POWER TRANSMISSION SYSTEMS

Introduction

The belts are used to transmit power from one shaft to another shaft by means of pulleys which rotate at the same speed or at different speeds. Belt drives are suitable when the power source (e.g. motor) is at some distance away from the load. These could be used for torque and speed conversion like gears. A belt is a strip of rubber or some other flexible material that is looped over two or more pulleys. They are used as a simple and efficient way to transmit power between two rotating shafts. Belts have very high power transmission efficiency at around 95%. Belts are inexpensive and easy to design. The maintenance of these devices is also easy. The elasticity present in the belts can provide damping and shock absorption which results in less vibration. Usually the belts have a composite structure. They have a rubber or a synthetic surface for providing a sufficient amount of friction. In order to provide increased tensile strength the belts are reinforced with steel wires.

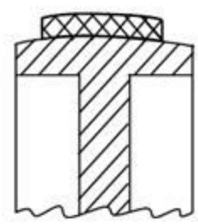
Types of Belts

Though there are many types of belts used these days, yet the following are important from the subject point of view:

1. Flat belt.

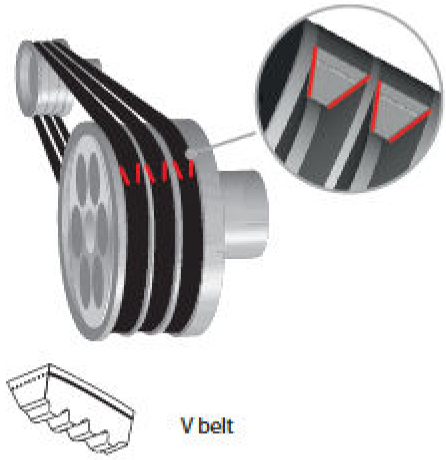
The flat belt as shown in Fig., is mostly used in the factories and work-shops, where a moderate amount of power is to be transmitted, from one pulley to another when the two pulleys are not more than 8 meters apart.

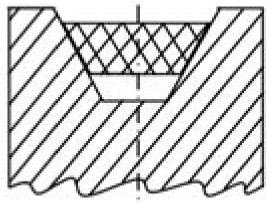
Flat belt



2. V- belt.

The V-belt as shown in Fig. 18.1 (b), is mostly used in the 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.

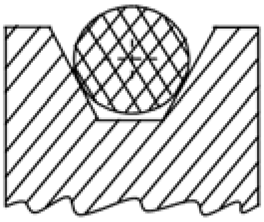




- ✓ Belt is shaped in a Shaped in a 'V'
- ✓ V belt allows higher torques to be transmitted
- ✓ The pulley circumference has grooves that would mate with the V-belt. These grooves wedge the belt at higher loads, allowing more torque to be placed on the belt.
- ✓ The grooves solve the problem of slipping and misalignment
- ✓ For higher power requirement, two or more belts can be joined side-by-side to form a multi-V belt
- ✓ When a belt cannot be specified, a linked V-belt can be used, which is made up of rubber links held together by metal fasteners. However, these are weaker and runs at slower speeds

3. Circular belt or rope.

The circular belt or rope as shown in Fig. 18.1 (c) is mostly used in the factories and workshops, where a great amount of power is to be transmitted, from one pulley to another, when the two pulleys are more than 8 meters apart

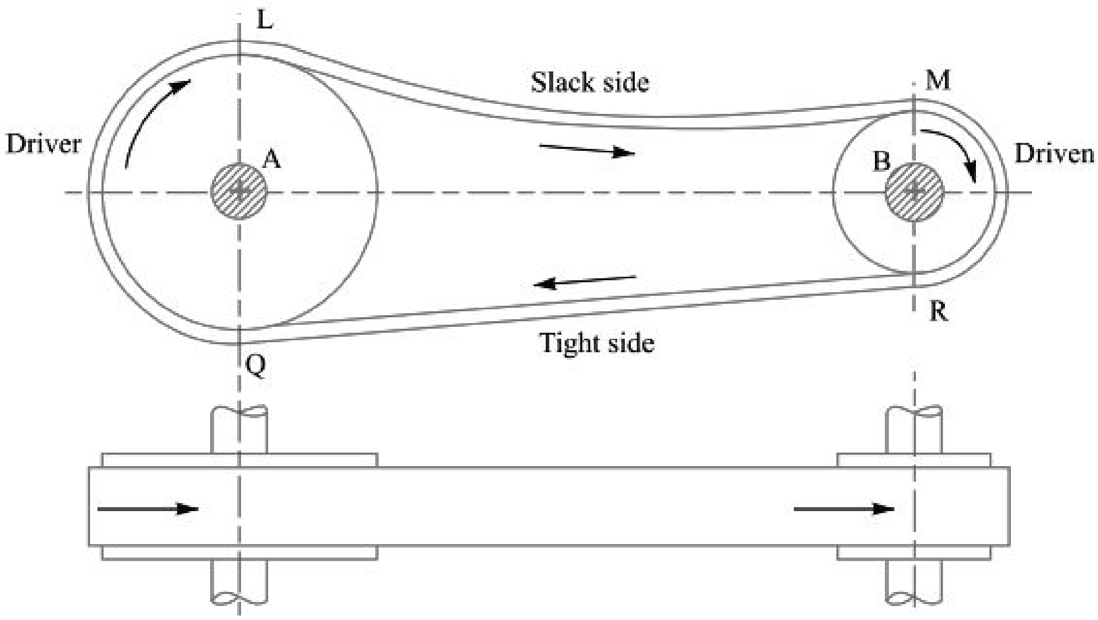


Types of Flat Belt Drives

The power from one pulley to another may be transmitted by any of the following types of belt drives.

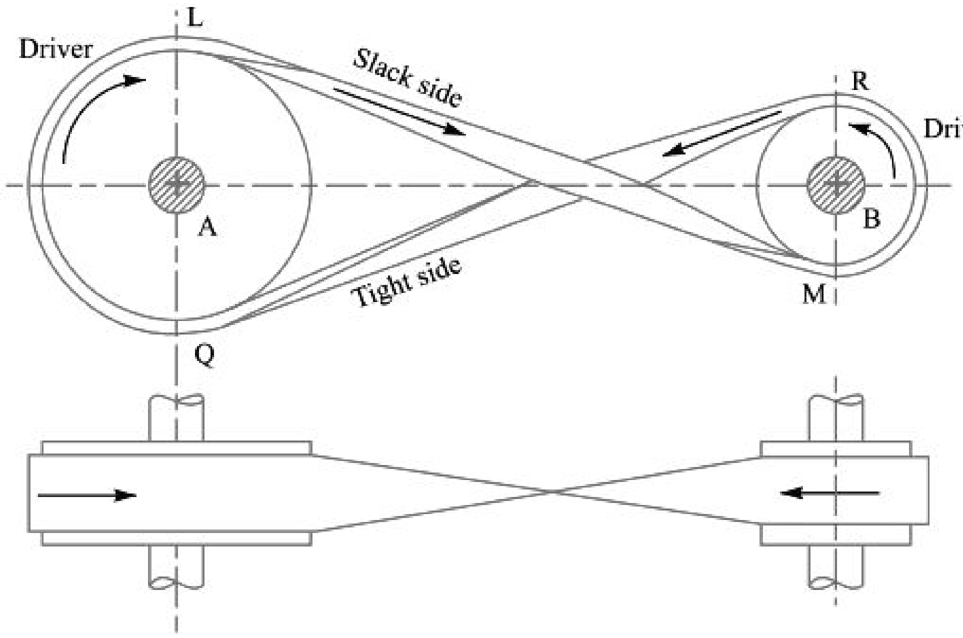
1. Open belt drive.

The open belt drive, as shown in Fig. 18.4, is used with shafts arranged parallel and rotating in the same direction. In this case, the driver A pulls the belt from one side (i.e. lower side RQ) and delivers it to the other side (i.e. upper side LM).Thus the tension in the lower side belt will be more than that in the upper side belt. The lower side belt (because of more tension) is known as tight side whereas the upper side belt (because of less tension) is known as slack side, as shown in Fig. 18.4.



2. Crossed or twist belt drive.

The crossed or twist belt drive, as shown in Fig. 18.5, is used with shafts arranged parallel and rotating in the opposite directions. In this case, the driver pulls the belt from one side (i.e. RQ) and delivers it to the other side (i.e. LM). Thus, the tension in the belt RQ will be more than that in the belt LM. The belt RQ (because of more tension) is known as tight side, whereas the belt LM (because of less tension) is known as slack side, as shown in Fig. 18.5.



Velocity Ratio of a Belt Drive

It is the ratio between the velocities of the driver and the follower or driven. It may be expressed, mathematically, as discussed below:

Let 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.,

∴ 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

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

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

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

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

Notes : 1.

The velocity ratio of a belt drive may also be obtained as discussed below:

We know that the peripheral velocity of the belt on the driving pulley,

ν1=11m/s60

dNπ

and peripheral velocity of the belt on the driven pulley,

ν2=22m/s60

dNπ

When there is no slip, thenν1 = ν2.

∴ 112221

12or6060

dNdNNdNdππ==2.

In case of a compound belt drive as shown in Fig. 18.7, the velocity ratio is given by 413

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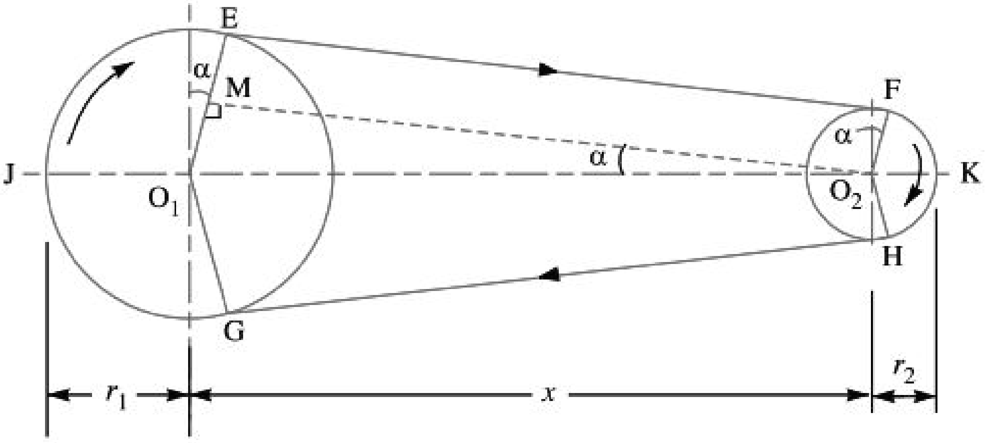
Ndd Ndd ×=× or

SpeedoflastdrivenProductofdiametersofdrivers=

SpeedoffirstdriverProductofdiametersofdrivens

Length of the open belt:



Let

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

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

L = Total length of the belt.

The belt leaves the larger pulley at E and G and the smaller pulley at F and H as shown in Fig. Through O_2 draw O_2M parallel to FE.

From the geometry of the figure, we find that O2M will be perpendicular to O1E.

Let the angle MO_2O_1 = α radians.

We know that the length of the belt,

L = Arc GJE + EF + Arc FKH + HG

= 2 (Arc JE + EF + Arc FK) .... (i)

From the geometry of the figure, we also find that

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

Since the angle α is very small, therefore putting

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

..................(ii)

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

Arc FK = $r_2\left[\frac{\pi}{2} - \alpha\right]$ ....(iv)

And EF = $\sqrt{OO_1^2 - O_1M^2} = \sqrt{x^2 - (r_1 - r_2)^2} = \sqrt{x^2 - \left(\frac{r_1 - r_2}{x}\right)^2}$

Expanding this equation by binomial theorem, we have

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

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

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

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

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

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

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

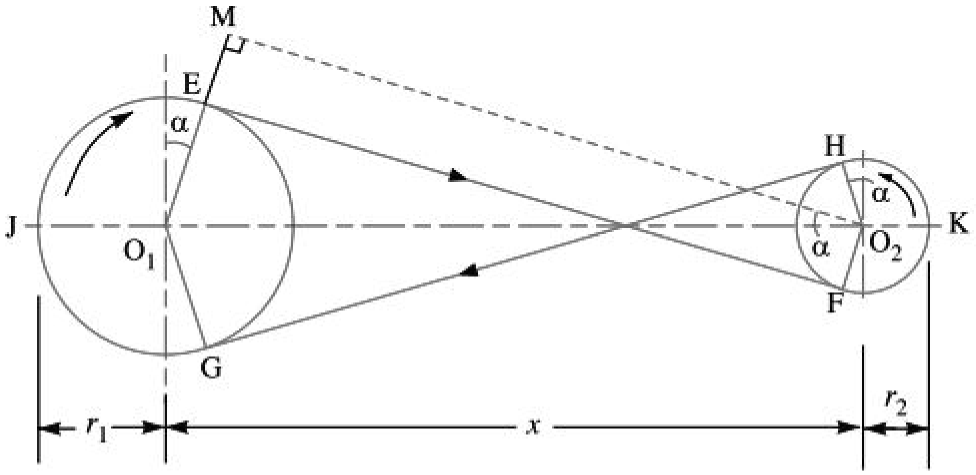
$$= \pi(r_1 + r_2) + 2\left(\frac{r_1 - r_2}{x}\right)(r_1 - r_2) + 2x - \frac{(r_1 - r_2)^2}{x}$$

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

$$= \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x} \text{ .... (Interms of radii)}$$

$$= \frac{\pi}{2}(d_1 + d_2) + 2x + \frac{(d_1 - d_2)^2}{4x} \text{ .... (Interms of diameter)}$$

Length of the cross belt:



Let

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

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

L = Total length of the belt.

The belt leaves the larger pulley at E and G and the smaller pulley at F and H as shown in Fig. Through O_2 draw O_2M parallel to FE.

From the geometry of the figure, we find that O2M will be perpendicular to O1E.

Let the angle MO_2O_1 = α radians.

We know that the length of the belt,

L = Arc GJE + EF + Arc FKH + HG

= 2 (Arc JE + EF + Arc FK) .... (i)

From the geometry of the figure, we also find that

$$\text{Sin } \alpha = \frac{O_1M}{O_1O_2} = \frac{O_1E - EM}{O_1O_2} = \frac{r_1 + r_2}{x}$$

Since the angle α is very small, therefore putting

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

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

Arc FK $= r_2\left[\frac{\pi}{2} + \alpha\right]$ ....(iv)

And EF $= \sqrt{OO_1^2 - O_1M^2} = \sqrt{x^2 - (r_1 - r_2)^2} = \sqrt{x^2 - \left(\frac{r_1 - r_2}{x}\right)^2}$

Expanding this equation by binomial theorem, we have

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

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

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

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

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

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

Sub $\alpha = \frac{r_1 + r_2}{x}$

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

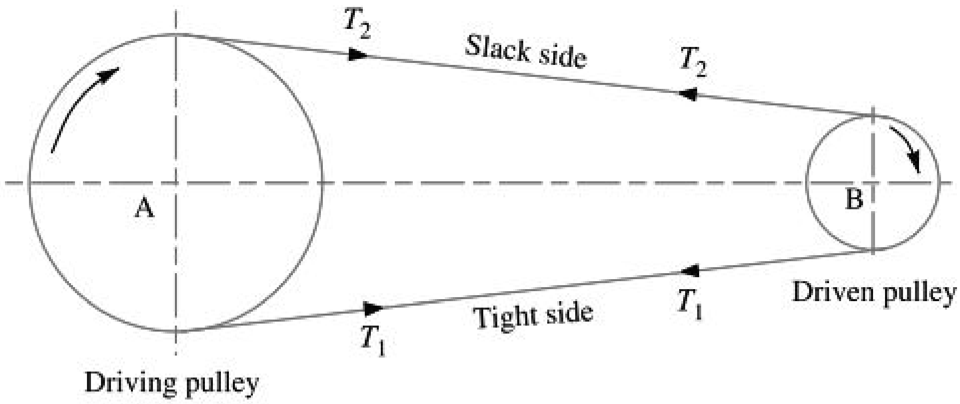
$$= \pi(r_1 + r_2) + \frac{2(r_1 + r_2)^2}{x} + 2x - \frac{(r_1 + r_2)^2}{x}$$

$$= \pi(r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x} \quad \ldots\ldots\ldots\ldots \text{(Interms of radii)}$$

$$= \frac{\pi}{2}(d_1 + d_2) + 2x + \frac{(d_1 + d_2)^2}{4x} \quad \ldots\ldots\ldots\ldots \text{(Interms of diameter)}$$

Power Transmitted by a Belt

Fig. shows the driving pulley (or driver) A and the driven pulley (or follower) B. As already discussed, the driving pulley pulls the belt from one side and delivers it to the other side. It is thus obvious that the tension on the former side (i.e. tight side) will be greater than the latter side (i.e. slack side) as shown in Fig. 18.15.



Let

T_1 and T_2 = Tensions in the tight side and slack side of the belt respectively in newtons,

r_1 and r_2 = Radii of the driving and driven pulleys respectively in metres, and

v = Velocity of the belt in m/s.

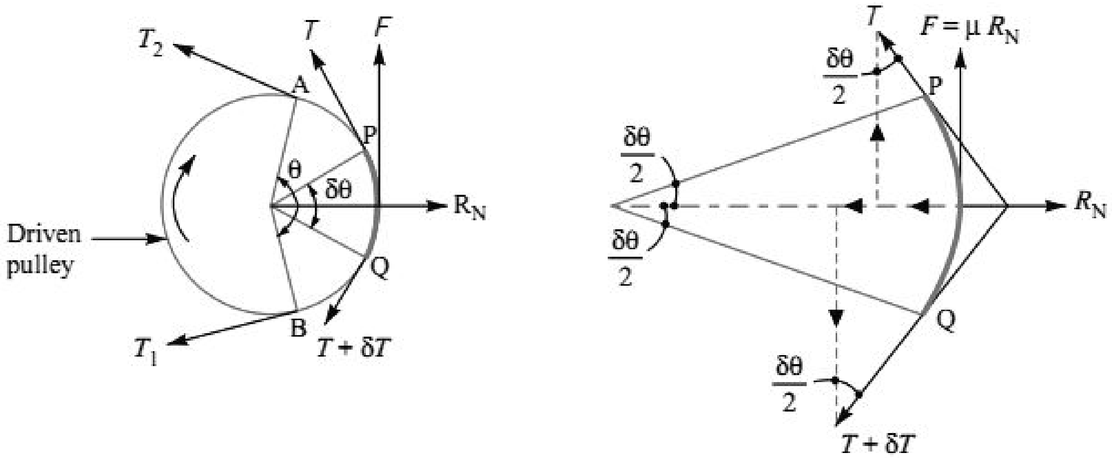
The effective turning (driving) force at the circumference of the driven pulley or follower is the difference between the two tensions (i.e. T1 – T2).

∴ Work done per second = $(T_1 - T_2)\, v$ N-m/s

and power transmitted = $(T1 - T2)\, v$ W ... (Q 1 N-m/s = 1W)

Ratio of Driving Tensions for Flat Belt Drive

Consider a driven pulley rotating in the clockwise direction as shown in Fig.



Let

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).

Now consider a small portion of the belt PQ, subtending an angle δθ at the centre of the pulley as shown in Fig. 18.16. The belt PQ is in equilibrium under the following forces:

1. Tension T in the belt at P,

2. Tension (T + δT) in the belt at Q,

3. Normal reaction RN, and

4. Frictional force $F = \mu \times R_N$, (where μ is the coefficient of friction between the belt and pulley.)

Design of belt flat belt

1. velocity of the belt $v = \frac{\pi\, d.N}{60}$
2. cross-sectional area of the belt, *a = b.t*
3. Power transmitted = $(T_1 - T_2)$ v W
4. Maximum or total tension in the tight side of the belt, $T = T_{t1} = \sigma.a$
5. We know that mass of the belt per metre length, m = Area × length × density = b×t×l×ρ
6. Centrifugal tension $T_C = m.v^2$
7. Centrifugal tension is considered, then
8. T (or T_{t1}) = $T_1 + T_C$ T (or T_{t2}) = $T_2 + T_C$
9. The relation between the tight side and slack side tensions, in terms of coefficient of friction and the angle of contact $2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta$
10. length of the belt
$$L = \pi\,(r_2 + r_1) + 2x + \frac{(r_2 - r_1)^2}{x}$$
11. Initial Tension in the Belt $T_0 = \frac{T_1 + T_2 + 2T_C}{2}$

Advantages and Disadvantages of V-belt Drive over Flat Belt Drive

Following are the advantages and disadvantages of the *V-belt drive over flat belt drive:*

Advantages

1. The *V-belt drive gives compactness due to the small distance between centres of pulleys.*
2. The drive is positive, because the slip between the belt and the pulley groove is negligible.
3. Since the *V-belts are made endless and there is no joint trouble, therefore the drive is smooth.*
4. It provides longer life, 3 to 5 years.
5. It can be easily installed and removed.
6. The operation of the belt and pulley is quiet.
7. The belts have the ability to cushion the shock when machines are started.
8. The high velocity ratio (maximum 10) may be obtained.
9. The wedging action of the belt in the groove gives high value of limiting ratio of tensions.
10. The *V-belt may be operated in either direction, with tight side of the belt at the top or* bottom. The centre line may be horizontal, vertical or inclined.

***Disadvantage**s*

1. The *V-belt drive can not be used with large centre distances, because of larger weight per* unit length.
2. The *V-belts are not so durable as flat belts.*
3. The construction of pulleys for V-belts is more complicated than pulleys of flat belts.
4. Since the *V-belts are subjected to certain amount of creep, therefore these are not suitable* for constant speed applications such as synchronous machines and timing devices.
5. The belt life is greatly influenced with temperature changes, improper belt tension and mismatching of belt lengths.
6. The centrifugal tension prevents the use of *V-belts at speeds below 5 m/ s and above 50 m / s.*

Design of V – belt

1. velocity of the belt $v = \frac{\pi\, d.N}{60}$
2. cross-sectional area of the belt, *a = b.t*
3. Power transmitted = $(T_1 - T_2)$ v * n W
4. Maximum or total tension in the tight side of the belt, T = T_{t1} = σ.a
5. We know that mass of the belt per metre length, m = Area × length × density = b×t×l×ρ
6. Centrifugal tension $T_C = m.v^2$
7. Centrifugal tension is considered, then
8. T (or T_{t1}) = $T_1 + T_C$ T (or T_{t2}) = $T_2 + T_C$
9. The relation between the tight side and slack side tensions, in terms of coefficient of friction and the angle of contact $2.3 \log (T_1 / T_2) = \mu.\theta \operatorname{cosec} \beta$
10. length of the belt

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

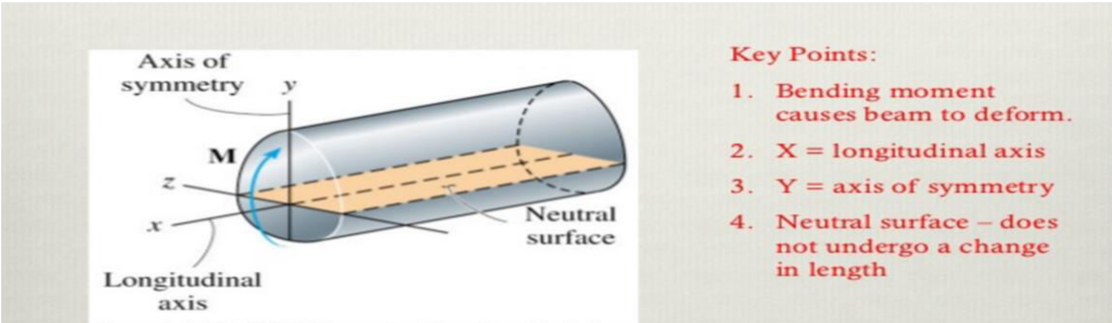
The following procedure may be followed while designing a wire rope.

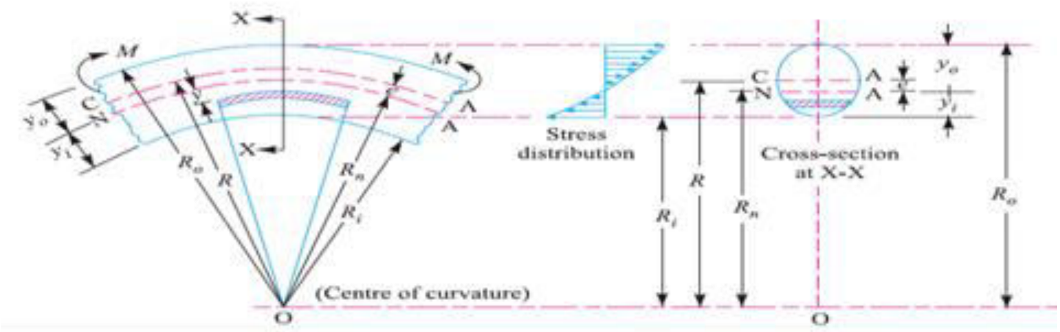
1. First of all, select a suitable type of rope for the given application.
2. Find the design load by assuming a factor of safety 2 to 2.5 times the factor of
3. Find the diameter of wire rope (d) by equating the tensile strength of the rope selected to the design load.
4. Find the diameter of the wire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

CURVED BEAMS

In straight beams, the **neutral axis of the section coincides with its centroidal axis** and the stress distribution in the beam is linear.

But in case of curved beams, the **neutral axis of the cross-section is shifted towards the centre of curvature** of the beam.





The general expression for the bending stress (σ_b)

$$\sigma_b = \frac{M}{A \cdot e}\left(\frac{y}{R_n - y}\right)$$

M = Bending moment acting at the given section about the centroidal axis,

A = Area of cross-section,

e = Distance from the centroidal axis to the neutral axis = $R - R_n$,

R = Radius of curvature of the centroidal axis,

R_n = Radius of curvature of the neutral axis, and

y = Distance from the neutral axis to the fibre under consideration. It is positive for the distances towards the centre of curvature and negative for the distances away from the centre of curvature.

$$\sigma_{bi} = \frac{M \cdot y_i}{A \cdot e \cdot R_i}$$

where y_i = Distance from the neutral axis to the inside fibre = $R_n - R_i$, and

R_i = Radius of curvature of the inside fibre.

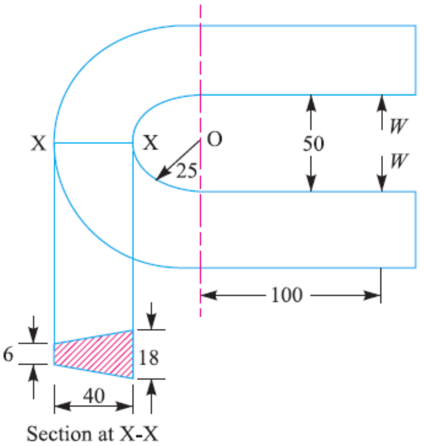
The maximum bending stress at the outside fibre is given by

$$\sigma_{bo} = \frac{M \cdot y_o}{A \cdot e \cdot R_o}$$

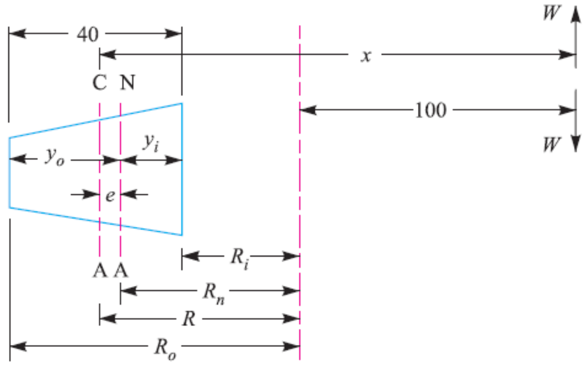
where y_o = Distance from the neutral axis to the outside fibre = $R_o - R_n$, and

R_o = Radius of curvature of the outside fibre.

It may be noted that the bending stress at the inside fibre is ***tensile*** while the bending stress at the outside fibre is ***compressive***.



All dimensions in mm.



All dimensions in mm.

Design procedure for curved beams:

1. Radius of curvature of the neutral axis $\mathbf{R_n}$
2. Radius of curvature of the centroidal axis **R**
3. Distance between the centroidal axis and neutral axis $\mathbf{e = R - R_n}$
4. The distance between the load and centroidal axis, $\mathbf{x = 100 + R}$
5. Bending moment about the centroidal axis M = **W.x**
6. Direct tensile stress $\boldsymbol{\sigma_t}$
7. Distance from the neutral axis to the inner surface $\mathbf{y_i = R_n - R_i}$
8. Distance from the neutral axis to the outer surface $\mathbf{y_o = R_o - R_n}$
9. Maximum bending stress at the inner surface $\boldsymbol{\sigma_{bi}}$
10. Maximum bending stress at the outer surface $\boldsymbol{\sigma_{b0}}$
11. Resultant stress on the inner surface = $\boldsymbol{\sigma_t + \sigma_{bi}}$
12. Resultant stress on the outer surface= $\boldsymbol{\sigma_t - \sigma_{bo}}$