DESIGN OF POWER TRANSMISSION DEVICES

NILESH PANCHOLI
B.E. (Mech.), M.E. (Mech.), Ph. D.
Email: *nhpancholi@gmail.com*
www.nileshpancholi.com

- Design of Belt Drive
- Design of Rope Drive
- Design of Chain Drive

Design of Belt Drive

prof.nilesh pancholi

Introduction

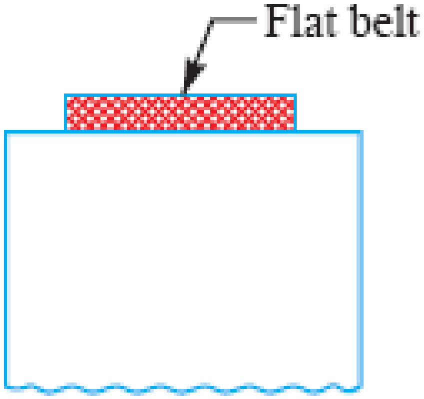
- The belts or ropes are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or at different speeds. The amount of power transmitted depends upon the following factors :

1. The velocity of the belt.
2. The tension under which the belt is placed on the pulleys.
3. The arc of contact between the belt and the smaller pulley.
4. The conditions under which the belt is used.

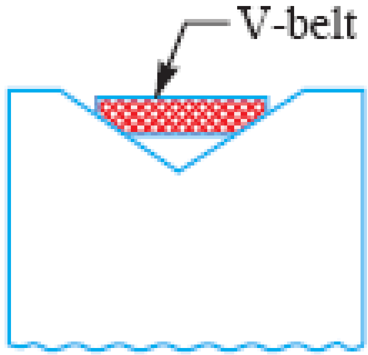
Selection of a Belt Drive

1. Speed of the driving and driven shafts,
2. Speed reduction ratio,
3. Power to be transmitted,
4. Centre distance between the shafts,
5. Positive drive requirements,
6. Shafts layout,
7. Space available, and
8. Service conditions.

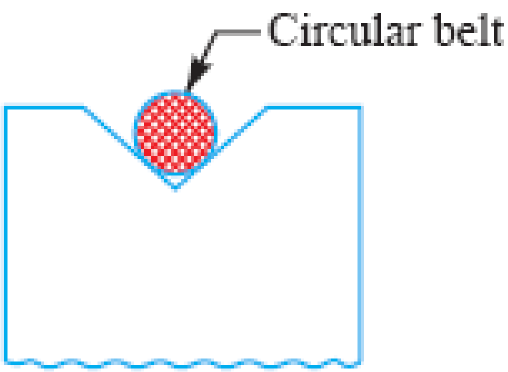
Types of Belts



(a) Flat belt.



(b) V-belt.

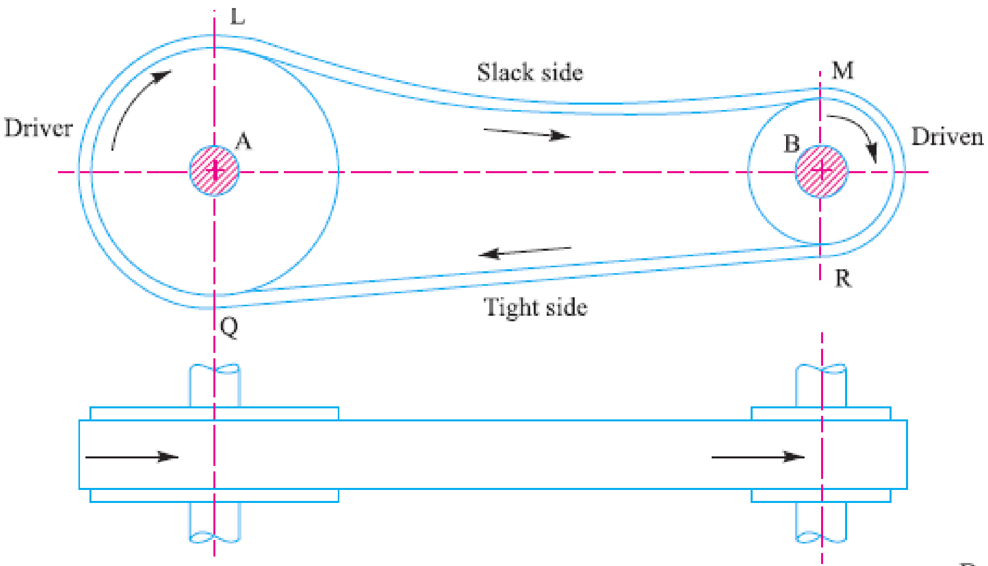


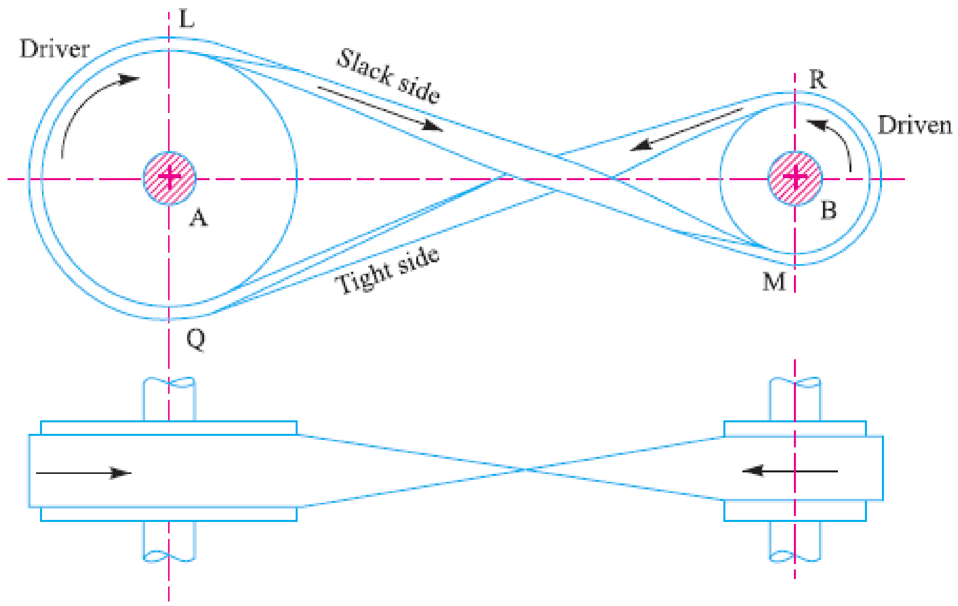
(c) Circular belt.

Material used for Belts

- The material used for belts and ropes must be strong, flexible, and durable. It must have a high coefficient of friction
- Leather, Cotton or fabric, Rubber, Balata etc.

Types of Flat Belt Drives





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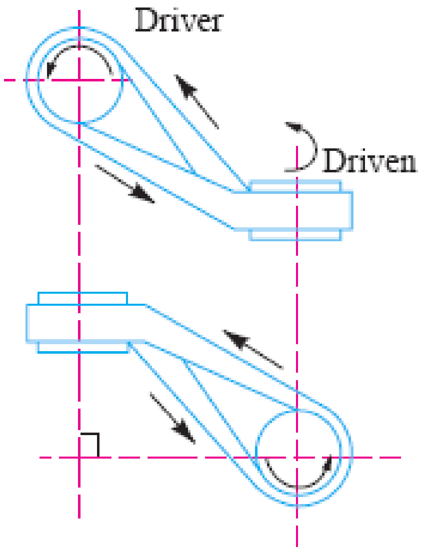
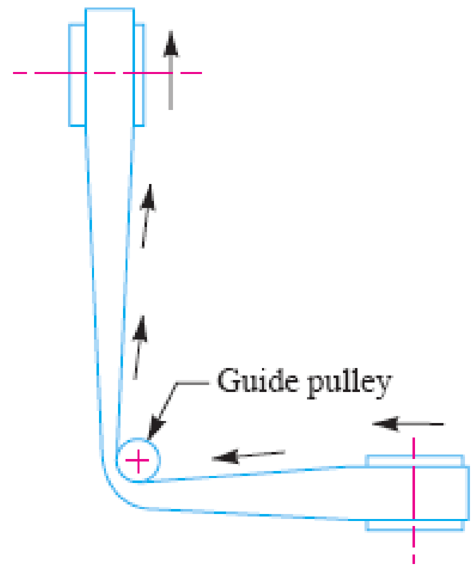
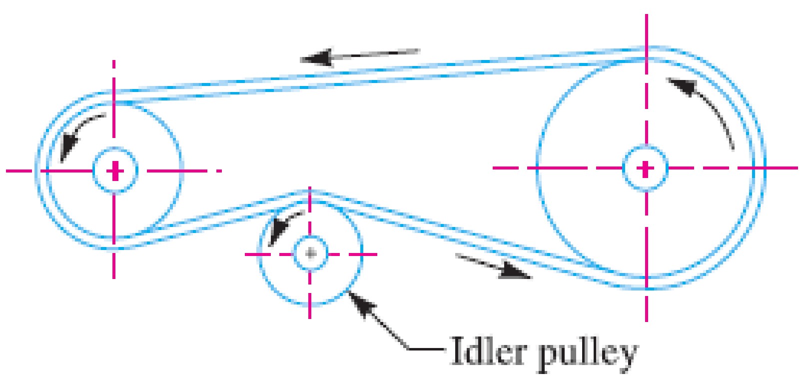
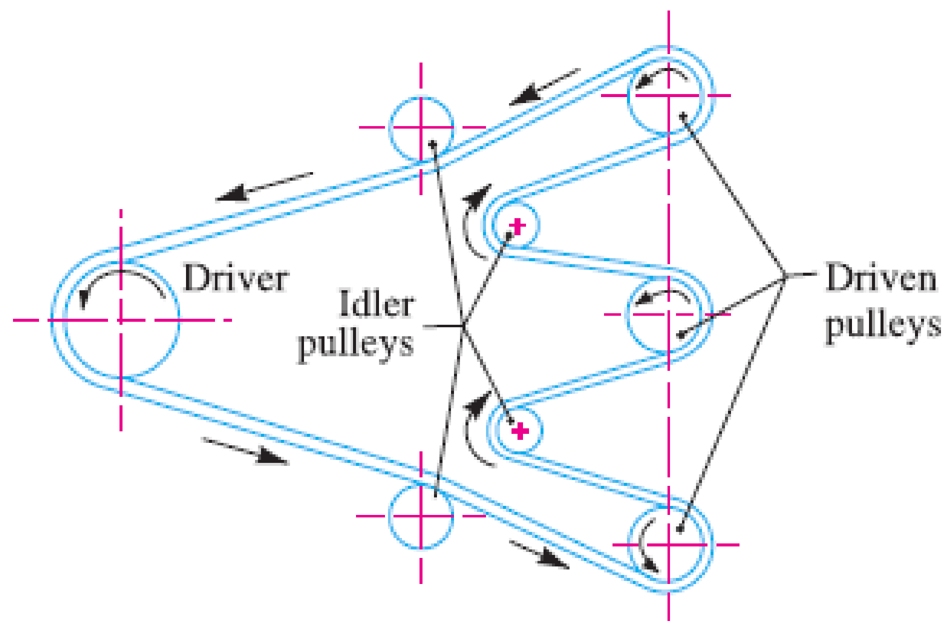

Driver
Driven


Guide pulley


Idler pulley


Driver
Idler pulleys
Driven pulleys

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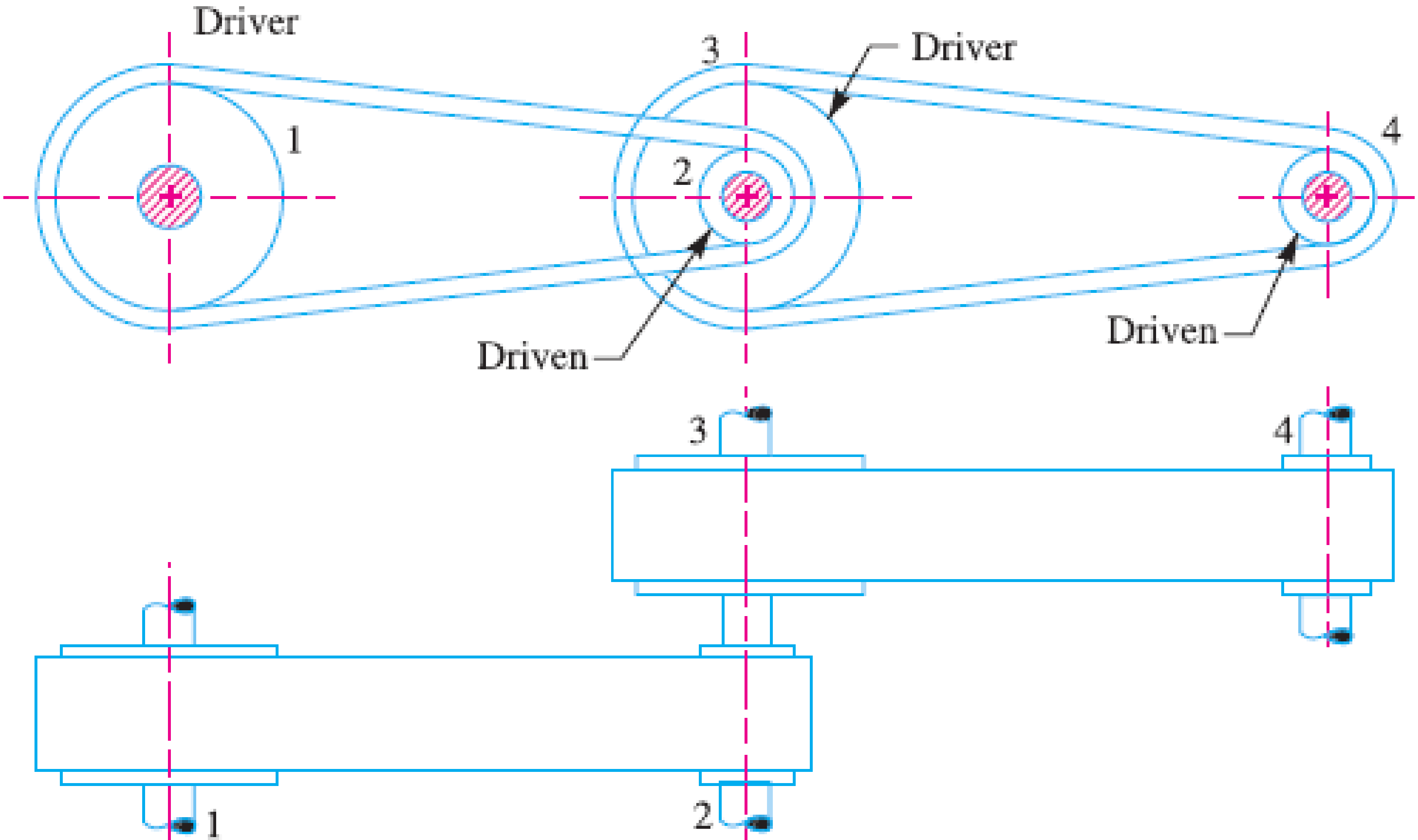
Flat Belt Drives



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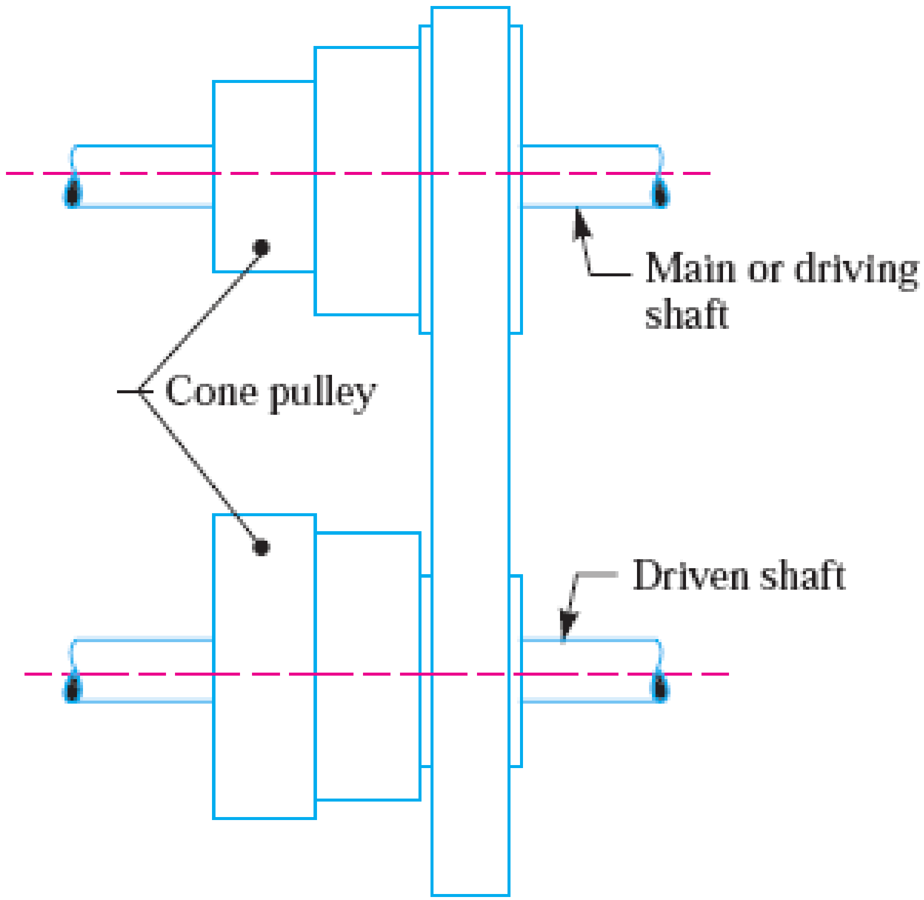
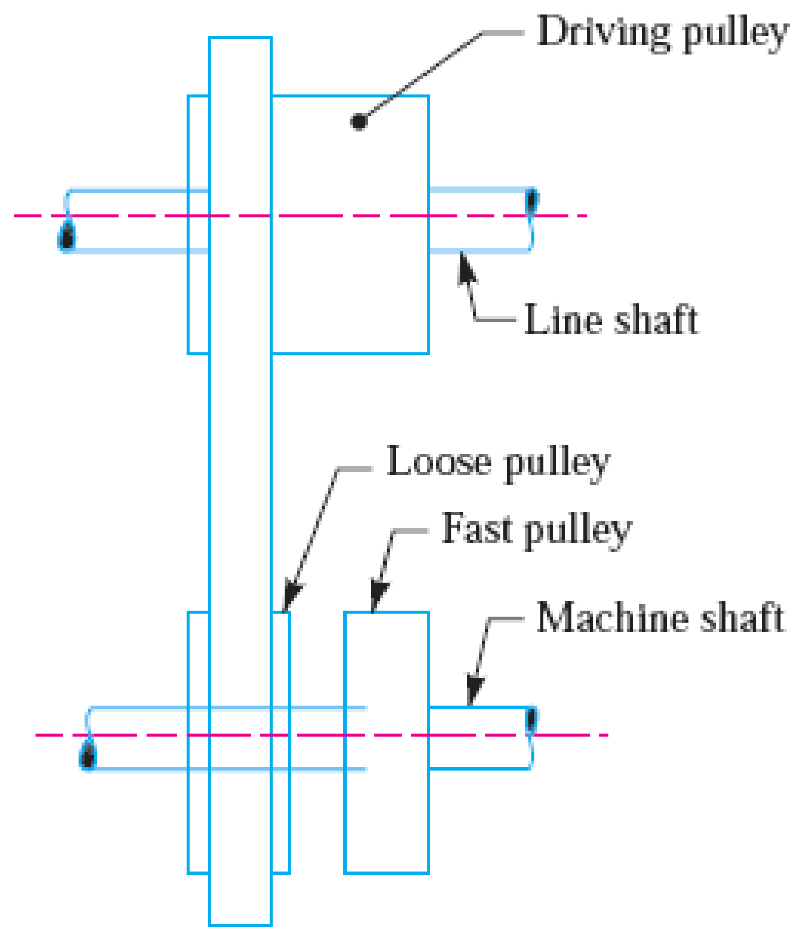


Main or driving shaft
Cone pulley
Driven shaft



Driving pulley
Line shaft
Loose pulley
Fast pulley
Machine shaft

Velocity Ratio of a Belt Drive

- It is the ratio between the velocities of the driver and the follower or driven.

d_1 = Diameter of the driver,
d_2 = Diameter of the follower,
N_1 = Speed of the driver in r.p.m.,
N_2 = Speed of the follower in r.p.m.,

Continue…

∴ Length of the belt that passes over the driver, in one minute

$$= \pi d_1 N_1$$

Similarly, length of the belt that passes over the follower, in one minute

$$= \pi d_2 N_2$$

Since the length of belt that passes over the driver in one minute is equal to the length of belt that passes over the follower in one minute, therefore

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

and velocity ratio, $$\frac{N_2}{N_1} = \frac{d_1}{d_2}$$

When thickness of the belt (t) is considered, then velocity ratio,

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

Continue…

- In case of a compound belt drive, the velocity ratio is given by

$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4} \quad \text{or} \quad \frac{\text{Speed of last driven}}{\text{Speed of first driver}} = \frac{\text{Product of diameters of drivers}}{\text{Product of diameters of drivens}}$$

Slip of the Belt

- The motion of belts and pulleys assuming a firm frictional grip between the belts and the pulleys. But sometimes, the frictional grip becomes insufficient. This may cause some forward motion of the driver without carrying the belt with it. This is called ***slip of the belt*** and is generally expressed as a percentage.

$s_1\,\%$ = Slip between the driver and the belt, and

$s_2\,\%$ = Slip between the belt and follower,

Continue…

∴ Velocity of the belt passing over the driver per second,

$$v = \frac{\pi\, d_1\, N_1}{60} - \frac{\pi\, d_1\, N_1}{60} \times \frac{s_1}{100}$$

$$= \frac{\pi\, d_1\, N_1}{60}\left(1 - \frac{s_1}{100}\right) \quad \ldots (i)$$

and velocity of the belt passing over the follower per second

$$\frac{\pi\, d_2\, N_2}{60} = v - v\left(\frac{s_2}{100}\right) = v\left(1 - \frac{s_2}{100}\right)$$

Substituting the value of v from equation (*i*), we have

Continue…

$$\frac{\pi\, d_2\, N_2}{60} = \frac{\pi\, d_1\, N_1}{60}\left(1 - \frac{s_1}{100}\right)\left(1 - \frac{s_2}{100}\right)$$

$$\therefore \quad \frac{N_2}{N_1} = \frac{d_1}{d_2}\left(1 - \frac{s_1}{100} - \frac{s_2}{100}\right) \qquad \ldots\left(\text{Neglecting } \frac{s_1 \times s_2}{100 \times 100}\right)$$

$$= \frac{d_1}{d_2}\left[1 - \left(\frac{s_1 + s_2}{100}\right)\right] = \frac{d_1}{d_2}\left(1 - \frac{s}{100}\right)$$

...(where $s = s_1 + s_2$ *i.e.* total percentage of slip)

If thickness of the belt (t) is considered, then

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}\left(1 - \frac{s}{100}\right)$$

Creep of Belt

- When the belt passes from the slack side to the tight side, a certain portion of the belt extends and it contracts again when the belt passes from the tight side to the slack side. Due to these changes of length, there is a relative motion between the belt and the pulley surfaces. This relative motion is termed as ***creep***.

$$\frac{N_2}{N_1} = \frac{d_1}{d_2} \times \frac{E + \sqrt{\sigma_2}}{E + \sqrt{\sigma_1}}$$

where σ_1 and σ_2 = Stress in the belt on the tight and slack side respectively, and

E = Young's modulus for the material of the belt.

Example: 1

- *An engine running at 150 r.p.m. drives a line shaft by means of a belt. The engine pulley is 750 mm diameter and the pulley on the line shaft is 450 mm. A 900 mm diameter pulley on the line shaft drives a 150 mm diameter pulley keyed to a dynamo shaft. Fine the speed of dynamo shaft, when 1. there is no slip, and 2. there is a slip of 2% at each drive*.

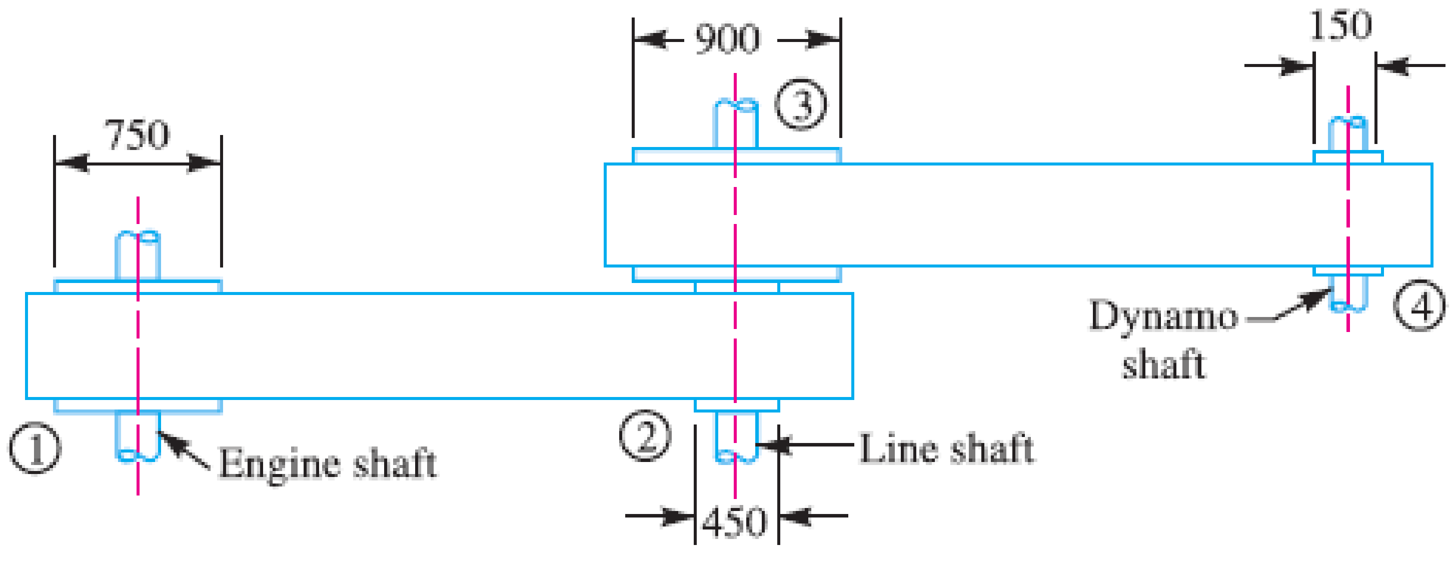
Solution: 1

- Given : $N1$ = 150 r.p.m. ;

  $d1$ = 750 mm ; $d2$ = 450 mm ;

  $d3$ = 900 mm ; $d4$ = 150 mm ;

  $s1$ = $s2$ = 2%



All dimensions in mm.

Continue…

- ***When there is no slip:***

$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4} \text{ or } \frac{N_4}{150} = \frac{750 \times 900}{450 \times 150} = 10$$

$$N_4 = 150 \times 10 = 1500 \text{ r.p.m. } \textbf{Ans.}$$

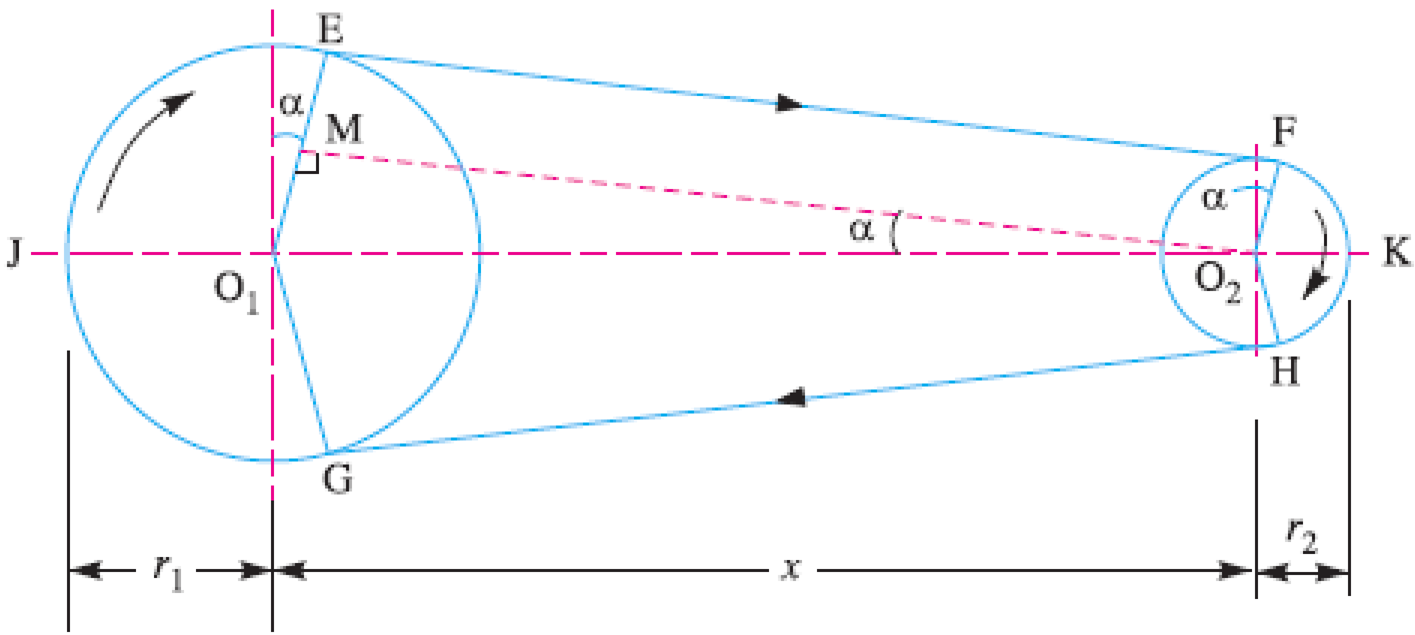
- ***When there is slip:***

$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4}\left(1 - \frac{s_1}{100}\right)\left(1 - \frac{s_2}{100}\right)$$

$$\frac{N_4}{150} = \frac{750 \times 900}{450 \times 150}\left(1 - \frac{2}{100}\right)\left(1 - \frac{2}{100}\right) = 9.6$$

$$N_4 = 150 \times 9.6 = 1440 \text{ r.p.m. } \textbf{Ans.}$$

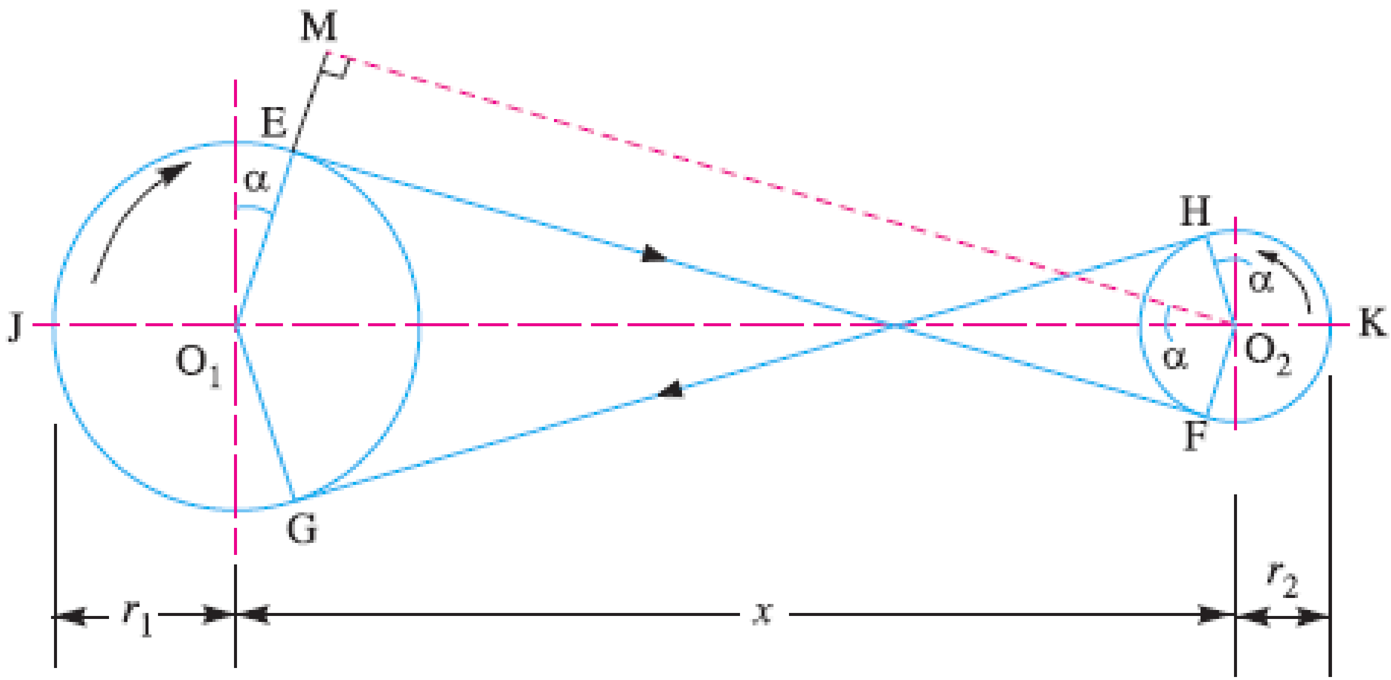
Length of a Open Belt Drive



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

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

Length of a Cross Belt Drive



.

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

.

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

Power Transmitted by a Belt

- power transmitted by belt; = $(T_1 - T_2)\, v$ W

$(\because 1 \text{ N-m/s} = 1\text{W})$

- A little consideration will show that torque exerted on the driving pulley is $(T_1 - T_2)\, r_1$. Similarly, the torque exerted on the driven pulley is $(T_1 - T_2)\, r_2$.

Ratio of Driving Tensions for Flat Belt Drive

T_1 = Tension in the belt on the tight side,

T_2 = Tension in the belt on the slack side, and

θ = Angle of contact in radians (*i.e.* angle subtended by the arc AB, along which the belt touches the pulley, at the centre).

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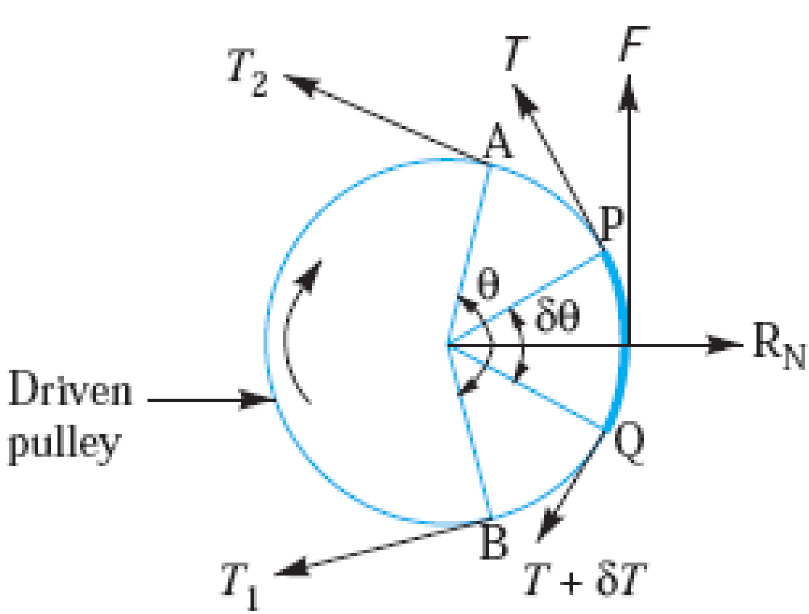
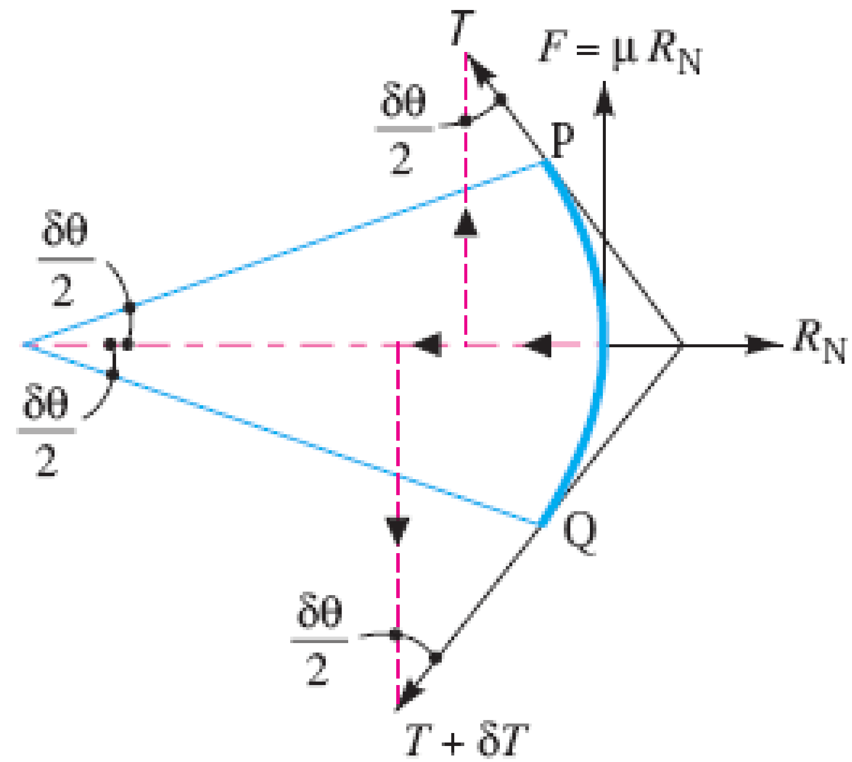
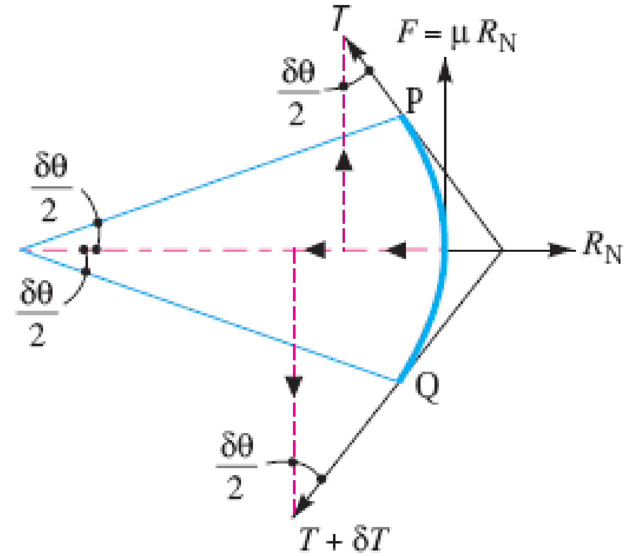




1. Tension T in the belt at P,
2. Tension $(T + \delta T)$ in the belt at Q,
3. Normal reaction R_N, and
4. Frictional force $F = \mu \times R_N$,

Continue…



Resolving all the forces horizontally, we have

$$R_N = (T + \delta T)\sin\frac{\delta\theta}{2} + T\sin\frac{\delta\theta}{2} \qquad \ldots(i)$$

Since the angle δθ is very small, therefore putting sin δθ/2 = δθ/2 in equation *(i)*, we have

$$R_N = (T + \delta T)\frac{\delta\theta}{2} + T\frac{\delta\theta}{2} = \frac{T.\delta\theta}{2} + \frac{\delta T.\delta\theta}{2} + \frac{T.\delta\theta}{2}$$

$$= T.\delta\theta \qquad \ldots\left(\text{Neglecting } \frac{\delta T.\delta\theta}{2}\right) \qquad \ldots(ii)$$

Now resolving the forces vertically, we have

$$\mu \times R_N = (T + \delta T)\cos\frac{\delta\theta}{2} - T\cos\frac{\delta\theta}{2} \qquad \ldots(iii)$$

Since the angle δθ is very small, therefore putting cos δθ/2 = 1 in equation *(iii)*, we have

$$\mu \times R_N = T + \delta T - T = \delta T \quad \text{or} \quad R_N = \frac{\delta T}{\mu} \qquad \ldots(iv)$$

Continue…

Equating the values of R_N from equations ***(ii)*** and ***(iv)***, we get

$$T.\delta\theta = \frac{\delta T}{\mu} \quad \text{or} \quad \frac{\delta T}{T} = \mu.\delta\theta$$

Integrating the above equation between the limits T_2 and T_1 and from 0 to θ, we have

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

$$\therefore \qquad \log_e \left(\frac{T_1}{T_2} \right) = \mu.\theta \quad \text{or} \quad \frac{T_1}{T_2} = e^{\mu.\theta} \qquad \text{...}(v)$$

The equation ***(v)*** can be expressed in terms of corresponding logarithm to the base 10, *i.e.*

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

Angle of Contact

Angle of contact or lap,

$$\theta = (180^\circ - 2\alpha)\frac{\pi}{180} \text{ rad}$$

$$= (180^\circ + 2\alpha)\frac{\pi}{180} \text{ rad}$$

- While determining the angle of contact, it must be remembered that it is the angle of contact at the smaller pulley, if both the pulleys are of the same material
- When the pulleys are made of different material (*i.e*. when the coefficient of friction of the pulleys or the angle of contact are different), then the design will refer to the pulley for which μ.θ is small

Example: 2

- *Two pulleys, one 450 mm diameter and the other 200 mm diameter, on parallel shafts 1.95 m apart are connected by a crossed belt. The larger pulley rotates at 200 rev/min, if the maximum permissible tension in the belt is 1 kN, and the coefficient of friction between the belt and pulley is 0.25?*
- *Find the (i) length of the belt required (ii) the angle of contact between the belt and each pulley. (iii) power can be transmitted by the belt*

Solution: 2

- Given : $d1 = 450$ mm = 0.45 m

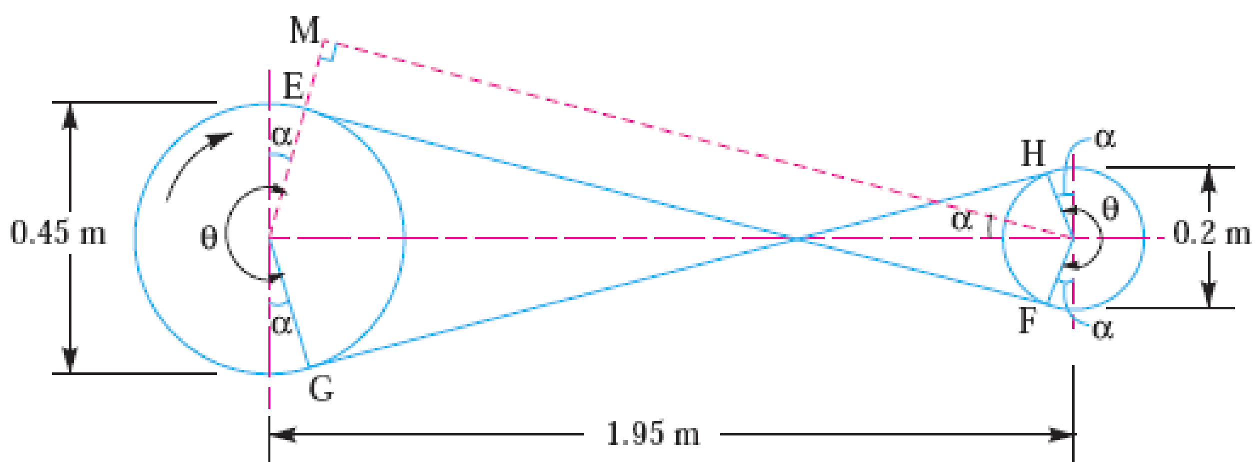
  or $r1 = 0.225$ m ;

  $d2 = 200$ mm = 0.2 m

  Or $r2 = 0.1$ m ; $x = 1.95$ m ;

  $N1 = 200$ r.p.m. ; $T1 = 1$ kN = 1000 N ;

  $\mu = 0.25$



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Length of the belt

We know that length of the belt,

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

$$= \pi (0.225 + 0.1) + 2 \times 1.95 + \frac{(0.225 + 0.1)^2}{1.95}$$

$$= 1.02 + 3.9 + 0.054 = 4.974 \text{ m}$$ **Ans.**

Continue…

Angle of contact between the belt and each pulley

Let θ = Angle of contact between the belt and each pulley.

We know that for a crossed belt drive,

$$\sin\alpha = \frac{r_1 + r_2}{x} = \frac{0.225 + 0.1}{1.95} = 0.1667$$

$\therefore$ $\alpha = 9.6°$

and $\theta = 180° + 2\alpha = 180 + 2 \times 9.6 = 199.2°$

$$= 199.2 \times \frac{\pi}{180} = 3.477 \text{ rad}$$ **Ans.**

Continue…

Power transmitted

Let T_1 = Tension in the tight side of the belt, and

T_2 = Tension in the slack side of the belt.

We know that

$$2.3 \log \left(\frac{T_1}{T_2} \right) = \mu.\theta = 0.25 \times 3.477 = 0.8693$$

$$\log \left(\frac{T_1}{T_2} \right) = \frac{0.8693}{2.3} = 0.378 \quad \text{or} \quad \frac{T_1}{T_2} = 2.387 \qquad \text{... (Taking antilog of 0.378)}$$

$$\therefore \qquad T_2 = \frac{T_1}{2.387} = \frac{1000}{2.387} = 419 \text{ N}$$

We know that the velocity of belt,

$$v = \frac{\pi \, d_1 N_1}{60} = \frac{\pi \times 0.45 \times 200}{60} = 4.713 \text{ m/s}$$

∴ Power transmitted,

$$P = (T_1 - T_2)\, v = (1000 - 419)\, 4.713 = 2738 \text{ W} = 2.738 \text{ kW}$$ **Ans.**

Centrifugal Tension in Belt Drive

- Since the belt continuously runs over the pulleys, therefore, some centrifugal force is caused, whose effect is to increase the tension on both the tight as well as the slack sides. The tension caused by centrifugal force is called ***centrifugal tension*.**

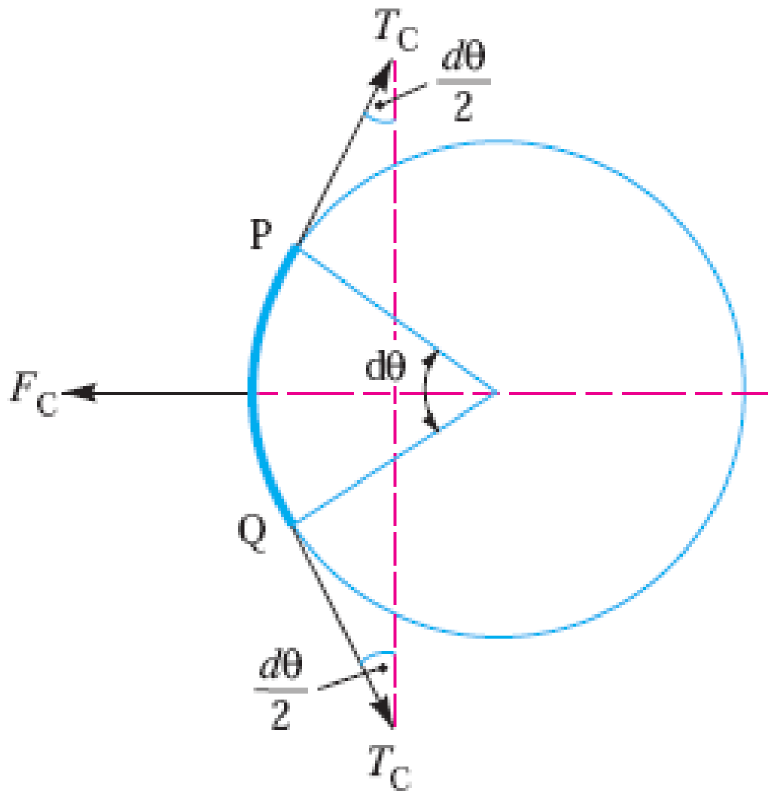
Continue…

m = Mass of belt per unit length in kg,

v = Linear velocity of belt in m/s,

r = Radius of pulley over which the belt runs in metres, and

T_C = Centrifugal tension acting tangentially at P and Q in newtons.



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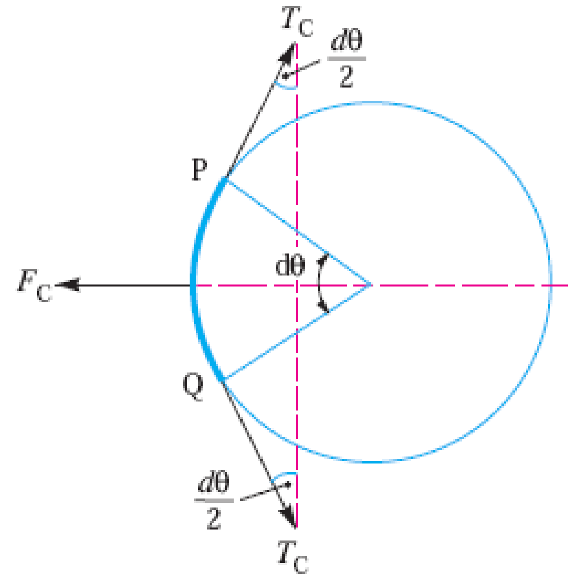


We know that length of the belt PQ

$$= r.d\theta$$

and mass of the belt PQ $= m.r.d\theta$

$\therefore$ Centrifugal force acting on the belt PQ,

$$F_C = m.r.d\theta \times \frac{v^2}{r} = m.d\theta.v^2$$

Resolving the forces (*i.e*. centrifugal force and centrifugal tension) horizontally

$$T_C \sin\left(\frac{d\theta}{2}\right) + T_C \sin\left(\frac{d\theta}{2}\right) = F_C = m.d\theta.v^2$$

$$2\,T_C\left(\frac{d\theta}{2}\right) = m.d\theta.v^2$$

$$T_C = m.v^2$$

$$\sin\left(\frac{d\theta}{2}\right) = \frac{d\theta}{2}$$

Maximum Tension in the Belt

σ = Maximum safe stress,

b = Width of the belt, and

t = Thickness of the belt.

$$T = \text{Maximum stress} \times \text{Cross-sectional area of belt} = \sigma . b . t$$

When centrifugal tension is neglected, then
$T = T1$, *i.e.* Tension in the tight side of the belt.

When centrifugal tension is considered, then
T (or $Tt1$) = $T1 + TC$

Condition for the Transmission of Maximum Power

$$P = (T_1 - T_2)\,v \quad \text{...}(i)$$

T_1 = Tension in the tight side in newtons,

T_2 = Tension in the slack side in newtons, and

v = Velocity of the belt in m/s.

$$\frac{T_1}{T_2} = e^{\mu\theta} \quad \text{or} \quad T_2 = \frac{T_1}{e^{\mu\theta}}$$

Substituting the value of T_2 in equation (*i*), we have

$$P = \left(T_1 - \frac{T_1}{e^{\mu\theta}}\right) v = T_1\left(1 - \frac{1}{e^{\mu\theta}}\right) v = T_1 . v . C \quad \text{..}(iii)$$

where

$$C = \left(1 - \frac{1}{e^{\mu\theta}}\right)$$

Continue…

The velocity of the belt for maximum power, (From Eq. (iv)

$$v = \sqrt{\frac{T}{3m}}$$

Also;

$T_1 = T - T_C$

T = Maximum tension to which the belt can be subjected in newtons, and

T_C = Centrifugal tension in newtons.

Substituting the value of T_1 in equation *(iii)*, we have

$$P = (T - T_C)\, v \times C$$

$$= (T - mv^2)\, v \times C = (T.v - m.v^3)\, C \qquad \text{... (Substituting } T_C = m.v^2)$$

For maximum power, differentiate the above expression with respect to v and equate to zero, *i.e.*

$$\frac{dP}{dv} = 0 \quad \text{or} \quad \frac{d}{dv}(T.v - m.v^3)\, C = 0$$

$$T - 3\, m.v^2 = 0 \qquad \text{...}(iv)$$

$$\therefore \quad T - 3\, T_C = 0 \quad \text{or} \quad T = 3T_C \qquad \text{... } (\because m.v^2 = T_C)$$

It shows that when the power transmitted is maximum, 1/3rd of the maximum tension is absorbed as centrifugal tension.

Example: 3

- *A leather belt 9 mm × 250 mm is used to drive a cast iron pulley 900 mm in diameter at 336 r.p.m. If the active arc on the smaller pulley is 120° and the stress in tight side is 2 MPa, find the* ***power*** *capacity of the belt considering C.F.. The density of leather may be taken as 980 kg/m3, and the coefficient of friction of leather on cast iron is 0.35*.

Solution: 3

- Given:
- t = 9 mm = 0.009 m ; b = 250 mm = 0.25 m; d = 900 mm = 0.9 m ;
- N = 336 r.p.m ;
- $\theta = 120° = 120 \times (\pi/180) = 2.1$ rad ;
- σ = 2 MPa = 2 N/mm2 ; ρ = 980 kg/m3 ;
- $\mu = 0.35$

Continue…

Hints:

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

$$v = \frac{\pi\, d.N}{60}$$

$$T_1 = T - T_C$$

$$T = \text{Maximum stress} \times \text{Cross-sectional area of belt} = \sigma.b.t$$

$$T_C = m.v^2 \qquad m = \text{Area} \times \text{length} \times \text{density} = b.t.l.\rho$$

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

Example: 4

- *Design a rubber belt to drive a dynamo generating 20 kW at 2250 r.p.m. and fitted with a pulley 200 mm diameter. Assume dynamo efficiency to be 85% and thickness 10 mm.*
- *Allowable stress for belt = 2.1 MPa*
- *Density of rubber = 1000 kg / m3*
- *Angle of contact for dynamo pulley = 165°*
- *Coefficient of friction between belt and pulley = 0.3*

Solution: 4

Given :

- P = 20 kW = 20 × 10^3 W ;
- N = 2250 r.p.m. ; d = 200 mm = 0.2 m ;
- η = 85% = 0.85 ;
- σ = 2.1 MPa = 2.1 × 106 N/m2 ;
- ρ = 1000 kg/m3 ;
- θ = 165° = 165 × π/180 = 2.88 rad ;
- μ = 0.3
- t = 10 mm

Continue…

Hints:

Calculation of "b": (Design "means")

$$T = T_1 + T_C = \text{Stress} \times \text{Area} = \sigma.b.t$$

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

$$P = (T_1 - T_2)\,v.\eta_d$$

$$T_C = m.v^2 :$$

$$m = \text{Area} \times \text{length} \times \text{density}$$

$$v = \frac{\pi\, d.N}{60}$$

Initial Tension in the Belt

- In order to increase frictionalgrip, the belt is tightened up. At this stage, even when the pulleys are stationary, the belt is subjected to some tension, called ***initial tension*****.**

T_0 = Initial tension in the belt,
T_1 = Tension in the tight side of the belt,
T_2 = Tension in the slack side of the belt, and
α = Coefficient of increase of the belt length per unit force.

Continue…

- the increase of tension in the tight side

  $= T_1 – T_0$

  and increase in the length of the belt on the tight side $= \alpha (T_1 – T_0)$
- Similarly, decrease in tension in the slack side

  $= T_0 – T_2$

  and decrease in the length of the belt on the slack side $= \alpha (T_0 – T_2)$

  From that;

$$\alpha (T_1 – T_0) = \alpha (T_0 – T_2)$$

Continue…

$$T_1 - T_0 = T_0 - T_2$$

$$T_0 = \frac{T_1 + T_2}{2}$$ ... (Neglecting centrifugal tension)

$$= \frac{T_1 + T_2 + 2T_C}{2}$$ ... (Considering centrifugal tension)

Example: 5

- *Two parallel shafts whose centre lines are 4.8 m apart, are connected by and open belt drive. The diameter of the larger pulley is 1.5 m and that of smaller pulley 1 m. The initial tension in the belt when stationary is 3 kN. The mass of the belt is 1.5 kg / m length. The coefficient of friction between the belt and the pulley is 0.3. Taking centrifugal tension into account, calculate the power transmitted, when the smaller pulley rotates at 400 r.p.m*.

Solution: 5

Given :

- x = 4.8 m ;
- d_1 = 1.5 m ; d_2 = 1 m ;
- T_0 = 3 kN = 3000 N ;
- m = 1.5 kg/m ;
- μ = 0.3 ;
- N_2 = 400 r.p.m.
- P = ?

Continue…

Hints:

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

$$v = \frac{\pi\, d_2 . N_2}{60}$$

$$T_0 = \frac{T_1 + T_2 + 2T_C}{2}$$

$$T_C = m.v^2$$

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

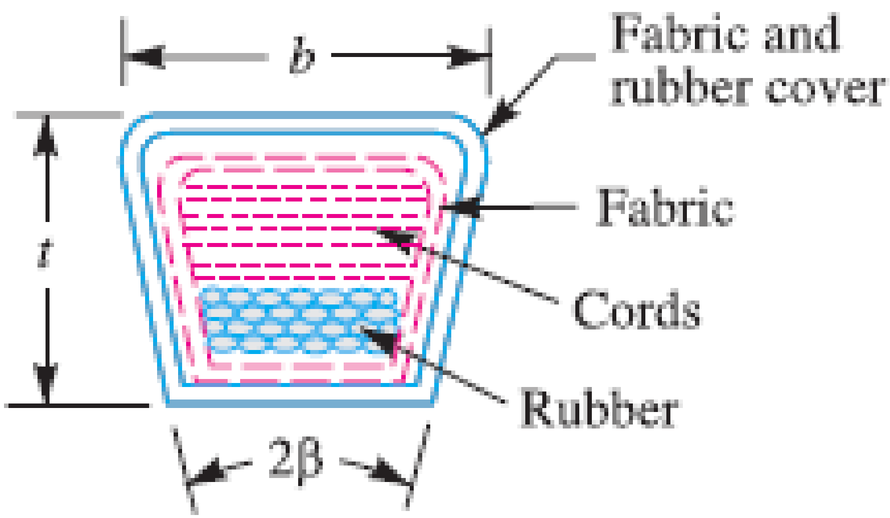
$$\theta = 180^\circ - 2\alpha \times \frac{\pi}{180}$$ **rad.**

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

V – Belt Drive Design

Concept:

- *V*-belt is mostly used in factories and workshops where a great amount of power is to be transmitted from one pulley to another when the two pulleys are very near to each other.



Continue...

Designation of V- Belt

Type of belt	Power ranges in kW	Minimum pitch diameter of pulley (D) mm	Top width (b) mm	Thickness (t) mm	Weight per metre length in newton
A	0.7 – 3.5	75	13	8	1.06
B	2 – 15	125	17	11	1.89
C	7.5 – 75	200	22	14	3.43
D	20 – 150	355	32	19	5.96
E	30 – 350	500	38	23	–

Advantages of V – Belt Drive

- Large power transmission
- Several belts can be used to increase power transmission
- Negligible slip
- Suitable for any shaft lay out
- High velocity ratio (7, 8 to 10)
- Compact

Disadvantages of V – Belt Drive

- Not suitable for large distance
- Complicated construction
- If $V > 50$ m/s, high C.F. developed and if $V < 5$ m/s not economical
- If one belt breaks, whole set must be replaced
- Not so durable

Design Equations

Ratio of Driving Tensions for V-belt:

$$2.3 \log (T_1 / T_2) = \mu.\theta \operatorname{cosec} \beta$$

$$\text{Number of } V\text{-belts} = \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}}$$

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

Example: 6

- *A belt drive consists of two V-belts in parallel, on grooved pulleys of the same size. The angle of the groove is 30°. The cross-sectional area of each belt is 750 mm2 and* μ = *0.12.*
- *Calculate the power that can be transmitted between pulleys of 300 mm diameter rotating at 1500 r.p.m. Find also the shaft speed in r.p.m. at which the power transmitted would be a maximum.*
- *The density of the belt material is 1.2 Mg / m3 and the maximum safe stress in the material is 7 MPa.*

Solution: 6

- Given : $n = 2$; $2\beta = 30°$ or $\beta = 15°$;
- $a = 750$ mm2 = 750×10^{-6} m2 ;
- $\mu = 0.12$;
- $\rho = 1.2$ Mg/m3 = 1200 kg/m3 ;
- $\sigma = 7$ MPa = 7×10^{6} N/m2 ;
- $d = 300$ mm = 0.3 m ;
- $N = 1500$ r.p.m.

Continue…

$$P = (T_1 - T_2)\, v \times n \qquad \Big| \qquad v = \frac{\pi d N}{60}$$

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \text{ cosec } \beta$$

$$T_1 = T - T_C$$

$$T = \sigma \times a$$

$$T_C = m.v^2$$

At maximum power, belt speed is;

$$v = \sqrt{\frac{T}{3m}} = \frac{\pi d N_1}{60}$$

Example: 7

- *A V-belt is driven on a flat pulley and a V-pulley. The drive transmits 20 kW from a 250 mm diameter V-pulley operating at 1800 r.p.m. to a 900 mm diameter flat pulley. The centre distance is 1 m, the angle of groove 40° and* μ = *0.2. If density of belting is 1110 kg / m3 and allowable stress is 2.1 MPa for belt material, what will be the number of belts required if C-size V-belts having 230 mm2 cross-sectional area are used.*

Solution: 7

Given :

- P = 20 kW ;
- $d1$ = 250 mm = 0.25 m ;
- $N1$ = 1800 r.p.m. ;
- $d2$ = 900 mm = 0.9 m ;
- x = 1 m = 1000 mm ;
- 2 β = 40° or β = 20° ;
- μ = 0.2 ;
- ρ = 1110 kg/m3 ;
- σ = 2.1 MPa = 2.1 N/mm2 ;
- a = 230 mm2 = 230 × 10–6 m2
- n = ?

Solution: 7

number of belts required

$$= \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}}$$

Power transmitted per belt

$$= (T_1 - T_2)\, v$$

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta$$

$$\theta = 180° - 2\alpha \times \frac{\pi}{180} \text{ rad.}$$

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

$$T_1 = T - T_C$$

$$T = \sigma \times a$$

$$T_C = m.v^2$$

$$m = \text{Area} \times \text{length} \times \text{density} = a \times l \times \rho$$

Design of Rope Drive

prof.nilesh pancholi

Design of Rope Drive

- The rope drives are widely used where a large amount of power is to be transmitted, from one pulley to another, over a considerable distance.
- The ropes drives use the following two types of ropes :

  1. Fibre ropes, and **2.** Wire ropes.
- The fibre ropes operate successfully when the pulleys are about 60 metres apart, while the wire
- ropes are used when the pulleys are upto 150 metres apart.

Advantages of fibre rope

The fibre rope drives have the following advantages :

1. They give smooth, steady and quiet service.

2. They are little affected by out door conditions.

3. The shafts may be out of strict alignment.

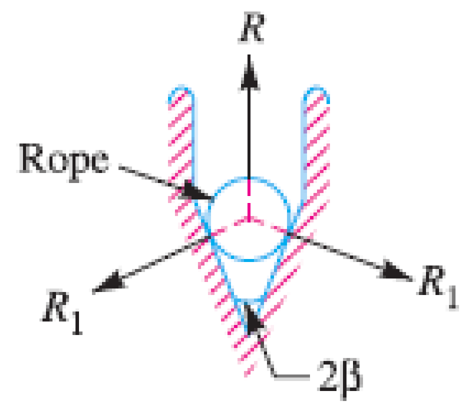
4. The power may be taken off in any direction and in fractional parts of the whole amount.

5. They give high mechanical efficiency.

Design of Fibre Rope Drive

Ratio of Driving Tensions for rope:

$$2.3 \log (T_1 / T_2) = \mu.\theta \operatorname{cosec} \beta$$



Number of ropes required

$$= \frac{\text{Total power transmitted}}{\text{Power transmitted per rope}}$$

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

Example: 8

- *A rope pulley with 10 ropes and a peripheral speed of 1500 m / min transmits 115 kW. The angle of lap for each rope is 180° and the angle of groove is 45°. The coefficient of friction between the rope and pulley is 0.2. Assuming the rope to be just on the point of slipping, find the tension in the tight and slack sides of the rope. The mass of each rope is 0.6 kg per metre length.*

Solution: 8

- Given :
- $n = 10$; $v = 1500$ m/min = 25 m/s ;
- $P = 115$ kW = 115 × 103 W ;
- $\theta = 180° = \pi$ rad ;
- $2\beta = 45°$ or $\beta = 22.5°$;
- $\mu = 0.2$;
- $m = 0.6$ kg / m

Continue…

$$T_{t1} = T_1 + T_C$$

$$T_{t2} = T_2 + T_C$$

$$T_C = m\,v^2$$

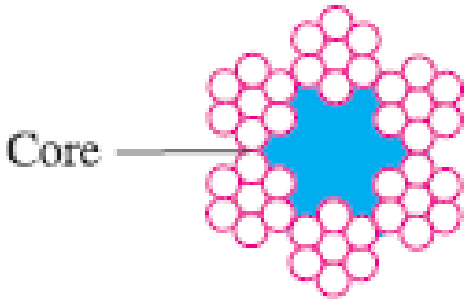
$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta$$

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

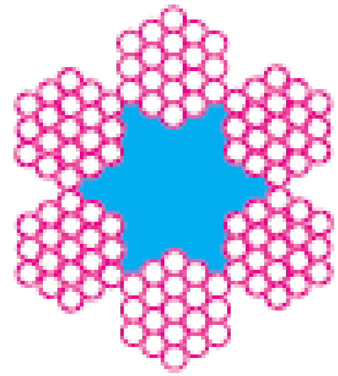
Design of Wire Rope Drive

- When a large amount of power is to be transmitted over long distances from one pulley to another (*i.e*. when the pulleys are upto 150 metres apart), then wire ropes are used. The wire ropes are widely used in elevators, mine hoists, cranes, conveyors, hauling devices and suspension bridges.

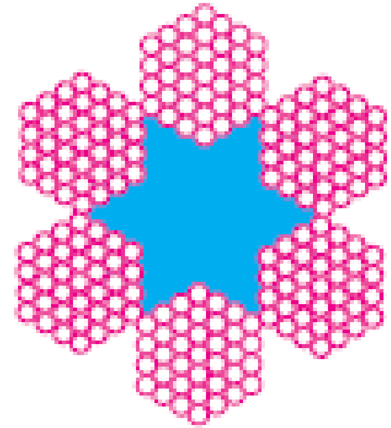
Cross section of rope



(a) 6 × 7 rope.



(b) 6 × 19 rope.



(c) 6 × 37 rope.

Standard designation	*Application*
6 × 7 rope	It is a standard coarse laid rope used as haulage rope in mines, tramways, power transmission.
6 × 19 rope	It is a standard hoisting rope used for hoisting purposes in mines, quarries, cranes, dredges, elevators, tramways, well drilling.
6 × 37 rope	It is an extra flexible hoisting rope used in steel mill laddles, cranes, high speed elevators.
8 × 19 rope	It is also an extra flexible hoisting rope.

Advantages of Wire Ropes

- These are lighter in weight,
- These offer silent operation,
- These can withstand shock loads,
- These are more reliable,
- These are more durable,
- They do not fail suddenly,
- The efficiency is high, and
- The cost is low.

Stresses in Wire Ropes

1. ***Direct stress due to axial load lifted and weight of the rope:***

W = Load lifted,

w = Weight of the rope, and

A = Net cross-sectional area of the rope.

$$\sigma_d = \frac{W + w}{A}$$

2. *Bending stress when the rope winds round the sheave or drum*.

$$\sigma_b = \frac{E_r \times d_w}{D}$$

$$W_b = \frac{\pi}{4}(d_w)^2\, n \times \sigma_b$$

n= No. of ropes

Equivalent bending load on the rope,

$$W_b = \sigma_b \times A = \frac{E_r \times d_w \times A}{D}$$

E_r = Modulus of elasticity of the wire rope,

d_w = Diameter of the wire,

D = Diameter of the sheave or drum, and

A = Net cross-sectional area of the rope.

3. *Stresses during starting and stopping:*

During starting and stopping, the rope and the supported load are to be accelerated. This induces additional load and stress in the rope which is given by

$$W_a = \frac{W + w}{g} \times a$$

$$\sigma_a = \frac{W + w}{g} \times \frac{a}{A}$$

a = Acceleration of the rope and load, and

g = Acceleration due to gravity.

The time (*t*) necessary to attain a speed (*v*) is known, then the value of '*a*' is given by *a* = *v* / 60 *t*

- In case of starting, due to slack (h); impact load induced is;

$$W_{st} = (W + w)\left[1 + \sqrt{1 + \frac{2a \times h \times E_r}{\sigma_d \times l \times g}}\right]$$

and velocity of the rope (v_r) at the instant when the rope is taut,

$$v_r = \sqrt{2a \times h}$$

where a = Acceleration of the rope and load,

h = Slackness in the rope, and

l = Length of the rope.

When there is no slackness in the rope, then $h = 0$ and $v_r = 0$, therefore

Impact load during starting,

$$W_{st} = 2\,(W + w)$$

and the corresponding stress,

$$\sigma_{st} = \frac{2\,(W + w)}{A}$$

4. *Stress due to change in speed:*

The additional stress due to change in speed may be obtained in the similar way as discussed previously in which the acceleration is given by;

$$a = (v2 - v1) / t$$

where ($v2 - v1$) is the change in speed in m/s and t is the time in seconds.

5. *Effective stress:*

The sum of the direct stress (σd) and the bending stress (σb) is called the effective stress in the rope during normal working

Effective stress in the rope during normal working

$$= \sigma_d + \sigma_b$$

Effective stress in the rope during starting

$$= \sigma_{st} + \sigma_b$$

effective stress in the rope during acceleration of the load

$$= \sigma_d + \sigma_b + \sigma_a$$

Design Procedure a Wire Rope

- First of all, select a suitable type of rope from design data book for the given application.
- Find the design load by assuming a factor of safety 2 to 2.5 times the factor of safety
- Find the diameter of wire rope (*d*) by equating the tensile strength of the rope selected to the design load.
- Find the diameter of the wire (*dw*) and area of the rope (*A*) from design data.
- Find the various stresses (or loads) in the rope as discussed
- Find the effective stresses (or loads) during normal working, during starting and during acceleration of the load.
- Now find the actual factor of safety and compare with the factor of safety given in design data. If the actual factor of safety is within permissible limits, then the design is safe.

Example: 9

- *Select a wire rope for a vertical mine hoist to lift a load of 55 kN from a depth 300 metres. A rope speed of 500 metres / min is to be attained in 10 seconds.*

Solution: 9

1. From design data, we find that the wire ropes for haulage purposes in mines are of two types, *i.e*. 6 × 7 and 6 × 19. Let us take a rope of type 6 × 19.

Type of rope	*Nominal diameter (mm)*	*Average weight (N/m)*	*Tensile strength (N)*	
			Tensile strength of wire	
			1600 MPa	*1800 MPa*
6 × 7	8, 9, 10, 11, 12, 13, 14, 16 18, 19, 20, 21, 22, 24, 25 26, 27, 28, 29, 31, 35	$0.0347\ d^2$	$530\ d^2$	$600\ d^2$
6 × 19	13, 14, 16, 18, 19, 20, 21 22, 24, 25, 26, 28, 29, 32 35, 36, 38	$0.0363\ d^2$	$530\ d^2$	$595\ d^2$

2. From design data, we find that the factor of safety for mine hoists from 300 to 600 m depth is 7. Since the design load is calculated by taking a factor of safety 2 to 2.5 times the factor of safety given data book, take the factor of safety as 15.

Design load for the wire rope

$= 15 \times 55 = 825$ kN $= 825\ 000$ N

Application of wire rope	*Factor of safety*	*Application of wire rope*	*Factor of safety*
Track cables	4.2	Derricks	6
Guys	3.5	Haulage ropes	6
Mine hoists : Depths		Small electric and air hoists	7
upto 150 m	8	Over head and gantry cranes	6
300 – 600 m	7	Jib and pillar cranes	6
600 – 900 m	6	Hot ladle cranes	8
over 900 m	5	Slings	8
Miscellaneous hoists	5		

3.From design data, the tensile strength of 6 × 19 rope made of wire with tensile strength of 1800 MPa is 595 d^2 (in newton), where d is the diameter of rope in mm. Equating this tensile strength to the design load,

$595\ d^2 = 825\ 000$; $d^2 = 825\ 000 / 595 = 1386.5$ or $d = 37.2$

Type of rope	*Nominal diameter (mm)*	*Average weight (N/m)*	*Tensile strength (N)*	
			Tensile strength of wire	
			1600 MPa	*1800 MPa*
6 × 7	8, 9, 10, 11, 12, 13, 14, 16 18, 19, 20, 21, 22, 24, 25 26, 27, 28, 29, 31, 35	$0.0347\ d^2$	$530\ d^2$	$600\ d^2$
6 × 19	13, 14, 16, 18, 19, 20, 21 22, 24, 25, 26, 28, 29, 32 35, 36, 38	$0.0363\ d^2$	$530\ d^2$	$595\ d^2$

4. From design data, for a 6 × 19 rope, Diameter of wire,

$d_w = 0.063\, d = 0.063 \times 38 = 2.4$ mm and area of rope,

$A = 0.38\, d^2 = 0.38\, (38)^2 = 550\ \text{mm}^2$

Type of wire rope	6 × 8	6 × 19	6 × 37	8 × 19
Wire diameter (d_w)	0.106 d	0.063 d	0.045 d	0.050 d
Area of wire rope (A)	0.38 d^2	0.38 d^2	0.38 d^2	0.35 d^2

5. Calculation of the various loads in the rope:

(*a*) From design data, weight of the rope,

$w = 0.0363\ d^2 \times \text{depth} = 0.0363\ (38)^2 \times 300$

= 15 720 N ... (As Depth = 300 m)

Type of rope	*Nominal diameter (mm)*	*Average weight (N/m)*	*Tensile strength (N)* *Tensile strength of wire*	
			1600 MPa	*1800 MPa*
6 × 7	8, 9, 10, 11, 12, 13, 14, 16 18, 19, 20, 21, 22, 24, 25 26, 27, 28, 29, 31, 35	$0.0347\ d^2$	$530\ d^2$	$600\ d^2$
6 × 19	13, 14, 16, 18, 19, 20, 21 22, 24, 25, 26, 28, 29, 32 35, 36, 38	$0.0363\ d^2$	$530\ d^2$	$595\ d^2$

(b) Bending Load (Wb): From design data , diameter of the sheave (*D*) may be taken as 60 to 100 times the diameter of rope (*d*).

$$D = 100\,d = 100 \times 38 = 3800 \text{ mm}$$

.

$$\sigma_b = \frac{E_r \times d_w}{D} = \frac{84 \times 10^3 \times 2.4}{3800} = 53 \text{ N/mm}^2$$

$$W_b = \sigma_b \times A = 53 \times 550 = 29\,150 \text{ N} \quad \text{...(Taking } E_r = 84 \times 10^3 \text{ N/mm}^2\text{)}$$

Type of wire rope	*Recommended sheave diameter (D)*		*Uses*
	Minimum sheave diameter	*Preferred sheave diameter*	
6 × 7	42 *d*	72 *d*	Mines, haulage tramways.
6 × 19	30 *d*	45 *d*	Hoisting rope.
	60 *d*	100 *d*	Cargo cranes, mine hoists
	20 *d*	30 *d*	Derricks, dredges, elevators, tramways, well drilling.
6 × 37	18 *d*	27 *d*	Cranes, high speed elevators and small shears.
8 × 19	21 *d*	31 *d*	Extra flexible hoisting rope.

(*c*) We know that the acceleration of the rope and load,

$$a = v / 60t = 500 / 60 \times 10 = 0.83 \text{ m/s}^2$$

∴ Additional load due to acceleration,

$$W_a = \frac{W + w}{g} \times a = \frac{55\,000 + 15\,720}{9.81} \times 0.83 = 5983 \text{ N}$$

(*d*) We know that the impact load during starting (when there is no slackness in the rope),

$$W_{st} = 2\,(W + w) = 2(55\,000 + 15\,720) = 141\,440 \text{ N}$$

6. Comparison of Actual FOS with FOS design data:

- The effective load on the rope during normal working (*i.e*. during uniform lifting or lowering of the load)

  $= W + w + W_b = 55\,000 + 15\,720 + 29150 = 99\,870$ N

$\therefore$ Actual factor of safety during normal working

$$= \frac{825\,000}{99\,870} = 8.26$$

Effective load on the rope during starting

$$= W_{st} + W_b = 141\,440 + 29\,150 = 170\,590 \text{ N}$$

∴ Actual factor of safety during starting

$$= \frac{825\,000}{170\,590} = 4.836$$

Effective load on the rope during acceleration of the load (*i.e.* during first 10 seconds after starting)

$$= W + w + W_b + W_a$$

$$= 55\,000 + 15\,7\,20 + 29\,150 + 5983 = 105\,853 \text{ N}$$

∴ Actual factor of safety during acceleration of the load

$$= \frac{825\,000}{105\,853} = 7.8$$

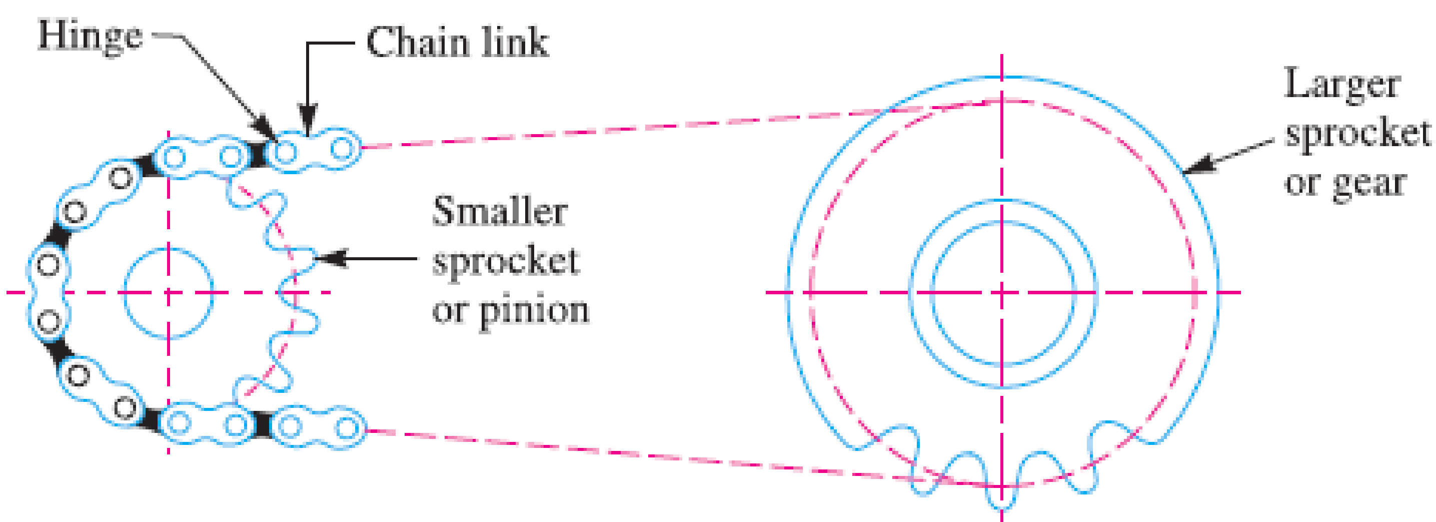
Since the actual factor of safety as calculated above are safe, therefore a wire rope of diameter 38 mm and 6 × 19 type is satisfactory. **Ans.**

Design of Chain Drive

prof.nilesh pancholi

Design of Chain Drive

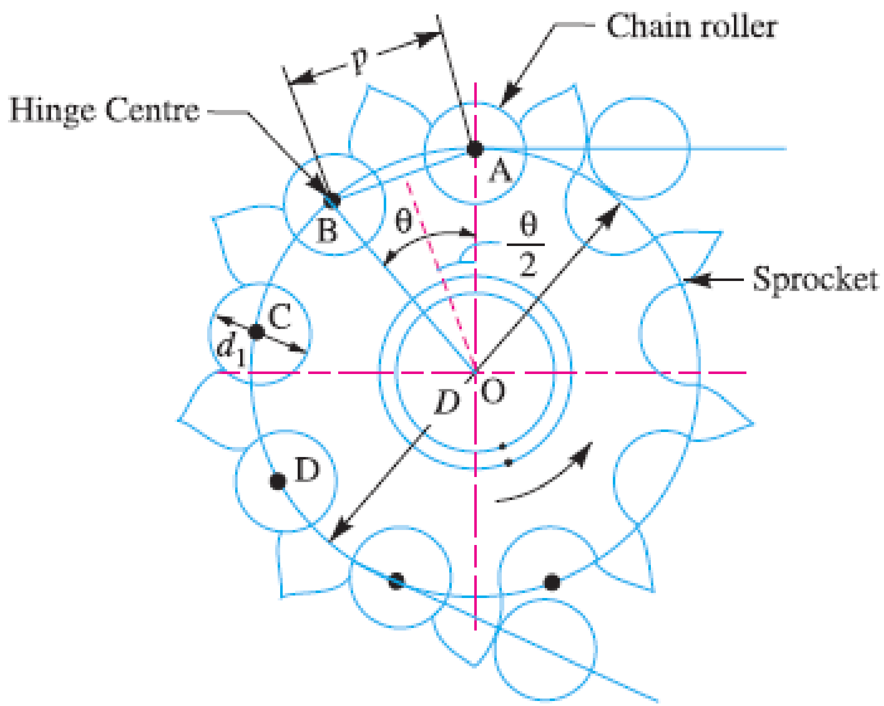
- The chains are mostly used to transmit motion and power from one shaft to another, when the centre distance between their shafts is short such as in bicycles, motor cycles, agricultural machinery, conveyors, rolling mills, road rollers etc. The chains may also be used for long centre distance of upto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



Terminology in Chain Drive

1. ***Pitch of chain (p):*** It is the distance between the hinge centre of a link and the corresponding hinge centre of the adjacent link

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



Relation between p and D

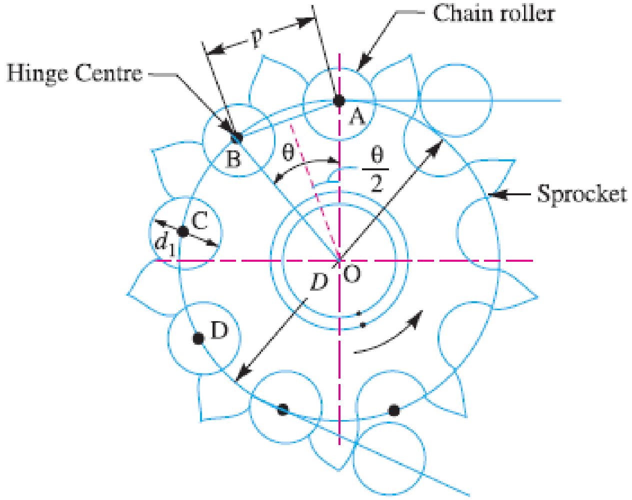
D = Diameter of the pitch circle, and
T = Number of teeth on the sprocket.

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

$$\theta = \frac{360^\circ}{T}$$

$$p = D\sin\left(\frac{360^\circ}{2T}\right) = D\sin\left(\frac{180^\circ}{T}\right)$$

$$D = p\,\text{cosec}\left(\frac{180^\circ}{T}\right)$$



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.

Velocity Ration in Chain Drive

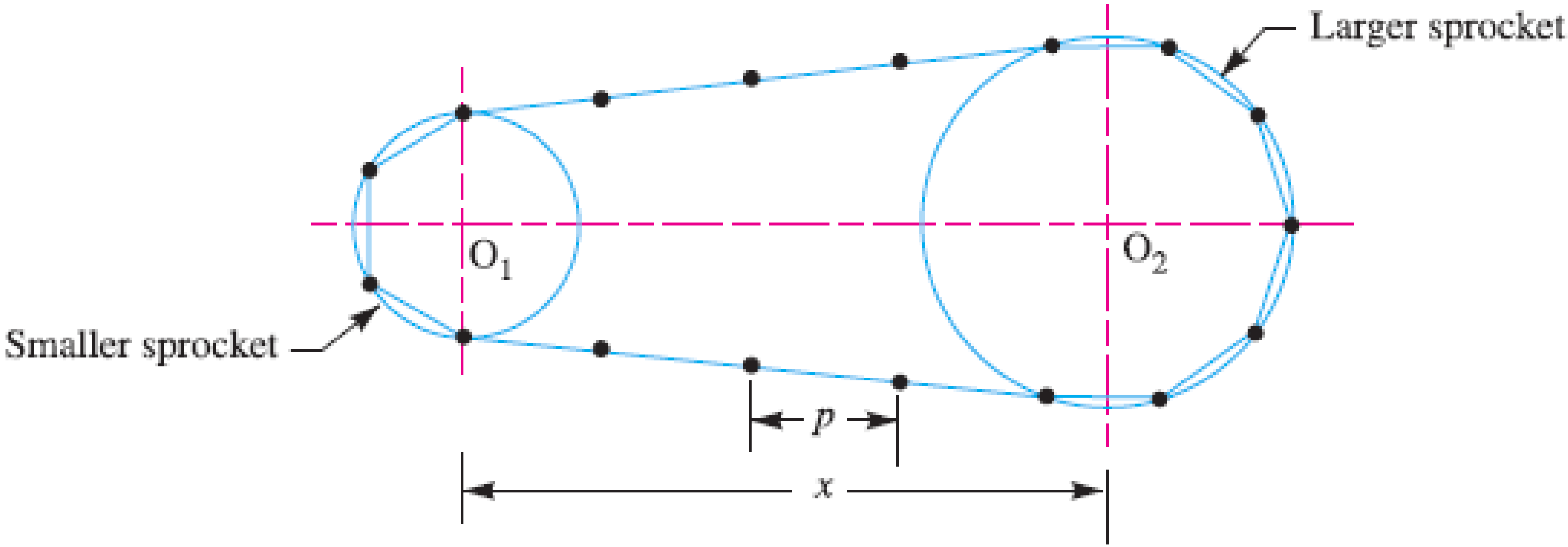
$$V.R. = \frac{N_1}{N_2} = \frac{T_2}{T_1}$$

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.

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

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

Length of Chain & Centre Distance



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.

K = No. of Chain links

$$L = K.p$$

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

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

Factor of Safety for Chain Drive

- 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,

  Factor of safety = W_B / W

Where;

$W_B = 106\,p^2$ (in newtons) for roller chains

$= 106\,p$ (in newtons) per mm width of chain for silent chains.

- Total Load; W = Ft + Fc + Fs

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

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

F_S = $k.mg.x$ (in newtons)

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 40°

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

Power Transmitted by Chain

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.

Data for Selection of Factors

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

Design Procedure for Chain Drive

Example:

- *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: 10

- Given :
- Rated power = 15 kW ;
- $N1$ = 1000 r.p.m ;
- $N2$ = 350 r.p.m.
- x = 500 mm
- Operating Hours = 16 Hrs.

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

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

- **2.** Select the minimum number of teeth on the smaller sprocket or pinion from design data

From design data, 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.

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

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

Number of teeth on the larger sprocket or gear,

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

- **4.** Determine the design power by using the service factor:

  Design power = Rated power × Service factor

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

$$= 1.25$$

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

and design power $= 15 \times 1.875 = 28.125$ kW

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

- From data, 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

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	–

p = 19.05 mm

Roller diameter, d = 12.07 mm

Minimum width of roller = w = 11.68 mm

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

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

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

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

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

and pitch circle diameter of the larger sprocket or gear

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

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

Pitch line velocity of the smaller sprocket,

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

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

W = Rated power / Pitch line velocity

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

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 design data.

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

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

This value is more than the value given in design data, which is equal to 11.

10. Fix the centre distance between the sprockets.

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,
$= 30\,p = 30 \times 19.05 = 572$ mm

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

Correct centre distance
$x = 572 - 4 = 568$ mm

11. Determine the length of the chain.

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

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

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

∴ Length of the chain,

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

12. The other dimensions may be fixed as standard proportions.

1. Tooth flank radius (r_e)

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

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

where d_1 = Roller diameter, and

T = Number of teeth.

2. Roller seating radius (r_i)

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

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

3. Roller seating angle (α)

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

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

4. Tooth height above the pitch polygon (h_a)

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

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

5. Pitch circle diameter (D)

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

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})

$$= 0.93\ b_1 \text{ when } p \leq 12.7 \text{ mm}$$

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

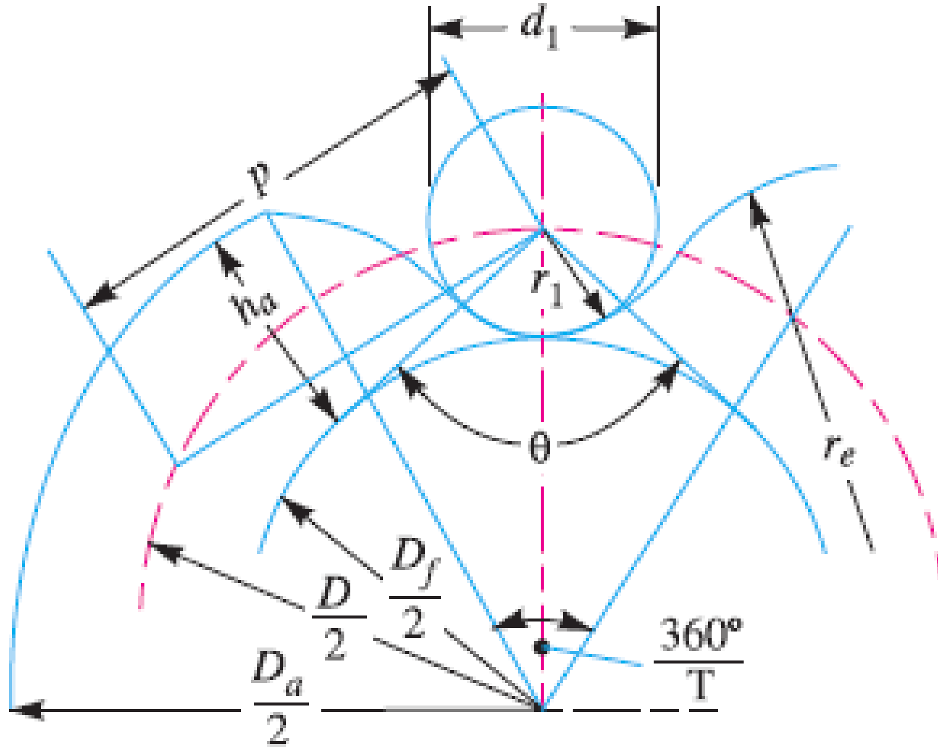
9. Tooth side radius (r_x) = p

10. Tooth side relief (b_a)

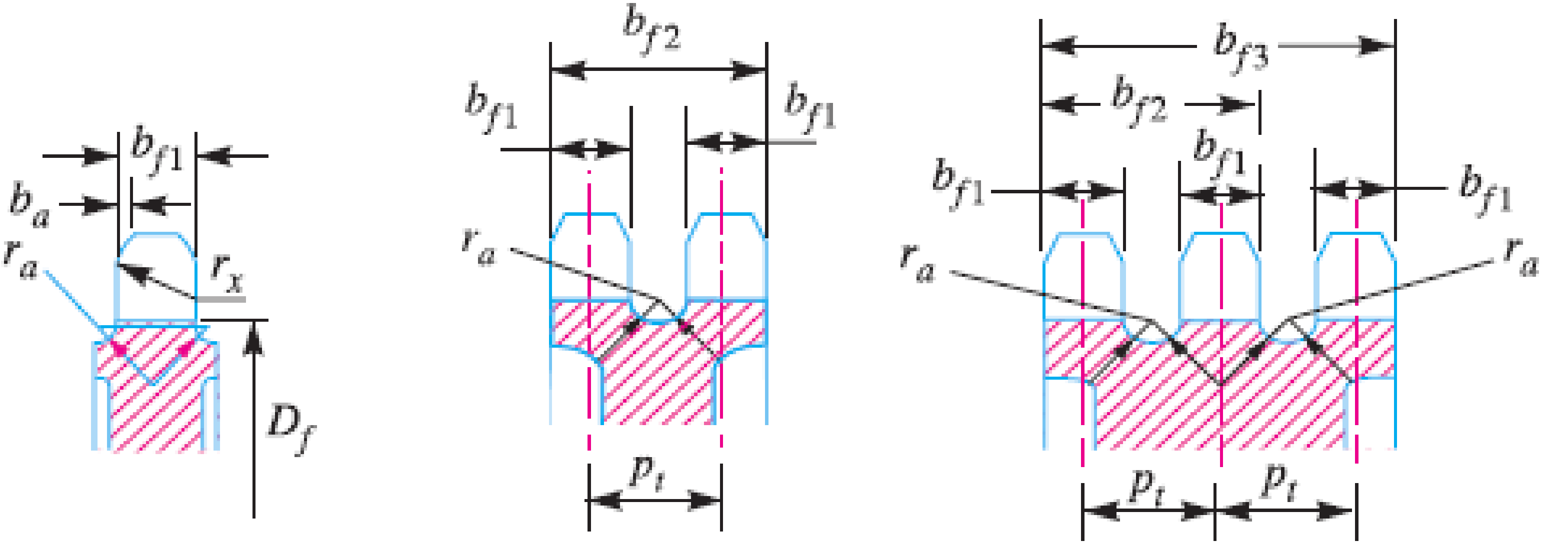
$$= 0.1\ p \text{ to } 0.15\ p$$

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

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



(*a*) Tooth profile of sprocket.



THANKS