CHAIN DRIVES

Introduction

In its simplest form, a chain drive consists of two toothed wheels, one driving the other joined by a flexible connector called a CHAIN. The toothed wheels are known as SPROCKETS and are mounted on shafts by keys or other means of fastening. (fig. below).

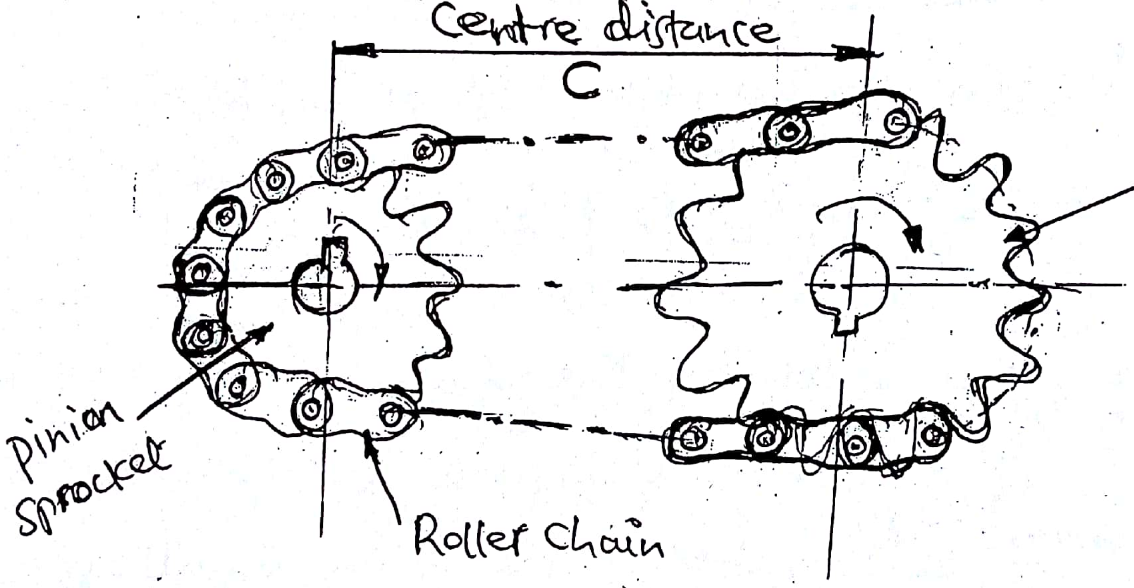


Fig: Roller chain drive layout

1. General remarks

1.1 Drive layout

- In any position with Long centre distance Excessive slack must be checked, Use an idler (for slack)

Fixed Centre drive - can operate with or without an idler.

Vertical position - care must be taken with excessive slack. Use of idler if necessary.

Least angle of contact

- between chain and sprocket = 90^0

1.2 Chain length (L)

The length of the chain is given by

$$\frac{L}{P} = \frac{2C}{P} + \left(\frac{Z_1 + Z_2}{2}\right) + \frac{(Z_2 - Z_1)^2 P}{4\pi^2 C} \qquad (1)$$

where P = pitch of the chain

C = Centre distance between sprockets.

Z_1 = number of teeth on smaller (pinion) sprocket

Z_2 = number of teeth on larger (wheel) sprocket.

1.3 Speed ratio (i)

$$\text{Speed ratio } i = \frac{n_1}{n_2} = \frac{Z_2}{Z_1} \qquad (2)$$

n = speed,

Z = no. of teeth on corresponding sprockets

1.4 Chain velocity (V)

$V = r\omega$, $r = \frac{D}{2}$, $\omega = \frac{\pi n}{30}$

$$\therefore V = \frac{D}{2} \times \frac{\pi n}{30} = \frac{\pi D n}{60}$$

D = pitch diameter of sprocket

but $\pi D = Z . p$

$$\therefore \boxed{V = \frac{\pi D n}{60} = \frac{Z p n}{60}} \qquad (3)$$

where D = pitch diameter of sprocket
Z = no. of teeth
p = pitch
n = speed in rpm.

1.5 Main types of power transmission chain drives are:

(i) Roller chain drives — most commonly used

(ii) Silent chain drives — used for heavier power transmission.

1.6 Advantages and Disadvantages of chain drives

Advantages

- No slip hence constant speed ratio.
- Long centre distance
- No tensioning devices
- Possible to drive several parallel shafts with same chain (fig. below)

Disadvantages

- Use of large space
- Noisy operation
- Careful and regular maintenance
- Use of covers for safety and for bath lubrication
- More costly than belt drives
- Wear of rollers / bushes reduces the useful life of the chain.

Note:
To reduce the slackness of the chain, use SLACK ADJUSTERS.

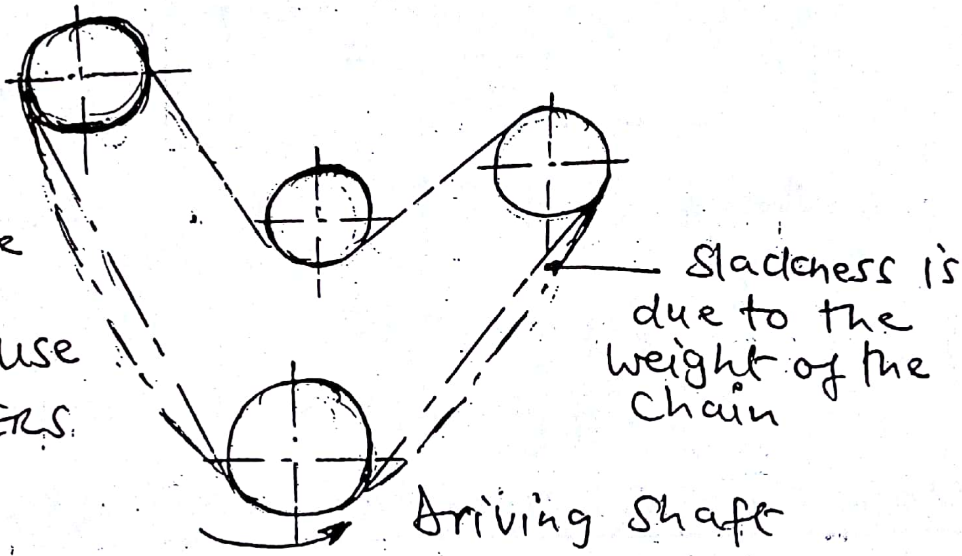


Fig: Driving several shafts

2.0 CHAIN TYPES / CONSTRUCTION

2.1 Silent chain — also known as the Inverted Tooth Chain.

- A silent chain consists of a series of toothed plates pinned together in rows accross the width of the chain. (fig. below).

- The advantage of silent chain over the roller type is their smooth and noiseless operation at high velocities.
- The disadvantage is a more intricate design. Hence more expensive and require good maintenance.

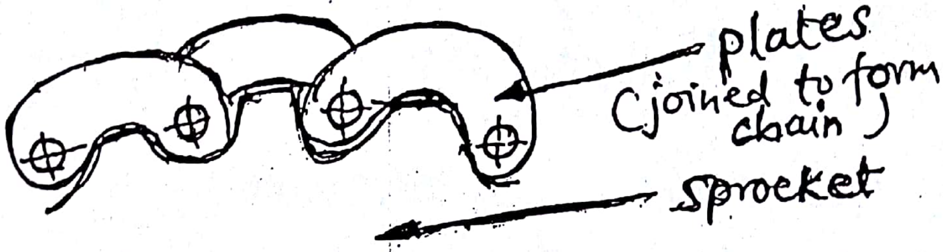
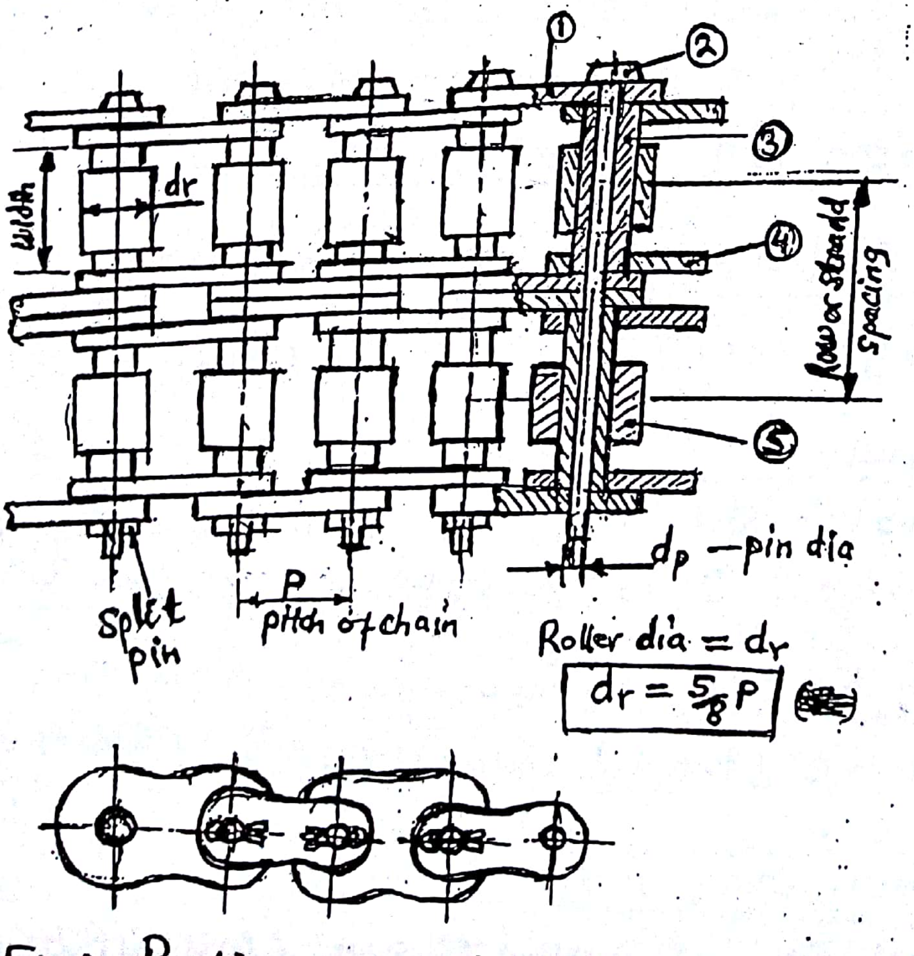


Fig: Silent Chain

2.2 Roller chain

1. Principal parts of a roller chain



$\boxed{d_r = \frac{5}{8}P}$

Fig: Portion of a double-strand roller chain

① Outer plate } Outer link
② pin

③ Bush } Inner link
④ Inner plate

⑤ Roller — freely revolving.

Types of fit

① / ④ Clearance
③ / ④ Interference
③ / ⑤ Clearance
① / ② Interference
② / ③ Clearance

.3 Classification of roller chains

Light-Duty Roller chain

— Is a single row / strand roller chain

Medium-Duty Roller chain

— Is a double strand roller chain (e.g. fig. above)
Also called DUPLEX CHAIN

Heavy-Duty Roller Chain

— Is a three strand roller chain (TRIPLEX CHAIN)

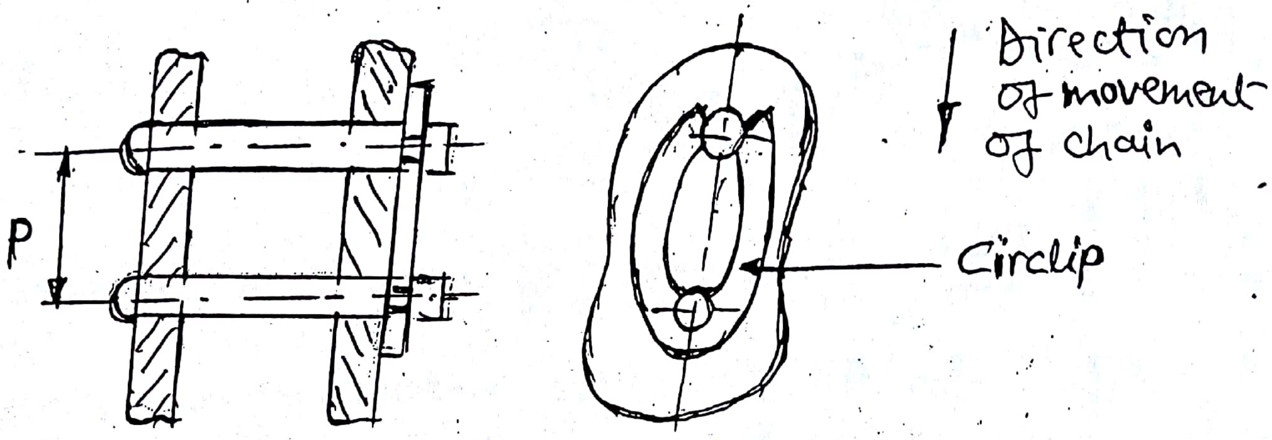
Extra-Heavy Duty Roller Chain

— Is a multi-strand roller chain (MULTIPLEX CHAIN)

2.4 Connecting Links

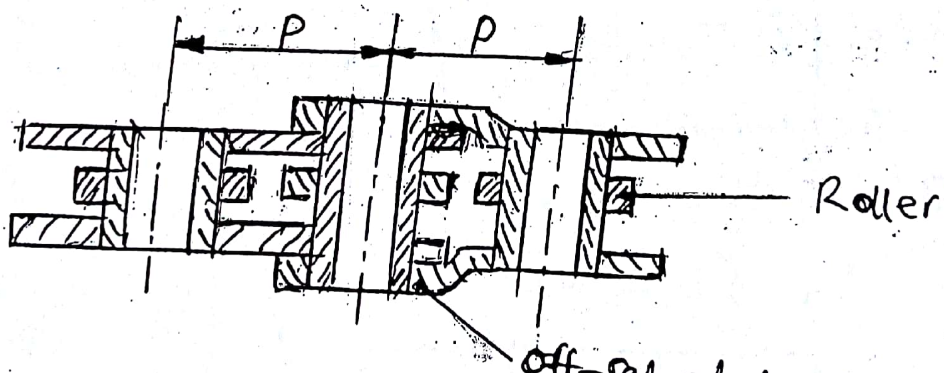
(a) Chain Link

– Is used for even number of chain links/pitches or odd number of sprocket teeth.



(b) Off-set link

– Is used for even number of sprocket teeth or odd number of chain pitches/links



2.5 Geometric and Kinematic Relations

Consider the engagement of a chain and sprocket. (Fig. below).

Let the sprocket drive in the direction shown. Also let the chain pitch be 'p' and the pitch angle be 'γ', and the pitch diameter of the sprocket be 'D'.

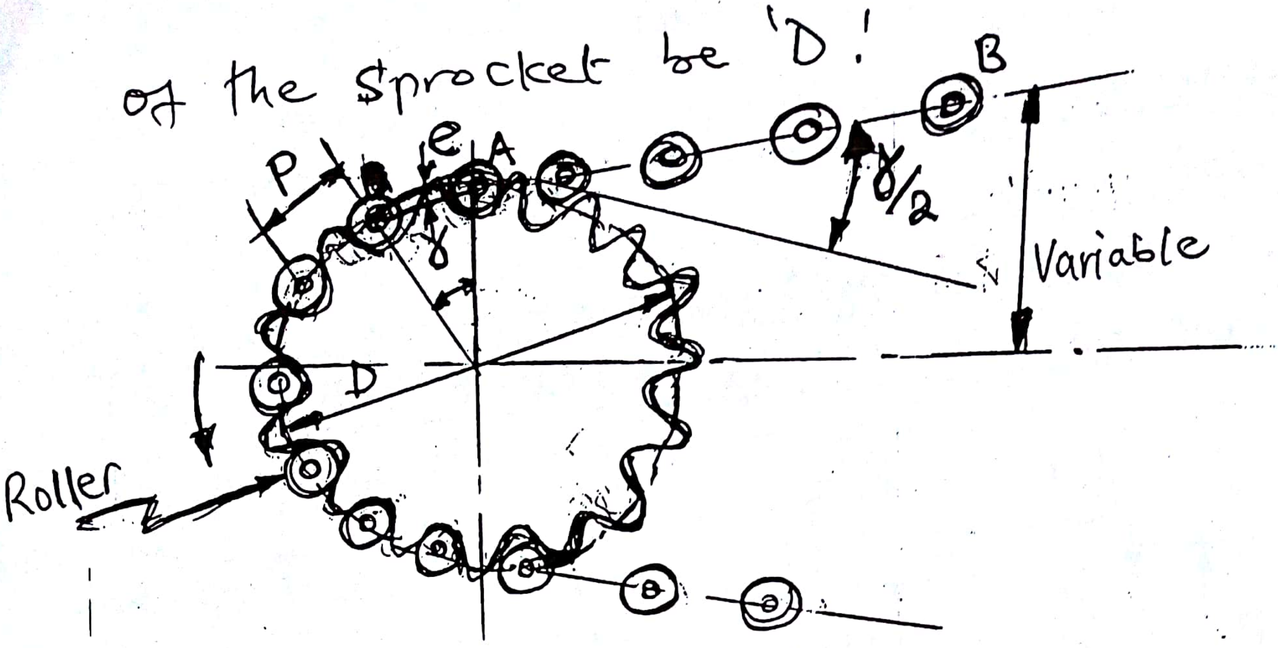
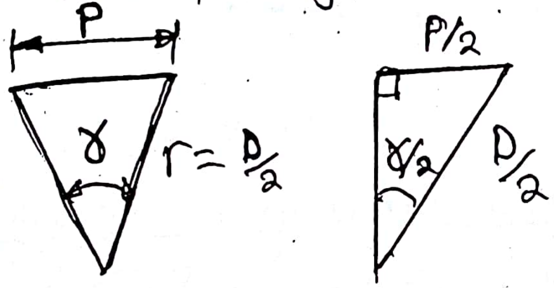


Fig: Engagement of a chain and sprocket

Then for the triangle below



$$\therefore \sin\frac{\gamma}{2} = \frac{P/2}{D/2}$$

Hence $\boxed{D = \dfrac{P}{\sin \gamma/2}}$ (4)

If 'z' is the number of teeth of the sprocket

$\therefore \gamma z = 360^\circ$

$$\therefore \boxed{D = \frac{P}{\sin\left(\frac{180^\circ}{z}\right)}} \quad (5)$$

Where D = pitch diameter of sprocket.

z = Number of teeth on sprocket

P = chain pitch

and γ = pitch angle.

Outer diameter of sprocket 'D_o' is given by

$$\boxed{D_o = D + 0.8\, d_r} \quad (6)$$

where d_r = diameter of roller $= \frac{5}{8} P$.

Articulation angle ($\gamma/2$)

omit

The angle ($\gamma/2$) in fig. above for the engagement is known as the 'articulation angle'. It is the angle through which the link swings as it enters contact with the sprocket teeth. Rotation of the link through this angle causes impact between the rollers and the sprocket teeth and also wear of the chain joint. It is therefore important to reduce this angle as much as possible.

Fig. below gives the magnitude of '($\gamma/2$)' as a function of the number of teeth 'Z'.

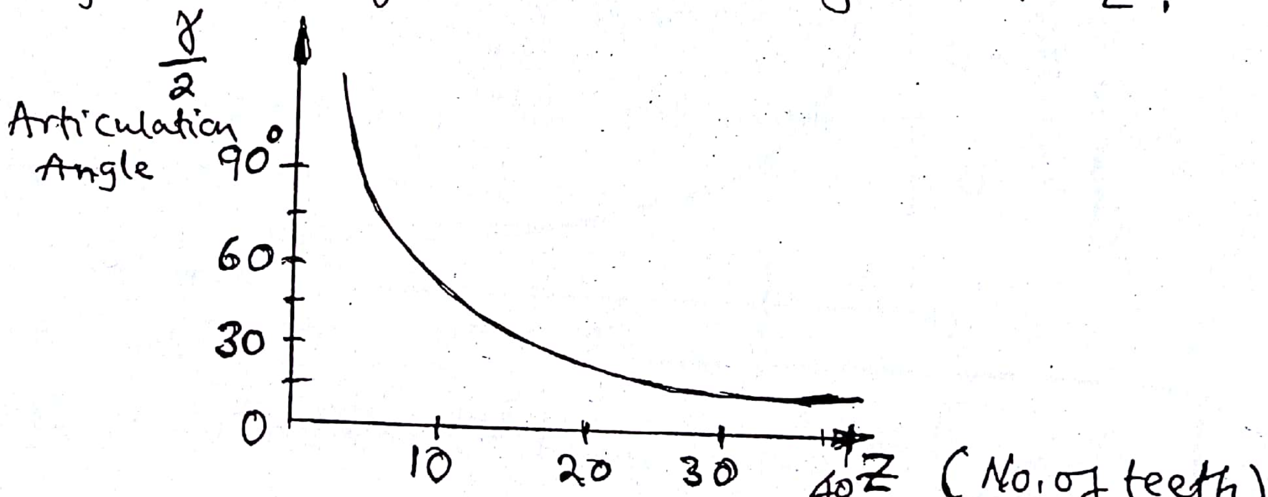


Fig: $\gamma/2$ Vs Z

Chordal Speed Variation

Refering to fig above for the engagement of chain and sprocket, when the sprocket has turned through an angle '$\gamma/2$', the pitch line AB is moved closer to the sprocket centreline by an amount 'e'. This means that the straight portion AB is moving up and down as the sprocket turns. That is it moves in chords instead of a curved path. Hence the lever arm also varies and therefore the velocity ratio is not constant for the rotation of the sprocket through the pitch angle. This is known as Chordal speed variation. This speed variation is also a function of the number of sprocket teeth. (Fig. below).

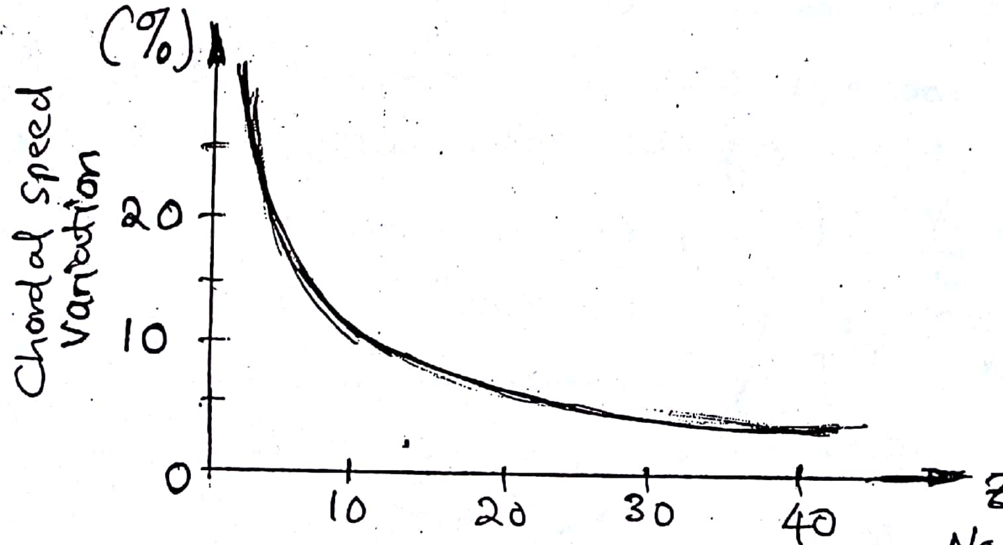


fig: Chordal Speed variation Vs No. of teeth omit

3.0 Analysis and Design of Roller Chain Drives

The main parameter of any chain drive is its pitch 'p'. Pitch values are standardized. The load carrying capacity (the power rating of a drive) increases with increasing pitch.

3.1 Number of teeth on Sprocket (Z)

Number of teeth on smaller sprocket (Z_1) should be taken as $Z_1 \geq 13$, since a smaller value of Z_1 would lead to excessive noise and heavier impacts. Where space is a problem and for very low speeds, smaller tooth numbers may be used at the expence of the chain life.

Z_2 for the larger sprocket should not exceed 120 for roller chain and 140 for silent chain drives, because the stretched chain may then run off the larger sprocket.

Most commonly, the optimum number of teeth Z_1 is found as

$$\boxed{Z_1 = 31 - 2i} \quad (7)$$

where i = speed ratio. $\boxed{i = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2} = \frac{Z_2}{Z_1}}$

3.2 Speed ratio (i):

As a rule $\boxed{i \leq 5}$ (8)

However in some low-speed drives $i = 8$ or even higher at the sacrifice of chain life.

3.3 Chain Velocity (V)

For design purposes the average chain velocity should be taken as given in eqn (3) above.

$$V \begin{cases} \leq 12 \text{ m/s for roller chain} \\ \leq 16 \text{ m/s for silent chain} \end{cases}$$

3.4 Chain Length (L):

Eqn (1) gives the chain length in the number of pitches (chain links). Therefore this should be rounded off to the nearest integer.

i.e. $\boxed{m = L/p}$ whole nos.

3.5 Centre distance (C):

In general $\boxed{C \leq 80p}$ (9)

Better value is

$$\boxed{30p \leq C \leq 50p} \quad (10)$$

Also for normal operation, the slack side of the chain should have a slight sag 'f'. For this purpose, the design sprocket centre distance 'C' is reduced by 0.002C to 0.004C.

3.6 TABLES

1. Factor of Safety for Bush roller chains (fs)

Chain width [mm]	Speed of Sprocket pinion [rpm]								
	50	200	400	600	800	1000	1200	1600	2000
12 - 15	7.0	7.8	8.55	9.35	10.2	11.0	11.7	13.2	14.8
20 - 25	7.0	8.2	9.35	10.3	11.7	12.9	14.0	16.3	—
30 - 35	7.0	8.55	10.2	13.2	14.8	16.3	19.5	—	—

2. No. of teeth Z_1 in sprocket pinion

Transmission ratio (i)	1 - 2	2 - 3	3 - 4	4 - 5	5 - 7
No. of teeth (Z_1)	30-27	27-25	25-23	23-21	21-17

3. Maximum velocity of Bush roller chain [rpm]

No. of teeth in pinion (Z_1)	Chain pitch p [mm]				
	12	15	20	25	30
15	2,300	1900	1350	1150	1000
19	2,400	2000	1450	1200	1050
23	2,500	2100	1500	1250	1100
27	2,550	2150	1550	1300	1100
30	2,600	2200	1550	1300	1100

4. Tooth correction factor K_T

No. of teeth on driving sprocket Z_1	11	12	13	14	15	16	17	18
Tooth correction factor K_T	0.53	0.62	0.70	0.78	0.85	0.92	1.00	1.05

Z_1	19	20	21	22	23	24	25	30	35	40	45	50	55	60
K_T	1.11	1.18	1.26	1.29	1.35	1.41	1.46	1.73	1.95	2.15	2.37	2.51	2.66	2.80

3.7 FORCES ON THE CHAIN

1. Transmitted force (F_t)

The turning force transmitted by a chain is given by

$$\boxed{F_t = \frac{P}{V} = \frac{2T}{D}}$$

Where
P = power transmitted
D = pitch diameter of sprocket
T = transmitted torque
and V = chain velocity.

2. Slack side tension (also known as ~~pretetension~~ pretension in the chain) – F_f

Is caused by the sagging of the slack side tension. F_f is given by

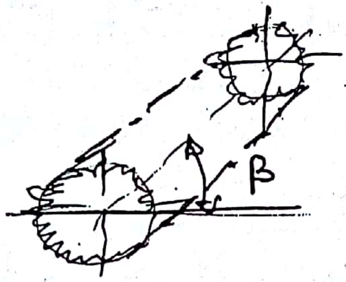
$$\boxed{F_f = K_f \cdot m_l \, g \, C}$$

where m_l = chain weight [kg/m]

C = centre distance [m]

g = 9.81 m/s² (gravitational acceleration)

K_f = Sag factor depending on the angle β between centreline of the chain and the horizontal.



i.e $K_f \begin{cases} = 6 & \text{for } \beta = 0 \quad \text{horizontal} \\ = 3 & \text{for } \beta \leq 40^\circ \\ = 1 & \text{for } \beta = 90^\circ \quad \text{vertical} \end{cases}$

3. <u>Centrifugal tension (F_c):</u>

The centrifugal tension is to be taken into account at $V \geq 5$ m/s. F_c is given by

$$\boxed{F_c \simeq m_l V^2}$$

where m_l = mass/length of chain [kg/m]

V = Chain speed

4. <u>Tight side tension (F_1)</u>

In operation, the load is carried by the tight side of the chain. F_1 is given by

$$\boxed{F_1 = F_t + F_c + F_f}$$

5. Load on sprocket shaft (F_s)

As in belt drives, the forces developed in the chain by the centrifugal action are not transferred to the sprocket shafts. The shafts are only loaded by the turning force F_t and $2F_f$ due to chain sag.

$$\therefore \boxed{F_s = K_{sh} \cdot F_t + 2F_f}$$

where K_{sh} = shaft loading factor. Generally $K_{sh} = 1.05 - 1.15$. The lower for vertical and higher for horizontal drives.

6. Chain breaking strength (F_u) and factor of safety (f_s)

The table for factors of safety (f_s) is as given above.

The chain ultimate load (F_u) is given by

$$\boxed{F_u = f_s \cdot K_s F_t}$$

or

$$\boxed{F_u = f_s F_1}$$

Where F_t = transmitted force
F_1 = tight side tension

and K_s = Service factor

3.8 SERVICE FACTOR (K_s)

K_s can be obtained directly in some other tables depending on the drive conditions.

If not available then K_s can be given as

$$\boxed{K_s = K_d K_c K_l K_\beta K_m K_{con}}$$

where

K_d = Dynamic factor which takes care of the manner of load application. ($K_d = 1$ for static loading, $K_d = 1.2 - 1.5$ for dynamic loading)

K_c = accounts for the effect of shaft centre distance. ($K_c = 1$ for $C = 30p - 50p$, and $K_c = 0.8$ for $C = 60p - 80p$).

K_l = Lubrication factor depending on the manner of lubrication. ($K_l = 0.8$ for immersion lubrication, $K_l = 1$ for drop feed lubrication, and $K_l = 1.5$ for periodic greasing).

K_β = factor of angle that centreline make with horizontal. ($K_\beta = 1$ for $\beta \leq 60°$ and $K_\beta = 1.25$ for $\beta > 60°$)

K_m = factor of mode of operation. ($K_m = 1$ for one shift operation, $K_m = 1.25$ for 2 shifts operation, and $K_m = 1.5$ for 24 hour operation).

K_{com} = factor of manner of tension control adjusted bases. ($K_{con} = 1.25$ with no tension control).

3.9 POWER RATING

Total rated Power (P_{RT})

The design power or total rated power P_{RT} is given by

$$\boxed{P_{RT} = P_a \cdot K_s}$$

Where P_a = Power available from the actuator (e.g. motor)

K_s = service factor (as above)

Rated power per strand (P_{RS})

For multi-strand chain drives, the rated power per strand is given by

$$\boxed{P_{RS} = P_s \cdot K_T}$$

where P_S = power per strand on a 17-tooth sprocket. [Depending on the type of the chain - Chain No, and the speed (rpm) of the sprocket] - from tables

K_T = Tooth correction factor (as above)

Number of strands (N)

Then the number of strands for the drive is given by

$$\boxed{N = P_{RT}/P_{RS}}$$

Strand correction factor (K_N)

The total rated power P_{RT} is given by

$$\boxed{P_{RT} = P_{RS} \cdot N \cdot K_N}$$

where K_N $\begin{cases} = \text{No. of strands correction factor} \\ = 1 \text{ for single strand} \\ < 1 \text{ for multi strands.} \end{cases}$

40 CRITERIA OF SERVICEABILITY

Majority of chain drives fail because of worn hinges which permanently lengthen the chain and spoil the mesh. Hence the wear resistance of the hinges appears to be the the most criteria of serviceablity of a chain drive. Therefore due to this, the design of a chain drive reduces ~~the~~ to the choice of a chain that would be sufficiently reliable and durable under given conditions.

A chain is said to have an adequate load carrying capacity if the design average pressure developed in a link hinge does not exceed a certain allowable value of pressure 'P_{all}'

$$\therefore \boxed{P_{max} = \frac{F_t \cdot K_s}{A} = P_{all}}$$

where P_{max} = maximum developed pressure on link hinge.

P_{all} = allowable pressure value.

F_t = Turning or transmitted force of a chain

A $\begin{cases} = \text{bearing area of the hinge.} \\ = dl; \; l = \text{bush length (width)} \end{cases}$

K_s = Service factor

For design purposes the above equation is rearranged to yield a required pitch 'p'. Substituting $F_t = \frac{2T_1}{D}$, and $D_1 = \frac{Z_1 p}{\pi}$

and taking into account that

$A \simeq 0.25p^2N - 0.28p^2N$ for most

standard chains, we get the following conditions for load-carrying capacity expressed in terms of the average pressure

$$\boxed{P = \frac{\pi \cdot 2T_1 \cdot K_s}{Z_1 (0.28) p^3 N} = P_{all}}$$

where p = pitch. Note: P_{max} for $A = 0.25p^2N$

The chain pitch 'p' is given by

$$\boxed{p = 2.8 \sqrt[3]{\frac{T_1 K_s}{Z_1 P_{all} \cdot N}}}$$

Where N = Number of strands in the chain.

Note: When selecting a chain the load service factor K_s should be taken as given in the manufacturer's catalogue.

5.0 INSTALLATION AND MAINTENANCE

Installation:- Shaft alignment, sprocket alignment should be correct.
Make proper chain tension.

Lubrication - Use light oil (free flowing oil), never use grease.

- lubricate the pin / bush joint.
- Check for evidence of poor lubrication.
- use covers to exclude dirt and moisture.

<u>Cleaning</u> — use paraffin or petrol to clean

<u>Maintenance</u> — Regular maintenance
- Check tension or slack
- Check lubrication
- Check for wear of sprocket teeth or rollers
- Check for alignment.

6.0 <u>EXAMPLES</u>

<u>Ex. 1</u>

Design a chain drive to actuate a compressor from a 15 h.p. electric motor running at 970 rpm, the compressor speed being 330 rpm. The compressor operates in two shifts. The centre distance should be minimum 500 mm. The chain tension can be adjusted by shifting the motor on slides.

<u>Soln:</u>

Velocity of pinion sprocket is low, therefore <u>roller chain is suitable.</u>

Transmission ratio $i = \frac{n_1}{n_2} = \frac{970}{330} = 2.94 \simeq 3$

From table $\therefore$ for $i = 3$, pinion teeth $\underline{Z_1 = 25}$

$\therefore$ Compressor sprocket, $Z_2 = i Z_1 = \frac{970}{330} \times 25 = \underline{74}$

From table of max. velocity / pinion teeth, any pitch length can be adopted.

$\therefore$ take $\underline{p = 15 \text{ mm}}$ as chain pitch

Mean peripheral speed of the chain (V)

$$V = \frac{Z p n}{60} \qquad \therefore V = \frac{25 (15) 970}{60 \times 1000} \text{ m/s}$$

$\therefore \underline{V = 6.05 \text{ m/s}}$ $\qquad \therefore$ O.K. for roller chain $\underline{V_{max} = 12 \text{ m/s}}$.

<u>Load Service factor K_s</u>

(from book Shigley). Generally ranges from 1 to 3. Let us adopt the average i.e. $\underline{K_s = 2}$.

<u>Driving force F_t</u>

$$F_t = \frac{P}{V}, \qquad P = 15 \text{ h.p.} = 0.75 (15) \text{ kW}$$

$$\therefore F_t = \frac{0.75 (15) \times 10^3}{6.05} = 1859.5 \text{ N}$$
$$\underline{\simeq 189 \text{ kgf}}$$

From table we take factor of safety $\underline{f_s = 11}$

Ultimate strength of chain $F_u = f_s . K_s . F_t$

$\therefore F_u = 11 (2) (189) = 4158 \text{ kgf}$
$\underline{\simeq 4200 \text{ kgf}}$

$\therefore$ Select a chain whose breaking load > 4200 kgf.

Centre distance (C)

Take $C = 40p$ $\therefore C = 40 \times 15 = 600$ mm

$C_{min} = 500$ mm $\therefore$ O.K.

No. of links (m)

$$m = \frac{L}{p} = \frac{Z_1 + Z_2}{2} + \frac{2C}{p} + \frac{(Z_2 - Z_1)^2 p}{4\pi^2 C}$$

$$= \frac{25+74}{2} + \frac{2 \times 600}{15} + \frac{(74-25)^2 \times 15}{4\pi^2 \times 600}$$

$$m = 49.5 + 80 + 1.52 = 131$$

or take $m = 130$ even

$\therefore$ take $m = 130$ links.

$\therefore$ The exact centre distance C_A is then

$$130 = 49.5 + \frac{2C_A}{15} + \frac{912.27}{C_A}$$

$\therefore C_A = 592.2$ mm $\simeq 594$ mm

Total length of the chain (L)

$L = mp$ $\therefore L = 130(15) = 1950$ mm

Diameter of sprockets

Pitch dias D_1, D_2

From $D = \dfrac{P}{\sin \frac{180°}{z}}$

$\therefore D_1 = \dfrac{15}{\sin \frac{180}{25}} = \underline{119.68 \text{ mm}}$

& $D_2 = \dfrac{15}{\sin \frac{180}{74}} = \underline{353.43 \text{ mm}}$

Roller diameter: $d_r = \frac{5}{8} P = \frac{5}{8}(15) = 9.375 \text{ mm}$
$\simeq \underline{10 \text{ mm}}$

Outside diameters D_{o1}, D_{o2}

$D_o = D + 0.8 d_r$; $0.8 d_r = 0.8(10) = \underline{8 \text{ mm}}$

$\therefore D_{o1} = 119.68 + 8 = \underline{127.68 \text{ mm}}$

& $D_{o2} = 353.43 + 8 = \underline{361.43 \text{ mm}}$

EX. 2

A $7\frac{1}{2}$ hp speed reducer which runs at 300 rpm is to drive a conveyor at 200 rpm. The centre distance is to be approximately 28 in. Select a suitable chain drive.

Soln:

1. Odd number of sprockets is prefered, but choose sprockets of 20 and 30 teeth for proper speed ratio. A 20-tooth

sprocket will have a longer life and generate less noise than a 16 or 18-tooth sprocket. It is chosen because space does not seem a problem.

2. From tables (Shigley book), a service factor $K_s = 1.4$ is chosen for a 24-hr operation with moderate shock.

$\therefore$ Total power rating $P_{RT} = P_a \cdot K_s$

$\therefore P_{RT} = 7.5(1.4)\,hp = 10.5\,hp.$

From table (Shigley book), with $n_1 = 300$ rpm, No. 50 or No. 60 chain may be satisfactory.

Tooth correction factor $K_T = 1.18$ (20-tooth sprocket.)

Power rating per strand P_{RS}.

No. 50 Chain $P_{RS} = 2.99(1.18) = 3.53\,h.p.$

No. 60 Chain $P_{RS} = 4.98(1.18) = 5.88\,h.p.$

i.e., $P_s = 2.99$ hp for No. 50 chain

and $P_s = 4.98$ hp. for No. 60 chain

all at 300 rpm

$\therefore$ Number of strands (N)

No. 50 Chain $N = \frac{P_{RT}}{P_{RS}} = \frac{10.5}{3.53} = 2.97 \simeq 3$

No. 60 chain $N = \frac{P_{Ri}}{P_{Rs}} = \frac{10.5}{5.88} = 1.79$
$\simeq 2$

No. 60 chain would require larger sprockets and hence it would run at a higher velocity, generate more noise, and have a shorter life. For prices No. 60 chain may be a better solution.

Taking No. 50 chain we have:

$\therefore p = 5/8''$ (from table – Shigley)

using centre distance $C = 28''$, then the required length of the triple-strand chain in pitches is

$$\frac{L}{p} = \frac{2C}{p} + \frac{Z_1 + Z_2}{2} + \frac{(Z_2 - Z_1)^2 p}{4\pi^2 C}$$

$$= \frac{2(28)}{0.625} + \frac{20+30}{2} + \frac{(30-20)^2(0.625)}{4\pi^2(28)}$$

$= 114.7$ pitches

$\therefore$ The nearest even number of pitches $= 114$ and this will be used. Solving back, the actual centre distance $C_A = 27\frac{3}{4}''$.

$\therefore \frac{C}{p} = 44.4$ or $C = 44.4p$ $\therefore$ O.K. because $30p \le C \le 50p$.

Pitch diameters and outside diameters of Sprockets can be determined.

$Z_1 = 20, \quad Z_2 = 30, \quad p = \frac{5}{8}''$

$$d_r = \frac{5}{8} p$$

and $$D = \frac{p}{\sin\left(\frac{180}{Z}\right)}$$

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

Also check speed of chain (V)

i.e. $V = \frac{\pi D n}{60} = \frac{Z p n}{60}$ it will be low.

FINI