UNIT 9 BELT AND CHAIN DRIVES

Structure

9.1 INTRODUCTION

Belt and chain drives are two common mechanical drives. While belts derive their driving force from friction between belt and pulley surfaces, chains have links which interfere with teeth on sprocket and thus develop the driving tension.

Belt and chain drives are two systems from a broad category of power transmission systems. A power transmission system is interposed between driving prime mover and driven machine for a number of reasons which are described as follows :

(a) The working velocity of driven machine is often not constant and it may not coincide with the constant speed of driving machine.

(b) Having a prime mover which can change speed with the requirement of driven machine may not be economical.

(c) The driven machine may require a torque which may not be produced by driving machine, hence, speed of driven machine may have to be reduced.

(d) A single prime mover may drive a number of machines.

(e) A driven machine even have to pause and driving machine may not sustain frequent stopping and starting.

(f) The driving and driven machines may not be directly coupled for reasons of safety and convenience of operation.

The examples of such systems are quite common in industrial and daily engineering application. An automobile is a common example, power plant equipment, washing machines, lathes, milling machines, rolling and crushing mills are other examples.

Power transmission systems in general can be classified as

(a) Electrical,

(b) Mechanical,

(c) Pneumatic, and

(d) Hydraulic.

These systems are judged on the basis of following characteristics :

(a) Centralized power supply – electrical and pneumatic systems

(b) Power transmission over long distance – electrical systems

(c) Accumulation of power – hydraulic systems

(d) Step-by-step velocity change – electrical and mechanical systems

(e) Accurate velocity ratio – mechanical drives which mesh

(f) High velocity of rotation – electrical and pneumatic

(g) No effect of ambient temperature – electrical and mechanical systems

(h) Automatic and remote control – electrical

Out of all above we will be concerned with the study of mechanical systems which can be further divided into two categories :

(a) Mechanical systems based on interference or mesh : gears and chains are such drives.

(b) Mechanical systems based on friction : belt drives, clutches and friction drives are such systems.

Objectives

After studying this unit, you should be able to

- describe belt types, characteristics and types of belt drives
- know materials for belt,
- calculate power transmitted by belt,
- calculate the strength of the belt, and
- design pulleys on which belt will run.

9.2 BELT DRIVES

Two types of belt are in common use. They are :

(a) Flat belt, and

(b) V-belt.

A third one similar in operation is the rope drive. We will confine to study the flat and V-belt drives. A flat belt has a rectangular cross-section and runs on a flat surface pulley. A V-belt has a trapezoidal cross-section and runs on a grooved pulley. The belt section and pulley grooves match exactly. Both flat belts and V-belts are available as manufactured items. Whereas a flat belt is cut from a sheet in desired width and length and then joined end to end to make it endless, the V-belts are much more standardised in single piece endless form. Their cross-sections are standardised and six of them are commercially available. The same section may be available in several pitch lengths and several of them may be used between same two grooved pulleys.

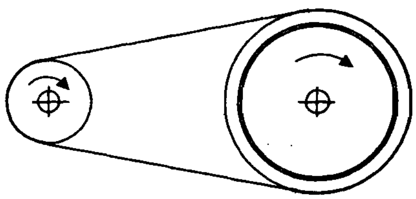
9.3 FLAT BELT DRIVES

For transmitting power between two shafts at considerable distance apart, flat belt drive is most commonly used. The centre distance may be as large as 15 m, and even may exceed this in some cases. Flat belt is a flexible connector of rectangular cross-section connecting two pulleys which are mounted upon two shafts. The power from one shaft is transmitted to another pulley by friction between belt and pulley surfaces. The force of friction between driving pulley and belt drives the belt and that between belt and driven pulley drives the driven pulley.

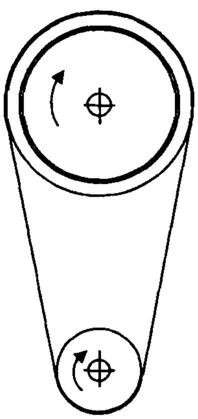
The flat belts connecting pulleys can be arranged in a number of ways depending upon the purpose of the drive. The arrangements are described below :

Open Belt Drive

An endless belt goes around two pulleys one of which is driving and another driven. Tight side of belt, as a rule, is kept lower so that the catenary action of the top portion is utilised in increasing angle of lap of the pulleys. In this arrangement, as shown in Figures 9.1(a) and (b), the two pulleys rotate in the same direction. The centre line of the pulleys could be horizontal, vertical and inclined to horizontal.



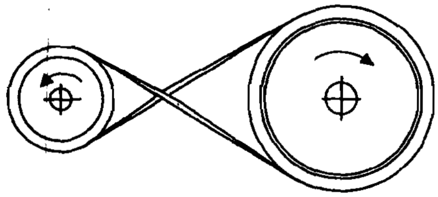
(a) Horizontal Open Drive



(b) Vertical Open Drive

Cross or Twist Belt Drive

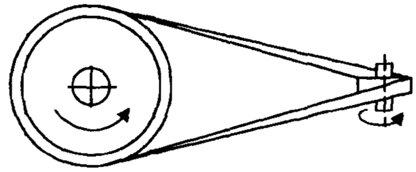
Belt running on two parallel shafts but crossing itself, as shown in Figure 9.1(c) would cause the pulleys to rotate in the direction opposite to each other. In such an arrangement excessive belt wear would occur because the belt will rub on itself. This wear is minimised if centre distance is larger than twenty times the width of belt. The speed of this type of drive must not exceed 15 m/s and for belts wider than 200 mm the idler pulley must be used to reduce rubbing. Capacity of such belt drive is only 75% of an equivalent open belt drive if the ratio of pulley diameters is less than 3 : 1. If, however, this ratio is higher the capacity of drive is only 50%.



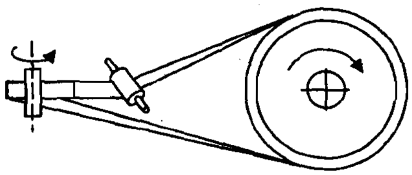
(c) Cross Belt Drive

Quarter-Twist Drive

In this arrangements two shafts are at right angle to each other. The chance of slipping can be avoided by providing larger face width of pulley. For this arrangement, illustrated in Figures 9.1(d) and (e), pulley face width exceeds the belt width by at least 40%.



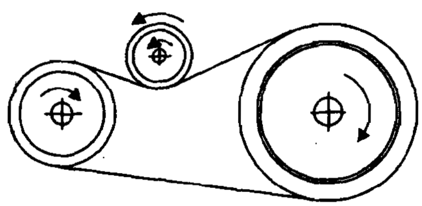
(d) Quarter Twist Drive



(e) Quarter Twist Drive with Idler

Belt Drive with an Idler Pulley

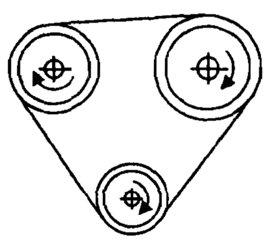
When centre distance between the shafts is short, the angle of lap on smaller pulley becomes small. This smaller angle of lap does not permit high belt tension. If it is not possible to increase belt tension by any other means to the desired value, then an idler pulley, as shown in Figure 9.1(f) is used to increase the belt tension. This drive has a disadvantages in the sense that belt bends in two opposite direction.



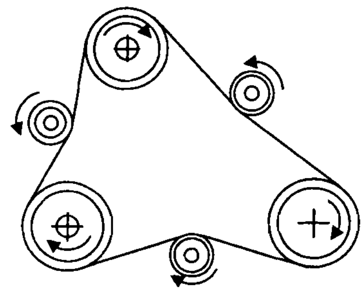
(f) Belt Drive an Idler Pulley

Belt Drives with many Pulleys

Figures 9.1(g) and (h) show an arrangement in which a number of pulleys are driven by a single belt. In such an arrangement either all the pulleys could rotate in the same direction [as in Figure 9.1(g)] or some may rotate in the direction opposite to that of others [as in Figure 9.1(h)].



(g) Belt Drive with Several Pulleys Rotating in the Same Direction



(h) Belt Drives with Several Pulleys Rotating in the Same and Opposite Direction

Figure 9.1 : Different Arrangements in Belt Drives

Pivoted Motor Drive

In this short centered belt drive the belt tension is enhanced by using the weight of the driving motor. This will be discussed in detail at a later stage.

9.4 CHARATERISTICS OF FLAT BELT DRIVES

The flat belt drives are simple in design and fabrication and are cheap (costlier than V-belt drives only). These drives can be used in dusty and abrasive atmosphere and can work at considerably high speeds. Owing to elasticity and flexibility of belts, these drives are capable of absorbing shocks. Thy protect the driven mechanism in case of sudden overload because belts slip out form the pulleys when overloaded. Their capability to transmit power between shafts at considerable distance apart is an added advantage. These drives operate noiselessly (particularly if compared with gear drives) and need minimum maintenance. Flat belts are capable of transmitting power between shafts arranged in several ways in space (Figure 9.1).

Belt drive is unique in one sense, that the effective driving peripheral force on the pulley is the difference of the belt tensions on two sides of the pulley but the sum of these two belt tension components act as bending load on the shaft. Thus, the shaft size in a belt driver is larger than in a great derive for comparable power. The velocity ratio of belt drives does not remain constant but changes with load. When belt has operated for some time it suffers from permanent stretching which often needs adjustment of centre distance or some other method of belt tensioning.

9.5 MATERIALS FOR FLAT BELTS

The usual requirements of materials for machine elements are strength and durability, so belt materials are also required to be strong and durable. In addition, a belt material must have such properties as flexibility and high coefficient of friction. In general, four different materials are used for flat belts; they are :

(a) Leather,

(b) Rubber,

(c) Woven cotton, and

(d) Woven wool.

9.5.1 Leather Belts

Leather belts are made of animal hides. The best quality of leather belts are manufactured from narrow strip of hide extending 375 mm on either side of back bone along a length of 1350 mm from tail. This portion, though not strongest, is best for beltings because it is flexible. Double play belts are obtained by cementing the strips along flesh side.

The commercially available belts are classified according to tanning treatment given to them. The belts are either oak-tanned or chrome tanned. Both the belts are equally strong but oak tanned belt is stiffer and should not be used when required to double on itself.

The leather belts as compared to others, are costlier. For this reason, they are used only when special design characteristics call of their use. However, the use of leather belt is decreasing fast in favour of rubber belts.

9.5.2 Rubber Belts

Rubber belts are made by vulcanising the layers of strong fabric or canvas with rubber in between. While the fabric or canvas provides all the strength rubber acts as a protector against atmospheric agencies, such as moisture and also prevents relative sliding of layers of fabric. Since these belts are cheaper than leather belts, they are largely used in practice. They remain unaffected by heat, light and oil. They deteriorate with time.

If rubber is replaced by balata gum, the strength of belts increases by about 25%, but balata gum belts lose strength beyond 48°C. Balata does not need vulcanising and does not deteriorate or oxidise in air. Although it is not affected by vegetable oil, mineral oil severely affects the balata belt and for this reason, the use is on decline.

Saw mills, creameries, chemical plants and paper mills largely use the rubber belts.

Modern high performance belts are made in multiple plies consisting of tension ply and friction ply. The tension ply takes the driving tension and centrifugal tension. The friction ply provides the necessary friction for generating tension in the tension ply. The tension ply is made of polymide strips or polyester cords. The friction ply firmly adhering to tension ply is made of either synthetic rubber or polyurethane or chrome tanned leather.

9.5.3 Fabric or Canvas Belts

Fabric or canvas belts are obtained in single ply by directly weaving them on looms. Alternatively thin layers of fabric can be stitched together to obtain multiply belts. These belts are often impregnated in linseed oil which fills in the voids and prevents atmosphere from reacting with the fabric. However, this filler makes the belts stiffer.

Table 9.1 : Characteristics of Flat Belt Materials

Characteristics	Belt Materials			
	Leather	Rubber	Woven Cotton	Woven Woolen
Width, b (mm)	20-300	20-500	30-250	50-500
Thickness, h (mm)	Single ply 3.0-5.5 Double ply 7.5-10.0	2.5-13.5	4.5-8.5	6.9-11.0
Ultimate tensile strength (MPa)	20.0	With layers 37.0 Without layers 44.0	35.0-40.0	30.0
Maximum elongation	10% at 10 MPa	18% at rupture	20% at rupture	60% at rupture
Ratio D_{min}/h Recommended	35	40	35	30
Ratio D_{min}/h Permissible	25	30	30	25
Maximum velocity v_{max} (m/s)	40	25	25	30
Specific weight (N/m^3)	9800	13750	9000	10700
Modulus elasticity (MPa)	125	100	45	–

These belts are much cheaper and are used where intermittent power supply is required. They find application specially in hot and dry atmosphere, and where much attention may not be available. They are most suitable for farm work, quarry and saw mills.

9.5.4 Woven Woolen Belts

In these belts woolen fibres and cotton picks of wefts are used for weaving. They are impregnated with a mixture of drying oil and chalk. These belts are not very strong.

Some of the important properties of normal belt materials are described in Table 9.1. Table 9.2 describes important properties of high performance belt.

Table 9.2 : Characteristics of High Performance Belts

Characteristics	Materials of Tension Ply	
	Polyamide	**Polyester**
Width, *b* (mm)	1000 max.	450 max
Thickness, *h* (mm)	1.0 to 8.0	0.8 to 4.0
Length (mm)	Unlimited	12000 max
Ultimate tensile strength (MPa)	450-600	700-900
Maximum elongation (%)	22	12-15
Service elongation (%)	1.5-3.0	1.0-1.5
Maximum tolerable bending frequency (1/s)	80-100	100-250
Maximum velocity (m/s)	60-80	80-150
Specific nominal peripheral force (N/mm)	4-80	10-40
Specific nominal power (kW/mm)	≤ 4.5	≤ 6.0
Elastic creep (%)	0.8-1.0	0.4-0.6

9.6 RATIO OF BELT TENSIONS

When a belt is moving round a pulley and transmitting power, the tension in belt on two sides of pulley will be different. The side of belt in which tension is higher is the tight side and other is called slack side. Figure 9.2(a) shows pulley on which a belt is running. The tension in tight side is T_t and that in slack side T_s.

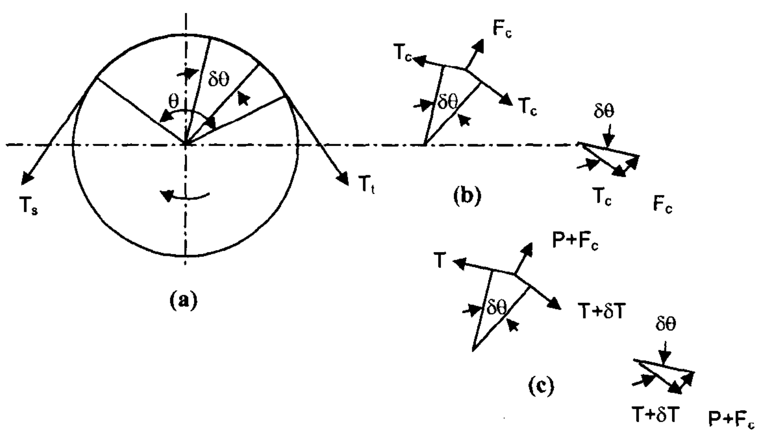
Following notations will be used.

v = Peripheral velocity of pulley or belt speed,

ω = Weight of belt per unit length,

r = Radius of pulley, and

θ = Angle of lap of belt over pulley.



(a) A Belt Round a Pulley. Pulley Rotates in Direction of T_t

(b) Belt Element under Action of Centrifugal Force F_c and Centrifugal Tension T_c

(c) Belt Element under Action of Differential Tension $T + \delta T$ and T, Normal Reaction and Centrifugal Force (P and F_c)

Figure 9.2

As the belt moves round the pulley it would experience a centrifugal force which has a tendency to separate the belt from the pulley surface. To maintain contact between pulley and belt, a tension, known as centrifugal tension would develop in the belt which would be in addition to the tension required for power transmission. Centrifugal tension becomes an important criterion in design of belting at high speeds.

In Figure 9.2(a) a small length of belt, subtending an angle $\delta\theta$ at the centre is chosen. The weight of this element of belt would be $\omega\, r\, \delta\, \theta$, and therefore, the centrifugal force acting on this element would be

$$F_c = \frac{\omega r \delta\theta \,.\, v^2}{rg} = \frac{\omega v^2}{g}\, \delta\theta \qquad \ldots (9.1)$$

Assuming that this centrifugal force causes centrifugal tension T_c on either side of the belt, a force triangle as shown in Figure 9.2(b) can be drawn from which

$$F_c = T_c\, \delta\theta$$

$$F_c = \frac{\omega v^2}{g} \qquad \ldots (9.2)$$

Since centrifugal tension is proportional to square of belt speed, it becomes an important factor in design if belt speed is high.

The tensions T_t and T_s tend to maintain the contact between pulley and belt causing a radial reaction between the two. In fact T_t and T_s both are composed of two parts, out of which one each equals the centrifugal tension and other provides driving force. Thus,

$$T_t = T_1 + T_c \qquad \ldots (9.3)$$

and $$T_s = T_2 + T_c$$

The driving force on pulley is the difference of T_1 and T_2. Let the radial reaction be P on the small belt element that subtends an angle $\delta\theta$ at the centre. Let the tension on one side of the belt element be T and on the other side it increases by a small amount δT, because of friction between belt and pulley. Hence, if coefficient of friction between belt and pulley is μ, then

$$P = \frac{\delta T}{\mu} \qquad \ldots (9.4)$$

The belt element could be regarded to be in equilibrium under the action of forces, P, F_c and $T + \delta T$. If these are represented by a force triangle as in Figure 9.2(c), then

$$T\,\delta\theta = P + F_c$$

From Eqs. (9.1), (3.2) and (9.4)

$$T\,\delta\theta = \frac{\delta T}{\mu} + T_c\, \delta\theta$$

i.e. $$(T - T_c) = \frac{\delta T}{\mu\,\delta\theta}$$

or $$\mu\,\delta\theta = \frac{\delta T}{T - T_c}$$

or $$\int_0^\theta \mu\,\delta\theta = \int_{T_s}^{T_t} \frac{dT}{T - T_c}$$

i.e. $$\mu\theta = \ln \frac{T_t - T_c}{T_s - T_c}$$

or
$$\frac{T_t - T_c}{T_s - T_c} = e^{\mu\theta} \qquad \ldots (9.5)$$

From Eq. (9.3)
$$\frac{T_1}{T_2} = e^{\mu\theta} \qquad \ldots (9.6)$$

9.7 POWER TRANSMITTED BY A BELT

Knowing tensions on tight and slack sides of a belt, it is straight forward to calculate the power, H. If the linear velocity of belt is v m/s then

$$H = (T_t - T_s)\, v, \text{ W} \qquad \ldots (9.7)$$

when tensions are measured in N.

from Eq. (9.3) it is obvious that power is also given by

$$H = (T_1 - T_2)\, v \qquad \ldots (9.8)$$

If the driving pulley having a diameter of D_1 rotates at N_1 rpm, then

$$H = T_1 \left(\frac{k-1}{k}\right) \frac{2\pi N_1}{60} \frac{D_1}{2}$$

or
$$H = 0.0524\, T_1 \frac{k-1}{k} N_1\, D_1, \text{ W} \qquad \ldots (9.9)$$

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

9.7.1 Speed for Maximum Power

It is apparent from Eq. (9.8) that if $(T_1 - T_2)$ remains unchanged power of drive can be increased by increasing speed. However, an optimum speed corresponding to maximum power will exist beyond which centrifugal tension will increase affecting power. Call this optimum speed v_{opt}.

$$H = T_1 \left(\frac{k-1}{k}\right) v$$

$$= \left(T_t - \frac{\omega v^2}{g}\right) v \left(\frac{k-1}{k}\right)$$

$$\frac{dH}{dv} = \left(T_t - \frac{3\omega v^2}{g}\right) \frac{k-1}{k}$$

For obtaining v_{opt}, replace v by v_{opt} in RHS of the above equation and equate $\frac{dH}{dv}$ to zero.

$$\frac{3\omega\, v_{opt}^2}{g} = T_t$$

or
$$v_{opt} = \sqrt{\frac{g\, T_t}{3\omega}} \qquad \ldots (9.10)$$

9.8 STRENGTH OF A FLAT BELT

The power transmitted by a belt drive depends upon tension $T_1 = T_t - T_c$, peripheral velocity v and ratio $\frac{T_1}{T_2} = e^{\mu\theta} = k$. An increase in velocity v can apparently increase power but also increases T_c whereby T_1 is reduced so that T_t is made constant and hence, power remains a function of v only. Under normal condition of operation this condition will hold. T_t depends upon the strength of the belt because increase in T_t will increase tensile stress in the belt which is limited by ultimate tensile strength of the belt material. The ratio k depends upon angle of lap θ and coefficient of friction μ. Higher value will apparently result in higher value of k and hence higher transmitted power.

The flat belt cross-section is a rectangle of width b and height h. While, T_t, the tension in the tight side of the belt, will cause tensile stress in the section, additional bending stress in the section will be induced because of change of radius of curvature as the belt passes over pulley. Let

σ_t = Tensile stress in the belt due to T_t, and

σ_b = Bending stress in the belt due to bending.

$$\sigma_t = \frac{T_t}{b \times h} \qquad \ldots (9.11)$$

since the radius of curvature of belt changes from radius of pulley, $\frac{D}{2}$ to infinity as the belt goes round the pulley, from theory of bending,

$$\frac{\sigma_b}{\frac{h}{2}} = \frac{E}{\frac{D}{2}}$$

or

$$\sigma_b = \frac{Eh}{D} \qquad \ldots (9.12)$$

Hence, maximum tensile stress in the belt

$$\sigma_{max} = \frac{T_t}{bh} + \frac{Eh}{D} \qquad \ldots (9.13)$$

from Eq. (9.12) it is seen that smaller the value of D higher the value of σ_b and hence σ_{max}. The pulley diameter and particularly the smaller pulley diameter becomes very important in a belt drive. Smaller value of smaller pulley diameter increases σ_{max} and also has smaller value of angle of lap θ.

Eq. (9.13) can be rewritten in the following form,

$$T_t = bh\left[(\sigma_{max}) - \frac{Eh}{D}\right] \qquad \ldots (9.14)$$

Here σ_{max} may be the permissible tensile stress in the flat belt material. This permissible stress can be a fraction of ultimate tensile strength or at best equal to the tensile strength itself. Eq. (9.13) can be used for design of a belt drive, though in a restricted way. The procedure may be exemplified later in the example.

The effective coefficient of friction between the belt and pulley depends upon materials of both pulley and belt and also upon the velocity of the belt. The pulleys are made in steel, cast iron, compressed paper and wood. The coefficient of friction in different combinations is described in Table 9.3.

Table 9.3 : Coefficients of Friction

Belt Material	Pulley Material						
	Steel and Cast Iron			Compressed Paper	Wood	Leather Face	Rubber Face
	Dry	Wet	Greasy				
Leather (oak tanned)	0.25	0.20	0.15	0.33	0.30	0.38	0.40
Leather (chorme tanned)	0.35	0.32	0.22	0.45	0.40	0.48	0.50
Rubber	0.30	0.18	–	0.35	0.32	0.40	0.52
Cotton (Woven) Woolen (Woven)	0.22	0.15	0.12	0.28	0.25	0.27	0.30

Pulleys with leather face or rubber face have considerably increased coefficient of friction as described in Table 9.3.

9.8.1 Maximum Belt Speed

Increasing belt speed will cause the centrifugal force to increase whose tendency will be to remove the belt from pulley contact. Interesting by this will be the situation when the belt will be subjected to tensile stress due to centrifugal tension and bending stress only and the tensile stress due to driving tension will become zero. Rewriting Eq. (9.13),

$$\sigma_{max} = \frac{\sigma_u}{n} = \frac{T_1}{bh} + \frac{T_c}{bh} + \frac{Eh}{D} \qquad \ldots (9.15)$$

where n is the factor of safety applied upon ultimate tensile strength σ_u. If the belt sped has increased to maximum, v_{max}, then $T_1 = 0$ but T_c will increase to maximum. At this point it is required that the stress should remain within permissible limit.

Hence,

$$\frac{T_c}{bh} = \frac{\omega\, v_{max}^2}{gbh} = \frac{\sigma_u}{n} - \frac{Eh}{D}$$

So that

$$v_{max} = \sqrt{\frac{g}{\omega}\left[\frac{\sigma_u}{n} - \frac{Eh}{D}\right] bh} \qquad \ldots (i)$$

From Eq. (9.14),

$$\left[\frac{\sigma_u}{n} - \frac{Eh}{D}\right] = \frac{T_t}{bh} \qquad \ldots (ii)$$

Using (ii) in (i),

$$v_{max} = \sqrt{\frac{gT_t}{\omega}} \qquad \ldots (iii)$$

From Eq. (9.10),

$$v_{max} = \sqrt{3}\, v_{opt}$$

or

$$v_{opt} = \frac{v_{max}}{\sqrt{3}} \qquad \ldots (9.16)$$

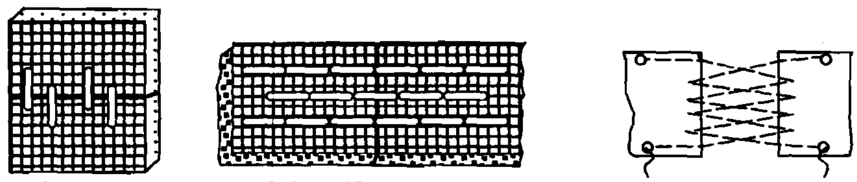
This gives optimum belt speed in terms of maximum belt speed, i.e. the speed at which belt will not transmit any power.

9.9 BELT JOINTS

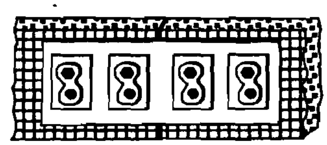
Flat belt could be made endless, that is without any joint and in such cases the strength of the joint is 80 to 100% of the belt material strength. However, the endless belts are not always used because they are available only in standarised length and centre distance of shafts is no more a degree of freedom in design. In addition, the endless belt would require tension adjuster for easy replacement, and overhanging position of pulleys and readily removable bearings. The endless belts are made in factory by cementing the two ends, or they may be cemented at works. The high performance belts have only such joints resulting in efficiency of 98%.

Because of these difficulties, belts with different types of joints are largely used. Various joints of belts are illustrated in Figure 9.3. These joints have efficiencies much below 80%. Further, the joints are more rigid and sometimes heavier than other portions of belt. The joints strike against the pulleys and cause variations in the velocity of motion. In certain cases, like spindles of precision lathes only endless belts should be used. The efficiencies of various belt joints are described in Table 9.4.

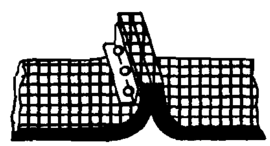
(a) Cemented Joint



(b) Laced Joints with Rawhide



(c) Belt Joint with Bolt and Plate



(d) Crest Joint

Figure 9.3 : Belt Joints

Table 9.4 : Efficiencies of Belt Joints

Type of Joints	Efficiency
Cemented in factory	100%
Cemented in work place	80-90%
Laced wire by machine	75-85%
Laced wire by hide	70-30%
Laced joint by raw hide	50-70%
Hinged by wire hooks	40-45%
Hinged by metal hooks	35-40%
Butt joints, bolt and place covers	30%
Crest but joints, bolts and place covers	25%

Thus, it would be seen that the belt joints have overall effect of reducing the belt power capacity or the strength. A factor of safety is normally applied upon ultimate tensile strength of the belt for several unspecified causes which tend to increase nominal load on the belt. A factor of safety of 10 is a common practice if ultimate tensile strength of belt is chosen from Table 9.1. In a subsequent section, various service factors which modify the load will be described. These factors are used in conjunction with the tabulated values of belt power capacity per unit width.

Example 9.1

A flat belt drive transmits power of 22 kW between pulleys of equal diameters. Each pulley is 400 mm in diameter and rotates at 400 rpm. The pulleys are made in cast iron and will run dry. The belt will have cemented joint which will be made in work place. Select an oak tanned leather belt to have a width of 220 mm. Take factor of safety of 3.

Solution

From Table 9.1 for leather belt note, width available is from 20 to 300 mm, hence 220 mm can be selected. Ultimate tensile strength $\sigma_u = 20$ N/mm^2, Recommended ratio $\frac{D_{min}}{h} = 35$, hence the calculated thickness should not be less than

$\frac{D_{min}}{35} = \frac{200}{35} = 5.7$ mm.

The problem requires calculation of thickness, since width is given. Specific weight $w = 9800$ N/m^3.

Coefficient of friction between oak tanned leather belt and cast iron pulley from Table 9.3 is 0.25 for dry condition.

The efficiency of the joint (cemented at work place), may be taken as 85% from Table 9.4. Modulus of elasticity of leather belt from Table 9.1, $E = 125$ N/mm^2.

The power transmitted,

$$H = (T_1 - T_2)\, v = (T_1 - T_2)\, \omega \frac{D_1}{2} = (T_1 - T_2) \frac{2\pi N_1}{60} . \frac{D_1}{2}$$

Here N_1 and D_1 are rpm and diameter of driving pulley. T_1 and T_2 are the tensions in belt in tight and slack sides such that $T_t = T_1 + T_c$ and $T_s = T_2 + T_c$.

From Eq. (9.6)

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

$\theta = 180^\circ = \pi$ rad and $\mu = 0.25$, hence, $\mu\theta = 0.25\,\pi = 0.7854$

$$\therefore \quad \frac{T_1}{T_2} = 2.2 \quad \text{or} \quad T_1 = 2.2\, T_2 \qquad \ldots \text{(i)}$$

$$\text{Also,} \quad H = 22 \times 10^3 = (T_1 - T_2) \frac{2\pi \times 400}{60} \times \frac{400 \times 10^{-3}}{2} = 8378\,(T_1 - T_2) \times 10^{-3}$$

$$\therefore \quad (T_1 - T_2) = (2.2\, T_2 - T_2) = \frac{22000}{8378} \times 10^3 = 2.63 \times 10^3$$

$$\therefore \quad T_2 = 2188 \text{ N} \quad \text{and} \quad T_1 = 4814 \text{ N} \qquad \ldots \text{(ii)}$$

From Eq. (9.2)

$$T_c = \frac{w v^2}{g}$$

Note w is the weight of 1 m length of belt and v is the belt speed in m/s, $g = 9.81$ m/s^2.

If b is width and h is thickness of the belt section in mm, then volume of 1 m length of belt

$$= b \times 10^{-3} \times h \times 10^{-3} \times 1 \text{ m}^3$$

$$\therefore \quad w = 9800 \times b\,h \times 10^{-6} \text{ N}$$

$$\therefore \quad T_c = \frac{9.8\, b\,h \times 10^{-3}}{9.81} \times \left(\frac{2\pi \times 400}{60} \times \frac{400}{2} \times 10^{-3} \right)^2 = 70.1 \times 10^{-3}\, b\,h \qquad \ldots \text{(iii)}$$

The stress in the belt is caused by tensions T_1 and T_c and due to belt bending over pulley, i.e. bending stress σ_b.

From Eq. (9.12),

$$\sigma_b = \frac{Eh}{D} = \frac{125h}{400} = 0.3125 \text{ N/mm}^2$$

Now to calculate h we set up equation for stress on belt cross-section which should not exceed $\frac{\sigma_u}{f.s.} \times$ Joint eff . Take $f.s. = 3$.

$$\therefore \qquad \sigma_{per} = \frac{20}{3} \times 0.85 = 5.67 \text{ N/mm}^2$$

and stress in belt

$$= \frac{T_1}{bh} + \frac{T_c}{bh} + \sigma_b = \frac{4814}{220h} + 70.12 \times 10^{-3} + 0.3125h \text{ (N/mm)}^2$$

$$\therefore \qquad 5.67 = \frac{21.9 + 70.12 \times 10^{-3}\, h + 0.3125h^2}{h}$$

$$\text{or} \qquad h^2 - 17.92\, h + 70 = 0 \qquad \ldots \text{(iv)}$$

$$\therefore \qquad h = \frac{17.92}{2} \pm \frac{1}{2}\sqrt{321 - 280} = 8.96 \pm 3.2$$

$$h = 12.16 \text{ mm} \quad \text{or} \quad 5.76 \text{ mm}$$

Out of two values which one will you choose? Remember both answers are correct and any value between 5.76 mm and 12.16 mm will be safe. Cost of the thinner belt will be less and hence, it should be chosen. The nearest standard value is 7.5 mm of double ply belt (from Table 9.1). You may like to check if Eq. (iv) is satisfied with $h = 30$ mm, i.e. the left hand side will be less than zero.

$$(7.5)^2 - 17.92\,(7.5) + 70 = 56.25 - 134.4 + 70 = -8.15$$

The other method of check is to see that stress is less than $\sigma_{per} = 5.67$ N/mm^2.

$$\frac{21.9 + 0.07 \times 7.5 + 0.3125\,(7.5)^2}{7.5} = 5.33$$

which is less than 5.67 N/mm^2.

Hence, leather belt of $b = 220$ mm and $h = 7.5$ mm is selected.

SAQ 1

(a) Enumerate methods of power transmission. What are mechanical power transmission systems?

(b) What materials are used to make flat belt?

(c) How do you calculate stresses in a flat belt?

(d) A 150 mm wide belt drives a pulley of 1500 mm diameter at 300 rpm while transmitting 33.5 kW of power. Calculate the thickness of a rubber belt of ultimate tensile strength 37 MPa and modulus of elasticity is 100 MPa. Belt runs dry on C.1 pulley to ensure a coefficient of friction of 0.3. Use a factor of safety of 10 to take care of overload and belt joint. The weight density of belt material 13750 N/m^3. The angle of leap is 165°.

9.10 BELT LENGTH

Belt covers smaller and larger pulleys over respective angles of contact. The larger length of belt covers two sides of centre line of the pulley (Figure 9.4). The figure shows an open belt drive in which C is the centre distance between the pulleys. The centre distance may be calculated on the basis of following equations if other conditions are not important.

$$C = (0.07 \text{ to } 0.1)\, v \qquad \ldots (9.17)$$

and

$$C \geq (1.5 \text{ to } 2.0)\,(D_1 + D_2) \qquad \ldots (9.18)$$

Shorter belt length will be due to shorter centre distance and belt will have to make larger number of trips over the pulleys. That means a shorter belt will undergo more cycles of stress variation during a given period of time than a longer belt with longer centre distance. The latter is preferred to have longer life. A shorter belt will have a shorter life.

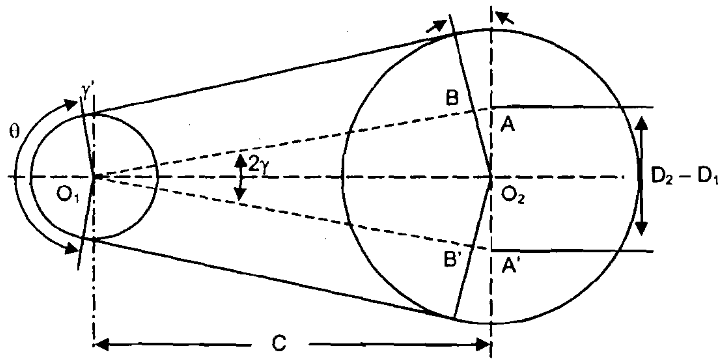


Figure 9.4 : Open Belt Drive

For a given situation we may be required to calculate the belt length for completing the design. O_1 and O_2 are centers of two pulleys. The line O_1BA is parallel to belt on top of pulley. Another line $O_1\,B'A'$ is parallel to belt below the pulleys. The angle of contact, θ, on smaller pulley is $\pi - 2\gamma$ where γ can be seen in Figure 9.4. With O_1 as centre and BB' as arc of a circle of radius C, we can write

$$2\gamma C = D_2 - D_1$$

or

$$2\gamma = \frac{(D_2 - D_1)}{C}$$

or

$$\theta = \left[\pi - \frac{D_2 - D_1}{C}\right] \text{ radians} \qquad \ldots (9.19)$$

The angle of contact on larger pulley is

$$(\pi + 2\gamma) = \left[\pi + \frac{D_2 - D_1}{C}\right]$$

The belt length is the sum of arcs of contact on smaller and larger pulleys and $2O_1B$

$$O_1\,B = O_1\,A - AB$$

$$AB = \frac{D_2 - D_1}{2}\,\gamma = \frac{D_2 - D_1}{2}\,\frac{D_2 - D_1}{2C} = \frac{(D_2 - D_1)^2}{4C}$$

$$O_1\,A = \sqrt{O_1\,O_2^2 + O_2\,A^2} = \sqrt{C^2 + \left(\frac{D_2 - D_1}{2}\right)^2}$$

Arcs of contact

$$= \theta\,\frac{D_1}{2} + (2\pi - \theta)\,\frac{D_2}{2}$$

$$= \frac{\pi}{2}\,(D_2 + D_1) + \frac{(D_2 - D_1)}{2C}$$

∴ Belt length

$$L = 2\sqrt{C^2 + \left(\frac{D_2 - D_1}{2}\right)^2} - 2\,\frac{(D_2 - D_1)^2}{4C}$$

$$+ \frac{\pi}{2}(D_2 + D_1) + \frac{(D_2 - D_1)^2}{2C}$$

$$= \frac{\pi}{2}(D_2 + D_1) + 2\sqrt{C^2 + \left(\frac{D_2 - D_1}{2}\right)^2} \quad \ldots (9.20)$$

9.11 PULLEYS FOR FLAT BELT

In a flat belt drive the belts and pulleys are equally important. Pulleys have to be perfectly aligned and their surfaces properly finished. Standard pulleys are available in stock. A pulley may be a solid disc, if small. A hole in the centre will be provided. Generally a pulley will have three distinct portions. A hub from which radial arms extend to join a rim of considerable width and small thickness. The normal geometry is shown in Figure 9.5. The width is normally 25% larger than the belt width.

Cast iron is the commonest material used for pulleys. The cost is least for C.I pulleys. They may be in solid or split construction. Small size pulley may look like cylinder in which hub joins the outer rim through a web. The C.I pulleys vary from 73 mm in diameter and same width to a 2 m diameter with 600 mm face width. Large pulleys will have high peripheral speeds which is limited by following expression,

$$v_{\max} = 10\sqrt{\frac{\sigma_u}{\rho}} \quad \ldots (9.21)$$

where σ_u is the allowable tensile stress and ρ the density. The maximum speed in cast iron pulleys is limited to 1650 m/min. The split construction is used when pulley is placed between the bearings.

The steel pulleys are built up or fabricated. The rim is made from pressed steel sheets and arms are welded or riveted to the rim. Steel pulleys are lighter and stronger than cast iron pulleys and can run at higher rim speeds. They are also made in solid or split construction. Steel pulleys are often used with special purpose machine when they are required in small numbers.

For small pulleys, particularly on electric motors, compressed paper fibre has found a wider use. These pulleys are also made with a metal centre, frequently cast iron. They are available in sizes : 38 mm in diameter with 50 mm face to 365 mm in diameter with 330 mm face. These pulleys are also suitable when centre distance is small.

Wooden pulleys are lighter – about 2/3 in weight of a comparable cast iron pulley. If wood is defect free and perfectly seasoned, wooden pulleys are capable of taking high loads at high speeds. These pulleys, though rarely used, are usually made with cast iron hub and arms, both in solid and split construction. Most frequent of their applications is with electric motors when they are small in sizes.

9.11.1 Width of Pulley Face and Height of Crown

Generally pulley face width must be 25% greater than the width of the belt. However, pulley face width b_f, can be calculated from the following expression,

$$b_f = [1.1\, b + (10 \text{ to } 15)] \text{ mm} \quad \ldots (9.22)$$

where b is width of the belt in mm.

The surface of the pulley rim is often crowned as shown in Figure 9.5. The crown on the rim forces the belt to return to the centre if, due to slight misalignment the belt has a tendency to run off. There are a few standards for crown which may be obtained either by a linear taper (Figure 9.5) or by curved taper. Some manufacturers use a taper of 1 mm in 200 mm. However, the height of crown, h can be calculated from following :

$$h = 0.092 \sqrt[3]{b_f^2} \text{ mm} \quad \ldots (9.23)$$

if face width of the pulley is in mm.

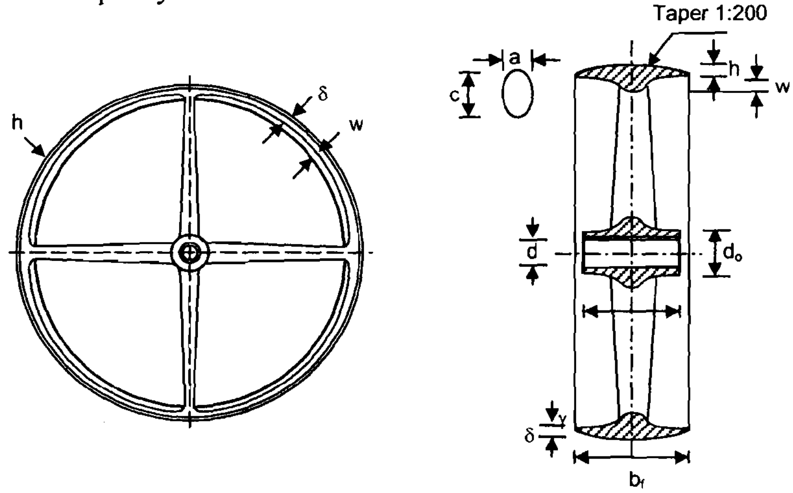


Figure 9.5 : A Pulley with Arms Connecting Hum and Rim

The underside of the cast pulley rim is taper from the centre line toward edges which may be 1 in 25 to 1 in 40. The thickness of the rim at edge is calculated from

$$\delta = (5D + 3) \text{ mm} \quad \ldots (9.24)$$

where D is the pulley diameter in m.

The outside hub diameter at the edge is given by

$$d_0 = (1.7 \text{ to } 2.0)\, d \quad \ldots (9.25)$$

where d is the shaft diameter (Figure 9.5).

The length of the hub is calculated from

$$l = (1.5 \text{ to } 2.0)\, d \quad \ldots (9.26)$$

The hub is tapered from centre to edges with the same taper as on the underside of the rim. The centre of the rim on the underside is ribbed with depth W equal to δ.

9.11.2 Arms of Pulleys

The number of spokes or arms depend upon the diameter of pulley. Following expression is used to calculate number of arms. n_a,

$$n_a = (10 \text{ to } 9)\, \frac{\sqrt{D}}{2} \quad \ldots (9.27)$$

where D is the diameter pulley in metres. Generally at least three arms are provided in a pulley. If the Eq. (9.27) results in $n_a < 4$, the pulleys are often made to have a solid web instead of arms. Many designers follow following rules for number of arms :

Three arms for $D \leq 300$ mm

Four arms for $300 \text{ mm} < D \leq 600$ mm

For $D > 600$ mm, two arms are added for each 600 to 1000 mm increase in diameter of pulley.

The cross-section of arms is calculated on the basis of bending moment caused by a load acting at the end of the arm which is supposed to act as cantilever free at the rim and supported at the hub. It is further assumed that his moment which is equal to the torque of the pulley is spread over one third of total number of arms. The section of the arm is generally elliptic with arm tapering from larger section at the hub to the rim. With major and minor axes, c and a, as shown in Figure 9.5, the section modulus would be

$$Z = \frac{\pi a c^2}{32}$$

and the general practice is to have $c = 2a$, so that

$$Z = 0.05\, c^3$$

As stated above the BM on the arm (the torque on pulley acts as BM or arm) will be $\frac{M_t}{\left(\frac{n_a}{3}\right)}$, where n_a is the number of arms. And thus, bending stress in arm will be

$$\sigma_b = \frac{3M_t}{Z} = \frac{3M_t}{0.05c^3\, n_a}$$

or

$$c = \left(\frac{60M_t}{n_a\, \sigma_b}\right)^{\frac{1}{3}} \quad \text{and} \quad a = \frac{c}{2} \qquad \ldots (9.28)$$

The size of the section at the rim will be a_1 and c_1 such that

$$a_1 = 0.8\, a \quad \text{and} \quad c_1 = 0.8\, c$$

In the absence of sufficient data the size of the pulley may be determined from Sarvin's formula

$$D_1 = 1.114\left(\frac{H}{N_1}\right)^{\frac{1}{3}} \text{ m} \qquad \ldots (9.29)$$

where H is the power in kW and N_1 is the rpm of the pulley.

Sizes of standard pulleys are described in Table 9.5.

Table 9.5 : Flat Belt Pulley Size (mm)

Standard Pulley Diameter	Crown	Tolerance of Concentricity
40	0.3	0.2
50	0.3	0.2
63	0.3	0.2
71	0.3	0.2
80	0.3	0.2
90	0.3	0.2
100	0.3	0.2
112	0.3	0.3
125	0.4	0.3
140	0.4	0.3
160	0.5	0.3
180	0.5	0.4
200	0.6	0.4
224	0.6	0.4
250	0.8	0.4
280	0.8	0.5

9.12 SELECTING BELT FROM MANUFACTURER'S CATALOGUE

What we have discussed was calculations of b or h for a belt considering stress in the cross-section. The tabulated properties are used for such calculation. However, the manufacturers publish data on their belt in terms of power per mm of width against belt speed. Different thicknesses of belt are given since power capacity of belt will increase with thickness. For example, you want to find the width and thickness of a leather belt to transmit 20 kW at a belt sped of 600 m/min. A table from manufacturer gives you power per mm width in kW for thickness of 4.36 mm, 7.14 mm and 11.9 mm, respectively as 0.105 kW and 0.243 kW, 0.147 kW at 610 m/min, which you may accept being closest to 600 m/min. Therefore, the belt widths of

$$\frac{20}{0.105} = 190.5 \text{ mm}, \text{ with } h = 4.36 \text{ mm}$$

$$\frac{20}{0.147} = 130.5 \text{ mm}, \text{ with } h = 7.14 \text{ mm}$$

and $$\frac{20}{0.243} = 82.3 \text{ mm}, \text{ with } h = 11.9 \text{ mm}$$

Which one you would choose will depend upon the pulley in the first place since belt width must be according to pulley width. Secondly the thickness of the belt will depend upon diameter to limit the bending stress. Apparently the belt width has not considered the angle of contact which decides $\frac{T_1}{T_2}$. The selection procedure did not consider, in addition, to above the coefficient of friction, the centre distance, whether the line of centers is horizontal, vertical or inclined, if a higher torque is applied at the start, the material of pulley (which may decide coefficient of friction), whether the load is steady, shock or any other condition and it did not consider under what atmospheric conditions the belt will operate. The leather belt especially may be affected by all or many of these factors. The rubber belt may also be affected by some other factors.

The manufacturers arrive at belt capacities by performing experiments and determining the maximum power transmitting capacity under specified conditions in laboratory. The conditions are clean and controlled atmosphere, both pulleys of same diameter, steady or gradually increasing load, centre line horizontal and may be with different pulley materials and electric motors. And for taking care of various varying factors they define correction factor to be applied on power transmitted to effectively increase it. They may also give angle of contact correction factor and ask us to use joint efficiency. All correction factors are called service factors and their product may be K.

Then, belt width, $$b = \frac{(\text{Power transmitted}) \times K}{(\text{Power per unit width of belt})\ \eta \times K_\theta} \qquad \ldots (9.30)$$

Here η is belt joint efficiency and K_θ is angle of contact correction factor. The service factors for leather belts, power per mm width and correction factor for small pulley diameter (in effect it is angle contact correction) are described in following tables. We will illustrate the use of these tables in a solved example.

For example suppose a belt drive operates under normal conditions with paper pulley on driving motor, the pulley centre line being inclined to horizontal at 60°, the service is continuous, the driven machine is compressor for which slipring motor is used as driver, hence, find service factor. From Table 9.6,

Atmospheric conditions, $K_1 = 1.011$ (normal)

Pulley material, $K_2 = 0.833$ (paper pulley)

Angle of belt center line, $K_3 = 1.001$ (60° inclination with horizontal)

Type of service, $K_4 = 1.25$ (continuous)

Type of load, $K_5 = 1.25$ (compressor)

Type of driving motor, $K_6 = 2.5$ (slipring)

Hence, gross service factor,

$$K = K_1\ K_2\ K_3\ K_4\ K_5\ K_6 = 3.3$$

Table 9.6 : Service Factor for Leather Belts

1.	Atmospheric conditions, K_1	Clean scheduled maintenance on large drives Normal condition Oily wet or dusty conditions	0.883 1.01 1.430
2.	Pulley material, K_2	Paper pulley on motor or small pulley Cast iron or steel pulley	0.833 1.004
3.	Angle of belt centre line, K_3	Pivoted motor drive horizontal to 60° from horizontal 60° to 75° from horizontal 75° to 90° from horizontal	1.001 1.001 1.111 1.251
4.	Type of service, K_4	Temporary or infrequent Normal Important or continuous	0.833 1.000 1.250
5.	Type of load, K_5	Steady belt loads with diesel engine, steam engine, turbine multicylinder gas engine, fan, centrifugal pump, steady line shaft Jerky loads with single cylinder gas engine, reciprocating machines, developing peak loads such as compressors, rock crushers and punch presser Shock and reverse belt loads such as printing press, elevator and laundry washers	1.000 1.250 1.670
6.	Type of driving motors, K_6	Squirrel cage, compensator start Squirrel cage, line start Slip ring and high starting torque	1.500 2.000 2.500

Continuing consideration of the same problem for which we calculated service factor, K, assume that the leather belt is transmitting 20 kW of power, from paper pulley of 150 mm diameter running at 950 mm. The driven pulley is 300 mm and made in cast iron. Find the belt width with service factor found earlier. Use Tables 9.7 and 9.8.

$$\text{Belt speed} = \frac{D_1}{2}\,\omega = \frac{150}{2} \times 10^{-3}\ \frac{2\pi\, 950}{60} = 7.46 \text{ m/s} = 447.7 \text{ m/min}$$

In Table 9.7 the speed closest to 447.7 is 425 m/min. Since, $\frac{D}{h}$ is recommended as

$35\ b \leq 25$ from Table 9.1, since $\frac{D}{h} = 21$ for $h = 7.14$ mm, only $h = 4.36$ or 5.16 mm are permissible.

Thus, for speed of belt of 425 m/min and thickness $h = 4.36$ mm and $h = 5.16$ mm power/width, b from Table 9.7 is 0.075 and 0.087 kW, respectively.

$$\therefore \qquad b_1 = \frac{20 \times 3.3}{0.075\ \eta\ K_\theta} = \frac{880}{\eta\ K_\theta} \qquad \text{for } h = 4.36 \text{ mm}$$

$$\text{or} \qquad b_2 = \frac{20 \times 3.3}{0.087\ \eta\ K_\theta} = \frac{758.6}{\eta\ K_\theta} \qquad \text{for } h = 5.16 \text{ mm}$$

And assuming centre distance 7 m and tight side below the factor $K_0 = 0.78$ for small pulley diameter of 150 mm. Assume joint is so chosen that $\eta = 0.85$, then

$$b_1 = \frac{880}{0.85 \times 0.78} = 1.33 \text{ m} \qquad \text{for } h = 4.36 \text{ mm}$$

and
$$b_2 = \frac{758.6}{0.85 \times 0.78} = 1.14 \text{ m} \qquad \text{for } h = 5.16 \text{ mm}$$

Table 9.7 : Power per mm of Width for Oak Tanned Leather Belt kW

Belt Speed (m/min)	Single Ply		Double Ply			Triple Ply	
	4.36 mm	5.16 mm	7.14 mm	7.94 mm	9.13 mm	11.9 mm	13.5 mm
183	0.03	0.036	0.045	0.054	0.066	0.075	0.084
244	0.042	0.051	0.060	0.072	0.087	0.09	0.108
305	0.054	0.063	0.078	0.093	0.108	0.123	0.135
366	0.063	0.075	0.093	0.111	0.120	0.147	0.162
425	0.075	0.087	0.105	0.129	0.147	0.171	0.189
488	0.084	0.09	0.120	0.147	0.168	0.195	0.213
549	0.096	0.111	0.135	0.162	0.186	0.219	0.240
610	0.105	0.123	0.147	0.180	0.207	0.243	0.267
671	0.117	0.135	0.162	0.198	0.228	0.264	0.291
732	0.126	0.147	0.177	0.213	0.186	0.285	0.313
792	0.135	0.159	0.189	0.231	0.267	0.309	0.342
853	0.147	0.168	0.204	0.246	0.285	0.330	0.363
915	0.156	0.177	0.210	0.261	0.300	0.348	0.384
975	0.162	0.189	0.228	0.276	0.318	0.369	0.405
1036	0.171	0.198	0.237	0.291	0.36	0.387	0.426
1097	0.177	0.207	0.249	0.303	0.351	0.402	0.444
1158	0.186	0.213	0.261	0.315	0.366	0.420	0.462
1220	0.192	0.222	0.270	0.327	0.378	0.435	0.480
1280	0.201	0.231	0.279	0.339	0.390	0.450	0.495
1340	0.207	0.237	0.288	0.351	0.402	0.462	0.507
1400	0.213	0.243	0.294	0.360	0.414	0.474	0.522
1460	0.216	0.249	0.303	0.369	0.423	0.486	0.434
1524	0.222	0.252	0.309	0.375	0.429	0.495	0.546
1585	0.225	0.258	0.315	0.384	0.438	0.504	0.555
1645	0.228	0.261	0.318	0.387	0.444	0.513	0.564
1705	0.231	0.264	0.324	0.393	0.450	0.519	0.576
1770	0.231	0.267	0.327	0.396	0.453	0.525	0.576

Table 9.8 : Correction Factor for Small Pulley Diameter for Oak Tanned Leather Belt (K_θ)

Small Pulley Diameter (mm)	Centre Distance (mm)							
	Upto 3048 mm		Upto 4572 mm		Upto 6096 mm		Upto 6096 mm	
	Tight Side		Tight Side		Tight Side		Tight Side	
	Above	Below	Above	Below	Above	Below	Above	Below
50	0.37	0.37	0.38	0.41	0.37	0.43	0.37	0.44
62.5	0.41	0.41	0.43	0.46	0.41	0.48	0.42	0.49
75	0.45	0.45	0.48	0.52	0.48	0.54	0.48	0.54
87.5	0.49	0.49	0.53	0.57	0.53	0.59	0.53	0.60
100	0.53	0.53	0.58	0.63	0.59	0.65	0.59	0.66
112.5	0.56	0.56	0.61	0.66	0.62	0.68	0.62	0.70
125	0.59	0.59	0.65	0.70	0.66	0.72	0.66	0.74
137.5	0.60	0.60	0.66	0.72	0.67	0.74	0.68	0.76
150	0.62	0.62	0.68	0.74	0.69	0.76	0.70	0.78
162.5	0.64	0.64	0.70	0.76	0.71	0.78	0.72	0.80
200	0.66	0.66	0.72	0.78	0.73	0.80	0.74	0.82
225	0.67	0.67	0.73	0.79	0.74	0.81	0.75	0.83
250	0.68	0.68	0.75	0.81	0.76	0.83	0.77	0.85
275	0.69	0.69	0.76	0.82	0.77	0.84	0.78	0.86
300	0.70	0.70	0.77	0.83	0.78	0.86	0.79	0.88
325	0.71	0.71	0.78	0.84	0.79	0.87	0.80	0.89
350	0.72	0.72	0.79	0.85	0.80	0.88	0.81	0.90
375	0.73	0.73	0.80	0.86	0.81	0.89	0.82	0.91
400	0.74	0.74	0.80	0.87	0.81	0.89	0.82	0.91
425	0.74	0.74	0.81	0.88	0.82	0.90	0.83	0.921
450	0.75	0.75	0.82	0.89	0.83	0.91	0.84	0.93
500	0.75	0.75	0.83	0.90	0.84	0.92	0.85	0.94
550	0.76	0.76	0.84	0.91	0.85	0.93	0.86	0.95
600	0.77	0.77	0.85	0.92	0.86	0.94	0.87	0.96
750	0.79	0.79	0.87	0.94	0.88	0.96	0.88	0.98
900	0.80	0.80	0.88	0.95	0.89	0.98	0.90	1.00

Service factors as described in Table 9.6 are not used for rubber belt but angle of contact (arc of contact) correction factor and efficiency of belt joint are used. Tables 9.9 and 9.10 are described for Goodyear Wing foot rubber belt.

Table 9.9 : Power Rating per mm Width at 180° Arc of Contact for Goodyear Wing Foot Belt (kW/mm)*

Ply	Small Pulley Dia. (mm)	Belt Speed in m/min																
		305	365	425	488	550	610	670	730	796	850	915	1065	1220	1370	1525	1675	1830
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)
3	76	0.18	0.21	0.24	0.27	0.27	0.30	0.33	0.36	0.36	0.39	0.39	0.42	0.45	0.45	0.45	0.42	0.33
	107	0.24	0.27	0.30	0.33	0.36	0.42	0.45	0.48	0.51	0.54	0.54	0.60	0.63	0.66	0.66	0.66	0.60
	127	0.27	0.33	0.39	0.42	0.45	0.51	0.54	0.57	0.63	0.66	0.69	0.75	0.81	0.87	0.90	1.99	0.87
	152	0.33	0.39	0.45	0.51	0.54	0.60	0.66	0.69	0.75	0.78	0.84	0.93	0.99	1.05	1.11	1.14	1.11
	203	0.42	0.48	0.54	0.60	0.66	0.72	0.78	0.84	0.90	1.96	1.02	1.14	1.23	1.32	1.41	1.44	1.47
	254	0.41	0.48	0.57	0.63	0.72	0.81	0.87	0.96	1.02	1.08	1.11	1.26	1.35	1.47	1.56	1.62	1.65
	305 and up	0.42	0.48	0.57	0.63	0.72	0.81	0.87	0.96	1.02	1.08	1.17	1.29	1.41	1.53	1.62	1.68	1.74
4	76	0.21	0.24	0.27	0.30	0.33	0.36	0.36	0.39	0.42	0.42	0.45	0.48	0.48	0.45	0.42	0.36	0.27
	107	0.27	0.30	0.33	0.39	0.42	0.45	0.48	0.51	0.54	0.57	0.60	0.03	0.66	0.66	0.66	0.60	0.54
	127	0.30	0.36	0.42	0.45	0.51	0.57	0.60	0.63	0.72	0.72	0.75	0.81	0.84	0.90	0.90	0.87	0.81
	152	0.36	0.42	0.48	0.54	0.60	0.66	0.69	0.75	0.84	0.84	0.87	0.96	1.02	1.08	1.11	1.11	1.17
	203	0.45	0.54	0.60	0.69	0.75	0.84	0.90	0.96	1.08	1.08	1.14	1.26	1.35	1.44	1.50	1.53	1.53
	254	0.54	0.63	0.72	0.81	0.87	0.96	1.02	1.11	1.23	1.23	1.32	1.44	1.56	1.68	1.77	1.83	1.86
	305	0.54	0.66	0.75	0.87	0.96	1.05	1.14	1.20	1.35	1.35	1.44	1.59	1.74	1.89	1.98	2.04	2.10
	365-406	0.54	0.66	0.78	0.90	0.99	1.08	1.17	1.26	1.41	1.41	1.59	1.68	1.86	1.98	2.10	2.19	2.25
	457 and up	0.54	0.66	0.78	0.90	0.99	1.08	1.20	1.29	1.44	1.44	1.56	1.74	1.89	2.04	2.16	2.25	2.31
5	152	0.33	0.39	0.45	0.51	0.54	0.60	0.63	0.66	0.69	0.75	0.78	0.84	0.87	0.90	0.90	0.87	0.78
	203	0.48	0.54	0.63	0.69	0.78	0.84	0.90	0.96	1.02	1.08	1.14	1.23	1.32	1.38	1.44	1.44	1.41
	254	0.57	0.66	0.75	0.84	0.93	0.02	1.08	1.14	1.23	1.29	1.38	1.50	1.65	1.74	1.83	1.86	1.89
	305	0.63	0.72	0.84	0.93	1.05	1.17	1.23	1.32	1.41	1.50	1.59	1.77	1.92	2.07	2.16	2.22	2.25
	356	0.66	0.78	0.90	1.02	1.14	1 26	1.35	1.44	1.56	1.65	1.74	1.92	2.10	2.25	2.22	2.46	2.52
	406-508	0.66	0.81	0.93	1.05	1.20	1.32	1.41	1.53	1.65	1.74	1.86	2.07	2.25	2.43	2.58	2.67	2.73
	558 and up	0.66	0.81	0.93	1.08	1.20	1.32	1.44	1.56	1.68	1.80	1.95	2.19	2.40	2.58	2.70	2.79	2.88
6	203	0.42	0.48	0.54	0.63	0.69	0.75	0.81	0.87	0.93	0.96	1.02	1.11	1.17	1.20	1.20	1.17	1.05
	245	0.57	0.66	0.75	0.84	0.93	1.02	1.11	1.17	1.26	1.32	1.38	1.53	1.62	1.71	1.74	1.74	1.71
	305	0.69	0.81	0.90	1.02	1.11	1.23	1.03	1.41	1.47	1.56	1.65	1.83	1.98	2.10	2.19	2.25	2.25
	356	0.75	0.87	0.99	1.11	1.23	1.35	1.44	1.56	1.65	1.74	1.86	2.04	2.22	2.40	2.52	2.58	2.61
	406	0.81	0.93	1.08	1.20	1.32	1.47	1.56	1.68	1.80	1.92	2.01	2.22	2.43	2.58	2.73	2.83	2.88
	457	0.81	0.96	1.11	1.26	1.58	1.53	1.62	1.77	1.89	2.01	2.10	2.34	2.55	2.76	2.91	3.00	3.06
	503-510	0.81	0.96	1.14	1.29	1.44	1.59	1.71	1.86	1.98	2.10	2.25	2.49	2.73	2.94	3.12	3.24	3.30
	708 and up	0.81	0.96	1.14	1.29	1.44	1.59	1.74	1.89	2.04	2.16	2.31	2.58	2.82	3.06	3.24	3.36	3.45

* The values given in above table are to be divided by 10.

Table 9.10 : Arc of Contact Correction Factors for Two Pulley Open Drives

Diff. of Pulley Dia. (mm)	Centres Distance in m																
	0.61	1.22	1.83	2.45	3.06	3.66	4.27	4.88	5.50	6.10	6.70	7.60	9.15	10.2	12.2	13.7	15.2
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)
51	0.98	0.9	0.99	0.99	1.00	1.00	1.00	1.00	1.00	1.10	1.00	1.00	1.00	1.00	1.00	1.00	1.00
102	0.96	0.98	0.98	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	1.00	1.00	1.00	1.00	1.00	1.00
152	0.94	0.97	0.98	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	1.00	1.00	1.00
203	0.92	0.96	0.97	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	1.00
254	0.90	0.95	0.97	0.97	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99
305	0.88	0.94	0.96	0.97	0.99	0.99	0.99	0.9	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99	0.99
356	0.85	0.93	0.95	0.96	0.97	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
406	0.83	0.92	0.95	0.96	0.97	0.97	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
457	0.81	0.91	0.94	0.95	0.96	0.97	0.97	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
508	0.79	0.90	0.93	0.95	0.96	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
559	0.76	0.89	0.93	0.94	0.96	0.96	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
610	0.74	0.88	0.92	0.94	0.95	0.96	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
660	0.71	0.87	0.91	0.93	0.95	0.96	0.96	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98	0.98	0.98
711	0.69	0.86	0.90	0.93	0.94	0.95	0.96	0.97	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98	0.98
762	0.66	0.85	0.90	0.92	0.94	0.95	0.96	0.96	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98	0.98
813	0.63	0.84	0.89	0.92	0.93	0.95	0.95	0.96	0.97	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98
864	0.58	0.83	0.88	0.91	0.93	0.94	0.95	0.96	0.96	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98
984	0.54	0.82	0.88	0.91	0.93	0.94	0.95	0.95	0.96	0.97	0.97	0.97	0.98	0.98	0.98	0.98	0.98
1067	–	0.81	0.85	0.89	0.91	0.93	0.94	0.95	0.95	0.96	0.97	0.97	0.97	0.98	0.98	0.98	0.98
1219	–	0.78	0.83	0.88	0.90	0.92	0.93	0.94	0.95	0.95	0.96	0.97	0.97	0.97	0.97	0.98	0.98
1372	–	0.74	0.81	0.88	0.89	0.91	0.92	0.93	0.94	0.94	0.95	0.96	0.96	0.97	0.97	0.98	0.98
1524	–	0.70	0.79	0.85	0.88	0.90	0.91	0.92	0.93	0.94	0.94	0.95	0.96	0.97	0.97	0.97	0.98
1674	–	0.66	0.76	0.83	0.86	0.88	0.90	0.90	0.92	0.93	0.94	0.95	0.96	0.97	0.97	0.97	0.97
1828	–	0.61	0.74	0.81	0.85	0.88	0.89	0.91	0.92	0.93	0.93	0.94	0.95	0.96	0.96	0.97	0.97
1981	–	0.54	0.71	0.79	0.84	0.87	0.89	0.90	0.91	0.92	0.93	0.94	0.95	0.95	0.96	0.97	0.97
2134	–	–	0.69	0.78	0.82	0.85	0.88	0.89	0.90	0.92	0.92	0.93	0.94	0.95	0.95	0.97	0.97
2286	–	–	0.6	0.76	0.81	0.84	0.87	0.88	0.90	0.91	0.92	0.93	0.94	0.95	0.95	0.96	0.97
2438	–	–	0.63	0.74	0.80	0.83	0.86	0.88	0.89	0.90	0.91	0.92	0.93	0.94	0.95	0.96	0.96
2591	–	–	0.58	0.72	0.78	0.82	0.85	0.87	0.88	0.90	0.90	0.92	0.93	0.94	0.95	0.95	0.96
2743	–	–	0.54	0.70	0.77	0.81	0.84	0.86	0.88	0.89	0.90	0.91	0.93	0.94	0.94	0.95	0.96
2896	–	–	–	0.68	0.75	0.80	0.83	0.85	0.87	0.88	0.89	0.91	0.92	0.93	0.94	0.95	0.95
3048	–	–	–	0.66	0.74	0.79	0.82	0.84	0.86	0.88	0.89	0.90	0.92	0.93	0.94	0.95	0.95
3200	–	–	–	0.64	0.73	0.78	0.81	0.83	0.85	0.87	0.88	0.80	0.91	0.93	0.94	0.94	0.95

9.13 BELT ELASTIC CREEP

An interesting phenomenon occurs in belt running over pulleys. It is not difficult to appreciate that since $T_1 > T_2$, the strain in tight side of pulley is greater than that part of belt which is slack, i.e. subjected to tension T_2. In driving pulley with tight side below, the tight side enters the pulley and leaves that pulley under lower tension or lower strain. If we consider a unit length under no tension, it will be changed to $(1 + \varepsilon_1)$ under T_1 and to $(1 + \varepsilon_2)$ under T_2 which means a length of $(1 + \varepsilon_1)$ will be received by driving pulley and a length of $(1 + \varepsilon_2)$ will be released where $\varepsilon_1 > \varepsilon_2$. Similarly, at driven pulley a length of $(1 + \varepsilon_2)$ will be received and $(1 + \varepsilon_1)$ will be released. But the mass of belt passing at an cross-section must remain constant, i.e. $\frac{v}{1 + \varepsilon}$ should be constant and hence,

$\frac{v_1}{1 + \varepsilon_1} = \frac{v_2}{1 + \varepsilon_2}$ where v_1 is the velocity of belt in tight side and v_2 is the velocity in slack side. The above equality requires that $v_1 > v_2$.

$$\therefore \qquad \frac{v_1}{v_2} - 1 = \frac{1 + \varepsilon_1}{1 + \varepsilon_2} - 1$$

or

$$\frac{v_1 - v_2}{v_2} = \frac{\varepsilon_1 - \varepsilon_2}{1 + \varepsilon_2} = \varepsilon_1 - \varepsilon_2 \qquad \ldots (9.31)$$

since $\varepsilon_2 << 1$.

Delivering of lesser length from driving pulley will require that some small length of belt creeps back on pulley. Similarly, delivering larger length of belt from driven pulley will require the belt to creep forward. The term creep is used to understand that belt does not slide backward or forward over entire are of contact for adjusting the length but only over a small fraction of arc of contact near the end of the contact and such adjustment does not comprise any sliding but a series of length contraction and elongation.

The ratio $\frac{v_1 - v_2}{v_2}$ or difference $\varepsilon_1 - \varepsilon_2$ is called coefficient of creep. If D_1 and D_2 are the diameters of driving and driven pulleys the velocity v_1 will coincide with peripheral velocity of driving pulley and v_2 will coincide with peripheral velocity of driven pulley.

$$v_1 = \frac{\pi N_1 D_1}{60}, \quad v_2 = \frac{\pi N_2 D_2}{60}$$

The ratio $\frac{N_1}{N_2}$ is often referred to as velocity ratio and denoted by i. From above

$$i = \frac{N_1}{N_2} = \frac{v_1}{v_2} \cdot \frac{D_2}{D_1}$$

or

$$\frac{v_1}{v_2} = 1 + S$$

$$\therefore \qquad i = \frac{N_1}{N_2} = (1 + S)\,\frac{D_2}{D_1} \qquad \ldots (9.32)$$

The meaning of Eq. (9.32) should be understood very carefully. If $S = 0$, i.e. when we assume that creep does not occur (which is never true) then $\frac{N_1}{N_2} = \frac{D_2}{D_1}$. If diameter D_2 is found this way then the actual velocity ratio will be different from that required. The creep in actual case may not be greater than 0.02 or 2%.

Suppose that we require a velocity ratio of 3 and creep is not considered then what will be the diameter D_2 for $D_1 = 200$ mm.

$$i = 3 = \frac{D_2}{D_1}, \quad D_2 = 200 \times 3 = 600 \text{ mm}$$

If 2% creep is considered then

$$D_2 = \frac{200 \times 3}{1.02} = 588.2 \text{ mm}$$

The actual diameter will be less.

Similarly, if the diameter D_2 is calculated by neglecting S, then

$$i = \frac{600}{200}\,(1.02) = 3.06$$

In which case if driving pulley is running at 300 rpm, the driven pulley will run at $\frac{300}{3.06} = 98.04$ rpm.

For convenience of calculation we denote various quantities by D_1, D_2, N_1 and N_2 when no consideration of creep is done. With creep under consideration we denote them by D_1, D'_2, N_1 and N'_2.

Then $$D_2 = D_1 \frac{N_1}{N_2} \quad \text{and} \quad N_2 = N_1 \frac{D_1}{D_2}$$

$$D'_2 = \frac{N_1 D_1}{1 + S} \cdot \frac{1}{N'_2} \quad \text{and} \quad N'_2 = \frac{N_1 D_1}{1 + S}\,\frac{1}{D'_2} \qquad \ldots (9.33)$$

It is again interesting to note that $S = \varepsilon_1 - \varepsilon_2 = \frac{T_1}{E} - \frac{T_2}{E}$ and hence if difference between T_1 and T_2 is large coefficient of creep will be large.

Example 9.2

A leather belt is used to supply power from a 5.5 kW, 1440 rpm compensator start electric motor to a reciprocating air compressor. The centre line between the pulleys is inclined to horizontal at 60°. The driving pulley on motor shaft is made of compressed paper and is 400 mm in diameter while the driven pulley 800 mm in diameter is made in C.I. The belt is to be cemented at the site to join the two ends and normal atmospheric and service conditions are maintained. Find what width of a single ply oak tanned belt will be necessary. Calculate the factor of safety if ultimate tensile strength of leather is 20 MPa and modulus of elasticity is 125 MPa. The weight density of oak tanned leather belt is 9800 N/m^3. Calculate dimensions of C.I pulley. Find actual speed of driven pulley if $S = 2\%$.

Solution

We will use tables to select belt.

The minimum centre distance, Eq. (9.18)

$$C = 1.5\,(D_1 + D_2) = 1.5\,(400 + 800) = 1800 \text{ mm} \qquad \ldots (i)$$

The belt velocity

$$= \frac{\pi N_1 D_1}{60} = \frac{\pi \times 1440 \times 400 \times 10^{-3}}{60} = 30.2 \text{ m/s or } 1810 \text{ m/min} \qquad \ldots (ii)$$

For tight side below, small pulley diameter of 400 mm and centre distance of 1800 mm < 3048 mm, from Table 9.9

$$K_\theta = 0.74 \qquad \ldots (iii)$$

Service Factor

Table 9.6,

Normal atmospheric conditions	$K_1 = 1.011$
Paper pulley on motor	$K_2 = 0.833$
Centre line inclined at 60° to horizontal	$K_3 = 1.111$
Normal service	$K_4 = 1.000$
Compressor load	$K_5 = 1.250$
Compensator start motor	$K_6 = 1.500$

$\therefore$ Service factor, $K = 1.011 \times 0.833 \times 1.111 \times 1.000 \times 1.25 \times 1.50 = 1.754$

Table 9.4, efficiency of belt joint made at work place, $\eta = 0.85$.

Belt Section

Table 9.7,

The belt velocity of 1800 m/min is close to 1770 m/min and hence at that velocity the single ply belt can transmit 0.231 kW with $h = 4.36$ mm or 0.267 kW for $h = 5.16$ mm.

Choosing higher thickness $\dfrac{D}{h} = \dfrac{400}{5.16} = 77.5$, permissible

$$\therefore \quad b = \frac{(\text{kW transmitted})\, K}{(\text{kW/mm width})\, \eta\, K_\theta} \qquad \text{Eq. (9.30)}$$

$$= \frac{5.5 \times 1.754}{0.267 \times 0.85 \times 0.74} = 57.44 \text{ say } 60 \text{ mm} \qquad \ldots \text{(iv)}$$

and $\quad h = 5.16$ mm $\qquad \ldots$ (v)

Factor of Safety

The service factors η and K_θ tend to increase actual power or

$$\text{Power} = \frac{K\,H}{0.85 \times K_\theta} = \frac{1.754 \times 5.5}{0.85 \times 0.75} = 15.05 \text{ kW}$$

$$\therefore \quad 15.05 \times 10^3 = (T_1 - T_2)\, v = (T_1 - T_2)\, 30.2$$

$$\therefore \quad T_1 - T_2 = 498 \text{ N} \qquad \ldots \text{(vi)}$$

$$\text{Angle of contact on smaller pulley} = \pi - \frac{D_2 - D_1}{C} = \pi - \frac{800 - 400}{1800}$$

or $\quad \theta = \pi - 0.222 = 2.92$ rad

Table 9.3, coefficient of friction between oak tanned belt and compressed paper pulley,

$$\mu = 0.33$$

$$\therefore \quad \frac{T_1}{T_2} = e^{\mu\theta} = e^{0.33 \times 2.92} = 2.64$$

$\therefore$ From (vi), $T_2\,(2.64 - 1) = 498$

or $\quad T_2 = 303.7$ N, $T_1 = 801.7$ N $\qquad \ldots$ (vii)

For $b = 60$ mm, $h = 5.16$ mm,

Weight per m, $w = 9800 \times 60 \times 5.16 \times 10^{-6} = 3.03$ N/m $\qquad \ldots$ (viii)

$$\therefore \quad T_c = \frac{wv^2}{g} = \frac{3.09 \times (30.2)^2}{9.81} = 281.7 \text{ N} \qquad \ldots \text{(ix)}$$

$\therefore$ Tensile stress in belt,

$$\sigma = \frac{T_1 + T_c}{bh} + \frac{Eh}{D_1}$$

or $$\sigma = \frac{801.7 + 281.7}{60 \times 5.16} + \frac{125 \times 5.16}{400}$$

$$= 2.6 + 1.6125 = 4.2125 \text{ N/mm}^2$$

$$\therefore \text{ Factor of safety } = \frac{\sigma_u}{\sigma} = \frac{20}{4.2125} = 4.75 \qquad \ldots \text{(x)}$$

Pulley, Cast Iron

Diameter $D_2 = 800$ mm

Eq. (9.22), $b_f = 1.1\,b + 10 \text{ mm} = 1.1 \times 60 + 10 = 76 \text{ mm}$

Eq. (9.23), height of crown

$$= 0.092\,(b_f)^{\frac{1}{3}} = 0.092\,(76)^{\frac{1}{3}} = 0.4 \text{ mm}$$

Eq. (9.24) $\delta = (5D_2 + 3) \text{ mm} = 5 \times 0.8 + 3 = 7 \text{ mm}$

Eq. (9.3) with $\tau = 78$ N/mm^2, $d = 36.5 \left(\dfrac{H}{\tau N}\right)^{0.33}$

$$d = 36.5 \left(\frac{15.05 \times 1000}{78 \times \dfrac{1440}{2}} \right)^{0.33}$$

or $d = 23.5$ mm say 29 mm (increased by 25%)

Eq. (9.25) $d_o = 2.0\,d = 58$ mm

Eq. (9.26) $l = 2.0\,d = 58$ mm

Eq. (9.27) $n_a = 10 \dfrac{\sqrt{D}}{2} = 10 \dfrac{\sqrt{0.8}}{2} = 4.47$ say 5.

$$\therefore \quad H = M_t\,\omega = M_t \frac{2\pi \times 1440}{60 \times 2} = 75.4\,M_t$$

$$M_t = \frac{15.05 \times 1000}{75.4} = 199.6 \text{ Nm} = 2 \times 10^5 \text{ Nmm}$$

Eq. (9.28) and $\sigma_b = 30$ N/mm^2

$$c = \left(\frac{60\,M_t}{\sigma_b}\right)^{\frac{1}{3}} = \left(\frac{60 \times 2 \times 10^5}{5 \times 30}\right)^{\frac{1}{3}} = 43 \text{ mm}$$

$a = 21.5$ mm

$a_1 = 17.23$ mm, $C_1 = 34.4$ mm

(Sketch pulley)

Eq. (9.33) $N_2' = \dfrac{N_1 D_1}{1 + S} \dfrac{1}{D_2} = \dfrac{1440 \times 400}{1.02} \times \dfrac{1}{800} = 705.9$ rpm

SAQ 2

(a) The pulleys of diameter 200 mm and 800 mm are mounted on parallel shafts separated by a distance of 2 m. Find angle of contact on smaller pulley and on larger pulley. What is the minimum distance by which these pulleys can be separated? In which case the tension in belt will be higher?

(b) Sketch a pulley for flat belt and show various dimensions. How do you calculate the arms of the pulley?

(c) Mention factors that will affect power to be transmitted by leather belt. How are the effects considered in selection of the belt?

(d) A prime mover running at 300 rpm drives a DC generator at 500 rpm by a belt drive. Diameter of the prime mover pulley is 600 mm. Assuming a creep coefficient of 3% determine the diameter of the generator pulley if belt is 6 mm thick.

(e) A winch driven by a flat belt drive is to have a velocity ratio of 4. The driving motor is 8 kW capacity and runs at 900 rpm. The load variation is considerable. Belt centre line is to be horizontal. Calculate belt width of a rubber belt. The belt carries a hook joint (wire hooks).

9.14 V-BELT DRIVES

Whereas flat belts, normally wide, run on wide faced pulleys with large centre distance, the V-belt, in reality, have trapezoidal section and run in grooved pulleys, which are better known as sheaves. The wedge section of a V-belt makes contact on two inclined faces and thus generates high frictional force. The V-belts are made endless and are available in fixed pitch lengths. The tensions T_1 and T_2 on two sides of a power transmitting belt are related in the same way as for a flat belt, i.e. $\frac{T_1}{T_2} = e^{\mu\theta}$. The difference is that μ in this case is enhanced because of inclined faces of contact in the same way as coefficient of friction in case of Acme threads is enhanced in comparison with the coefficient of friction in case of square threads. The groove angle and angle between two sides of wedge section of belt is same and is normally 37°. If coefficient of friction between belt and pulley, as a property, is μ and the groove angle is 2α, then enhanced or effective coefficient of friction, μ_{eff} is

$$\mu_{eff} = \frac{\mu}{\sin \alpha} = \frac{\mu}{\sin 18.5} = 3\mu \qquad \ldots (9.34)$$

Thus, the coefficient of friction in V-belt is quite high, which effectively increases $\frac{T_1}{T_2}$.

In most cases $T_2 << T_1$. Groove angle which is still smaller will further increase μ_{eff} but a groove angle less than 20° is never used because the belt will be locked in the groove. Even for a groove angle of 20°, effective coefficient of friction will be 5.8 μ. The higher coefficient of friction results in higher pressure between the belt and groove surface, which will generate faster wear of belt. The worn surface will have reduced coefficient of friction. Therefore, the main advantage of the V-belts, viz., higher coefficient of friction, is not used to full extent. In V-belts the ratio of pressure intensity to the tensile stress is 4.5 to 5 times that of same ratio in flat belts. If the velocity reduction from driving to driven pulley is 4 : 1 or larger then the driven pulley may be flat. The absence of grooves on larger driven pulley reduces cost considerably.

The advantages of V-belt drives may be summed up below :

(a) Compact layout. The minimum centre distance is $\frac{1}{2}(D_1 + D_2) + 3h$, where h is the height of belt section. A V-belt drive is accommodated in a machine.

(b) High velocity ratios upto 10 are permissible.

(c) Noiseless operation.

(d) The drive can take up shocks, especially at starting.

(e) The reversal of direction whereby the tight side can be on top does not change power.

(f) The angle of inclination of centre line does not affect power.

(g) The V-belt is not affected by heat, moisture and dust.

(h) Though power capacity of one belt is not very high, several of them run on the same set of pulleys and increase power of drive. Even the failure of one belt in a multiple drive does not incapacitate the drive and the remaining belts can take the load, being overloaded, till such time that drive can be stopped conveniently and failed belt replaced.

(i) The drives can be made to have variable speed drives.

However, the centre distance variation upto 10% is to be incorporated in the drive to facilitate putting the belts on sheave. Such adjustability is shown in Figure 9.6. The efficiency of V-belt drive is slightly lower than flat belt drive and V-belt creeps more because of higher difference between T_1 and T_2. V-belt drive is cheaper, even 5% cheaper than flat belt drive.

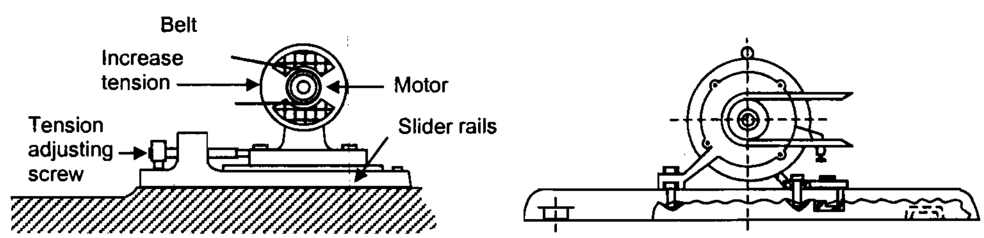


Figure 9.6 : Adjustable Motor Base to Help Adjust the Center Distance of the Belt Drive

The V-belt drive has a few distinct drawbacks as compared to flat belt drive. These are stated as under :

(a) V-belts have lower life than flat belts mainly because of high bending stress that is caused due to higher ratio of belt depth to diameter of pulley than that in case of flat belt drive and also because of high frequency of stress variation.

(b) The design of pulleys for V-belt drives is more complicated than for flat belt drives.

(c) V-belt drives cannot be used for larger power.

(d) The centre distance in case of V-belts may not be large.

The V-belt drives are used quite widely and within their power range (i.e. upto 1100 kW), employment of V-belt drives is second only to gear drives.

9.15 STRUCTURE AND TYPES OF V-BELTS

V-belts are trapezoidal in cross-section and run in grooved pulleys, grooves being V-shaped or having two inclined sides with flat bottoms. A properly installed belt should fit tightly against the sides of the pulley grooves without projecting beyond the rim or touching the bottom of the groove. The correct method of mounting a wedge-shaped belt in grooved pulley is illustrated in Figure 9.7. Such belts as shown in Figures 9.8(a), (b) and (c) are of classical type. The ultraflex narrow section belts have now become popular and their cross-section is shown in Figure 9.8(d) and Figure 9.8(a) shows the structure of a belt. The belt section consists of several layers of rubberised fabric

(a) which carry the tensile load, the rubberised cords,

(b) which also carry the tension and rubber, and

(c) which takes compression during bending of belts. A rubberised fabric coating, and

(d) covers the above material and mainly serves to protect them from mechanical and other damages.

Figure 9.6(b) shows another V-belt section which consists of cords placed along the neutral plane to take up the tensile load only. These cords are surrounded by rubber and this combination is then surrounded by protective rubberised fabric cover.

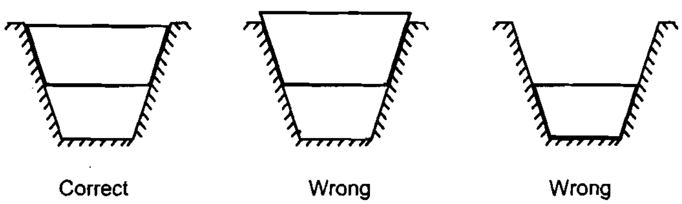


Figure 9.7 : Correct and Wrong Mountings of V-belt in Sheave

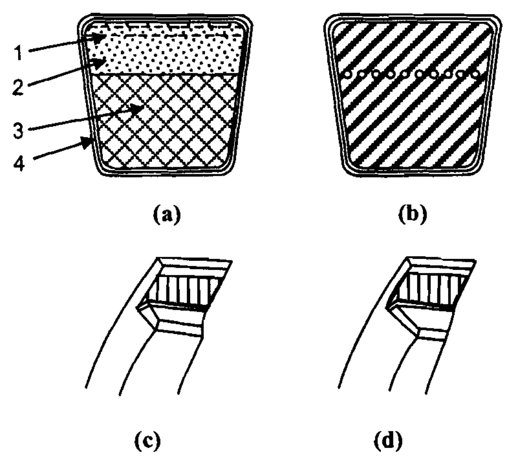


Figure 9.8 : V-belt Sections

Generally the V-belts have flat bottom. Alternative constructions of V-belts are shown in Figure 9.9. A cogged bottom, Figure 9.9(a), provides greater flexibility without loss of transverse rigidity. Cogged bottom permits smaller diameter of driving pulley and also increases the heat dissipation due to increased surface area. A ribbed bottom belt Figure 9.9(b) combines, the strength and simplicity of flat belt with wedgring action of V-belts.

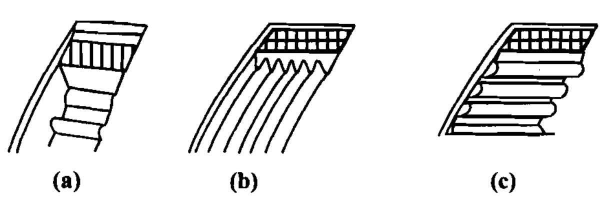


Figure 9.9

Recent developments in V-belt technology have brought into practice several types of sections. Synchronous (toothed) belts which combine the characteristics of belts and chain have replaced chains in many applications. The important characteristics of these belts is no slippage whereby the ratio between speeds of driving and driven shafts remains constant (Figure 9.9(c)).

9.16 STANDARD SIZES OF V-BELTS

Standard or classic V-belts are still in wide use. They are referred to as V-belt here. Five different sizes of V-belt cross-sections are commercially available and they cover wide range of general requirements. The sections are named as *A*, *B*, *C*, *D* and *E* in order of increasing dimensions. Figure 9.9 shows the nominal dimensions of these sections. Actual dimensions may vary slightly from these but will operate satisfactory in standard grooved sheaves. Table 9.11 gives the cross-sectional areas for these sections.

Table 9.11 : Cross-sectional Areas of V-belt Sections

Belt Section	*A*	*B*	*C*	*D*	*E*
Area (mm)^2	87.74	118.71	280	535.5	729

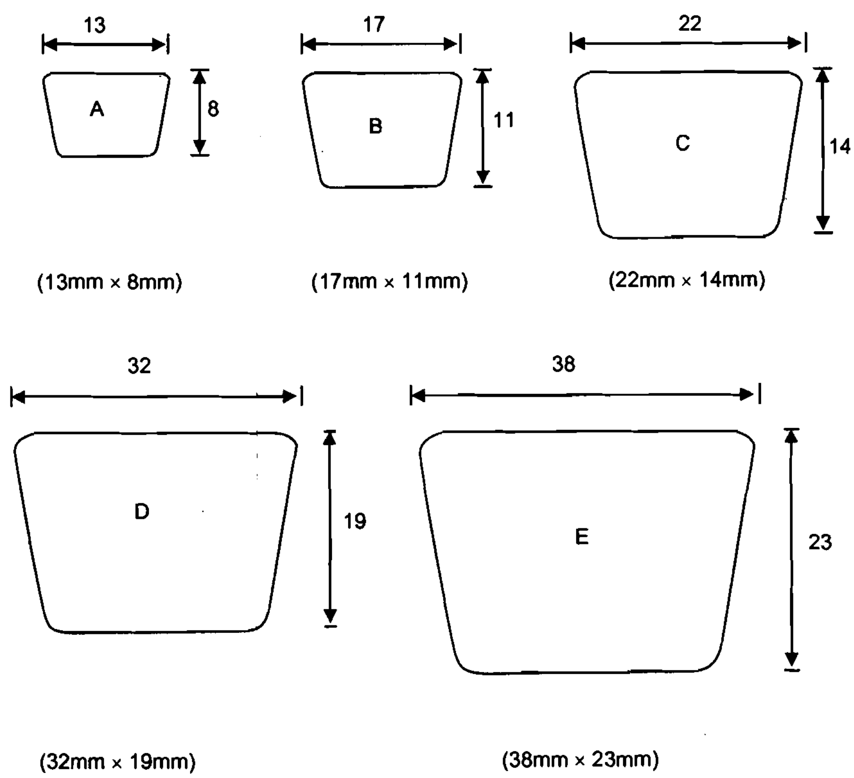


Figure 9.10 : Various V-belt Sections

A V-belt is generally recognised by its nominal length which follows the letter representing the size of the cross-section. The actual pitch length of the belt may differ from the nominal length used for its recognition. Thus, an A 26 belt has its nominal length equal to 26 inches but actual pitch length would be 27.4 inches or 696 mm. Similarly, C 51 belt has actual pitch lengths of 53.2 inches or 1351.3 mm. Standard pitch lengths of V-belt, manufactured by Goodyear (India) are described in Table 9.12. However, for calculating actual length of V-belt we may refer to Figure 9.4 and Eq. (9.20). The angle of contact on smaller pulley can be calculated by use of Eq. (9.19). Table 9.13 describes the V-belt data according to Russian Standards, which provides for seven different cross-sections; *O*, *A*, *B*, *C*, *D*, *E* and *F*.

Table 9.12 : Standard V-Belts Pitch Lengths

Nominal Length inches (mm)	Standard Pitch Lengths, inches (mm)			
	A Section	*B* Section	*C* Section	*D* Section
(1)	(2)	(3)	(4)	(5)
26 (660)	27.4 (696)	–	–	–
31 (787)	32.4 (823)	–	–	–
33 (838)	34.4 (874)	–	–	–
35 (889)	36.4 (924.6)	36.7 (932.4)	–	–
38 (965)	39.4 (1001)	39.7 (1008.4)	–	–
42 (1066.8)	43.4 (1102.4)	43.7 (1110)	–	–
46 (1168)	47.4 (1204)	47.7 (1212)	–	–
48 (1219)	49.5 (1257)	–	–	–
51 (1295)	52.4 (1331)	52.7 (1339)	53.2 (1351.3)	–
53 (1346)	54.4 (1382)	54.7 (1389.4)	–	–
55 (1397)	56.4 (1433)	56.7 (1440.2)	–	–
60 (1524)	61.4 (1561)	61.7 (1567.2)	62.2 (1580)	–
62 (1575)	63.4 (1610.4)	63.7 (1618)	–	–
64 (1625)	65.4 (1661.2)	65.7 (1668.8)	–	–
68 (1727)	69.4 (1762.8)	69.7 (1770.4)	70.2 (1783.1)	–
75 (1905)	76.4 (1940.6)	76.7 (1948.2)	77.2 (1960.9)	–
78 (1981)	79.4 (2016.8)	79.7 (2024.4)	–	–
80 (2032)	81.4 (2067.6)	–	–	–
81 (2057)	–	82.7 (2100.6)	83.2 (2113.3)	–
83 (2108)	84.4 (2143.8)	84.7 (2151.4)	–	–
85 (2159)	86.4 (2194.6)	86.7 (2202.2)	87.2 (2215)	–
90 (2286)	91.4 (2474)	91.7 (2329.27)	92.2 (2342)	–
96 (2438)	98.4 (2499.4)	–	98.2 (2493)	–
97 (2464)	106.4 (2702.6)	98.7 (2507)	–	–
105 (267)	113.4 (2880.4)	106.7 (2710.2)	107.2 (2723)	–
112 (2845)	121.4 (3083.6)	113.7 (2888)	114.2 (2900.7)	–
120 (3048)	129.4 (3286.8)	121.7 (3091.2)	122.2 (3104)	123.1 (3126.7)
128 (3251)	137.4 (3490)	129.7 (3294.4)	130.2 (3307.1)	131.1 (3330)
136 (2454)	145.4 (3693.2)	137.7 (3497.6)	–	–
144 (3658)	–	145.7 (3700.8)	146.2 (3713.5)	147.1 (3736.3)
158 (4013)	–	159.7 (4056.4)	160.2 (4069.1)	161.1 (4092)
162 (4115)	–	163.7 (4158)	164.2 (4171)	165.1 (4193.5)
173 (4394)	–	174.7 (4437.4)	175.2 (4450)	176.1 (4473)
180 (4572)	–	181.7 (4615.2)	182.2 (4628)	183.1 (4651)
195 (4953)	–	196.7 (4991.2)	197.2 (5009)	198.1 (5032)
210 (5334)	–	211.7 (5377.2)	212.2 (5390)	213.1 (5413)
240 (6096)	–	241.7 (6139.2)	242.2 (6152)	243.1 (6175)
270 (6858)	–	271.7 (6901.2)	272.2 (6914)	273.1 (6937)
300 (7620)	–	301.7 (7663.2)	302.2 (7676)	303.1 (769)
330 (8382)	–	–	332.2 (8438)	333.1 (8461)
360 (9144)	–	–	362.2 (9200)	363.1 (9223)
396 (10058)	–	–	–	39.1 (10137.1)
408 (10363)	–	–	–	411.1 (10442)

Table 9.13 : V-Belts (Russian Standards)

V-Belt Characteristics	V-Belt Cross-section						
	O	*A*	*B*	*C*	*D*	*E*	*F*
Cross-sectional area (mm^2)	50	80	140	230	480	700	1170
Nominal length (mm) Minimum Maximum	 500 2500	 500 4000	 630 6300	 1800 9000	 3150 11000	 4500 14000	 6300 14000
Difference between design and nominal inner length (mm)	25	33	40	55	76	95	120
Minimum design diameter of pulley (mm)	63	90	125	200	315	500	800
Maximum belt speed (m/s)	25	25	25	25	30	30	30
Size of pulley groove (mm)	10 2.5 12 8	12.5 3.5 16 10	15 5 20 12.5	21 6 26 17	28.1 8.5 37.5 24	34 10 44.5 29	43 12.5 58 38

9.17 DESIGN OF V-BELT DRIVES

The power transmitted by a V-belt drive can be calculated from the general formula

$$H = T_e \, v, \text{ W} \qquad \dots (9.35)$$

where T_e = Effective tension in belt (N), and

v = Belt speed (m/s).

The effective tension in above equation in first step of calculation is replaced by $(T_t - T_c)$ where T_t and T_c are respectively tension in tight side and centrifugal tension. T_e could be taken as $(T_1 - T_2)$ but since $T_2 << T_1$ it is neglected and hence $T_e = T_1 = T_t - T_c$. $\frac{T_t}{A}$, where A is the cross-sectional area of the belt, is the stress (tensile) generated in the belt because of T_t. Similarly, $\frac{T_c}{A}$ is the tensile stress in the belt section due to centrifugal tension. Calling these two stresses as σ_t and σ_c in N/mm^2 and dividing Eq. (9.35) by A

$$H_1 = \frac{H}{A} = (\sigma_1 - \sigma_c) \, v, \text{ W/mm}^2 \qquad \dots (9.36)$$

The permissible values for σ_1 and σ_c (for different belt speeds) are used for calculating power/mm^2. By using Table 9.11 the belt can be selected. Table 9.14 describes the permissible values of σ_c . σ_c is called centrifugal force coefficient. The permissible value of σ_1 = 2.245 N/mm^2 for standard V-belt.

Table 9.14 : Centrifugal Force Coefficient for Standard V-Belts

Belt Speed (m/min)	457	610	760	915	1065	1220	1370	1525
σ_c (MPa)	0.060	0.116	0.181	0.261	0.344	0.450	0.550	0.690

For example let us find belt section for transmitting 7 kW of power at a belt speed of 15.2 m/s. The speed is $15.2 \times 60 = 912$ m/min which is closest to 915 m/min in Table 9.15. Corresponding to 915 m/min $\sigma_c = 0.261$ N/mm^2.

Hence, from Eq. (9.36)

$$7 \times 10^3 = A\,(2.245 - 0.261)\,15.2 = 30A \text{ W}$$

$$\therefore \qquad A = \frac{7000}{30} = 233.33 \text{ mm}^2$$

i.e. the area of cross-section of belt must be 233.3 mm^2.

Search in Table 9.12 to find that two belts of B-section will have area of 237.42 mm^2 which is closest. Hence, two B-section belts are selected.

9.18 SELECTING V-BELT FROM MANUFACTURER'S CATALOGUE

For selection of belts the data supplied by belt manufacturers are commonly used. The power capacities of belt as given in manufacturer's catalogue are according to standard conditions and, therefore, corrections for angle of contact, belt and service conditions have to be applied.

Table 9.15 describes the power ratings of standard V-belts of various types at 180° arc of contact. All the values of power rating in this table are under standard conditions and are to be corrected. Figure 9.11 describes effective horsepower, of various belt sections as function of small sheave rpm. Effective power is the product of rated power of driving machine and service factors. Service factors will be described later. Figure 9.11 is divided into five distinct regions, for five standard belt sections. For a point lying upon the line of demarcation or near it, both the sections of the belt should be investigated and final choice may depend upon considerations of space limitation and economy, etc. Table 9.15 correlates effective diameter and belt speed with belt ratings of different sections.

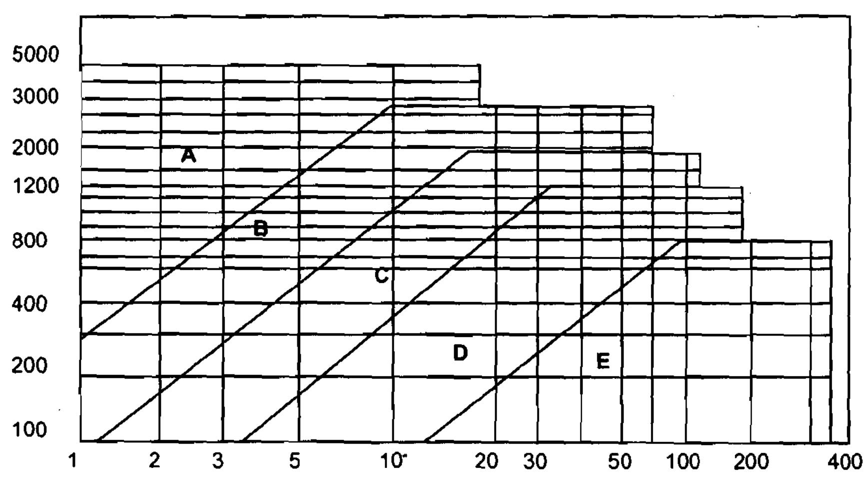


Figure 9.11 : Effective Power of Belts as Function of RPM of Small Shear (Based on Goodyear Hand Book)

The effective diameter, d_e, is defined as :

$$d_e = \text{Small pulley pitch diameter} \times \text{small diameter factors.}$$

Small diameter factor is detailed in Table 9.16.

Table 9.15 : Power Rating per Standard V-Belt at 180° Arc of Contact (W)

Belt Section	Effective Dia. (mm)	Belt Speed (m/min)													
		120	180	305	366	488	610	730	915	1100	1220	1400	1525	1645	1830
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)
A	66	261	335.7	462.5	515	582	627	634	582	448	306	15	–	–	–
	71	305	403	574	642	761	843	895	910	843	746	515	298	30	–
	76	343	462.5	671	761	910	1030	1126	1194	1186	1126	947	776	545	90
	86	403	552	821	947	1164	1343	1500	1664	1746	1746	1664	1552	1380	1022
	91	425	597	888	1022	1261	1477	1656	1805	1977	2007	1962	1872	1731	1410
	102	470	656.5	1000	1156	1440	1693	1917	2186	2372	2447	2469	2425	2327	2074
	112	507	716	1089	1261	1582	1872	2134	2462	2701	2812	2887	2880	2820	2618
	122	537	761	1164	1350	1708	2029	2313	2686	2969	3111	3238	3253	3223	3066
	127 & up	552	776	1201	1395	1761	2089	2328	2783	3088	3245	3387	3424	3409	3267
B	117	627	858	1253	1425	1723	1969	2163	2342	2372	2253	2052	1783	1417	686
	122	664	918	1350	1537	1880	2163	2387	2626	2708	2678	2492	2253	1932	1253
	127	701	970	1432	1641	2014	2335	2596	2865	3021	3029	2887	2686	2395	1776
	137	761	1060	1590	1828	2260	2641	2969	3350	3581	3648	3596	3461	3230	2701
	142	791	1104	1656	1910	2372	2783	3133	3551	3827	3924	3917	3805	3603	3118
	152	843	1171	1783	2060	2574	3029	3432	3924	4275	4416	4484	4446	4275	3857
	163	880	1238	1887	2186	2745	3245	3670	4252	4663	4849	4983	4968	4857	4513
	173	925	1298	1984	2305	2894	3435	3924	4536	5006	5237	5423	5446	5379	5088
	178 & up	940	1320	2029	2357	2969	3521	4028	4670	5162	5416	5625	5662	5610	5349
C	178	1194	1634	2395	2723	3312	3805	4193	4573	4700	4625	4252	3805	3208	1969
	191	1306	1798	2671	3059	3752	4357	4849	5401	5692	5729	5520	5185	4692	3626
	203	1403	1940	2909	3342	4140	4834	5431	6125	6565	6692	6624	6393	5998	5073
	229	1559	2178	3312	3827	4782	5640	6401	7333	7982	8281	8504	8430	8206	7460
	241	1626	2283	3476	4028	5058	5983	6804	7833	8654	8952	9250	9250	9101	8504
	254	1686	2380	3633	4894	5297	6281	7169	8281	9176	9623	9922	9996	9922	9400
	279	1798	2536	3894	4528	5722	6811	7833	9101	10146	10668	11190	11339	11339	10966
	292	1843	2604	4014	1663	5901	7042	8057	9399	10519	11115	11712	11936	11936	11712
	305 & up	1880	2663	4118	4797	6072	7251	8355	9773	10892	11488	12160	12458	12533	12309
D	305	2663	3655	5394	6162	7460	8728	9698	10668	11265	11190	10668	9922	8877	6542
	318	2798	3857	5737	6572	8057	9400	10519	11712	12533	12533	11488	11638	10668	8579
	330	2924	4051	6050	6953	8579	9996	11265	12682	13577	13801	13726	13204	12384	10519
	356	3148	4387	6617	7609	9474	11190	12607	14398	15591	16114	16263	16039	15442	13876
	368	3253	4543	6871	7908	9922	11638	13204	15144	16487	16561	17456	17307	16785	15368
	381	3343	4685	7102	8206	10295	12160	13801	15815	17382	18053	18575	18501	18053	16785
	406	3514	4939	7535	8728	10966	12980	14771	17158	18874	19769	20515	20590	20366	19396
	419	3596	5058	7758	8252	11265	13353	15293	17680	19620	20515	21410	21559	21485	20590
	432 & up	3670	5162	7908	9176	11563	13726	15741	18277	20261	21261	22231	22529	22455	21634

Table 9.16 : Small Diameter Factor

Speed Ratio	Small Diameter Factor	Speed Ratio	Small Diameter Factor	Speed Ratio	Small Diameter Factor
1.00	1.00	1.1	1.05	1.34	1.10
1.02	1.01	1.14	1.06	1.43	1.11
1.04	1.02	1.18	1.07	1.56	1.12
1.06	1.03	1.22	1.08	1.82	1.13
1.08	1.04	1.28	1.09	3.004 Larger	1.14

9.19 CORRECTION AND SERVICE FACTORS

Correction factors are multiplying factors on belt power capacity from tables to bring into consideration the difference in actual belt condition and those used in laboratory where capacity of belt was determined. The service factors are also multiplying factors to take care of machines which are driving and driven. The service factors multiply the power to be transmitted.

Correction Factor for Angle of Contact, K_{AC}

It is known that belt tension is function of angle of contact. Smaller angles have to be used if velocity ratio is high. The standard power capacity is based on 180° contact angle. If both pulleys are grooved, no correction is needed for 180° contact. But it was pointed out earlier that driven pulley may be flat in which case some correction may be needed. Table 9.17 can be used to find K_{AC}.

Table 9.17 : Correction Factor of Angle of Contact, K_{AC}

Angle of Contact (deg.)	Correction Factor	
	V-V Combination	V-Flat Combination
180	1.00	0.75
170	0.98	0.77
160	0.95	0.80
150	0.92	0.82
140	0.89	0.84
130	0.86	0.86
120	0.82	0.82
110	0.78	0.78
100	0.74	0.74
90	0.69	0.69

Length Correction Factor, K_L

A short length belt will pass round the pulleys more number of times than a longer belt for same revolutions of driving pulley. This will cause greater number of stress reversal in smaller belt which will tend to have reduced fatigue life in terms of number of hours. Some standards prescribe the frequency of flexing and introduce the limiting frequency in belt selection itself. We present the length correction factor as recommended by Indian manufacturer.

Table 9.18 : Belt Length Correction Factor, K_L

Belt Section	Short Length Group $K_L = 0.9$	Nominal Length Group $K_L = 1.0$	Longer Length Group $K_L = 1.1$
A	Under 1295 mm	1295 to 1981 mm	Above 1951 mm
B	Under 1803 mm	1803 to 2671 mm	Above 2667 mm
C	Under 2667 mm	2667 to 4013 mm	Above 4013 mm
D	Under 4115 mm	4013 to 6858 mm	Above 6858 mm
E	Under 6096 mm	6858 to 9144 mm	Above 9144 mm

Service Factor, K_S

This factor enhances the power that is transmitted. The driven machines could be of several types. Some will exert smooth load, yet many of them apply sudden and shock loads like crushers and hammers. The prime movers are also several like electric motors, IC engines, turbines, etc. They have different characteristics, particularly in respect of starting torque.

Further, the conditions of operation may vary widely. Table 9.19 describes service factors for combination of driving and driven machines. The higher value of K_o must be chosen for following operating conditions.

(a) Drive operates for 24 hours a day.

(b) Idle pulley is used on the belt.

(c) The drive is often stopped and started.

(d) The drive is often reversed.

(e) The drive is a multiplier.

Table 9.19 : Service Factor

Driving Machines / Driven Machines	AC Split Phase, Squirrel Cage, Synchronous, DC Shunt Wound Motor, Water Steam Turbines, IC Engines 4 Cylinders and more	Motors : AC Single ph, Series Wound, Induction, Slip Ring, Sq. Cage, DC Compd. or Series Wound, IC Engine Less than 4 Cylinders	Motors : AC Slip Ring, DC Series Wound, Line shaft with Clutch on Driving or Driven Shaft
Centrifugal pumps and blowers, engraving machines, floatation machine, multiple cylinder compressors, rotor pumps and small fans	1.0 to 1.2	1.1 to 1.3	1.2 to 1.5
Conveyors generators, hammer mills, large fans, line shafts, machine tools, oil pumping equipment, plunger pump, saw mill, textile machine, 1 or 2 cylinder compressor	1.1 to 1.5	1.2 to 1.6	1.3 to 2.0
Ball mills, crushers, DC exciters, hammers, hoists, oil well drills, excavators, rolling mills, stamp mills	1.2 to 1.6	1.4 to 2.0	1.6 to 3.0

If H = Power transmitted, and

h = Power capacity of one belt of a specified section.

$$\text{Then number of belts} = \frac{K_S H}{K_{AS} K_L h} \qquad \ldots (9.37)$$

A few machines have been named in first column of Table 9.19. These machines are driven by standard V-belts which are of five standard section and shown in Figures 9.8(a) and (b). Soil compactors, paper shredders, rotary press, grinders, extruders (plastic), cylindrical dryers, and CNC lathes use cogged belts as shown in Figure 9.9(a). V-ribbed (Figure 9.9(b)) are used to drive franking machines, hard copy printer, hand hold planes, lift door mechanisms, washing machine, floor polishers, IC engine auxilliary units, etc. synchronous belts (Figure 9.9(c)) are used in conjunction with camshaft drive, cultivators, weaving machines, positioning drives, adhesive bonding equipment, etc. We will be concerned only with standard or classic V-belts.

Example 9.3

A single phase series wound AC motor drives a heavy blower through V-belt. The rpm of motor is 1440 while the blower rotates at 482 rpm. The pitch diameters of driving and driven sheaves are respectively 100 and 300 mm. The power of electric motor is 7460 W. The blower works continuously for 8 hours. Select V-belt and number of belts. The centre distance is 220 mm.

Solution

Use Eq. (9.20) to calculate pitch length of the belt

$$L = 2C + \frac{\pi}{2}(D_2 + D_1) + \frac{(D_2 - D_1)^2}{4C}$$

Here, D_1 = 100 mm, D_2 = 300 mm, C = 220 mm

$$L = 2 \times 220 + \frac{\pi}{2}(300 + 100) + \frac{(300 - 100)^2}{4 \times 220} = 440 + 673.75 = 1113.5 \text{ mm} \quad \ldots \text{(i)}$$

For 8 hours operation/day 1-phase series wound AC motor to drive a blower, K_s = 1.1 from Table 9.20.

$$\therefore \quad \text{Effective power, } H = 1.1 \times 7460 = 8206 \text{ W} \quad \ldots \text{(ii)}$$

$$\text{The belt velocity} = \frac{\pi D_1 N_1}{60} = \frac{\pi \times 100 \times 1440}{60} = 7540 \text{ mm/s} = 7.54 \text{ m/s} \quad \ldots \text{(iii)}$$

Calculation of Belt Section Area

Since T_c is proportional to v^2, using σ_c corresponding to 457 m/min, i.e. 7.62 m/s as 0.06 N/mm^2, then corresponding to 7.54 m/s,

$$\sigma_c = \left(\frac{7.54}{7.62}\right)^2 \times 0.06 = 0.0587 \text{ N/mm}^2$$

Using Eq. (9.36),

$$H = 8206 = A(\sigma_1 - \sigma_c)\, v = A\,(2.245 - 0.0587)\, 7.54$$

$$A = \frac{8206}{16.485} = 497.8 \text{ mm}^2$$

Table 9.11 gives areas of several sections from A to E but it does not help in choosing the section. If we select A section, then

$$\text{Number of belts} = \frac{497.8}{87.74} = 5.67 \quad \text{say } 6 \quad \ldots \text{(iv)}$$

If we choose B section, then

$$\text{Number of belts} = \frac{497.8}{118.71} = 4.19 \quad \text{say} \quad 5$$

However, we can draw upon Figure 9.11 to decide the section. from this figure for power of 8.2 kW and small sheave rpm of 1440 a B section belt can be chosen.

$\therefore$ A B section belt is chosen whose actual length is 1113.5 mm which is not available in Table 9.17 for B section. The closest standard actual length is 1110 mm for which nominal length is 1066.6 mm or 42 in. Five 42 B belts are to be used.

Selection of Belt Using Tabulated Values

We can go for this procedure as an alternative to calculation method. The calculation of length, service factor, effective power and decision to use B section will still hold.

For velocity or speed ratio of $\frac{1440}{482} = 3$ from Table 9.16 the small diameter factor is 1.14.

$\therefore$ Effective diameter of small pulley = $1.14 \times 100 = 14$ mm

Belt speed = $7.54 \times 60 = 452.4$ m/min

From Table 9.15 for B section belt for effective small sheave diameter of 117 mm (closest to 114 mm) and belt speed of 488 m/min (closest to 452.4 m/min) the power capacity of one belt $h = 1723$ W.

From Table 9.18 for nominal length of 1066 is less than 1803 mm, so that length correction factor $K_L = 0.9$.

From Eq. (9.19) angle of contact on small pulley

$$\theta = \left(\pi - \frac{D_2 - D_1}{C}\right) = \pi - \frac{300 - 100}{220} = 2.23 \text{ rad} = 128 \text{ deg}$$

From Table 9.17, the angle of contact correction factor for $\theta = 130^\circ$ (closest to 128°).

$$K_{AC} = 0.86$$

Use Eq. (9.37),

Number of B section belts

$$= \frac{K_S H}{K_{AC} K_L h} = \frac{1.1 \times 7460}{0.86 \times 0.9 \times 1723} = 6.15$$

So seven 42 B belts are to be used.

Apparently the method of calculation could not consider K_{AC} and K_L. If we use both these factors on second of (iv). The number of belts by calculation

$$= \frac{4.19}{0.86 \times 0.9} = 5.4 \quad \text{say} \quad 6.$$

It is advisable to use tables of manufacturers in practice.

SAQ 3

(a) What are standard V-belt section? Give areas of cross-sections.

(b) Describe structure of V-belt. What are different types of V-belts.

(c) Show how should the V-belt be placed in pulley. Why should the angle of groove not be less than 18°.

(d) Why a small length of belt should have lower K_L?

(e) In which case you can have a flat surface driven pulley.

(f) A V-belt drive with a velocity ratio of 4 has belt running at a speed of 600 m/min and transmits 11 kW. The prime mover is 3 cylinder IC engine which with speed of 750 rpm runs the pulley on the shaft of a belt conveyor. The operation has to be uninterrupted for 16 hours everyday. Make driven pulley with flat surface. The coefficient of creep is 3%, hence, find diameter of larger pulley to maintain velocity ratio of 4. Take minimum permissible centre distance.

9.20 SUMMARY

Belt drives of flat and V-types provide power transmission system between parallel shafts. Although variants for shafts at right angle in case of flat belts and with variable speed ratio in case of V-belts are available, their applications are much restricted and drives with parallel shafts and constant velocity ratios only have been discussed. The belt is subjected to direct tensile stress due to driving and centrifugal tension and additive bending stress due to bending over circumference of the pulley. All these stresses together should not exceed the permissible tensile stress, becomes the governing equation for calculating the dimensions of the belt section. In case of V-belt the effective coefficient of friction becomes three times the basic property resulting much higher tension in tight side than in slack side. The latter is neglected in comparison with former and thus, equation used for calculating cross-section of V-belts which are classified as five groups of sections A, B, C, D and E with increasing size becomes function of T_t and T_c.

The manufacturers provide data for their products. In case of flat belts it is in terms of power per mm width while in case of V-belt it is on the basis of per belt. The recommendations for modification of power transmitted and belt power capacity depending upon conditions of operation through correction/service factors have been incorporated. Such factors when chosen judiciously can help attain a realistic design.

The methods of calculation of length, angle of contact, centre distance and creep phenomenon in belt have also been introduced. The structure of belts and materials have been discussed.

9.21 KEY WORDS

Flat Belt : A belt of rectangular cross-section that connects flat surface pulleys.

V-belt : A belt of trapezoidal cross-section that connects V-grooved pulleys.

Pulley : Made up of a hub that sits on shaft and drum or solid web connecting hub with cylindrical surface.

Idler Pulley : A pulley which does not transmit power.

Sheave : Grooved pulley.

Creep : Small movement of belt on pulley either in forward or backward direction.

Service Factor : Numerical factor that multiplies nominal power to take care of types of driving and driven machines.

9.22 ANSWERS TO SAQs

SAQ 1

(d) $\theta = 165^\circ = \dfrac{165 \times \pi}{180} = 2.88$ rad. Use $\mu = 0.3$.

$$\frac{T_1}{T_2} = e^{\mu\theta} = e^{0.3 \times 2.88} = e^{0.864} = 2.37$$

$$\therefore \quad T_1 = 2.37\, T_2$$

Power, $H = 33.5 \times 1000 = 33500 \text{ W} = (T_1 - T_2)\, v$

$$= T_2\,(2.37 - 1)\,\frac{1.5}{2} \cdot \frac{2\pi\, 300}{60}$$

$$\therefore \quad T_2 = \frac{35000}{32.3} = 1037.8 \text{ N}$$

$$T_1 = 2.37 \times 1037.8 = 2459.6 \text{ N} \qquad \ldots \text{(i)}$$

Area of the belt section $= b\,h \text{ mm}^2 = 150\,h \times 10^{-6} \text{ m}^2$

$\therefore$ Belt weight per meter $= w = 13750 \times 150\,h \times 10^{-6}$ N $= 2.06\,h$ N

Belt velocity, $v = \dfrac{2\pi N_1}{60} \cdot \dfrac{D_1}{2} = \dfrac{1.5}{2}\,\dfrac{2\pi\, 300}{60} = 23.56$ m/s

Check that permissible belt velocity from Table 9.1 is 25 m/s.

$\therefore$ Centrifugal tension, $T_c = \dfrac{w v^2}{g} = \dfrac{2.06\,h \times 23.56^2}{9.81} = 116.6\,h$ N $\qquad \ldots$ (ii)

Now use $\dfrac{T_1}{A} + \dfrac{T_c}{A} + \dfrac{E h}{D_1} = \dfrac{\sigma_u}{f\,.\,s} = \dfrac{37}{10}$

or $\dfrac{2459.6 + 116.6\,h}{150\,h} + \dfrac{100\,h}{1500} = 3.7$

i.e. $2459.6 + 116.6\,h + 10\,h^2 = 555\,h$

or $h^2 - 43.84\,h + 246 = 0$

$\therefore \quad h = 21.92 \pm \dfrac{1}{2}\sqrt{1922 - 984}$

$$= 21.9 \pm 15.3$$

$$h = 37.23 \quad \text{or} \quad 6.6 \text{ mm}$$

Choose $h = 6.6$ mm

SAQ 2

(a) $D_1 = 200$ mm, $D_2 = 800$ mm, $C = 2$ m

$$\theta = \pi - \frac{D_2 - D_1}{C} = \pi - \frac{800 - 200}{2000} = \pi - \frac{6}{20} = 2.84 \text{ rad} = 162.8 \text{ deg}$$

θ is angle of contact on smaller pulley. On larger pulley it is

$$\pi + \frac{D_2 - D_1}{C} = \pi + 0.3 = 3.44 \text{ rad} = 197.2 \text{ deg}$$

Sum of two angles $= 162.8 + 197.2 = 360$ deg or 2π rad

$$C_{min} = 1.5\,(D_1 + D_2) = 1.5 \times 1000 = 1500 \text{ mm}$$

Making $\quad \theta = \pi - \dfrac{800 - 200}{1500} = 2.74 \text{ rad} = 157.1 \text{ deg}$

This will reduce $\dfrac{T_1}{T_2}$ resulting in lower value of T_1.

(d) $N_1 = 300$ rpm, $N_2 = 500$ rpm, $D_1 = 600$ mm + 6 mm, $D_2 = (D + 6)$ mm

Half of the thickness of belt on all sides of pulley is added to diameter to obtain effective diameter.

$$i = \frac{300}{500} = (1 + S)\,\frac{D_2}{D_1} = (1.03)\,\frac{D + 6}{606}$$

$$D + 6 = 353$$

or $\quad D = 353 - 6 = 347$ mm, the diameter of driven pulley.

(e) Nothing is said about driving pulley diameter the same may be calculated from Savrin's formula, Eq. (9.29)

$$D_1 = 1.114\left(\frac{H}{N_1}\right)^{\frac{1}{3}}$$

H is to be in kW, i.e. 8, $N_1 = 900$ rpm

$$\therefore \quad D_1 = 1.114\left(\frac{8}{900}\right)^{\frac{1}{3}} = 0.231 \text{ m or } 231 \text{ mm}$$

Select a standard diameter of 254 mm.

For $i = 4$, $\quad N_2 = \dfrac{900}{4} = 225$ rpm $\quad$ . . . (i)

Assume a creep coefficient of 2%

$$i = 4 = (1 + S)\,\frac{D_2}{D_1} = 1.02\,\frac{D_2}{254}$$

$$\therefore \quad D_2 = \frac{254 \times 4}{1.02} = 996 \text{ mm} \quad \text{. . . (ii)}$$

Belt speed $\quad v = \pi\, N_1\, D_1 = \pi \times 900 \times 254 \times 10^{-3} = 718.2$ m/min $= 12$ m/s

From Table 9.9 we can select rubber belt for 254 mm diameter driving pulley in 3, 4 or 5 plies. The belt speed of 718.2 m falls between 670 and 730 m/min. We select power capacity for lower speed so that larger width will be obtained. The power capacities for 3, 4 and 5 plies are : 0.87, 0.102 and 0.108 kW/mm, respectively.

The centre distance should be at least

$$1.5\,(D_1 + D_2) = 1.5\,(254 + 996) = 1875 \text{ mm}$$

The higher limit for centre distance is

$$2\,(D_1 + D_2) = 2\,(254 + 996) = 2500 \text{ mm}$$

Hence, from Table 9.10 choose a standard value for centre distance as 2450 m.

The difference in pulley diameters is 996 – 254 = 74 2 mm.

This difference is close to 762 in Table 9.10.

The angle of contact correction factor for pulley diameter difference of 763 mm and centre distance of 2.45 m is 0.92.

Allowing for wire hinges hook joint the $\eta = 0.45$.

So the width of belt for 3, 4 and 5 plies are calculated.

For 3-ply belt, $$b_3 = \frac{8}{0.92 \times 0.45 \times 0.087} = 222.1 \text{ mm}$$

For 4-ply belt, $$b_4 = \frac{8}{0.92 \times 0.45 \times 0.102} = 189.45 \text{ mm}$$

For 5-ply belt, $$b_5 = \frac{8}{0.92 \times 0.45 \times 0.108} = 178.92 \text{ mm}$$

As standard practice,

3 plies are recommended upto width of 100 mm.

4 plies are recommended upto width of 300 mm.

5 plies are recommended upto width of 350 mm.

Hence, a 4-ply rubber belt of 189.45 mm say 190 mm is chosen.

SAQ 3

(f) Belt speed, $v = 600 \text{ m/min} = \pi\, D_1\, N_1 = \pi\, D_1 \times 750$

$$\therefore \quad D_1 = \frac{600}{\pi \times 750} = 0.2546 \text{ m} = 254.6 \text{ mm}$$

Table 9.16, small diameter factor for $i > 3.004 = 1.14$

$\therefore$ Effective diameter of driving pulley $= 1.14 \times 254.6 = 290.24$ mm

Eq. (9.33), $$D_2 = \frac{i\, D_1}{1 + S} = \frac{4 \times 254.6}{1.03} = 988.73 \text{ mm}$$

The centre distance can be as low as

$$\frac{D_1 + D_2}{2} + 3h$$

h is the height of belt section.

For 3 cylinder IC engine driving a belt conveyor for 16 hr/day operation, from Table 9.19, K_s is between 1.2 to 1.6. Choose $K_s = 1.5$.

$\therefore$ Effective power $= 1.5 \times 11 = 16.5$ kW

From Figure 9.11, for effective power of 16.5 kW and small sheave rpm of 750, a C section belt is chosen. $h = 14$ mm (Figure 9.20).

$$\therefore \quad C = \frac{D_1 + D_2}{2} + 3 \times 14 = \frac{254.6 + 988.73}{2} + 42 = 663.665 \text{ mm}$$

From Eq. (9.20), belt length,

$$L = 2C + \frac{\pi}{2}(D_2 - D_1) + \frac{(D_2 - D_1)^2}{4C}$$

$$= 2 \times 663.7 + \frac{\pi}{2}(988.7 + 254.6) + \frac{(988.7 - 254.6)^2}{4 \times 663.7}$$

$$= 1327.4 + 1953 + 203 = 3483.4 \text{ mm}$$

The nearest standard C section belt has a length of 3713.5 mm = 146.2 in with nominal pitch length of 144 in.

From Table 9.18, since this length lies between 2667 and 4013 mm the length correction factor is $K_L = 1.0$.

Angle of contact on small pulley

$$\theta = \left(\pi - \frac{D_2 - D_1}{C}\right) = \pi - \frac{988.7 - 254.6}{663.7}$$

$$= \pi - 1.1 \doteq 2.036 \text{ rad} = 116.6 \text{ deg}$$

From Table 9.17, the angle of contact correction factor can be mean of 0.82 and 0.78, i.e. $K_{AC} = 0.8$ (for flat-V combination).

From Table 9.15 power rating of one C section belt for 292 mm effective diameter of small pulley and belt speed of 610 m/min. is

$$h = 7042 \text{ W} \quad \text{or} \quad 7.042 \text{ kW}$$

$$\therefore \quad \text{Number of belts} = \frac{K_S H}{K_L K_{AC} h} = \frac{1.5 \times 11}{1.0 \times 0.8 \times 7.042} = 2.93 \text{ say } 3$$

Thus, 3 belts 144 C are selected.