Example 20.5. *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. Given : $P = 20$ kW ; $d_1 = 250$ mm $= 0.25$ m ; $N_1 = 1800$ r.p.m. ; $d_2 = 900$ mm $= 0.9$ m ; $x = 1$ m $= 1000$ mm ; $2\beta = 40°$ or $\beta = 20°$; $\mu = 0.2$; $\rho = 1110$ kg/m³ ; $\sigma = 2.1$ MPa $= 2.1$ N/mm² ; $a = 230$ mm² $= 230 \times 10^{-6}$ m²

Fig. 20.5 shows a V-flat drive. First of all, let us find the angle of contact for both the pulleys. From the geometry of the Fig. 20.5, we find that

$$\sin\alpha = \frac{O_2M}{O_1O_2} = \frac{r_2 - r_1}{x} = \frac{d_2 - d_1}{2x} = \frac{900 - 250}{2 \times 1000} = 0.325$$

$\therefore \quad \alpha = 18.96°$

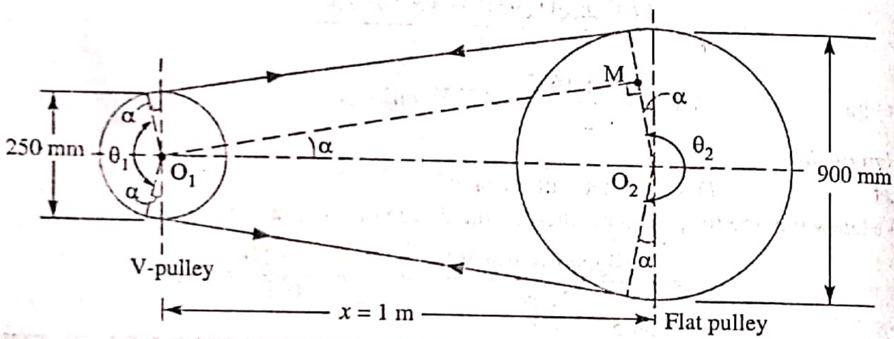


Fig. 20.5

We know that angle of contact on the smaller or V-pulley,

$$\theta_1 = 180° - 2\alpha = 180° - 2 \times 18.96 = 142.08°$$
$$= 142.08 \times \pi/180 = 2.48 \text{ rad}$$

and angle of contact on the larger or flat pulley,

$$\theta_2 = 180° + 2\alpha = 180° + 2 \times 18.96 = 217.92°$$
$$= 217.92 \times \pi / 180 = 3.8 \text{ rad}$$

We have already discussed that when the pulleys have different angle of contact (θ), then the design will refer to a pulley for which μ.θ is small.

We know that for a smaller or V-pulley,

$$\mu.\theta = \mu.\theta_1 \operatorname{cosec}\beta = 0.2 \times 2.48 \times \operatorname{cosec} 20° = 1.45$$

and for larger or flat pulley,

$$\mu.\theta = \mu.\theta_2 = 0.2 \times 3.8 = 0.76$$

Since (μ.θ) for the larger or flat pulley is small, therefore the design is based on the larger or flat pulley.

We know that peripheral velocity of the belt,

$$v = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 0.25 \times 1800}{60} = \ldots$$

①

$\therefore$ Centrifugal tension,

$$T_C = m.v^2 = 0.253\ (23.56)^2 = 140.4 \text{ N}$$

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 maximum tension in the belt,

$$T = \text{Stress} \times \text{area} = \sigma \times a = 2.1 \times 230 = 483 \text{ N}$$

We also know that maximum or total tension in the belt,

$$T = T_1 + T_C$$

$\therefore \quad T_1 = T - T_C = 483 - 140.4 = 342.6$ N

We know that

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta_2 = 0.2 \times 3.8 = 0.76$$

$$\log\left(\frac{T_1}{T_2}\right) = 0.76 / 2.3 = 0.3304 \quad \text{or} \quad \frac{T_1}{T_2} = 2.14 \qquad \text{...(Taking antilog of 0.3304)}$$

and $\quad T_2 = T_1 / 2.14 = 342.6 / 2.14 = 160$ N

$\therefore$ Power transmitted per belt

$$= (T_1 - T_2)\, v = (342.6 - 160)\ 23.56 = 4302 \text{ W} = 4.302 \text{ kW}$$

We know that number of belts required

$$= \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}} = \frac{20}{4.302} = 4.65 \text{ say 5 Ans.}$$

20.7 Rope Drives

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. It may be noted that the use of flat belts is limited for the transmission of moderate power from one pulley to another when the two pulleys are not more than 8 metres apart. If large amounts of power are to be transmitted, by the flat belt, then it would result in excessive belt cross-section.

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.

20.8 Fibre Ropes

The ropes for transmitting power are usually made from fibrous materials such as hemp, manila and cotton. Since the hemp and manila fibres are rough, therefore the ropes made from these fibres are not very flexible and possesses poor mechanical properties. The hemp ropes have less strength as compared to manila ropes. When the hemp and manila ropes are bent over the sheave, there is some sliding of the fibres, causing the rope to wear and chafe internally. In order to minimise this defect, the rope fibres are lubricated with a tar, tallow or graphite. The lubrication also makes the rope moisture proof. The hemp ropes are suitable only for hand operated hoisting machinery and as tie ropes for lifting tackle ...

Costlier than manila ropes

Notes : 1. The diameter of manila and cotton ropes usually ranges from 38 mm to 50 mm. The size of the rope is usually designated by its circumference or '*girth*'.

2. The ultimate tensile breaking load of the fibre ropes varies greatly. For manila ropes, the average value of the ultimate tensile breaking load may be taken as 500 d^2 kN and for cotton ropes, it may be taken as 350 d^2 kN, where d is the diameter of rope in mm.

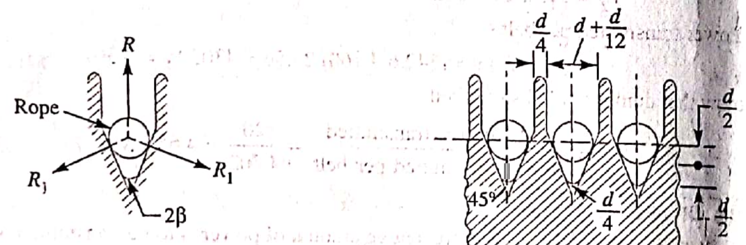
20.9 Advantages of Fibre Rope Drives

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.

20.10 Sheave for Fibre Ropes

The fibre ropes are usually circular in cross-section as shown in Fig. 20.6 (*a*). The sheave for the fibre ropes, is shown in Fig. 20.6 (*b*). The groove angle of the pulley for rope drives is usually 45°.



(*a*) Cross-section of a rope. (*b*) Sheave (grooved pulley) for ropes.

Fig. 20.6. Rope and sheave.

The grooves in the pulleys are made narrow at the bottom and the rope is pinched between the edges of the *V*-groove to increase the holding power of the rope on the pulley. The grooves should be finished smooth to avoid chafing of the rope. The diameter of the sheaves should be large to reduce the wear on the rope due to internal friction and bending stresses. The proper size of sheave wheels is 40 d and the minimum size is 36 d, where d is the diameter of rope in cm.

Note : The number of grooves should not be more than 24.

Rope drives

20.11 Ratio of Driving Tensions for Fibre Rope

A fibre rope with a grooved pulley is shown in Fig. 20.6 (*a*). The fibre ropes are designed in the similar way as V-belts. We have discussed in Art. 20.5, that the ratio of driving tensions is

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

where μ, θ and β have usual meanings.

2

Example 20.6. *A pulley used to transmit power by means of ropes has a diameter of 3.6 metres and has 15 grooves of 45° angle. The angle of contact is 170° and the coefficient of friction between the ropes and the groove sides is 0.28. The maximum possible tension in the ropes is 960 N and the mass of the rope is 1.5 kg per metre length. Determine the speed of the pulley in r.p.m. and the power transmitted if the condition of maximum power prevail.*

Solution. Given : d = 3.6 m ; n = 15 ; 2 β = 45° or β = 22.5° ; θ = 170° = 170 × π / 180 = 2.967 rad ; μ = 0.28 ; T = 960 N ; m = 1.5 kg / m

Speed of the pulley

Let N = Speed of the pulley in r.p.m.

We know that for maximum power, speed of the pulley,

$$v = \sqrt{\frac{T}{3m}} = \sqrt{\frac{960}{3 \times 1.5}} = 14.6 \text{ m/s}$$

We also know that speed of the pulley (v),

$$14.6 = \frac{\pi d . N}{60} = \frac{\pi \times 3.6 \times N}{60} = 0.19\, N$$

$$\therefore \quad N = 14.6 / 0.19 = 76.8 \text{ r.p.m. } \textbf{Ans.}$$

Power transmitted

We know that for maximum power, centrifugal tension,

$$T_C = T / 3 = 960 / 3 = 320 \text{ N}$$

∴ Tension in the tight side of the rope,

$$T_1 = T - T_C = 960 - 320 = 640 \text{ N}$$

Let T_2 = Tension in the slack side of the rope.

We know that

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta = 0.28 \times 2.967 \times \operatorname{cosec} 22.5° = 2.17$$

$$\therefore \quad \log\left(\frac{T_1}{T_2}\right) = \frac{2.17}{2.3} = 0.9435 \quad \text{or} \quad \frac{T_1}{T_2} = 8.78 \qquad \text{...(Taking antilog of 0.9435)}$$

and $T_2 = T_1 / 8.78 = 640 / 8.78 = 73$ N

∴ Power transmitted,

$$P = (T_1 - T_2)\, v \times n = (640 - 73)\, 14.6 \times 15 = 124\,173 \text{ W}$$
$$= 124.173 \text{ kW } \textbf{Ans.}$$

Example 20.7. *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. Given : n = 10 ; v = 1500 m/min = 25 m/s ; P = 115 kW = 115 × 10³ W ; θ = 180° = π rad ; 2 β = 45° or β = 22.5° ; μ = 0.2 ; m = 0.6 kg / m

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

T_2 = Tension in the slack side of the rope.

We know that total power transmitted (P),

$$115 \times 10^3 = (T_1 - T_2)\, v \times n = (T_1 - T_2)\, 25 \times 10 = 250\, (T_1 - T_2)$$

$$T_1 - T_2 = 115 \times 10^3 / 250 = 460 \qquad \text{...(i)}$$

20.12 Wire Ropes

When a large amount of power is to be transmitted over long distances from one pulley 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. The wire ropes run on grooved pulleys but they rest on the bottom of the *grooves and are not wedged between the sides of the grooves.

The wire ropes are made from cold drawn wires in order to have increase in strength and durability. It may be noted that the strength of the wire rope increases as its size decreases. The various materials used for wire ropes in order of increasing strength are wrought iron, cast steel, extra strong cast steel, plough steel and alloy steel. For certain purposes, the wire ropes may also be made of copper, bronze, aluminium alloys and stainless steels.

20.13 Advantages of Wire Ropes

The wire ropes have the following advantages as compared to fibre ropes.

1. These are lighter in weight,
2. These offer silent operation,
3. These can withstand shock loads,
4. These are more reliable,
5. These are more durable,
6. They do not fail suddenly,
7. The efficiency is high, and
8. The cost is low.

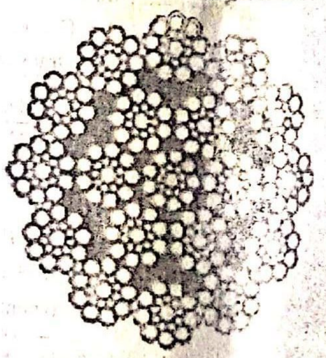
20.14 Construction of Wire Ropes

The wire ropes are made from various grades of steel wire having a tensile strength ranging from 1200 to 2400 MPa as shown in the following table :

Table 20.4. Grade and tensile strength of wires.

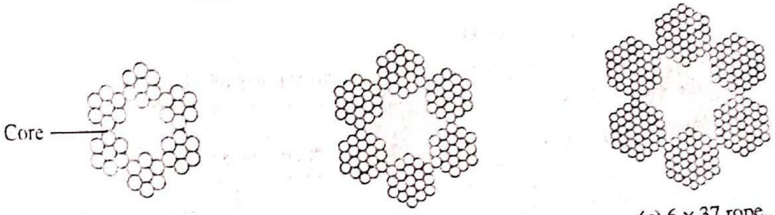
Grade of wire	120	140	160	180	200
Tensile strength range (MPa)	1200 – 1500	1400 – 1700	1600 – 1900	1800 – 2100	2000 – 2400

The wires are first given special heat treatment and then cold drawn in order to have high strength and durability of the rope. The steel wire ropes are manufactured by special machines. First of all, a number of wires such as 7, 19 or 37 are twisted into a strand and then a number of strands, usually 6 or 8 are twisted about a core or centre to form the rope as shown in Fig. 20.7. The core may be made of hemp, jute, asbsestos or a wire of softer steel. The core must be continuously saturated with lubricant for the long life of the core as well as the entire rope. The asbestos or soft wire core is used when ropes are subjected to radiant heat such as cranes operating near furnaces. However, a wire core reduces the flexibility of the rope and thus such ropes are used only where they are subjected to high compression as in the case of several layers wound over a rope drum.



Wire strands

* The fibre ropes do not rest at the bottom of the groove.



(*a*) 6 × 7 rope. (*b*) 6 × 19 rope. (*c*) 6 × 37 rope.

Fig. 20.7. Cross-sections of wire rope.

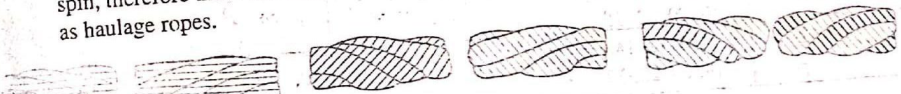
Wire rope

20.15 Classification of Wire Ropes

According to the direction of twist of the individual wires and that of strands, relative to each other, the wire ropes may be classified as follows :

1. *Cross or regular lay ropes*. In these types of ropes, the direction of twist of wires in the strands is opposite to the direction of twist of the stands, as shown in Fig. 20.8 (*a*). Such type of ropes are most popular.

2. *Parallel or lang lay ropes*. In these type of ropes, the direction of twist of the wires in the strands is same as that of strands in the rope, as shown in Fig. 20.8 (*b*). These ropes have better bearing surface but is harder to splice and twists more easily when loaded. These ropes are more flexible and resists wear more effectively. Since such ropes have the tendency to spin, therefore these are used in lifts and hoists with guide ways and also as haulage ropes.



(*i*) right handed (*ii*) left handed (*i*) right handed (*ii*) left handed

(*a*) Cross or regular lay ropes. (*b*) Parallel or lang lay ropes. (*c*) Composite or reverse laid ropes.

Fig. 20.8. Wire ropes classified according to the direction of twist of the individual wires.

3. *Composite or reverse laid ropes*. In these types of ropes, the wires in the two adjacent strands are twisted in the opposite direction, as shown in Fig. 20.8 (*c*).

Note : The direction of the lay of the ropes may be right handed or left handed, depending upon whether the strands form right hand or left hand helixes, but the right hand lay ropes are most commonly used.

20.16 Designation of Wire Ropes

The wire ropes are designated by the number of strands and the number of wires in each strand. For example, a wire rope having six strands and seven wires in each strand is designated by 6 × 7 rope. Following table shows the standard designation of ropes and their applications :

Table 20.5. Standard designation of ropes and their applications.

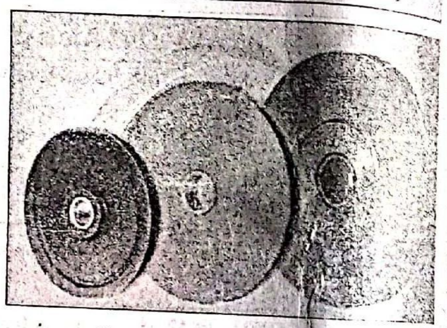
Standard designation	*Application*
6 × 7 rope	It is a standard coarse laid rope used as haulage rope in mines, tramways, power transmission.
6 × 19 rope	It is a standard hoisting rope used for hoisting purposes in mines, quarries, cranes, dredges, elevators, tramways, well drilling.
6 × 37 rope	It is 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.

Table 20.12. Sheave diameters (D) for wire ropes.

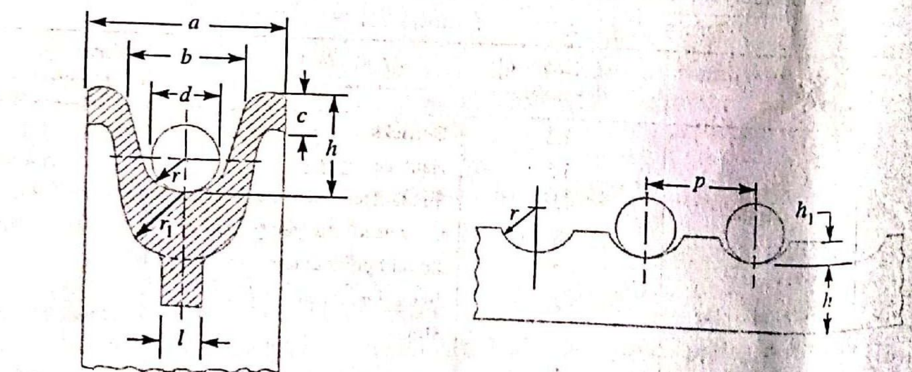
Type of wire rope	Recommended sheave diameter (D)		Uses
	Minimum sheave diameter	Preferred sheave diameter	
6 × 7	42 d	72 d	Mines, haulage tramways.
6 × 19	30 d	45 d	Hoisting rope.
	60 d	100 d	Cargo cranes, mine hoists.
	20 d	30 d	Derricks, dredges, elevators, tramways, well drilling.
6 × 37	18 d	27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

However, if the space allows, then the large diameters should be employed which give better and more economical service.

The sheave groove has a great influence on the life and service of the rope. If the groove is bigger than rope, there will not be sufficient support for the rope which may, therefore, flatten from its normal circular shape and increase fatigue effects. On the other hand, if the groove is too small, then the rope will be wedged into the groove and thus the normal rotation is prevented. The standard rim of a rope sheave is shown in Fig. 20.9 (*a*) and a standard grooved drum for wire ropes is shown in Fig. 20.9 (*b*).



Sheave or pulleys for winding ropes



$r = 0.53\ d$; $r_1 = 1.1\ d$; $a = 2.7\ d$; $b = 2.1\ d$; $c = 0.4\ d$; $h = 1.6\ d$; $l = 0.75\ d$

(*a*) Wire rope sheave rim.

$p = 1.15\ d$; $h_1 = 0.25\ d$; $r = 0.53\ d$; $h = 1.1\ d$

(*b*) Grooved rope drum.

Fig. 20.9

For light and medium service, the sheaves are made of cast iron, but for heavy crane service they are often made of steel castings. The sheaves are usually mounted on fixed axles on antifriction bearings or bronze bushings.

The small drums in hand hoists are made plain. A hoist operated by a motor or an engine has a drum with helical grooves, as shown in Fig. 20.9 (*b*). The pitch (*p*) of the grooves must be made slightly larger than the rope diameter to avoid friction and wear between the coils.

20.21 Wire Rope Fasteners

The various types of rope fasteners are shown in Fig. 20.10. The splices in wire ropes should be avoided because it reduces the strength of the rope by 25 to 30 percent of the normal ultimate strength.

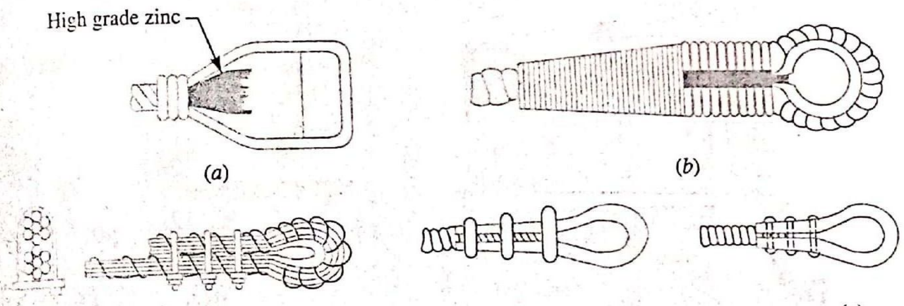


Fig. 20.10. Types of wire rope fastners.

The efficiencies of various types of fasteners are given in the following table.

Table 20.13. Efficiencies of rope fasteners.

Type of fastening	Efficiency (%)
(*a*) Wire rope socket with zinc, Fig. 20.10 (*a*)	100
(*b*) Thimble with four or five wire tucks, Fig. 20.10 (*b*)	90
(*c*) Special offset thimble with clips, Fig. 20.10 (*c*)	90
(*d*) Regular thimble with clips, Fig. 20.10 (*d*)	85
(*e*) Three bolt wire clamps, Fig. 20.10 (*e*)	75

20.22 Stresses in Wire Ropes

A wire rope is subjected to the following types of stresses :

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

Let W = Load lifted,

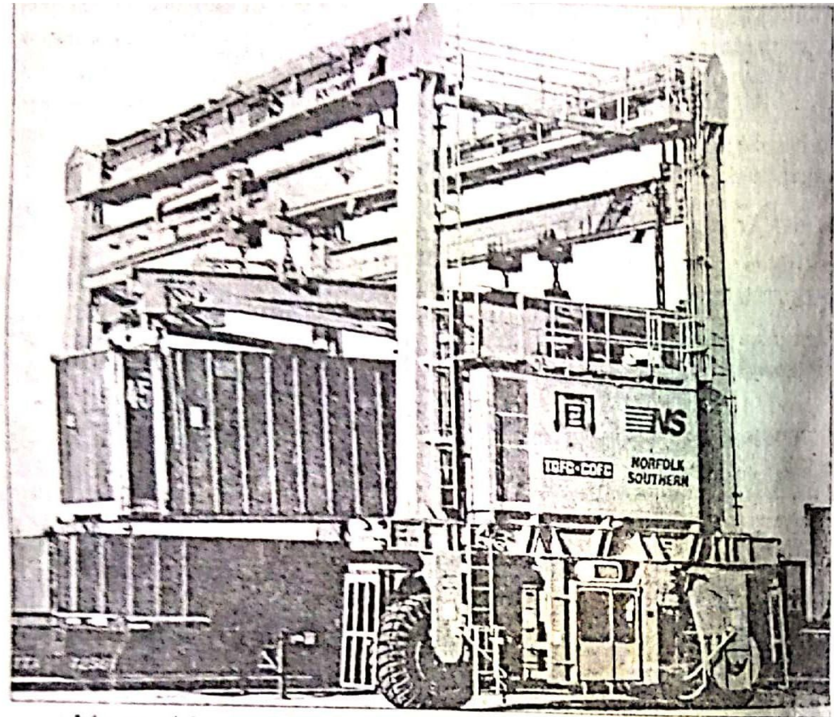
w = Weight of the rope, and

A = Net cross-sectional area of the rope.

∴ Direct stress,
$$\sigma_d = \frac{W + w}{A}$$

2. ***Bending stress when the rope winds round the sheave or drum.*** When a wire rope is wound over the sheave, then the bending stresses are induced in the wire which is tensile at the top and compressive at the lower side of the wire. The bending stress induced depends upon many factors such as construction of rope, size of wire, type of centre and the amount of restraint in the grooves. The approximate value of the bending stress in the wire as proposed by Reuleaux, is

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



A heavy duty crane. Cranes use rope drives in addition to gear drives

and equivalent bending load on the rope,

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

where 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.

It may be noted that E_r is not the modulus of elasticity for the wire material, but it is of the entire rope. The value of E_r may be taken as 77 kN/mm² for wrought iron ropes and 84 kN/mm² for steel ropes. It has been found experimentally that E_r = 3/8 E, where E is the modulus of elasticity of the wire material.

If σ_b is the bending stress in each wire, then the load on the whole rope due to bending may be obtained from the following relation, *i.e.*

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

where n is the total number of wires in the rope section.

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

$$W_a = \frac{W + w}{g} \times a \qquad \text{...(}W \text{ and } w \text{ are in newton)}$$

and the corresponding stress,

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

where a = Acceleration of the rope and load, and

g = Acceleration due to gravity.

If the time (t) necessary to attain a speed (v) is known, then the value of 'a' is given by

$$a = v / 60\, t$$

The general case of starting is when the rope has a slack (h) which must be overcome before the rope is taut and starts to exert a pull on the load. This induces an impact load on the rope.

The impact load on starting may be obtained by the impact equation, *i.e.*

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

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 above in which the acceleration is given by

$$a = (v_2 - v_1)/t$$

where $(v_2 - v_1)$ is the change in speed in m/s and t is the time in seconds.

It may be noted that when the hoist drum is suddenly stopped while lowering the load, it produces a stress that is several times more than the direct or static stress because of the kinetic energy of the moving masses is suddenly made zero. This kinetic energy is absorbed by the rope and the resulting stress may be determined by equating the kinetic energy to the resilience of the rope. If during stopping, the load moves down a certain distance, the corresponding change of potential energy must be added to the kinetic energy. It is also necessary to add the work of stretching the rope during stopping, which may be obtained from the impact stress.

5. *Effective stress.* The sum of the direct stress (σ_d) and the bending stress (σ_b) is called the effective stress in the rope during normal working. Mathematically,

Effective stress in the rope during normal working

$$= \sigma_d + \sigma_b$$

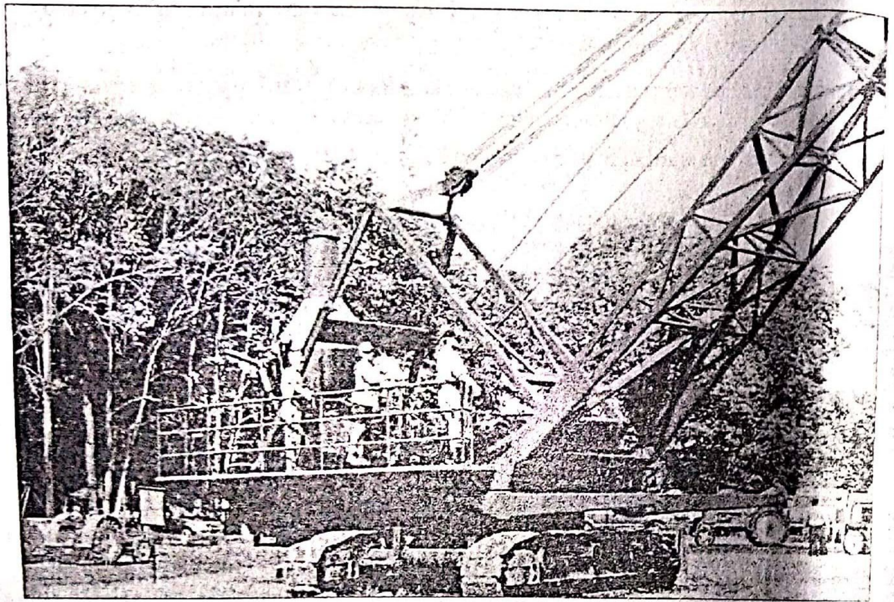
Effective stress in the rope during starting

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

and effective stress in the rope during acceleration of the load

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

While designing a wire rope, the sum of these stresses should be less than the ultimate strength divided by the factor of safety.

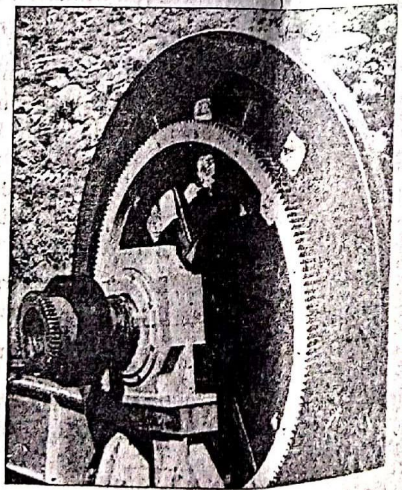


Ropes on a pile driver

20.23 Procedure for Designing a Wire Rope

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

1. First of all, select a suitable type of rope from Tables 20.6, 20.7, 20.8 and 20.9 for the given application.
2. Find the design load by assuming a factor of safety 2 to 2.5 times the factor of safety given in Table 20.11.
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) from Table 20.10.
5. Find the various stresses (or loads) in the rope as discussed in Art. 20.22.
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 safety given in Table 20.11. If the actual factor of safety is within permissible limits, then the design is safe.



Wheel that winds the metal rope.

(6)

Example 20.10. *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. Given : $W = 55$ kN $= 55\ 000$ N ; Depth = 300 m ; $v = 500$ m/min ; $t = 10$ s

The following procedure may be adopted in selecting a wire rope for a vertical mine hoist.

1. From Table 20.6, 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.
2. From Table 20.11, 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 in Table 20.11, therefore let us take the factor of safety as 15.

$\therefore$ Design load for the wire rope

$$= 15 \times 55 = 825 \text{ kN} = 825\ 000 \text{ N}$$

3. From Table 20.6, we find that 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, we get

$$595\ d^2 = 825\ 000$$

$$\therefore \quad d^2 = 825\ 000 / 595 = 1386.5 \quad \text{or} \quad d = 37.2 \text{ say } 38 \text{ mm}$$

4. From Table 20.10, we find that for a 6×19 rope,

Diameter of wire, $d_w = 0.063\ d = 0.063 \times 38 = 2.4$ mm

and area of rope, $A = 0.38\ d^2 = 0.38\ (38)^2 = 550$ mm^2

5. Now let us find out the various loads in the rope as discussed below :

(*a*) From Table 20.6, we find that weight of the rope,

$$w = 0.0363\ d^2 = 0.0363\ (38)^2 = 52.4 \text{ N/m}$$

$$= 52.4 \times 300 = 15\ 720 \text{ N} \quad \text{...}(\because \text{Depth} = 300 \text{ m})$$

(*b*) From Table 20.12, we find that diameter of the sheave (D) may be taken as 60 to 100 times the diameter of rope (d). Let us take

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

$\therefore$ Bending stress,

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

...(Taking $E_r = 84 \times 10^3$ N/mm^2)

and the equivalent bending load on the rope,

$$W_b = \sigma_b \times A = 53 \times 550 = 29\ 150 \text{ N}$$

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

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

$\therefore$ 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. We know that the effective load on the rope during normal working (*i.e.* during uniform lifting or lowering of the load)

$$= W + w + W_b = 55\ 000 + 15\ 720 + 29150 = 99\ 870 \text{ N}$$

$\therefore$ Actual factor of safety during normal working

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

Effective load on the rope during starting

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

$\therefore$ 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 ... starting)

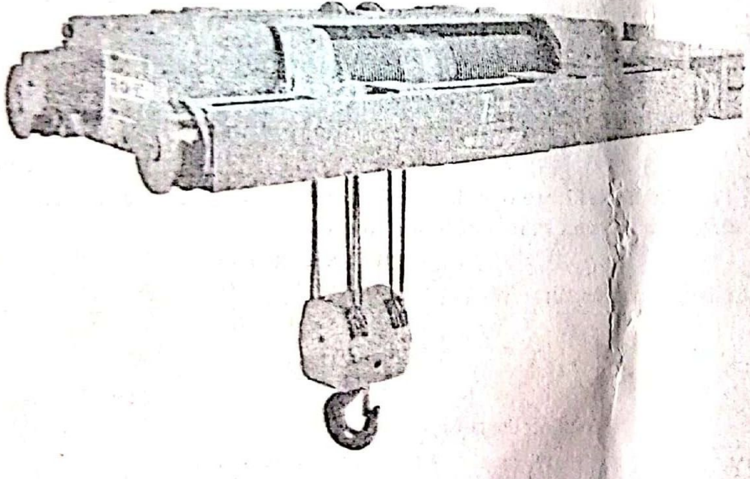
$$= W + w + W_b + W_a$$

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

$\therefore$ 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 diame... 38 mm and 6 × 19 type is satisfactory. Ans.



A vertical hoist with metal ropes

Example 20.11. *An extra flexible 8 × 19 plough steel wire rope of 38 mm diameter is used ... a 2m diameter hoist drum to lift 50 kN of load. Find the factor of safety (ratio of the breaking loa... the maximum working load) under the following conditions of operation :*

The wire rope is required to lift from a depth of 900 metres. The maximum speed is 3 m/s ... the acceleration is 1.5 m / s², when starting under no slack condition. The diameter of the wire ... be taken as 0.05 d, where d is the diameter of wire rope. The breaking strength of plough ste... 1880 N/mm² and modulus of elasticity of the entire rope is 84 × 10³ N/mm². The weight of the ro... is 53 N/m length.

Solution. Given : $d = 38$ mm ; $D = 2$ m $= 2000$ mm ; $W = 50$ kN $= 50\,000$ N ; Depth = ... $v = 3$ m/s ; $a = 1.5$ m/s² ; $d_w = 0.05\,d$; Breaking strength = 1880 N/mm² ; $E_r = 84 \times 10^3$ N/... $w = 53$ N/m $= 53 \times 900 = 47\,700$ N

Since the wire rope is 8 × 19, therefore total number of wires in the rope,

$$n = 8 \times 19 = 152$$

We know that diameter of each wire,

$$d_w = 0.05\,d = 0.05 \times 38 = 1.9 \text{ mm}$$

$\therefore$ Cross-sectional area of the wire rope,

$$A = \frac{\pi}{4}(d_w)^2\, n = \frac{\pi}{4}(1.9)^2\, 152 = 431 \text{ mm}^2$$

Minimum breaking strength of the rope

$$= \text{Breaking strength} \times \text{Area} = 1880 \times 431 = 810\,280 \text{ N}$$

We know that bending stress,

$$\sigma_b = \frac{E_r \times d_w}{D} = \frac{84 \times 10^3 \times 1.9}{2000} = 79.8 \text{ N/mm}^2$$

Equivalent bending load on the rope,

$$W_b = \sigma_b \times A = 79.8 \times 431 = 34\,390 \text{ N}$$

Additional load due to acceleration of the load lifted and rope,

$$W_a = \frac{W + w}{g} \times a = \frac{50\,000 + 47\,700}{9.81} \times 1.5 = 14\,940 \text{ N}$$

Impact load during starting (when there is no slackness in the rope),

$$W_{st} = 2\,(W + w) = 2(50\,000 + 47\,700) = 195\,400 \text{ N}$$

We know that the effective load on the rope during normal working

$$= W + w + W_b = 50\,000 + 47\,700 + 34\,390 = 132\,090 \text{ N}$$

$\therefore$ Factor of safety during normal working

$$= 810\,280 / 132\,090 = 6.13 \text{ Ans.}$$

Effective load on the rope during starting

$$= W_{st} + W_b = 195\,400 + 34\,390 = 229\,790 \text{ N}$$

$\therefore$ Factor of safety during starting

$$= 810\,280 / 229\,790 = 3.53 \text{ Ans.}$$

Effective load on the rope during acceleration of the load (*i.e.* during the first 2 second after ...)

$$= W + w + W_b + W_a = 50\,000 + 47\,700 + 34\,390 + 14\,940$$

$$= 147\,030 \text{ N}$$

$\therefore$ Factor of safety during acceleration of the load

$$= 810\,280 / 147\,030 = 5.51 \text{ Ans.}$$

Example 20.12. *A workshop crane is lifting a load of 25 kN through a wire rope and a hook. ... weight of the hook etc. is 15 kN. The rope drum diameter may be taken as 30 times the diameter ... rope. The load is to be lifted with an acceleration of 1 m/s². Calculate the diameter of the wire ... take a factor of safety of 6 and Young's modulus for the wire rope 80 kN/mm². The ultimate ... may be taken as 1800 MPa. The cross-sectional area of the wire rope may be taken as 0.38 ... square of the wire rope diameter.*

Solution. Given : $W = 25$ kN $= 25\,000$ N ; $w = 15$ kN $= 15\,000$ N ; $D = 30\,d$; $a = 1$ m/s² ; ... kN/mm² $= 80 \times 10^3$ N/mm² ; $\sigma_u = 1800$ MPa $= 1800$ N/mm² ; $A = 0.38\,d^2$