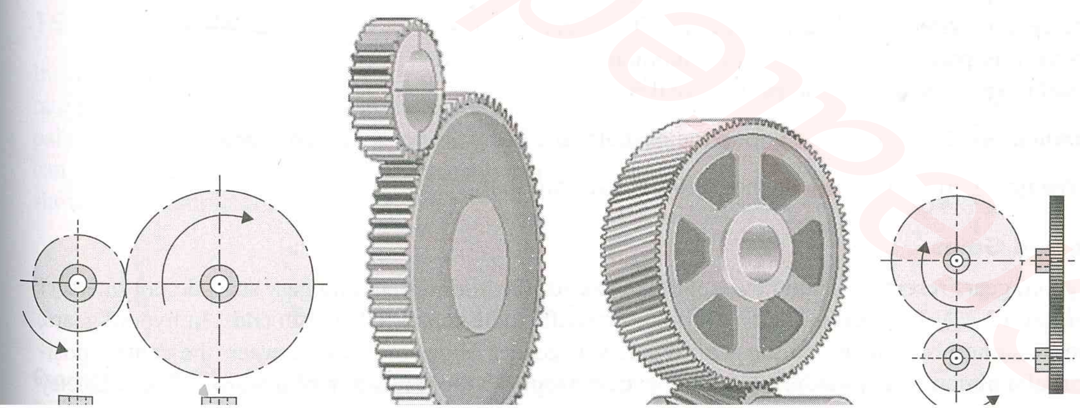
Helical Gears

Helical gear is similar to the spur gear except that the teeth are at an angle to the shaft, rather than parallel to its axis as in a spur gear. The resulting teeth are longer than the teeth on a spur gear of equivalent pitch diameter. The longer teeth cause helical gears to have the following differences from spur gears of the same size:

- Tooth strength is greater because the teeth are longer.
- Greater surface contact on the teeth allows a helical gear to carry more load than a spur gear.
- The longer surface of contact reduces the efficiency of a helical gear relative to a spur gear.

Helical gears may be used to mesh two shafts that are not parallel, although they are still primarily used in parallel shaft applications. A special application in which helical gears are used is a crossed gear mesh, in which the two shafts are perpendicular to each other. The basic geometry for a helical gear is shown in Figure 15.19.

Advantages: Helical gears can be used on non-parallel and even perpendicular shafts, and can carry higher loads than can spur gears.



Limitations: Helical gears are expensive and much more difficult to manufacture. They are also slightly less efficient than a spur gear of the same size.

Double Helical/Herringbone Gears

Double helical gears have one a right-hand helix and the other a left-hand helix. The teeth of two rows are separated by a groove used for tool run out. Axial thrust which occurs in case of single helical gears, two rows of teeth cancel each other. This can be run at high speeds with less noise and vibrations. Figure 15.20 shows the basic structure of double helical gear and herringbone or double helical gear.

Herringbone gears have opposed teeth to eliminate side thrust loads the same as double helical, but they are joined in the middle of the gear circumference. This arrangement makes herringbone gears more compact than double helicals. However, the gear centres must be precisely aligned to avoid interference between the mating helixes.

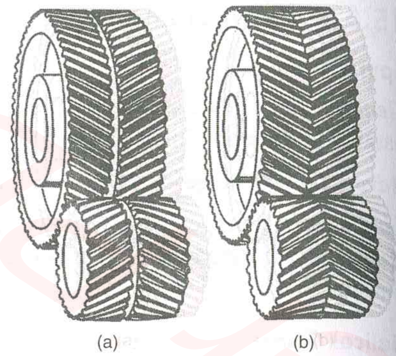


Figure 15.20 (a) Double Helical Gearing with Two Pairs of Opposed Gears, (b) Herringbone Gears Having Opposed Teeth Joined in the Middle

15.6.2 Intersecting Shaft

Bevel Gears

Bevel gears are primarily used to transfer power between intersecting shafts. The teeth of these gears are formed on a conical surface. Standard bevel gears have teeth which are cut straight and are all parallel to the line pointing the apex of the cone on which the teeth are based as shown in Figure 15.21. Spiral bevel gears are also available with teeth form arcs. Hypocycloid bevel gears are a special type of spiral gear that will allow non-intersecting, non-parallel shafts to mesh. Straight tool bevel gears are generally considered the best choice for systems with lower speeds. They become noisy above this point. One of the most common applications of bevel gears is the differential in automobiles.

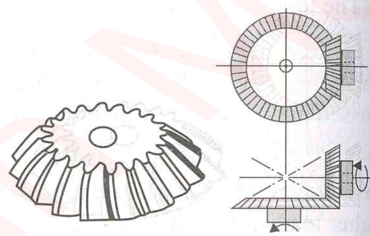


Figure 15.21 Bevel Gears

Limitations: It cannot be used for parallel shafts and becomes noisy at high speeds.

Advantages: It is an excellent choice for intersecting shaft systems.

Hypoid Gears

Hypoid gears resemble spiral bevels, but the axes of the pinion shaft and gear shaft do not intersect (Figure 15.22). This configuration allows both shafts to be supported at both ends. In hypoid gears, the meshing point of the pinion with the driven gear is about midway between the central position of a pinion in a spiral bevel and the extreme top or bottom position of a worm. This geometry

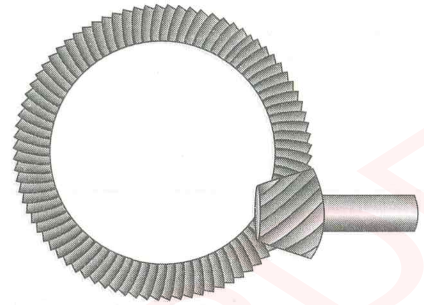


Figure 15.22 Hypoid Gear

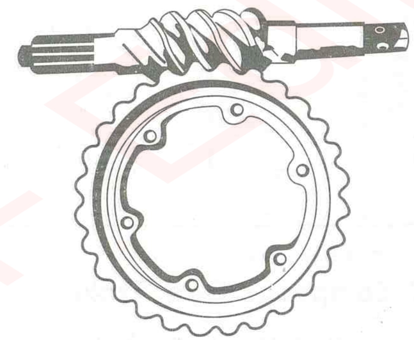


Figure 15.23 Worm Gear

Worm Gears

If a tooth of a helical gear makes complete revolutions on the pitch cylinder, the resulting gear is known as worm. The mating gear is called worm wheel as shown in Figure 15.23. Worm may be single start, double start, or triple start.

15.7 GEAR TRAINS

Gear trains are various types of combination of gears to transfer power from one shaft to another. Gear trains can be classified as follows:

- Simple gear train.
- Compound gear train.
- Reverted gear train.
- Planetary gear train.
- Sun and planet gear.

15.7.1 Simple Gear Train

In this gear train, all the gears are mounted on their separate shafts and the gear axes remain fixed in a frame. All the paired gears are moved in opposite directions. All the gears can be in straight line or in zig-zag manner. Figure 15.24 shows the example of simple gear train.

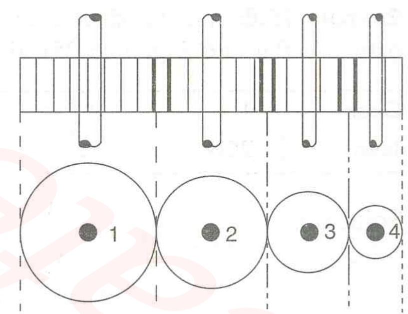


Figure 15.24 Simple Gear Train

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

Example 15.7: There are four gears mounted on parallel shafts. The speed of driver gear is 1,000 rpm.

Solution:

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

$$\Rightarrow\quad N_4=N_1\times\frac{T_1}{T_4}=1{,}000\times\frac{80}{50}=1{,}600\,\text{rpm}$$

15.7.2 Compound Gear Train

When two or more gears rotate about same axis and have same angular velocity, it is known as compound gear train. Figure 15.25 shows the example of compound gear train.

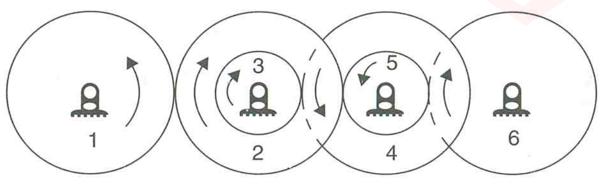


Figure 15.25 Compound Gear Train

$$\frac{N_2}{N_1}\times\frac{N_4}{N_2}\times\frac{N_6}{N_4}=\frac{T_1}{T_2}\times\frac{T_3}{T_4}\times\frac{T_5}{T_6}\quad\text{or}\quad\frac{N_6}{N_1}=\frac{T_1\times T_3\times T_5}{T_2\times T_4\times T_6}$$

i.e.,

$$\text{Train value}=\frac{\text{Product of number of teeth on driving gears}}{\text{Product of number of teeth on driven gears}}$$

Example 15.8: A motor shaft is connected to gear 1 which rotates at 50 rpm. Find the speed of output shaft gear 6 (Figure 15.25). The number of teeth on each gear is given in the following table:

Gear	1	2	3	4	5	6
Teeth	75	30	60	25	40	20

Solution:

$$\frac{N_6}{N_1}=\frac{T_1\times T_3\times T_5}{T_2\times T_4\times T_6}=\frac{75\times60\times40}{30\times25\times20}=12$$

$$N_6=12\times N_1=12\times50=600\text{ rpm}$$

15.7.3 Reverted Gear Train

If axes of the first and the last wheels of a compound gear coincide, it is called reverted gear train.

Example 15.9: In a reverted gear train, speed ratio is 10 as shown in Figure 15.26. The module of gear 1 and 4 is 3 mm and module of gear 2 and 3 is 2 mm. Calculate the number of teeth on gears.

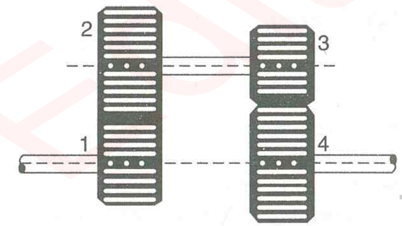


Figure 15.26 Reverted Gear Train

Solution:

$$\frac{N_1}{N_4}\times\frac{N_3}{N_2}=10$$

But $$\frac{N_1}{N_4}=\frac{N_3}{N_2}=\sqrt{10}$$

The speed radio is inverse of teeth ratio

i.e., $$\frac{T_4}{T_1}=\frac{T_2}{T_3}=\sqrt{10}=3.1622$$

Distance between shaft is 180 mm

$$C=r_1+r_4=r_2+r_3=180\text{ mm}$$

$$\frac{m_1T_1}{2}+\frac{m_4T_4}{2}=\frac{m_2T_2}{2}+\frac{m_3T_3}{2}=180\,\text{mm}$$

$$3\,(T_1+T_4)=2\,(T_2+T_3)=180\text{ mm}$$

$$T_1+T_4=60$$

$$T_2+T_3=90$$

15.7.4 Planetary or Epicyclic Gear Train

If the axis of at least one gear in gear train moves relative to fixed axis or frame, such type of gear train is known as epicyclic gear train. In this gear train, one gear rotates over pitch circle of other gear as shown in Figure 15.27. Consider two gear wheels P and Q, the axes of which are connected by an arm 'a' is fixed. The wheels P and Q constitute a simple gear train.

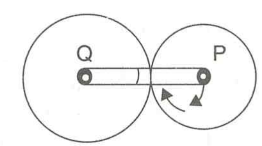
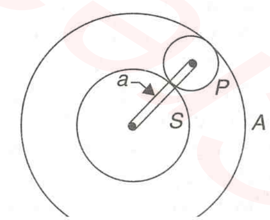


Figure 15.27 Epicyclic Gear Train

However, if the wheel Q is fixed so that the arm can rotate about the axes of Q, the wheel P would also move around Q. Thus, it is epicyclic gear train.

15.7.5 Sun and Planet Gear



When an annular gear A is used in epicyclic gear train, it is known as sun and planet gear train. The annular wheel meshes with wheel P and

SUMMARY

The chapter can be summarized as follows:

- Power transmission is a process to transmit motion from one shaft to another by using some connection between them like belt, rope, chain, and gears.
- Belt, rope, and chain are flexible connectors where gears are rigid connectors.
- In belt drive, the velocity of two shafts can be varied by variation of the diameter of pulley on which the belt is mounted.
- In chain or gear drive, the velocity of two shafts is varied by variation in number of teeth on sprocket and gear, respectively.
- The effect of slip is a decrease in the speed of belt on driving shaft and then driven shaft.
- When belt passes from slack to tight side, a certain portion of belt extends and again contracts when belt passes through tight to slack side. Due to fluctuation in length of the belt, there is relative motion between belt and pulley surface. This relative motion is known as creep.
- Open-belt drive is used to provide same direction of rotation to driven shaft as the direction of rotation of driving shaft.
- Cross-belt drive is used to provide reverse direction of rotation to driven shaft as the direction of rotation of driving shaft.
- When velocity of the belt is more than 10 m/s, the centrifugal force due to self-weight of the belt becomes predominant.
- To overcome the problem of slip in belt drive or rope drive, chain drive is used.
- Gears are compact power transmission device that controls the speed, torque, and direction of rotation of driven shaft.
- The point of contact between pitch circles of two gears is known as pitch point.
- The circle passing through point of contacts of two gears is known as pitch circle.
- Diameter of pitch circle is known as pitch diameter.
- Circular pitch is the distance measured along the circumference of the pitch circle from a point on one tooth of the corresponding point on the adjacent tooth.
- Diametral pitch is the number of teeth per unit length of the pitch circle diameter.
- Module is the ratio of pitch diameter to the number of teeth.
- *Face* is the tooth surface between the pitch circle and the top land.
- *Flank* is the curved portion of the tooth flank at the root circle.
- The angle between the pressure line and the common tangent at the pitch point is known as the pressure angle or angle of obliquity.
- Law of gearing: If angular velocities of two mating gears remain constant, the common normal at the point of the two teeth should always pass through a fixed point P which divides the line joining the centres in the inverse ratio of angular velocities of the gears.
- A cycloid is the locus of points on the circumference of a circle that rolls without slipping on a fixed straight line.
- An epicycloid is the locus of points on the circumference of a circle that rolls without slipping outside the circumference of another circle.
- A hypocycloid is the locus of points on the circumference of a circle that rolls without slipping inside the circumference of another circle.
- An involutes profile is a plane curve generated by the points on tangent on a circle which rolls without slipping or by points on a taught string which is unwrapped from a reel.
- Spur gears are the most commonly used gear. They are characterized by teeth which are parallel to the axis.
- Helical gears are similar to the spur gear except that the teeth are at an angle to the shaft, rather than parallel to its axis as in a spur gear.

- Hypoid gears resemble spiral bevels, but the axes of the pinion shaft and gear shaft do not intersect.
- If a tooth of a helical gear makes complete revolutions on the pitch cylinder, the resulting gear is known as worm. The mating gear is called worm wheel.
- In a simple gear train, all the gears are mounted on their separate shafts and the gear axes remain fixed in a frame. All the paired gears are moved in opposite directions.
- When two or more gears rotate about same axis and have same angular velocity, it is known as compound gear train.
- If axes of the first and the last wheels of a compound gears coincide, it is called reverted gear train.
- If the axis of at least one gear in gear train moves relative to fixed axis or frame, such type of gear train is known as epicyclic gear train.
- When an annular gear A is used in epicyclic gear train, it is known as sun and planet gear train.

MULTIPLE CHOICE QUESTIONS

1. In a belt drive pulley acts as
   (a) sliding pair (b) rolling pair
   (c) turning pair (d) none of these
2. When two pulleys are connected by a cross-belt drive, then both the pulleys rotate in
   (a) same direction (b) opposite direction
   (c) not necessary (d) none of these
3. Length of open belt connecting two pulleys of radii r_1 and r_2 and at a centre distance D apart, is
   (a) $\pi(r_1+r_2)+(r_1-r_2)^2/D+2D$
   (b) $\pi(r_1+r_2)+(r_1+r_2)^2/D+2D$
   (c) $\pi(r_1-r_2)+(r_1+r_2)^2/D+2D$
   (d) $\pi(r_1+r_2)+(r_1-r_2)^2/D+2D$
4. Length of cross belt connecting two pulleys of radii r_1 and r_2 and at a centre distance D apart, is
   (a) $\pi(r_1+r_2)+(r_1-r_2)^2/D+2D$
   (b) $\pi(r_1+r_2)+(r_1+r_2)^2/D+2D$
   (c) $\pi(r_1-r_2)+(r_1+r_2)^2/D+2D$
   (d) $\pi(r_1+r_2)+(r_1-r_2)^2/D+2D$
5. Angle of contact in cross-belt drive in comparison to open-belt drive is
   (a) more (b) less
   (c) same (d) none
6. Slip in belt drive is difference between
   (a) angular velocities between two pulleys
   (b) linear speed of the rim of pulleys and the belt on it
   (c) the velocities of two pulleys
   (d) none of these
7. In belt drives, effect of centrifugal tension is
   (a) to increase the driving power
8. If T_1 and T_2 are tensions on tight and slack side of belt, θ is angle of contact and μ is coefficient of friction between belt and pulley, then ratio of tension is
   (a) $T_1/T_2=\mu\theta$ (b) $T_1/T_2=e^{\mu\theta}$
   (c) $T_1/T_2=e\mu\theta$ (d) $T_1/T_2=e^{1/\mu}\theta$
9. For maximum power transmission, the maximum tension T_{max} in the belt is equal to
   (a) T_c (b) $2T_c$
   (c) $3T_c$ (d) $T_c/3$
10. Creep in belt is due to
    (a) elasticity of belt material
    (b) elongation of belt due to tension
    (c) differential elongation of belt due to difference in tension on two sides of a pulley
    (d) plasticity of belt material
11. Included angle of V-belt is generally
    (a) 10° to 20° (b) 20° to 30°
    (c) 30° to 40° (d) 50° to 60°
12. In designation 6 by 19 rope, 6 and 19, respectively, stand for
    (a) diameter of wire rope and number of strands
    (b) diameter of wire rope and number of wires
    (c) number of wires and number of strands
    (d) number of strands and number of wires
13. A chain drive is used for
    (a) short distance
    (b) medium distance
    (c) long distance
    (d) distance is no barrier

(c) three or more roller chains
(d) inverted tooth and overlapping links

15. Wire ropes are used for
(a) low speeds and low tension
(b) low speeds and high tension
(c) high speeds and low tension
(d) none of these

16. The ratio of the number of teeth and pitch circle diameter is called
(a) pitch (b) circular pitch
(c) diametral pitch (d) module

17. The circle passing through the bottom of the teeth of gear is known as
(a) inner circle (b) base circle
(c) addendum circle (d) dedendum circle

18. The circle passing through the top of the teeth of gear is known as
(a) inner circle (b) base circle
(c) addendum circle (d) dedendum circle

19. Pitch circle diameter of an involute gear is
(a) independent of any other factor
(b) dependent on pressure angle
(c) constant for set of meshing gears
(d) proportional to base diameter

20. The surface of the gear below the pitch circle is called
(a) face (b) flank
(c) bottom tooth (d) tooth depth

21. Law of the gearing is satisfied if
(a) two surfaces slide smoothly
(b) common normal at the point of contact passes through pitch point on the line joining the centres of rotation
(c) addendum is greater than dedendum
(d) none of these

Answers

1. (b), 2. (b), 3. (a), 4. (b), 5. (a), 6. (b), 7. (c), 8. (b), 9. (c), 10. (c), 11. (c), 12. (d), 13. (a), 14. (d), 15. (b), 16. (c), 17. (d), 18. (c), 19. (b), 20. (b), 21. (b)

FILL IN THE BLANKS

1. The gear train in which the first and last gear are on the same axis is known as ________.
2. The difference between dedendum and addendum is known as ________.
3. Best profile to obtain resistance against wear is ________.
4. The product of circular pitch and dimetral pitch is equal to ________.

Answers

1. Reverted gear train, 2. Clearance, 3. 14½° full depth involute, 4. Π.

REVIEW QUESTIONS

1. In a flat bet drive prove that $\frac{T_1}{T_2} = e^{\mu\theta}$;

where T_1 is tension in tight side, T_2 is tension in slack side, μ is coefficient of friction, θ is angle of lapin radian.

2. What is initial tension in belt? Explain the effect of centrifugal tension in belt drive.

5. Find the expression for the length of belt in cross-belt drive (Refer Figure 15.25).
6. What are the various types of chain drive? Explain with neat sketches.
7. What are the relative merits and demerits of belt, rope, and chain drive?
8. State the law of gearing.
9. Define: (i) module, (ii) pressure angle, (iii) pitch point, (iv) addendum, (v) dedendum, (vi) flank, (vii) face, (viii) circular pitch, (ix) dimetral pitch, and (x) pitch circle.
10. Differentiate involute and cycloidal profiles of gear teeth.
11. Classify the gears and explain them with neat sketches.
12. Explain the phenomena interference and undercutting in gear drive.

PROBLEMS FOR PRACTICE

1. The speed of a driving shaft is 80 rpm and the speed of driven shaft is 120 rpm. Diameter of the driving pulley is given as 600 mm. Find the diameter of driven pulley in the following cases:
(a) If belt thickness is negligible
(b) If belt thickness is 5 mm
(c) If total slip is 10% (considering thickness of belt)
(d) If a slip of 2% on each pulley (considering thickness of belt)

2. Two shafts are arranged parallel to each other at a distance of 8 m. If the pulleys' diameters mounted on the shafts are of 600 and 1,000 mm, find the ratio of length of belts for open and cross-belt drives.

3. A leather belt of density 1,000 kg/m^3, thickness of 10 mm is used to transmit a power of 10 kW from a pulley of diameter 1.2 m and running at 250 rpm. Determine the width of the belt taking centrifugal tension into account. If the angle of lap is 160° and coefficient of friction between belt and pulley is 0.25. Assuming allowable stress for leather belt is 1.5 MPa.

4. An open-belt drive transmits a power of 5 kW. The linear velocity of the belt is 8 m/s. Angle of lap on smaller pulley is 165°. The coefficient of friction between belt and pulley is 0.25. Determine the effect of the following on power transmission:
(a) Initial tension in belt is increased by 5%.
(b) Angle of lap is increased by 5% using idler pulley for same speed and tension in tight side.
(c) Coefficient of friction is increased by 5%.

5. The gear 1, mounted on motor shaft, rotates at 600 rpm. Find the speed of gear 6 mounted on output shaft. The number of teeth on each gear is given below. (Refer Figure 15.25).

Gear	1	2	3	4	5	6
Teeth	20	30	60	40	80	100