ME 308
MACHINE ELEMENTS II

CHAPTER 9

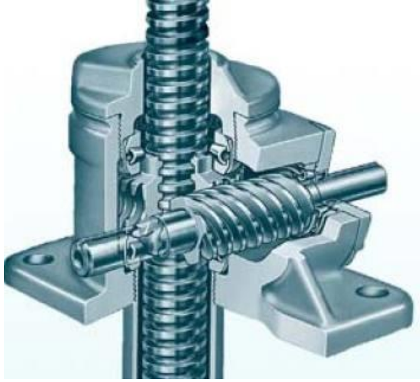
FLEXIBLE MECHANICAL ELEMENTS

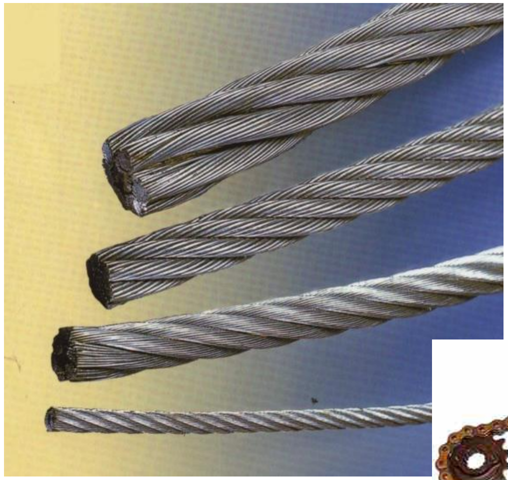
This chapter is about those machine elements which
can transmit power and
have a flexible nature

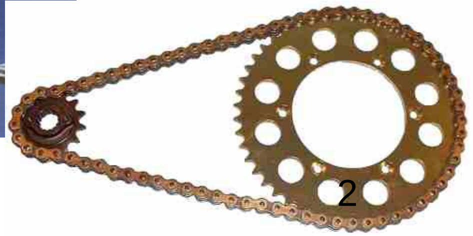
contrary to those rigid ones like gears, shafts, etc.

Power transmission elements can vary from

- Power Screws,
- Wedge,
- Gears and gearboxes
- Couplings
- Pulleys,
- Belts,
- Chains
- Sprockets(zincir dişlisi)
- Ropes
- Pneumatics and hydraulics,
- To motors, and
- more….







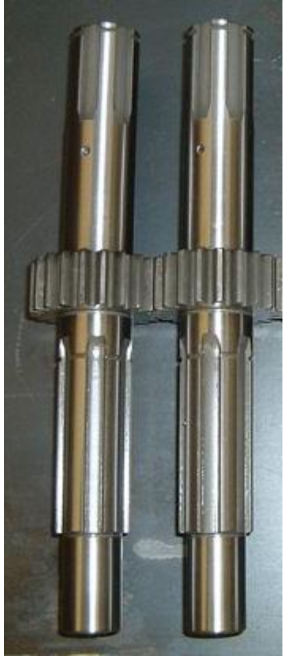
The ones which do not themselves

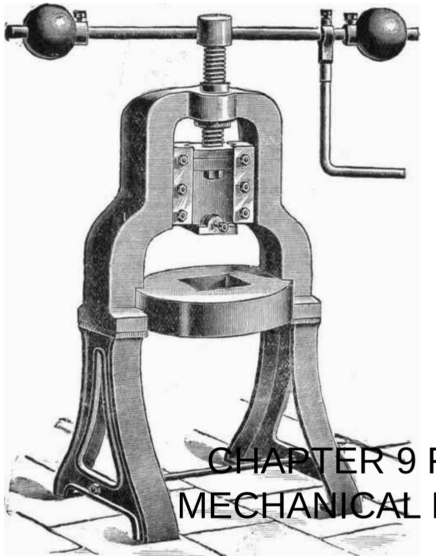
bend/change shape or

allow some kind of flexiblity

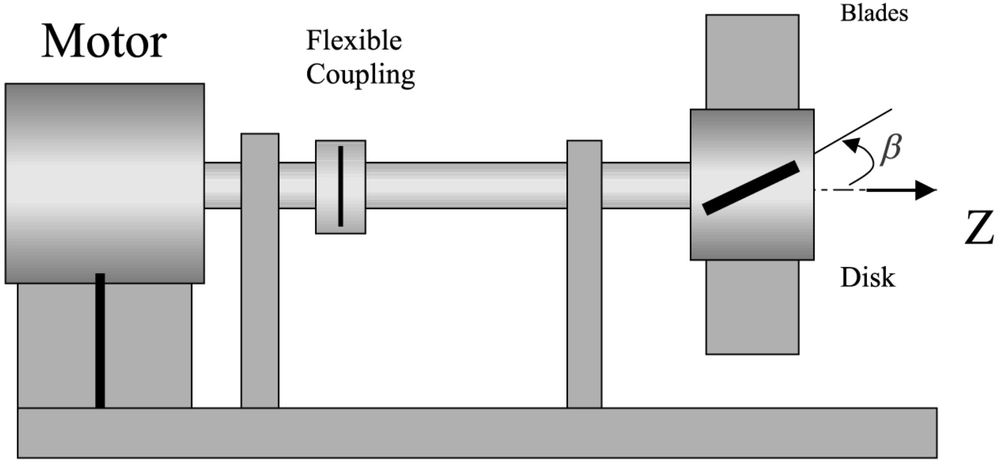
are called rigid power transmission elements like:

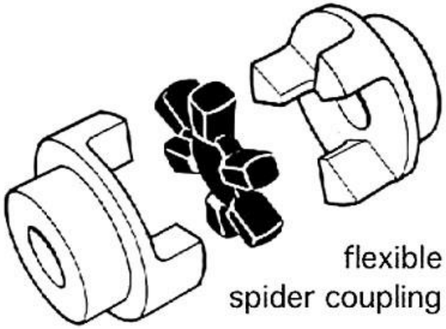
- Shafts,
- Gears,
- Screws,
- Some couplings,

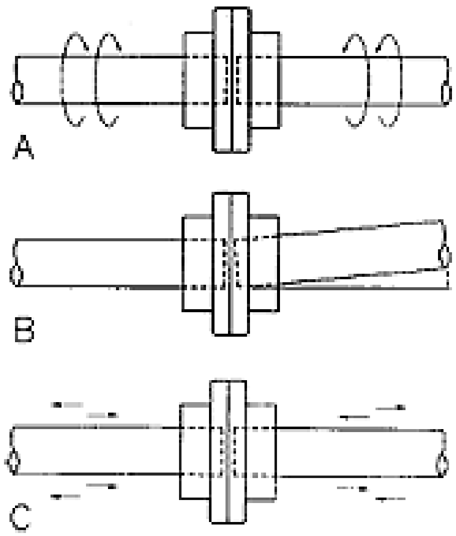


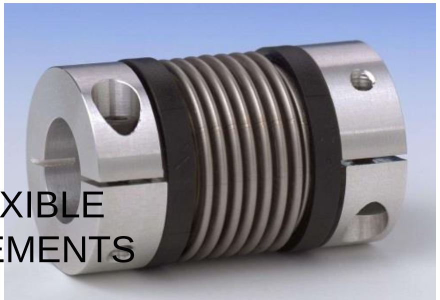


Here are different coupling types



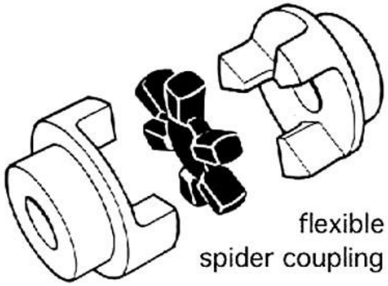


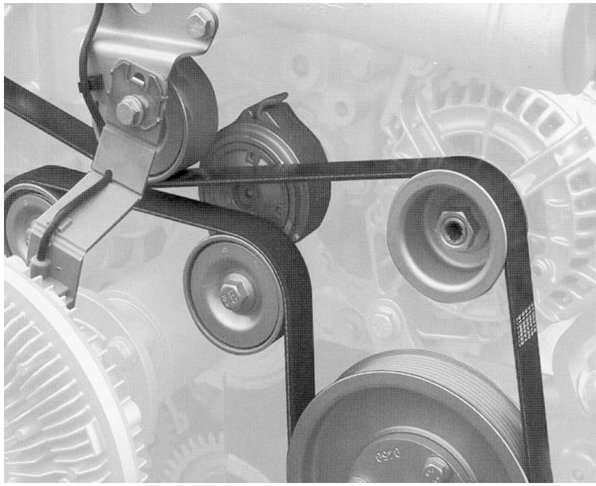


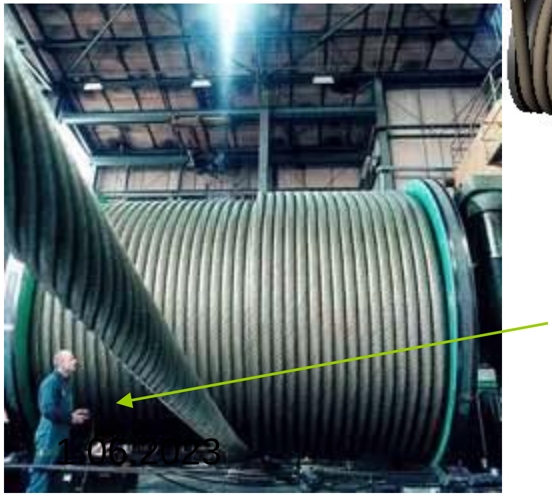


The ones which themselves
bend/change shape or
allow some kind of flexiblity
are called flexible power transmission elements like:

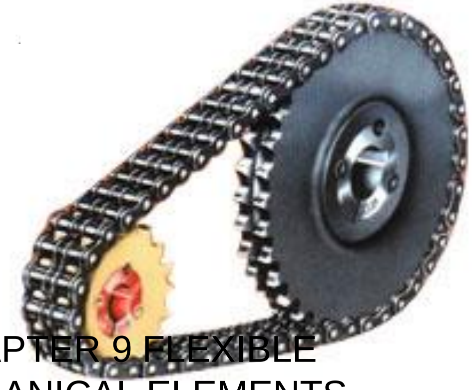
- Some couplings
- Belts
- Chains
- Ropes

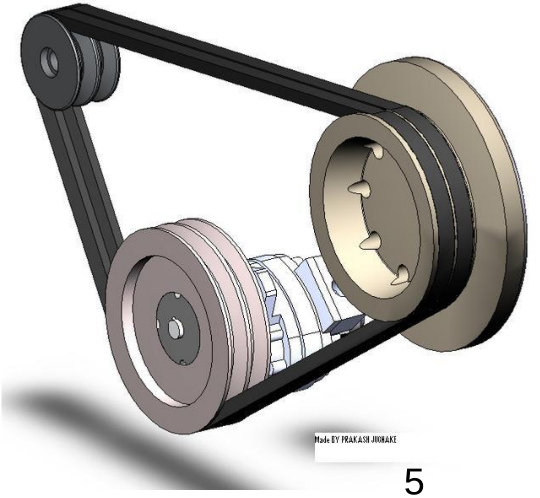


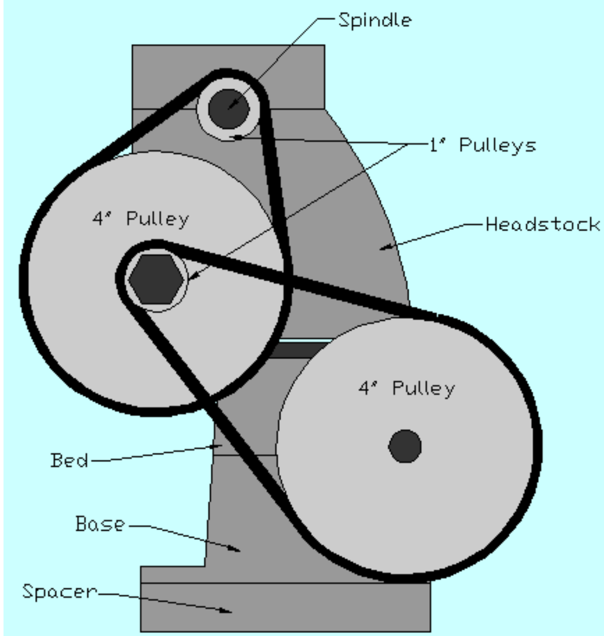


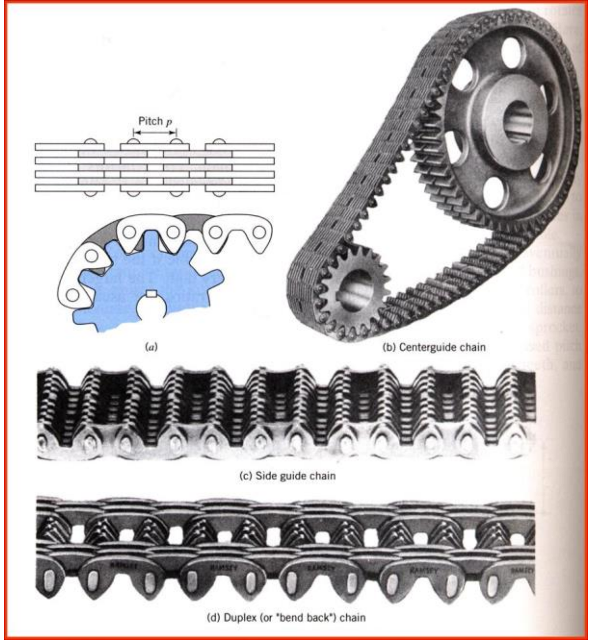


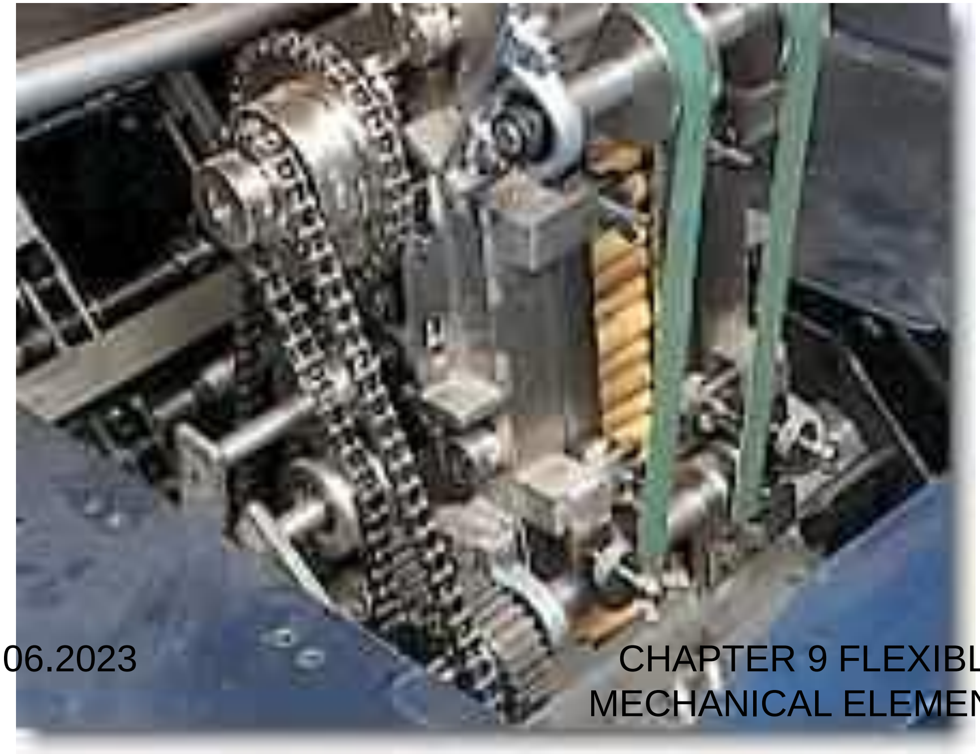
Did you see him?

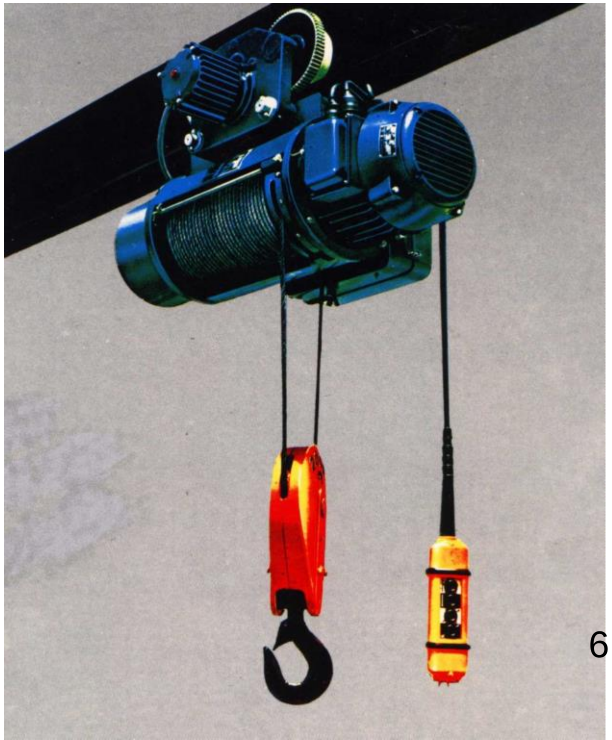





Spindle
1" Pulleys
4" Pulley
Headstock
4" Pulley
Bed
Base
Spacer


Pitch p
(a)
(b) Centerguide chain
(c) Side guide chain
(d) Duplex (or "bend back") chain



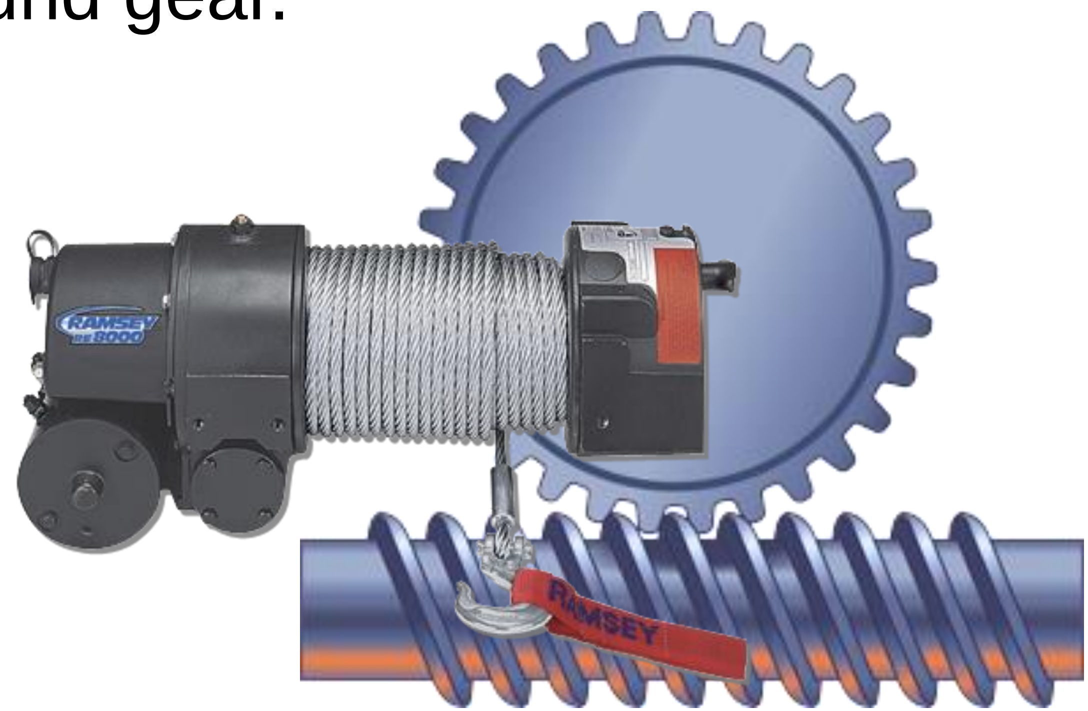


Winch Components

Gear Reducer

Worm Gear Reducers consist of a cylindrical worm and a typical round gear.

- ✓Heavy duty operation
- ✓Slower line speed
- ✓Automatic self locking
- ✓Generally used for towing(çekme) and recovery



A **Worm Gear** Winch

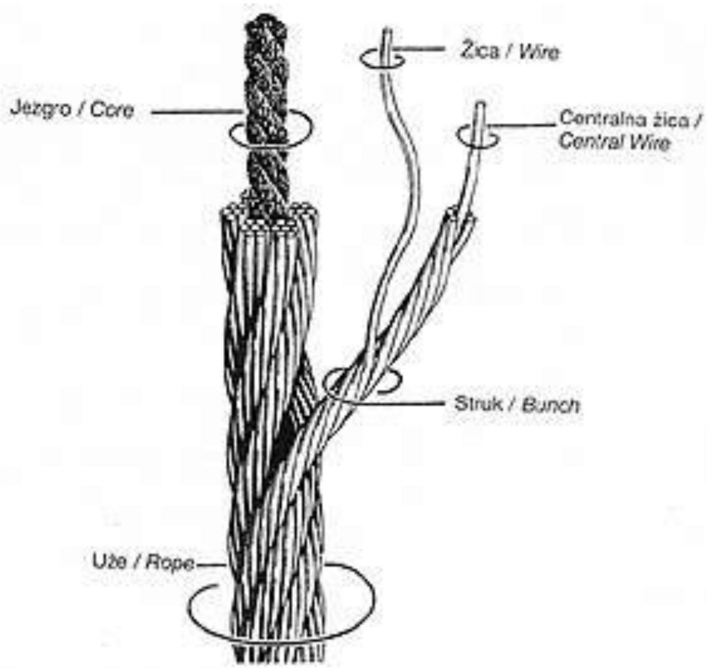
A **Worm Gear** winch will have a housing that sits perpendicular to the drum.

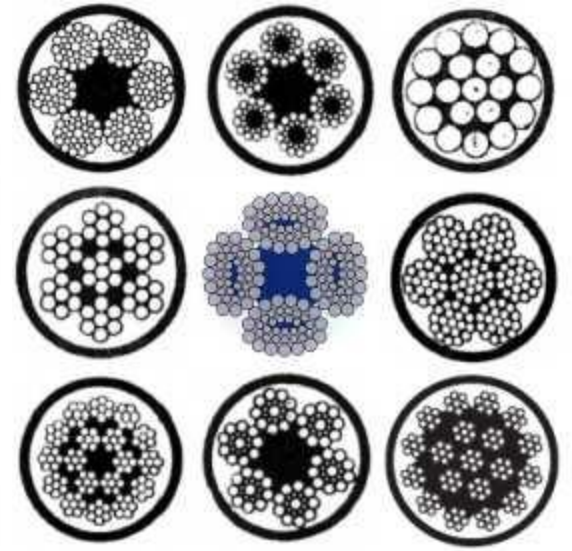
A **Worm Gear** system offers a **constant engagement** that prevents back-spooling, so the worm gear is a self braking design. It is typically found on **heavy duty** winches that will be used under the most demanding conditions. Because it is easier for the motor to turn the worm gear, duty cycles on worm gear winches tend to be lengthened. Disadvantages to a worm gear setup is its' additional size and weight as well as a slower line pull.

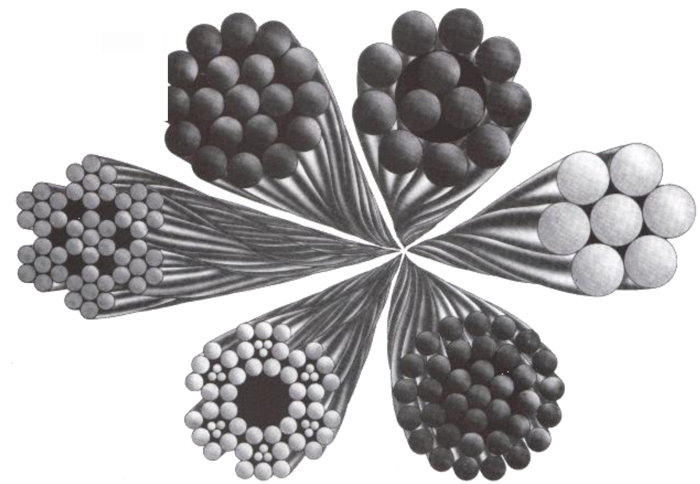
Again, duty cycle refers to the time the winch can be working before it needs to cool down. All powered winches will be subject to a duty cycle. If the motor housing is hot to the touch, the winch should be rested till it cools down.

Safety Tip! Winches should never be left for long periods to secure a load. Possible damage to the winch and the loss of the load may occur.


2010.07.03


Žica / Wire
Jezgro / Core
Centralna žica /
Central Wire
Struk / Bunch
Uže / Rope





9.1 BELTS INTRODUCTION

Fig. 9.1 V-Belts

Belts, ropes, chains, and other similar elastic or flexible machine elements are used in conveying systems and in the transmission of power over comparatively long distances. It often happens that these elements can be used as a replacement for gears, shafts, bearings, and other relatively rigid power-transmission devices. In many cases their use simplifies the design of a machine and substantially reduces the cost.

9.1.1 TYPES OF BELTS

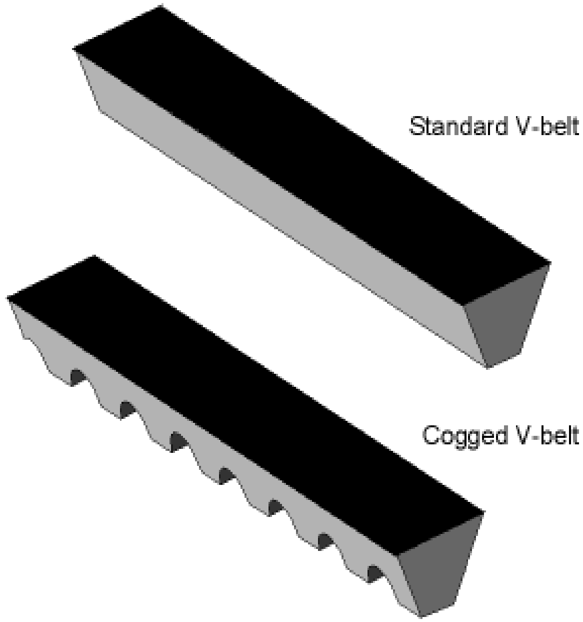


Fig. 9.2 Types of V-Belts

In addition, since these elements are elastic and usually quite long, they play an important part in absorbing shock loads and in damping out and isolating the effects of vibration. This is an important advantage as far as machine life is concerned.

Most flexible elements do not have an infinite life. When they are used, it is important to establish an inspection schedule to guard against wear, aging, and loss of elasticity. The elements should be replaced at the first sign of deterioration.

The most used flexible machine elements for power transmission are:

ropes,
belts*,
chains*,

and all these can be used for transmission of power over long distances.

These elements replace a group of gears, shafts and bearings or similar power transmission devices which would otherwise be used.

Since there are some equipment (shaft, gear, bearing) reduction in the system, the machine is greatly simplified and consequently these flexible elements are major cost reducing elements.

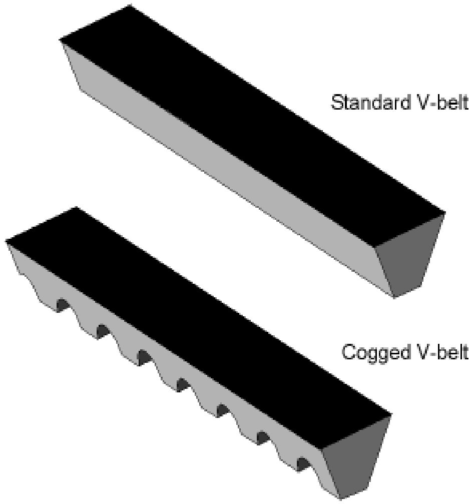
Since these elements are usually long and elastic and they play an important role in absorbing shock loads and damping out the effects of vibrating forces.

Belts are used to transmit power between two parallel shafts. Shafts have to be separated a certain minimum distance to work efficiently. This distance C_{min} depends on type of belt.

They are usually made of either leather, or fabric or synthetic nylons or a combination of some other materials. To strength them in power transmission generally steel or some other elements are used in the along the length wise.

Types of belts used in industry are:

- Flat belts (not continuous, but joined at two ends to make it endless)
- V-belt (generally standardized length of endless type)
- Link V-belt (made of small portions of V-belts added end-to-end like chains to make endless of different lengths)
- Timing belt (flat or V-belt with teeth on inner side of the belt fitting on to teeth of the pulley with similar teeth)



Kaplamak
Kapamak

Salınmak
sallanmak

Practical Notes

1) Pulleys need to be crowned to prevent belt from wandering off. Belts tend to move to tightest position

2) Tension required to enable belt to operate. Tensions normally set by adjusting center distance between pulleys to ensure some stretch of belts (say 2%).

3) Best drives result from belts with high flexibility, low mass, and with surfaces engineered to provide a high coefficient of friction

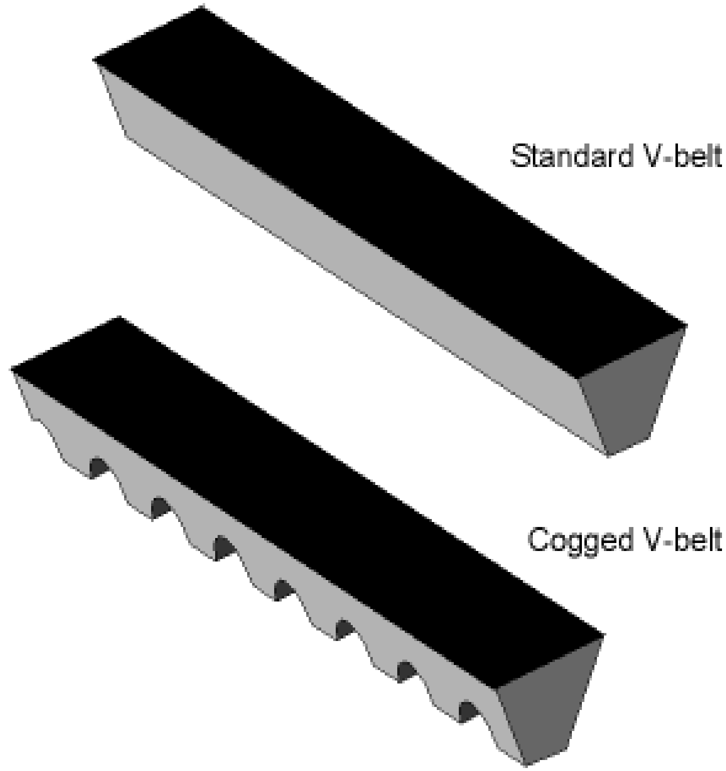
Flat belts are very efficient (e=98%) for high speeds , they are quiet,

- They can transmit large amounts of power over long center distances,
- They do not require large pulleys,
- They can transmit power around corners or between pulley at right angles to each other.
- speed range : belt speed: 2500-7000 fpm (12.5 m/sec-35 m/sec) ideal=4000 fpm (20 m/sec)

V-belts are slightly less efficient (e=70-96%) than flat belts (e=98%) but a number of them can be used on a single sheave, this making a multiple drive.

(5 m/sec<V<25 m/sec, V_{opt}= 20 m/sec)

Sheave: çıkrık, makara, bobin, oluklu kasnak



Timing belts do not stretch or slip on pulleys and consequently transmits power at a constant angular velocity ratio.

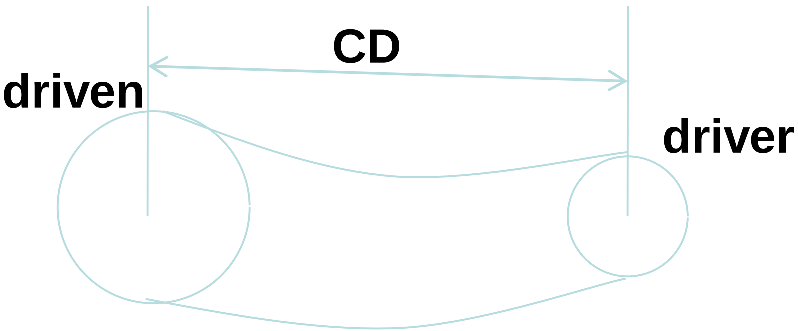
No initial tension is required , so fixed center drives can be used. The speed limitation is eliminated since teeth on belts make is possible to run at any speed, slow or fast.

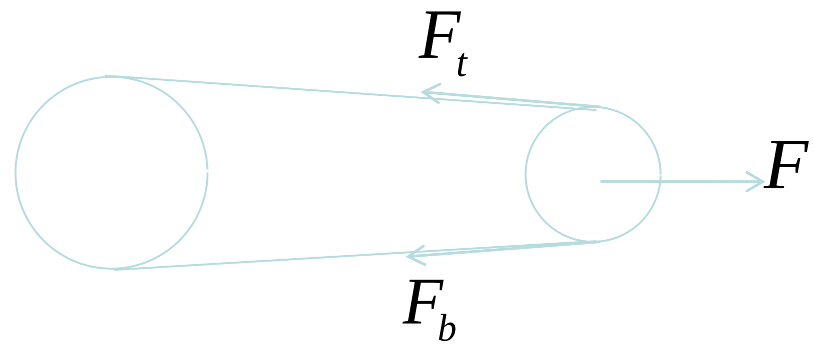
On the other hand, the initial cost is high, the teeth on pulley are required and some dynamic loads are attended the belt at the belt meshing frequency.

Some characteristics of belts in general are:

- They may be used for long center distances,
- Some adjustments of the center distance is usually required when belts are used (except timing belt),
- Because of the slip and creep of the belts, the angular velocity ratio between the shafts carrying the pulleys is neither constant nor exactly equal to the ratio of pulley diameters (exception is timing belt),
- By employing step pulleys on economical means of changing the velocity ratio may be obtained,
- When using V- belts, pulleys with spring-loaded sides can allow some variation in angular speed variation by changing the effective pulley diameter.

9.2 Force Analysis of Flat Belt Drives





Without tightening the belt by extending the CD of the shaft the driving pulley will slip only and will not transmit any power or rotational motion.

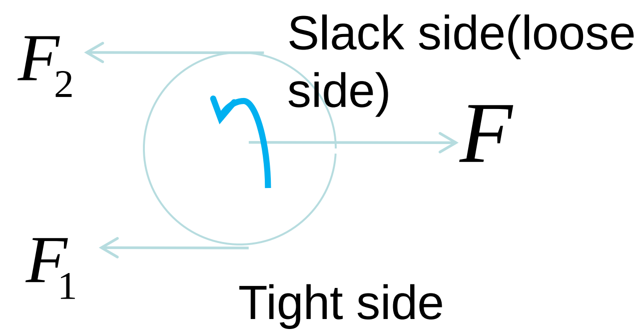
Without any rotation

$$\sum F_x = 0 \qquad F_t + F_b = F$$

$$F_t = F_b = F_{initial}$$

When pulley start rotation there will be an increase in tension of the bottom belt while a reduction in tension of top belt.

Thus



$$F_1 > F_i \; ; \quad F_1 = F_i + \Delta F$$

$$F_2 < F_i \; ; \quad F_2 = F_i - \Delta F$$

To transmit a rotation or power a friction is required between pulley and the belt inside surface. Friction coefficient $f \cong 0.30 - 0.70$

$$\sum F_x = 0 \quad dN - (P + dP)\sin\frac{d\theta}{2} - P\sin\frac{d\theta}{2} = 0$$

$$\underline{\underline{dN = Pd\theta}} \quad \text{since,} \quad \sin\frac{d\theta}{2} = \frac{d\theta}{2} \quad \text{for small angles.} \qquad \textbf{...(1)}$$

$$\sum F_y = 0 \quad fdN + P\cos\frac{d\theta}{2} - (P + dP)\cos\frac{d\theta}{2} = 0$$

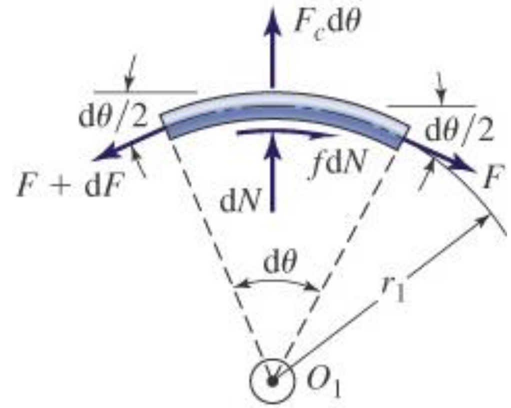
$$\underline{\underline{dP = fdN}} \quad \text{since,} \quad \cos\frac{d\theta}{2} = 1 \quad \text{for small angles.} \qquad \textbf{...(2)}$$

Using (1) and (2)

$$dP = fPd\theta$$

$$\int_{P_2}^{P_1} \frac{dP}{P} = \int_0^{\theta} fd\theta$$

$$\ln\frac{P_1}{P_2} = f\theta; \quad \underline{\underline{\frac{P_1}{P_2} = e^{f\theta}}}$$



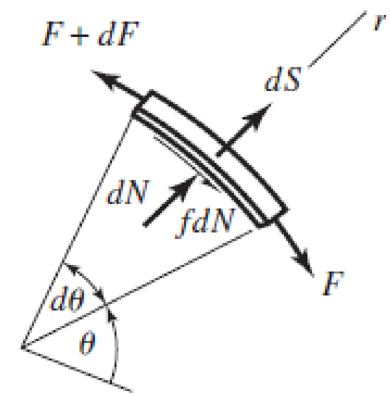


Figure 9–1
Free body of an infinitesimal element of a flat belt in contact with a pulley.

$$\frac{F_1}{F_2} \leq e^{f\theta}$$

is the relation between tight side and loose side forces of a belt when transmitting power;
where f is coefficient of friction (0.2-0.7)
θ is the angle of contact in radians for the driving pulley.

The torque transmitted is;

$$T = (F_1 - F_2) \times radius$$

Power transmitted is;

$$P = (F_1 - F_2) \times V$$

$$\left(or\ HP = \frac{(F_1 - F_2) \times V}{33000} \right)$$

F's in Newton (lb)
V belt speed in m/sec (fpm)
P in watts (hp)

$$\begin{bmatrix} P = T \times w \\ P = (F_1 - F_2) \times r \times w \\ P = (F_1 - F_2) \times (w \times r) \end{bmatrix}$$

Since each small part of the belt runs at a speed of V the total centrifugal force due to mass of the belt is given by

$$F_c = mV^2$$ Where *m* is the mass per unit length of belt (kg/m)
V is the speed of belt in m/sec

Thus the belt equation becomes;

$$\frac{F_1 - F_c}{F_2 - F_c} \leq e^{f\theta}$$

Also we know that when first initial tightening is done and rotation starts;

$$F_1 = F_i + \Delta F$$
$$F_2 = F_i - \Delta F$$
$$\longrightarrow$$
$$\Delta F = F_1 - F_i = F_i - F_2$$
$$\Rightarrow F_i = \frac{F_1 + F_2}{2}$$

Since F_1 is at tight side where the belt carries maximum load and hence exposed to maximum tensile stresses they have to be chosen from catalogues to have required tensile strength.

OR the force F_1 have to be limited to a certain value at which the tensile strength of the catalogue belt is not exceeded.

Since ΔF is the same at both (tight and loose sides) the maximum value ΔF can reach is F_i at which $F_2 = F_i - F_i = 0$ and belt can not carry compression load thus $\Delta F_{max} = F_i$

Hence

$$F_1 = F_i + \Delta F = F_i + F_i = 2F_i$$

$$\Rightarrow F_{1\max} = 2F_i \quad or \quad F_i = \frac{F_{1\max}}{2}$$

This is the maximum belt tension on tight side and to increase the power transmission F_i has to be increased.

The maximum tensile stress or the tensile load for belts are generally given in catalogues and Tables (T.17-1)

Once F_1 is determined from catalogue (and F_i is determined to be less than $F_i \le F_1 /2$); then from

$$F_i = \frac{F_1 + F_2}{2} \quad \Rightarrow F_2 = 2F_i - F_1$$

Hence power ;

$$P = \left(F_1 - F_2\right) \times V = \left(F_1 - \left(2F_i - F_1\right)\right) \times V = \left(2F_1 - 2F_i\right) \times V$$

$$P = 2\left(F_1 - F_i\right) \times V \quad \left(\text{or} \quad P = \frac{\left(F_1 - F_i\right) \times V}{16500}\right)$$

In design having selected an initial tension F_i and evaluated F_1 and F_2

through $P = 2\left(F_1 - F_i\right) \times V$ & $F_2 = 2F_i - F_1$ equation's then the

slippage has to be checked via

Either $\dfrac{F_1}{F_2} \le e^{f\theta}$ or $\dfrac{F_1 - F_c}{F_2 - F_c} \le e^{f\theta}$ if centrifugal forces are included

To avoid slippage between belt and pulley. $e^{f\theta}$ has to be larger than the ratio terms.

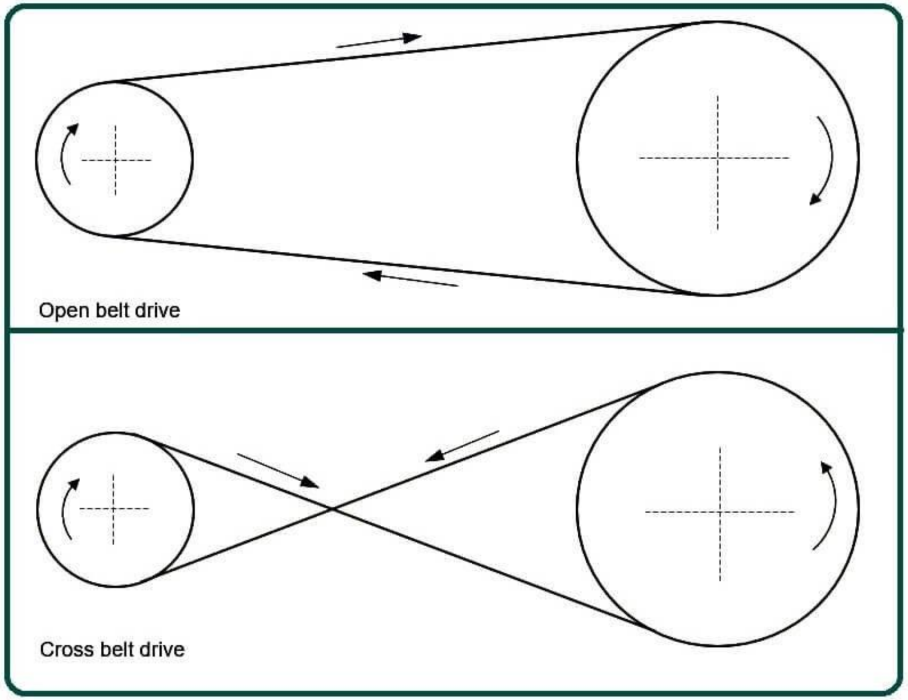
$$\frac{F_1}{F_2} \leq e^{f\theta}$$

If this is not satisfied;

either f has to be increased

or θ has to be increased

or a larger belt section have to be used.

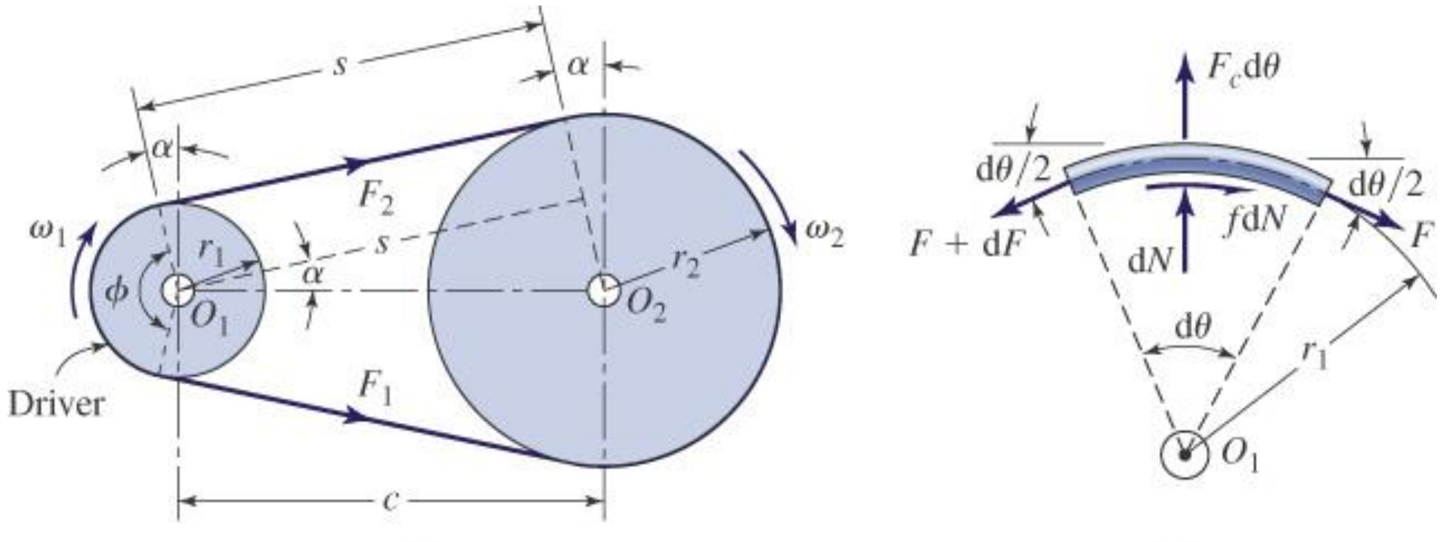


Two types of the belt configuration are possible for flat belts

- Open belt
- Crossed belt

9.2.1 OPEN BELT (for speed reduction)

Assume that small pulley is driving one and large pulley is driven one



(a) (b)

angle of contact of driving pulleys : θ_s (in radians)

angle of contact of driven pulleys : θ_L (in radians)

$$\sin\theta = \frac{D-d}{2}\Big/C = \frac{D-d}{2C}$$

diameter of driven pulleys : D (in mm)

diameter of driving pulleys : d (in mm)

Total length of the belt : L (in mm)

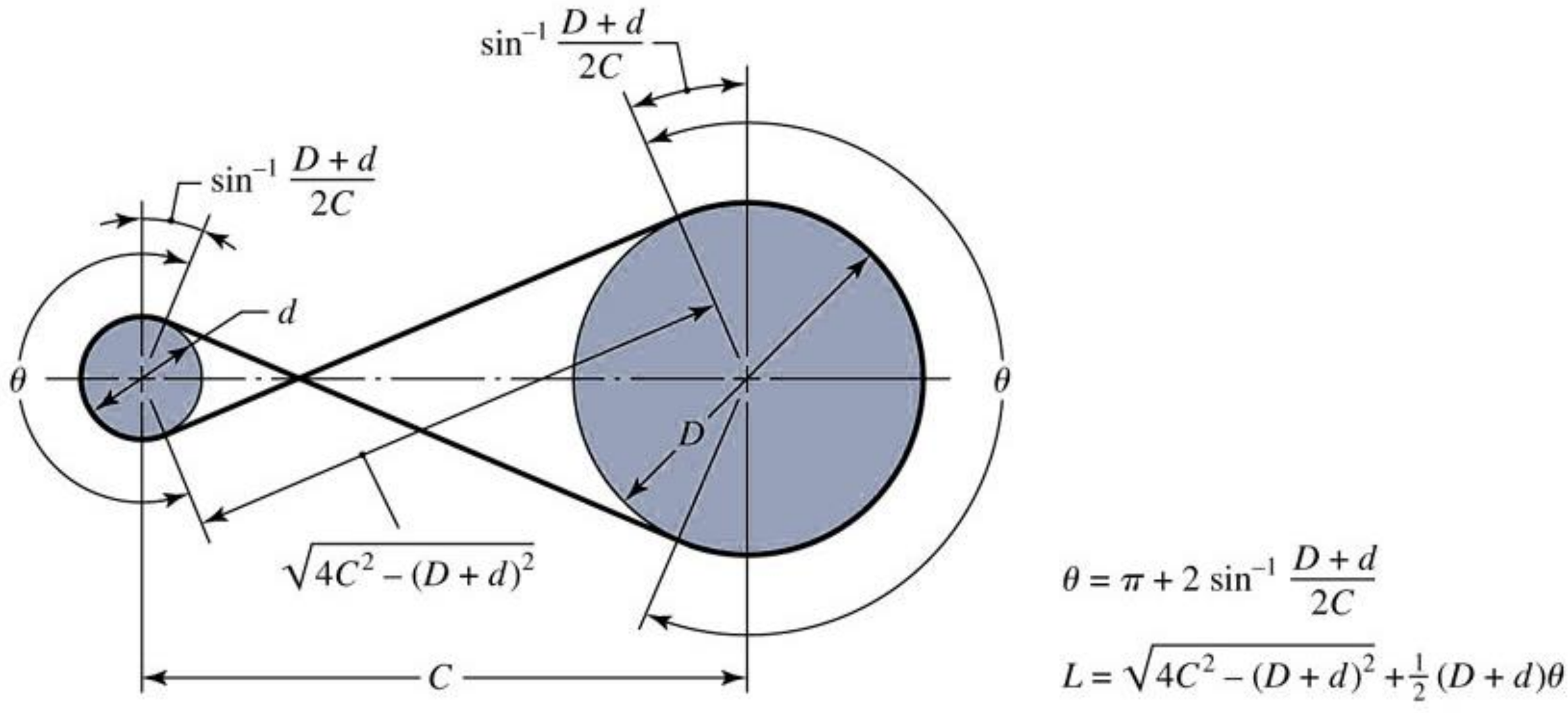
$$\theta_S = \pi - 2\theta = \pi - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$L = \sqrt{4C^2 - (D-d)^2} + \frac{1}{2}\left(D\theta_L + d\theta_s\right)$$

$$\theta_L = \pi + 2\theta = \pi + 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

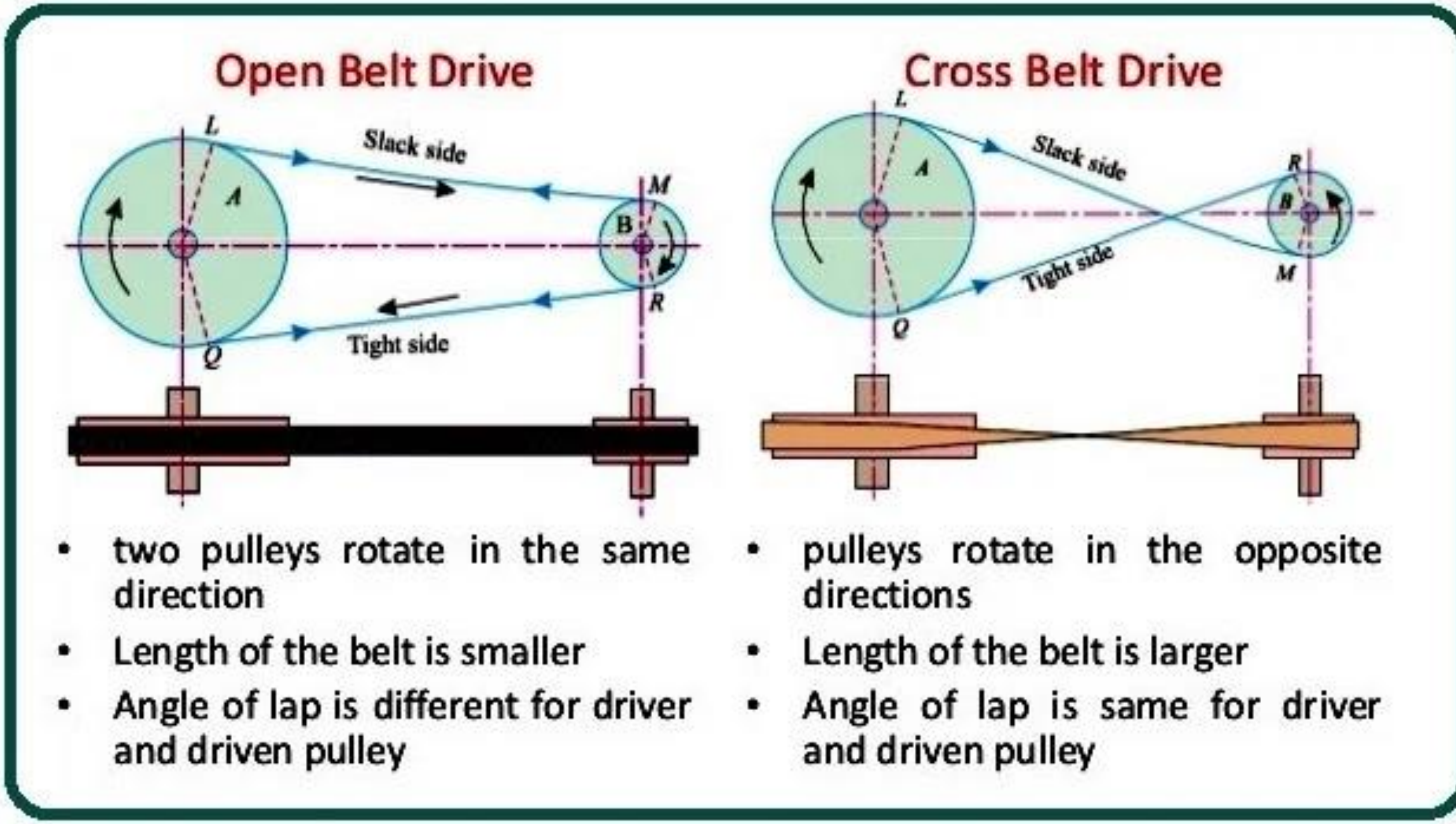
9.2.2 CROSSED BELT (for speed reduction)

Regardless of driving pulley size (whether small pulley or large pulley) angle of contact for both pulley is the same ($\theta_S = \theta_L$)



(b)

$$\theta_S = \theta_L = \pi + 2\sin^{-1}\left(\frac{D+d}{2C}\right) \qquad L = \sqrt{4C^2 - (D+d)^2} + \frac{\theta}{2}(D+d)$$



$$\theta_S = \pi - 2\theta = \pi - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$\theta_L = \pi + 2\theta = \pi + 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$\theta_S = \theta_L = \pi + 2\sin^{-1}\left(\frac{D+d}{2C}\right)$$

Angle of contacts are in radians

EXAMPLE 9.1

A nylon-core flat belt has an elastomeric envelope, is 200 mm wide and transmits *60 kW* at a belt speed of *25 m/sec*.

The belt has a mass of 2 kg/m of belt length. The belt is used in **a crossed configuration** to connect a 300 mm diameter driving pulley to a 900 mm diameter driven pulley at a shaft spacing of 6 m.

a) Calculate the belt length and the angles of wrap.

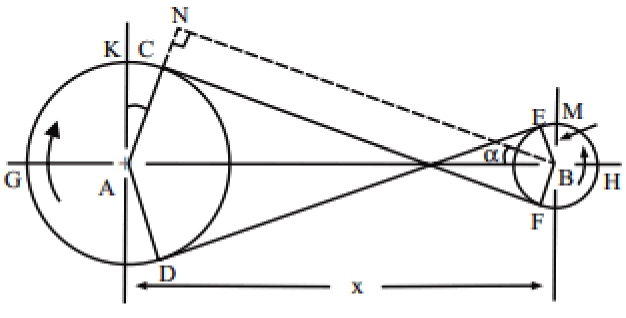
$$L=?,\quad \theta_S=?,\quad \theta_L=?$$

b) Compute the belt tensions based on a friction coefficient of 0.38.

$$F_1=?,\quad F_2=?$$

c) Calculate the speed ratio and rotational speeds.

$$w_1=?,\quad w_2=?\quad w_2/w_1=?$$



Given values

$$P = 60\ kW \qquad mass = 2\ kg/m$$

$$V = 25\ m/\sec \qquad width = 200\ mm = 0.2\ m$$

a) $$L = \sqrt{4C^2 - (D+d)^2} + \frac{\theta}{2}(D+d) \qquad \theta_S = \theta_L = \pi + 2\sin^{-1}\left(\frac{D+d}{2C}\right)$$

$$\theta = \theta_S = \theta_L = \pi + 2\sin^{-1}\left(\frac{D+d}{2C}\right) \text{ for driving pulley}$$

$$\theta = \pi + 2\sin^{-1}\left(\frac{900+300}{2\times 6000}\right) = 3.1415 + 2\times 0.1$$

$$\theta = 3.3415\ radian = 191.45^0$$

$$L = \sqrt{4\times(6000)^2 - (900+300)^2} + \frac{3.3415}{2}(900+300)$$

$$L = 11940 + 2005 = 13945\ mm = 13.945\ m$$

b)

$F_1 = ?, \quad F_2 = ?$ 2 equations and 2 unknowns (F_1 and F_2) so it is soluble

$$\frac{F_1}{F_2} = e^{f\theta} \qquad P = (F_1 - F_2)\times V \qquad (1)$$

$$\frac{F_1 - F_c}{F_2 - F_c} = e^{f\theta} \qquad (2)$$

$$F_C = m\times V^2 = 2\times(25)^2 = 1250\ N$$

$$\frac{F_1 - 1250}{F_2 - 1250} = e^{0.38\times 3.3415} = 3.56 \Rightarrow F_1 - 1250 = 3.56F_2 - 4450$$

$$F_1 = 3.56F_2 - 3200 \quad \text{into (1)}$$

$$60000 = \left((3.56F_2 - 3200) - F_2\right)\times 25$$

$$F_2 = \frac{5600}{2.56} = 2187.5\ N$$

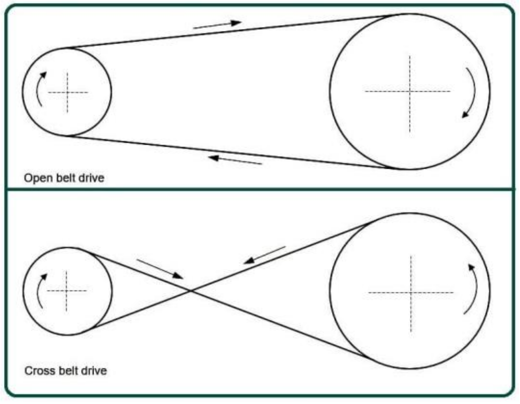
$$F_1 = 4587.5\ N$$

$$F_i = ?$$

$$F_i \le \frac{F_1}{2} \le 2293.75\ N$$

c) Calculate the speed ratio and rotational speeds.

Speed reduction



$$V_{belt} = w \times r = \text{uniform} = \text{constant}$$

$$V_{belt} = w_1 \times \frac{d}{2} = w_2 \times \frac{D}{2}$$

$$\frac{w_1}{w_2} = \frac{D}{d} = \frac{900}{300} = 3$$

Rotational Speeds

$$rad/\sec \times \frac{60 \sec}{1 \min} \times \frac{1}{2\pi\ rad} = rpm$$

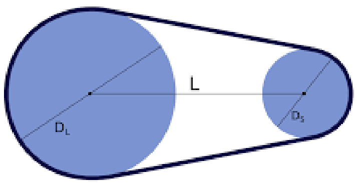
$$w_1 = \frac{V}{d/2} = \frac{2 \times 25\ m/s}{0.3\ m} = 166.66\ rad/\sec = 1591.5\ rpm$$

$$w_2 = \frac{w_1}{3} = 55.55\ rad/\sec = 530.5\ rpm$$

EXAMPLE 9.2

A flat belt is 250 mm wide and 8 mm thickness.

The **open belt** connects a 400 mm diameter cast iron driving pulley to another cast iron pulley 900 mm diameter.



The mass of the belt is 1.9 kg/m of belt length.

The coefficient of friction between belt and driving pulley is 0.28.

a) Using a center distance of 5 m, a maximum tensile stress in the belt of 1400 kPa and a belt velocity of 20 m/sec, calculate the maximum power that can be transmitted.

b) What are the shaft loads?

SOLUTION

$$Pover = F_{net} \times V = (F_1 - F_2) \times V$$

$$F_1 = ? \qquad F_2 = ? \qquad V = 20\ m/s$$

$$\begin{bmatrix} P = T \times w \\ P = (F_1 - F_2) \times r \times w \\ P = (F_1 - F_2) \times (w \times r) \end{bmatrix}$$

a)

$$\frac{F_1 - F_c}{F_2 - F_c} = e^{f\theta} \qquad \mu = 0.28$$

$$\text{Tensile stress} \quad \sigma_{\max} = 1400\, kPa$$

$$F_{\max} = F_1 = \sigma_{\max} \times Area$$

$$F_{1_{\max}} = 1400 \times 10^3 \times (0.25 \times 0.008) = 2800\ N$$

$\theta_S =$ driving pulley angle of wrap

$$\theta_S = \pi - 2\sin^{-1}\left(\frac{D-d}{2C}\right) = 3.1415 - 2\sin^{-1}\left(\frac{900-400}{2\times 5000}\right)_{rad}$$

$$\theta_S = 3.1415 - 2\times 0.05 = 3.0415\ radian$$

$$F_C = m \times V^2 = 1.9 \times (20)^2 = 760\ N$$

$$F_2 = \frac{F_1 - F_c}{e^{f\theta_S}} + F_c = \frac{2800 - 760}{e^{0.28\times 3.0415}} + 760 = 1630.6\ N$$

$$P_{\max} = (F_1 - F_2)\times V = (2800 - 1630.6)\times 20 = 23388\, Watt = 23.4\, kW$$

(with a safety factor of 1)

$$\text{if } n = 2 \quad F_1 = \frac{F_{1\max}}{2} = 1400\ N$$

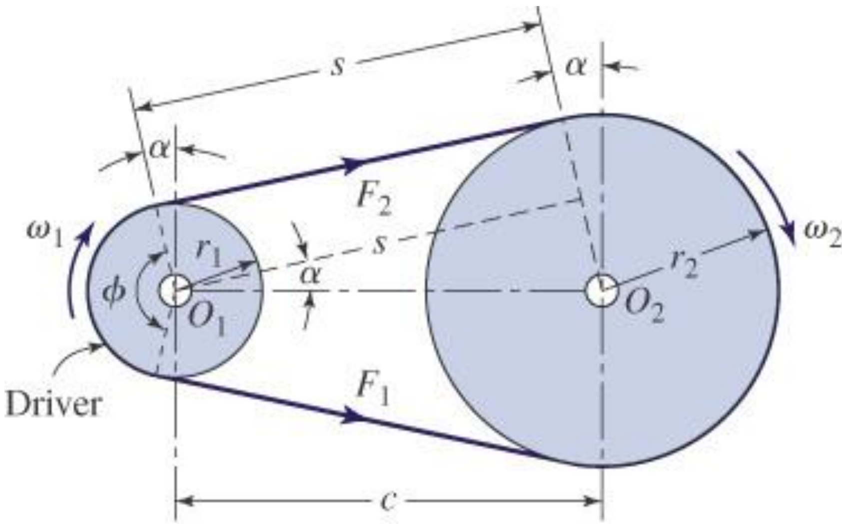
$$F_2 = 1033\ N$$

$$P_{\max} = (1400 - 1033)\times 20 = 7340\, Watt = 7.34\, kW$$

b)

$$F_{shaft_X} = F_1 \cos\theta + F_2 \cos\theta$$

$$F_{shaft_Y} = (F_1 - F_2)\sin\theta$$



(a)

$$\theta = \sin^{-1}\left(\frac{D-d}{2C}\right) = 0.05\,radian$$

$$\theta = 2.866^o$$

$$F_{shaft_X} = F_1 \cos\theta + F_2 \cos\theta$$

$$F_{shaft_X} = (2800 + 1630.6) \times \cos 2.866 = 4425\ N$$

$$F_{shaft_Y} = (F_1 - F_2)\sin\theta$$

$$F_{shaft_Y} = (2800 - 1630.6)\sin 2.866 = 58.5\ N$$

$$F = \sqrt{(4425)^2 + (58.5)^2} = 4425.4\ N$$

9.3 V-BELTS

V-belts as in rolling contact bearings are selected from standards rather than designing them.

A standard V-belt is chosen from the catalogue and its (or their if multiples) power transmitting capacity is calculated and checked with the required power transmission. If one is not enough to transmit the required power, then multiples of them can be used;

$$\text{no. of V-belts} = \frac{\text{power required}}{\text{transferred power/belt}}$$

Power transmitting capacity of a single V-belt is given by

$$H_r = \left[c_1 - \frac{c_2}{d} - c_3(r \times d)^2 - c_4 \log(r \times d)\right](r \times d) + c_2 \times r\left(1 - \frac{1}{K_A}\right)$$

Constants $c_1 \ldots\ldots c_4$ depend on belt section and given in Table 17.5.

$$r \quad = \quad \text{rpm of high speed shaft divided by } 1000$$

$$K_A = \text{ speed ratio factor given in Table } 17.6$$

$$d \quad = \text{pitch diameter of small sheave (pulley) in mm}$$

$$H_r = \text{rated power in kWatts}$$

$$\text{(If d in inches then } H_r \text{ in horse power)}$$

The power H_r is calculated on the assumption that $\Theta_s = 180^o$.
If $\Theta_s \neq 180^o$ (<180 generally since V-belts can be used only in open configuration.)
Then

$$H' = H_r \times K_1 \times K_2$$

K_1 given in Table 17-8 for tooth correction factor
K_2 given Table 17-9 for angle of wrap correction

Tables 17-2 and 17-3 are used to choose standard V-belts

The length (pitch length) L_p of a V-belt is calculated as:

$$L_p = 2C + 1.57(D+d) + \frac{(D-d)^2}{4C}$$

$$L_p = L_s + conv.\ factor\ (Table\ 17.4)$$

$$\text{where } C = \frac{K + K^2 - 32(D-d)^2}{16} \rightarrow D \le C \le 3(D+d)$$

$$K = 4 \times L_p - 6.28(D+d)$$

Thus,

$$\text{req. no. of V-belts} = \frac{\text{power required} \times \text{service factor (Table 17.9)}}{\text{H}' \quad \text{(per belt)}} = n.38 \cong n+1$$

EXAMPLE 9.3

Sheave: çıkrık, makara, bobin, oluklu kasnak

Select on SI heavy duty V-belt to transmit 50 kW at a speed of 1590 rpm for the small sheave. There is to be a speed reduction of 3:1

Select pulley sizes (D and d) centerline distance (C), belt type and number. The driver is a short motor and the driven is a liquid agitator with 24 hour service.

Select from Table 17-2 heavy-duty SI conventional “22c” with

Let $d_{min} = 224\ mm$

$$d = 230\ mm \rightarrow \quad D = 3 \times d = 690\ mm$$

$$D < C < 3(D+d) \quad let\ C = 800\ mm$$

$$L_P = 2C + 1.57(D+d) + \frac{(D-d)^2}{4C}$$

$$L_P = 2 \times 800 + 1.57(690+230) + \frac{(690-230)^2}{4 \times 800} = 3110.5\ mm$$

From Table 17.3 choose 22c series $L_P = 3150 \text{ with} \quad K_2 = 1.00$

For $D = 690\ mm$ and *C*= *820* mm for above values

$d = 230\ mm$

$L_P = 3150\ mm$

where $C = \frac{K + K^2 - 32(D-d)^2}{16} \rightarrow D \le C \le 3(D+d)$

$K = 4 \times L_p - 6.28(D+d)$

$$H_r = \left[c_1 - \frac{c_2}{d} - c_3 (r \times d)^2 - c_4 \log(r \times d) \right] \times r \times d + c_2 \times r \left(1 - \frac{1}{K_A}\right)$$

$$r = \frac{1590}{1000} = 1.59$$

$$d = 230\ mm$$

$$c_1 = 0.10002$$

$$c_2 = 7.040$$

$$c_3 = 3.326 \times 10^{-8}$$

$$c_4 = 1.5 \times 10^{-2}$$

$$K_A = 1.1106 \quad for\ \frac{D}{d} > 1.64$$

$$H_r = 10.9\ kW/belt$$

$$\theta_S = \pi - 2\theta = \pi - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$H^{'} = H_r \times K_1 \times K_2$$

$$K_1 = f(\theta_s)$$

$$\theta_s = \pi - 2\sin^{-1}(\frac{D-d}{2C}) = 2\cos^{-1}(\frac{D-d}{2C})$$

$$\theta_s = 2\cos^{-1}(\frac{690-230}{2\times 820}) = 147.42^o \rightarrow$$

$$K_1 = 0.92 \quad (Fig. 17.4)$$

$$K_2 = 1.00 \quad (in\ Table\ 17.7)$$

$$\#\text{of V-belts} = \frac{\text{power req.} \times \text{serv. factor (Tbl 17.9)}}{\text{H}^{'} \quad \text{(per belt)}}$$

$$H^{'} = 10.9 \times 0.92 \times 1.0 = 10.028\ kW\ /\ belt$$

$$\text{service factor} = 1.3$$

$$\#\ of\ \ belts = \frac{50 \times 1.3}{10.028} = 6.480 \rightarrow 7\ belts$$

(This is a large number for multiple V-belts) (1-4 usually)

Re-choose 32c series with d_{min} = *355 mm*

$$L_P = 2C + 1.57(D+d) + \frac{(D-d)^2}{4C}$$

$$Let \rightarrow d = 360\ mm \rightarrow D = 3\times 360 = 1080\ mm$$

$$Let \rightarrow C = 2000\ mm \begin{pmatrix} 1080 & - & 4320 \\ D & & 3(D+d) \end{pmatrix}$$

$$L_P = 2\times 2000 + 1.57(1080+360) + \frac{(1080-360)^2}{4\times 2000} = 6325.6\ mm$$

From table 17-3 32c series

$$L_P = 6560 \text{ with } \quad K_2 = 1.02$$

for

$$D = 1080\ mm$$

$$d = 360\ mm \quad \rightarrow C = 2119\ mm$$

$$L_P = 6560\ mm$$

$$H_r = \left[c_1 - \frac{c_2}{d} - c_3(r\times d)^2 - c_4 \times \log(r\times d) \right] \times r \times d + c_2 \times r(1 - \frac{1}{K_A})$$

$$H_r = \left[0.2205 - \frac{26.62}{360} - 7.037\times 10^{-8}(1.59\times 360)^2 - 0.0317\times \log(1.59\times 360) \right] \times$$

$$(1.59\times 360) + 26.62\times 1.59(1 - \frac{1}{1.1106})$$

$$H_r = 24.55\ kW/belt$$

$$\theta_s = \pi - 2\sin^{-1}\left(\frac{D-d}{2C}\right) = 2\cos^{-1}\left(\frac{D-d}{2C}\right)$$

$$D = 1080\ mm$$
$$d = 360\ mm \quad \rightarrow C = 2119\ mm$$
$$L_P = 6560\ mm$$

$$\theta_s = 2\cos^{-1}\left(\frac{1080-360}{2\times 2119}\right) = 160.437^o \rightarrow$$

where $C = \dfrac{K + K^2 - 32(D-d)^2}{16} \rightarrow D \le C \le 3(D+d)$

$K = 4\times L_p - 6.28(D+d)$

$$\theta_s = 160.437^o \rightarrow$$
$$K_1 = 0.96 \quad (Fig. 17.4)$$
$$K_2 = 1.00 \quad (in\ Table\ 17.7)$$

$$H_r = 24.55\ kW/belt$$
$$H' = H_r \times K_1 \times K_2$$

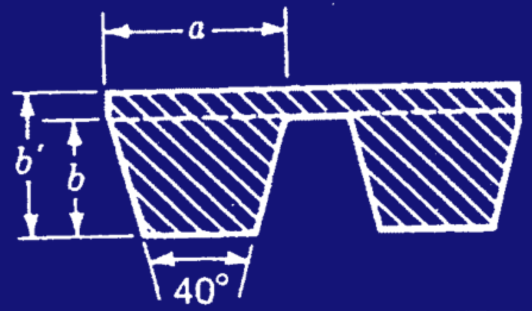
$$H' = 24.55\times 0.96\times 1.0 = 23.57\ kW/belt$$

$$\text{service factor} = 1.3$$

$$\text{req. no. of V-belts} = \frac{\text{power required} \times \text{service factor (Table 17.9)}}{\text{H}' \ \text{(per belt)}} = n.38 \cong n+1$$

$$\#\ of\ belts = \frac{50\times 1.3}{23.57} = 2.75 \rightarrow 3\ belts\ are\ SUITABLE\ !...$$

Table 17-2 SIZES AND RATINGS OF HEAVY-DUTY SI CONVENTIONAL V-BELT SECTIONS
(Sizes in millimeters)



Section	Width a	Type of belt: Single thickness b	Type of belt: Multiple thickness b'	Power range per belt, kW	Minimum sheave size†
13C(SPA)*	13	8	10	0.1–3.6	80
16C(SPB)	16	10	13	0.5–72	140
22C(SPC)	22	13	17	0.7–15	224
32C	32	19	21	1.3–39	355

* Designation in parentheses is from BS 3790: 1973.
† Smaller sheaves are available but their use will shorten belt life.
Sources: ANSI/RMA-IP-1977 and M. J. Neale (ed.), *Tribology Handbook*, Butterworths, London, 1973, p. A-47.

Select

Table 17-3 STANDARD PITCH LENGTHS FOR HEAVY-DUTY SI CONVENTIONAL V BELTS

Section	Length, mm
13C	710, 750, 800, 850, 900, 950, 1000, 1075, 1120, 1150, 1230, 1300, 1400, 1500, 1585, 1710, 1790, 1865, 1965, 2120, 2220, 2350, 2500, 2600, 2730, 2910, 3110, 3310
16C	960, 1040, 1090, 1120, 1190, 1250, 1320, 1400, 1500, 1600, 1700, 1800, 1900, 1980, 2110, 2240, 2360, 2500, 2620, 2820, 2920, 3130, 3330, 3530, 3740, 4090, 4200, 4480, 4650, 5040, 5300, 5760, 6140, 6520, 6910, 7290, 7670
22C	1400, 1500, 1630, 1830, 1900, 2000, 2160, 2260, 2390, 2540, 2650, 2800, 3030, 3150, 3350, 3550, 3760, 4120, 4220, 4500, 4680, 5060, 5440, 5770, 6150, 6540, 6920, 7300, 7680, 8060, 8440, 8820, 9200
32C	3190, 3390, 3800, 4160, 4250, 4540, 4720, 5100, 5480, 5800, 6180, 6560, 6940, 7330, 8090, 8470, 8850, 9240, 10 000, 10 760, 11 530, 12 290

Select

Table 17-8 TOOTH CORRECTION FACTORS K_1 BASED UPON THE NUMBER OF TEETH N ON THE DRIVING SPROCKET

N	K_1	N	K_1	N	K_1	N	K_1
11	0.53	17	1.00	23	1.35	35	1.95
12	0.62	18	1.05	24	1.41	40	2.15
13	0.70	19	1.11	25	1.46	45	2.37
14	0.78	20	1.18	27	1.57	50	2.51
15	0.85	21	1.26	29	1.68	55	2.66
16	0.92	22	1.29	31	1.77	60	2.80

Table 17-9 MULTIPLE-STRAND FACTORS K_2

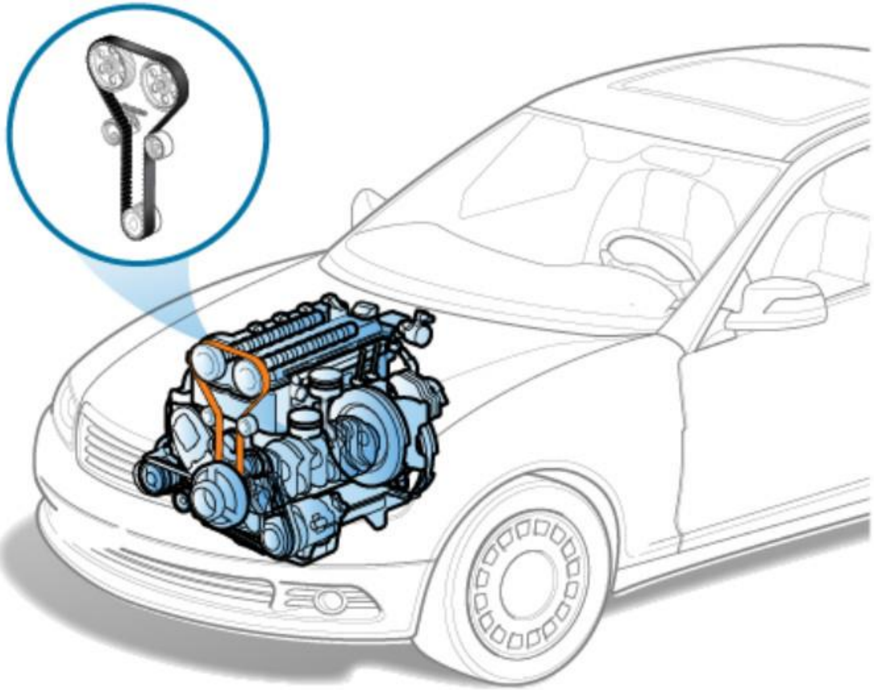
Number of strands	K_2
1	1.0
2	1.7
3	2.5
4	3.3

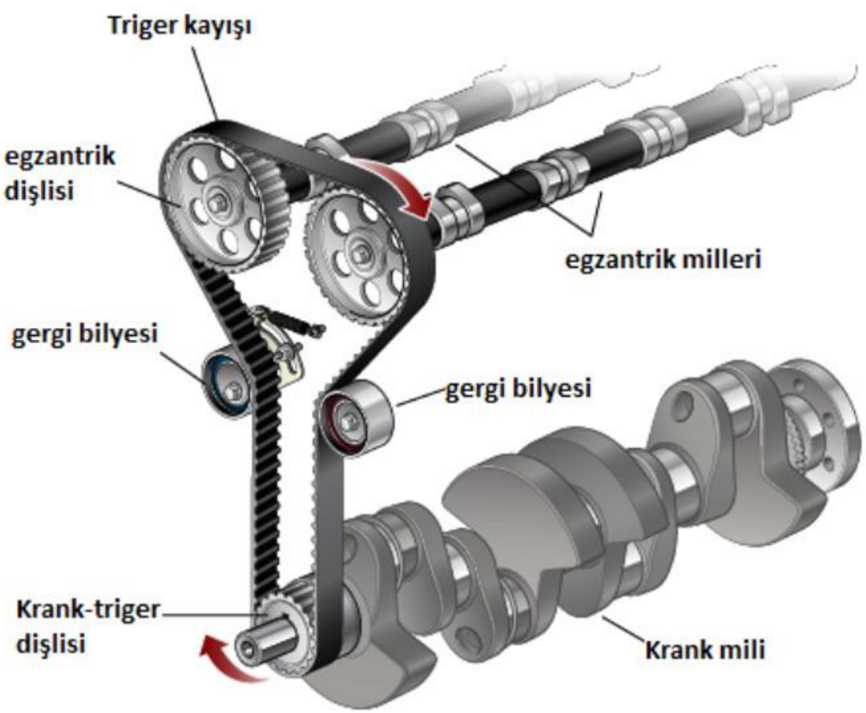
Table 17-10 LOAD SERVICE FACTORS K_s

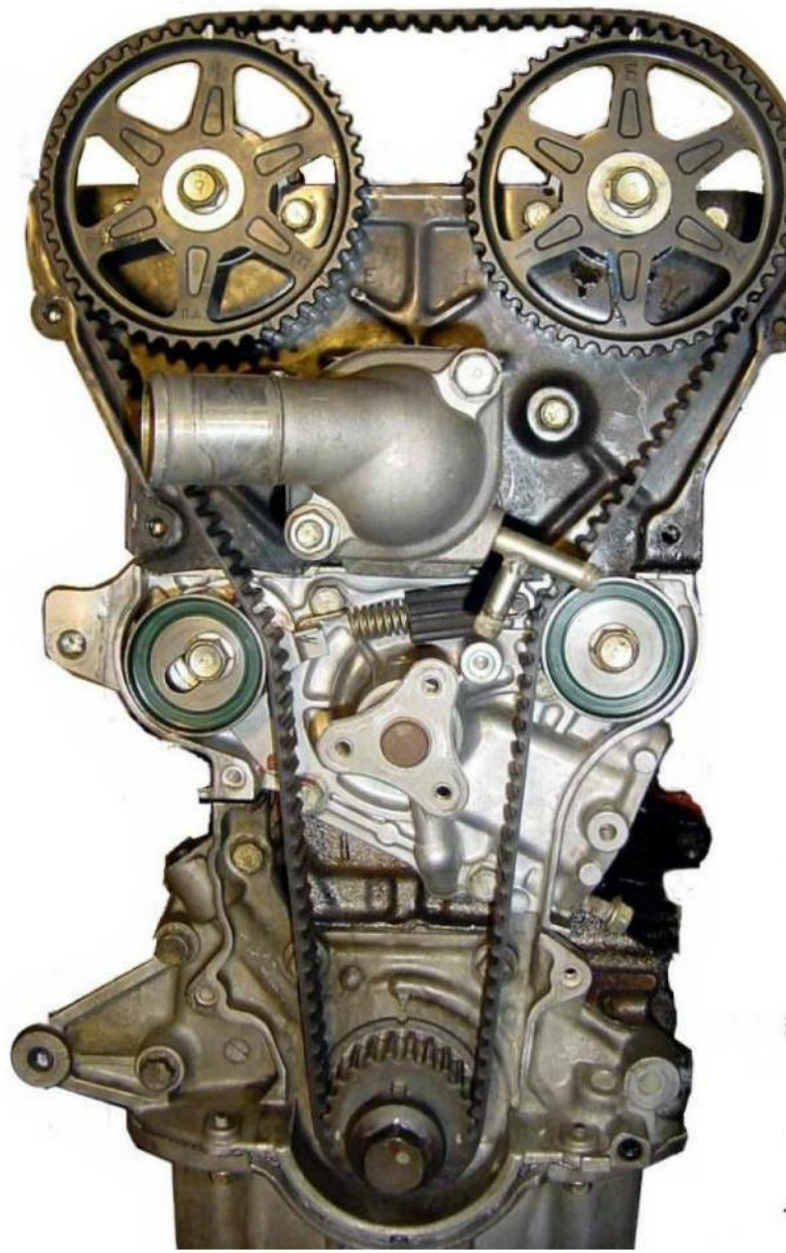
	Driving source		
Driven machinery	Internal combustion engine with hydraulic drive	Electric motor or turbine	Internal combustion engine with mechanical drive
Smooth	1.00	1.00	1.2
Moderate shock	1.2	1.3	1.4
Heavy shock	1.4	1.5	1.7

Select


T304




Triger kayışı
egzantrik dişlisi
egzantrik milleri
gergi bilyesi
gergi bilyesi
Krank-triger dişlisi
Krank mili




Devridaim pompası
Triger kayışı
Gergi bilyesi