- Belt and Chain Drives
- (Flexible Drive Elements)

Why Flexible Drives?

+

- Long Distances Between Shafts
- Less Expensive
- Adjustable Centers
- Tolerates some mis alignment better than gears

-

- Not as compact as gears
- Some speed limits
- Power and torque limits

Common Belt Types

Type	**Pulley**
1. Flat (conveyor belts)	Crowned pulley
2. Round (O-ring)	Grooved pulley
3. V-belt	Flanged pulleys
4. Timing (toothed) (no stretch or slip)	Cogged pulley
5. Proprietary belt designs	

Characteristics of Some Common Belt Types

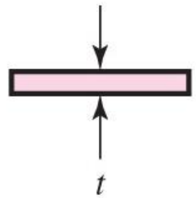
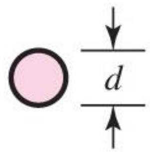
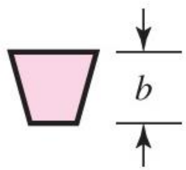
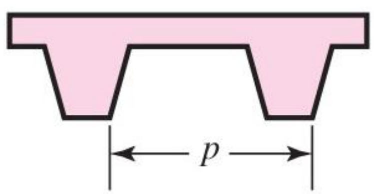
Belt Type	Figure	Joint	Size Range	Center Distance
Flat		Yes	$t = \begin{cases} 0.03 \text{ to } 0.20 \text{ in} \\ 0.75 \text{ to } 5 \text{ mm} \end{cases}$	No upper limit
Round		Yes	$d = \frac{1}{8} \text{ to } \frac{3}{4} \text{ in}$	No upper limit
V		None	$b = \begin{cases} 0.31 \text{ to } 0.91 \text{ in} \\ 8 \text{ to } 19 \text{ mm} \end{cases}$	Limited
Timing		None	$p = 2 \text{ mm and up}$	Limited

Table 17–1

Flat Belt Drive

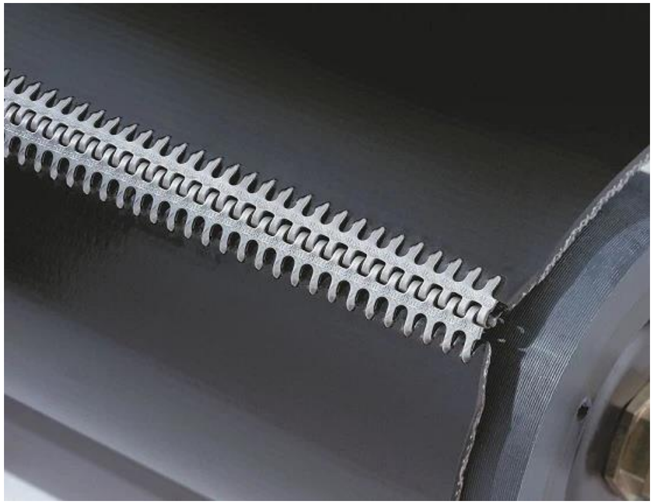
Better for high speed applications,
rather than power.

Good for tight clearances

Can allow slip, to prevent breakage

Flat Belts as Conveyer Belts

Crowned Pulleys for Tracking

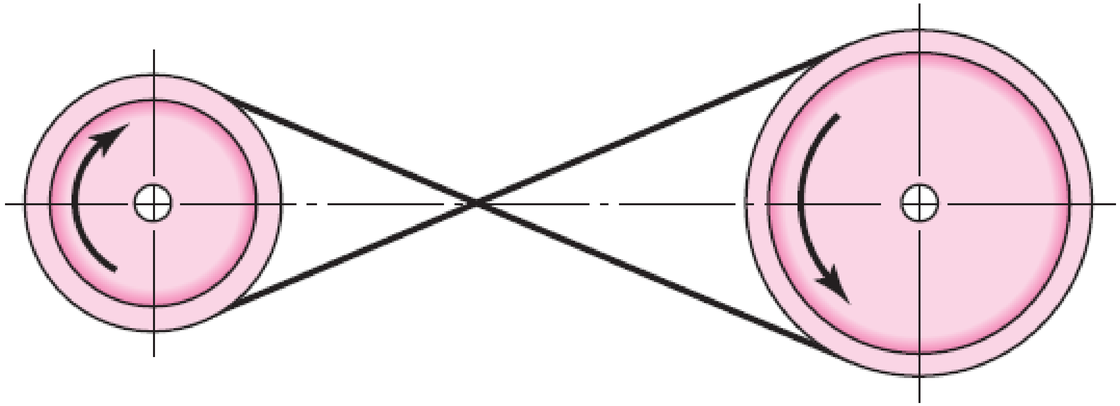


Usually seamed/stitched

O-Ring Drive

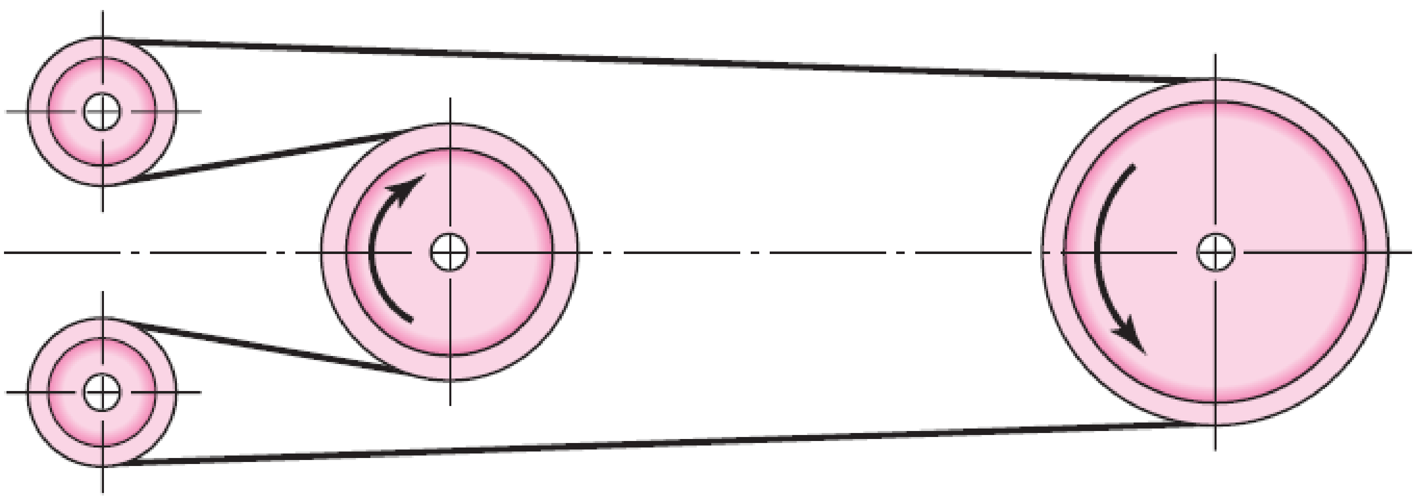
Good for low cost.
Good for driving multiple shafts

Reversing Belts



(b)

Typically
O-Ring
Drives



(c)

Fig.17–2

Timing Belts

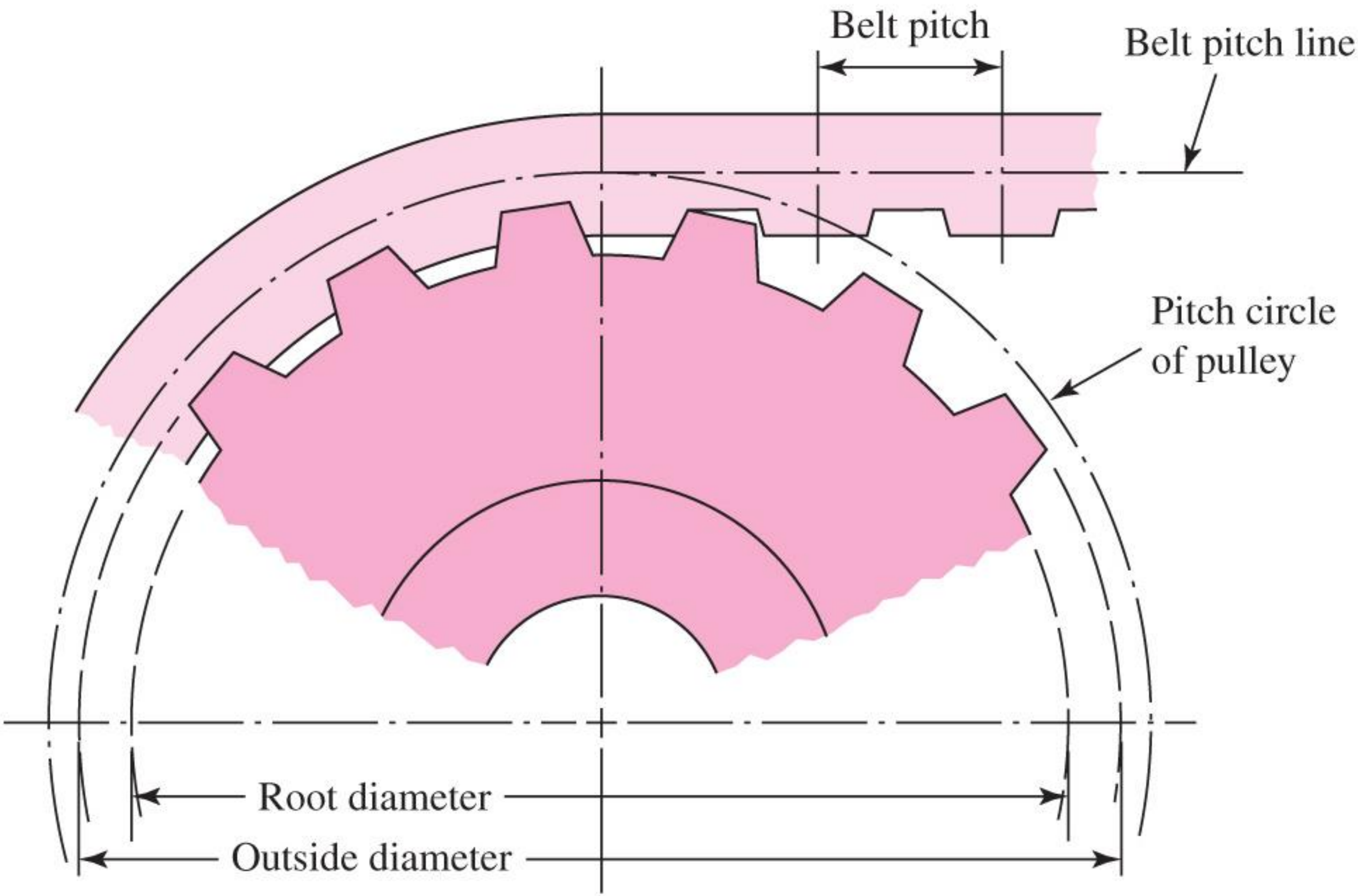


Fig.17–15

Flat-belt with Out-of-plane Pulleys

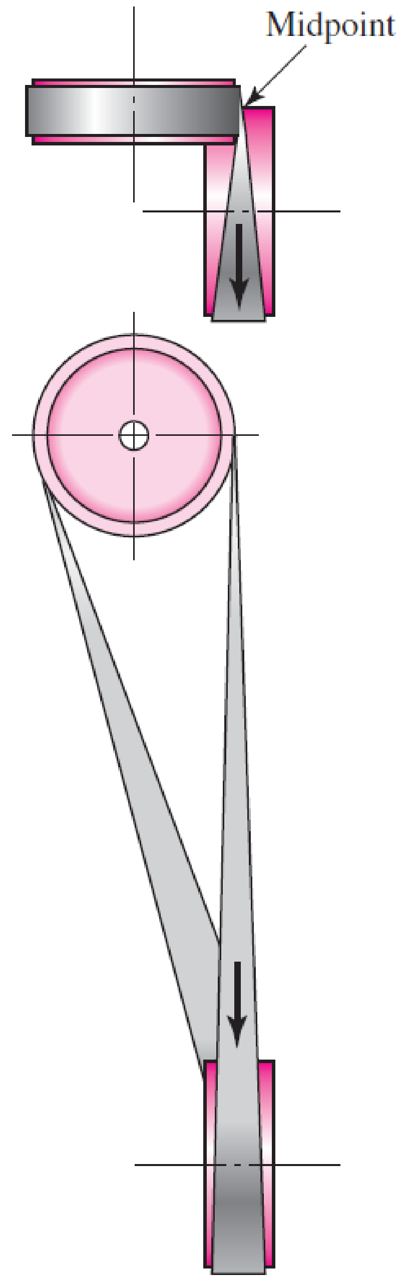


Fig.17–3

Variable-Speed Belt Drives

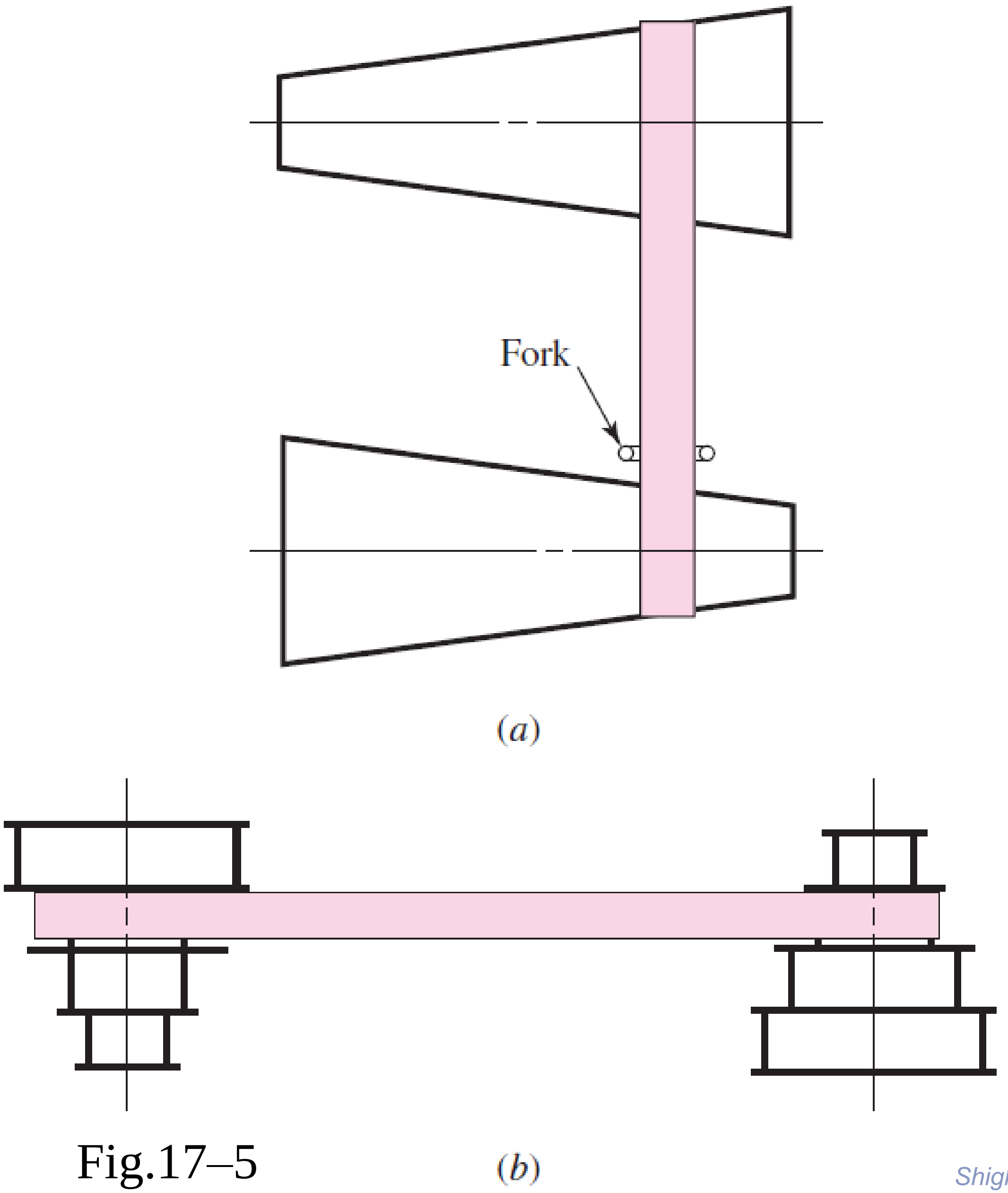


Fig.17–5

Flat-Belt Geometry – Open Belt

$$\theta_d = \pi - 2\sin^{-1}\frac{D-d}{2C}$$

$$\theta_D = \pi + 2\sin^{-1}\frac{D-d}{2C}$$

$$L = \sqrt{4C^2 - (D-d)^2} + \tfrac{1}{2}(D\theta_D + d\theta_d)$$

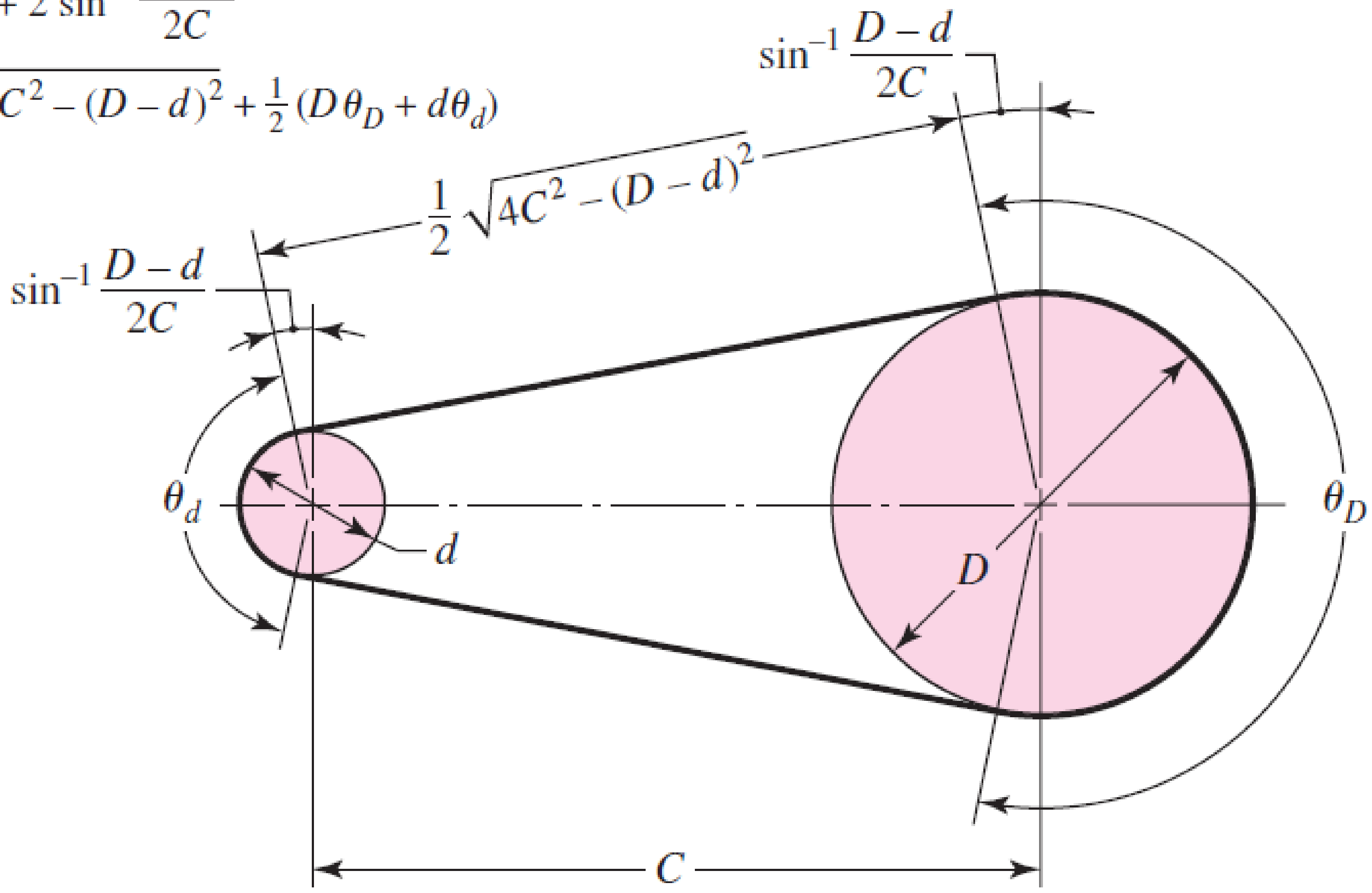


Fig.17–1*a*

Free Body of Infinitesimal Element of Flat Belt

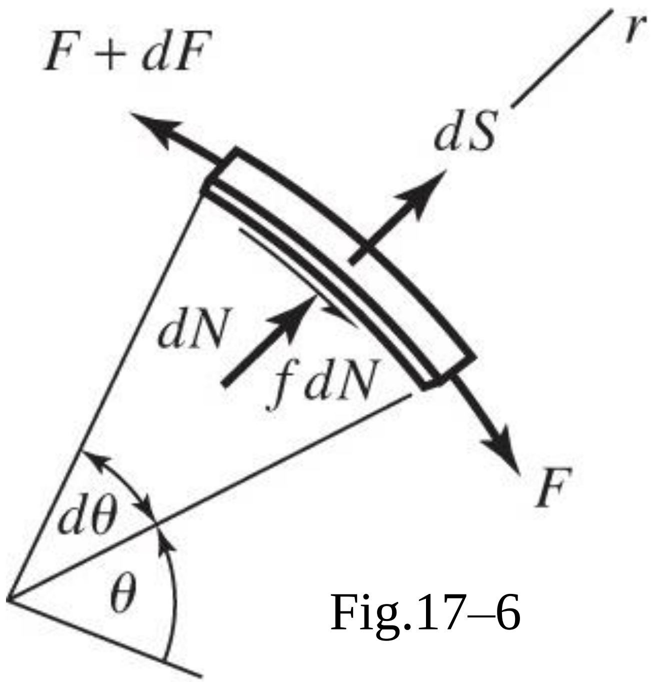


Fig.17–6

$$dS = (mr\,d\theta)r\omega^2 = mr^2\omega^2\,d\theta = mV^2\,d\theta = F_c\,d\theta \qquad (a)$$

$$\sum F_r = -(F + dF)\frac{d\theta}{2} - F\frac{d\theta}{2} + dN + dS = 0$$

$$dN = F\,d\theta - dS \qquad (b)$$

Free Body of Infinitesimal Element of Flat Belt

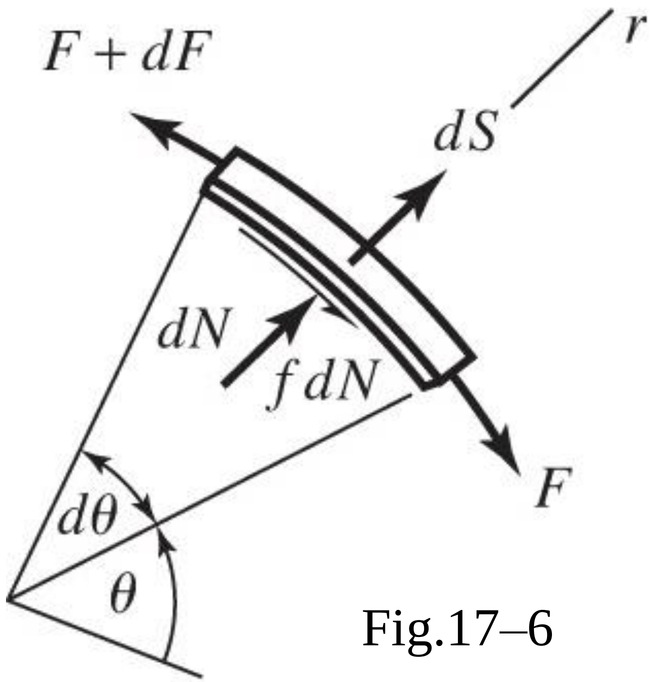


Fig.17–6

$$\sum F_t = -f\,dN - F + (F + dF) = 0$$

$$dF = f\,dN = f F\,d\theta - f\,dS = f F\,d\theta - f m r^2 \omega^2\,d\theta$$

$$\frac{dF}{d\theta} - f F = -f m r^2 \omega^2 \qquad (c)$$

Analysis of Flat Belt, The Belting Equation

$$\frac{dF}{d\theta} - fF = -fmr^2\omega^2 \qquad \text{(c)}$$

$$F = A\exp(f\theta) + mr^2\omega^2 \qquad \text{(d)}$$

$$F \text{ at } \theta = 0 \text{ equals } F_2 \text{ gives } A = F_2 - mr^2\omega^2$$

$$F = (F_2 - mr^2\omega^2)\exp(f\theta) + mr^2\omega^2 \qquad \text{(17–5)}$$

$$F|_{\theta=\phi} = F_1 = (F_2 - mr^2\omega^2)\exp(f\phi) + mr^2\omega^2 \qquad \text{(17–6)}$$

$$\boxed{\frac{F_1 - mr^2\omega^2}{F_2 - mr^2\omega^2} = \frac{F_1 - F_c}{F_2 - F_c} = \exp(f\phi)} \qquad \text{(17–7)}$$

The Belting Equation

$$\boxed{F_c = mr^2\omega^2}$$

$$F_1 - F_2 = (F_1 - F_c)\frac{\exp(f\phi) - 1}{\exp(f\phi)} \qquad \text{(17–8)}$$

Fc = Hoop Tension Due to Centrifugal Force

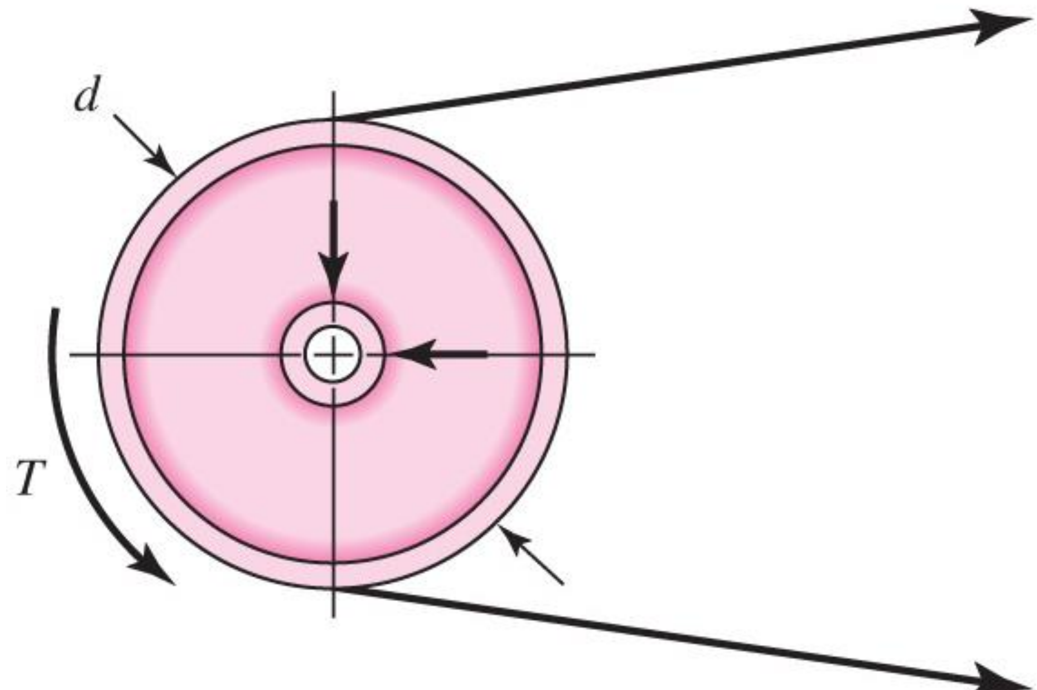
$$\boxed{F_c = mr^2\omega^2}$$

$$F_c = \frac{w}{g}\left(\frac{V}{60}\right)^2 = \frac{w}{32.17}\left(\frac{V}{60}\right)^2 \qquad (e)$$

$w = 12\gamma bt$ lbf/ft where b and t are in inches

$V = \pi\, dn/12$ ft/min

Forces and Torques on a Pulley



$$F_1 = F_i + F_c + \Delta F/2 = F_i + F_c + \frac{T}{d}$$

$$F_2 = F_i + F_c - \Delta F/2 = F_i + F_c - \frac{T}{d}$$

Fig.17–7

F_i = initial tension

F_c = hoop tension due to centrifugal force

$\Delta F/2$ = tension due to the transmitted torque T

d = diameter of the pulley

Initial Tension

$$F_1 - F_2 = \frac{2T}{d} \qquad (h)$$

$$F_1 + F_2 = 2F_i + 2F_c$$

$$F_i = \frac{F_1 + F_2}{2} - F_c \qquad (i)$$

$$\frac{F_i}{T/d} = \frac{(F_1 + F_2)/2 - F_c}{(F_1 - F_2)/2} = \frac{F_1 + F_2 - 2F_c}{F_1 - F_2} = \frac{(F_1 - F_c) + (F_2 - F_c)}{(F_1 - F_c) - (F_2 - F_c)}$$

$$= \frac{(F_1 - F_c)/(F_2 - F_c) + 1}{(F_1 - F_c)/(F_2 - F_c) - 1} = \frac{\exp(f\phi) + 1}{\exp(f\phi) - 1}$$

$$F_i = \frac{T}{d}\,\frac{\exp(f\phi) + 1}{\exp(f\phi) - 1} \qquad (17\text{–}9)$$

Flat Belt Tensions

$$F_1 = F_i + F_c + \frac{T}{d} = F_c + F_i + F_i \frac{\exp(f\phi) - 1}{\exp(f\phi) + 1}$$

$$= F_c + \frac{F_i[\exp(f\phi) + 1] + F_i[\exp(f\phi) - 1]}{\exp(f\phi) + 1}$$

$$F_1 = F_c + F_i \frac{2\exp(f\phi)}{\exp(f\phi) + 1} \qquad (17\text{–}10)$$

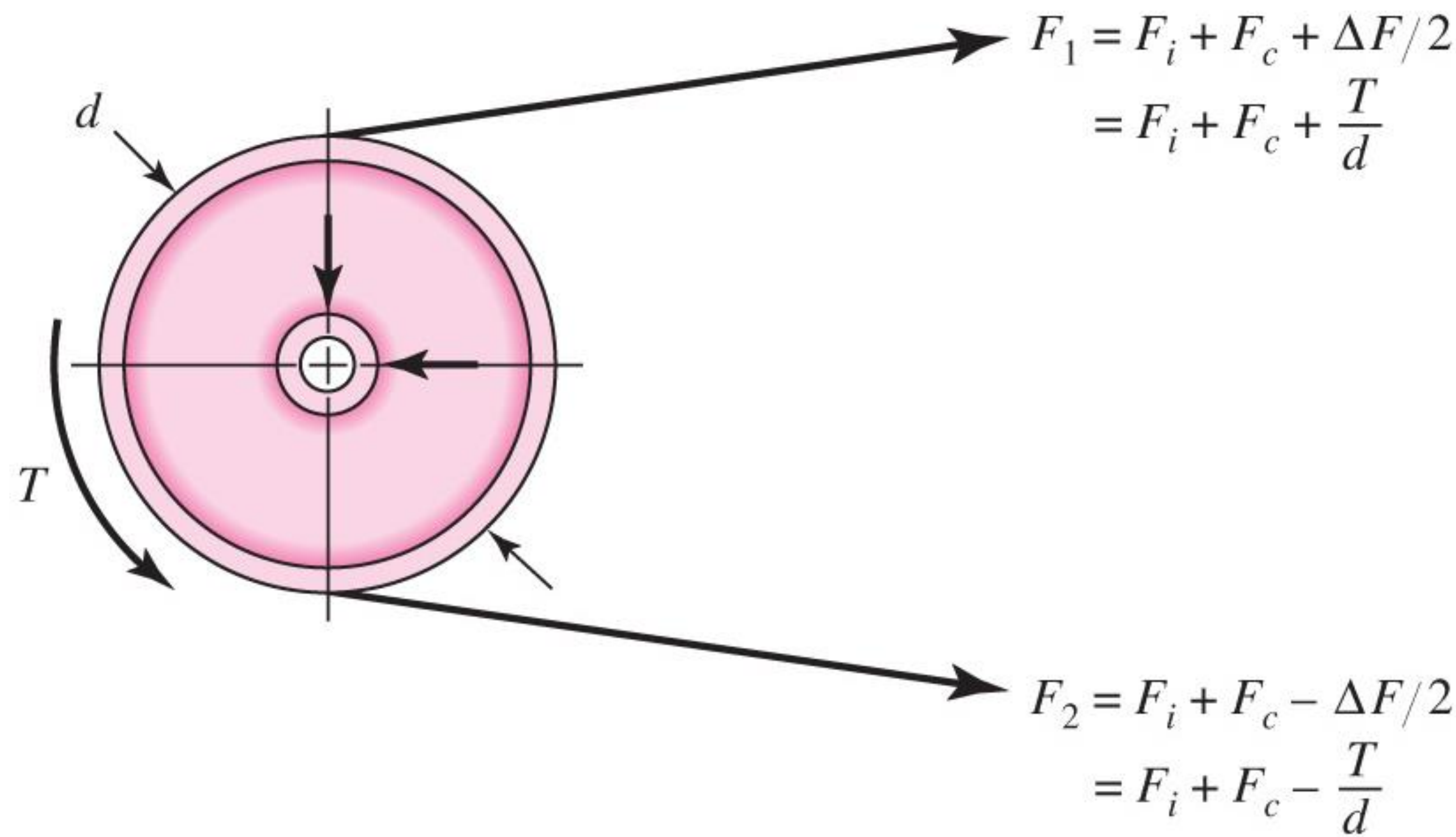
$$F_2 = F_i + F_c - \frac{T}{d} = F_c + F_i - F_i \frac{\exp(f\phi) - 1}{\exp(f\phi) + 1}$$

$$= F_c + \frac{F_i[\exp(f\phi) + 1] - F_i[\exp(f\phi) - 1]}{\exp(f\phi) + 1}$$

$$F_2 = F_c + F_i \frac{2}{\exp(f\phi) + 1} \qquad (17\text{–}11)$$

Transmitted Horsepower

$$H = \frac{(F_1 - F_2)V}{33\,000} \qquad (j)$$



Correction Factors for Belts, Based on Manufacturer Data

$$(F_1)_a = bF_aC_pC_v \tag{17–12}$$

where $(F_1)_a$ = allowable largest tension, lbf

b = belt width, in

F_a = manufacturer's allowed tension, lbf/in

C_p = pulley correction factor (Table 17–4)

C_v = velocity correction factor

Velocity Correction Factor C_v for Leather Belts

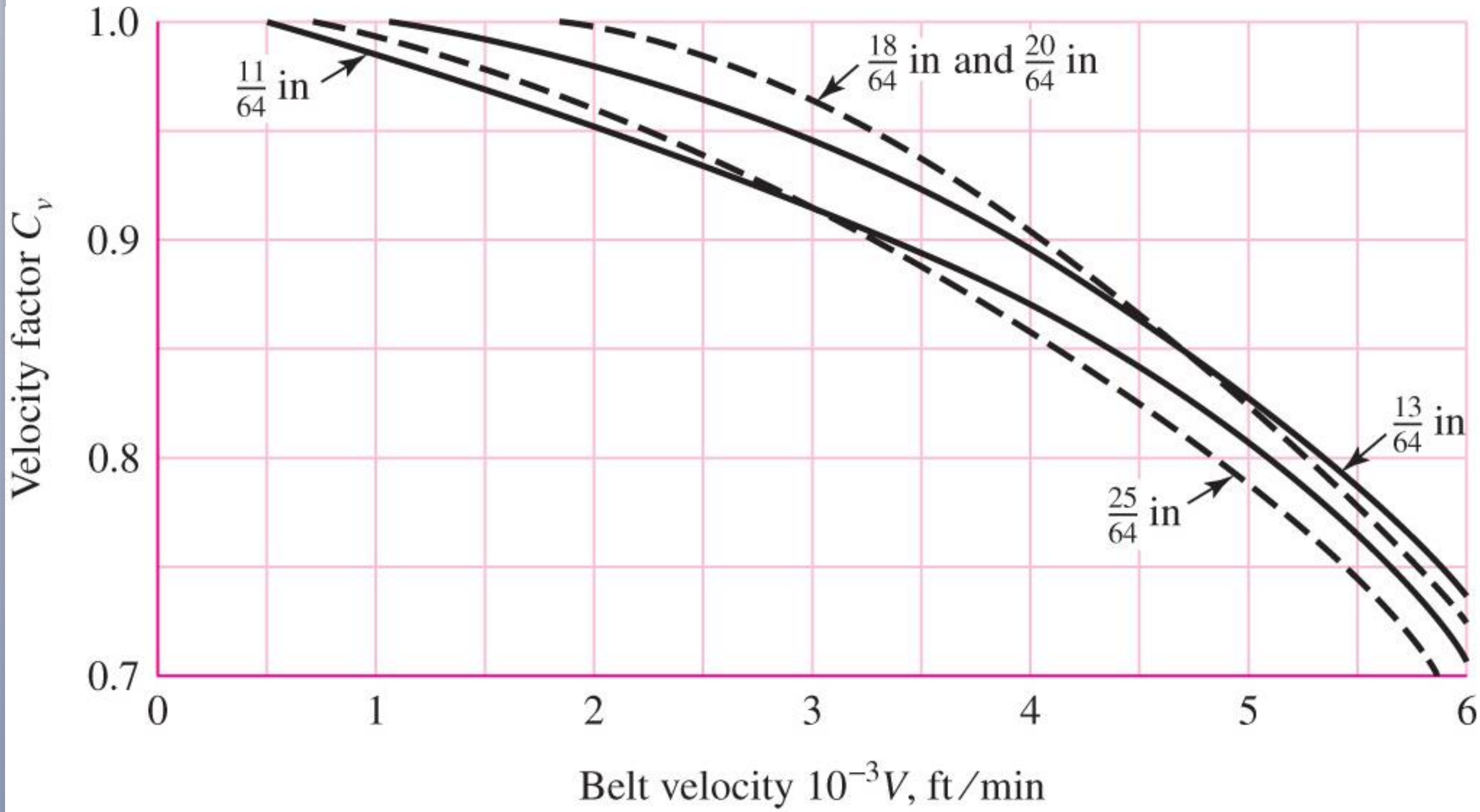


Fig.17–9

Pulley Correction Factor C_P for Flat Belts

Table 17–4

Pulley Correction Factor C_P for Flat Belts*

Material	Small-Pulley Diameter, in 1.6 to 4	4.5 to 8	9 to 12.5	14, 16	18 to 31.5	Over 31.5
Leather	0.5	0.6	0.7	0.8	0.9	1.0
Polyamide, F–0	0.95	1.0	1.0	1.0	1.0	1.0
F–1	0.70	0.92	0.95	1.0	1.0	1.0
F–2	0.73	0.86	0.96	1.0	1.0	1.0
A–2	0.73	0.86	0.96	1.0	1.0	1.0
A–3	—	0.70	0.87	0.94	0.96	1.0
A–4	—	—	0.71	0.80	0.85	0.92
A–5	—	—	—	0.72	0.77	0.91

*Average values of C_P for the given ranges were approximated from curves in the *Habasit Engineering Manual,* Habasit Belting, Inc., Chamblee (Atlanta), Ga.

Steps for Flat-Belt Analysis

1 Find $\exp(f\phi)$ from belt-drive geometry and friction

2 From belt geometry and speed find F_c

3 From $T = 63\,025 H_{\text{nom}} K_s n_d / n$ find necessary torque

4 From torque T find the necessary $(F_1)_a - F_2 = 2T/d$

5 From Tables 17–2 and 17–4, and Eq. (17–12) determine $(F_1)_a$.

6 Find F_2 from $(F_1)_a - [(F_1)_a - F_2]$

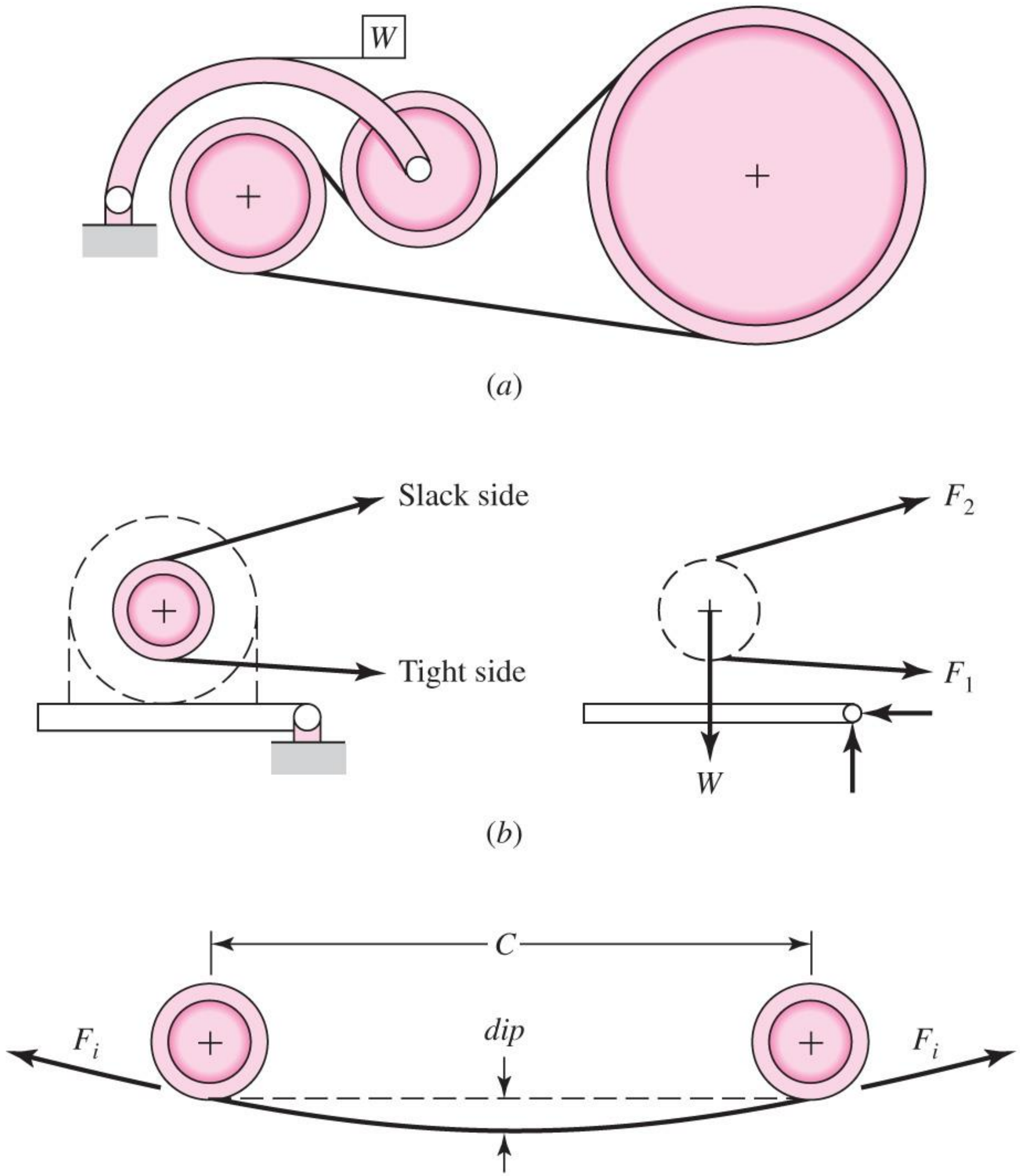
7 From Eq. (i) find the necessary initial tension F_i

8 Check the friction development, $f' < f$. Use Eq. (17–7) solved for f':

$$f' = \frac{1}{\phi} \ln \frac{(F_1)_a - F_c}{F_2 - F_c}$$

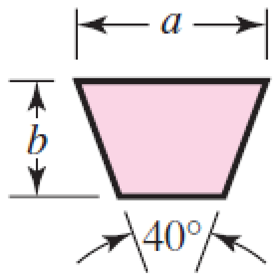
9 Find the factor of safety from $n_{fs} = H_a/(H_{\text{nom}} K_s)$

Belt-Tensioning Schemes



(c) Fig.17–11

Standard V-Belt Sections



Belt Section	Width a, in	Thickness b, in	Minimum Sheave Diameter, in	hp Range, One or More Belts
A	$\frac{1}{2}$	$\frac{11}{32}$	3.0	$\frac{1}{4}$–10
B	$\frac{21}{32}$	$\frac{7}{16}$	5.4	1–25
C	$\frac{7}{8}$	$\frac{17}{32}$	9.0	15–100
D	$1\frac{1}{4}$	$\frac{3}{4}$	13.0	50–250
E	$1\frac{1}{2}$	1	21.6	100 and up

Table 17–9

Inside Circumferences of Standard V-Belts

Section	Circumference, in
A	26, 31, 33, 35, 38, 42, 46, 48, 51, 53, 55, 57, 60, 62, 64, 66, 68, 71, 75, 78, 80, 85, 90, 96, 105, 112, 120, 128
B	35, 38, 42, 46, 48, 51, 53, 55, 57, 60, 62, 64, 65, 66, 68, 71, 75, 78, 79, 81, 83, 85, 90, 93, 97, 100, 103, 105, 112, 120, 128, 131, 136, 144, 158, 173, 180, 195, 210, 240, 270, 300
C	51, 60, 68, 75, 81, 85, 90, 96, 105, 112, 120, 128, 136, 144, 158, 162,173, 180, 195, 210, 240, 270, 300, 330, 360, 390, 420
D	120, 128, 144, 158, 162, 173, 180, 195, 210, 240, 270, 300, 330, 360,390, 420, 480, 540, 600, 660
E	180, 195, 210, 240, 270, 300, 330, 360, 390, 420, 480, 540, 600, 660

Table 17–10

Length Conversion Dimensions

Table 17–11

Length Conversion Dimensions (Add the listed quantity to the inside circumference to obtain the pitch length in inches).

Belt section	A	B	C	D	E
Quantity to be added	1.3	1.8	2.9	3.3	4.5

V-Belt Pitch Length and Center-to-Center Distance

$$L_p = 2C + \pi(D + d)/2 + (D - d)^2/(4C) \tag{17–16a}$$

$$C = 0.25\left\{\left[L_p - \frac{\pi}{2}(D + d)\right] + \sqrt{\left[L_p - \frac{\pi}{2}(D + d)\right]^2 - 2(D - d)^2}\right\} \tag{17–16b}$$

Horsepower Ratings of Standard V-Belts

Table 17–12

Belt Section	Sheave Pitch Diameter, in	Belt Speed, ft/min 1000	2000	3000	4000	5000
A	2.6	0.47	0.62	0.53	0.15	
	3.0	0.66	1.01	1.12	0.93	0.38
	3.4	0.81	1.31	1.57	1.53	1.12
	3.8	0.93	1.55	1.92	2.00	1.71
	4.2	1.03	1.74	2.20	2.38	2.19
	4.6	1.11	1.89	2.44	2.69	2.58
	5.0 and up	1.17	2.03	2.64	2.96	2.89
B	4.2	1.07	1.58	1.68	1.26	0.22
	4.6	1.27	1.99	2.29	2.08	1.24
	5.0	1.44	2.33	2.80	2.76	2.10
	5.4	1.59	2.62	3.24	3.34	2.82
	5.8	1.72	2.87	3.61	3.85	3.45
	6.2	1.82	3.09	3.94	4.28	4.00
	6.6	1.92	3.29	4.23	4.67	4.48
	7.0 and up	2.01	3.46	4.49	5.01	4.90
C	6.0	1.84	2.66	2.72	1.87	
	7.0	2.48	3.94	4.64	4.44	3.12
	8.0	2.96	4.90	6.09	6.36	5.52
	9.0	3.34	5.65	7.21	7.86	7.39
	10.0	3.64	6.25	8.11	9.06	8.89
	11.0	3.88	6.74	8.84	10.0	10.1
	12.0 and up	4.09	7.15	9.46	10.9	11.1
D	10.0	4.14	6.13	6.55	5.09	1.35
	11.0	5.00	7.83	9.11	8.50	5.62
	12.0	5.71	9.26	11.2	11.4	9.18
	13.0	6.31	10.5	13.0	13.8	12.2
	14.0	6.82	11.5	14.6	15.8	14.8
	15.0	7.27	12.4	15.9	17.6	17.0
	16.0	7.66	13.2	17.1	19.2	19.0
	17.0 and up	8.01	13.9	18.1	20.6	20.7
E	16.0	8.68	14.0	17.5	18.1	15.3
	18.0	9.92	16.7	21.2	23.0	21.5
	20.0	10.9	18.7	24.2	26.9	26.4
	22.0	11.7	20.3	26.6	30.2	30.5
	24.0	12.4	21.6	28.6	32.9	33.8
	26.0	13.0	22.8	30.3	35.1	36.7
	28.0 and up	13.4	23.7	31.8	37.1	39.1

Adjusted Power

$$H_a = K_1 K_2 H_{tab} \qquad (17\text{–}17)$$

where H_a = allowable power, per belt

K_1 = angle-of-wrap (ϕ) correction factor, Table 17–13

K_2 = belt length correction factor, Table 17–14

Angle of Wrap Correction Factor

$\frac{D-d}{C}$	θ, deg	K_1 VV	K_1 V Flat
0.00	180	1.00	0.75
0.10	174.3	0.99	0.76
0.20	166.5	0.97	0.78
0.30	162.7	0.96	0.79
0.40	156.9	0.94	0.80
0.50	151.0	0.93	0.81
0.60	145.1	0.91	0.83
0.70	139.0	0.89	0.84
0.80	132.8	0.87	0.85
0.90	126.5	0.85	0.85
1.00	120.0	0.82	0.82
1.10	113.3	0.80	0.80
1.20	106.3	0.77	0.77
1.30	98.9	0.73	0.73
1.40	91.1	0.70	0.70
1.50	82.8	0.65	0.65

Table 17–13

Belt-Length Correction Factor

	Nominal Belt Length, in				
Length Factor	A Belts	B Belts	C Belts	D Belts	E Belts
0.85	Up to 35	Up to 46	Up to 75	Up to 128	
0.90	38–46	48–60	81–96	144–162	Up to 195
0.95	48–55	62–75	105–120	173–210	210–240
1.00	60–75	78–97	128–158	240	270–300
1.05	78–90	105–120	162–195	270–330	330–390
1.10	96–112	128–144	210–240	360–420	420–480
1.15	120 and up	158–180	270–300	480	540–600
1.20		195 and up	330 and up	540 and up	660

*Multiply the rated horsepower per belt by this factor to obtain the corrected horsepower.

Table 17–14

Belting Equation for V-Belt (Gates Belting)

$$\frac{F_1 - F_c}{F_2 - F_c} = \exp(0.5123\phi) \qquad (17\text{–}18)$$

Design Power for V-Belt

$$H_d = H_{\text{nom}} K_s n_d \tag{17–19}$$

where H_{nom} is the nominal power

K_s is the service factor given in Table 17–15

n_d is the design factor

Number of belts:

$$N_b \geq \frac{H_d}{H_a} \qquad N_b = 1, 2, 3, \ldots \tag{17–20}$$

where H_a = allowable power, per belt

V-Belt Tensions

$$F_c = K_c \left(\frac{V}{1000} \right)^2 \qquad (17\text{–}21)$$

where K_c is from Table 17–16

$$\Delta F = \frac{63\,025 H_d/N_b}{n(d/2)} \qquad (17\text{–}22)$$

$$F_1 = F_c + \frac{\Delta F \exp(f\phi)}{\exp(f\phi) - 1} \qquad (17\text{–}23)$$

$$F_2 = F_1 - \Delta F \qquad (17\text{–}24)$$

$$F_i = \frac{F_1 + F_2}{2} - F_c \qquad (17\text{–}25)$$

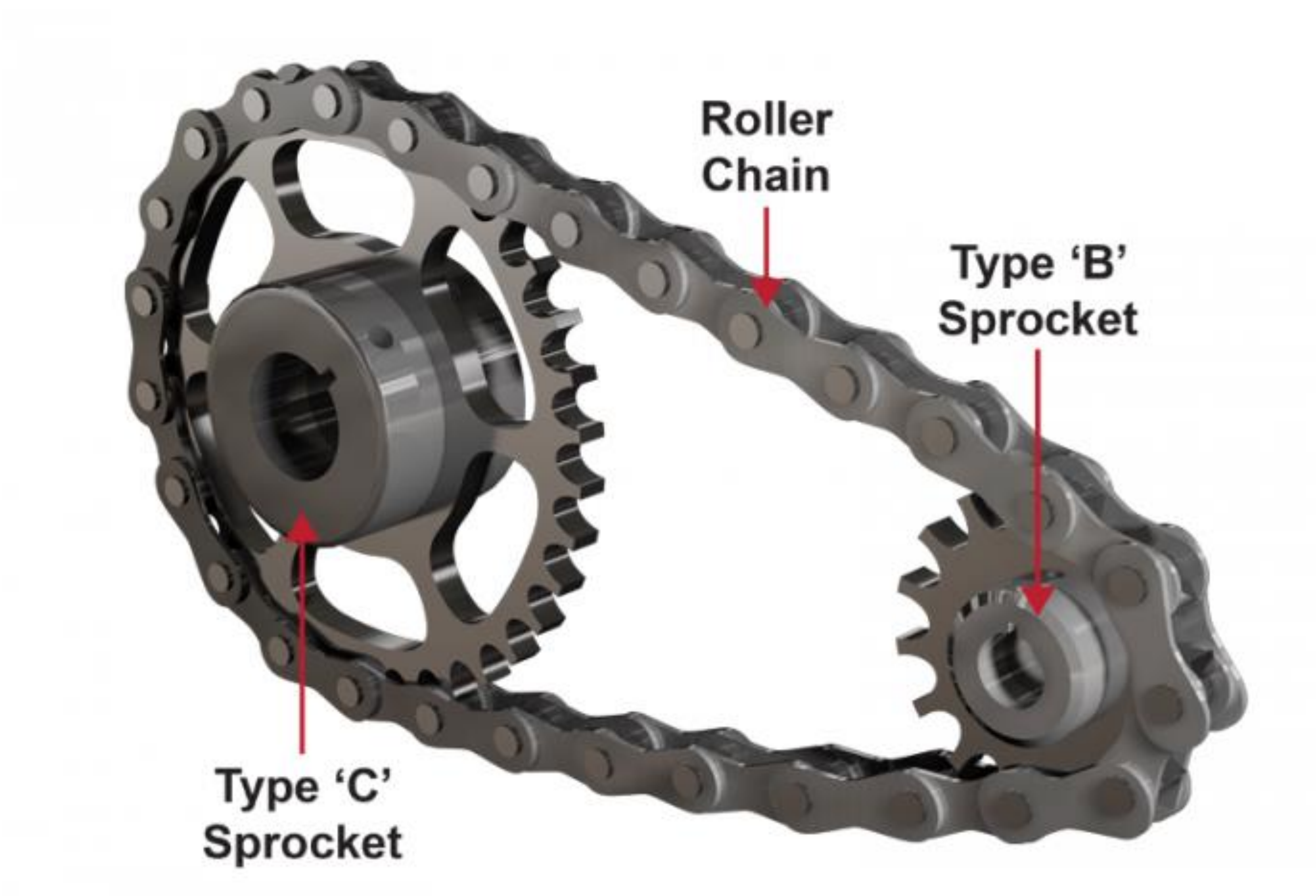
Belting Selection

Use Belting Manufacture Specific Data
For examples Gates Belts (know mostly for V-belts)
https://www.gates.com/us/en.html

Available Resources from the Manufacturer

- Products and Applications (by Industry)
- Engineering Applications & Design Software
- Calculators
- Resources Library

Roller Chain



Roller Chain Characteristics

- Similar to Timing Belts, no slip
- Long life
- Fairly large power can be transmitted
- Costs less than gears
- Easy to adjust centers, tolerant of large variation
- Input and Output sprocket usually rotate in same direction
- Only need light to no chain preload tension
  when installing, but may require a tensioner/idler

- Usually requires lubrication
- Can use multiple strands for increased power
- Usually made of steel
- Stainless steel versions available for food industry

Roller Chain

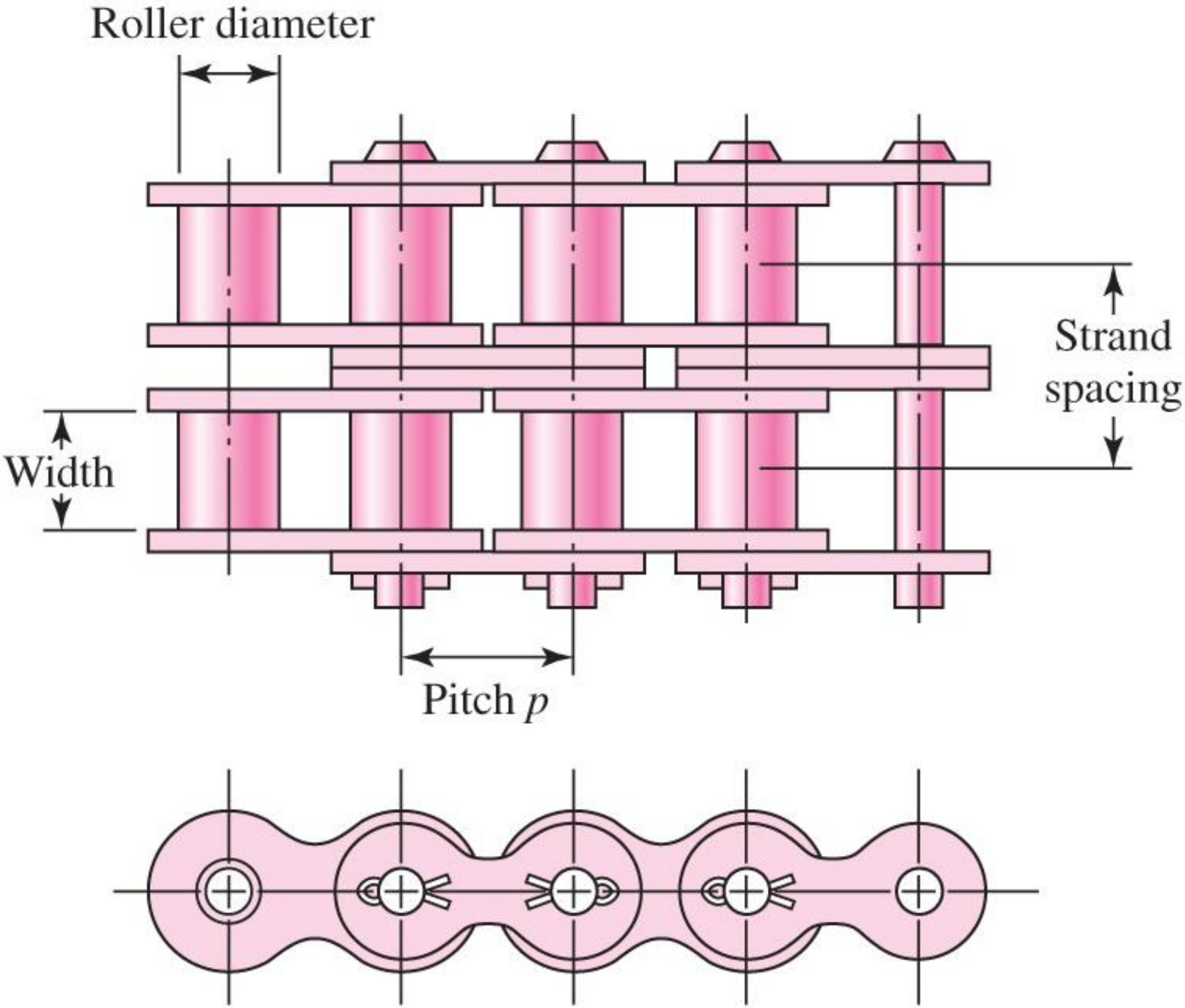


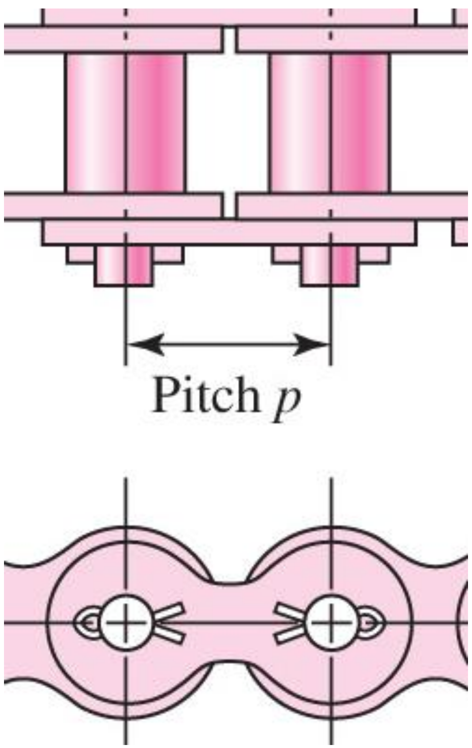
Fig.17–16

ANSI Chain Size Number

First number(s) pitch p in number of 1/8”
ANSI 40 chain is 4 x1/8” =0.50” pitch

XXX

0=Standard
1=Light Duty



ANSI 41 chain is 4 x1/8” =0.50” pitch
But Light Duty

Dimensions of American Standard Roller Chains

ANSI Chain Number	Pitch, in (mm)	Width, in (mm)	Minimum Tensile Strength, lbf (N)	Average Weight, lbf/ft (N/m)	Roller Diameter, in (mm)	Multiple-Strand Spacing, in (mm)
25	0.250 (6.35)	0.125 (3.18)	780 (3 470)	0.09 (1.31)	0.130 (3.30)	0.252 (6.40)
35	0.375 (9.52)	0.188 (4.76)	1 760 (7 830)	0.21 (3.06)	0.200 (5.08)	0.399 (10.13)
41	0.500 (12.70)	0.25 (6.35)	1 500 (6 670)	0.25 (3.65)	0.306 (7.77)	— —
40	0.500 (12.70)	0.312 (7.94)	3 130 (13 920)	0.42 (6.13)	0.312 (7.92)	0.566 (14.38)
50	0.625 (15.88)	0.375 (9.52)	4 880 (21 700)	0.69 (10.1)	0.400 (10.16)	0.713 (18.11)
60	0.750 (19.05)	0.500 (12.7)	7 030 (31 300)	1.00 (14.6)	0.469 (11.91)	0.897 (22.78)
80	1.000 (25.40)	0.625 (15.88)	12 500 (55 600)	1.71 (25.0)	0.625 (15.87)	1.153 (29.29)
100	1.250 (31.75)	0.750 (19.05)	19 500 (86 700)	2.58 (37.7)	0.750 (19.05)	1.409 (35.76)
120	1.500 (38.10)	1.000 (25.40)	28 000 (124 500)	3.87 (56.5)	0.875 (22.22)	1.789 (45.44)
140	1.750 (44.45)	1.000 (25.40)	38 000 (169 000)	4.95 (72.2)	1.000 (25.40)	1.924 (48.87)
160	2.000 (50.80)	1.250 (31.75)	50 000 (222 000)	6.61 (96.5)	1.125 (28.57)	2.305 (58.55)
180	2.250 (57.15)	1.406 (35.71)	63 000 (280 000)	9.06 (132.2)	1.406 (35.71)	2.592 (65.84)
200	2.500 (63.50)	1.500 (38.10)	78 000 (347 000)	10.96 (159.9)	1.562 (39.67)	2.817 (71.55)
240	3.00 (76.70)	1.875 (47.63)	112 000 (498 000)	16.4 (239)	1.875 (47.62)	3.458 (87.83)

Table 17–19

Engagement of Chain and Sprocket

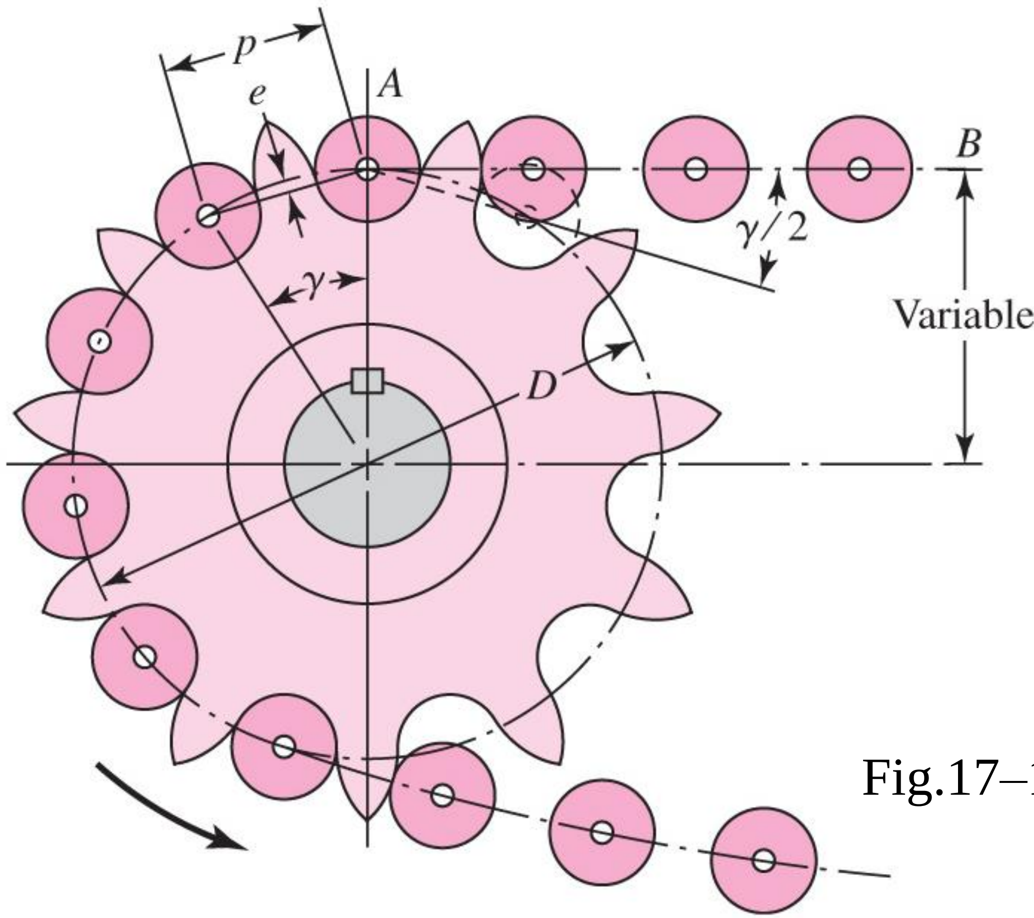
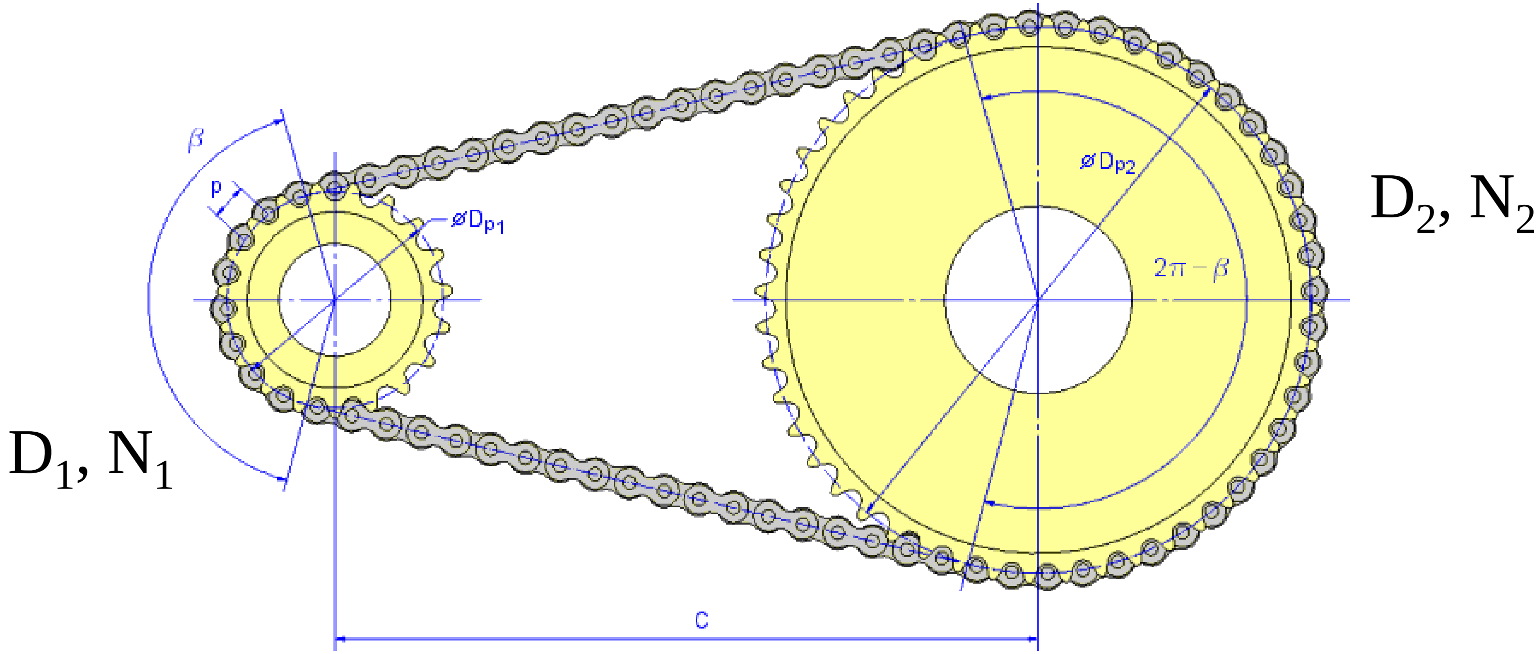


Fig.17–17

$$\sin\frac{\gamma}{2} = \frac{p/2}{D/2} \qquad \text{or} \qquad D = \frac{p}{\sin(\gamma/2)} \qquad (a)$$

$$D = \frac{p}{\sin(180°/N)} \qquad (17\text{–}29)$$

Roller Chain Drive



length of the chain L in pitches is

$$\boxed{\frac{L}{p} \approx \frac{2C}{p} + \frac{N_1 + N_2}{2} + \frac{(N_2 - N_1)^2}{4\pi^2 C/p}} \qquad (17\text{–}34)$$

The center-to-center distance C is given by

$$C = \frac{p}{4}\left[-A + \sqrt{A^2 - 8\left(\frac{N_2 - N_1}{2\pi}\right)^2}\,\right] \qquad (17\text{–}35)$$

Roller Chain Design Guide Rules of Thumb

- Min Teeth per Sprocket ~ 12 to 25
- Max Teeth per Sprocket ~120
- Max Speed Reduction ~7:1
- Min Wrap Angle ~ 120˚

Chain Velocity

$$\boxed{V = \frac{Npn}{12} \quad \text{feet per minute}} \tag{17–30}$$

where N = number of sprocket teeth

p = chain pitch, in

n = sprocket speed, rev/min

$$v_{\max} = \frac{\pi D n}{12} = \frac{\pi n p}{12 \sin(\gamma/2)} \tag{b}$$

$$d = D \cos \frac{\gamma}{2} \tag{c}$$

$$v_{\min} = \frac{\pi d n}{12} = \frac{\pi n p}{12} \frac{\cos(\gamma/2)}{\sin(\gamma/2)} \tag{d}$$

Chordal Speed Variation (as chain not perfectly round)

$$\frac{\Delta V}{V} = \frac{v_{\max} - v_{\min}}{V} = \frac{\pi}{N}\left[\frac{1}{\sin(180^\circ/N)} - \frac{1}{\tan(180^\circ/N)}\right] \qquad (17\text{–}31)$$

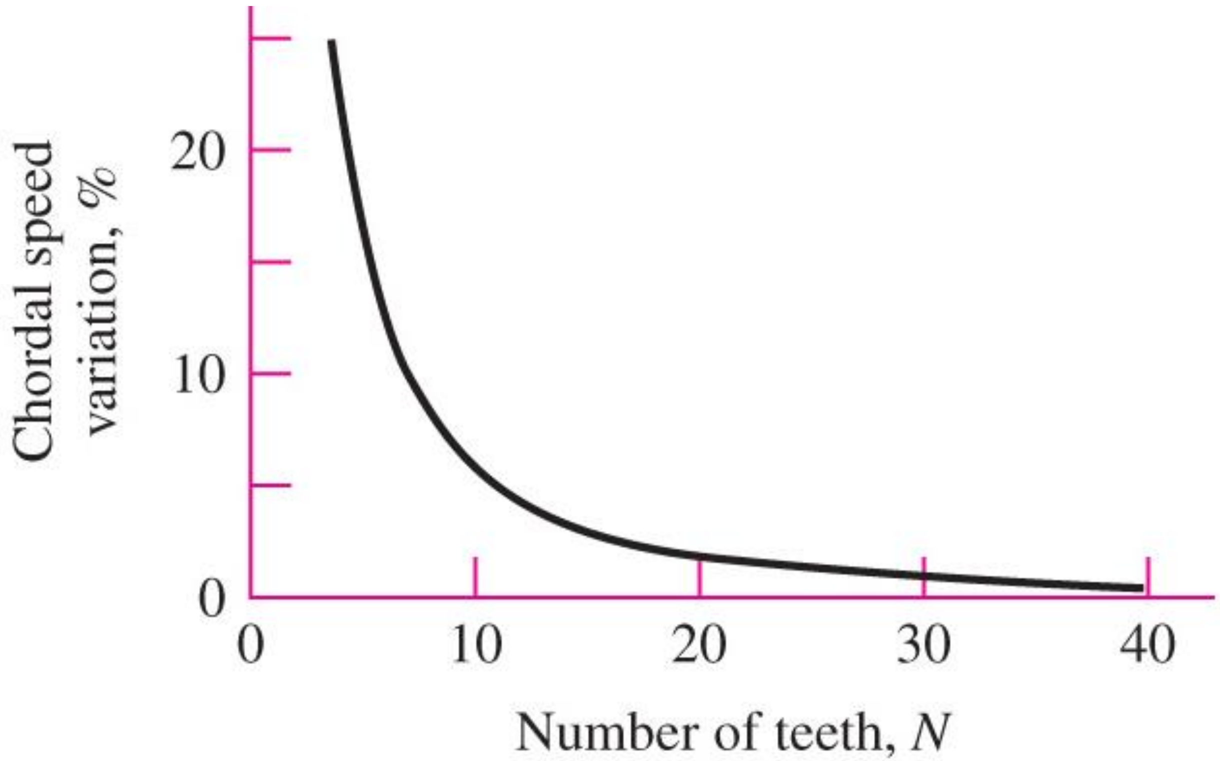


Fig.17–18

Roller Chain Rated Horsepower Capacity

Table 17–20

Rated Horsepower Capacity of Single-Strand Single-Pitch Roller Chain for a 17-Tooth Sprocket

Source: Compiled from ANSI B29.1-1975 information only section, and from B29.9-1958.

Sprocket Speed, rev/min	ANSI Chain Number 25	35	40	41	50	60
50	0.05	0.16	0.37	0.20	0.72	1.24
100	0.09	0.29	0.69	0.38	1.34	2.31
150	0.13*	0.41*	0.99*	0.55*	1.92*	3.32
200	0.16*	0.54*	1.29	0.71	2.50	4.30
300	0.23	0.78	1.85	1.02	3.61	6.20
400	0.30*	1.01*	2.40	1.32	4.67	8.03
500	0.37	1.24	2.93	1.61	5.71	9.81
600	0.44*	1.46*	3.45*	1.90*	6.72*	11.6
700	0.50	1.68	3.97	2.18	7.73	13.3
800	0.56*	1.89*	4.48*	2.46*	8.71*	15.0
900	0.62	2.10	4.98	2.74	9.69	16.7
1000	0.68*	2.31*	5.48	3.01	10.7	18.3
1200	0.81	2.73	6.45	3.29	12.6	21.6
1400	0.93*	3.13*	7.41	2.61	14.4	18.1
1600	1.05*	3.53*	8.36	2.14	12.8	14.8
1800	1.16	3.93	8.96	1.79	10.7	12.4
2000	1.27*	4.32*	7.72*	1.52*	9.23*	10.6
2500	1.56	5.28	5.51*	1.10*	6.58*	7.57
3000	1.84	5.64	4.17	0.83	4.98	5.76

Type A | Type B | Type C

Roller Chain Rated Horsepower Capacity

Table 17–20

Rated Horsepower Capacity of Single-Strand Single-Pitch Roller Chain for a 17-Tooth Sprocket (*Continued*)

Sprocket Speed, rev/min		ANSI Chain Number 80	100	120	140	160	180	200	240
50	Type A	2.88	5.52	9.33	14.4	20.9	28.9	38.4	61.8
100		5.38	10.3	17.4	26.9	39.1	54.0	71.6	115
150		7.75	14.8	25.1	38.8	56.3	77.7	103	166
200		10.0	19.2	32.5	50.3	72.9	101	134	215
300		14.5	27.7	46.8	72.4	105	145	193	310
400		18.7	35.9	60.6	93.8	136	188	249	359
500	Type B	22.9	43.9	74.1	115	166	204	222	0
600		27.0	51.7	87.3	127	141	155	169	
700		31.0	59.4	89.0	101	112	123	0	
800		35.0	63.0	72.8	82.4	91.7	101		
900		39.9	52.8	61.0	69.1	76.8	84.4		
1000		37.7	45.0	52.1	59.0	65.6	72.1		
1200		28.7	34.3	39.6	44.9	49.9	0		
1400		22.7	27.2	31.5	35.6	0			
1600		18.6	22.3	25.8	0				
1800		15.6	18.7	21.6					
2000		13.3	15.9	0					
2500		9.56	0.40						
3000		7.25	0						
Type C					Type C′				

Available Sprocket Tooth Counts

Table 17–21

Single-Strand Sprocket Tooth Counts Available from One Supplier*

No.	Available Sprocket Tooth Counts
25	8-30, 32, 34, 35, 36, 40, 42, 45, 48, 54, 60, 64, 65, 70, 72, 76, 80, 84, 90, 95, 96, 102, 112, 120
35	4-45, 48, 52, 54, 60, 64, 65, 68, 70, 72, 76, 80, 84, 90, 95, 96, 102, 112, 120
41	6-60, 64, 65, 68, 70, 72, 76, 80, 84, 90, 95, 96, 102, 112, 120
40	8-60, 64, 65, 68, 70, 72, 76, 80, 84, 90, 95, 96, 102, 112, 120
50	8-60, 64, 65, 68, 70, 72, 76, 80, 84, 90, 95, 96, 102, 112, 120
60	8-60, 62, 63, 64, 65, 66, 67, 68, 70, 72, 76, 80, 84, 90, 95, 96, 102, 112, 120
80	8-60, 64, 65, 68, 70, 72, 76, 78, 80, 84, 90, 95, 96, 102, 112, 120
100	8-60, 64, 65, 67, 68, 70, 72, 74, 76, 80, 84, 90, 95, 96, 102, 112, 120
120	9-45, 46, 48, 50, 52, 54, 55, 57, 60, 64, 65, 67, 68, 70, 72, 76, 80, 84, 90, 96, 102, 112, 120
140	9-28, 30, 31, 32, 33, 34, 35, 36, 37, 39, 40, 42, 43, 45, 48, 54, 60, 64, 65, 68, 70, 72, 76, 80, 84, 96
160	8-30, 32–36, 38, 40, 45, 46, 50, 52, 53, 54, 56, 57, 60, 62, 63, 64, 65, 66, 68, 70, 72, 73, 80, 84, 96
180	13-25, 28, 35, 39, 40, 45, 54, 60
200	9-30, 32, 33, 35, 36, 39, 40, 42, 44, 45, 48, 50, 51, 54, 56, 58, 59, 60, 63, 64, 65, 68, 70, 72
240	9-30, 32, 35, 36, 40, 44, 45, 48, 52, 54, 60

Example 17–5

Select drive components for a 2:1 reduction, 90-hp input at 300 rev/min, moderate shock, an abnormally long 18-hour day, poor lubrication, cold temperatures, dirty surroundings, short drive $C/p = 25$.

Solution

Function: $H_{\text{nom}} = 90$ hp, $n_1 = 300$ rev/min, $C/p = 25$,
Design factor: Choose $n_d = 1.5$
Service factor: Choose $K_s = 1.3$ for moderate shock
Sprocket teeth: $N_1 = 17$ teeth, $N_2 = 34$ teeth,

Example 17–5

Form a table:

Number of Strands	176/K2 (Table 17–23)	Chain Number (Table 17–20)	Lubrication Type
1	176/1 = 176	200	C′
2	176/1.7 = 104	160	C
3	176/2.5 = 70.4	140	B
4	176/3.3 = 53.3	140	B

3 strands of number 140 chain (H_{tab} is 72.4 hp).

Number of pitches in the chain:

$$\frac{L}{p} = \frac{2C}{p} + \frac{N_1 + N_2}{2} + \frac{(N_2 - N_1)^2}{4\pi^2 C/p}$$

$$= 2(25) + \frac{17 + 34}{2} + \frac{(34 - 17)^2}{4\pi^2(25)} = 75.79 \text{ pitches}$$

Use 76 pitches. Then $L/p = 76$.

Example 17–5

Identify the center-to-center distance: From Eqs. (17–35) and (17–36),

$$A = \frac{N_1 + N_2}{2} - \frac{L}{p} = \frac{17 + 34}{2} - 76 = -50.5$$

$$C = \frac{p}{4}\left[-A + \sqrt{A^2 - 8\left(\frac{N_2 - N_1}{2\pi}\right)^2}\right]$$

$$= \frac{p}{4}\left[50.5 + \sqrt{50.5^2 - 8\left(\frac{34 - 17}{2\pi}\right)^2}\right] = 25.104p$$

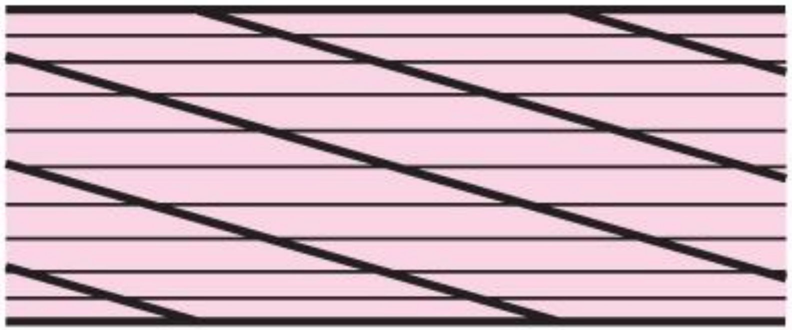
For a 140 chain, $p = 1.75$ in. Thus,

$$C = 25.104p = 25.104(1.75) = 43.93 \text{ in}$$

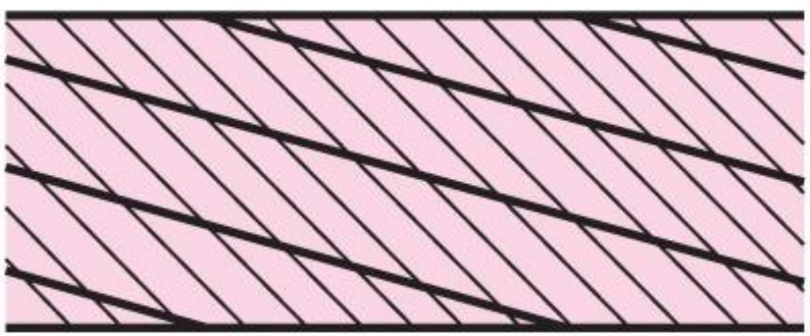
Lubrication: Type B

Comment: This is operating on the pre-extreme portion of the power, so durability estimates other than 15 000 h are not available. Given the poor operating conditions, life will be much shorter.

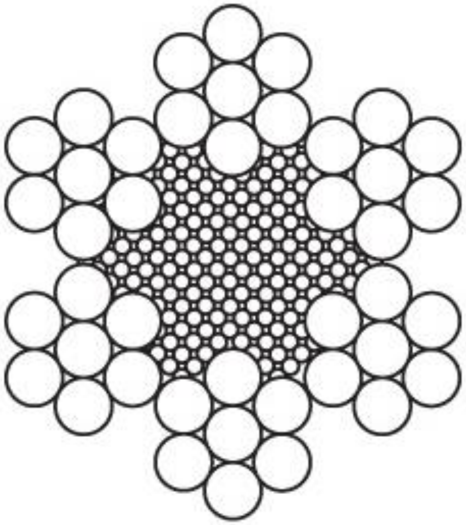
Wire Rope



(*a*) Regular lay



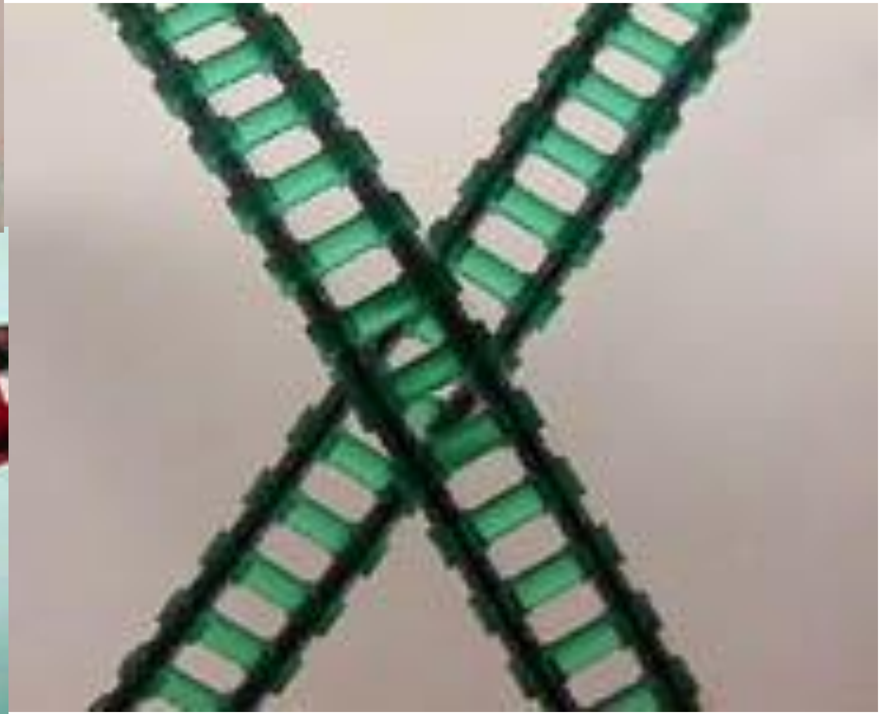
(*b*) Lang lay



(*c*) Section of 6×7 rope

Fig.17–19

Berg cable timing belts



Stress in Wire Rope

$$M = \frac{EI}{\rho} \qquad \text{and} \qquad M = \frac{\sigma I}{c} \qquad (a)$$

$$\sigma = \frac{Ec}{\rho} \qquad (b)$$

$$c = d_w/2$$

where d_w is the wire diameter

$$\sigma = E_r \frac{d_w}{D} \qquad (c)$$

where E_r is the *modulus of elasticity of the rope*, not the wire

Wire-Rope Data

Rope	Weight per Foot, lbf	Minimum Sheave Diameter, in	Standard Sizes d, in	Material	Size of Outer Wires	Modulus of Elasticity,* Mpsi	Strength,† kpsi
6 × 7 haulage	$1.50d^2$	$42d$	$\frac{1}{4}$–$1\frac{1}{2}$	Monitor steel	$d/9$	14	100
				Plow steel	$d/9$	14	88
				Mild plow steel	$d/9$	14	76
6 × 19 standard hoisting	$1.60d^2$	$26d$–$34d$	$\frac{1}{4}$–$2\frac{3}{4}$	Monitor steel	$d/13$–$d/16$	12	106
				Plow steel	$d/13$–$d/16$	12	93
				Mild plow steel	$d/13$–$d/16$	12	80
6 × 37 special flexible	$1.55d^2$	$18d$	$\frac{1}{4}$–$3\frac{1}{2}$	Monitor steel	$d/22$	11	100
				Plow steel	$d/22$	11	88
8 × 19 extra flexible	$1.45d^2$	$21d$–$26d$	$\frac{1}{4}$–$1\frac{1}{2}$	Monitor steel	$d/15$–$d/19$	10	92
				Plow steel	$d/15$–$d/19$	10	80
7 × 7 aircraft	$1.70d^2$	—	$\frac{1}{16}$–$\frac{3}{8}$	Corrosion-resistant steel	—	—	124
				Carbon steel	—	—	124
7 × 9 aircraft	$1.75d^2$	—	$\frac{1}{8}$–$1\frac{3}{8}$	Corrosion-resistant steel	—	—	135
				Carbon steel	—	—	143
19-wire aircraft	$2.15d^2$	—	$\frac{1}{32}$–$\frac{5}{16}$	Corrosion-resistant steel	—	—	165
				Carbon steel	—	—	165

Table 17–24

Equivalent Bending Load in a Pulley, diameter D

- Wire rope tension giving same tensile stress as sheave bending is called *equivalent bending load* F_b

$$F_b = \sigma A_m = \frac{E_r d_w A_m}{D} \tag{17-41}$$

- Clearly the smaller the sheave diameter the higher the added bending tension of the sheave on the cable

Percent Strength Loss for D/d

D is sheave (pulley) diameter
d is the cable diameter

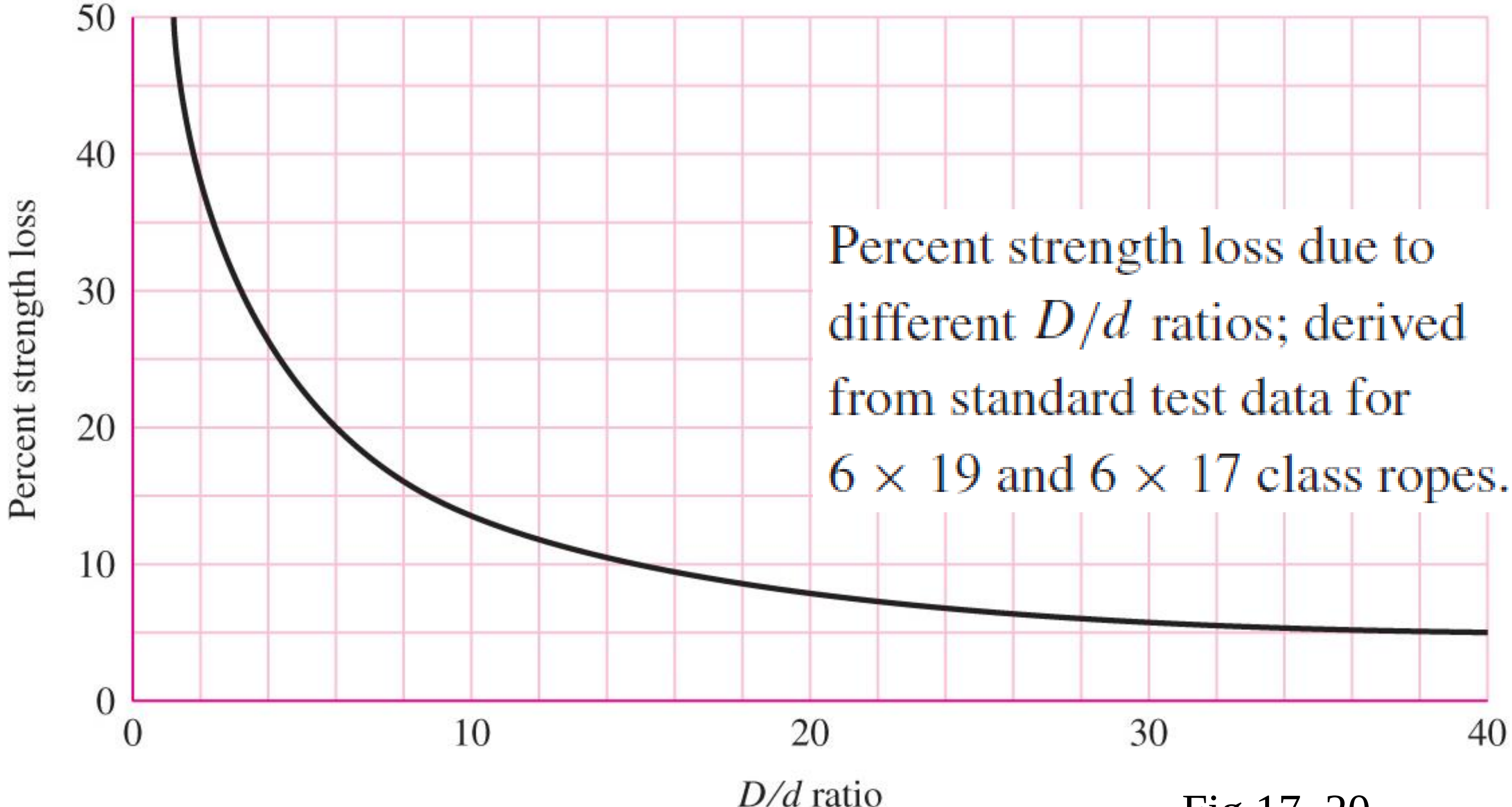


Fig.17–20

Minimum Factors of Safety for Wire Rope

Track cables	3.2	Passenger elevators, ft/min:	
Guys	3.5	50	7.60
Mine shafts, ft:		300	9.20
Up to 500	8.0	800	11.25
1000–2000	7.0	1200	11.80
2000–3000	6.0	1500	11.90
Over 3000	5.0	Freight elevators, ft/min:	
Hoisting	5.0	50	6.65
Haulage	6.0	300	8.20
Cranes and derricks	6.0	800	10.00
Electric hoists	7.0	1200	10.50
Hand elevators	5.0	1500	10.55
Private elevators	7.5	Powered dumbwaiters, ft/min:	
Hand dumbwaiter	4.5	50	4.8
Grain elevators	7.5	300	6.6
		500	8.0

Table 17–25

Bearing Pressure of Wire Rope in Sheave Groove

$$p = \frac{2F}{dD} \tag{17-42}$$

where F = tensile force on rope

d = rope diameter

D = sheave diameter

Maximum Allowable Bearing Pressures (in psi)

Rope	Wood[a]	Cast Iron[b]	Cast Steel[c]	Chilled Cast Irons[d]	Manganese Steel[e]
			Sheave Material		
Regular lay:					
6 × 7	150	300	550	650	1470
6 × 19	250	480	900	1100	2400
6 × 37	300	585	1075	1325	3000
8 × 19	350	680	1260	1550	3500
Lang lay:					
6 × 7	165	350	600	715	1650
6 × 19	275	550	1000	1210	2750
6 × 37	330	660	1180	1450	3300

[a]On end grain of beech, hickory, or gum.

[b]For H_B (min.) = 125.

[c]30–40 carbon; H_B (min.) = 160.

[d]Use only with uniform surface hardness.

[e]For high speeds with balanced sheaves having ground surfaces.

Table 17–26

Relation Between Fatigue Life of Wire Rope and Sheave Pressure

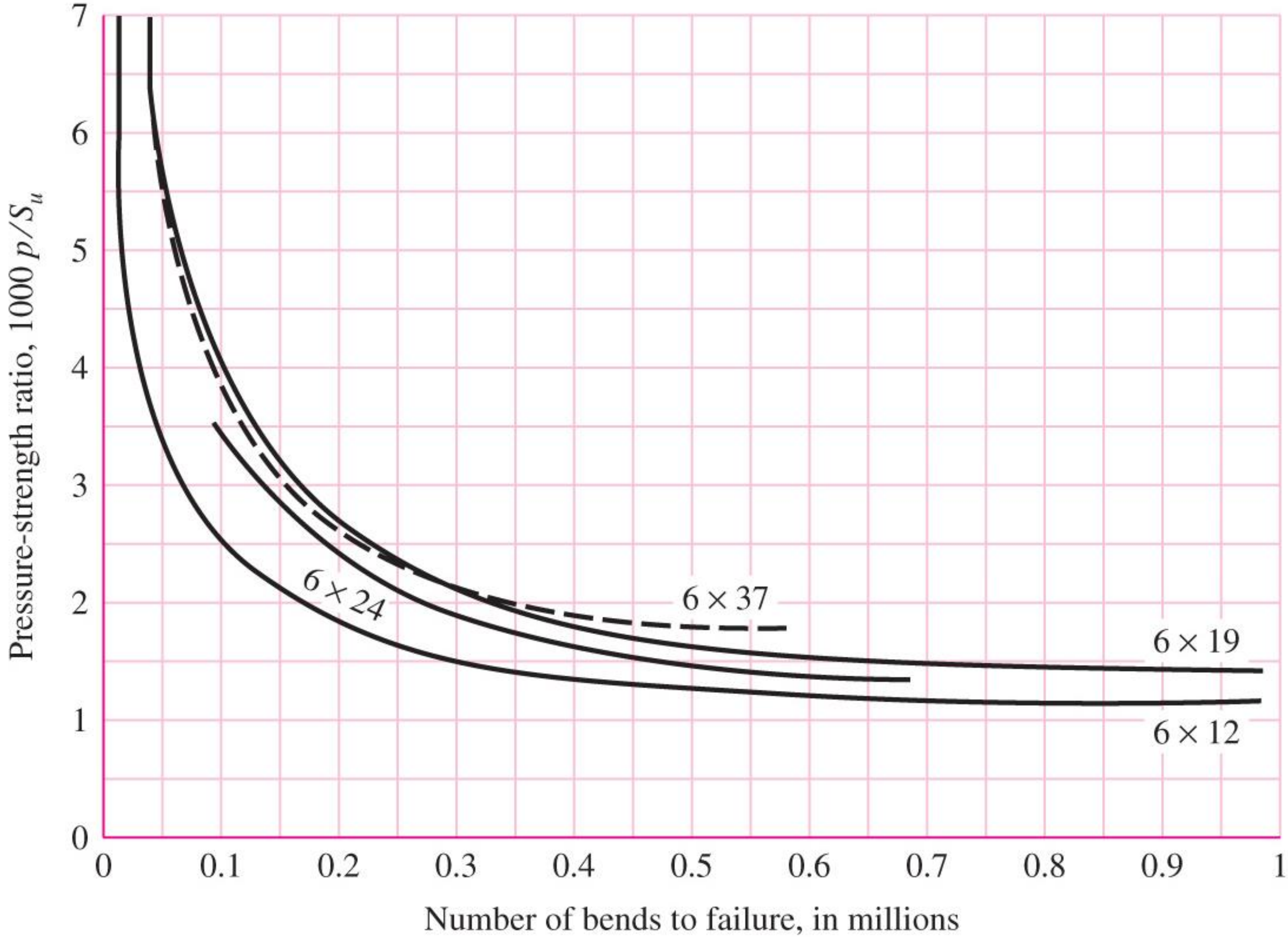


Fig.17–21

Fatigue of Wire Rope

- Fig. 17–21does not preclude failure by fatigue or wear
- It does show long life if p/S_u is less than 0.001.
- Substituting this ratio in Eq. (17–42),

$$S_u = \frac{2000F}{dD} \tag{17–43}$$

- Dividing both sides of Eq. (17–42) by S_u and solving for F, gives allowable fatigue tension,

$$F_f = \frac{(p/S_u)S_u dD}{2} \tag{17–44}$$

- Factor of safety for fatigue is

$$n_f = \frac{F_f - F_b}{F_t} \tag{17–45}$$

Factor of Safety for Static Loading

- The factor of safety for static loading is

$$n_s = \frac{F_u - F_b}{F_t} \qquad (17\text{–}46)$$

Typical Strength of Individual Wires

Improved plow steel (monitor)	$240 < S_u < 280$ kpsi
Plow steel	$210 < S_u < 240$ kpsi
Mild plow steel	$180 < S_u < 210$ kpsi

Service-Life Curve Based on Bending and Tensile Stresses

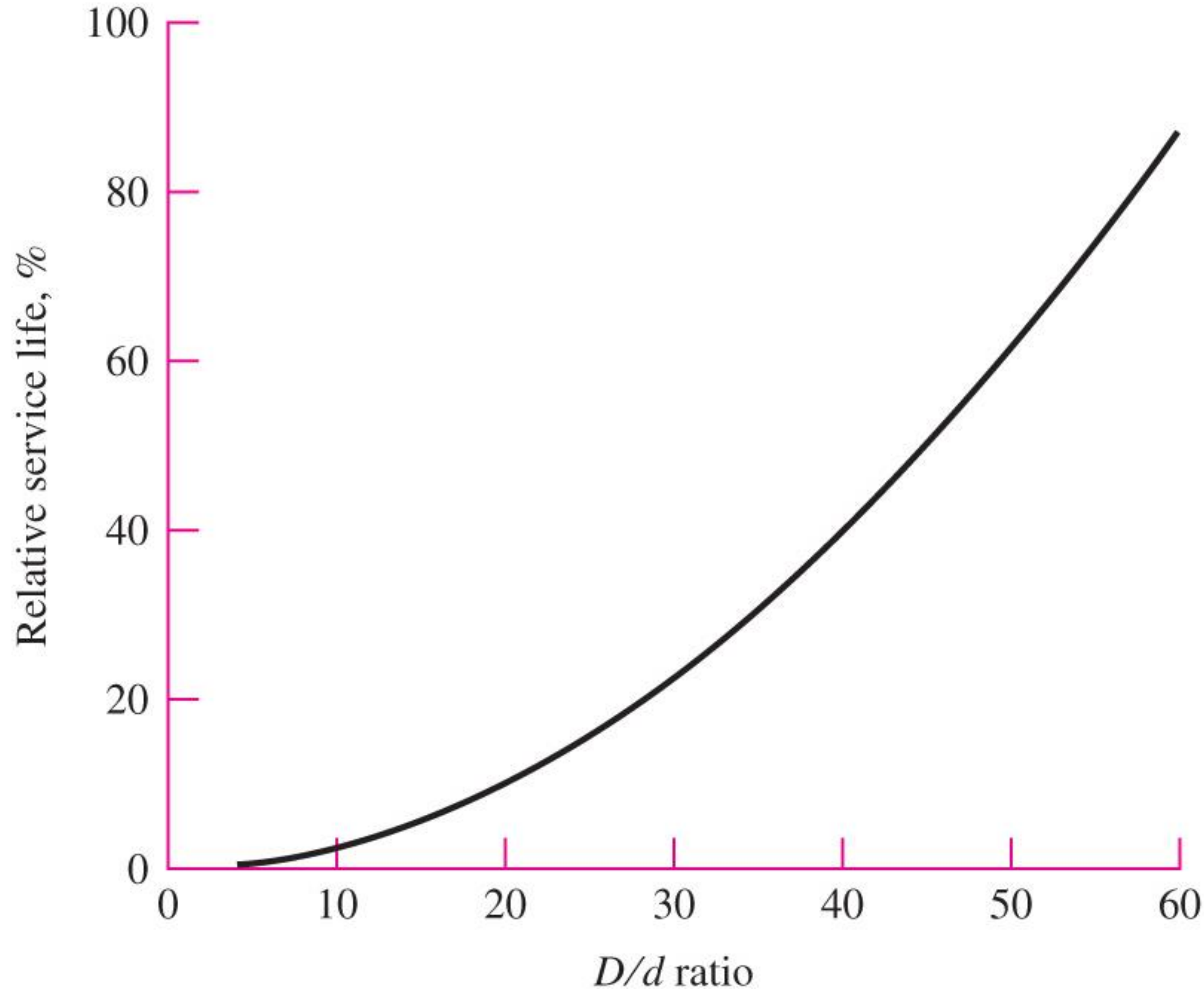


Fig.17–22

Some Wire-Rope Properties

Table 17–27

Some Useful Properties of 6 × 7, 6 × 19, and 6 × 37 Wire Ropes

Wire Rope	Weight per Foot w, lbf/ft	Weight per Foot Including Core w, lbf/ft	Minimum Sheave Diameter D, in	Better Sheave Diameter D, in	Diameter of Wires d_w, in	Area of Metal A_m, in^2	Rope Young's Modulus E_r, psi
6 × 7	$1.50d^2$		$42d$	$72d$	$0.111d$	$0.38d^2$	13×10^6
6 × 19	$1.60d^2$	$1.76d^2$	$30d$	$45d$	$0.067d$	$0.40d^2$	12×10^6
6 × 37	$1.55d^2$	$1.71d^2$	$18d$	$27d$	$0.048d$	$0.40d^2$	12×10^6

Working Equations for Mine-Hoist Problem

$$F_t = \left(\frac{W}{m} + wl\right)\left(1 + \frac{a}{g}\right) \qquad (17\text{–}47)$$

where W = weight at the end of the rope (cage and load), lbf

m = number of wire ropes supporting the load

w = weight/foot of the wire rope, lbf/ft

l = maximum suspended length of rope, ft

a = maximum acceleration/deceleration experienced, ft/s^2

g = acceleration of gravity, ft/s^2

Working Equations for Mine-Hoist Problem

$$F_f = \frac{(p/S_u)S_u D d}{2} \qquad (17\text{–}44)$$

where (p/S_u) = specified life, from Fig. 17–21

S_u = ultimate tensile strength of the wires, psi

D = sheave or winch drum diameter, in

d = nominal wire rope size, in

Working Equations for Mine-Hoist Problem

$$F_b = \frac{E_r d_w A_m}{D} \qquad (17\text{–}41)$$

where E_r = Young's modulus for the wire rope, Table 17–24 or 17–27, psi

d_w = diameter of the wires, in

A_m = metal cross-sectional area, Table 17–27, in^2

D = sheave or winch drum diameter, in

$$n_s = \frac{F_u - F_b}{F_t} \qquad (17\text{–}46)$$

$$n_f = \frac{F_f - F_b}{F_t} \qquad (17\text{–}45)$$

Example 17–6

Given a 6 × 19 monitor steel ($S_u = 240$ kpsi) wire rope.

(*a*) Develop the expressions for rope tension F_t, fatigue tension F_f, equivalent bending tensions F_b, and fatigue factor of safety n_f for a 531.5-ft, 1-ton cage-and-load mine hoist with a starting acceleration of 2 ft/s^2 as depicted in Fig. 17–23. The sheave diameter is 72 in.

(*b*) Using the expressions developed in part (*a*), examine the variation in factor of safety n_f for various wire rope diameters d and number of supporting ropes m.

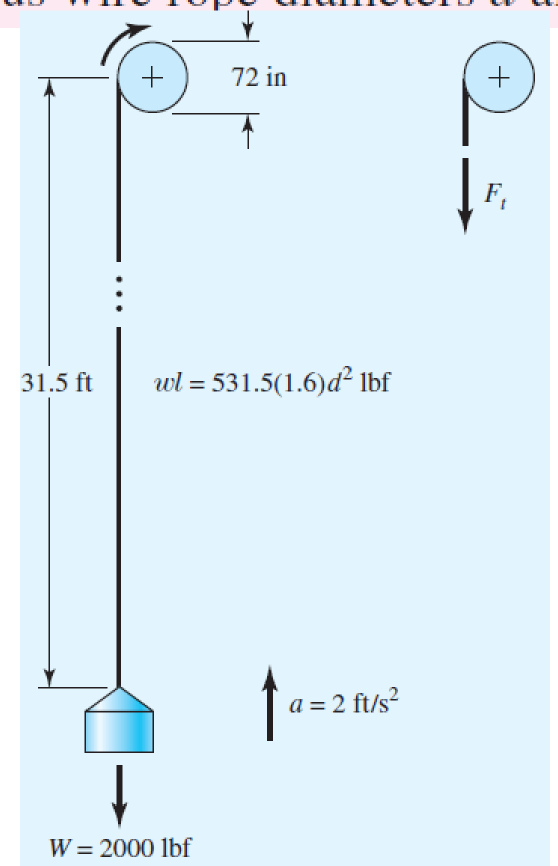


Fig.17–23

Example 17–6

(*a*) Rope tension F_t from Eq. (17–47), using Table 17–24 for w, is given by

$$F_t = \left(\frac{W}{m} + wl\right)\left(1 + \frac{a}{g}\right) = \left[\frac{2000}{m} + 1.60d^2(531.5)\right]\left(1 + \frac{2}{32.2}\right)$$

$$= \frac{2124}{m} + 903d^2 \qquad \text{Answer}$$

From Fig. 17–21, use $p/S_u = 0.0014$. Fatigue tension F_f from Eq. (17–44) is given by

$$F_f = \frac{(p/S_u)S_u Dd}{2} = \frac{0.0014(240\,000)72d}{2} = 12\,096d \qquad \text{Answer}$$

Equivalent bending tension F_b from Eq. (17–41) and Table 17–27 is given by

$$F_b = \frac{E_r d_w A_m}{D} = \frac{12(10^6)\,0.067d(0.40d^2)}{72} = 4467d^3 \qquad \text{Answer}$$

Example 17–6

Factor of safety n_f in fatigue from Eq. (17–45) is given by

$$n_f = \frac{F_f - F_b}{F_t} = \frac{12\,096d - 4467d^3}{2124/m + 903d^2}$$ Answer

(*b*) Using a spreadsheet program, form a table as follows:

	n_f			
d	$m = 1$	$m = 2$	$m = 3$	$m = 4$
0.25	1.355	2.641	3.865	5.029
0.375	1.910	3.617	5.150	6.536
0.500	2.336	4.263	5.879	7.254
0.625	2.612	4.573	6.099	7.331
0.750	2.731	4.578	5.911	6.918
0.875	2.696	4.330	5.425	6.210
1.000	2.520	3.882	4.736	5.320

Example 17–6

Wire rope sizes are discrete, as is the number of supporting ropes. Note that for each m the factor of safety exhibits a maximum. Predictably the largest factor of safety increases with m. If the required factor of safety were to be 6, only three or four ropes could meet the requirement. The sizes are different: $\frac{5}{8}$-in ropes with three ropes or $\frac{3}{8}$-in ropes with four ropes. The costs include not only the wires, but the grooved winch drums.

Flexible Shaft Configurations

Fig.17–24*b*

Flexible Shaft Construction Details

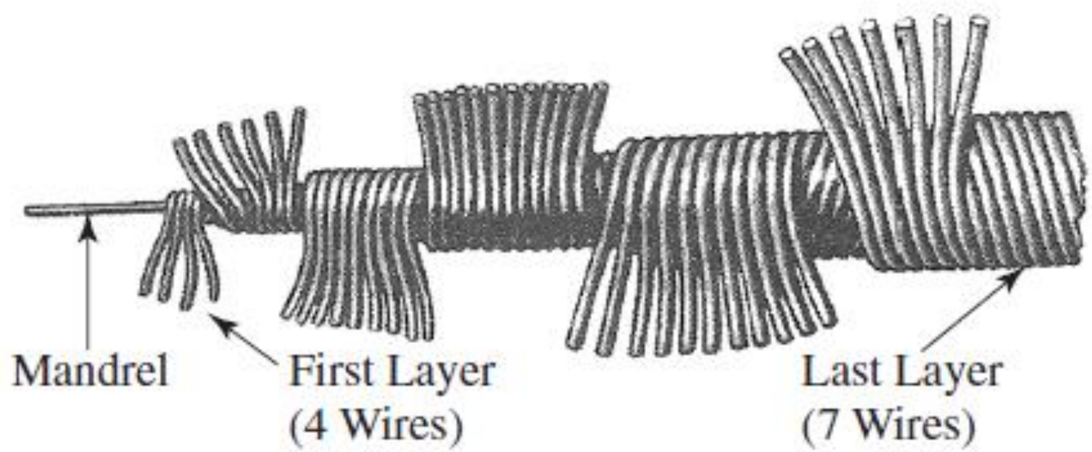


Fig.17–24*a*