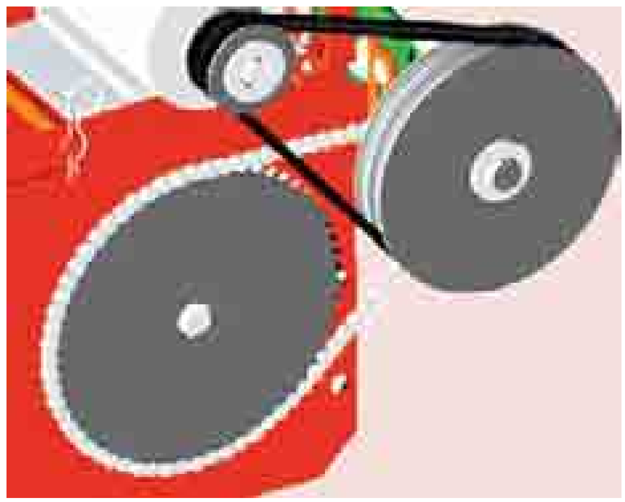
Contents

Dr. ALI KAMIL
4th CLASS
LECTURE No. 7

CHAPTER

20

V-Belt and Rope Drives



20.1 Introduction

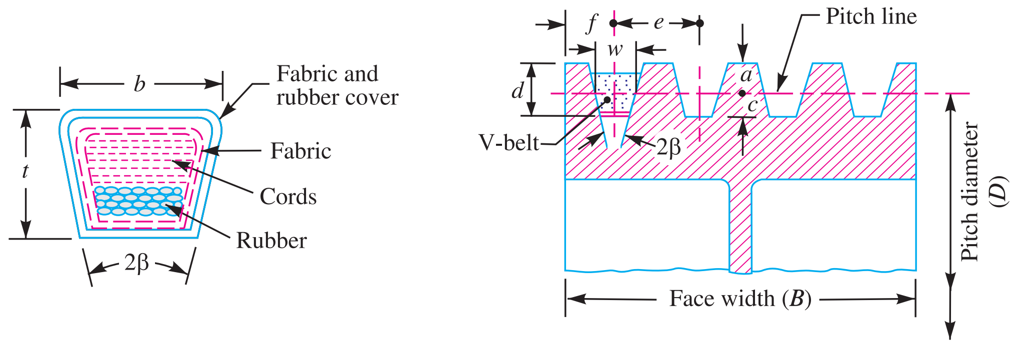
We have already discussed that a *V*-belt is mostly used in factories and workshops where a great amount of power is to be transmitted from one pulley to another when the two pulleys are very near to each other.

The *V*-belts are made of fabric and cords moulded in rubber and covered with fabric and rubber as shown in Fig. 20.1 (*a*). These belts are moulded to a trapezoidal shape and are made endless. These are particularly suitable for short drives. The included angle for the *V*-belt is usually from 30° to 40°. The power is transmitted by the *wedging

* The wedging action of the *V*-belt in the groove of the pulley results in higher forces of friction. A little consideration will show that the wedging action and the transmitted torque will be more if the groove angle of the pulley is small. But a small groove angle will require more force to pull the belt out of the groove which will result in loss of power and excessive belt wear due to friction and heat. Hence the selected groove angle is a compromise between the two. Usually the groove angles of 32° to 38° are used.

Top

action between the belt and the *V*-groove in the pulley or sheave. A clearance must be provided at the bottom of the groove as shown in Fig. 20.1 (*b*), in order to prevent touching of the bottom as it becomes narrower from wear. The *V*-belt drive may be inclined at any angle with tight side either at top or bottom. In order to increase the power output, several *V*-belts may be operated side by side. It may be noted that in multiple *V*-belt drive, all the belts should stretch at the same rate so that the load is equally divided between them. When one of the set of belts break, the entire set should be replaced at the same time. If only one belt is replaced, the new unworn and unstretched belt will be more tightly stretched and will move with different velocity.



(*a*) Cross-section of a V-belt. (*b*) Cross-section of a V-grooved pulley.

Fig. 20.1. V-Belt and V-grooved pulley.

20.2 Types of V-belts and Pulleys

According to Indian Standards (IS: 2494 – 1974), the *V*-belts are made in five types *i.e.* *A*, *B*, *C*, *D* and *E*. The dimensions for standard *V*-belts are shown in Table 20.1. The pulleys for *V*-belts may be made of cast iron or pressed steel in order to reduce weight. The dimensions for the standard *V*-grooved pulley according to IS: 2494 – 1974, are shown in Table 20.2.

Table 20.1. Dimensions of standard V-belts according to IS: 2494 – 1974.

Type of belt	*Power ranges in kW*	*Minimum pitch diameter of pulley (D) mm*	*Top width (b) mm*	*Thickness (t) mm*	*Weight per metre length in newton*
A	0.7 – 3.5	75	13	8	1.06
B	2 – 15	125	17	11	1.89
C	7.5 – 75	200	22	14	3.43
D	20 – 150	355	32	19	5.96
E	30 – 350	500	38	23	–

Table 20.2. Dimensions of standard V-grooved pulleys according to IS : 2494–1974. (All dimensions in mm)

Type of belt	*w*	*d*	*a*	*c*	*f*	*e*	*No. of sheave grooves (n)*	*Groove angle (2β) in degrees*
A	11	12	3.3	8.7	10	15	6	32, 34, 38
B	14	15	4.2	10.8	12.5	19	9	32, 34, 38
C	19	20	5.7	14.3	17	25.5	14	34, 36, 38
D	27	28	8.1	19.9	24	37	14	34, 36, 38
E	32	33	9.6	23.4	29	44.5	20	–

Note : Face width (*B*) = (*n* – 1) *e* + 2 *f*

20.3 Standard Pitch Lengths of V-belts

According to IS: 2494-1974, the V-belts are designated by its type and nominal inside length. For example, a *V*-belt of type *A* and inside length 914 mm is designated as *A* 914–IS: 2494. The standard inside lengths of *V*-belts in mm are as follows :

610, 660, 711, 787, 813, 889, 914, 965, 991, 1016, 1067, 1092, 1168, 1219, 1295, 1372, 1397, 1422, 1473, 1524, 1600, 1626, 1651, 1727, 1778, 1905, 1981, 2032, 2057, 2159, 2286, 2438, 2464, 2540, 2667, 2845, 3048, 3150, 3251, 3404, 3658, 4013, 4115, 4394, 4572, 4953, 5334, 6045, 6807, 7569, 8331, 9093, 9885, 10 617, 12 141, 13 665, 15 189, 16 713


Material handler.

According to IS: 2494-1974, the pitch length is defined as the circumferential length of the belt at the pitch width (*i.e.* the width at the neutral axis) of the belt. The value of the pitch width remains constant for each type of belt irrespective of the groove angle.

The pitch lengths are obtained by adding to inside length: 36 mm for type *A*, 43 mm for type *B*, 56 mm for type *C*, 79 mm for type *D* and 92 mm for type *E*. The following table shows the standard pitch lengths for the various types of belt.

Table 20.3. Standard pitch lengths of V-belts according to IS: 2494-1974.

Type of belt	*Standard pitch lengths of V-belts in mm*
A	645, 696, 747, 823, 848, 925, 950, 1001, 1026, 1051, 1102 1128, 1204, 1255, 1331, 1433, 1458, 1509, 1560, 1636, 1661, 1687, 1763, 1814, 1941, 2017, 2068, 2093, 2195, 2322, 2474, 2703, 2880, 3084, 3287, 3693.
B	932, 1008, 1059, 1110, 1212, 1262, 1339, 1415, 1440, 1466, 1567, 1694, 1770, 1821, 1948, 2024, 2101, 2202, 2329, 2507, 2583, 2710, 2888, 3091, 3294, 3701, 4056, 4158, 4437, 4615, 4996, 5377.
C	1275, 1351, 1453, 1580, 1681, 1783, 1834, 1961, 2088, 2113, 2215, 2342, 2494, 2723, 2901, 3104, 3205, 3307, 3459, 3713, 4069, 4171, 4450, 4628, 5009, 5390, 6101, 6863, 7625, 8387, 9149.
D	3127, 3330, 3736, 4092, 4194, 4473, 4651, 5032, 5413, 6124, 6886, 7648, 8410, 9172, 9934, 10 696, 12 220, 13 744, 15 268, 16 792.
E	5426, 6137, 6899, 7661, 8423, 9185, 9947, 10 709, 12 233, 13 757, 15 283, 16 805.

Note: The *V*-belts are also manufactured in non-standard pitch lengths (*i.e.* in oversize and undersize). The standard pitch length belt is designated by grade number 50. The oversize belts are designated by a grade

Contents

number more than 50, while the undersize belts are designated by a grade number less than 50. It may be noted that one unit of a grade number represents 2.5 mm in length from nominal pitch length. For example, a *V*-belt marked *A* – 914 – 50 denotes a standard belt of inside length 914 mm and a pitch length 950 mm. A belt marked *A* – 914 – 52 denotes an oversize belt by an amount of (52 – 50) = 2 units of grade number. Since one unit of grade number represents 2.5 mm, therefore the pitch length of this belt will be 950 + 2 × 2.5 = 955 mm. Similarly, a belt marked *A* – 914 – 48 denotes an undersize belt, whose pitch length will be 950 – 2 × 2.5 = 945 mm.

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

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

Advantages

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

Disadvantages

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

20.5 Ratio of Driving Tensions for V-belt

A *V*-belt with a grooved pulley is shown in Fig. 20.2.

Let R_1 = Normal reactions between belts and sides of the groove.

R = Total reaction in the plane of the groove.

μ = Coefficient of friction between the belt and sides of the groove.

Resolving the reactions vertically to the groove, we have

$$R = R_1 \sin \beta + R_1 \sin \beta = 2R_1 \sin \beta$$

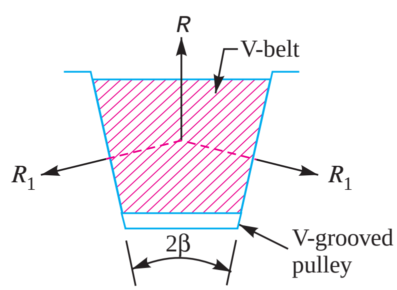


Fig. 20.2. V-belt with pulley.

* The ratio of tensions in *V*-belt drive is cosec β times the flat belt drive.

Top

or $$R_1 = \frac{R}{2\sin\beta}$$

We know that the frictional force

$$= 2\,\mu.R_1 = 2\,\mu \times \frac{R}{2\sin\beta} = \frac{\mu.R}{\sin\beta} = \mu.R.\mathrm{cosec}\,\beta$$

Consider a small portion of the belt, as in Art. 18.19, subtending an angle $\delta\theta$ at the centre, the tension on one side will be T and on the other side $(T + \delta T)$. Now proceeding in the same way as in Art. 18.19, we get the frictional resistance equal to $\mu\,R$. cosec β against $\mu.R$. Thus the relation between T_1 and T_2 for the *V*-belt drive will be

$$2.3 \log (T_1 / T_2) = \mu.\theta \text{ cosec } \beta$$

20.6 V-flat Drives

In many cases, particularly, when a flat belt is replaced by *V*-belt, it is economical to use flat-faced pulley, instead of large grooved pulley, as shown in Fig. 20.3. The cost of cutting the grooves is thereby eliminated. Such a drive is known as ***V-flat drive.***

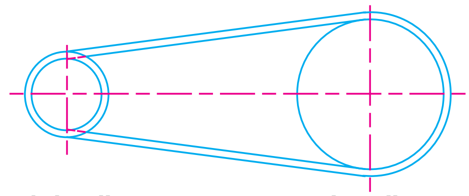


Fig. 20.3. V-flat drive.

5-tine clamps of a material handlesr

Example 20.1. *A compressor, requiring 90 kW, is to run at about 250 r.p.m. The drive is by V-belts from an electric motor running at 750 r.p.m. The diameter of the pulley on the compressor shaft must not be greater than 1 metre while the centre distance between the pulleys is limited to 1.75 metre. The belt speed should not exceed 1600 m / min.*

Determine the number of V-belts required to transmit the power if each belt has a cross-sectional area of 375 mm², density 1000 kg / m³ and an allowable tensile stress of 2.5 MPa. The groove angle of the pulleys is 35°. The coefficient of friction between the belt and the pulley is 0.25. Calculate also the length required of each belt.

Solution. Given : P = 90 kW = 90 × 10^3 W ; N_2 = 250 r.p.m. ; N_1 = 750 r.p.m. ; d_2 = 1 m ; x = 1.75 m ; v = 1600 m/min = 26.67 m/s ; a = 375 mm^2 = 375 × 10^{-6} m^2 ; ρ = 1000 kg / m^3 ; σ = 2.5 MPa = 2.5 N/mm^2 ; 2β = 35° or β = 17.5° ; μ = 0.25

First of all, let us find the diameter of pulley on the motor shaft (d_1). We know that

$$\frac{N_1}{N_2} = \frac{d_2}{d_1} \quad \text{or} \quad d_1 = \frac{d_2 N_2}{N_1} = \frac{1 \times 250}{750} = 0.33 \text{ m}$$

For an open belt drive, as shown in Fig. 20.4,

$$\sin \alpha = \frac{O_2M}{O_1O_2} = \frac{r_2 - r_1}{x} = \frac{d_2 - d_1}{2x} = \frac{1 - 0.33}{2 \times 1.75} = 0.1914$$

$$\therefore \quad \alpha = 11.04°$$

and angle of lap on the smaller pulley (*i.e.* pulley on the motor shaft),

$$\theta = 180° - 2\alpha = 180 - 2 \times 11.04 = 157.92°$$

$$= 157.92 \times \frac{\pi}{180} = 2.76 \text{ rad}$$

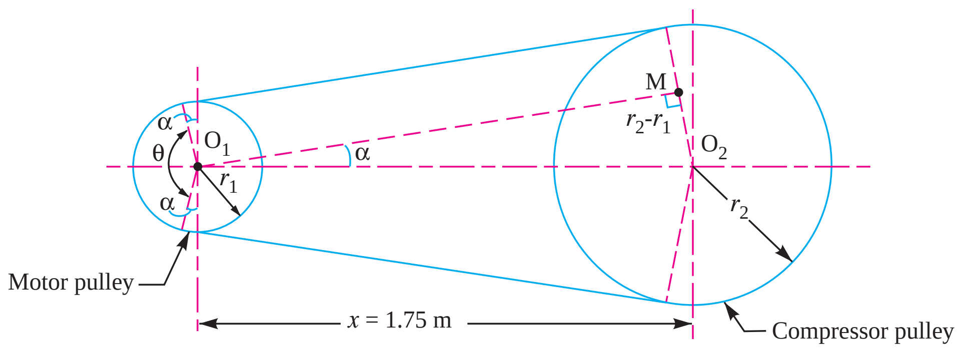


Fig. 20.4

We know that mass of the belt per metre length,

$$m = \text{Area} \times \text{length} \times \text{density} = 375 \times 10^{-6} \times 1 \times 1000 = 0.375 \text{ kg / m}$$

∴ Centrifugal tension,

$$T_C = m.v^2 = 0.375\,(26.67)^2 = 267 \text{ N}$$

and maximum tension in the belt,

$$T = \sigma \times a = 2.5 \times 375 = 937.5 \text{ N}$$

∴ Tension in the tight side of the belt,

$$T_1 = T - T_C = 937.5 - 267 = 670.5 \text{ N}$$

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

We know that

$$2.3 \log \left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta = 0.25 \times 2.76 \times \operatorname{cosec} 17.5°$$

$$= 0.69 \times 3.3255 = 2.295$$

$$\therefore \quad \log \left(\frac{T_1}{T_2}\right) = \frac{2.295}{2.3} = 0.9976 \quad \text{or} \quad \frac{T_1}{T_2} = 9.95 \qquad \text{...(Taking antilog of 0.9976)}$$

and $T_2 = T_1 / 9.95 = 670.5 / 9.95 = 67.4$ N

Number of V-belts

We know that the power transmitted per belt,

$$= (T_1 - T_2)\, v = (670.5 - 67.4)\, 26.67 = 16\,085 \text{ W} = 16.085 \text{ kW}$$

∴ Number of *V*-belts

$$= \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}} = \frac{90}{16.085} = 5.6 \text{ say } 6$$ **Ans.**

Length of each belt

We know that radius of pulley on motor shaft,

$$r_1 = d_1 / 2 = 0.33 / 2 = 0.165 \text{ m}$$

and radius of pulley on compressor shaft,

$$r_2 = d_2 / 2 = 1 / 2 = 0.5 \text{ m}$$

We know that length of each belt,

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

$$= \pi (0.5 + 0.165) + 2 \times 1.75 + \frac{(0.5 - 0.165)^2}{1.75}$$

$$= 2.09 + 3.5 + 0.064 = 5.654 \text{ m}$$ **Ans.**

Example 20.2. *A belt drive consists of two V-belts in parallel, on grooved pulleys of the same size. The angle of the groove is 30°. The cross-sectional area of each belt is 750 mm^2 and μ = 0.12. The density of the belt material is 1.2 Mg / m^3 and the maximum safe stress in the material is 7 MPa. Calculate the power that can be transmitted between pulleys of 300 mm diameter rotating at 1500 r.p.m. Find also the shaft speed in r.p.m. at which the power transmitted would be a maximum.*

Solution. Given : $n = 2$; $2\beta = 30°$ or $\beta = 15°$; $a = 750 \text{ mm}^2 = 750 \times 10^{-6} \text{ m}^2$; $\mu = 0.12$; $\rho = 1.2$ Mg/m^3 = 1200 kg/m^3 ; $\sigma = 7 \text{ MPa} = 7 \times 10^6 \text{ N/m}^2$; $d = 300 \text{ mm} = 0.3 \text{ m}$; $N = 1500$ r.p.m.

We know that mass of the belt per metre length,

$$m = \text{Area} \times \text{length} \times \text{density} = 750 \times 10^{-6} \times 1 \times 1200 = 0.9 \text{ kg/m}$$

and speed of the belt, $$v = \frac{\pi d N}{60} = \frac{\pi \times 0.3 \times 1500}{60} = 23.56 \text{ m/s}$$

∴ Centrifugal tension,

$$T_C = m.v^2 = 0.9 (23.56)^2 = 500 \text{ N}$$

and maximum tension, $$T = \sigma \times a = 7 \times 10^6 \times 750 \times 10^{-6} = 5250 \text{ N}$$

We know that tension in the tight side of the belt,

$$T_1 = T - T_C = 5250 - 500 = 4750 \text{ N}$$

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

Since the pulleys are of the same size, therefore angle of lap (θ) = 180° = π rad.

We know that

$$2.3 \log \left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta = 0.12 \times \pi \times \operatorname{cosec} 15° = 0.377 \times 3.8637 = 1.457$$

$$\therefore \quad \log \left(\frac{T_1}{T_2}\right) = \frac{1.457}{2.3} = 0.6335 \quad \text{or} \quad \frac{T_1}{T_2} = 4.3$$ ...(Taking antilog of 0.6335)

and $$T_2 = T_1 / 4.3 = 4750 / 4.3 = 1105 \text{ N}$$

Power transmitted

We know that power transmitted,

$$P = (T_1 - T_2)\, v \times n = (4750 - 1105)\, 23.56 \times 2 = 171\,750 \text{ W}$$

$$= 171.75 \text{ kW}$$ **Ans.**

Shaft speed

Let N_1 = Shaft speed in r.p.m., and

v_1 = Belt speed in m/s.

We know that for maximum power, centrifugal tension,

$$T_C = T/3 \quad \text{or} \quad m(v_1)^2 = T/3$$

or $$0.9\,(v_1)^2 = 5250/3 = 1750$$

∴ $$(v_1)^2 = 1750/0.9 = 1944.4 \quad \text{or} \quad v_1 = 44.1 \text{ m/s}$$

We know that belt speed (v_1),

$$44.1 = \frac{\pi\, d\, N_1}{60} = \frac{\pi \times 0.3 \times N_1}{60} = 0.0157\ N_1$$

∴ $$N_1 = 44.1/0.0157 = 2809 \text{ r.p.m.}$$ **Ans.**

Example 20.3. *Two shafts whose centres are 1 metre apart are connected by a V-belt drive. The driving pulley is supplied with 95 kW power and has an effective diameter of 300 mm. It runs at 1000 r.p.m. while the driven pulley runs at 375 r.p.m. The angle of groove on the pulleys is 40°. Permissible tension in 400 mm² cross-sectional area belt is 2.1 MPa. The material of the belt has density of 1100 kg / m³. The driven pulley is overhung, the distance of the centre from the nearest bearing being 200 mm. The coefficient of friction between belt and pulley rim is 0.28. Estimate: 1. The number of belts required ; and 2. Diameter of driven pulley shaft, if permissible shear stress is 42 MPa.*

Solution. Given : x = 1 m ; P = 95 kW = 95 × 10^3 W ; d_1 = 300 mm = 0.3 m ; N_1 = 1000 r.p.m. ; N_2 = 375 r.p.m ; 2 β = 40° or β = 20° ; a = 400 mm² = 400 × 10^{-6} m² ; σ = 2.1 MPa = 2.1 N/mm² ; ρ = 1100 kg/m³; μ = 0.28 ; τ = 42 MPa = 42 N/mm²

First of all, let us find the diameter of the driven pulley (d_2). We know that

$$\frac{N_1}{N_2} = \frac{d_2}{d_1} \quad \text{or} \quad d_2 = \frac{N_1 \times d_1}{N_2} = \frac{1000 \times 300}{375} = 800 \text{ mm} = 0.8 \text{ m}$$

For an open belt drive,

$$\sin\alpha = \frac{r_2 - r_1}{x} = \frac{d_2 - d_1}{2x} = \frac{0.8 - 0.3}{2 \times 1} = 0.25$$

∴ $$\alpha = 14.5°$$

and angle of lap on the smaller or driving pulley,

$$\theta = 180° - 2\,\alpha = 180° - 2 \times 14.5 = 151°$$

$$= 151 \times \frac{\pi}{180} = 2.64 \text{ rad}$$

We know that the mass of the belt per metre length,

$$m = \text{Area} \times \text{length} \times \text{density} = 400 \times 10^{-6} \times 1 \times 1100 = 0.44 \text{ kg/m}$$

and velocity of the belt,

$$v = \frac{\pi\, d_1 . N_1}{60} = \frac{\pi \times 0.3 \times 1000}{60} = 15.71 \text{ m/s}$$

∴ Centrifugal tension,

$$T_C = m.v^2 = 0.44\,(15.71)^2 = 108.6 \text{ N}$$

and maximum tension in the belt,

$$T = \sigma \times a = 2.1 \times 400 = 840 \text{ N}$$

∴ Tension in the tight side of the belt,

$$T_1 = T - T_C = 840 - 108.6 = 731.4 \text{ N}$$

We know that

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec}\beta = 0.28 \times 2.64 \operatorname{cosec} 20° = 0.74 \times 2.9238 = 2.164$$

$$\therefore \quad \log\left(\frac{T_1}{T_2}\right) = \frac{2.164}{2.3} = 0.9407 \quad \text{or} \quad \frac{T_1}{T_2} = 8.72 \quad \text{...(Taking antilog of 0.9407)}$$

and
$$T_2 = \frac{T_1}{8.72} = \frac{731.4}{8.72} = 83.9 \text{ N}$$

1. *Number of belts required*

We know that the power transmitted per belt

$$= (T_1 - T_2)\, v = (731.4 - 83.9)\, 15.71 = 10\,172 \text{ W} = 10.172 \text{ kW}$$

$\therefore$ Number of belts required

$$= \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}} = \frac{95}{10.172} = 9.34 \text{ say } 10 \textbf{ Ans.}$$

2. *Diameter of driven pulley shaft*

Let D = Diameter of driven pulley shaft.

We know that torque transmitted by the driven pulley shaft,

$$T = \frac{P \times 60}{2\pi N_2} = \frac{95 \times 10^3 \times 60}{2\pi \times 375} = 2420 \text{ N-m} = 2420 \times 10^3 \text{ N-mm}$$

Since the driven pulley is overhung and the distance of the centre from the nearest bearing is 200 mm, therefore bending moment on the shaft due to the pull on the belt,

$$M = (T_1 + T_2 + 2T_C)\, 200 \times 10 \quad \text{...}(\because \text{ No. of belts} = 10)$$
$$= (731.4 + 83.9 + 2 \times 108.6)\, 200 \times 10 = 2065 \times 10^3 \text{ N-mm}$$

$\therefore$ Equivalent twisting moment,

$$T_e = \sqrt{T^2 + M^2} = \sqrt{(2420 \times 10^3)^2 + (2065 \times 10^3)^2} \text{ N-mm}$$
$$= 3181 \times 10^3 \text{ N-mm}$$

We know that equivalent twisting moment (T_e),

$$3181 \times 10^3 = \frac{\pi}{16} \times \tau \times D^3 = \frac{\pi}{16} \times 42\, D^3 = 8.25\, D^3$$

$$\therefore \quad D^3 = 3181 \times 10^3 / 8.25 = 386 \times 10^3$$

or
$$D = 72.8 \text{ say } 75 \text{ mm } \textbf{Ans.}$$

Example 20.4. *Power of 60 kW at 750 r.p.m. is to be transmitted from an electric motor to compressor shaft at 300 r.p.m. by V-belts. The approximate larger pulley diameter is 1500 mm. The approximate centre distance is 1650 mm, and overload factor is to be taken as 1.5. Give a complete design of the belt drive. A belt with cross-sectional area of 350 mm² and density 1000 kg / m³ and having an allowable tensile strength 2 MPa is available for use. The coefficient of friction between the belt and the pulley may be taken as 0.28. The driven pulley is overhung to the extent of 300 mm from the nearest bearing and is mounted on a shaft having a permissible shear stress of 40 MPa with the help of a key. The shaft, the pulley and the key are also to be designed.*

Solution. Given : P = 60 kW ; N_1 = 750 r.p.m. ; N_2 = 300 r.p.m. ; d_2 = 1500 mm ; x = 1650 mm ; Overload factor = 1.5 ; $a = 350 \text{ mm}^2 = 350 \times 10^{-6} \text{ m}^2$; $\rho = 1000 \text{ kg/m}^3$; $\sigma = 2$ MPa = 2 N/mm² ; $\mu = 0.28$; $\tau = 40$ MPa = 40 N/mm²

1. *Design of the belt drive*

First of all, let us find the diameter (d_1) of the motor pulley. We know that

$$\frac{N_1}{N_2} = \frac{d_2}{d_1} \quad \text{or} \quad d_1 = \frac{d_2 \times N_2}{N_1} = \frac{1500 \times 300}{750} = 600 \text{ mm} = 0.6 \text{ m}$$

and
$$\sin\alpha = \frac{r_2 - r_1}{x} = \frac{d_2 - d_1}{2x} = \frac{1500 - 600}{2 \times 1650} = 0.2727 \quad \text{or} \quad \alpha = 15.83°$$

We know that the angle of contact,

$$\theta = 180° - 2\alpha = 180 - 2 \times 15.83 = 148.34°$$

$$= 148.34 \times \pi / 180 = 2.6 \text{ rad}$$

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

T_2 = Tension in the slack side of the belt.

Assume the groove angle of the pulley, $2\beta = 35°$ or $\beta = 17.5°$. We know that

$$2.3 \log \left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta = 0.28 \times 2.6 \times \operatorname{cosec} 17.5° = 2.42$$

$$\therefore \quad \log \left(\frac{T_1}{T_2}\right) = 2.42 / 2.3 = 1.0526 \quad \text{or} \quad \frac{T_1}{T_2} = 11.28 \quad \ldots(i)$$

...(Taking antilog of 1.0526)

We know that the velocity of the belt,

$$v = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 0.6 \times 750}{60} = 23.66 \text{ m/s}$$

and mass of the belt per metre length,

$$m = \text{Area} \times \text{length} \times \text{density} = 350 \times 10^{-6} \times 1 \times 1000 = 0.35 \text{ kg / m}$$

∴ Centrifugal tension in the belt,

$$T_C = m.v^2 = 0.35 (23.66)^2 = 196 \text{ N}$$

and maximum tension in the belt,

$$T = \text{Stress} \times \text{area} = \sigma \times a = 2 \times 350 = 700 \text{ N}$$

∴ Tension in the tight side of the belt,

$$T_1 = T - T_C = 700 - 196 = 504 \text{ N}$$

and

$$T_2 = \frac{T_1}{11.28} = \frac{504}{11.28} = 44.7 \text{ N} \quad \ldots[\text{From equation } (i)]$$

We know that the power transmitted per belt

$$= (T_1 - T_2)\, v = (504 - 44.7)\, 23.66 = 10\,867 \text{ W} = 10.867 \text{ kW}$$

Since the over load factor is 1.5, therefore the belt is to be designed for 1.5 × 60 = 90 kW.

∴ Number of belts required

$$= \frac{\text{Designed power}}{\text{Power transmitted per belt}} = \frac{90}{10.867} = 8.3 \text{ say } 9 \textbf{ Ans.}$$

Since the *V*-belt is to be designed for 90 kW, therefore from Table 20.1, we find that a '*D*' type of belt should be used.

We know that the pitch length of the belt,

$$L = \pi (r_2 + r_1) + 2x + \frac{(r_2 - r_1)^2}{x} = \frac{\pi}{2} (d_2 + d_1) + 2x + \frac{(d_2 - d_1)^2}{4x}$$

$$= \frac{\pi}{2} (1500 + 600) + 2 \times 1650 + \frac{(1500 - 600)^2}{4 \times 1650}$$

$$= 3300 + 3300 + 123 = 6723 \text{ mm}$$

Subtracting 79 mm for '*D*' type belt, we find that inside length of the belt

$$= 6723 - 79 = 6644 \text{ mm}$$

According to IS: 2494 – 1974, the nearest standard inside length of *V*-belt is 6807 mm.

∴ Pitch length of the belt,

$$L_1 = 6807 + 79 = 6886 \text{ mm } \textbf{Ans.}$$

Now let us find out the new centre distance (x_1) between the two pulleys. We know that

$$L_1 = \frac{\pi}{2}(d_2 + d_1) + 2x_1 + \frac{(d_2 - d_1)^2}{4x_1}$$

$$6886 = \frac{\pi}{2}(1500 + 600) + 2x_1 + \frac{(1500 - 600)^2}{4x_1}$$

$$= 3300 + 2x_1 + \frac{810\ 000}{4x_1} \quad \text{....(}d_1 \text{ and } d_2 \text{ are taken in mm)}$$

$$6886 \times 4x_1 = 3300 \times 4x_1 + 2x_1 \times 4x_1 + 810\ 000$$

or $$3443\, x_1 = 1650\, x_1 + x_1^2 + 101\ 250$$

or $$x_1^2 - 1793\, x_1 + 101\ 250 = 0$$

$$\therefore \quad x_1 = \frac{1793 \pm \sqrt{(1793)^2 - 4 \times 101\ 250}}{2}$$

$$= \frac{1793 \pm 1677}{2} = 1735 \text{ mm } \textbf{Ans.} \quad \text{...(Taking + ve sign)}$$

2. *Design of shaft*

Let D = Diameter of the shaft.

We know that the torque transmitted by the driven or compressor pulley shaft,

$$T = \frac{\text{Designed power} \times 60}{2\pi N_2} = \frac{90 \times 10^3 \times 60}{2\pi \times 300} = 2865 \text{ N-m}$$

$$= 2865 \times 10^3 \text{ N-mm}$$

Since the overhang of the pulley is 300 mm, therefore bending moment on the shaft due to the belt tensions,

$$M = (T_1 + T_2 + 2T_C)\, 300 \times 9 \quad \text{...(}\because \text{ No. of belts = 9)}$$

$$= (504 + 44.7 + 2 \times 196)\, 300 \times 9 = 2540 \times 10^3 \text{ N-mm}$$

∴ Equivalent twisting moment,

$$T_e = \sqrt{T^2 + M^2} = \sqrt{(2865 \times 10^3)^2 + (2540 \times 10^3)^2}$$

$$= 3830 \times 10^3 \text{ N-mm}$$

We also know that equivalent twisting moment (T_e),

$$3830 \times 10^3 = \frac{\pi}{16} \times \tau \times D^3 = \frac{\pi}{16} \times 40 \times D^3 = 7.855\, D^3$$

$$\therefore \quad D^3 = 3830 \times 10^3 / 7.855 = 487.6 \times 10^3 \quad \text{or} \quad D = 78.7 \text{ say } 80 \text{ mm } \textbf{Ans.}$$

3. *Design of the pulley*

The dimensions for the standard *V*-grooved pulley (Refer Fig. 20.1) are shown in Table 20.2, from which we find that for '*D*' type belt

w = 27 mm, d = 28 mm, a = 8.1 mm, c = 19.9 mm, f = 24 mm, and e = 37 mm.

We know that face width of the pulley,

$$B = (n - 1)\, e + 2f = (9 - 1)\, 37 + 2 \times 24 = 344 \text{ mm } \textbf{Ans.}$$

4. *Design for key*

The standard dimensions of key for a shaft of 80 mm diameter are

Width of key = 25 mm **Ans.**

and thickness of key = 14 mm **Ans.**

Contents

Example 20.5. *A V-belt is driven on a flat pulley and a V-pulley. The drive transmits 20 kW from a 250 mm diameter V-pulley operating at 1800 r.p.m. to a 900 mm diameter flat pulley. The centre distance is 1 m, the angle of groove 40° and μ = 0.2. If density of belting is 1110 kg / m^3 and allowable stress is 2.1 MPa for belt material, what will be the number of belts required if C-size V-belts having 230 mm^2 cross-sectional area are used.*

Solution. Given : P = 20 kW ; d_1 = 250 mm = 0.25 m ; N_1 = 1800 r.p.m. ; d_2 = 900 mm = 0.9 m ; x = 1 m = 1000 mm ; 2 β = 40° or β = 20° ; μ = 0.2 ; ρ = 1110 kg/m^3 ; σ = 2.1 MPa = 2.1 N/mm^2 ; a = 230 mm^2 = 230 × 10^{-6} m^2

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

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

$$\therefore \quad \alpha = 18.96°$$

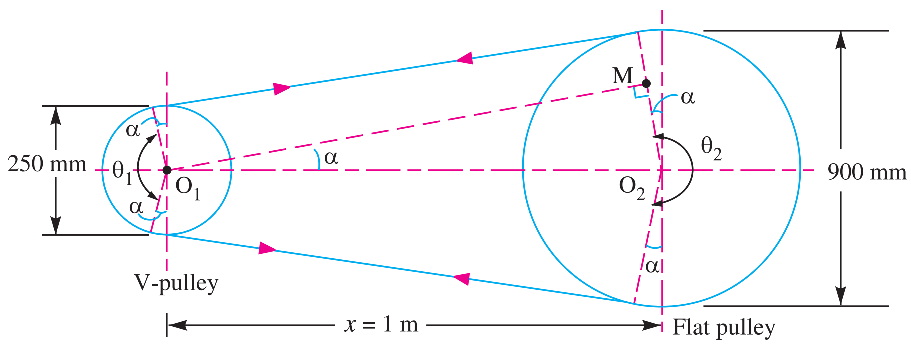


Fig. 20.5

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

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

and angle of contact on the larger or flat pulley,

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

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

We know that for a smaller or *V*-pulley,

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

and for larger or flat pulley,

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

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

We know that peripheral velocity of the belt,

$$v = \frac{\pi\, d_1\, N_1}{60} = \frac{\pi \times 0.25 \times 1800}{60} = 23.56 \text{ m/s}$$

Mass of the belt per metre length,

$$m = \text{Area} \times \text{length} \times \text{density} = a \times l \times \rho$$
$$= 230 \times 10^{-6} \times 1 \times 1100 = 0.253 \text{ kg / m}$$

Top

Contents

∴ Centrifugal tension,

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

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

T_2 = Tension in the slack side of the belt.

We know that maximum tension in the belt,

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

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

$$T = T_1 + T_C$$

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

We know that

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

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

and

$$T_2 = T_1 / 2.14 = 342.6 / 2.14 = 160 \text{ N}$$

∴ Power transmitted per belt

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

We know that number of belts required

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

20.7 Rope Drives

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

The ropes drives use the following two types of ropes :

1. Fibre ropes, and **2.** *Wire ropes.

The fibre ropes operate successfully when the pulleys are about 60 metres apart, while the wire ropes are used when the pulleys are upto 150 metres apart.

20.8 Fibre Ropes

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

The cotton ropes are very soft and smooth. The lubrication of cotton ropes is not necessary. But if it is done, it reduces the external wear between the rope and the grooves of its sheaves. It may be noted that the manila ropes are more durable and stronger than cotton ropes. The cotton ropes are costlier than manila ropes.

* Wire ropes are discussed in Art. 20.12.

Top