Example 1.3 *It is required to select a flat-belt drive for a fan running at 360 r.p.m. which is driven by a 10 kW, 1440 r.p.m. motor. The belt drive is open-type and space available for a centre distance of 2 m approximately. The diameter of a driven pulley is 1000 mm.*

Given Data : N_1 = 1440 r.p.m. ; N_2 = 360 r.p.m. ; P = 10 kW = 10 × 10^3 W ;

C = 2 m ; D = 1000 mm.

To find : Select (or design) a open flat belt drive.

☺ ***Solution :*** The given arrangement is shown in Fig.1.14.

1. Calculation of pulley diameters :

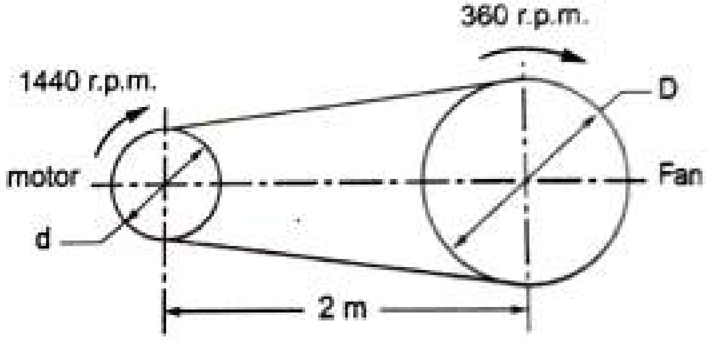

Fig. 1.14.

Driven pulley diameter, D = **1000 mm**

We know that velocity ratio $= \dfrac{D}{d}$

$$= \frac{\text{Driver pulley speed}}{\text{Driven pulley speed}} = \frac{N_1}{N_2} = \frac{1440}{360} = 4$$

$$\therefore \quad \text{Driver pulley diameter, } d = \frac{D}{4} = \frac{1000}{4} = 250 \text{ mm}$$

754 Consulting Table 1.5, the recommended driver pulley diameter = **250 mm Ans.**

2. Calculation of design power in kW :

$$\text{Design kW} = \frac{\text{Rated kW} \times \text{Load correction factor } (K_s)}{\text{Arc of contact factor } (K_\alpha) \times \text{Small pulley factor } (K_d)}$$

(i) Rated kW = 10 kW ... [Given]

7.53 (ii) Referring to Table 1.9, load correction factor, $K_s = 1.2$ for steady load.

(iii) To find arc of contact factor (K_α) :

754 $$\text{Arc of contact} = 180° - \left(\frac{D-d}{C}\right) \times 60°$$

$$= 180^\circ - \left(\frac{1000-250}{2000}\right) \times 60^\circ = 157.5^\circ$$

Consulting Table 1.10, arc of contact factor for 157.5°, $K_\alpha = 1.08$.

(iv) Consulting Table 1.11, small pulley factor, $K_d = 0.7$

$$\therefore \quad \text{Design kW} = \frac{10 \times 1.2}{1.08 \times 0.7} = \mathbf{15.873\ kW} \quad \textbf{Ans.}$$

3. *Selection of belt :*

7.54

Consulting Table 1.12, ***HI-SPEED duck belting*** is selected. Its capacity is given as 0.023 kW/mm/ply.

4. *Load rating correction :*

$$\text{Velocity of the belt, } V = \frac{\pi d N_1}{60} = \frac{\pi \times 0.25 \times 1440}{60} = 18.85 \text{ m/s}$$

$$\text{Load rating at V m/s} = \text{Load rating at 10 m/s} \times \frac{V}{10}$$

$$\therefore \quad \text{Load rating at 18.85 m/s} = \text{Load rating at 10 m/s} \times (18.85 / 10)$$
$$= 0.023 \times (18.85 / 10) = 0.04335 \text{ kW / mm / ply}$$

5. *Determination of belt width :*

7.52

For 250 mm smaller pulley diameter and velocity of 18.85 m/s, consulting Table 1.8, the number of plies can be selected as 5.

$$\therefore \quad \text{Width of belt} = \frac{\text{Design power}}{\text{Load rating} \times \text{No. of plies}}$$
$$= \frac{15.873}{0.04335 \times 5} = 73.23 \text{ mm}$$

Consulting Table 1.13, the calculated belt width should be rounded off to the standard belt width.

∴ For 5 ply belt, standard belt width = **76 mm** **Ans.**

6. *Determination of pulley width :*

Consulting Table 1.6(a), the pulley width is given by

$$\text{Pulley width} = \text{Belt width} + 13 \text{ mm} = 76 + 13 = 89 \text{ mm}$$

∴ Referring Table 1.6(b), the standard pulley width is **90 mm** **Ans.**

7. *Calculation of length of the belt (L) :*

We know that the length of an open belt, 7.59 7.53

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

$$= 2 \times 2000 + \frac{\pi}{2}(1000 + 250) + \frac{(1000-250)^2}{4 \times 2000} = \mathbf{6033.8\ mm} \quad \textbf{Ans.}$$

Example 1.7 ***A flat belt is required to transmit 35 kW from a pulley of 1.5 m effective diameter running at 300 r.p.m. The angle of lap is 165° and $\mu = 0.3$. Determine, taking centrifugal tension into account, width of the belt required. It is given that the belt thickness is 9.5 mm, density of its material is 1.1 Mg/m³ and the related permissible working stress is 2.5 MPa.***

Given Data : $P = 35 \text{ kW} = 35 \times 10^3 \text{ W}$; $d = 1.5$ m; $N = 300$ r.p.m. ;

Mu Rho

$\alpha = 165° = 165° \times \dfrac{\pi}{180°} = 2.88$ rad; $\mu = 0.3$; $t = 9.5$ mm ; $\rho = 1.1 \text{ Mg/m}^3 = 1100 \text{ kg/m}^3$

Sigma

$\sigma = 2.5 \text{ MPa} = 2.5 \times 10^6 \text{ N/m}^2$.

To find : Width of the belt (b).

☺ ***Solution :*** Velocity of belt, $\boxed{v = \dfrac{\pi d N}{60}} = \dfrac{\pi \times 1.5 \times 300}{60} = 23.56$ m/s.

Let b = Belt width in mm.

We know that,

$$\boxed{P = (T_1 - T_2)\, v}$$

$$35 \times 10^3 = (T_1 - T_2)\ 23.56 \quad \text{or} \quad T_1 - T_2 = 1485.45 \qquad \ldots \text{(i)}$$

$$\frac{T_1}{T_2} = e^{\mu\alpha} = e^{0.3 \times 2.88} = 2.373 \quad \text{or} \quad T_1 = 2.373\, T_2 \qquad \ldots \text{(ii)}$$

Solving (i) and (ii), $T_1 = 2568$ N and $T_2 = 1082.19$ N

Cross-sectional area of the belt $\boxed{= b \times t} = 9.5\, b \text{ mm}^2 = 9.5\, b \times 10^{-6} \text{ m}^2$

We know that mass of the belt per meter length,

$$m = \text{Density} \times \text{Area} \times \text{Length} = \boxed{\rho \times (b \times t) \times l}$$

$$= 1100 \times 9.5\, b \times 10^{-6} \times 1 = 0.01045\, b \text{ kg/m}$$

$\therefore$ Centrifugal tension, $\boxed{T_C = m v^2} = 0.01045\, b\, (23.56)^2 = 5.8\, b$ N and Maximum tension in the belt, $T = \sigma(b \times t)$

$$= 2.5 \times 10^6 \times 9.5\,b \times 10^{-6} = 23.75\,b \text{ N}$$

We also know that $T = T_1 + T_C$

$$23.75\,b = 2568 + 5.8\,b \quad \text{or} \quad b = 143 \text{ mm}$$

Consulting Table 1.13, standard width of the belt = **152 mm** **Ans.**

Model
Sem

Example 2.2 ***Design a V-belt drive to the following specifications :***

P	*Power to be transmitted*	=	*7.5 kW*
N_1	*Speed of driving wheel*	=	*1440 r.p.m.*
N_2	*Speed of driven wheel*	=	*400 r.p.m.*
d	*Diameter of driving wheel*	=	*300 mm*
C	*Centre distance*	=	*1000 mm*
	Service	=	*16 hours / day*

Given Data : P = 7.5 kW ; N_1 = 1440 r.p.m. ; N_2 = 400 r.p.m. ;

d = 300 mm = 0.3 m ; C = 1000 mm = 1 m.

PSG 7.58 → V belt

To find : Design a V-belt drive.

☺ Solution :

1. Selection of the belt section : PSG 7.58

Consulting Table 2.3, for power 7.5 kW ***B section*** is selected. 2.4

2. Selection of pulley diameters (d and D) :

$i = \frac{D}{d}$

$$\text{Speed ratio} = \frac{D}{d} = \frac{N_1}{N_2} = \frac{1440}{400} = 3.6 \qquad D = i \times d$$

D = 3.6 × 300

Smaller pulley diameter, d = 300 mm ... (Given)

Referring Table 1.5, the preferred smaller pulley diameter, d = **315 mm.** 7.54 1.14

∴ Larger pulley diameter, D = 3.6 d = 3.6 × 315 = 1134 mm

Referring Table 1.5, the preferred larger pulley diameter, D = **1250 mm.** 7.54 1.14

3. Selection of centre distance (C) : 7.61 PSG

Centre distance, C = **1000 mm** ... (Given)

4. Determination of nominal pitch length : L 7.53 7.61

$$\text{Nominal inside length, } L = 2\,C + \left(\frac{\pi}{2}\right)(D+d) + \frac{(D-d)^2}{4\,C}$$

$$= (2 \times 1000) + \left(\frac{\pi}{2}\right)(1250+315) + \frac{(1250-315)^2}{4 \times 1000}$$

(2000) + (2458.3) + (218.6)

$$= 4676.85 \text{ mm}$$

For this nominal inside length and B section, consulting Table 2.5, the next standard pitch length is selected as **4996 mm.** 7.60 2.7   L = 4996

5. Selection of various modification factors :

hours	1.3	Fa Service	→ 7.69
speed ratio	1.14	Fb	→ 7.68
length	1.18	Fc Correction	→ 7.60
Arc of contact	0.83	Fd Arc of contact	→ 7.68

(i) Length correction factor (F_c) : For B section, referring Table 2.5,

length correction factor, F_c = 1.18 7.60

(ii) Correction factor for arc of contact (F_d) : 7.68, 7.54

$$\text{Arc of contact} = 180° - \left(\frac{D-d}{C}\right) \times 60°$$ 7.54

$$= 180° - \left(\frac{1250-315}{1000}\right) \times 60° = 123.9°$$

= 180 − 0.935 × 60

= 180 − 56.1

Arc of contact = 123.9 7.68

F_d Correction Factor ⇒ 0.83

F_d = 0.83

23.4 → 0.83 7.68 2.7

For this arc of contact, consulting Table 2.6, correction factor for arc of contact is selected as F_d = 0.83.

7.69 2.8

(iii) Service factor (F_a): Consulting Table 2.7, for light duty 16 hours continuous service, for driving machines of type II, service factor is selected as F_a = 1.3.

6. *Calculation of maximum power capacity :*

Consulting Table 2.8, for B section, power capacity formula is given as

$$\text{kW} = \left(0.79\, S^{-0.09} - \frac{50.8}{d_e} - 1.32 \times 10^{-4} S^2\right) S$$ 7.62

$S = \frac{\pi d N_1}{60}$

where S = Belt speed $= \frac{\pi\, d\, N_1}{60} = \frac{\pi \times 0.315 \times 1440}{60}$ = 23.75 m/s → S

d_e = Equivalent pitch diameter = $d_p \times F_b$ $d_e = d_p \cdot F_b$ 7.62

d_p = Pitch diameter of the smaller pulley = d = 315 mm

7.62 2.9

F_b = Small diameter factor for speed ratio of 3.6 = 1.14 ... [From Table 2.9]

$\therefore$ d_e = 315 × 1.14 = 359.1 mm → d_e → Max 175 > Ans 7.62

7.62 2.8 7.62

But from Table 2.8, maximum value of d_e in the formula should be 175 mm. → d_e

$$\text{Power, kW} = \left(0.79 \times 23.75^{-0.09} - \frac{50.8}{175} - 1.32 \times 10^{-4} \times 23.75^2\right) 23.75 = 5.445\ \text{kW}$$

for B → 175

7. *Determination of number of belts* (n_b):

7.70

We know that
$$n_b = \frac{P \times F_a}{\text{kW} \times F_c \times F_d}$$
$$= \frac{7.5 \times 1.3}{5.445 \times 1.18 \times 0.83} = 1.828 \approx \textbf{2 belts} \quad \textbf{Ans.}$$ → n_b

8. *Calculation of actual centre distance :* C_{act}

Actual centre distance is given by

$$C_{\text{actual}} = A + \sqrt{A^2 - B}$$ 7.61

where
$$A = \frac{L}{4} - \pi\left[\frac{D+d}{8}\right] = \frac{4996}{4} - \pi\left[\frac{1250+315}{8}\right]$$
$$= 634.42$$

and
$$B = \frac{(D-d)^2}{8} = \frac{(1250-315)^2}{8} = 109278$$

$$\therefore \quad C_{\text{actual}} = 634.42 + \sqrt{(634.42)^2 - 109278}$$
$$C_{act} = \textbf{1175.92 mm} \quad \textbf{Ans.}$$

Example 2.3 *A centrifugal pump running at 340 r.p.m is to be driven by a 100 kW motor running at 1440 r.p.m. The drive is to work for atleast 20 hours every day. The centre distance between the motor shaft and the pump shaft is 1200 mm. Suggest a suitable multiple V-belt drive for this application. Also calculate the actual belt tensions and stress induced.*

=feed (driven) Power speed (driver) centre distance

Given Data : N_2 = 340 r.p.m. ; P = 100 kW ; N_1 = 1440 r.p.m. ; C = 1200 mm = 1.2 m

To find : (i) Design a V-belt drive, and Service = 20 Hours/day

(ii) Actual belt tensions and stress induced.

Solution :

1. Selection of the belt section : 7.58

Consulting Table 2.3, for power 100 kW, ***D section*** is selected.

2. Selection of pulley diameters (d and D) :

Since diameters of both pulleys are not given, therefore first select the smaller pulley diameter from Table 2.3. 7.58

∴ Consulting Table 2.3, for power 100 kW, smaller pulley diameter, d = **355 mm.** 7.58

$$\text{Speed ratio} = \frac{D}{d} = \frac{N_1}{N_2} = \frac{1440}{340} = 4.235$$

∴ Larger pulley diameter, $D = 4.235 \times d = 4.235 \times 355 = 1503.53$ mm

Consulting Table 1.5, the preferred larger pulley diameter, **D = 1600 mm.** 7.54

3. Selection of centre distance (C) :

Centre distance, **C = 1200 mm** ... (Given)

4. Determination of nominal pitch length :

7.53

$$\text{Nominal inside length, } L = 2\,C + \frac{\pi}{2}(D + d) + \frac{(D - d)^2}{4\,C}$$

$$= 2 \times 1200 + \frac{\pi}{2}(1600 + 355) + \frac{(1600 - 355)^2}{4 \times 1200}$$

$$= 5793.83 \text{ mm}$$

For this nominal inside length and D section, consulting Table 2.5, the next standard pitch length is selected as **6124 mm.** (L) 7.60

5. Selection of various modification factors :

(i) Length correction factor : For D section, referring Table 2.5, length correction factor, F_c = 1.00. 7.60

Correction factor for Industrial service	F_a	7.64
Small dia factor	F_b	7.62
Correction factor	F_c	7.60
for arc of contact	F_d	7.68

Fa 1.3
Fb 1.14
Fc 1.00
Fd 0.81

(ii) Correction factor for arc of contact (F_d) :

$$\text{Arc of contact} = 180° - \left(\frac{D-d}{C}\right) \times 60° = 180° - \left(\frac{1600-355}{1200}\right) 60°$$

$$= 117.75°$$

7.68

For 117.75°, consulting Table 2.6, correction factor for arc of contact, $F_d = 0.81$.

(iii) Service factor (F_a) : For light duty, for over 16 hours continuous service, for driving machines of type II, consulting Table 2.7, the service factor, $F_a = 1.3$. 7.69

6. Calculation of maximum power capacity (kW) :

Consulting Table 2.8, for D section, power capacity formula is given as

$$\text{kW} = \left(3.22\, S^{-0.09} - \frac{506.7}{d_e} - 4.78 \times 10^{-4}\, S^2\right) S$$ 7.62

$V = \frac{\pi d N}{60}$

where $S = \text{Belt speed} = \frac{\pi d N_1}{60} = \frac{\pi \times 0.355 \times 1440}{60} = 26.76 \text{ m/s}$

$d_e = d_p \times F_b$

d_p = Smaller pulley diameter = 355 mm

F_b = Small diameter factor, for speed ratio of 4.235, from Table 2.9 = 1.14 (7.62)

$\therefore \quad d_e = 355 \times 1.14 = 404.7$

$$\text{Power, kW} = \left(3.22 \times 26.76^{-0.09} - \frac{506.7}{404.7} - 4.78 \times 10^{-4} \times 26.76^2\right) 26.76$$

$$\text{kW} = 21.44 \text{ kW}$$

7. Determination of number of belts (n_b) :

We know that
$$n_b = \frac{P \times F_a}{\text{kW} \times F_c \times F_d}$$ 7.70

$$= \frac{100 \times 1.3}{21.44 \times 1 \times 0.81} = 7.486 \approx 8 \text{ belts} \quad \textbf{Ans.}$$

8. Calculation of actual centre distance :

Actual centre distance, $C_{actual} = A + \sqrt{A^2 - B}$ 7.61

where
$$A = \frac{L}{4} - \pi\left[\frac{D+d}{8}\right] = \frac{6124}{4} - \pi\left[\frac{1600+355}{8}\right] = 763.27$$

$$B = \frac{(D-d)^2}{8} = \frac{(1600-355)^2}{8} = 193753.125$$

L = 6124

$$\therefore \quad C_{actual} = 763.27 + \sqrt{763.27^2 - 193753.125} = 1386.83 \text{ mm} \quad \textbf{Ans.}$$

If need only

9. *Calculation of belt tensions (T_1 and T_2) :*

We know that,

$$\text{Power transmitted per belt} = (T_1 - T_2)\, v$$

$$\frac{100 \times 10^3}{8} = (T_1 - T_2)\, 26.76 \text{ or } T_1 - T_2 = 467.12 \qquad \ldots (i)$$

7.58 weight

From Table 2.3, mass per metre length, $m = 0.596$ kg/m

7.70

From Table 2.2, groove angle $(2\beta) = 34°$.

Angle A

Already found that arc of contact for smaller pulley, $\alpha = 117.75° \times \frac{\pi}{180°}$

$$= 2.055 \text{ radians}$$

We know that the tension ratio for V-belts considering centrifugal tension,

$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\mu\alpha / \sin\beta} = e^{\mu\alpha \cdot \mathrm{cosec}\,\beta}$$

$$\frac{T_1 - 0.596\,(26.76)^2}{T_2 - 0.596\,(26.76)^2} = e^{0.3 \times 2.055 \times \mathrm{cosec}\, 17°} = 8.237$$

or $$T_1 - 8.237\, T_2 = -3088.68 \qquad \ldots (ii)$$

Solving (i) and (ii), we get $T_1 =$ **958.45 N** and $T_2 =$ **491.33 N** **Ans.**

10. *Calculation of stress induced :*

Consulting Table 2.3, cross-sectional area of D section = 475 mm²

$$\therefore \quad \text{Stress induced} = \frac{\text{Maximum tension}}{\text{Cross-sectional area}} = \frac{958.45}{475}$$

$$= \mathbf{2.02\ N/mm^2} \quad \textbf{Ans.}$$

Example 4.1 ***A truck equipped with a 9.5 kW engine uses a roller chain as the final drive to the rear axle. The driving sprocket runs at 900 r.p.m. and the driven sprocket at 400 r.p.m. with a centre distance of approximately 600 mm. Select the roller chain.***

Given Data : $N = 9.5$ kW ; $N_1 = 900$ r.p.m. ; $N_2 = 400$ r.p.m. ; $a_0 = 600$ mm.

To find : Select (*i.e.*, design) the roller chain.

Chain drive

☺ ***Solution :***

1. Determination of the transmission ratio (i) :

$$\text{Transmission ratio, } i = \frac{N_1}{N_2} = \frac{900}{400} = 2.25$$

(Since the transmission ratio can be calculated from the given data, therefore we need not to consult Table 4.2)

2. Selection of number of teeth on the driver sprocket (z_1) :

From Table 4.3, $z_1 = 27$ (for $i = 2$ to 3) is selected.

3. Determination of number of teeth on the driven sprocket (z_2) :

$$z_2 = i \times z_1 = 2.25 \times 27 = 60.75 \approx 61$$

Recommended value, $z_{2max} = 100$ to 120

$\therefore$ $z_2 = $ **61** is satisfactory.

4. Selection of standard pitch (p) :

We know that Centre distance, $a = (30 - 50)\,p$

$$\therefore \quad \text{Maximum pitch, } p_{max} = \frac{a}{30} = \frac{600}{30} = 20 \text{ mm}$$

$$\text{and} \quad \text{Minimum pitch, } p_{min} = \frac{a}{50} = \frac{600}{50} = 12 \text{ mm}$$

Any standard pitch between 12 mm and 20 mm can be chosen. But to get a quicker solution, it is always preferred to take the standard pitch closer to p_{max}. Refer Table 4.4.

$\therefore$ Standard pitch, p = **15.875 mm** is chosen.

5. Selection of the chain :

Assume the chain to be duplex. Consulting Table 4.5, the selected ***chain number*** is ***10A-2 / DR50.***

6. Calculation of total load on the driving side of the chain (P_T) :

(i) Tangential force (P_t) :

$$P_t = \frac{1020\,N}{v}$$

where N = Transmitted power in kW = 9.5 kW

v = Chain velocity in m/s

$$= \frac{z_1 \times p \times N_1}{60 \times 1000} = \frac{27 \times 15.875 \times 900}{60 \times 1000} = 6.43 \text{ m/s}$$

$$\therefore \quad P_t = \frac{1020\,N}{v} = \frac{1020 \times 9.5}{6.43} = 1507 \text{ N}$$

(ii) Centrifugal tension (P_c) :

$$P_c = mv^2$$

From Table 4.5, $m = 1.78$ kg/m

$\therefore$ $P_c = 1.78\,(6.43)^2 = 73.59$ N

(iii) Tension due to sagging (P_s) :

$$P_s = k \cdot w \cdot a$$

From Table 4.6, $k = 6$ (for horizontal)

$w = mg = 1.78 \times 9.81 = 17.46$ N

a = Initial centre distance = 0.6 m

$\therefore \quad P_s = 6 \times 17.46 \times 0.6 = 62.82$ N

(iv) Total load (P_T) : $\quad P_T = P_t + P_c + P_s$

$$= 1507 + 73.59 + 62.82 = 1643.4 \text{ N}$$

7. Calculation of service factor (k_s) :

We know that the service factor,

$$k_s = k_1 \cdot k_2 \cdot k_3 \cdot k_4 \cdot k_5 \cdot k_6$$

From Table 4.7, $k_1 = 1.25$ (for load with mild shocks)

From Table 4.8, $k_2 = 1$ (for adjustable supports)

From Table 4.9, $k_3 = 1$ ($\because$ we have used $a_p = (30 \text{ to } 50)\, p$)

From Table 4.10, $k_4 = 1$ (for horizontal drive)

From Table 4.11, $k_5 = 1$ (for drop lubrication)

From Table 4.12, $k_6 = 1.25$ (for 16 hours / day running)

$\therefore \quad k_s = 1.25 \times 1 \times 1 \times 1 \times 1 \times 1.25 = 1.5625$

8. Calculation of design load :

$$\text{Design load} = P_T \times k_s = 1643.4 \times 1.5625 = 2567.8 \text{ N}$$

9. Calculation of working factor of safety (FS_w) :

$$FS_w = \frac{\text{Breaking load Q from Table 4.5}}{\text{Design load}} = \frac{44400}{2567.8} = 17.29$$

10. Check for factor of safety :

Consulting Table 4.13, for smaller sprocket speed of 900 r.p.m. and pitch 15.875 mm, the required minimum factor of safety is 11. Therefore the working factor of safety is greater than the recommended minimum factor of safety. Thus ***the design is safe and satisfactory.***

11. Check for the bearing stress in the roller :

We know that $\quad \sigma_{roller} = \dfrac{P_t \times k_s}{A}$; where $A = 140$ mm^2 from Table 4.5.

$$= \frac{1507 \times 1.5625}{140} = 16.8 \text{ N/mm}^2$$

Consulting Table 4.14, for smaller sprocket speed of 900 r.p.m. and pitch 15.875 mm, the allowable bearing stress is 22.4 N/mm². Therefore the induced stress is less than the allowable bearing stress. Thus ***the design is safe and satisfactory.***

12. Calculation of length of chain (L) :

$$\text{Number of links, } l_p = 2\,a_p + \left(\frac{z_1 + z_2}{2}\right) + \frac{[\,(z_2 - z_1)/2\pi\,]^2}{a_p}$$

where $$a_p = \frac{a_0}{p} = \frac{\text{Centre distance}}{\text{pitch}} = \frac{600}{15.875} = 37.795$$

$$\therefore \quad l_p = 2\,(37.795) + \left(\frac{27 + 61}{2}\right) + \frac{[\,(61 - 27)/2\pi\,]^2}{37.795} = 120.36$$

$$\approx 122 \text{ links (rounded off to an even number)}$$

$\therefore$ Actual length of chain, $L = l_p \times p = 122 \times 15.875 =$ **1936.75 mm**

13. Calculation of exact centre distance (a) :

We know that $$a = \frac{e + \sqrt{e^2 - 8\,M}}{4} \times p$$

where $$e = l_p - \left(\frac{z_1 + z_2}{2}\right) = 122 - \left(\frac{27 + 61}{2}\right) = 78$$

and $$M = \left(\frac{z_2 - z_1}{2\pi}\right)^2 = \left(\frac{61 - 27}{2\pi}\right)^2 = 29.28$$

$$\therefore \quad a = \frac{78 + \sqrt{78^2 - 8 \times 29.28}}{4} \times 15.875 = 613.11 \text{ mm}$$

Decrement in centre distance for an initial sag $= 0.01\,a = 0.01\,(613.11) = 6.1311$ mm

$\therefore$ Exact centre distance $= 613.11 - 6.1311 =$ **606.978 mm**

14. Calculation of sprocket diameters :

Smaller sprocket :

$$\text{Pcd of smaller sprocket, } d_1 = \frac{p}{\sin(180/z_1)}$$

$$= \frac{15.875}{\sin(180/27)} = \mathbf{136.74\ mm}$$

and Sprocket outside diameter, $d_{01} = d_1 + 0.8\,d_r$

where d_r = Diameter of roller, from Table 4.5 = 10.16 mm

$$\therefore \quad d_{01} = 136.74 + 0.8 \times 10.16 = \mathbf{144.868\ mm}$$

chain

Larger sprocket :

$$\text{Pcd of larger sprocket, } d_2 = \frac{p}{\sin(180/z_2)} = \frac{15.875}{\sin(180/61)}$$

$$= \mathbf{308.38\ mm}$$

and Sprocket outside diameter, $d_{02} = d_2 + 0.8\, d_r$

$$= 308.38 + 0.8 \times 10.16 = \mathbf{316.51\ mm}$$

Example 4.2 ***The transporter of a heat treatment furnace is driven by a 4.5 kW, 1440 r.p.m. induction motor through a chain drive with a speed reduction ratio of 2.4. The transmission is horizontal with bath type of lubrication. Rating is continuous with 3 shifts per day. Design the complete chain drive.***

Chain drive

Given Data : N = 4.5 kW ; N_1 = 1440 r.p.m. ; i = 2.4.

To find : Design the chain drive.

☺ ***Solution :***

1. ***Transmission ratio,*** i = 2.4 (Given) $\therefore\ N_2 = \frac{N_1}{i} = \frac{1440}{2.4} = 600$ r.p.m.

2. ***To find z_1 :*** From Table 4.3, $z_1 = \mathbf{27}$ (for $i = 2$ to 3) is chosen.

3. ***To find z_2 :*** $z_2 = i \times z_1 = 2.4 \times 27 = 64.8 \approx 65$

   Recommended $z_{2\,max}$ = 100 to 120. $\therefore\ z_2 = \mathbf{65}$ is satisfactory.

4. ***Standard pitch (p) :*** Since the centre distance is not given, we have to assume the initial centre distance, say $a = 500$ mm.

We know that $a = (30 - 50)\, p$

$$\therefore \quad p_{max} = \frac{a}{30} = \frac{500}{30} = 16.6 \text{ mm}$$

$$\text{and} \quad p_{min} = \frac{a}{50} = \frac{500}{50} = 10 \text{ mm}$$

From Table 4.4, in between 10 and 16.6 mm, a standard pitch, $p = \mathbf{15.875}$ mm is chosen.

5. ***Selection of chain :*** Assume the chain to be ***simplex.***

From Table 4.5, the ***10A-1 / R50 chain number*** is chosen.

6. ***Calculation of total load on the driving side (P_T) :***

$$P_T = P_t + P_c + P_s$$

(i) $$P_t = \frac{1020\, N}{v}$$

where N = Transmitted power in kW = 4.5 kW (Given)

$$v = \text{Velocity of chain in m/s}$$

$$= \frac{z_1 \times p \times N_1}{60 \times 1000} = \frac{27 \times 15.875 \times 1440}{60 \times 1000} = 10.287 \text{ m/s}$$

$\therefore$ $$P_t = \frac{1020 \times 4.5}{10.287} = 446.19 \text{ N}$$

(ii) $$P_c = mv^2$$

From Table 4.5, $m = 1.01$ kg/m

$\therefore$ $$P_c = 1.01\,(10.287)^2 = 106.88 \text{ N}$$

(iii) $$P_s = k \cdot w \cdot a$$

From Table 4.6, $k = 6$ (for horizontal)

$w = mg = 1.01 \times 9.81 = 9.908$ N/m and $a = 0.5$ m

$\therefore$ $$P_s = 6 \times 9.908 \times 0.5 = 29.72 \text{ N}$$

(iv) $\therefore$ Total load, $P_T = 466.19 + 106.88 + 29.72 = 582.79$ N

7. Service factor : $k_s = k_1 \cdot k_2 \cdot k_3 \cdot k_4 \cdot k_5 \cdot k_6$

From Table 4.7, $k_1 = 1.25$ (for load with mild shocks)

From Table 4.8, $k_2 = 1$ (for adjustable supports)

From Table 4.9, $k_3 = 1$ (since we have used $a = (30 \text{ to } 50)\,p$)

From Table 4.10, $k_4 = 1$ (for horizontal drive)

From Table 4.11, $k_5 = 0.8$ (for bath type lubrication)

From Table 4.12, $k_6 = 1.5$ (for continuous running *i.e.*, 3 shifts / day)

$\therefore$ $$k_s = 1.25 \times 1 \times 1 \times 1 \times 0.8 \times 1.5 = 1.5$$

8. Design load $= P_T \times k_s = 582.79 \times 1.5 = 874.19$ N

9. Working factor of safety $= \dfrac{\text{Breaking load Q from Table 4.5}}{\text{Design load}}$

$$FS_w = \frac{22200}{874.19} = 25.39$$

From Table 4.13, for smaller sprocket speed 1440 r.p.m. and pitch 15.875 mm, the recommended minimum value of factor of safety (n') is 13.2. Since the working factor of safety is greater than the recommended minimum value of factor of safety, therefore ***the design is safe and satisfactory.***

10. Bearing stress in the roller : From Table 4.5, $A = 70 \text{ mm}^2$

$$\sigma = \frac{P_t \times k_s}{A} = \frac{446.19 \times 1.5}{70} = 9.56 \text{ N/mm}^2$$

From Table 4.14, for smaller sprocket speed 1440 r.p.m. and pitch 15.875 mm, the allowable bearing stress is 18.5 N/mm². Since the induced stress is less than the allowable bearing stress, ***the design is safe and satisfactory.***

11. Actual length of chain (L) :

$$\text{Number of links, } l_p = 2\,a_p + \left(\frac{z_1 + z_2}{2}\right) + \frac{[(z_2 - z_1)/2\pi]^2}{a_p}$$

where $$a_p = \frac{a_0}{p} = \frac{500}{15.875} = 31.496$$

$$\therefore \quad l_p = 2\,(31.496) + \left(\frac{27 + 65}{2}\right) + \frac{[(65 - 27)/2\pi]^2}{31.496}$$

$$= 110.153 \approx 112 \qquad \text{(rounded off to an even number)}$$

$$\therefore \quad \text{Actual length of chain, L} = l_p \times p = 112 \times 15.875 = \mathbf{1778\ mm}$$

12. Exact centre distance :

$$a = \frac{e + \sqrt{e^2 - 8\,M}}{4} \times p$$

where $$e = l_p - \left(\frac{z_1 + z_2}{2}\right) = 112 - \left(\frac{27 + 65}{2}\right) = 66$$

and $$M = \left[\frac{(z_2 - z_1)}{2\pi}\right]^2 = \left(\frac{65 - 27}{2\pi}\right)^2 = 36.57$$

$$\therefore \quad a = \frac{66 + \sqrt{66^2 - 8 \times 36.57}}{4} \times 15.875 = 514.92\ \text{mm}$$

Decrement in centre distance for an initial sag, $\Delta a = 0.01\,a = 5.149$ mm

$$\therefore \quad \text{Exact centre distance} = 514.92 - 5.149 = \mathbf{509.77\ mm}$$

13. Sprocket diameters :

For smaller sprocket :

$$\text{Pcd} = \frac{p}{\sin(180/z_1)} = \frac{15.875}{\sin(180/27)} = \mathbf{136.74\ mm}$$

and Sprocket outside diameter, $d_{01} = d_1 + 0.8\,d_r$

From Table 4.5, d_r = Diameter of roller = 10.16 mm

$$\therefore \quad d_{01} = 136.74 + 0.8 \times 10.16 = \mathbf{144.87\ mm}$$

For larger sprocket :

$$\text{Pcd} = \frac{p}{\sin(180/z_2)} = \frac{15.875}{\sin(180/65)} = \mathbf{328.58\ mm}$$

and Sprocket outside diameter, $d_{02} = d_2 + 0.8\,d_r = 328.58 + 0.8 \times 10.16$

$$= \mathbf{336.71\ mm}$$

Chain drive

Sem
16 Mark

Example 3.1 ***Design a wire rope for an elevator in a building 60 metres high and for a total load of 20 kN. The speed of the elevator is 4 m/sec and the full speed is reached in 10 seconds.***

Wire rope

Given Data : Height = 60 m ; $W = 20 \text{ kN} = 20 \times 10^3 \text{ N}$; $v = 4$ m/sec = 240 m/min; $t = 10$ sec.

To find : Design a wire rope.

☺ ***Solution :***

1. Selection of suitable wire rope : Given that the wire rope is used for an elevator, *i.e.*, for hoisting purpose. So lets use **6 × 19 rope** (refer Table 3.1).

2. Calculation of design load : Assuming a larger factor of safety of 15, the design load is calculated.

$$\text{Design load} = \text{Load to be lifted} \times \text{Assumed factor of safety}$$

$$= 20 \times 15 = \mathbf{300\ kN}$$

3. Selection of wire rope diameter (d) : From Table 3.4, taking the design load as the breaking strength, the wire rope diameter is selected as 25 mm.

$\therefore$ d = **25 mm** for σ_u = 1600 to 1750 N/mm² and breaking strength = 340 kN.

4. Calculation of sheave diameter (D) : From Table 3.5, for 6 × 19 rope and class 4,

$$\frac{D_{min}}{d} = 27 \text{ (for velocity upto 50 m/min)}$$

Since the given lifting speed is 240 m/min (= 4 m/s), therefore D_{min}/d ratio should be modified. Thus for every additional speed of 50 m/min, D_{min}/d ratio has to be increased by 8%.

$$\therefore \quad \text{Modified } \frac{D_{min}}{d} = 27 \times (1.08)^{5-1} = 36.73 \text{ say } 40. \qquad \left[\because \frac{240}{50} \approx 5\right]$$

The sheave diameter, $D = 40 \times d = 40 \times 25 = \mathbf{1000\ mm}$

5. Selection of the area of useful cross-section of the rope (A) : From Table 3.6, for 6 × 19 rope,

$$A = 0.4\, d^2 = 0.4\,(25)^2 = \mathbf{250\ mm^2}$$

6. Calculation of wire diameter (d_w) :

$$\text{Wire diameter, } d_w = \frac{d}{1.5\sqrt{i}}$$

where i = Number of strands × Number of wires in each strand

$$= 6 \times 19 = 114$$

$$\therefore \quad d_w = \frac{25}{1.5\sqrt{114}} = 1.56 \text{ mm}$$

7. Selection of weight of rope (W_r) :

From Table 3.4,

$$\text{Approximate mass} = 2.41 \text{ kg / m}$$

$$\therefore \quad \text{Weight of rope / m} = 2.41 \times 9.81 = 23.6 \text{ N/m}$$

and $\quad$ Weight of rope, $W_r = 23.6 \times 60 = \mathbf{1416\ N}$

8. Calculation of various loads :

(i) $\quad$ Direct load, $W_d = W + W_r = 20000 + 1416 = 21416$ N

(ii) Bending load, $W_b = \sigma_b \times A = \dfrac{E_r \cdot d_w}{D} \times A$

$$= \frac{0.84 \times 10^5 \times 1.56}{1000} \times 250 = 32760 \text{ N}$$

[Take $E_r = 0.84 \times 10^5$ N / mm²]

(iii) Acceleration load, $W_a = \left(\dfrac{W + W_r}{g}\right) a$

where a = Acceleration of the load $= \dfrac{v_2 - v_1}{t_1} = \dfrac{4-0}{10}$

$= 0.4$ m/s²

$\therefore$ $W_a = \left(\dfrac{20000 + 1416}{9.81}\right) 0.4 = 873.23$ N

(iv) Starting load (W_{st}) :

When there is no slack in the rope, starting load is given by

$$W_{st} = 2 \cdot W_d = 2\,(W + W_r) = 2\,(20000 + 1416) = 42832 \text{ N}$$

9. Calculation of effective loads on the rope :

(i) Effective load during normal working, $W_{en} = W_d + W_b$

$= 21416 + 32760 = 54176$ N

(ii) Effective load during acceleration of the load, $W_{ea} = W_d + W_b + W_a$

$= 21416 + 32760 + 873.23$

$= 55049.23$ N

(iii) Effective load during starting, $W_{est} = W_b + W_{st}$

$= 32760 + 42832 = 75592$ N

10. Calculation of working factor of safety (FS_w) :

$$\text{Working factor of safety} = \frac{\text{Breaking load from Table 3.4 for the selected rope}}{\text{Effective load during acceleration } (W_{ea})}$$

$$= \frac{340000}{55049.23} = 6.176$$

11. Check for safe design : From Table 3.2, for hoists and class 4, the recommended factor of safety = 6.

Since the working factor of safety is greater than the recommended factor of safety therefore the ***design is safe.***

wire rope