21

Chain Drives

21.1 Introduction

We have seen in previous chapters on belt and rope drives that slipping may occur. In order to avoid slipping, steel chains are used. The chains are made up of number of rigid links which are hinged together by pin joints in order to provide the necessary flexibility for wraping round the driving and driven wheels. These wheels have projecting teeth of special profile and fit into the corresponding recesses in the links of the chain as shown in Fig. 21.1. The toothed wheels are known as *sprocket wheels or simply sprockets*. The sprockets and the chain are thus constrained to move together without slipping and ensures perfect velocity ratio.

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* These wheels resemble to spur gears.

Hinge — — Chain link
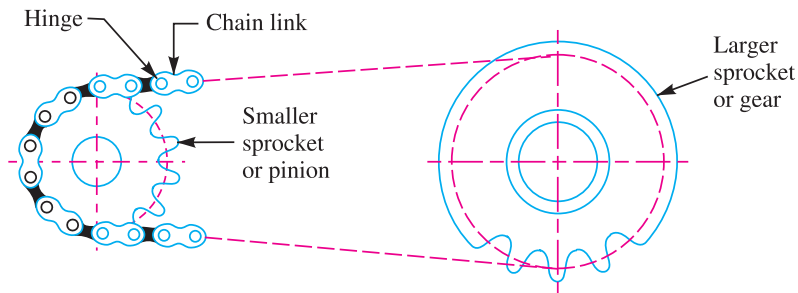Larger sprocket or gear
Smaller sprocket or pinion

Fig. 21.1. Sprockets and chain.

The chains are mostly used to transmit motion and power from one shaft to another, when the centre distance between their shafts is short such as in bicycles, motor cycles, agricultural machinery, conveyors, rolling mills, road rollers etc. The chains may also be used for long centre distance of upto 8 metres. The chains are used for velocities up to 25 m / s and for power upto 110 kW. In some cases, higher power transmission is also possible.

21.2 Advantages and Disadvantages of Chain Drive over Belt or Rope Drive

Following are the advantages and disadvantages of chain drive over belt or rope drive:

Advantages

1. As no slip takes place during chain drive, hence perfect velocity ratio is obtained.
2. Since the chains are made of metal, therefore they occupy less space in width than a belt or rope drive.
3. It may be used for both long as well as short distances.
4. It gives a high transmission efficiency (upto 98 percent).
5. It gives less load on the shafts.
6. It has the ability to transmit motion to several shafts by one chain only.
7. It transmits more power than belts.
8. It permits high speed ratio of 8 to 10 in one step.
9. It can be operated under adverse temperature and atmospheric conditions.

Disadvantages

1. The production cost of chains is relatively high.
2. The chain drive needs accurate mounting and careful maintenance, particularly lubrication and slack adjustment.
3. The chain drive has velocity fluctuations especially when unduly stretched.

Sports bicycle gear and chain drive mechanism

21.3 Terms Used in Chain Drive

The following terms are frequently used in chain drive.

1. *Pitch of chain.* It is the distance between the hinge centre of a link and the corresponding hinge centre of the adjacent link, as shown in Fig. 21.2. It is usually denoted by p.

Fig. 21.2. Terms used in chain drive.

2. *Pitch circle diameter of chain sprocket.* It is the diameter of the circle on which the hinge centres of the chain lie, when the chain is wrapped round a sprocket as shown in Fig. 21.2. The points A, B, C, and D are the hinge centres of the chain and the circle drawn through these centres is called pitch circle and its diameter (D) is known as pitch circle diameter.

21.4 Relation Between Pitch and Pitch Circle Diameter

A chain wrapped round the sprocket is shown in Fig. 21.2. Since the links of the chain are rigid, therefore pitch of the chain does not lie on the arc of the pitch circle. The pitch length becomes a chord. Consider one pitch length AB of the chain subtending an angle θ at the centre of sprocket (or pitch circle),

Let D = Diameter of the pitch circle, and

T = Number of teeth on the sprocket.

From Fig. 21.2, we find that pitch of the chain,

$$p = AB = 2\,A\,O \sin\left(\frac{\theta}{2}\right) = 2 \times \left(\frac{D}{2}\right) \sin\left(\frac{\theta}{2}\right) = D \sin\left(\frac{\theta}{2}\right)$$

We know that $\theta = \dfrac{360°}{T}$

$\therefore$ $$p = D \sin\left(\frac{360°}{2\,T}\right) = D \sin\left(\frac{180°}{T}\right)$$

or $$D = p \operatorname{cosec}\left(\frac{180°}{T}\right)$$

The sprocket outside diameter (D_o), for satisfactory operation is given by

$$D_o = D + 0.8\, d_1$$

where d_1 = Diameter of the chain roller.

Note: The angle $\theta/2$ through which the link swings as it enters contact is called **angle of articulation.**

21.5 Velocity Ratio of Chain Drives

The velocity ratio of a chain drive is given by

$$V.R. = \frac{N_1}{N_2} = \frac{T_2}{T_1}$$

where
N_1 = Speed of rotation of smaller sprocket in r.p.m.,
N_2 = Speed of rotation of larger sprocket in r.p.m.,
T_1 = Number of teeth on the smaller sprocket, and
T_2 = Number of teeth on the larger sprocket.

The average velocity of the chain is given by

$$v = \frac{\pi \, D \, N}{60} = \frac{T \, p \, N}{60}$$

where
D = Pitch circle diameter of the sprocket in metres, and
p = Pitch of the chain in metres.

21.6 Length of Chain and Centre Distance

An open chain drive system connecting the two sprockets is shown in Fig. 21.3.
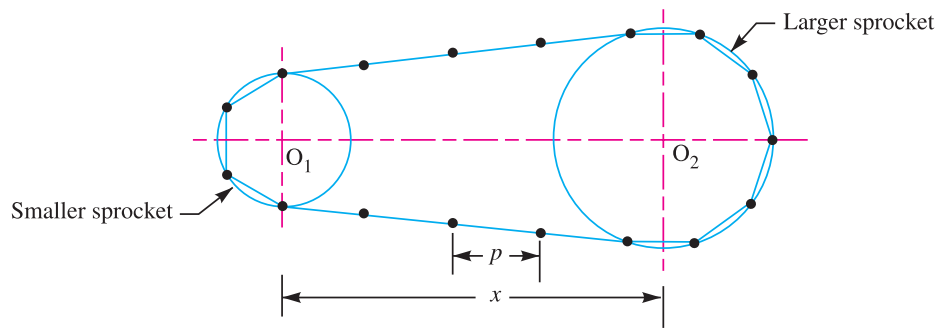
Fig. 21.3. Length of chain.

Let
T_1 = Number of teeth on the smaller sprocket,
T_2 = Number of teeth on the larger sprocket,
p = Pitch of the chain, and
x = Centre distance.

The length of the chain (L) must be equal to the product of the number of chain links (K) and the pitch of the chain (p). Mathematically,

$$L = K.p$$

The number of chain links may be obtained from the following expression, *i.e.*

$$K = \frac{T_1 + T_2}{2} + \frac{2 \, x}{p} + \left[\frac{T_2 - T_1}{2 \, \pi}\right]^2 \frac{p}{x}$$

The value of K as obtained from the above expression must be approximated to the nearest even number.

The centre distance is given by

$$x = \frac{p}{4}\left[K - \frac{T_1 + T_2}{2} + \sqrt{\left(K - \frac{T_1 + T_2}{2}\right)^2 - 8\left(\frac{T_2 - T_1}{2 \, \pi}\right)^2}\right]$$

In order to accommodate initial sag in the chain, the value of the centre distance obtained from the above equation should be decreased by 2 to 5 mm.

Notes: 1. The minimum centre distance for the velocity transmission ratio of 3, may be taken as

$$x_{min} = \frac{d_1 + d_2}{2} + 30 \text{ to } 50 \text{ } mm$$

where d_1 and d_2 are the diameters of the pitch circles of the smaller and larger sprockets.

2. For best results, the minimum centre distance should be 30 to 50 times the pitch.

3. The minimum centre distance is selected depending upon the velocity ratio so that the arc of contact of the chain on the smaller sprocket is not less than 120º. It may be noted that larger angle of arc of contact ensures a more uniform distribution of load on the sprocket teeth and better conditions of engagement.

21.7 Classification of Chains

The chains, on the basis of their use, are classified into the following three groups:

1. Hoisting and hauling (or crane) chains,

2. Conveyor (or tractive) chains, and

3. Power transmitting (or driving) chains.

These chains are discussed, in detail, in the following pages.

21.8 Hoisting and Hauling Chains

These chains are used for hoisting and hauling purposes and operate at a maximum velocity of 0.25 m / s. The hoisting and hauling chains are of the following two types:

1. *Chain with oval links*. The links of this type of chain are of oval shape, as shown in Fig. 21.4 (*a*). The joint of each link is welded. The sprockets which are used for this type of chain have receptacles to receive the links. Such type of chains are used only at low speeds such as in chain hoists and in anchors for marine works.
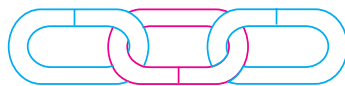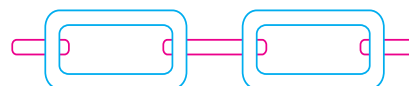
(*a*) Chain with oval links.   (*b*) Chain with square links.

Fig. 21.4. Hoisting and hauling chains.

2. *Chain with square links*. The links of this type of chain are of square shape, as shown in Fig. 21.4 (*b*). Such type of chains are used in hoists, cranes, dredges. The manufacturing cost of this type of chain is less than that of chain with oval links, but in these chains, the kinking occurs easily on overloading.
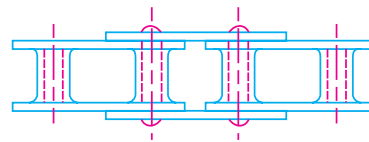
21.9 Conveyor Chains

These chains are used for elevating and conveying the materials continuously at a speed upto 2 m / s. The conveyor chains are of the following two types:

1. Detachable or hook joint type chain, as shown in Fig. 21.5 (*a*), and

2. Closed joint type chain, as shown in Fig. 21.5 (*b*).
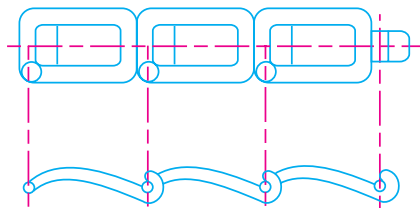
(*a*) Detachable or hook joint type chain.   (*b*) Closed joint type chain.

Fig. 21.5. Conveyor chains.

The conveyor chains are usually made of malleable cast iron. These chains do not have smooth running qualities. The conveyor chains run at slow speeds of about 0.8 to 3 m / s.

21.10 Power Transmitting Chains

These chains are used for transmission of power, when the distance between the centres of shafts is short. These chains have provision for efficient lubrication. The power transmitting chains are of the following three types.

1. *Block or bush chain.* A block or bush chain is shown in Fig. 21.6. This type of chain was used in the early stages of development in the power transmission.
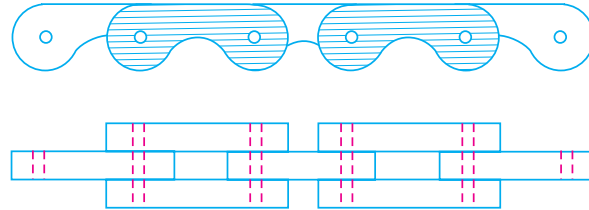

Fig. 21.6. Block or bush chain.

It produces noise when approaching or leaving the teeth of the sprocket because of rubbing between the teeth and the links. Such type of chains are used to some extent as conveyor chain at small speed.

2. *Bush roller chain.* A bush roller chain as shown in Fig. 21.7, consists of outer plates or pin link plates, inner plates or roller link plates, pins, bushes and rollers. A pin passes through the bush which is secured in the holes of the roller between the two sides of the chain. The rollers are free to rotate on the bush which protect the sprocket wheel teeth against wear. The pins, bushes and rollers are made of alloy steel.
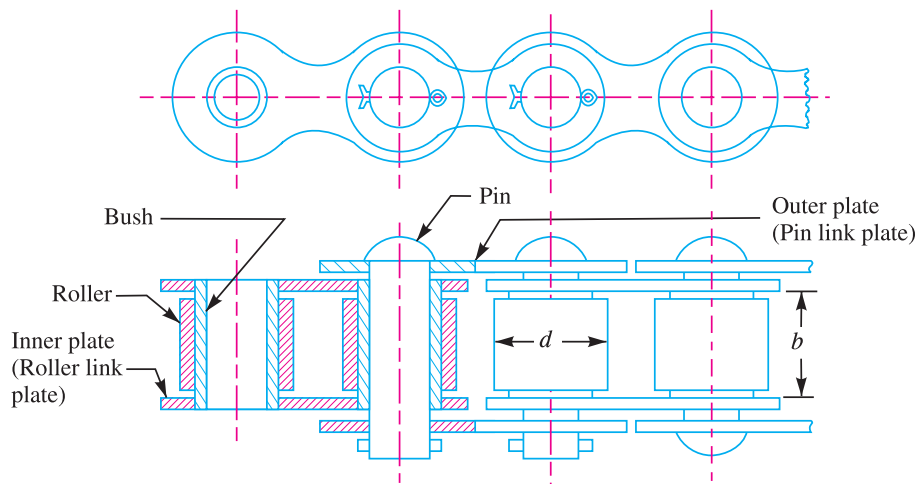

Fig. 21.7. Bush roller chain.

A bush roller chain is extremely strong and simple in construction. It gives good service under severe conditions. There is a little noise with this chain which is due to impact of the rollers on the sprocket wheel teeth. This chain may be used where there is a little lubrication. When one of these chains elongates slightly due to wear and stretching of the parts, then the extended chain is of greater pitch than the pitch of the sprocket wheel teeth. The rollers then fit unequally into the cavities of the wheel. The result is that the total load falls on one teeth or on a few teeth. The stretching of the parts increase wear of the surfaces of the roller and of the sprocket wheel teeth.

Rear wheel chain drive of a motorcycle

The roller chains are standardised and manufactured on the basis of pitch. These chains are available in single-row or multi-row roller chains such as simple, duplex or triplex strands, as shown in Fig. 21.8.
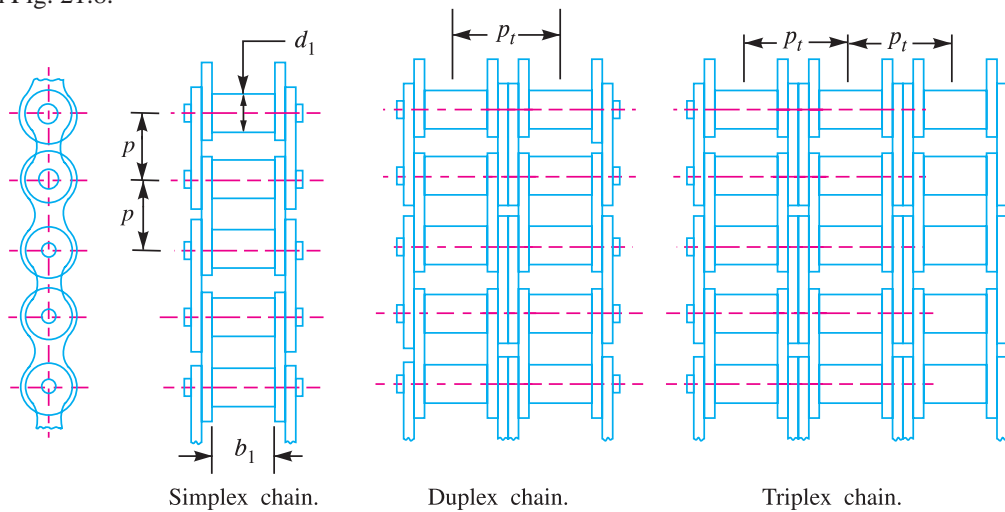

Simplex chain.   Duplex chain.   Triplex chain.

Fig. 21.8. Types of roller chain.

3. Silent chain. A silent chain (also known as *inverted tooth chain*) is shown in Fig. 21.9.
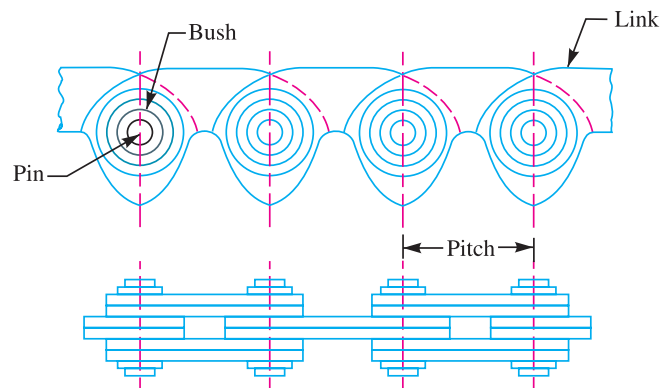

Fig. 21.9. Silent chain.

It is designed to eliminate the evil effects caused by stretching and to produce noiseless running. When the chain stretches and the pitch of the chain increases, the links ride on the teeth of the sprocket wheel at a slightly increased radius. This automatically corrects the small change in the pitch. There is no relative sliding between the teeth of the inverted tooth chain and the sprocket wheel teeth. When properly lubricated, this chain gives durable service and runs very smoothly and quietly.

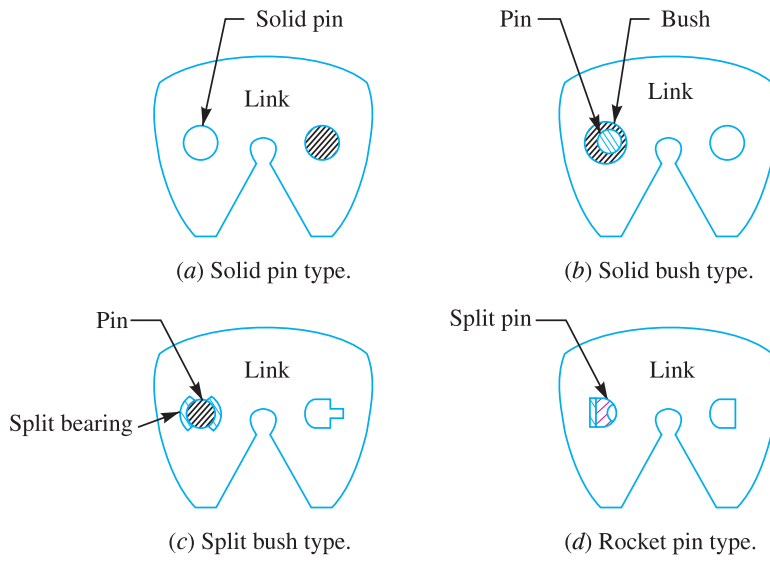The various types of joints used in a silent chain are shown in Fig. 21.10.

(*a*) Solid pin type.   (*b*) Solid bush type.

(*c*) Split bush type.   (*d*) Rocket pin type.

Fig. 21.10. Silent chain joints.

21.11 Characteristics of Roller Chains

According to Indian Standards (IS: 2403 —1991), the various characteristics such as pitch, roller diameter, width between inner plates, transverse pitch and breaking load for the roller chains are given in the following table.

Table 21.1. Characteristics of roller chains according to IS: 2403 — 1991.

ISO Chain number	Pitch (p) mm	Roller diameter (d_1) mm Maximum	Width between inner plates (b_1) mm Maximum	Transverse pitch (p_1)mm	Breaking load (kN) Minimum		
					Simple	Duplex	Triplex
05 B	8.00	5.00	3.00	5.64	4.4	7.8	11.1
06 B	9.525	6.35	5.72	10.24	8.9	16.9	24.9
08 B	12.70	8.51	7.75	13.92	17.8	31.1	44.5
10 B	15.875	10.16	9.65	16.59	22.2	44.5	66.7
12 B	19.05	12.07	11.68	19.46	28.9	57.8	86.7
16 B	25.4	15.88	17.02	31.88	42.3	84.5	126.8
20 B	31.75	19.05	19.56	36.45	64.5	129	193.5
24 B	38.10	25.40	25.40	48.36	97.9	195.7	293.6
28 B	44.45	27.94	30.99	59.56	129	258	387
32 B	50.80	29.21	30.99	68.55	169	338	507.10
40 B	63.50	39.37	38.10	72.29	262.4	524.9	787.3
48 B	76.20	48.26	45.72	91.21	400.3	800.7	1201

21.12 Factor of Safety for Chain Drives

The factor of safety for chain drives is defined as the ratio of the breaking strength (W_B) of the chain to the total load on the driving side of the chain (W). Mathematically,

$$\text{Factor of safety} = \frac{W_B}{W}$$

The breaking strength of the chain may be obtained by the following empirical relations, *i.e.*

$$W_B = 106 \, p^2 \text{ (in newtons) for roller chains}$$
$$= 106 \, p \text{ (in newtons) per mm width of chain for silent chains.}$$

where p is the pitch in mm.

The total load (or total tension) on the driving side of the chain is the sum of the tangential driving force (F_T), centrifugal tension in the chain (F_C) and the tension in the chain due to sagging (F_S).

We know that the tangential driving force acting on the chain,

$$F_T = \frac{\text{Power transmitted (in watts)}}{\text{Speed of chain in m/s}} = \frac{P}{v} \text{ (in newtons)}$$

Centrifugal tension in the chain,

$$F_C = m.v^2 \text{ (in newtons)}$$

and tension in the chain due to sagging,

$$F_S = k.mg.x \text{ (in newtons)}$$

where
$m =$ Mass of the chain in kg per metre length,

$x =$ Centre distance in metres, and

$k =$ Constant which takes into account the arrangement of chain drive

$= 2$ to 6, when the centre line of the chain is inclined to the horizontal at an angle less than 40º

$= 1$ to 1.5, when the centre line of the chain is inclined to the horizontal at an angle greater than 40º.

The following table shows the factor of safety for the bush roller and silent chains depending upon the speed of the sprocket pinion in r.p.m. and pitch of the chains.

Table 21.2. Factor of safety (*n*) for bush roller and silent chains.

Type of chain	Pitch of chain (mm)	Speed of the sprocket pinion in r.p.m.								
		50	200	400	600	800	1000	1200	1600	2000
Bush roller chain	12 – 15	7	7.8	8.55	9.35	10.2	11	11.7	13.2	14.8
	20 – 25	7	8.2	9.35	10.3	11.7	12.9	14	16.3	–
	30 – 35	7	8.55	10.2	13.2	14.8	16.3	19.5	–	–
Silent chain	12.7 – 15.87	20	22.2	24.4	28.7	29.0	31.0	33.4	37.8	42.0
	19.05 – 25.4	20	23.4	26.7	30.0	33.4	36.8	40.0	46.5	53.5

21.13 Permissible Speed of Smaller Sprocket

The following table shows the permissible speed of the smaller sprocket or pinion (in r.p.m.) for the bush roller and silent chain corresponding to different pitches.

Common bicycle is the best example of a chain drive

Table 21.3. Permissible speed of smaller sprocket or pinion in r.p.m.

Type of Chain	Number of teeth on sprocket pinion	Pitch of chain (p) in mm				
		12	15	20	25	30
Bush roller chain	15	2300	1900	1350	1150	1000
	19	2400	2000	1450	1200	1050
	23	2500	2100	1500	1250	1100
	27	2550	2150	1550	1300	1100
	30	2600	2200	1550	1300	1100
Silent chain	17 – 35	3300	2650	2200	1650	1300

Note: The chain velocity for the roller chains may be as high as 20 m / s, if the chains are properly lubricated and enclosed, whereas the silent chain may be operated upto 40 m / s.

21.14 Power Transmitted by Chains

The power transmitted by the chain on the basis of breaking load is given by

$$P = \frac{W_B \times v}{n \times K_S} \quad \text{(in watts)}$$

where
W_b = Breaking load in newtons,
v = Velocity of chain in m/s
n = Factor of safety, and
K_S = Service factor = $K_1.K_2.K_3$

The power transmitted by the chain on the basis of bearing stress is given by

$$P = \frac{\sigma_b \times A \times v}{K_S}$$

where
σ_b = Allowable bearing stress in MPa or N/mm^2,
A = Projected bearing area in mm^2,
v = Velocity of chain in m/s, and
K_S = Service factor.

The power rating for simple roller chains depending upon the speed of the smaller sprocket is shown in the following table.

Table 21.4. Power rating (in kW) of simple roller chain.

Speed of smaller sprocket or pinion (r.p.m.)	Power (kW)				
	06 B	08 B	10 B	12 B	16 B
100	0.25	0.64	1.18	2.01	4.83
200	0.47	1.18	2.19	3.75	8.94
300	0.61	1.70	3.15	5.43	13.06
500	1.09	2.72	5.01	8.53	20.57
700	1.48	3.66	6.71	11.63	27.73
1000	2.03	5.09	8.97	15.65	34.89
1400	2.73	6.81	11.67	18.15	38.47
1800	3.44	8.10	13.03	19.85	–
2000	3.80	8.67	13.49	20.57	–

The service factor (K_S) is the product of various factors, such as load factor (K_1), lubrication factor (K_2) and rating factor (K_3). The values of these factors are taken as follows:

1. Load factor (K_1)   = 1, for constant load
                    = 1.25, for variable load with mild shock
                    = 1.5, for heavy shock loads

2. Lubrication factor (K_2) = 0.8, for continuous lubrication
                    = 1, for drop lubrication
                    = 1.5, for periodic lubrication

3. Rating factor (K_3)   = 1, for 8 hours per day
                    = 1.25, for 16 hours per day
                    = 1.5, for continuous service

21.15 Number of Teeth on the Smaller or Driving Sprocket or Pinion

Consider an arrangement of a chain drive in which the smaller or driving sprocket has only four teeth, as shown in Fig. 21.11 (a). Let the sprocket rotates anticlockwise at a constant speed of N r.p.m. The chain link AB is at a distance of d / 2 from the centre of the sprocket and its linear speed is given by
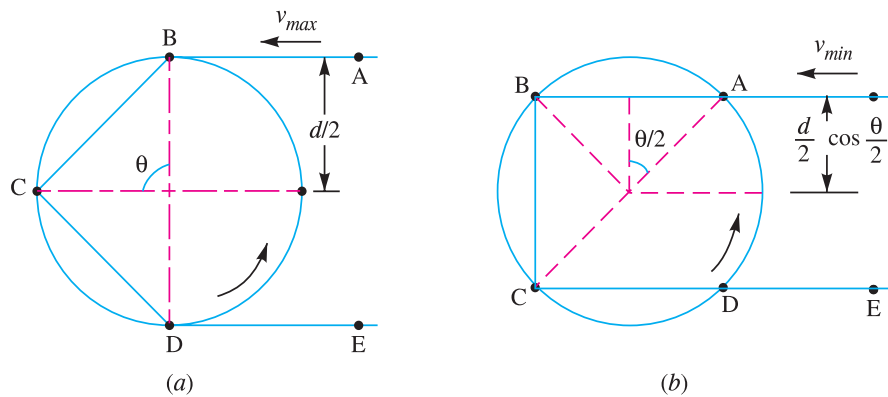

Fig. 21.11. Number of teeth on the smaller sprocket.

$$v_{max} = \frac{\pi \, d \, N}{60} \quad m/s$$

where d = Pitch circle diameter of the smaller or driving sprocket in metres.

When the sprocket rotates through an angle $\theta/2$, the link AB occupies the position as shown in Fig. 21.11 (b). From the figure, we see that the link is now at a distance of $\left(\dfrac{d}{2} \times \cos \dfrac{\theta}{2} \right)$ from the centre of the sprocket and its linear velocity is given by

$$v_{min} = \frac{\pi \, d \, N \cos \theta / 2}{60} \quad m/s$$

From above, we see that the linear velocity of the sprocket is not uniform but varies from maximum to minimum during every cycle of tooth engagement. This results in fluctuations in chain transmission and may be minimised by reducing the angle θ or by increasing the number of teeth on the sprocket. It has been observed that for a sprocket having 11 teeth, the variation of speed is 4 percent and for the sprockets having 17 teeth and 24 teeth, the variation of speed is 1.6 percent and 1 percent respectively.

In order to have smooth operation, the minimum number of teeth on the smaller sprocket or pinion may be taken as 17 for moderate speeds and 21 for high speeds. The following table shows the number of teeth on a smaller sprocket for different velocity ratios.

Table 21.5. Number of teeth on the smaller sprocket.

Type of chain	Number of teeth at velocity ratio					
	1	2	3	4	5	6
Roller	31	27	25	23	21	17
Silent	40	35	31	27	23	19

Note: The number of teeth on the smaller sprocket plays an important role in deciding the performance of a chain drive. A small number of teeth tends to make the drive noisy. A large number of teeth makes chain pitch smaller which is favourable for keeping the drive silent and reducing shock, centrifugal force and friction force.

21.16 Maximum Speed for Chains

The maximum allowable speed for the roller and silent chains, depending upon the number of teeth on the smaller sprocket or pinion and the chain pitch is shown in the following table.

Table 21.6. Maximum allowable speed for chains in r.p.m.

Type of chain	Number of teeth on the smaller sprocket (T_1)	Chain pitch (p) in mm				
		12	15	20	25	30
Roller chain	15	2300	1900	1350	1150	1100
	19	2400	2000	1450	1200	1050
	23	2500	2100	1500	1250	1100
	27	2550	2150	1550	1300	1100
	30	2600	2200	1550	1300	1100
Silent chain	17–35	3300	2650	2200	1650	1300

Note: The r.p.m. of the sprocket reduces as the chain pitch increases for a given number of teeth.

21.17 Principal Dimensions of Tooth Profile

The standard profiles for the teeth of a sprocket are shown in Fig. 21.12. According to Indian Standards (IS: 2403 – 1991), the principal dimensions of the tooth profile are as follows:
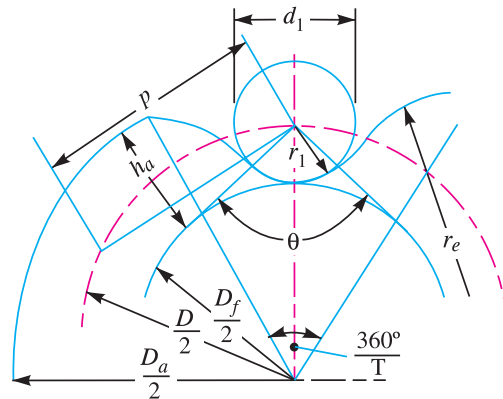
1. Tooth flank radius (r_e)

$$= 0.008 \, d_1 \, (T^2 + 180) \qquad\qquad\qquad\qquad ...(\text{Maximum})$$

$$= 0.12 \, d_1 \, (T + 2) \qquad\qquad\qquad\qquad\qquad ...(\text{Minimum})$$

where $\qquad\qquad d_1$ = Roller diameter, and

$\qquad\qquad\quad T$ = Number of teeth.

2. Roller seating radius (r_i)

$$= 0.505 \, d_1 + 0.069 \, \sqrt[3]{d_1} \qquad\qquad\qquad ...(\text{Maximum})$$

$$= 0.505 \, d_1 \qquad\qquad\qquad\qquad\qquad\qquad\; ...(\text{Minimum})$$

3. Roller seating angle (α)

$$= 140º - \frac{90º}{T} \qquad\qquad\qquad\qquad\qquad\quad ...(\text{Maximum})$$

$$= 120º - \frac{90º}{T} \qquad\qquad\qquad\qquad\qquad\quad ...(\text{Minimum})$$

4. Tooth height above the pitch polygon (h_a)

$$= 0.625 \, p - 0.5 \, d_1 + \frac{0.8 \, p}{T} \qquad\qquad\quad ...(\text{Maximum})$$

$$= 0.5 \, (p - d_1) \qquad\qquad\qquad\qquad\qquad\; ...(\text{Minimum})$$

(a) Tooth profile of sprocket.
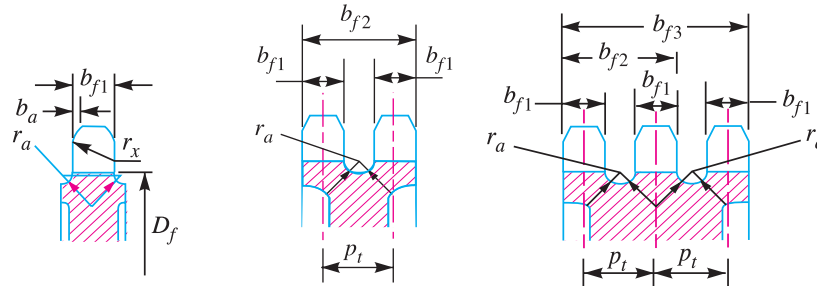
(b) Rim profile of sprocket.

Fig. 21.12

5. Pitch circle diameter (D)

$$= \frac{p}{\sin\left(\frac{180}{T}\right)} = p \text{ cosec}\left(\frac{180}{T}\right)$$

6. Top diameter (D_a)

$$= D + 1.25\, p - d_1$$

...(Maximum)

$$= D + p\left(1 - \frac{1.6}{T}\right) - d_1$$

...(Minimum)

7. Root diameter (D_f)

$$= D - 2\, r_i$$

8. Tooth width (b_{f1})

$$= 0.93\, b_1 \text{ when } p \le 12.7 \text{ mm}$$
$$= 0.95\, b_1 \text{ when } p > 12.7 \text{ mm}$$

9. Tooth side radius (r_x) $= p$

10. Tooth side relief (b_a)

$$= 0.1\, p \text{ to } 0.15\, p$$

11. Widths over teeth (b_{f2} and b_{f3})

$$= (\text{Number of strands} - 1)\, p_t + b_{f1}$$

Chain drive of an automobile

21.18 Design Procedure of Chain Drive

The chain drive is designed as discussed below:

1. First of all, determine the velocity ratio of the chain drive.

2. Select the minimum number of teeth on the smaller sprocket or pinion from Table 21.5.

3. Find the number of teeth on the larger sprocket.

4. Determine the design power by using the service factor, such that

Design power = Rated power × Service factor

5. Choose the type of chain, number of strands for the design power and r.p.m. of the smaller sprocket from Table 21.4.

6. Note down the parameters of the chain, such as pitch, roller diameter, minimum width of roller etc. from Table 21.1.

7. Find pitch circle diameters and pitch line velocity of the smaller sprocket.

8. Determine the load (W) on the chain by using the following relation, *i.e.*

$$W = \frac{\text{Rated power}}{\text{Pitch line velocity}}$$

9. Calculate the factor of safety by dividing the breaking load (W_B) to the load on the chain (W). This value of factor of safety should be greater than the value given in Table 21.2.

10. Fix the centre distance between the sprockets.

11. Determine the length of the chain.

12. The other dimensions may be fixed as given in Art. 21.17.

Example 21.1. *Design a chain drive to actuate a compressor from 15 kW electric motor running at 1000 r.p.m., the compressor speed being 350 r.p.m. The minimum centre distance is 500 mm. The compressor operates 16 hours per day. The chain tension may be adjusted by shifting the motor on slides.*

Solution. Given : Rated power = 15 kW ; N_1 = 1000 r.p.m ; N_2 = 350 r.p.m.

We know that the velocity ratio of chain drive,

$$V.R. = \frac{N_1}{N_2} = \frac{1000}{350} = 2.86 \text{ say } 3$$

From Table 21.5, we find that for the roller chain, the number of teeth on the smaller sprocket or pinion (T_1) for a velocity ratio of 3 are 25.

∴ Number of teeth on the larger sprocket or gear,

$$T_2 = T_1 \times \frac{N_1}{N_2} = 25 \times \frac{1000}{350} = 71.5 \text{ say } 72 \text{ **Ans.**}$$

We know that the design power

= Rated power × Service factor (K_S)

The service factor (K_S) is the product of various factors K_1, K_2 and K_3. The values of these factors are taken as follows:

Load factor (K_1) for variable load with heavy shock

= 1.5

Lubrication factor (K_2) for drop lubrication

= 1

Rating factor (K_3) for 16 hours per day

= 1.25

∴ Service factor, $K_S = K_1.K_2.K_3 = 1.5 \times 1 \times 1.25 = 1.875$

and design power   = 15 × 1.875 = 28.125 kW

From Table 21.4, we find that corresponding to a pinion speed of 1000 r.p.m. the power transmitted for chain No. 12 is 15.65 kW per strand. Therefore, a chain No. 12 with two strands can be used to transmit the required power. From Table 21.1, we find that

Pitch, p = 19.05 mm

Chain drive

Roller diameter, $d = 12.07$ mm

Minimum width of roller,

$$w = 11.68 \text{ mm}$$

Breaking load, $W_B = 59 \text{ kN} = 59 \times 10^3 \text{ N}$

We know that pitch circle diameter of the smaller sprocket or pinion,

$$d_1 = p \text{ cosec} \left(\frac{180}{T_1}\right) = 19.05 \text{ cosec} \left(\frac{180}{25}\right) \text{ mm}$$

$$= 19.05 \times 7.98 = 152 \text{ mm} = 0.152 \text{ m } \textbf{Ans.}$$

and pitch circle diameter of the larger sprocket or gear

$$d_2 = p \text{ cosec} \left(\frac{180}{T_2}\right) = 19.05 \text{ cosec} \left(\frac{180}{72}\right) \text{ mm}$$

$$= 19.05 \times 22.9 = 436 \text{ mm} = 0.436 \text{ m } \textbf{Ans.}$$

Pitch line velocity of the smaller sprocket,

$$v_1 = \frac{\pi \, d_1 \, N_1}{60} = \frac{\pi \times 0.152 \times 1000}{60} = 7.96 \text{ m/s}$$

∴ Load on the chain,

$$W = \frac{\text{Rated power}}{\text{Pitch line velocity}} = \frac{15}{7.96} = 1.844 \text{ kN} = 1844 \text{ N}$$

and factor of safety $= \dfrac{W_B}{W} = \dfrac{59 \times 10^3}{1844} = 32$

This value is more than the value given in Table 21.2, which is equal to 11.

The minimum centre distance between the smaller and larger sprockets should be 30 to 50 times the pitch. Let us take it as 30 times the pitch.

∴ Centre distance between the sprockets,

$$= 30 \, p = 30 \times 19.05 = 572 \text{ mm}$$

In order to accomodate initial sag in the chain, the value of centre distance is reduced by 2 to 5 mm.

∴ Correct centre distance

$$x = 572 - 4 = 568 \text{ mm}$$

We know that the number of chain links

$$K = \frac{T_1 + T_2}{2} + \frac{2 \, x}{p} + \left[\frac{T_2 - T_1}{2 \pi}\right]^2 \frac{p}{x}$$

$$= \frac{25 + 72}{2} + \frac{2 \times 568}{19.05} + \left[\frac{72 - 25}{2 \pi}\right]^2 \frac{19.05}{568}$$

$$= 48.5 + 59.6 + 1.9 = 110$$

∴ Length of the chain,

$$L = K.p = 110 \times 19.05 = 2096 \text{ mm} = 2.096 \text{ m } \textbf{Ans.}$$

EXERCISES

1. Design a roller chain to transmit power from a 20 kW motor to a reciprocating pump. The pump is to operate continuously 24 hours per day. The speed of the motor is 600 r.p.m. and that of the pump is 200 r.p.m. Find: 1. number of teeth on each sprocket; 2. pitch and width of the chain.

2. Design a chain drive to run a blower at 600 r.p.m. The power to the blower is available from a 8 kW motor at 1500 r.p.m. The centre distance is to be kept at 800 mm.

3. A chain drive using bush roller chain transmits 5.6 kW of power. The driving shaft on an electric motor runs at 1440 r.p.m. and velocity ratio is 5. The centre distance of the drive is restricted to $550 \pm 2\%$ mm and allowable pressure on the pivot joint is not to exceed 10 N/mm^2. The drive is required to operate continuously with periodic lubrication and driven machine is such that load can be regarded as fairly constant with jerk and impact. Design the chain drive by calculating leading dimensions, number of teeth on the sprocket and specify the breaking strength of the chain. Assume a factor of safety of 13.

QUESTIONS

1. State the advantages and disadvantages of the chain drive over belt and rope drive.
2. Explain, with the help of a neat sketch, the construction of a roller chain.
3. What do you understand by simplex, duplex and triplex chains?
4. Write in brief on
   (a)Hoisting and hauling chains,
   (b)Conveyor chais, and
   (c)Silent chains.
5. Write the design procedure for a chain drive.

OBJECTIVE TYPE QUESTIONS

1. Which one of the following is a positive drive?
   (a) Crossed flat belt drive   (b) Rope drive
   (c) V-belt drive   (d) Chain drive
2. The chain drive transmits .... power as compared to belt drive.
   (a) more   (b) less
3. The relation between the pitch of the chain (p) and pitch circle diameter of the sprocket (D) is given by

   (a) $p = D \sin \left(\dfrac{90°}{T} \right)$ (b) $p = D \sin \left(\dfrac{120°}{T} \right)$

   (c) $p = D \sin \left(\dfrac{180°}{T} \right)$ (d) $p = D \sin \left(\dfrac{360°}{T} \right)$

   where T = Number of teeth on the spoocket.
4. In order to have smooth operation, the minimum number of teeth on the smaller sprocket, for moderate speeds, should be
   (a) 15   (b) 17
   (c) 21   (d) 25
5. The speed of the sprocket reduces as the chain pitch .... for a given number of teeth.
   (a) increases   (b) decreases

ANSWERS

1. (d)   **2.** (a)   **3.** (c)   **4.** (b)   **5.** (a)