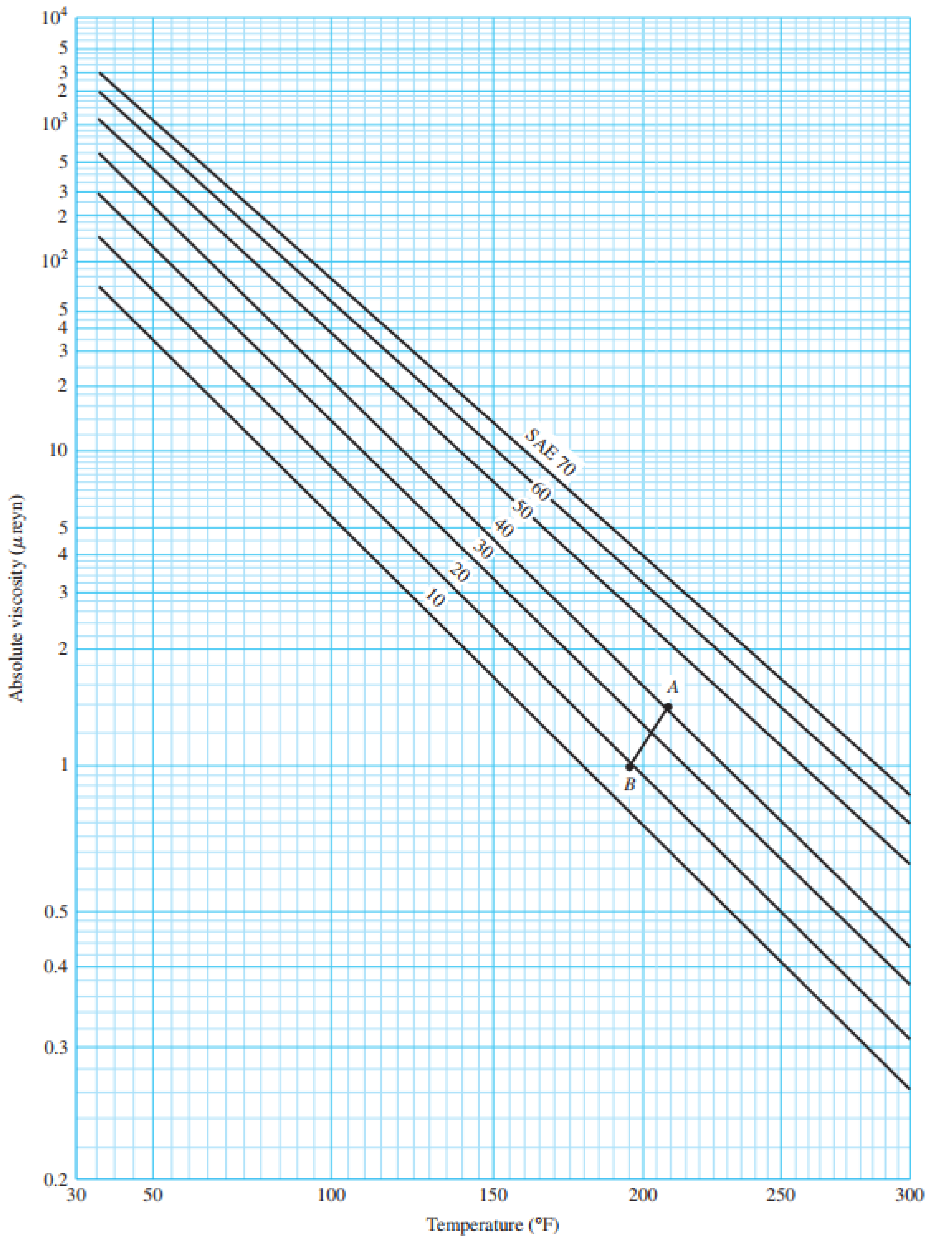
Chapter 12 Journal Bearings

To find Viscosity (μ) Using T_{av}

US units

Figure 12–12

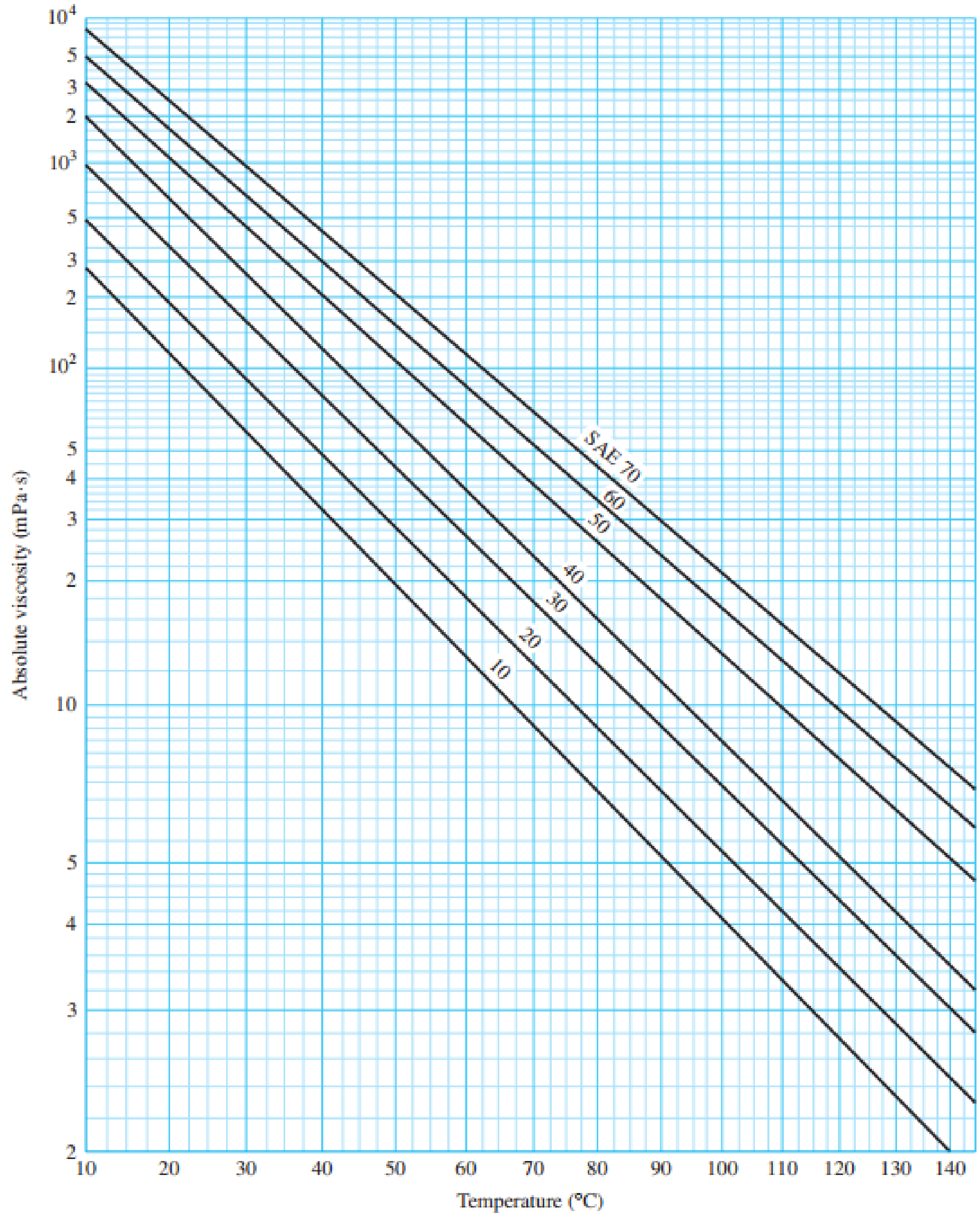
Viscosity–temperature chart in U.S. customary units. *(Raimondi and Boyd.)*



SI Units

Figure 12–13

Viscosity–temperature chart in SI units. *(Adapted from Fig. 12–12.)*



To find h_o

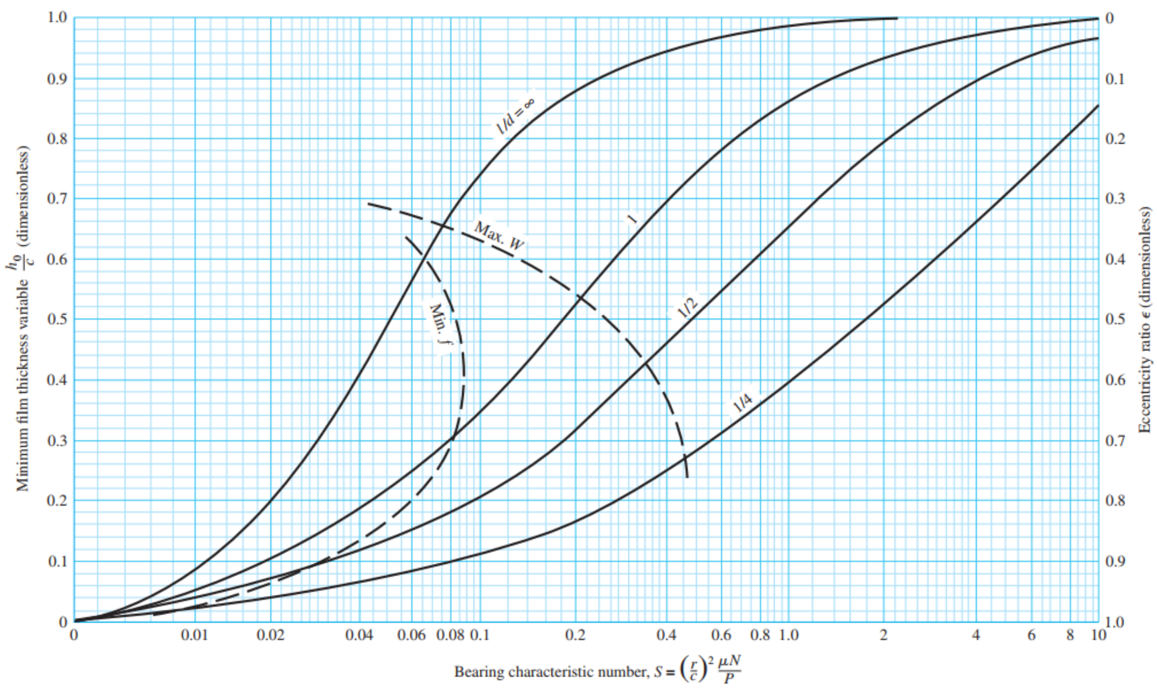


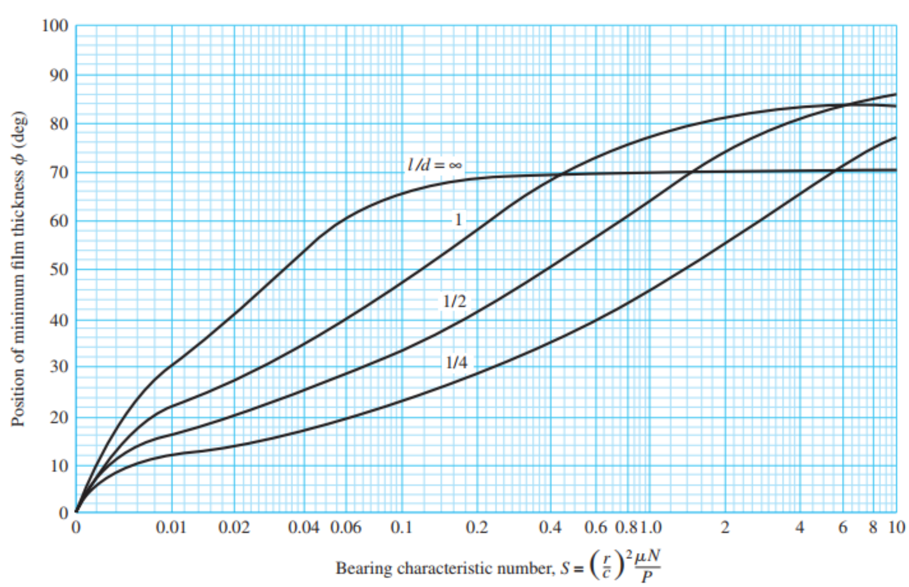
Figure 12–16

Chart for minimum film thickness variable and eccentricity ratio. The left boundary of the zone defines the optimal h_0 for minimum friction; the right boundary is optimum h_0 for load. *(Raimondi and Boyd.)*

To find Φ (Position of h_o)

Figure 12–17

Chart for determining the position of the minimum film thickness h_0. *(Raimondi and Boyd.)*



To find friction coefficient f

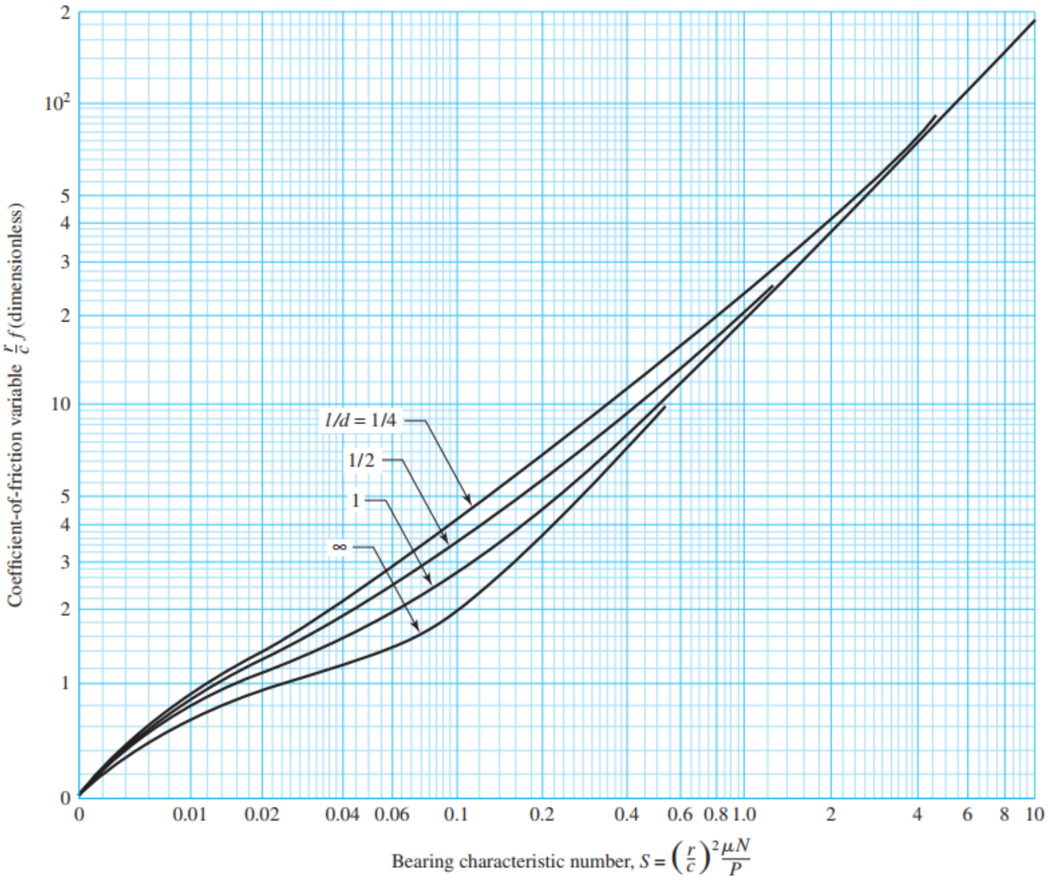


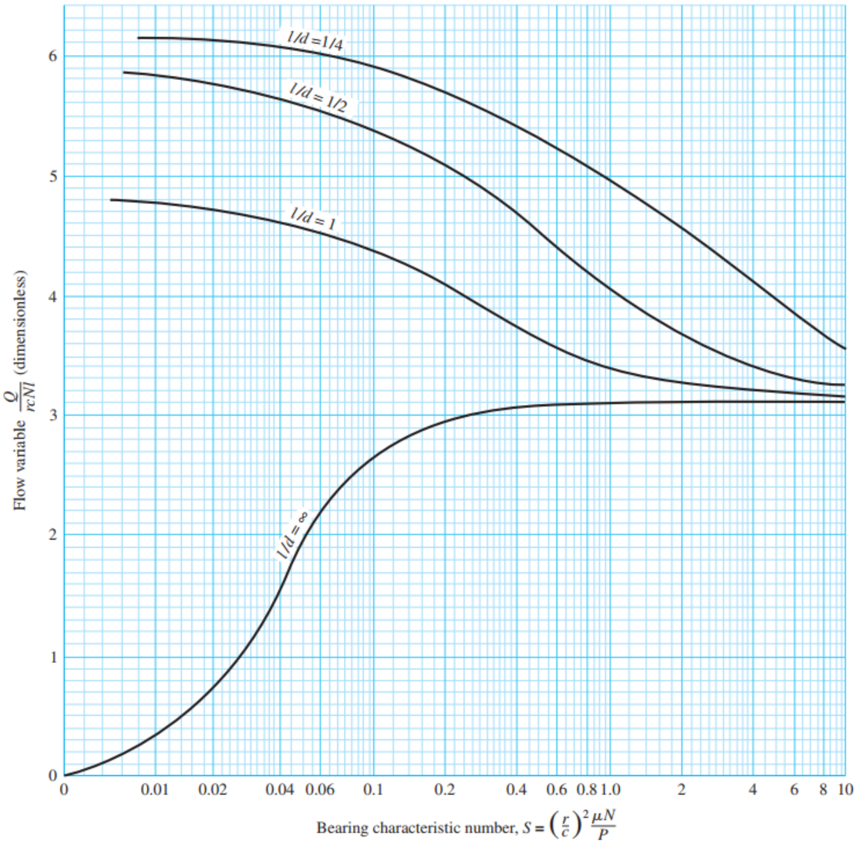
Figure 12–18

Chart for coefficient-of-friction variable; note that Petroff's equation is the asymptote. *(Raimondi and Boyd.)*

To find Q

Figure 12–19

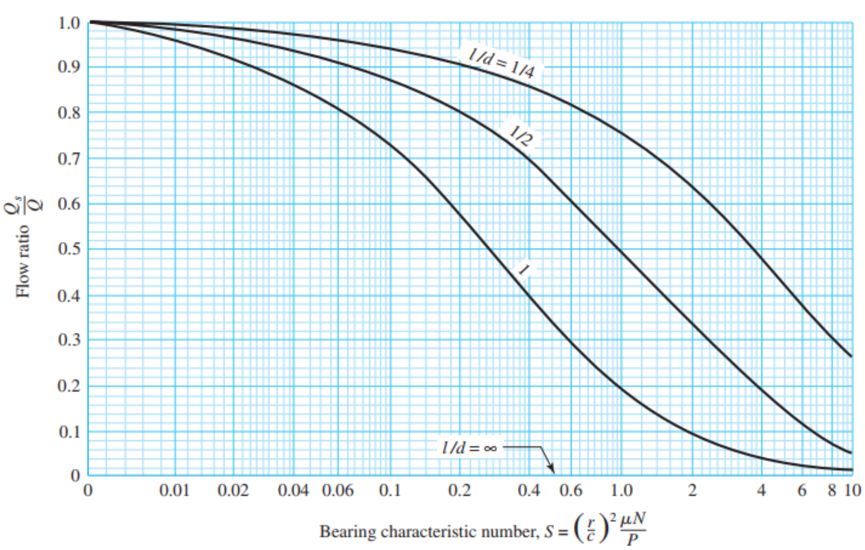
Chart for flow variable.
Note: Not for pressure-fed bearings. *(Raimondi and Boyd.)*



To find Q_s

Figure 12–20

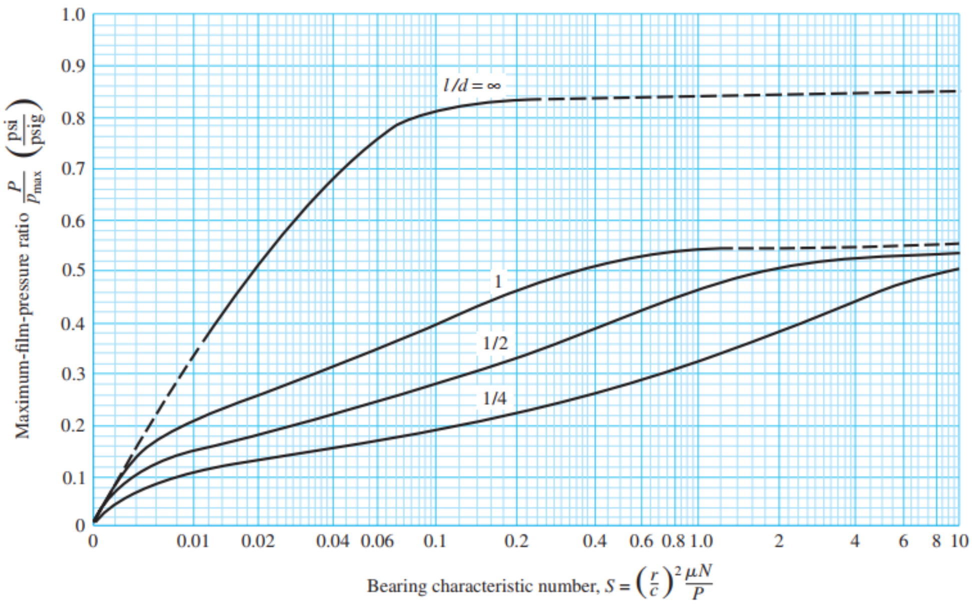
Chart for determining the ratio of side flow to total flow. *(Raimondi and Boyd.)*



To find P_{max}

Figure 12–21

Chart for determining the maximum film pressure. *Note:* Not for pressure-fed bearings. *(Raimondi and Boyd.)*



To find $\theta_{p\,max}$, $\theta_{p\,min}$

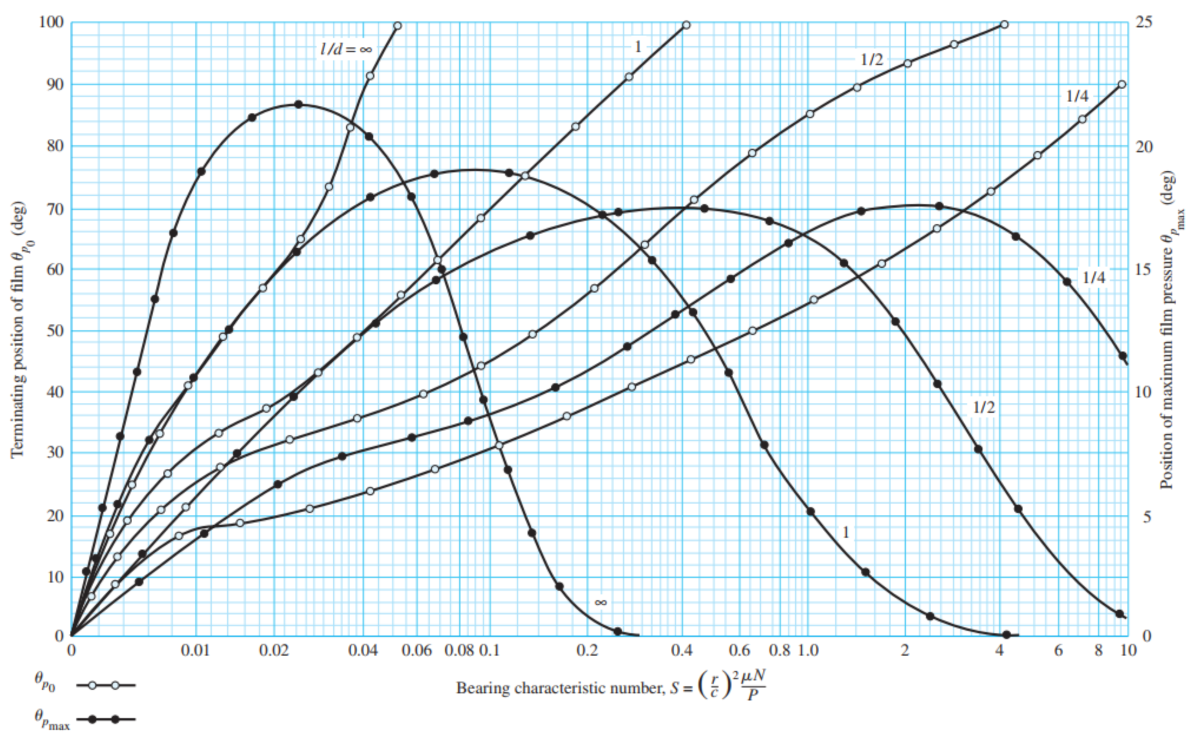


Figure 12–22

Chart for finding the terminating position of the lubricant film and the position of maximum film pressure. *(Raimondi and Boyd.)*

To Find ΔT

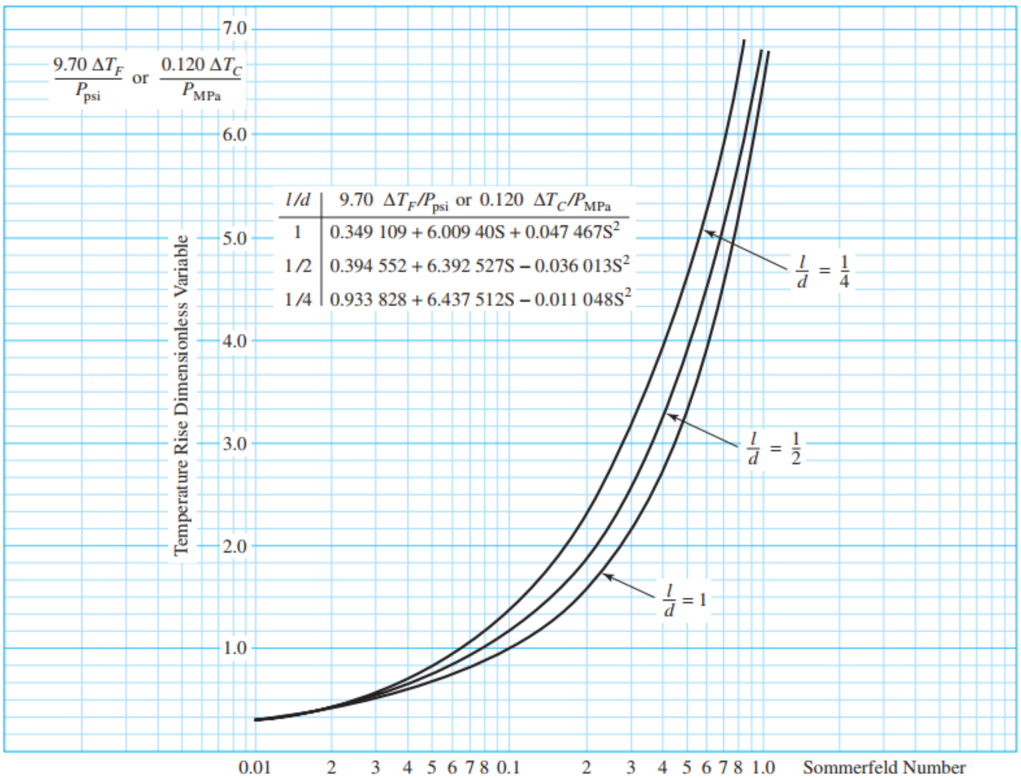


Figure 12–24

Figures 12–18, 12–19, and 12–20 combined to reduce iterative table look-up. *(Source: Chart based on work of Raimondi and Boyd boundary condition (2), i.e., no negative lubricant pressure developed. Chart is for full journal bearing using single lubricant pass, side flow emerges with temperature rise $\Delta T/2$, thru flow emerges with temperature rise ΔT, and entire flow is supplied at datum sump temperature.)*

To find h_{CR} or C_T

$$\hbar_{CR} = \begin{cases} 2\ \text{Btu}/(\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}) & \text{for still air} \\ 2.7\ \text{Btu}/(\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}) & \text{for shaft-stirred air} \\ 5.9\ \text{Btu}/(\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}) & \text{for air moving at 500 ft/min} \end{cases} \qquad (12\text{–}18)$$

To find α

Table 12–2

Lubrication System	Conditions	Range of α
Oil ring	Moving air	1–2
	Still air	$\frac{1}{2}$–1
Oil bath	Moving air	$\frac{1}{2}$–1
	Still air	$\frac{1}{5}$–$\frac{2}{5}$

To find P (for design)

Table 12–5

Range of Unit Loads in Current Use for Sleeve Bearings

Application	Unit Load psi	Unit Load MPa
Diesel engines:		
Main bearings	900–1700	6–12
Crankpin	1150–2300	8–15
Wristpin	2000–2300	14–15
Electric motors	120–250	0.8–1.5
Steam turbines	120–250	0.8–1.5
Gear reducers	120–250	0.8–1.5
Automotive engines:		
Main bearings	600–750	4–5
Crankpin	1700–2300	10–15
Air compressors:		
Main bearings	140–280	1–2
Crankpin	280–500	2–4
Centrifugal pumps	100–180	0.6–1.2

Chapter 17 Flexible mechanical elements

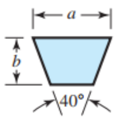
Belts

V-Belts

To find Standard size

Table 17–9

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 17–10

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 17–11

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

To find Tabulated rated power

Table 17–12

Horsepower Ratings of Standard V Belts

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

To find K_1

Table 17–13

Angle of Contact Correction Factor K_1 for VV* and V-Flat Drives

$\frac{D-d}{C}$	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

*A curve fit for the VV column in terms of θ is
$K_1 = 0.143\,543 + 0.007\,468\,\theta - 0.000\,015\,052\,\theta^2$
in the range $90° \le \theta \le 180°$.

To find K_2

Table 17–14

Belt-Length Correction Factor K_2^*

Length Factor	Nominal Belt Length, in: 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.

To find K_s

Table 17–15

Suggested Service Factors K_S for V-Belt Drives

Driven Machinery	Source of Power: Normal Torque Characteristic	Source of Power: High or Nonuniform Torque
Uniform	1.0 to 1.2	1.1 to 1.3
Light shock	1.1 to 1.3	1.2 to 1.4
Medium shock	1.2 to 1.4	1.4 to 1.6
Heavy shock	1.3 to 1.5	1.5 to 1.8

To find Kc, Kc

Table 17–16

Some V-Belt Parameters*

Belt Section	K_b	K_c
A	220	0.561
B	576	0.965
C	1 600	1.716
D	5 680	3.498
E	10 850	5.041
3V	230	0.425
5V	1098	1.217
8V	4830	3.288

*Data courtesy of Gates Rubber Co., Denver, Colo.

To find K,b for tension relation with cycles

Table 17–17

Durability Parameters for Some V-Belt Sections

Source: M. E. Spotts, *Design of Machine Elements,* 6th ed. Prentice Hall, Englewood Cliffs, N.J., 1985.

Belt Section	10^8 to 10^9 Force Peaks		10^9 to 10^{10} Force Peaks		Minimum Sheave Diameter, in
	K	*b*	*K*	*b*	
A	674	11.089			3.0
B	1193	10.926			5.0
C	2038	11.173			8.5
D	4208	11.105			13.0
E	6061	11.100			21.6
3V	728	12.464	1062	10.153	2.65
5V	1654	12.593	2394	10.283	7.1
8V	3638	12.629	5253	10.319	12.5

Flat and round belts

Table 17–2

Properties of Some Flat- and Round-Belt Materials. (Diameter = d, thickness = t, width = w)

Material	Specification	Size, in	Minimum Pulley Diameter, in	Allowable Tension per Unit Width at 600 ft/min, lbf/in	Specific Weight, lbf/in^3	Coefficient of Friction
Leather	1 ply	$t = \frac{11}{64}$	3	30	0.035–0.045	0.4
		$t = \frac{13}{64}$	$3\frac{1}{2}$	33	0.035–0.045	0.4
	2 ply	$t = \frac{18}{64}$	$4\frac{1}{2}$	41	0.035–0.045	0.4
		$t = \frac{20}{64}$	6^a	50	0.035–0.045	0.4
		$t = \frac{23}{64}$	9^a	60	0.035–0.045	0.4
Polyamide[b]	F–0[c]	$t = 0.03$	0.60	10	0.035	0.5
	F–1[c]	$t = 0.05$	1.0	35	0.035	0.5
	F–2[c]	$t = 0.07$	2.4	60	0.051	0.5
	A–2[c]	$t = 0.11$	2.4	60	0.037	0.8
	A–3[c]	$t = 0.13$	4.3	100	0.042	0.8
	A–4[c]	$t = 0.20$	9.5	175	0.039	0.8
	A–5[c]	$t = 0.25$	13.5	275	0.039	0.8
Urethane[d]	$w = 0.50$ in	$t = 0.062$	See	5.2^e	0.038–0.045	0.7
	$w = 0.75$ in	$t = 0.078$	Table	9.8^e	0.038–0.045	0.7
	$w = 1.25$ in	$t = 0.090$	17–3	18.9^e	0.038–0.045	0.7
	Round	$d = \frac{1}{4}$	See	8.3^e	0.038–0.045	0.7
		$d = \frac{3}{8}$	Table	18.6^e	0.038–0.045	0.7
		$d = \frac{1}{2}$	17–3	33.0^e	0.038–0.045	0.7
		$d = \frac{3}{4}$		74.3^e	0.038–0.045	0.7

[a]Add 2 in to pulley size for belts 8 in wide or more.

[b]*Source: Habasit Engineering Manual,* Habasit Belting, Inc., Chamblee (Atlanta), Ga.

[c]Friction cover of acrylonitrile-butadiene rubber on both sides.

[d]*Source:* Eagle Belting Co., Des Plaines, Ill.

[e]At 6% elongation; 12% is maximum allowable value.

To find C_p

Table 17–4

Pulley Correction Factor C_P for Flat Belts*

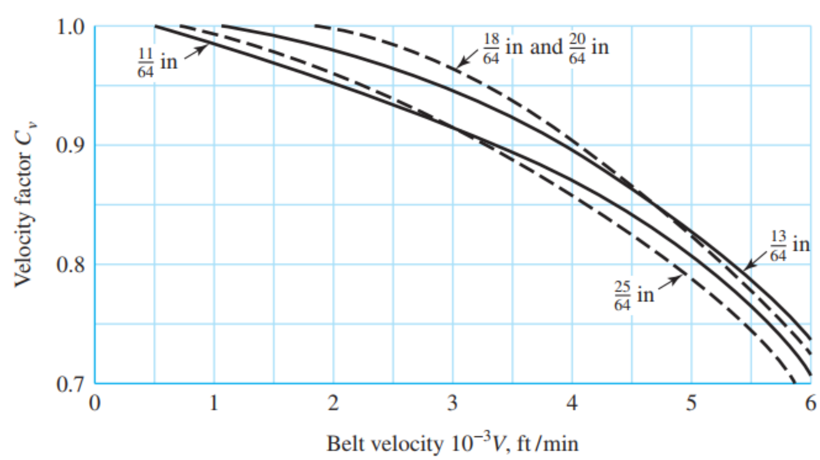
	Small-Pulley Diameter, in					
Material	**1.6 to 4**	**4.5 to 8**	**9 to 12.5**	**14, 16**	**18 to 31.5**	**Over 31.5**
Leather	0.5	0.6	0.7	0.8	0.9	1.0
Polyamide, F–0	0.95	1.0	1.0	1.0	1.0	1.0
F–1	0.70	0.92	0.95	1.0	1.0	1.0
F–2	0.73	0.86	0.96	1.0	1.0	1.0
A–2	0.73	0.86	0.96	1.0	1.0	1.0
A–3	—	0.70	0.87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

To find C_v

Figure 17–9

Velocity correction factor C_v for leather belts for various thicknesses. *(Data source:* Machinery's Handbook, 20th ed., *Industrial Press, New York, 1976, p. 1047.)*



Ropes and Wires

To find Sut of rope and weight per foot

Table 17–24

Wire-Rope Data *Source:* Compiled from *American Steel and Wire Company Handbook.*

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

*The modulus of elasticity is only approximate; it is affected by the loads on the rope and, in general, increases with the life of the rope.

†The strength is based on the nominal area of the rope. The figures given are only approximate and are based on 1-in rope sizes and $\frac{1}{4}$-in aircraft-cable sizes.

To find factor of safety

Table 17–25

Minimum Factors of Safety for Wire Rope*

Source: Compiled from a variety of sources, including ANSI A17.1-1978.

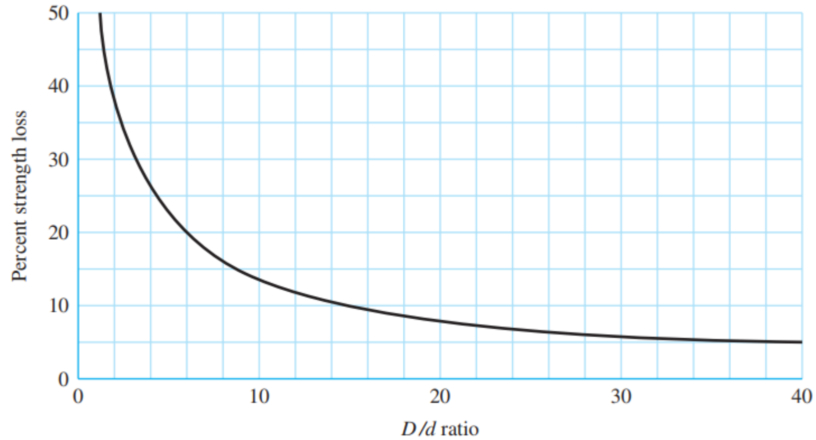
Track cables	3.2	Passenger elevators, ft/min:	
Guys	3.5	50	7.60
Mine shafts, ft:		300	9.20
Up to 500	8.0	800	11.25
1000–2000	7.0	1200	11.80
2000–3000	6.0	1500	11.90
Over 3000	5.0	Freight elevators, ft/min:	
Hoisting	5.0	50	6.65
Haulage	6.0	300	8.20
Cranes and derricks	6.0	800	10.00
Electric hoists	7.0	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

*Use of these factors does not preclude a fatigue failure.

To find %

Figure 17–20

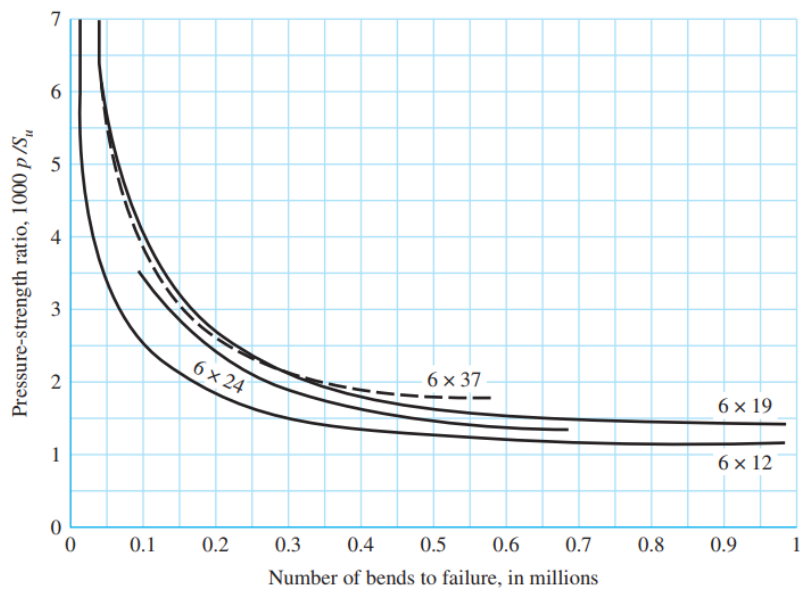
Percent strength loss due to different D/d ratios; derived from standard test data for 6×19 and 6×17 class ropes. *(Materials provided by the Wire Rope Technical Board (WRTB),* Wire Rope Users Manual Third Edition, *Second printing. Reprinted by permission.)*



To find 1000 P/S_u

Figure 17–21

Experimentally determined relation between the fatigue life of wire rope and the sheave pressure.



Chains

ANSI Chains

Table 17–19

Dimensions of American Standard Roller Chains—Single Strand

Source: Compiled from ANSI B29.1-1975.

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)

Rated horse Power

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	

*Estimated from ANSI tables by linear interpolation.

Note: Type A—manual or drip lubrication; type B—bath or disk lubrication; type C—oil-stream lubrication.

(*Continued*)

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′					

To find N (number of teeth)

Table 17–21 type C′—type C, but this is a galling region; submit design to manufacturer for evaluation.

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

*Morse Chain Company, Ithaca, NY, Type B hub sprockets.

904

To find K_1

Table 17–22

Tooth Correction Factors, K_1

Number of Teeth on Driving Sprocket	K_1 Pre-extreme Horsepower	K_1 Post-extreme Horsepower
11	0.62	0.52
12	0.69	0.59
13	0.75	0.67
14	0.81	0.75
15	0.87	0.83
16	0.94	0.91
17	1.00	1.00
18	1.06	1.09
19	1.13	1.18
20	1.19	1.28
N	$(N_1/17)^{1.08}$	$(N_1/17)^{1.5}$

To find K_2

Table 17–23

Multiple-Strand Factors, K_2

Number of Strands	K_2
1	1.0
2	1.7
3	2.5
4	3.3
5	3.9
6	4.6
8	6.0

Chapter 15 Bevel and Worm Gears

Bevel gears

Bending

To find K_o

Table 15–2

Overload Factors K_o (K_A)

Source: ANSI/AGMA 2003-B97.

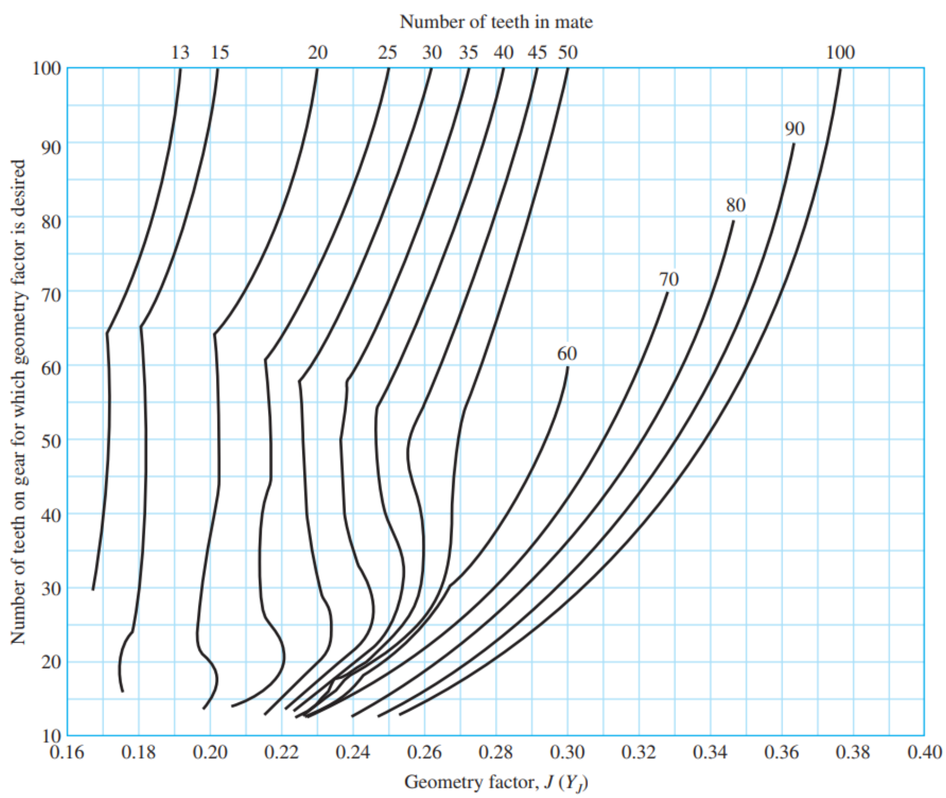
Character of Prime Mover	Character of Load on Driven Machine: Uniform	Light Shock	Medium Shock	Heavy Shock
Uniform	1.00	1.25	1.50	1.75 or higher
Light shock	1.10	1.35	1.60	1.85 or higher
Medium shock	1.25	1.50	1.75	2.00 or higher
Heavy shock	1.50	1.75	2.00	2.25 or higher

Note: This table is for speed-decreasing drives. For speed-increasing drives, add $0.01(N/n)^2$ or $0.01(z_2/z_1)^2$ to the above factors.

To find J: When Φ = 20 and angles sum is 90

Figure 15–7

Bending factor J (Y_J) for coniflex straight-bevel gears with a 20° normal pressure angle and 90° shaft angle. *(Source: ANSI/AGMA 2003-B97.)*



To find K_m

$$K_m = K_{mb} + 0.0036F^2 \qquad \text{(U.S. customary units)}$$
$$K_{H\beta} = K_{mb} + 5.6(10^{-6})b^2 \qquad \text{(SI units)} \qquad (15\text{–}11)$$

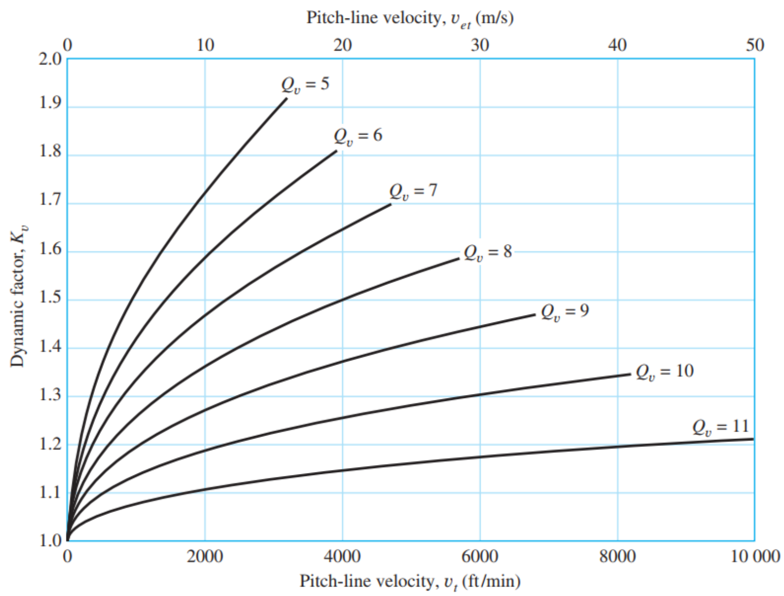
where

$$K_{mb} = \begin{cases} 1.00 & \text{both members straddle-mounted} \\ 1.10 & \text{one member straddle-mounted} \\ 1.25 & \text{neither member straddle-mounted} \end{cases}$$

To find K_v

Figure 15–5

Dynamic factor K_v. *(Source: ANSI/AGMA 2003-B97.)*



Or equations

$$K_v = \left(\frac{A + \sqrt{v_t}}{A}\right)^B \qquad \text{(U.S. customary units)}$$
$$K_v = \left(\frac{A + \sqrt{200v_{et}}}{A}\right)^B \qquad \text{(SI units)} \qquad (15\text{–}5)$$

where

$$A = 50 + 56(1 - B)$$
$$B = 0.25(12 - Q_v)^{2/3} \qquad (15\text{–}6)$$

To find Ks

Size Factor for Bending K_s (Y_x)

$$K_S = \begin{cases} 0.4867 + 0.2132/P_d & 0.5 \le P_d \le 16 \text{ teeth/in} \\ 0.5 & P_d > 16 \text{ teeth/in} \end{cases} \quad \text{(U.S. customary units)}$$

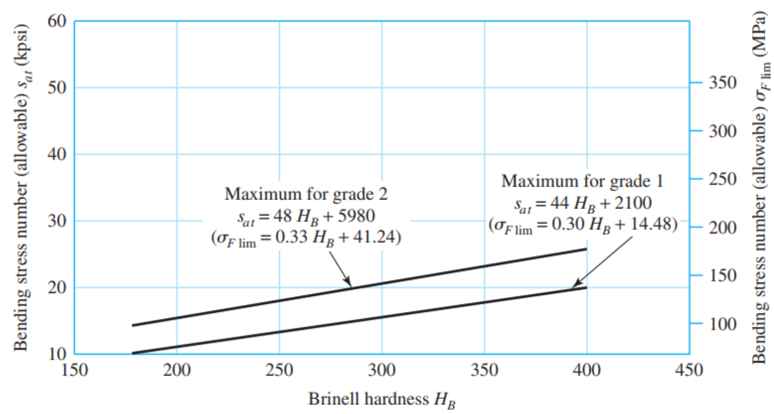
(15–10)

$$Y_x = \begin{cases} 0.5 & m_{et} < 1.6 \text{ mm} \\ 0.4867 + 0.008\,339 m_{et} & 1.6 \le m_{et} \le 50 \text{ mm} \end{cases} \quad \text{(SI units)}$$

To find S_t

Figure 15–13

Allowable bending stress number for through-hardened steel gears, $s_{at}(\sigma_{F\,\mathrm{lim}})$. *(Source: ANSI/AGMA 2003-B97.)*



Or

Table 15–6

Allowable Bending Stress Numbers for Steel Gears, s_{at} $(\sigma_{F\,\mathrm{lim}})$ *Source: ANSI/AGMA 2003-B97.*

Material Designation	Heat Treatment	Minimum Surface Hardness	Bending Stress Number (Allowable), s_{at} $(\sigma_{F\,\mathrm{lim}})$ lbf/in² (N/mm²)		
			Grade 1*	Grade 2*	Grade 3*
Steel	Through-hardened	Fig. 15–13	Fig. 15–13	Fig. 15–13	
	Flame or induction hardened				
	Unhardened roots	50 HRC	15 000 (85)	13 500 (95)	
	Hardened roots		22 500 (154)		
	Carburized and case hardened[†]	2003-B97 Table 8	30 000 (205)	35 000 (240)	40 000 (275)
AISI 4140	Nitrided[†,‡]	84.5 HR15N		22 000 (150)	
Nitralloy 135M	Nitrided[†,‡]	90.0 HR15N		24 000 (165)	

*See ANSI/AGMA 2003-B97, Tables 8–11, for metallurgical factors for each stress grade of steel gears.

[†]The allowable stress numbers indicated may be used with the case depths prescribed in 21.1, ANSI/AGMA 2003-B97.

[‡]The overload capacity of nitrided gears is low. Since the shape of the effective S-N curve is flat, the sensitivity to shock should be investigated before proceeding with the design.

To find K_L

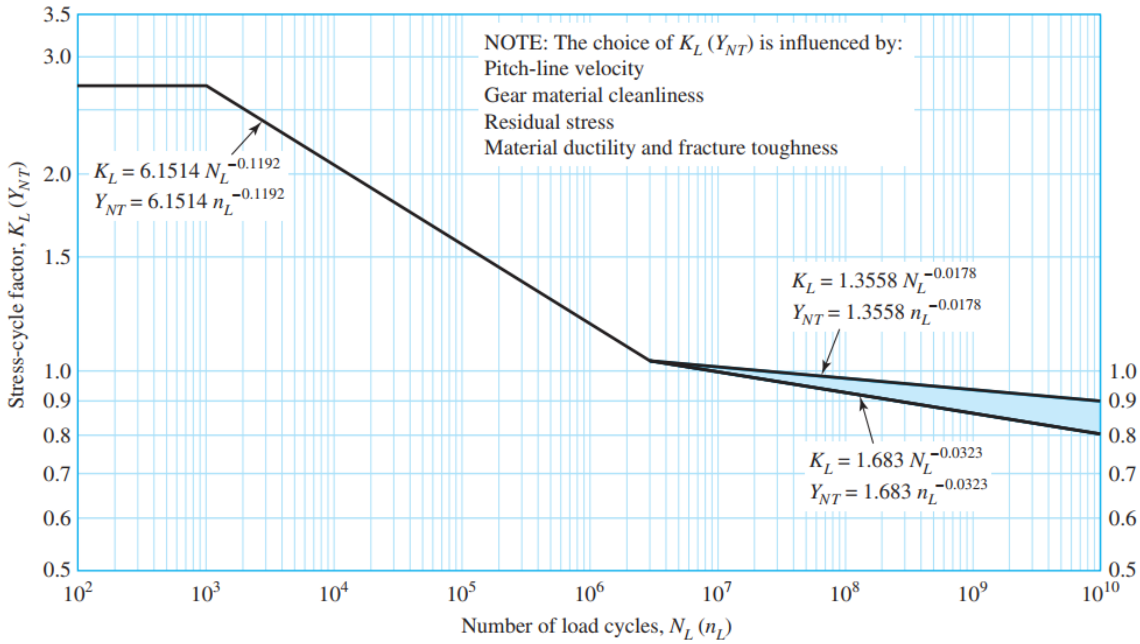


Figure 15–9

Stress-cycle factor for bending strength K_L (Y_{NT}) for carburized case-hardened steel bevel gears. *(Source: ANSI/AGMA 2003-B97.)*

To find K_r

Table 15–3

Reliability Factors

Source: ANSI/AGMA 2003-B97.

Requirements of Application	Reliability Factors for Steel* C_R (Z_Z)	Reliability Factors for Steel* K_R (Y_Z)†
Fewer than one failure in 10 000	1.22	1.50
Fewer than one failure in 1000	1.12	1.25
Fewer than one failure in 100	1.00	1.00
Fewer than one failure in 10	0.92	0.85‡
Fewer than one failure in 2	0.84	0.70§

*At the present time there are insufficient data concerning the reliability of bevel gears made from other materials.

†Tooth breakage is sometimes considered a greater hazard than pitting. In such cases a greater value of K_R (Y_Z) is selected for bending.

Or

$$Y_Z = K_R = \begin{cases} 0.50 - 0.25\log(1-R) & 0.99 \le R \le 0.999 \\ 0.70 - 0.15\log(1-R) & 0.90 \le R < 0.99 \end{cases} \qquad \begin{matrix} (15\text{–}19) \\ (15\text{–}20) \end{matrix}$$

Contact

To find C_s

$$C_s = \begin{cases} 0.5 & F < 0.5 \text{ in} \\ 0.125F + 0.4375 & 0.5 \le F \le 4.5 \text{ in} \\ 1 & F > 4.5 \text{ in} \end{cases} \qquad \text{(U.S. customary units)}$$

$$Z_x = \begin{cases} 0.5 & b < 12.7 \text{ mm} \\ 0.004\,92b + 0.4375 & 12.7 \le b \le 114.3 \text{ mm} \\ 1 & b > 114.3 \text{ mm} \end{cases} \qquad \text{(SI units)} \qquad (15\text{–}9)$$

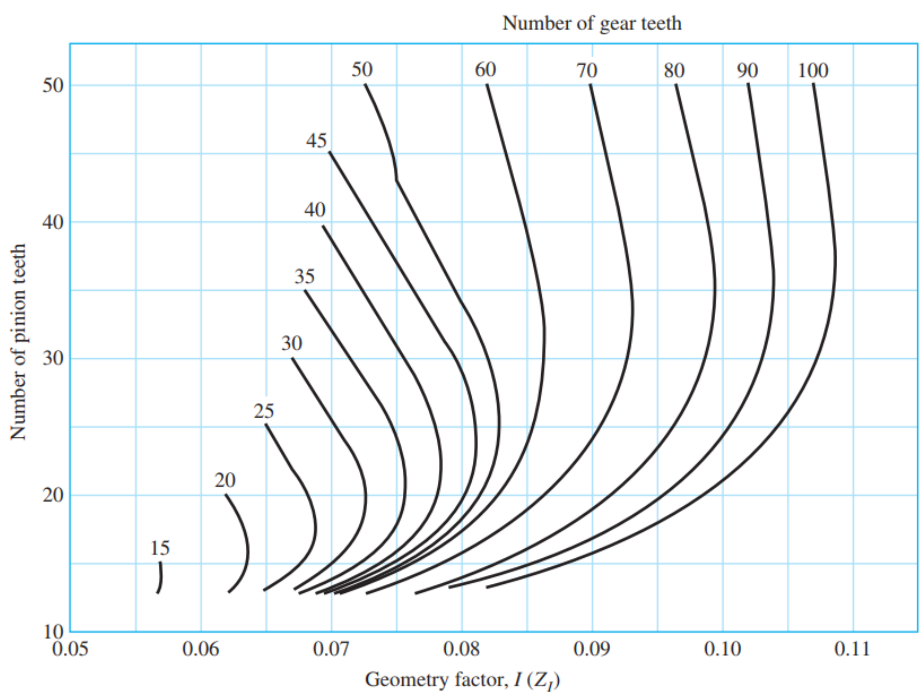
To find C_c

$$C_{xc} = Z_{xc} = \begin{cases} 1.5 & \text{properly crowned teeth} \\ 2.0 & \text{or larger uncrowned teeth} \end{cases} \qquad (15\text{–}12)$$

To find I: When $\Phi = 20$ and angles sum is 90

Figure 15–6

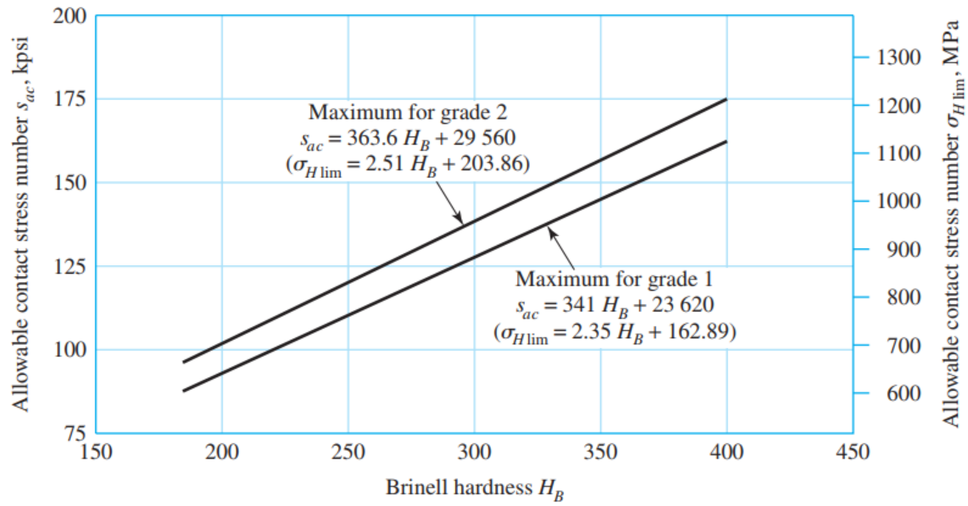
Contact geometry factor I (Z_I) for coniflex straight-bevel gears with a 20° normal pressure angle and a 90° shaft angle. *(Source: ANSI/AGMA 2003-B97.)*



To find Sc

Figure 15–12

Allowable contact stress number for through-hardened steel gears, $s_{ac}(\sigma_{H\ \mathrm{lim}})$. *(Source: ANSI/AGMA 2003-B97.)*



Or

Table 15–4

Allowable Contact Stress Number for Steel Gears, s_{ac} $(\sigma_{H\ \mathrm{lim}})$ *Source: ANSI/AGMA 2003-B97.*

Material Designation	Heat Treatment	Minimum Surface* Hardness	Allowable Contact Stress Number, s_{ac} $(\sigma_{H\ \mathrm{lim}})$ lbf/in^2 (N/mm^2) Grade 1†	Grade 2†	Grade 3†
Steel	Through-hardened‡	Fig. 15–12	Fig. 15–12	Fig. 15–12	
	Flame or induction hardened§	50 HRC	175 000 (1210)	190 000 (1310)	
	Carburized and case hardened§	2003-B97 Table 8	200 000 (1380)	225 000 (1550)	250 000 (1720)
AISI 4140	Nitrided§	84.5 HR15N		145 000 (1000)	
Nitralloy 135M	Nitrided§	90.0 HR15N		160 000 (1100)	

*Hardness to be equivalent to that at the tooth middepth in the center of the face width.

†See ANSI/AGMA 2003-B97, Tables 8 through 11, for metallurgical factors for each stress grade of steel gears.

To find C_L

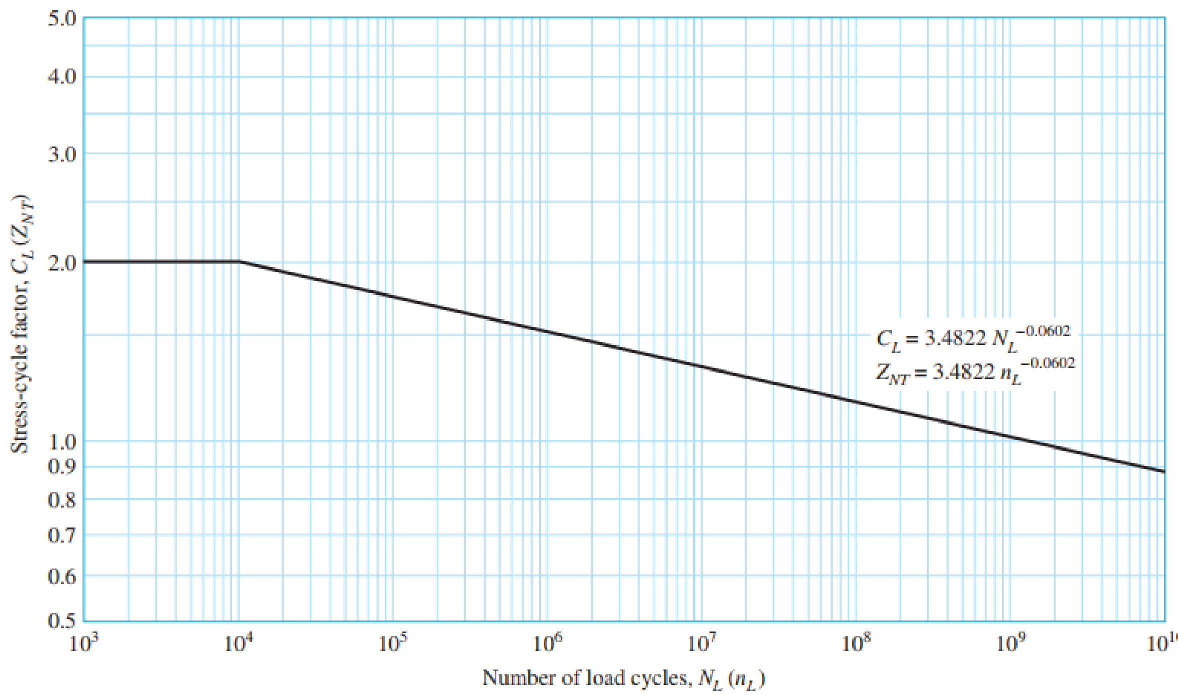


Figure 15–8

Contact stress-cycle factor for pitting resistance C_L (Z_{NT}) for carburized case-hardened steel bevel gears. *(Source: ANSI/AGMA 2003-B97.)*

To find C_H

$$C_H = 1.0 + A'(m_G - 1.0) \qquad (14\text{–}36)$$

where

$$A' = 8.98(10^{-3})\left(\frac{H_{BP}}{H_{BG}}\right) - 8.29(10^{-3}) \qquad 1.2 \le \frac{H_{BP}}{H_{BG}} \le 1.7$$

The terms H_{BP} and H_{BG} are the Brinell hardness (10-mm ball at 3000-kg load) of the pinion and gear, respectively. The term m_G is the speed ratio and is given by Eq. (14–22). See Fig. 14–12 for a graph of Eq. (14–36). For

$$\frac{H_{BP}}{H_{BG}} < 1.2, \qquad A' = 0$$

$$\frac{H_{BP}}{H_{BG}} > 1.7, \qquad A' = 0.006\,98$$

To find C_R

$$Y_Z = K_R = \begin{cases} 0.50 - 0.25\log(1-R) & 0.99 \le R \le 0.999 \qquad (15\text{–}19) \\ 0.70 - 0.15\log(1-R) & 0.90 \le R < 0.99 \qquad (15\text{–}20) \end{cases}$$

Worm gears

Contact

To find C_s

where D_m is the mean gear diameter.

The parameters in Eq. (15–28) are, quantitatively,

$$C_s = 720 + 10.37C^3 \qquad C \le 3 \text{ in} \tag{15–32}$$

For sand-cast gears,

$$C_s = \begin{cases} 1000 & C > 3 \quad D_m \le 2.5 \text{ in} \\ 1190 - 477 \log D_m & C > 3 \quad D_m > 2.5 \text{ in} \end{cases} \tag{15–33}$$

For chilled-cast gears,

$$C_s = \begin{cases} 1000 & C > 3 \quad D_m \le 8 \text{ in} \\ 1412 - 456 \log D_m & C > 3 \quad D_m > 8 \text{ in} \end{cases} \tag{15–34}$$

For centrifugally cast gears,

$$C_s = \begin{cases} 1000 & C > 3 \quad D_m \le 25 \text{ in} \\ 1251 - 180 \log D_m & C > 3 \quad D_m > 25 \text{ in} \end{cases} \tag{15–35}$$

To find C_v

The velocity factor C_v is given by

$$C_v = \begin{cases} 0.659 \exp(-0.0011 V_s) & V_s < 700 \text{ ft/min} \\ 13.31\, V_s^{-0.571} & 700 \le V_s < 3000 \text{ ft/min} \\ 65.52\, V_s^{-0.774} & V_s > 3000 \text{ ft/min} \end{cases} \tag{15–37}$$

To find C_m

The ratio correction factor C_m for gear ratio m_G is given by

$$C_m = \begin{cases} 0.02\sqrt{-m_G^2 + 40m_G - 76} + 0.46 & 3 < m_G \le 20 \\ 0.0107\sqrt{-m_G^2 + 56m_G + 5145} & 20 < m_G \le 76 \\ 1.1483 - 0.006\,58m_G & m_G > 76 \end{cases} \tag{15–36}$$

To find f coefficient of friction

$$f = \begin{cases} 0.15 & V_s = 0 \\ 0.124 \exp(-0.074 V_s^{0.645}) & 0 < V_s \leq 10 \text{ ft/min} \\ 0.103 \exp(-0.110 V_s^{0.450}) + 0.012 & V_s > 10 \text{ ft/min} \end{cases} \quad (15\text{–}38)$$

Wear

To find K_w

Table 15–11

Wear Factor K_w for Worm Gearing

Source: Earle Buckingham, *Design of Worm and Spiral Gears,* Industrial Press, New York, 1981.

Material		Thread Angle ϕ_n			
Worm	Gear	$14\frac{1}{2}°$	20°	25°	30°
Hardened steel*	Chilled bronze	90	125	150	180
Hardened steel*	Bronze	60	80	100	120
Steel, 250 BHN (min.)	Bronze	36	50	60	72
High-test cast iron	Bronze	80	115	140	165
Gray iron†	Aluminum	10	12	15	18
High-test cast iron	Gray iron	90	125	150	180
High-test cast iron	Cast steel	22	31	37	45
High-test cast iron	High-test cast iron	135	185	225	270
Steel 250 BHN (min.)	Laminated phenolic	47	64	80	95
Gray iron	Laminated phenolic	70	96	120	140

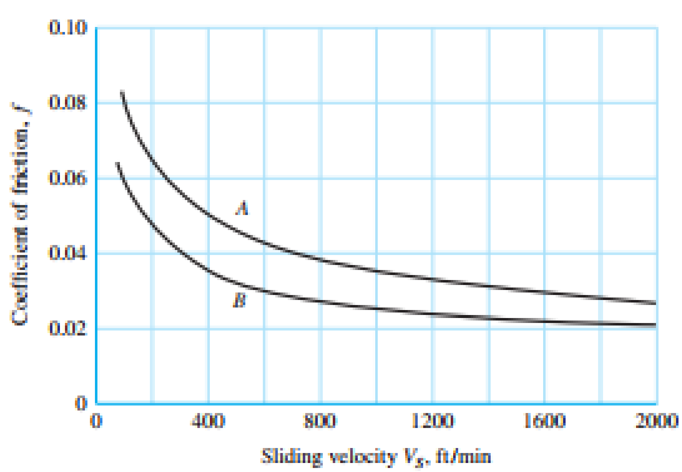
*Over 500 BHN surface.

†For steel worms, multiply given values by 0.6.

To find f

Figure 13–42

Representative values of the coefficient of friction for worm gearing. These values are based on good lubrication. Use curve *B* for high-quality materials, such as a case-hardened steel worm mating with a phosphor-bronze gear. Use curve *A* when more friction is expected, as with a cast-iron worm mating with a cast-iron worm gear.



To find a, b

Table 13–5

Recommended Pressure Angles and Tooth Depths for Worm Gearing

Lead Angle λ, deg	Pressure Angle ϕ_n, deg	Addendum a	Dedendum b_G
0–15	$14\frac{1}{2}$	$0.3683p_x$	$0.3683p_x$
15–30	20	$0.3683p_x$	$0.3683p_x$
30–35	25	$0.2865p_x$	$0.3314p_x$
35–40	25	$0.2546p_x$	$0.2947p_x$
40–45	30	$0.2228p_x$	$0.2578p_x$

Relation between Normal pressure angle and lead angle

Table 15–9

Largest Lead Angle Associated with a Normal Pressure Angle ϕ_n for Worm Gearing

ϕ_n	Maximum Lead Angle λ_{max}
14.5°	16°
20°	25°
25°	35°
30°	45°

To find f coefficient of friction

$$f = \begin{cases} 0.15 & V_s = 0 \\ 0.124 \exp(-0.074 V_s^{0.645}) & 0 < V_s \leq 10 \text{ ft/min} \\ 0.103 \exp(-0.110 V_s^{0.450}) + 0.012 & V_s > 10 \text{ ft/min} \end{cases} \qquad (15\text{–}38)$$