CHAPTER 31
BELT DRIVES

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NOMENCLATURE

A	Cross section
b	Width
c_β	Angular factor
c_B	Service factor
d_1	Diameter of driving pulley
d_2	Diameter of driven pulley
e	Center distance
E	Modulus of elasticity
F	Force
f_b	Bending frequency
l	Datum length of flexible connector
M	Torque
n	Speed
P	Power
q	Mass per length
r	Radius
s	Belt thickness
t	Pitch
v	Velocity
z	Number

α	Included angle
β	Angle of wrap
ε	Elongation (strain)
μ	Coefficient of friction
η	Efficiency
ψ	Slip
ρ	Specific mass
σ	Stress

Indices

1	Driving
2	Driven
b	Bending
f	Centrifugal
max	Maximum
w	Effective
zul	Allowable
N	Nominal

31.1 GENERAL

Flexible-connector drives are simple devices used to transmit torques and rotational motions from one to another or to several other shafts, which would usually be parallel. Power is transmitted by a flexible element (flexible connector) placed on pulleys, which are mounted on these shafts to reduce peripheral forces. The transmission ratios of torques and speeds at the driving and driven pulleys are determined by the ratio of pulley diameters. Peripheral forces may be transmitted by either frictional (nonpositive) or positive locking of the flexible connector on the pulleys.

Because of their special characteristics, flexible-connector drives have the following advantages and disadvantages as compared with other connector drives:

Advantages:

- Small amount of installation work
- Small amount of maintenance
- High reliability
- High peripheral velocities
- Good adaptability to the individual application
- In some cases, shock- and sound-absorbing
- In some cases, with continuously variable speed (variable-speed belt drive)

Disadvantages:

- Limited power transmission capacity
- Limited transmission ratio per pulley step
- In some cases, synchronous power transmission impossible (slip)
- In some cases, large axle and contact forces required

31.1.1 Classification According to Function

According to function, flexible-connector drives are classified as (1) nonpositive and (2) positive.

Nonpositive flexible-connector drives transmit the peripheral force by means of friction (mechanical force transmission) from the driving pulley to the flexible connector and from there to the driven pulley(s). The transmissible torque depends on the friction coefficient of the flexible connector and the pulleys as well as on the surface pressure on the pulley circumference. The power transmission capacity limit of the drive is reached when the flexible connector starts to slip. By use of wedge-shaped flexible connectors, the surface pressure can be increased, with shaft loads remaining constant, so that greater torques are transmitted. Since nonpositive flexible-connector drives tend to slip, synchronous power transmission is impracticable.

The positive flexible-connector drive transmits the peripheral force by positive locking of transverse elements (teeth) on the connector and the pulleys. The surface pressure required is small. The transmissible torque is limited by the distribution of the total peripheral force to the individual teeth in engagement and by their functional limits. The power transmission capacity limit of the drive is reached when the flexible connector slips. Power transmission is slip-free and synchronous.

31.1.2 Geometry

The dimensions of the different components [pulley diameter, center distance, datum length (pitch length) of the flexible connector] and the operational characteristics (speed ratio, angle of wrap, included angle) are directly interrelated.

Two-Pulley Drives. For the standard two-pulley drive, the geometry is simple (Fig. 31.1). In general, this drive is designed with the center distance and the speed ratio as parameters. The individual characteristics are related as follows: Speed ratio:

$$i = \frac{n_1}{n_2} = \frac{d_2}{d_1} \tag{31.1}$$

Included angle:

$$\sin\alpha = \frac{d_2 - d_1}{2e} = \frac{d_1}{2e}\,(i-1) \tag{31.2}$$

Angles of wrap:

$$\begin{aligned} \beta_1 &= 180° - 2\alpha = 180° - 2\arcsin\frac{d_1}{2e}\,(i-1) \\ \beta_2 &= 180° + 2\alpha = 180° + 2\arcsin\frac{d_1}{2e}\,(i-1) \end{aligned} \tag{31.3}$$

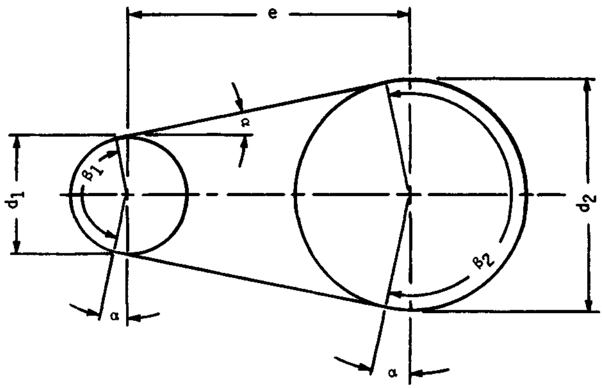

FIGURE 31.1 Two-pulley drive.

Datum length of flexible connector:

$$
\begin{aligned}
l &= 2e \cos \alpha + \pi \left(d_1 \frac{\beta_1}{360°} + d_2 \frac{\beta_2}{360°} \right) \\
&= 2e \cos \alpha + \frac{\pi d_1}{360°} \left[180° - 2\alpha + i(180° + 2\alpha) \right]
\end{aligned}
\tag{31.4}
$$

Approximate equation:

$$
\begin{aligned}
l &\approx 2e + 1.57(d_1 + d_2) + \frac{(d_2 - d_1)^2}{4e} \\
&= 2e + 1.57 d_1 (i + 1) + \frac{d_1^2}{4e} (i - 1)^2
\end{aligned}
\tag{31.5}
$$

The minimum diameter allowable for the flexible connector selected is often substituted for the unknown parameter d_1 (driving-pulley diameter) required for the design.

Multiple-Pulley Drives. For the multiple-pulley drive (one driving pulley, two or more driven pulleys), the geometry is dependent on the arrangement of the pulleys (Fig. 31.2). These drives have the following characteristics: Speed ratios:

$$
i_{12} = \frac{n_1}{n_2} = \frac{d_2}{d_1} \qquad i_{13} = \frac{n_1}{n_3} = \frac{d_3}{d_1} \qquad i_{1m} = \frac{n_1}{n_m} = \frac{d_m}{d_1}
$$

Included angles:

$$
\sin \alpha_{12} = \frac{d_1}{2e_{12}} (i_{12} - 1) \tag{31.6}
$$

$$
\sin \alpha_{13} = \frac{d_1}{2e_{13}} (i_{13} - 1) \tag{31.7}
$$

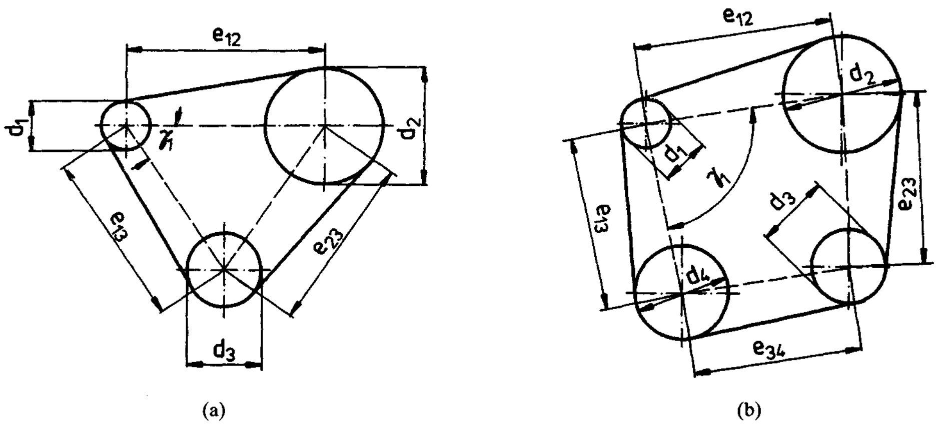

FIGURE 31.2 Multiple-pulley drives.

$$\sin \alpha_{1m} = \frac{d_1}{2e_{1m}} (i_{1m} - 1) \tag{31.8}$$

$$\sin \alpha_{km} = \frac{d_k}{2e_{km}} (i_{km} - 1) \tag{31.9}$$

Angles of wrap:

$$\beta_j = 180° - \alpha_{j,j-1} - \alpha_{j,j+1} - \gamma_j \tag{31.10}$$

where j = index of pulley
γ_j = angle between center distances

$$l = \frac{\beta_1 \pi d_1}{360} + e_{12} \cos \alpha_{12} + \frac{\beta_2 \pi d_2}{360} + e_{23} \cos \alpha_{23} + \cdots + \frac{\beta_k \pi d_k}{360} + e_{km} \cos \alpha_{km} + \frac{\beta_m \pi d_m}{360} + e_{1m} \cos \alpha_{1m} \tag{31.11}$$

31.1.3 Forces in Moving Belt

Friction is employed in transmitting the peripheral forces between the belt and the pulley. The relation of the friction coefficient μ, the arc of contact β, and the belt forces is expressed by Eytelwein's equation. For the extreme case, i.e., slippage along the entire arc of contact, this equation is

$$\frac{F_1'}{F_2'} = \exp \frac{\mu \beta \pi}{180} \tag{31.12}$$

For normal operation of the drive without belt slip, the peripheral force is transmitted only along the active arc of contact $\beta_w < \beta$ (according to Grashof), resulting in a force ratio between the belt sides of

$$\frac{F_1'}{F_2'} = \exp\frac{\mu\beta_w\pi}{180} \tag{31.13}$$

The transmission of the peripheral force between the belt and the pulley then occurs only within the active arc of contact β_w with belt creep at the driven pulley and the corresponding contraction slip at the driving pulley. During operation, the belt moves slip-free along the inactive arc of contact, then with creep along the active arc of contact. If the inactive arc of contact equals zero, the belt slips and may run off the pulley.

Along the inactive arc of contact, the angular velocity in the neutral plane equals that of the pulley. Along the active arc of contact, the velocity is higher in the tight side of the belt owing to higher tension in that side than in the slack side. Since this velocity difference has to be offset, slip results. This slip leads to a speed difference between the engagement point and the delivery point on each pulley, which amounts up to 2 percent depending on the belt material (modulus of elasticity), and load:

$$\psi = \frac{v_1 - v_2}{v_1} = \frac{(l_2 + \Delta l) - l_2}{l_2 + \Delta l} \approx \Delta\varepsilon = \frac{\sigma_1 - \sigma_2}{E} = \frac{\sigma_n}{E} \tag{31.14}$$

For practical design purposes, the calculations for a belt drive are usually based on the entire arc of contact β of the smaller pulley (full load), since the active arc of contact is not known, and the belt slips at the smaller pulley first.

$$\frac{F_1'}{F_2'} = m = \exp\frac{\mu\beta\pi}{180} \tag{31.15}$$

Centrifugal forces acting along the arcs of contact reduce the surface pressure there. As these forces are supported by the free belt sides, they act uniformly along the entire belt:

$$F_f = \rho v^2 A = q v^2 \tag{31.16}$$

With increasing belt velocity v, constant center distance e, and constant torques, the forces F_1 and F_2 acting along the belt sides as well as the peripheral force (usable force) F_u remain constant, whereas the surface pressure and the usable forces F_1' and F_2' in the belt sides are reduced. Usable forces in belt sides:

$$\begin{aligned} F_1' &= F_1 - F_f = mF_2' \\ F_2' &= F_2 - F_f = \frac{F_1'}{m} \end{aligned} \tag{31.17}$$

Peripheral force:

$$\begin{aligned} F_u = F_1' - F_2' = F_1 - F_2 &= F_1'\left(1 - \frac{1}{m}\right) \\ &= F_2'(m - 1) \end{aligned} \tag{31.18}$$

Because

$$F_u = F_2'(m-1) \qquad m = \exp\frac{\mu\beta_w\pi}{180} \tag{31.19}$$

β_w becomes greater, until the belt slips on the pulley with the smaller arc of contact when $\beta_w = \beta$. When $F_f = F_2$, there are no usable forces; that is, $F_2' = F_1' = F_u = 0$. In this case, no torque can be transmitted. If belt velocity v is increased further, the belt runs off the pulley.

The maximum force in the belt sides is given by

$$F_{\max} = F_1 = F_2' + F_u + F_f \tag{31.20}$$

With only the centrifugal forces acting, the belt is in equilibrium. They do not act on the pulleys at all. Hence, the shaft load F_w of a belt drive results from only the usable forces F_1' and F_2' in the belt sides (Fig. 31.3):

$$F_w = \sqrt{F_1'^2 + F_2'^2 - 2F_1'F_2'\cos\beta} \tag{31.21}$$

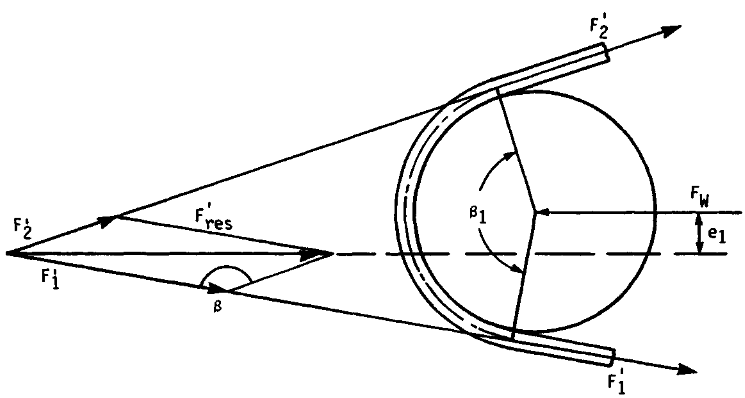

FIGURE 31.3 Equilibrium of forces.

The force rating

$$\Phi = \frac{F_u}{F_w} = (m-1)\sqrt{m^2 + 1 - 2m\cos\beta}$$

defines the minimum shaft tensioning force required for peripheral force production as a function of the friction coefficient μ and the arc of contact β.

The rated output $K = F_u/F_1' = 1 - 1/m$ defines the peripheral force F_u which can be produced by the permissible force F_1' as a function of the friction coefficient μ and the arc of contact β. The reduction in rated output with decreasing arc of contact is defined by an angular factor c_β, based on $\beta = 180°$, that is, a speed ratio of $i = 1$.

The tensions in a homogeneous belt result from the forces acting in the belt and the belt cross section, $A = bs$. For multiple-ply belts, these tensions can be used only as theoretical mean values.

Bending of the belt around the pulley produces the bending stress σ_b. This stress can be calculated from the elongation of the belt fibers with respect to the neutral axis:

$$\Delta l = \beta(r+s) - \beta\left(r + \frac{s}{2}\right) = \beta\,\frac{s}{2}$$

$$\varepsilon = \frac{\Delta l}{l} = \frac{\beta s/2}{\beta(r+s/2)} = \frac{s}{2r+s} \approx \frac{s}{d} \tag{31.22}$$

$$\sigma_b = \varepsilon E_b \approx \frac{s}{d}\,E_b \tag{31.23}$$

The strain ε increases with decreasing pulley diameter d. For practical design purposes, σ_b is not taken into consideration, since belt life depends much less on σ_b than on the bending frequency.

The maximum stress is in the tight side of the belt at the beginning and end of the arc of contact, i.e., the points where it passes onto or off the smaller pulley (Fig. 31.4):

$$\sigma_{max} = \sigma_1' + \sigma_f + \sigma_b = \frac{F_1'}{A} + \rho v^2 + E_b\frac{s}{d} \tag{31.24}$$

The safety stress depends on the bending frequency and the smallest pulley diameter as well as on the material and the construction of the belt as indicated by the manufacturer. With z = number of pulleys, the bending frequency is given by

$$f_b = \frac{vz}{l} \tag{31.25}$$

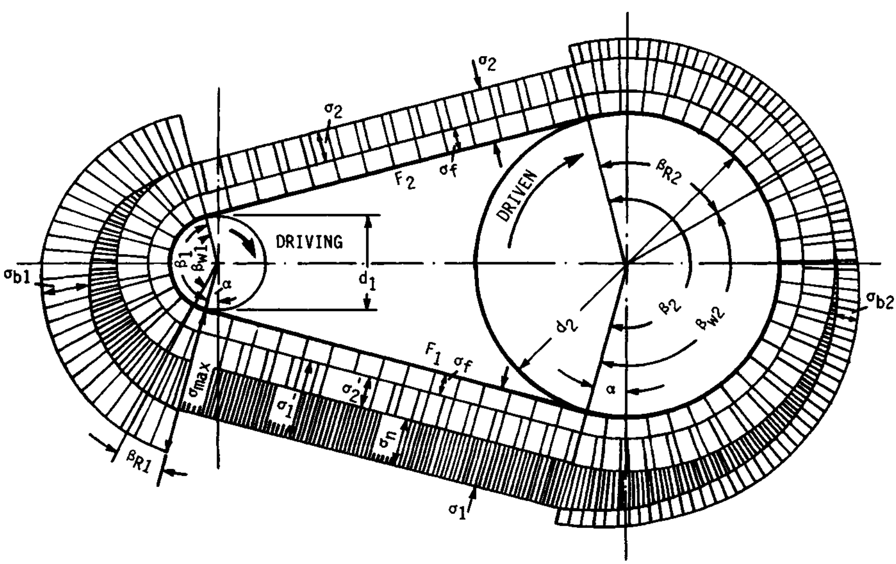

FIGURE 31.4 Stress distribution.

The maximum power transmission capacity of a belt drive can be determined as follows: The power transmission capacity

$$P = F_u v = \sigma_n A v$$

equals zero if the belt velocity v either equals zero or reaches a maximum at which the belt safety stress limit is approached by the centrifugal and bending stresses alone, so that

$$\sigma_1' = \sigma_2' = \sigma_n = 0$$

Then

$$\sigma_{zul} = \sigma_f + \sigma_b = \rho v_{max}^2 + \sigma_b \tag{31.26}$$

from which the maximum belt velocity can be calculated as follows:

$$v_{max} = \sqrt{\frac{\sigma_{zul} - \sigma_b}{\rho}} \tag{31.27}$$

Optimum power transmission is possible only at the optimum belt velocity v_{opt} within the range of $v = 0$ and $v = v_{max}$. It depends on the belt safety stress and is given by

$$v_{opt} = \sqrt{\frac{\sigma_{zul} - \sigma_b}{3\rho}} = \frac{v_{max}}{\sqrt{3}} \tag{31.28}$$

In theory, this equation applies to all flexible connectors, under the assumption of σ_{zul} (belt safety stress) [or F_{zul} (allowable load)] being independent of belt velocity. Since σ_{zul} decreases with increasing belt velocity, though, the stress and power transmission capacity diagrams are as shown in Fig. 31.5.

31.1.4 Arrangement and Tensioning Devices

Because of their good twistability, flexible connectors are suited for drives with pulleys in different planes and nonparallel shafts of equal or opposite directions of rotation. Since the outer fibers of a twisted flat belt or synchronous belt are strained more than the center fibers, stress is higher there, resulting in the reduction of the belt power transmission capacity.

Figures 31.6 and 31.7 show several belt drives with pulleys in different planes. Note that for drives with crossed belts (Fig. 31.6), endless belts have to be used, in order to avoid damage. For half- or quarter-turn belt drives (Fig. 31.7), the side of delivery must lie in the plane of the mating pulley. By the use of step (cone) pulleys, different speed ratios may be obtained (Fig. 31.8). Pulley diameters have to be selected to ensure equal belt lengths on all steps.

The belt rim running onto the larger diameter of a cone pulley (Fig. 31.9) has a higher velocity than the opposite rim. Thus, the following belt portion is skewed and then runs onto a larger diameter. The drive is balanced when the bending moment due to the bending deformation of the belt is compensated by the skew of the belt side running off (Fig. 31.10).

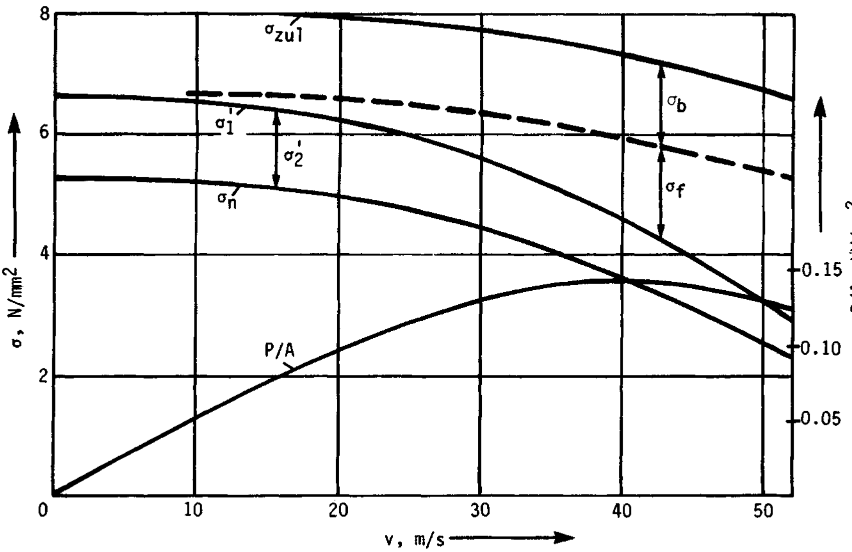

FIGURE 31.5 Stress and power transmission capacity.

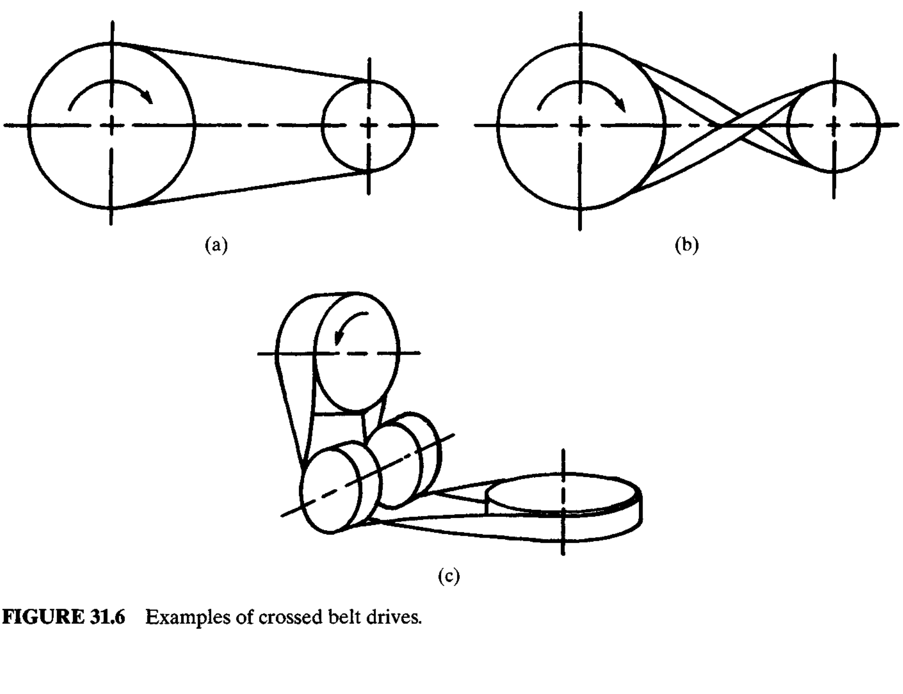

FIGURE 31.6 Examples of crossed belt drives.

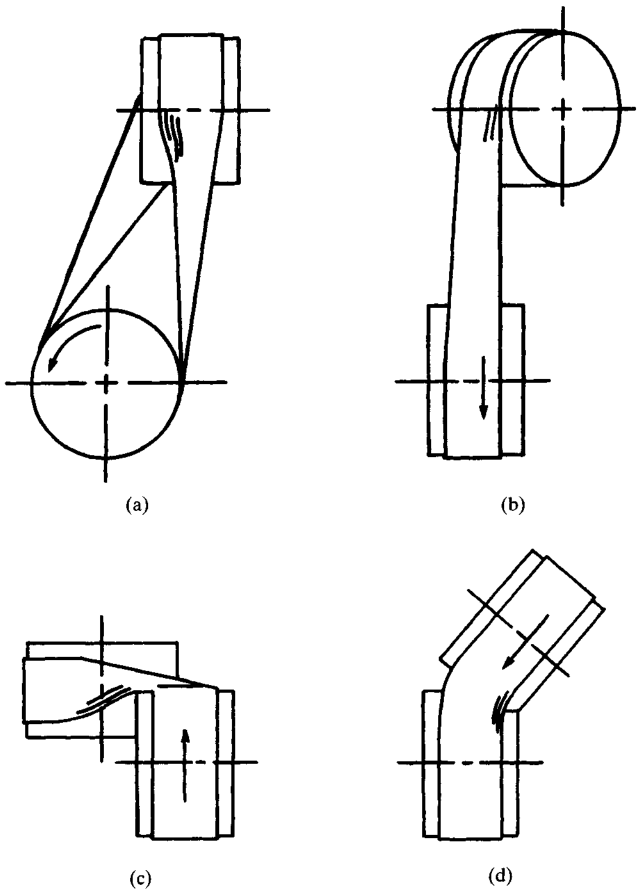

FIGURE 31.7 Belt drives with pulleys in different planes.

The minimum shaft tensioning force F_w required for nonpositive-type force transmission can be produced as shown in Figs. 31.11 to 31.14.

1. *Pretensioning by belt strain:* The belt is cut to such a length that it is elastically preloaded when it is placed on the pulleys. Since both the forces F' in the belt sides and the shaft initial tensioning force are reduced by the action of centrifugal forces, σ_f has to be added to the initial tension of the belt to ensure proper transmission of peripheral forces by friction.

2. *Pretensioning by adjustment of center distance:* The shaft tensioning force F_w is produced by shifting the driving motor on a slide. The belt drive may be preloaded either by adjustment of a threaded spindle or by spring action or weighting.

3. *Pretensioning by means of a belt tightener acting on the slack side:* The slack side of the belt is provided with a pulley to tighten it—with its own weight or by means of counterweights or by spring action—and increase the arcs of contact on

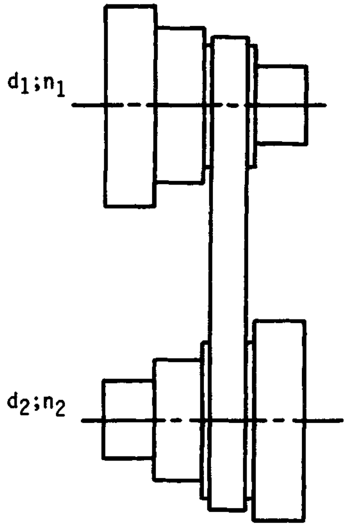

FIGURE 31.8 Step pulleys.

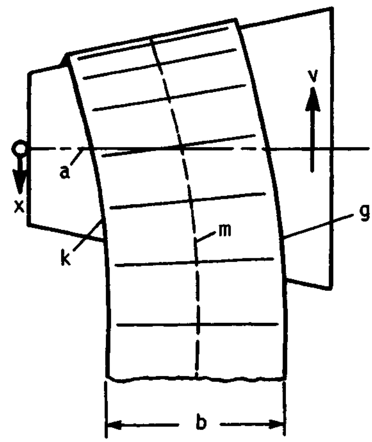

FIGURE 31.9 Cone pulley.

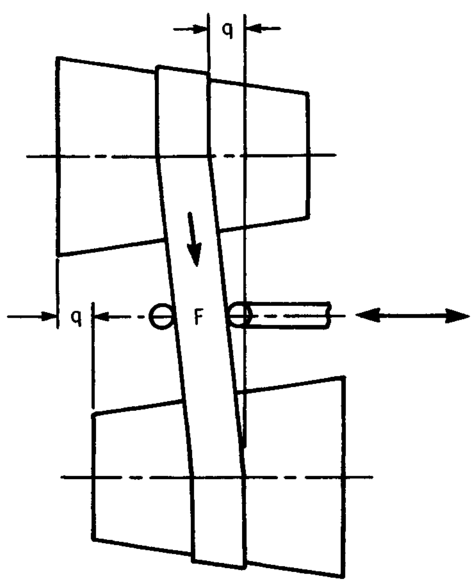

FIGURE 31.10 Cone-pulley drive.

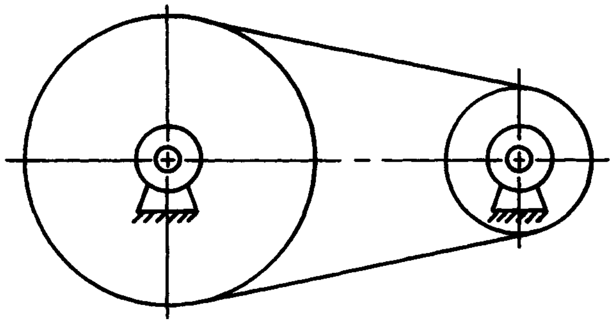

FIGURE 31.11 Pretensioning by belt strain.

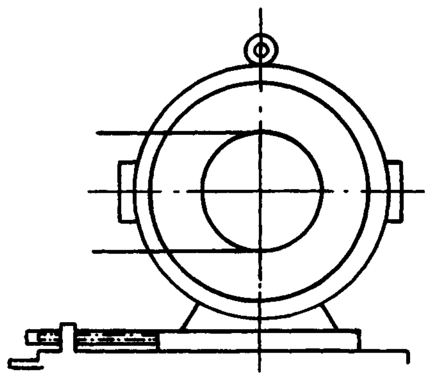

FIGURE 31.12 Pretensioning by adjustment of center distance.

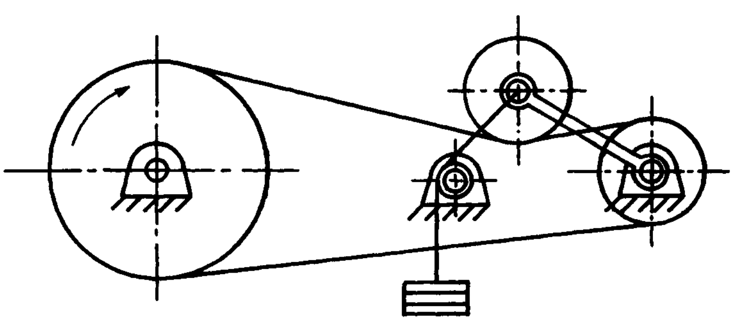

FIGURE 31.13 Pretensioning with a belt tightener.

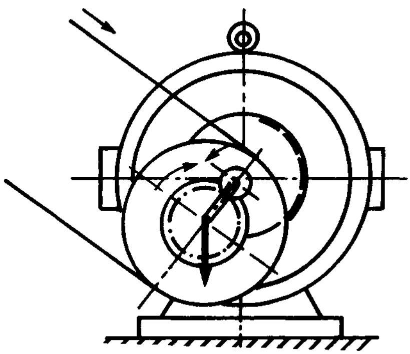

FIGURE 31.14 Pretensioning by torque.

both the driving and the driven pulleys. The belt tightener produces a constant force F_2' for all operating conditions but increases the bending frequency f_b, thus reducing the safety stress of the belt. Taking this into consideration, we see that the belt tightener used should have a minimum diameter equaling that of the smaller pulley, if possible.

4. *Pretensioning by torque making use of a rocker or pivoting pulley:* Figure 31.14 shows an arrangement with an eccentric pulley shaft pivoting the motor pulley shaft. The pulley is driven by a gear assembly. The shaft tensioning force F_w increases almost in proportion to the tooth force F_Z, and with the correct ratio of h_2/h_1, adapts automatically to the specific torque to be transmitted. Belt slipping is impossible with this method of pretensioning.

For all belt pretensioning methods using tensioning devices which can be adjusted during operation of the drive (e.g., methods 2 to 4 described above), the shaft tensioning force F_w and the usable forces F_1' and F_2' in the belt sides are not influenced by centrifugal force. Centrifugal force increases the belt stress by σ_f, though.

31.2 FLAT-BELT DRIVE

Calculations for flat-belt drives are based on Eytelwein's fundamental equation of belt friction:

$$\frac{F_1'}{F_2'} = \exp\frac{\mu\beta\pi}{180} \tag{31.29}$$

For calculations for a belt drive, the arc of contact of the smaller pulley is substituted for the arc of contact β, since the belt slips at the smaller pulley first, and the active arc of contact β_w is not known.

The transmissible power P is calculated from the peripheral forces

$$F_u = F_1' - F_2' = F_2'\left[\exp\left(\frac{\mu\beta\pi}{180}\right) - 1\right] \tag{31.30}$$

and the belt velocity v and is expressed by

$$P_1 = v_1 F_u P_2 = v_2 F_u \tag{31.31}$$

The belt velocities are

$$v_1 = \pi n_1 d_1 \qquad v_2 = \pi n_2 d_2 \tag{31.32}$$

Taking into consideration a service correction factor c_B (Table 31.1) for peak overloads due to heavy duty, we can calculate the power rating of the drive from P_1 by

$$P = c_B P_1 \tag{31.33}$$

The speed ratio, which is slightly dependent on the load because of slip, is

$$i = \frac{n_1}{n_2} = \frac{d_2 v_1}{d_1 v_2} = \frac{d_2}{d_1(1 - \sigma_n E)} = \frac{d_2}{d_1(1 - \psi)} \tag{31.34}$$

In calculations, the approximation

$$i \approx \frac{d_2}{d_1} \tag{31.35}$$

may be used.

The efficiency depends on the belt slip only, with the bearing friction and windage being neglected, since the pull of either belt side at both pulleys must be assumed to be equal:

$$\eta = \frac{P_2}{P_1} = \frac{M_2}{iM_1} = 1 - \frac{\sigma_n}{E} = 1 - \psi \tag{31.36}$$

where

$$M_1 = F_u \frac{d_1}{2} \qquad M_2 = F_u \frac{d_2}{2} \tag{31.37}$$

The bending frequency, which is of particular importance for belt life, is calculated by

$$f_b = \frac{vz}{l} \tag{31.38}$$

It should not exceed the limit specified for the particular belt material.

Modern high-performance flat belts are designed as multiple-ply belts. They consist of two or three plies, each serving a special purpose. Leather belts made from hide and Balata belts are no longer used. In some cases, the improvements in power transmission capacity brought about by the development of modern high-performance flat belts are not fully utilized in standard belt-drive applications. The utilization of a drive unit is influenced by the design characteristics on which a flat-belt drive is based. Thus, it is advisable to always consider the whole flat-belt drive unit instead of just the flat belt.

The major components of a multiple-ply belt are the tension ply and the friction ply. The purpose of the tension ply is to absorb the forces resulting from the deformation of the belt by tensioning. The energy stored by tensioning of the belt at minimum elongation is the basis of the power transmission. In addition, the tension ply has to absorb the centrifugal forces acting on the belt during operation.

Since the materials used for the tension ply do not have the required frictional characteristics, a separate, laminated friction ply is used as the second layer. This friction ply, which is adapted to the operating conditions with regard to material and surface finish, transmits the friction forces from the pulley surface finish to the tension ply and vice versa (Fig. 31.15). The tension ply is usually made of highly drawn

TABLE 31.1 Service Factor c_B

Application: Operating conditions	Application: Examples	Prime movers: Alternating-current and three-phase motors with a low starting torque (up to 1.5 times nominal torque); dc shunt motors; internal combustion engines with eight or more cylinders	Prime movers: Alternating-current and three-phase motors with moderate starting torque (1.5 to 2.5 times nominal torque); internal combustion engines with six cylinders	Prime movers: Alternating-current and three-phase motors with high starting torque (above 2.5 times nominal torque); internal combustion engines with four cylinders or fewer
Continuous service, small accelerated masses	Liquid-stirring apparatus, agitators, calenders and drying equipment for paper manufacture, setters, slitters and folders, centrifugal pumps and compressors, fans up to 7.5 kW, light-duty woodworking machinery, sifting plants	1.2	1.4	1.6
Interrupted service without bumps, medium-sized accelerated masses	Agitators and mixers for semifluid media, machine tools (such as grinding, turning, drilling and milling machines), punches, embossing machines, presses, textile machinery, laundry machinery, fans above 7.5 kW, generators and exciters, rotary presses, vibrating screens	1.3	1.5	1.7
Interrupted service with bumps, medium-sized accelerated masses	Elevators and worm conveyors, centrifuges, paper manufacturing machinery such as grinding gear, pumps, shredders, beaters, piston pumps and compressors, blowers, high-power fans	1.5	1.7	1.8
Service with severe bumps, large accelerated masses	Crushers and rolling mills, ball mills, tile-molding machines, compressors and high-capacity pumps, hoists	1.6	1.8	1.9

†The service factor c_B takes into account the type of prime movers and driven machines. Special operating conditions are not taken into account in these values. The factors stated are guide values.

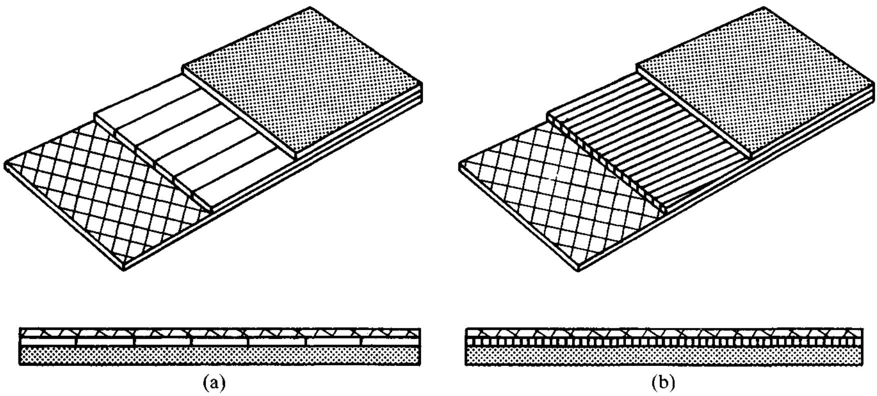

FIGURE 31.15 Multiple-ply belt.

polyamide strips or polyester cord. The friction ply, firmly attached to the tension ply, is made of either synthetic rubber or polyurethane or chrome leather. Table 31.2 shows the most important physical data for high-performance flat belts of the most commonly used tension ply materials, polyamide and polyester. The belts are manufactured in endless form according to the user length requirements or are made endless by heat-cementing the two beveled, feather-edged ends. Table 31.3 shows sizes of pulleys for flat belt drives and tolerances.

Calculations for a high-performance belt drive are usually based on data supplied by the belt manufacturer. Since the latest developments are always taken into consideration in this information, use of the latest manufacturers' data for the calculation is mandatory.

TABLE 31.2 Physical Data of High-Performance Flat Belts

Notion	Unit	Tension ply	
		Polyamid	Polyestercord
Tensile strength	N/mm^2	450–600	700–900
	N/cm	1300–18 000	1300–6600
Elongation at rupture	%	~22	~12–15
Stress at 1% elongation	N/cm	30–400	100–400
Service elongation	%	1.5–3.0	1.0–1.5
Specific nominal peripherical force	N/cm	40–800	100–400
Specific nominal power P_N	kW/cm	$\leq$45	$\leq$60
Maximum belt velocity	m/s	60–80	80–150
Maximum tolerable bending frequency	1/s	80–100	100–250
Elongation slip at nominal peripherical force	%	~0.8–1.0	~0.4–0.6
Attenuation (logarithmic decrement) ϑ		~0.28	~0.25
Efficiency η		0.98–0.99	0.985–0.99
Total thickness a	mm	1.0–8.0	0.8–4.0
Belt width	mm	Max. 1000	Max. 450
Belt length	mm	Unlimited	Max. 12 000

TABLE 31.3 Size of Pulleys for Flat-Belt Drives

Diameter d_1 nominal size	40	50	63	71	80	90	100	112	125
Allowable off size	0.5	0.6	0.8	1	1	1.2	1.2	1.2	1.6
Height of convexity	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.4
Tolerance of concentricity	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.3	0.3
Diameter d_1 nominal size	140	160	180	200	224	250	280	315	355
Allowable off size	1.6	2	2	2	2.5	2.5	3.2	3.2	3.2
Height of convexity	0.4	0.5	0.5	0.6	0.6	0.8	0.8	1	1
Tolerance of concentricity	0.3	0.3	0.4	0.4	0.4	0.4	0.5	0.5	0.5

In addition to the applicable ratings of the various high-performance flat belts, their types, and their configurations, the following data are necessary for the calculations for a single-step flat-belt drive:

1. Type of prime mover (driving assembly), e.g., electric motor, combustion engine, water turbine, etc.; this is important for the determination of the corresponding service-correction factors.
2. Type of machine (driven assembly); this determines corresponding load factors, dependent on acceleration, forces of gravity, changing loads, etc.
3. Power to be transmitted P, in kilowatts.
4. Speed of driving pulley n_1, in revolutions per minute.
5. Diameter of the driving pulley d_1, in millimeters.
6. Speed of the driven pulley n_2, in revolutions per minute.
7. Diameter of the driven pulley d_2, in millimeters.
8. Center distance e, in millimeters.
9. Adjustment range available (of tensioning device).
10. Allowable radial shaft loads of prime mover and driven assembly, loads on which the maximum shaft tensioning force F_w depends.

With the use of the above data and the manufacturer's data, calculations for the flat-belt drive can be made, giving the designer the type of belt, belt width, dynamic and static shaft stresses, and the required elongation, expressed as the percentage of belt strain.

Because of the special characteristics of flat-belt drives with high-performance belts, the determination of the drive data should be based on the following:

1. The belt velocity should be as high as possible (v_{opt}). The higher the belt speed, the smaller the belt width and thus the shaft load.
2. In calculating drives with changing loads or cyclic variations, you should determine to what extent the damping properties of the tension ply materials can be utilized.
3. You have to examine whether the initial tension of the belt or the shaft load can be accurately calculated from belt strain data.

4. The manufacturer can supply belts of all widths and lengths; you should examine, however, whether there are restrictions from the design point of view.
5. Finally, you must examine whether the belt can be manufactured in endless form or has to be assembled open-ended, with the ends being closed by welding after assembly.

The following general guidelines apply to pulley design:

1. The pulleys for open and crossed flat-belt drives are crowned in accordance with ISO R 100 (Table 31.3) in order to align the belt, which tends to move toward the larger pulley diameter.
2. For speed ratios higher than 1/3 ($i > 3$), the smaller pulley may be cylindrical. Spatial belt drives are equipped with cylindrical pulleys.
3. The requirements for smooth running of the belt are as follows: Parallelism of both shafts, smooth pulley faces, static balancing up to belt velocities v of 25 meters per second (m/s), and dynamic balancing for velocities above 25 m/s. When certain aluminum alloys are used, abrasion may occur, reducing friction between belt and pulley to such a degree as to make power transmission impossible.

Flat-belt drives are nonpositive flexible-connector drives used for the transmission of forces and motions between two or more shafts, particularly at greater center distances. This type of drive is superior because of its elasticity, enabling it to absorb shock loads, and its low-noise running. Its disadvantages are the greater forces acting on the shafts and bearings, resulting from the required initial tension, and the unavoidable belt slip.

These properties are decisive for the preferred applications of flat-belt drives, e.g., in machine tools, textile machinery, mixers and grinders, paper machines, gang saws, wire-drawing machines, presses, punches, and compressors. Flat belts with suitable contours may also be used as conveyor belts.

Figure 31.16 shows the drive of a hobbing machine. A high-performance flat belt was used in this case not because of its efficiency or damping properties, but because of the uniformity of rotational transmission from one pulley to the other. Preliminary studies have shown that even slight transmission deviations affect the dimensional accuracy of the tools manufactured on such machines. Belts for this application are subjected to a transmission accuracy test on a special test stand before delivery.

Figure 31.17 shows the tangential belt drive of a textile machine. This drive of a ring spinning frame is typical of a so-called multipoint drive or, in particular, a tangential belt drive. In this machine, a high-performance flat belt of 35-mm width and approximately 82-m length drives a total of 500 spindles on each side of the machine. The total power of 25 to 30 kW per machine side is thus distributed to 500 separate work positions. Depending on the spindle speed, the belt velocity ranges from 25 to 45 m/s. The absolute constancy of the belt operating tension throughout the life of the drive is a necessary prerequisite for this type of application.

31.3 V-BELT DRIVE

V-belt drives are nonpositive drives. The peripheral force F_u is transmitted by frictional forces acting on the flanks of the pulley-and-belt combination (Fig. 31.18). Bottoming of the belt in the groove leads to a reduction of the transmissible peripheral force, to belt slip, and to damages owing to overheating.

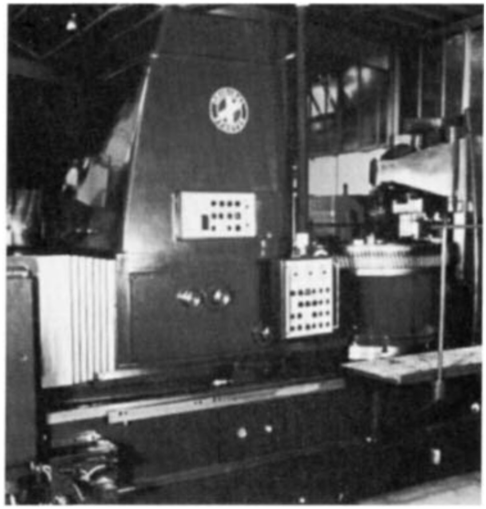

FIGURE 31.16 Drive of a hobbing machine. *(Siegling.)*

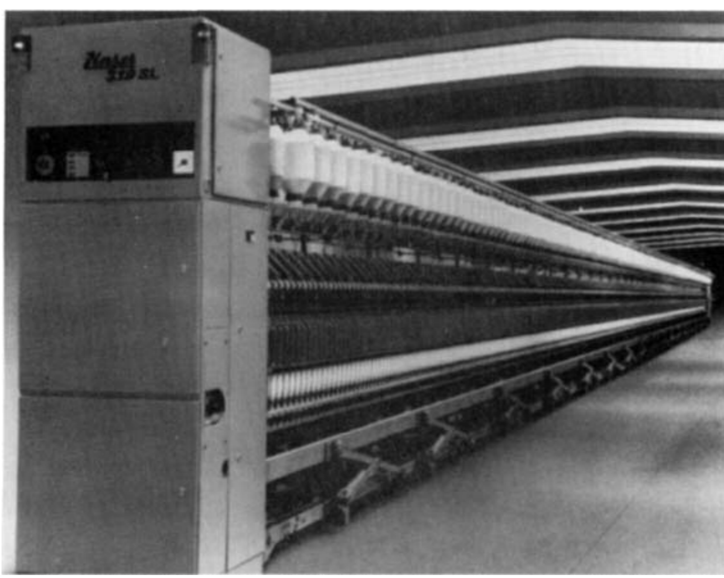

FIGURE 31.17 Tangential belt drive of a textile machine. *(Siegling.)*

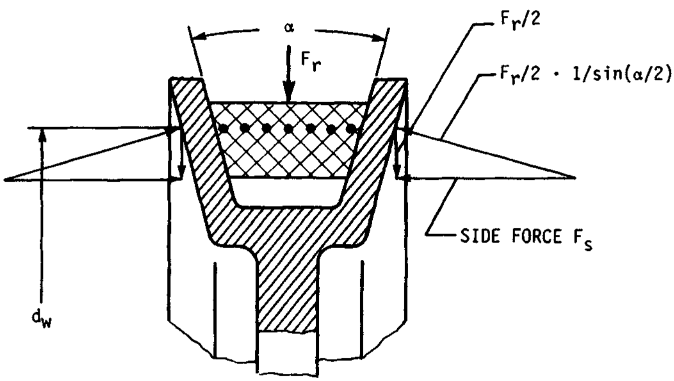

FIGURE 31.18 Section of a V-belt and sheave.

The known form of Eytelwein's equation cannot be used for the calculations for V-belt drives, since the belt creeps in the direction of travel and is simultaneously pulled radially into the groove by the radial component of the belt side force, with friction being reduced by this. Hence

$$\frac{F_1'}{F_2'} > \exp\left[\frac{\mu\beta\pi}{(\sin\alpha/2)180}\right] \tag{31.39}$$

The arc of contact β of the smaller pulley has to be used in the calculation of a V-belt drive.

In general, a belt angle α on the order of 36° is selected, since $\alpha < 20°$ would cause self-locking. Such a belt would operate with a lot of jerking and little efficiency. By bending around the pulley during operation, the belt is stretched on the outside and compressed on the inside, the belt angle thus being reduced compared to that of the straight belt. The smaller the pulley diameter, the larger the reduction of the belt angle. Since snug fit between belt and groove sides has to be ensured, the groove angle must be adjusted accordingly. Incorrect groove angles will reduce power transmission capacity and belt life.

The calculations for V-belt drives are internationally standardized (ISO R 155). In general, manufacturers' supply data for the calculations for V-belts and other belt types, too, are as follows:

Speed ratio:

$$i = \frac{n_1}{n_2} = \frac{d_{w2}}{d_{w1}} \tag{31.40}$$

For pulley diameters, standard series have been specified. When these pulley diameters are used, standardized speed ratios will result. Pulley diameters below the minimum values recommended for the belt section in question should not be used because the higher bending stress materially reduces belt life.

Distance between shaft centers:

$$\begin{aligned} &\text{Recommended lower limit: } e \geq 0.7(d_2 + d_1) \quad \text{mm} \\ &\text{Recommended upper limit: } e \leq 2(d_2 + d_1) \quad \text{mm} \end{aligned} \tag{31.41}$$

Shaft center distances that are too short (short belts) result in high bending frequencies, causing excessive heating and thus premature failure of the belt. Shaft center distances that are too long (long belts) may result in belt vibrations, especially of the slack side, also causing higher belt stress.

Adjustment of shaft center distance (Fig. 31.19):

$$X \geq 0.03 L_w \quad \text{mm}$$
$$X \geq 0.015 L_w \quad \text{mm} \tag{31.42}$$

The importance of the adjustability of the shaft center distance for tensioning and retensioning of the belt (X) as well as for easy application of the belt (Y) is often underestimated. The quantity Y in particular is often neglected, resulting in application problems, and the use of tools may cause belt damage when the belt is first applied.

The transmissible power of the belt is given by

$$P = 2\pi n M \tag{31.43}$$

where M = known torque and n = speed of the corresponding pulley.

Taking into consideration a service-correction factor c_B (Table 31.1), we can calculate the power rating of the drive from P by

$$P = c_B P_1 \tag{31.44}$$

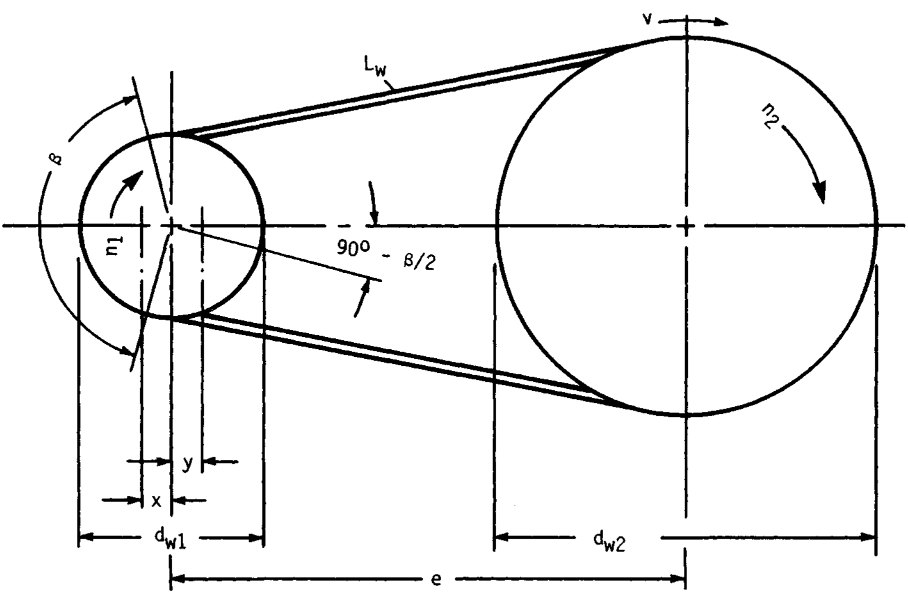

FIGURE 31.19 Adjustment of shaft center distance.

The calculation of belt velocity (peripheral velocity) is not really necessary, as the power rating tables are based on pulley diameter and speed:

$$v = \pi d_1 n_k \approx \pi d_2 n_g \tag{31.45}$$

For multiple drives, the number of V-belts required is calculated from

$$z = \frac{P c_B}{P_N c_1 c_3} \tag{31.46}$$

The arc of contact factor c_1, the service factor c_B, and the belt length correction factor c_3 can be found in the manuals of the manufacturers. The power rating P_N of each belt—based on the selected pitch diameter d_{w1} of the smaller pulley, the corresponding speed n_1, and the speed ratio—can be derived from tables. These tables also contain the nominal values of the service-correction factors to be used. When the guidelines mentioned in the design of a V belt are observed, the belt life to be expected is 24 000 hours (h) of operation.

As for flat belts, the bending frequency is

$$f_b = \frac{vz}{l} \tag{31.47}$$

Normal bending frequencies are

$f_b < 30$ per second for endless standard V-belts

$f_b < 60$ per second for endless narrow-V-belts

The belts have to be pretensioned to limit belt slip to 1 percent. Improperly pretensioned belts have a life substantially shorter than the 24 000 h mentioned. The necessary initial tension will lead to equivalent shaft or bearing loads.

The approximate equation for the calculation of the required average shaft tension force is

$$F_w \cong \frac{aP}{v} \qquad 1.5 \leq a \leq 2 \tag{31.48}$$

where P = power to be transmitted and v = belt velocity. More accurate methods may be found in the manufacturers' publications containing a description of the correct initial tension adjustment by force and deflection measurements at the center of the belt side.

The different belt types are distinguished by section dimensions; the configurations, however, are distinguished by belt construction. Figure 31.20 shows the most widely used V-belt configurations.

V-belts consist of the following firmly vulcanized elements:

1. Intermediate ply of high-strength cord (cotton, polyester) for transmission of the rope-pull force
2. Highly elastic belt body (rubber, plastic) for transmission of the peripheral forces between the belt flanks and the cord ply

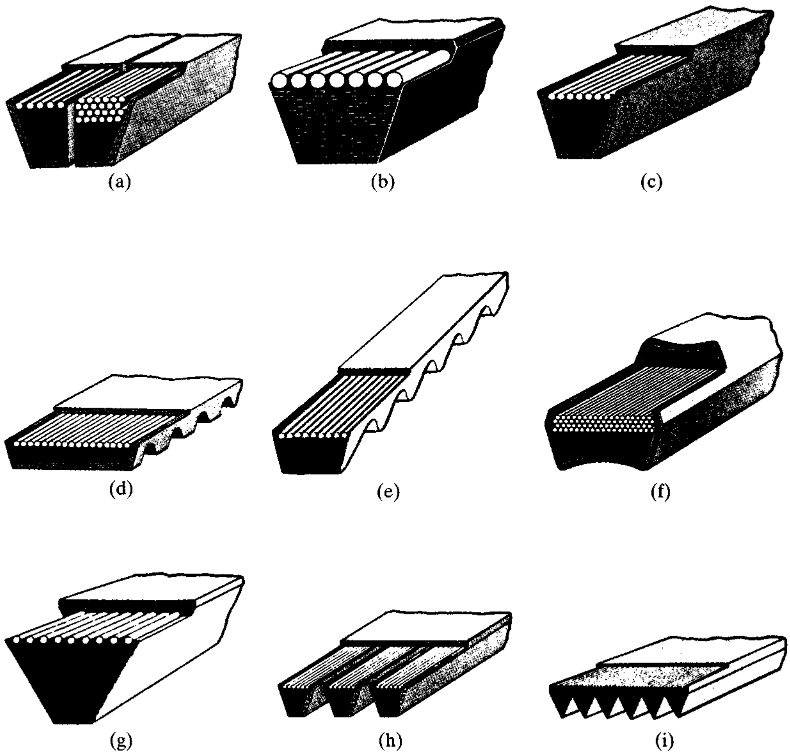

FIGURE 31.20 V-belt configurations.

3. Impregnated wear-resisting fabric coat (textile fibers) for transmission of the friction forces between the belt flanks and the V-belt pulley

Because of its higher modulus of elasticity, the cord ply forms both the neutral axis and the pitch diameter d_w of the V-belt in the groove, i.e., that diameter which is relevant for the transmission of peripheral velocity.

The endless standard V-belts (Fig. 31.20*a*) in accordance with ISO (ISO R 52, R253, R434 and R606) are supplied in specified lengths and cannot be shortened.

Endless narrow-V-belts (Fig. 31.20*c*) in accordance with ISO R 459 and R 460 are most widely used today. Their power transmission capacity is higher than that of standard V-belts of the same pitch width.

FO-type V-belts (Fig. 31.20*b*), made by the Continental Gummi-Werke AG, are distinguished by their basically different construction. They are a promising new development superior to others for the unusually small change of length their tension members undergo when loaded. Transverse short fibers in the rubber filler lead to high lateral stiffness with high flexibility in the direction of belt travel. By grinding the flanks, the traveling accuracy of the FO-type V-belt can be improved systematically. The Continental belts of this type are manufactured as endless standard V-belts of small section dimensions and as endless narrow-V-belts.

The flexibility of the V-belts mentioned thus far may be increased by cross grooves worked into the inner surface of the profile (Fig. 31.20*e*). These grooved belts allow smaller pulley diameters and require less space at slightly reduced power transmission capacity than other belts; however, the grooves are the cause of periodic running in of shock loads and noise.

A V-belt assembly (Fig. 31.20*h*) is composed of up to five standard or narrow-V-belts connected by an elastic cover band which does not rest on the pulleys. The cover band prevents twisting or excessive vibration of individual belts.

The grooves of the V-belt pulleys are standardized (ISO R 52, R 253 for standard V belts and ISO R 459 for narrow-V-belts). The pulleys are castings or weldings or steel-plate parts. In general, pulley diameters below the minimum specified should not be used. The groove flanks must be smooth and clean to ensure a sufficiently long belt life.

Applications of V-belts are practically unlimited. Because of the variety of sections available, the V-belt may be used for fractional-horsepower drives with minimum power transmission capacity in precision machines, phonographic equipment, and domestic appliances; for light drives, e.g., of centrifugal pumps and fans; and for all sizes of drives for general mechanical engineering purposes up to heavy-duty drives, such as rolling mill, rock crusher, excavator, and crane drives.

In general, the power transmission capacity is approximately 700 kW maximum; it may be increased to 1000 kW and in very rare cases to approximately 5000 kW.

Because of the extreme variety of applications in mechanical engineering, general belt-life data cannot be given. Another explanation is that belt life largely depends on the conditions of use. Assembly and operating conditions affect belt life, as do environmental conditions such as oil, dust, and climatic conditions.

The power transmission capacities specified in the relevant standards are based on the very high empiric belt life of 24 000 h, which can be reached only under optimum operating conditions, i.e., no misalignment of the pulleys, no overload, correct belt tension, normal ambient conditions, etc.

One of the particularly important areas of V-belt application is the automotive industry with its large primary equipment production series and substantial spare-part need. The V-belt as a drive unit connects the crankshaft and different accessory units, e.g., generator, cooling-water circulating pump, and fan. The V-belt may also be used for driving air-conditioning equipment or turbosuperchargers. The life of car V-belts is calculated for a total distance of 50 000 to 80 000 kilometers (km).

If higher power transmission capacities are required in mechanical engineering, multiple drives (Fig. 31.21) are used. When used as main drives in situations involving high cost in case of failure, such as mine fan drives and drives used in the metallurgical, glass, and cement industries, multiple drives also meet the demand for safety, since the probability of sudden failure of a complete belt set is very small. The uniformity of initial tensions and speed ratios of all belts of multiple drives is a prerequisite for smooth running and equal distribution of total power to all belts. Differences in speed ratios and initial tensions will lead to a reduction in belt life. Thus the complete set of belts should be replaced, if necessary, not just a single endless belt.

31.4 SYNCHRONOUS-BELT DRIVE

The synchronous belt is a relatively new machine element combining the advantages of positive and nonpositive flexible connectors. In contrast to the nonpositive belt drives, the pulleys and belts of synchronous-belt drives have meshing teeth, allowing synchronous power transmission with angular accuracy (Fig. 31.22). To prevent rid-

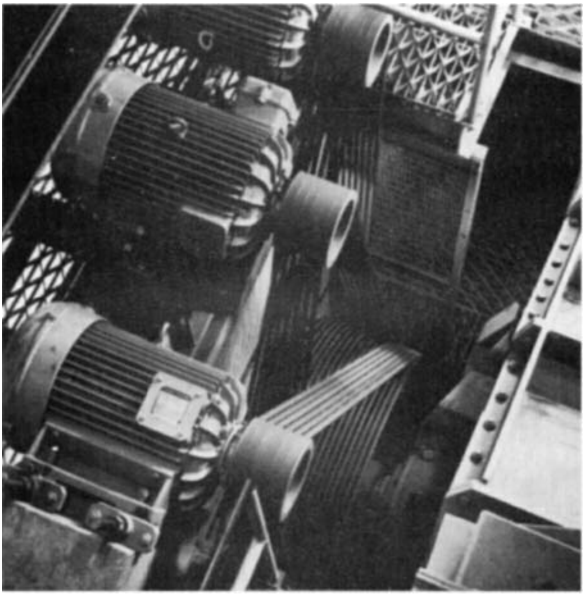

FIGURE 31.21 Multiple-V-belt drive of a crusher in mining. *(Arntz-Optibelt-KG.)*

ing off of the belt, at least one of the pulleys has flanges which are slightly beveled to reduce lateral friction.

Because of their special properties, synchronous belts are used wherever synchronous power transmission is asked for and the safety and freedom-from-maintenance requirements are strict.

As for all positive drives, the conformity of belt and pulley pitches is of utmost importance. The advancing belt tooth must correctly mesh with the corresponding pulley groove and remain there until leaving it. To achieve this, the pitch of the deflected belt must correspond exactly with that of the pulley. Hence, the belt must be made of high-strength material experiencing little length change under load; thus, the neutral axis may be assumed to be in the center. The pitch line is situated outside the pulley-tip-circle radius at a distance equaling that of the neutral axis (u value). This value is a largely invariable quantity which depends on the belt construction and must be taken into consideration in the design of the pulleys. Figure 31.22 shows the geometric relations between belt and pulley.

Despite the use of materials with little change of length under load for the tension member (fiber glass, steel cord), the belt pitch is a variable dependent on the strain properties of the belt and the applicable belt pull. Along the arcs of contact there is a step-by-step change in the belt pull with a corresponding change in local belt pitch, so that, in compensating belt and pulley pitch deviation, the teeth are deformed differently and the load differs accordingly. In addition, there is a minor pitch difference between belt and pulley, which is attributable to the production process and has to be compensated by deformation of the belt teeth, too.

The load distribution along the arc of contact as a measure of peripheral force distribution to the individual teeth is thus mainly dependent on the pitch deviation. The pitch deviation within the pretensioned belt (idling) is of particular importance:

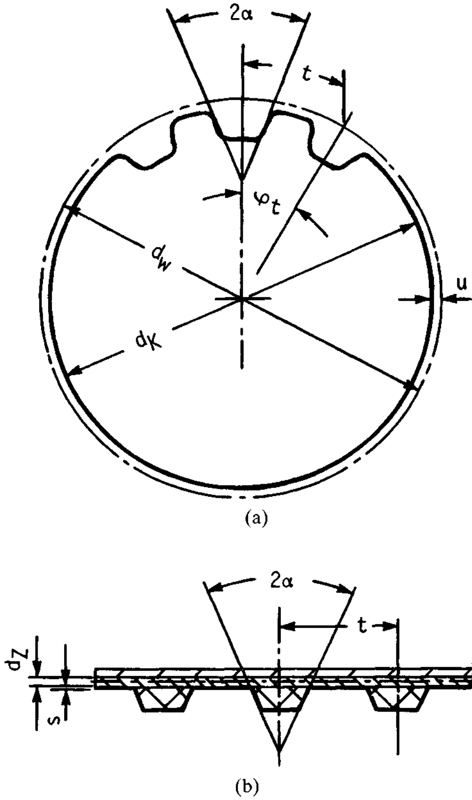

FIGURE 31.22 Synchronous-belt drive. (*a*) Pulley dimensions are d_w = working diameter, d_k = peripheral diameter, t = pitch, φ_t = pitch angle, u = theoretical distance to working diameter, and 2α = angle of tooth; (*b*) belt dimensions are d_z = diameter of tensile member and s = thickness of cover.

- Pitches of the pretensioned belt and the pulleys are exactly the same ($\Delta t = 0$). Load distribution is symmetric. The teeth at the beginning and at the end of the arc of contact carry a higher load than those along the center of the arc.
- The pitch of the pretensioned belt is smaller than the pulley pitch ($\Delta t < 0$). Load distribution is unsymmetric. Power is mainly transmitted at the beginning of the arc of contact (driven pulley) or the end of the arc of contact (driving pulley).
- The pitch of the pretensioned belt is larger than the pulley pitch ($\Delta t > 0$). Load distribution is unsymmetric. Power is mainly transmitted at the end of the arc of contact (driven pulley) or the beginning of the arc of contact (driving pulley).

For the driven pulley Fig. 31.23 shows the pull varying as a function of load-distribution deviations from the beginning of the arc of contact ($n/z = 0$) to the end of the arc of contact ($n/z = 1$).

Synchronous belts are available in different dimensions (pitch, belt width, and tooth geometry) according to the required application and range of capacity. The

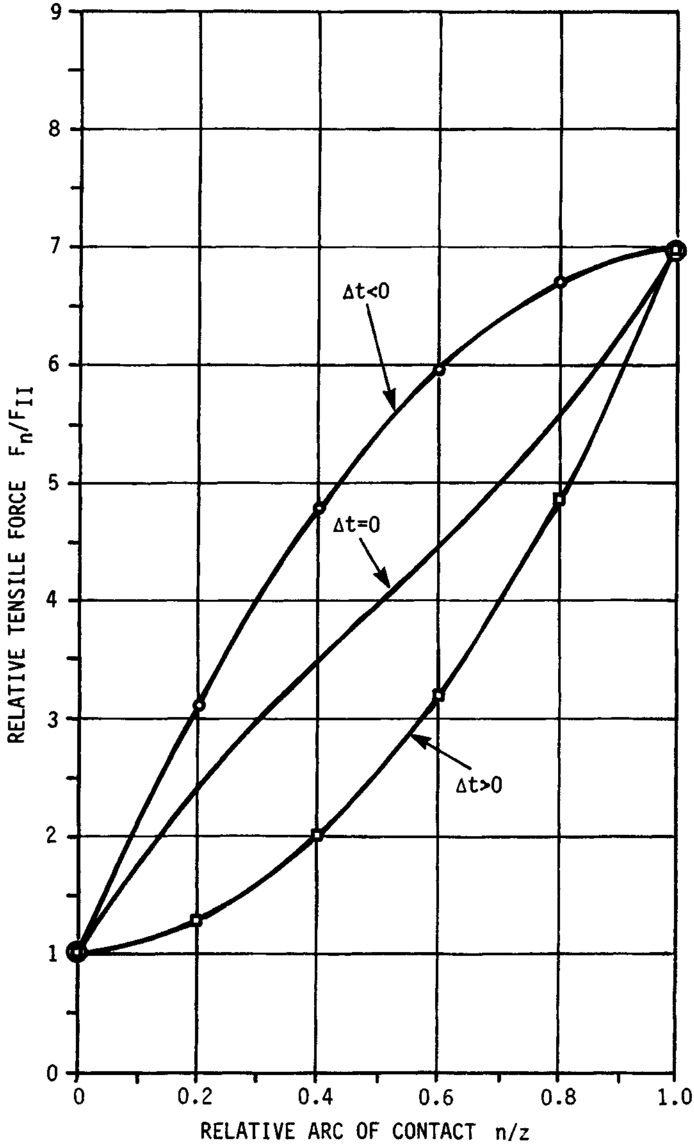

FIGURE 31.23 Theoretical tensile force diagram for the driven pulley.

transmissible power P is determined by the standard capacity P_N of the selected belt and by the operating conditions, expressed by the service-correction factor c_B:

$$P = \frac{P_N}{c_B} \tag{31.49}$$

The service-correction factors range from 1.0 to 2.10. The accurate value is determined by the type of drive and the type of application as well as the daily period of operation. For special applications, further restrictions may be required; details (meshing factor, acceleration factor, etc.) are contained in the manufacturers' catalogs.

The ranges of power transmission capacity are fixed in accordance with belt pitch. According to ISO 5296, standard belts have the standard pitch values shown in Table 31.4.

The capacities listed in Table 31.4 are nominal values for medium belt velocities; they may vary considerably to the positive or negative depending on the operating conditions. Accurate calculation has to be based on manufacturers' data.

When making calculations for a synchronous-belt drive, you must take into account that the pitch may be changed only in integral numbers, in contrast to non-positive belt drives. The individual design characteristics are as follows:

Speed ratio:

$$i = \frac{z_1}{z_2} = \frac{d_2}{d_1} \tag{31.50}$$

where z_2 = number of teeth of larger pulley
z_1 = number of teeth of smaller pulley
d_2 = pitch diameter of larger pulley
d_1 = pitch diameter of smaller pulley

Datum length:

$$l = 2e \sin\frac{d}{2} + \frac{t}{2}\left[z_1 + z_2 + \frac{\beta}{90^\circ}(z_2 - z_1)\right] \tag{31.51}$$

Approximation:

$$l \approx 2e + \frac{t}{2}(z_1 + z_2) + \left(\frac{t}{2\pi}\right)^2 \frac{(z_2 - z_1)^2}{l} \tag{31.52}$$

where z_1 = number of teeth of smaller pulley
z_2 = number of teeth of larger pulley
t = pitch, in
l = belt datum length = $z_R t$
z_R = number of belt teeth

TABLE 31.4 Standard Pitch Values

	Extra light (XL)	Light (L)	Heavy (H)	Extra heavy (XH)	Double extra heavy (XXH)
Belt pitch, in	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{4}$
Nominal power, kW	0.15	1.0	10	40	100

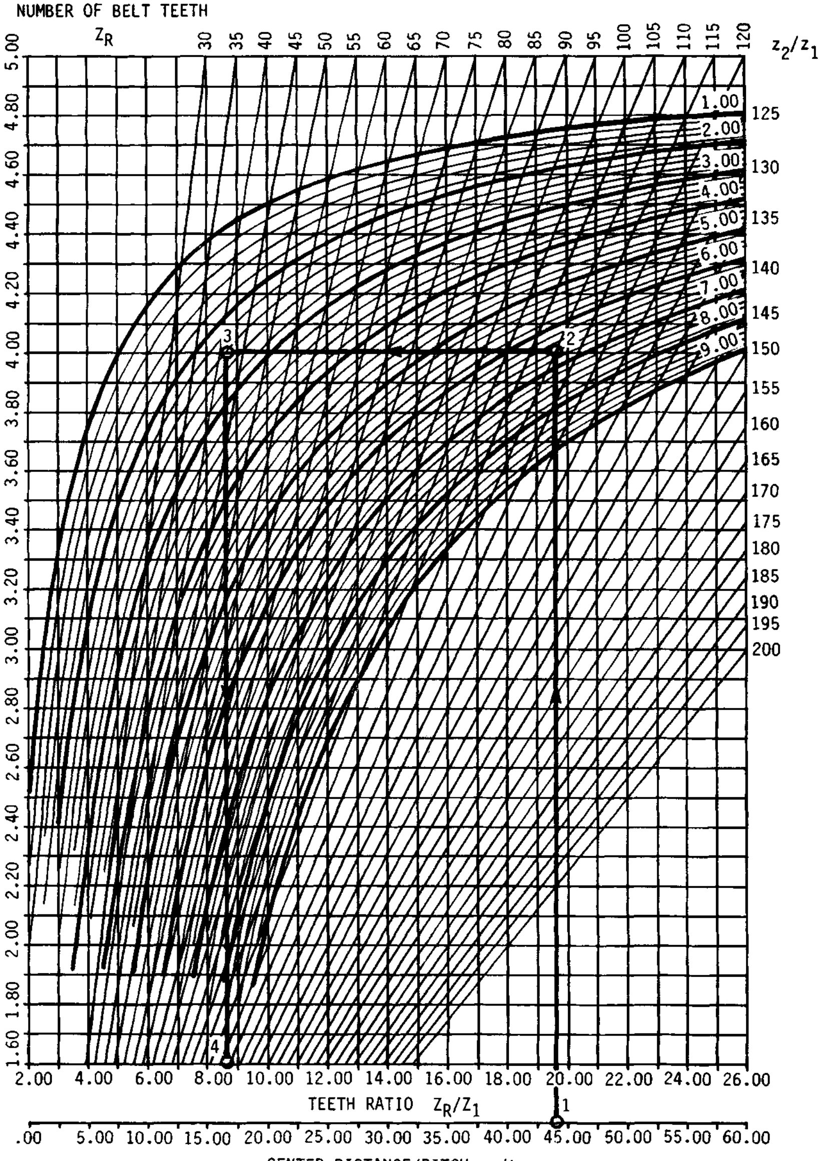

FIGURE 31.24 Determination of center distance of synchronous belts.

The accurate number of pulley and belt teeth for a given center distance may be found from detailed tables included in the manufacturers' catalogs. Guideline values may also be determined by using the nomogram in Fig. 31.24. Enter the nomogram at the ratio of given center distance to pitch (point 1), and proceed vertically to intersection 2 with the straight line indicating the number of teeth of the belt intended for use; then proceed horizontally from this point to intersection 3 with the curve of the given speed ratio. The number of teeth of the smaller pulley can be calculated from the applicable value on the axis of the tooth ratio of the belt and the smaller pulley (point 4), and the number of teeth of the larger pulley from the speed ratio; the results are rounded to the next higher whole numbers.

When calculating the number of teeth or the geometric design data, the designer has to ensure that the number of meshing teeth is at least six to eight. For the transmission of larger peripheral forces, the minimum number of meshing teeth has to be increased in accordance with manufacturers' data.

The transmissible peripheral force F_u is closely related to the effective shaft tensioning force F_w, which is dependent on the initial tensioning force F_A and the operating conditions. As in chain drives, the effective shaft tensioning force F_w increases with increasing load moment. As a guideline, assume the initial tensioning force F_A to be in the range

$$F_u \leq F_A \leq 1.5F_u \tag{31.53}$$

and the belt side-force ratio to be $F_I/F_{II} \cong 5$, where F_I is the tight-side force and F_{II} is the slack-side force.

The permissible initial tensioning force F_A is determined by the construction and the dimensions of the selected belt. Particular attention has to be paid to the fact that the belt should not be strained to such a degree as to prevent proper functioning of the belt-and-pulley combination because of the resultant pitch deviation.

In addition to the standard synchronous belts, according to ISO 5296, other types of synchronous belts with metric pitch but changed tooth configuration or shape are used for special applications. Figure 31.25 shows the different constructions of common synchronous belts.

The teeth of the standard synchronous belt (Fig. 31.25*a*) are trapezoidal. The belt is made of highly elastic synthetic rubber (polychloroprene) with an intermediate ply (tension member) of high-strength fiber-glass cord with little change of length under load. The pulley side of the belt is made of abrasion-resistant polyamide fabric. The back of the belt is ground to ensure nonpositive driving of belt tighteners or additional pulleys.

The polyurethane synchronous belt, shown in Fig. 31.25*b*, has a similar tooth shape. This belt is made of a highly elastic and abrasion-resistant plastic (polyurethane) and needs no pulley-side fabric cover to reduce wear. The intermediate ply taking the tension consists of steel cord, and so the belt strain is reduced to a minimum. The belt pitch is metric. Common pitch values are 5 mm for belt type T5, 10 mm for T10, and 20 mm for T20.

Heavy-duty belts are innovations with a tooth shape designed to ensure optimum distribution of the flank pressure over the entire tooth surface. The belt teeth have an enlarged cross-sectional area for the reduction of transverse (shear) stress, their edges being chamfered. Trade names differ among manufacturers (HTD, Super-Torque, etc.). Neither the tooth shape nor the pitch is standardized and both may vary with the product, although the pitch is usually metric. Belt materials and construction are similar to those of standard belts. Common pitch values are 8, 14, and 22 mm.

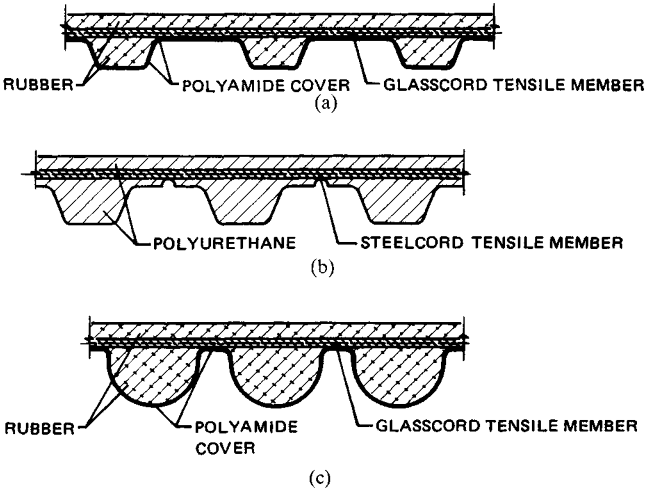

FIGURE 31.25 Construction and differences of timing belts. (*a*) Polychloroprene (rubber); (*b*) polyurethane; (*c*) heavy-duty.

In addition to the belt types mentioned above, there are special belts with pitches or tooth shapes adapted to particular applications. There exist also synchronous belts with teeth on the back to provide for a two-side running capability and others with special back configurations, e.g., for transportation purposes. Geometric data for these special belts may be found in the catalogs of the respective manufacturers. Figure 31.26 shows an assortment.

Because of their ability to transmit power synchronously with the required angular accuracy, synchronous belts are used to an ever-increasing extent, with new modes of application being added constantly. Classic applications are as follows:

- Light-duty conveyors (office machinery, food industry)
- Positioning drives (peripheral equipment of electronic data processing systems, machine tools, screening machinery)
- Synchronizing gears [camshaft drives (OHC), textile machinery, paper machinery]

Figures 31.27 to 31.29 show some typical synchronous-belt applications.

To ensure trouble-free operation and a long life of the drive, the following recommendations should be observed:

- The belt should be pretensioned only to the degree necessary to prevent skipping of the belt on starting or braking. Excessive initial tension leads to reduced belt life and in some cases to extreme noise levels.
- To prevent tension member damage (e.g., kinking or breaking), the belt has to be placed on the pulleys without tilting them. To facilitate this, an axial-shifting device should be provided on one of the pulleys.
- To prevent excessive wear of belt sides and noise from lateral contact of belt and pulley flanges, shafts and pulleys have to be aligned with extreme care. Slight lateral contact of belt and pulley flanges is attributable to the manufacturing process

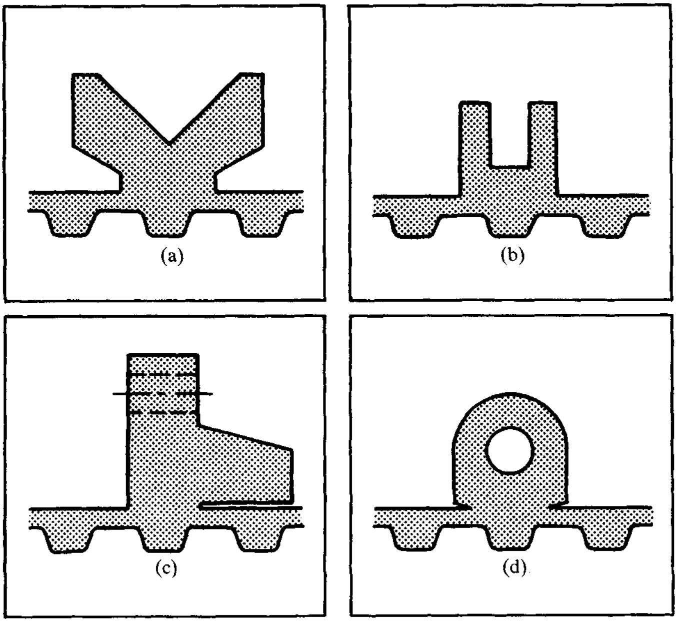

FIGURE 31.26 Synchronous belts with special back configuration.

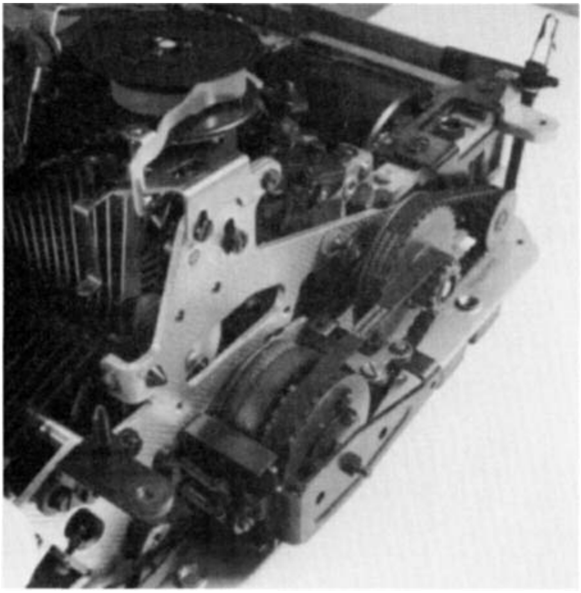

FIGURE 31.27 Synchrobelt timing belts for sophisticated typewriter drive mechanism. *(Continental Gummi-Werke Aktiengesellschaft.)*

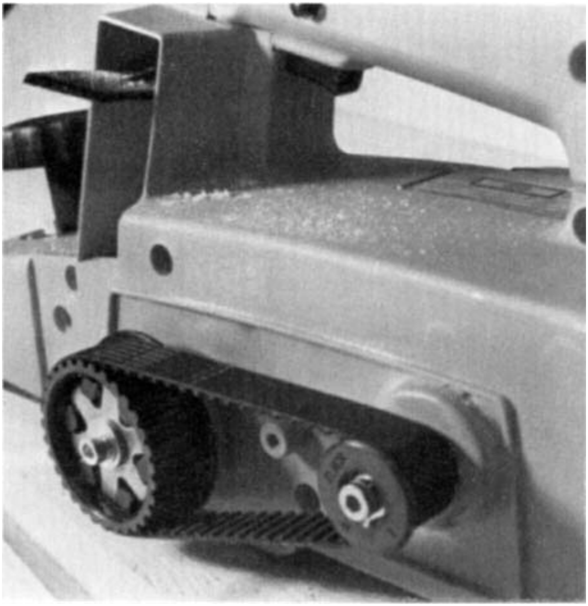

FIGURE 31.28 Planer machine drive with Synchrobelt timing belt; the driving speed is 30 000 min^{-1}. *(Continental Gummi-Werke Aktiengesellschaft.)*

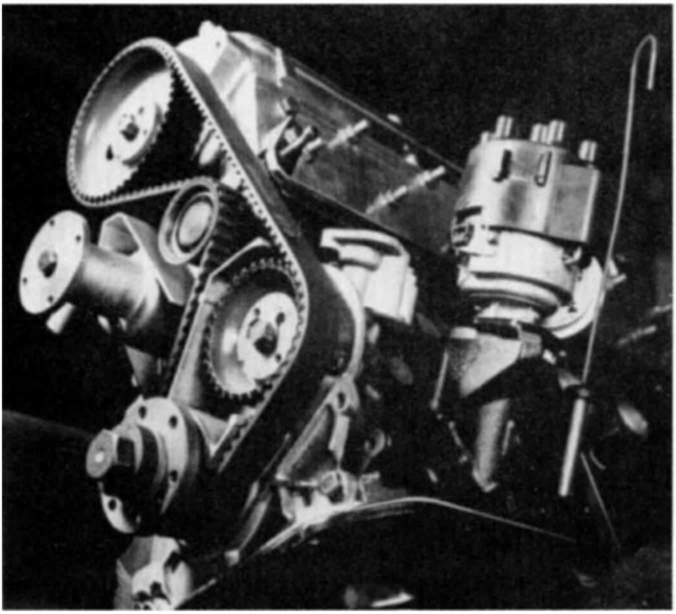

FIGURE 31.29 Camshaft control with Synchrobelt HTD timing belt. *(Continental Gummi-Werke Aktiengesellschaft.)*

(twisting of tension member and its slight inclination) and does not affect the running characteristics or the life of the belt.

- If feasible, belt tighteners should be arranged to act on the inner surface of the slack side of the belt; they should be toothed, too. The arc of contact should be as small as possible.

31.5 OTHER BELT DRIVES

31.5.1 Special Types

Poly-V-belts are similar to V-belt assemblies; however, their cover band rests on the pulleys, contains the cord, and takes the tension, whereas the individual V-sections, called *ribs*, solely transmit the peripheral force between pulley and cover band. The ribs have no cords (Fig. 31.20*i*) and fill up the pulley grooves almost to the bottom. Thus they are susceptible to foreign matter entering between the belt and the pulley. Because of the relatively rigid cover band resting on the face of the pulley, particularly accurate alignment of the pulleys is required, as it is with flat-belt drives. The belts are remarkably quiet.

V-belts with a belt-groove angle of 60° are cast belts (Fig. 31.20*g*) made of polyurethane with polyester tension members. Because of the higher friction coefficient of polyurethane on steel, the large included angle of 60° is required to prevent self-locking. Because of the manufacturing process, these belts show a substantially greater dimensional accuracy, resulting in particularly quiet operation. They are designed for maximum velocities of up to 50 m/s and are used mainly in machine tools.

Hexagonal (double-V) belts (Fig. 31.20*f*) have a symmetric special profile. The ratio of the maximum width to the height of the section is approximately 1.25. Double-V-belts may be operated in one plane on counterrotating pulleys. The power transmission capacity is approximately equal to that of conventional V-belts of equal width with small pulley diameters. Minimum pulley diameters are also about equal to those of conventional V-belts. Double-V-belts are suited for multiple-shaft drives operated in one plane with counterrotating pulleys. They are especially suited for coupled operation, with a tensioning or coupling shaft riding on the back of the belt and being able to transmit power, too.

Double-V-belts are used for medium-duty drives (combine harvester) as well as for light-duty tools (gardening equipment, rotary sweepers).

Toothed V-belts, which must not be confused with synchronous belts, are available as standard, narrow-, and wide-V-belts with punched or preformed teeth in the belt carcass to increase flexibility. Thus, with only a slight reduction in power transmission capacity, the minimum permissible diameter of the smaller pulley may almost be halved, so that the space requirement of such a belt drive with equal ratio or speed range is reduced substantially. These toothed V-belts are subject to the effects produced by their polygonal shape; i.e., the teeth cause running in of shock loads (which may result in irregular transmission of motion), additional dynamic stress of bearings, and even (in the case of high peripheral velocities) noise.

Round belts are versatile, simple, and reliable connectors for the transmission of small torques and medium velocities. Their main advantage is polydirectional flexibility. There are two different types: homogeneous round belts made of one material (rubber, plastic) and round belts with tension member (Fig. 31.30).

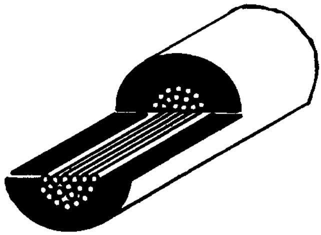

FIGURE 31.30 Continental round section belt.

31.5.2 Variable-Speed Belt Drives

For variable-speed belt drives (with a continuously variable speed ratio), especially wide-V-belts have been developed in toothed and nontoothed configurations. The construction corresponds to that of endless standard V-belts with the exception of the substantially greater belt width. This great belt width (the ratio of the upper width of the belt to the section height $b/h = 2/5$) is required for radial shifting on pulleys, the halves of which can be moved in an axial direction (Fig. 31.20*d*). These variable-speed belt drives allow for a continuously variable speed ratio within the range of $i_{max}/i_{min} = 4/10$. With larger ratios of belt width to section height, greater speed ranges are possible in general.

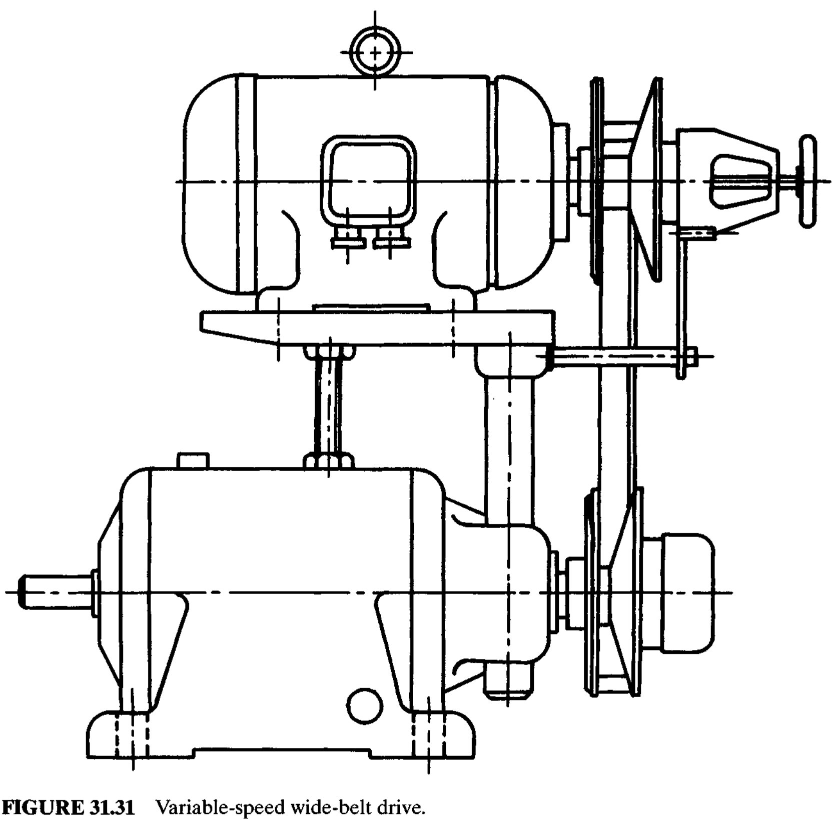

FIGURE 31.31 Variable-speed wide-belt drive.

The power transmission capacity of variable-speed belts is approximately equal to that of standard V-belts of equal section height; with increasing *b*/*h* ratio, however, it decreases by up to 20 percent. Depending on the speed range and the design of the variable-speed pulleys, the included angles range from 22° to 34° (ISO recommendation is 26°), with the smaller angles resulting in greater speed ranges but, because of the approach to the self-locking limit, reduced rated outputs.

A variable-speed wide-belt drive combined with an electric motor and a gear train is shown in Fig. 31.31. The top right half of the pulley may be shifted by turning the handwheel. The bottom pulley is an adjusting pulley (Fig. 31.32) which adapts itself automatically to the respective speed ratio.

31.6 COMPARISON OF BELT DRIVES

A flexible connector for a particular application should be selected on the basis of the following considerations:

- *Performance:* The flexible-connector drive has to perform reliably for an adequate period under the given operating conditions (speeds, moments, space

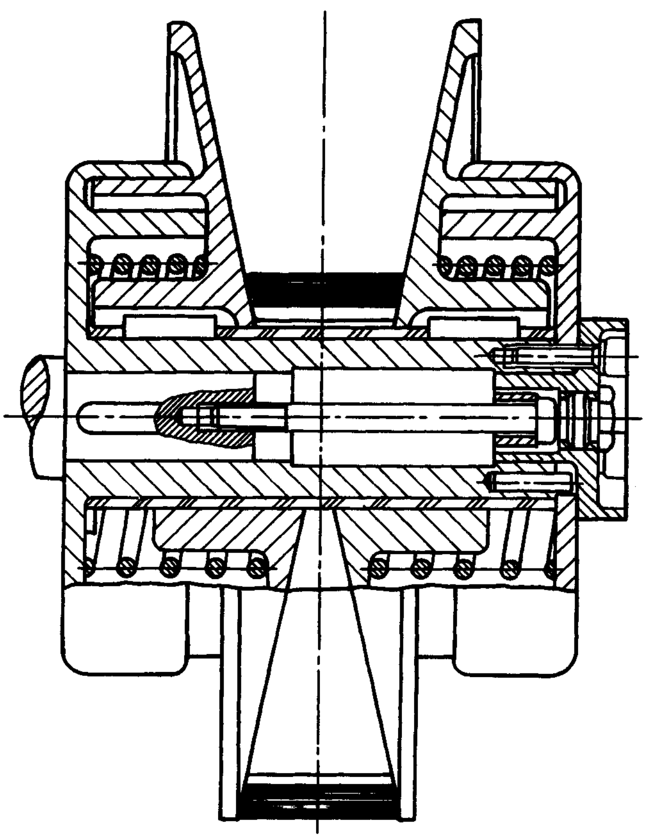

FIGURE 31.32 Adjusting pulley.

requirement, overloads, shaft dislocations, temperatures, and other environmental conditions).

- *Economy:* Those flexible-connector drives suitable under performance criteria are further investigated with respect to their applicability from the economic point of view. The economy of flexible drives is influenced by the following factors:

  1. Production cost of the flexible connector
  2. Subsequent design expenses required because of the selection made (e.g., pulley costs, costs of larger bearings to take up the required initial tensioning force, costs of lubrication facilities and seals required, of special installation arrangements, and of tensioning devices)
  3. Transportation and installation costs, mainly dependent on the space requirement and the weight of the drive
  4. Operation and maintenance costs
  5. Failure risk as well as costs of repair and subsequent costs

The relative importance of these factors varies with the machines used, and the costs of the flexible connector as the drive element are the least important. A drive unit will prevail wherever its special technical properties ensure the most economical drive.

To facilitate the selection of suitable flexible connectors, Table 31.5 compares the most important operating characteristics of flexible connectors and remarks on investment costs as well as maintenance and installation.

Figure 31.33 shows the results of a comparison of costs, when different flexible connectors are used, for a system consisting of two pulleys, the flexible connector, and the driven shaft, including the required roller bearings. The costs were determined following calculations for the drive with the power transmission capacity, the center distance, and the speed of the driving pulley given.

Figure 31.34 shows the specific power transmission capacity of different flexible-connector drives as a function of velocity. This specific power transmission capacity indicates the rating of the flexible connector and thus the space required. The higher the specific power transmission capacity of the flexible connector, the smaller the space required for the installation of the drive.

TABLE 31.5 Properties of Belt Drives

	Flat belts	Toothed belts†	Poly-V belts	Standard V belts	Narrow-V belts	Connected V belts†	Round belts†
Installation cost index	1.1	1.4	1.2	1.4	1.0	ND	ND
Maintenance	Yes	No	Yes	Yes	Yes	Yes	Yes
Power per volume, kW/cm^3	0.8	1.9	1.7	0.7	1.8	ND	ND
Maximum bending frequency, Hz	200	200	100	40	80	40	40
Shaft load	$2F_u$–$3F_u$	F_u	$2F_u$–$2.5F_u$	$2F_u$–$2.5F_u$	$2F_u$–$2.5F_u$	$2F_u$–$2.5F_u$	$2F_u$–$3F_u$
Efficiency, %	98	98	97	95	96	94	95
Diameter transmission	Constant	Constant	Constant		Variable		Constant
Bending rate d/s, dimensionless	15	15–30	5–11	8–14	8–12	8–14	8–10
Girder rate F_u/F_w, dimensionless	0.3–0.4	1	0.4–0.5	0.5–0.6	0.5–0.6	0.4–0.5	0.4
Admissible temperature, °C	−40 to +80	−50 to +120	−50 to +100	−55 to +70	−55 to +70	−55 to +70	−40 to +100
Adjustment of shaft center distance X, mm	$0.02L_w$	NA	$0.014L_w$	$0.02L_w$	$0.02L_w$	$0.02L_w$	$0.03L_w$
Mounting adjustment Y, mm	$0.01L_w$	$0.01L_w$	$0.02L_w$	$0.015L_w$	$0.015L_w$	$0.015L_w$	$0.01L_w$

†NA, not applicable; ND, no data.

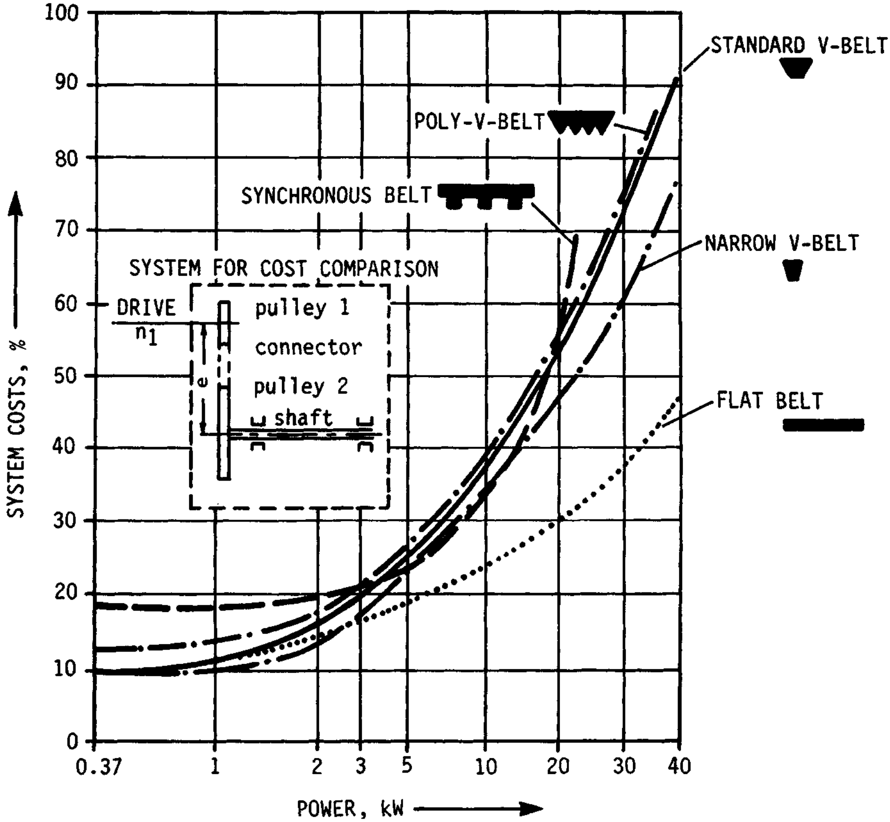

FIGURE 31.33 Comparison of system costs.

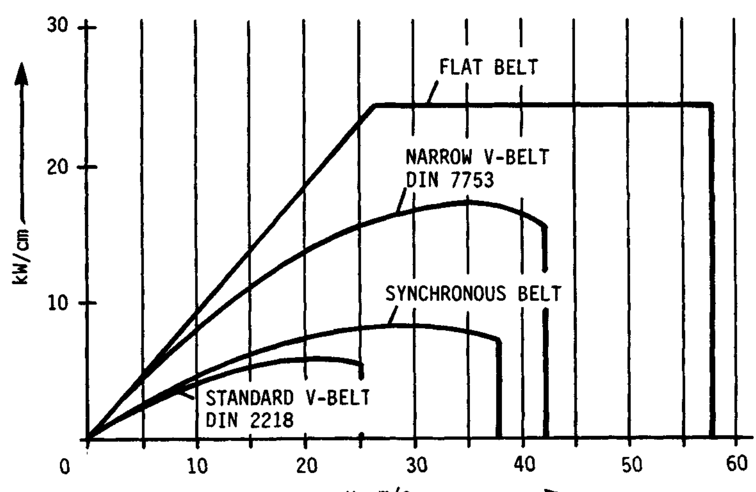

FIGURE 31.34 Specific power transmission capacity.

The figure also shows the range of applications of various flexible connectors and their maximum peripheral velocities. Belt drives reach their maximum specific power transmission capacity only at higher peripheral velocities.

It is not the maximum velocities, though, that determine the transmission of high specific powers, but the optimum velocities. The latter vary for the different flexible connectors because of the differing ratio of average density to the permissible tensile strain, as shown in Fig. 31.35. Accordingly, flat belts are particularly suited for maximum speeds and V-belts for most of the medium speeds of common electric motor and piston engines.

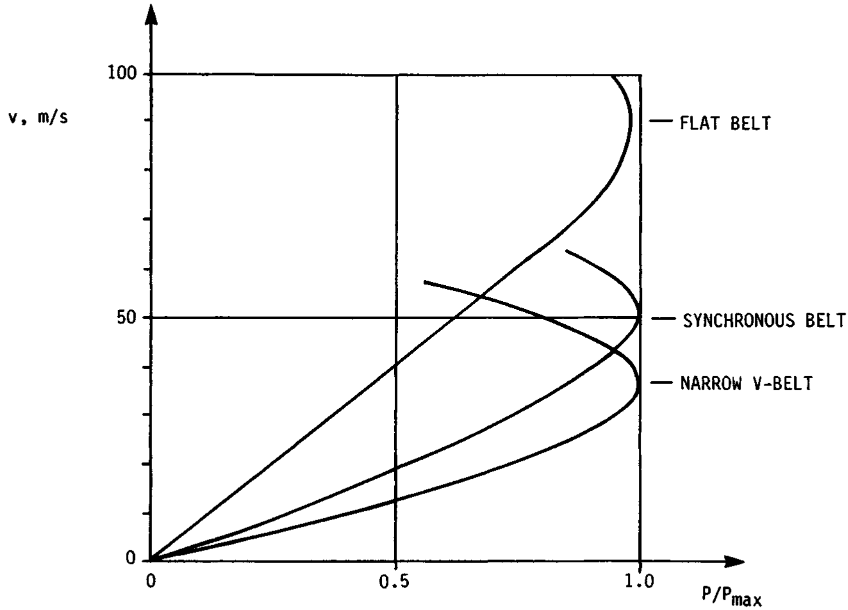

FIGURE 31.35 Optimum velocities.