Stupid Mech 325 Stupid Summary

Fuck this course

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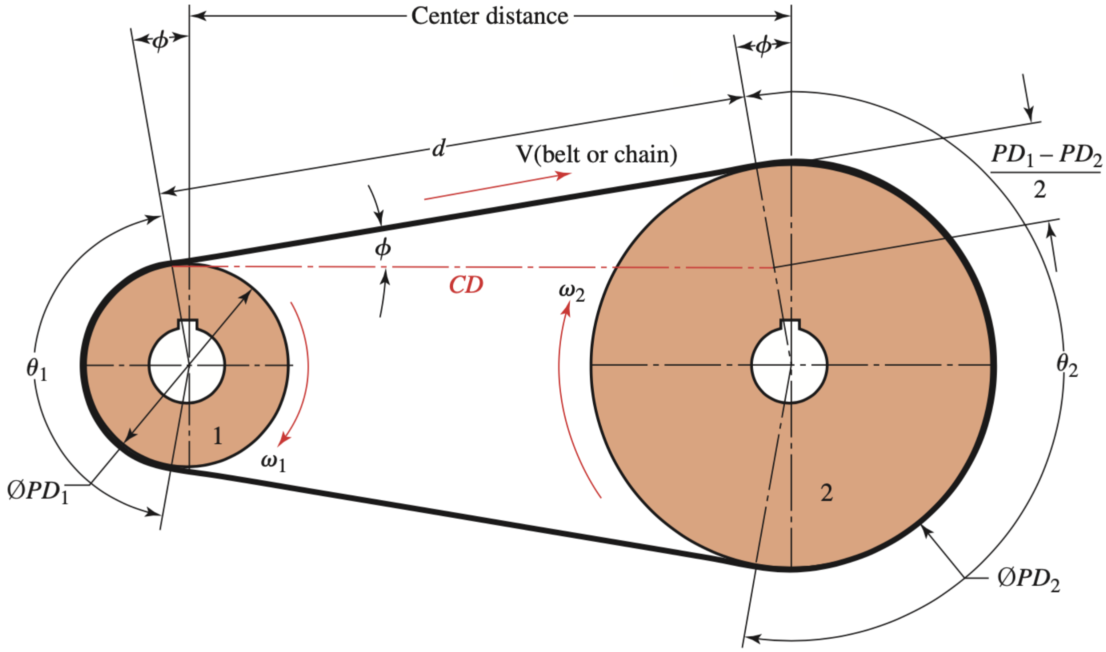
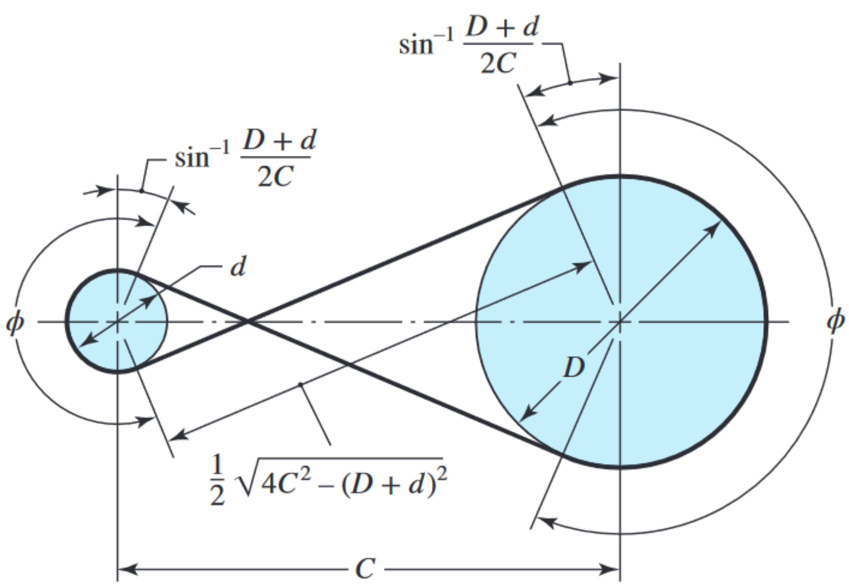
Contents

A note on the text:
Mech 325 is a stupid course that follows no laws or logic. I don't know what in the fuck is going on so use this guide at your own risk.

1 Belts and Shit

1.1 General Anatomy





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

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

1.2 General Nomenclature

$PD = D =$ pitch diameter (in)
$\omega = n =$ angular speed of sprocket/sheave (rpm)

VR = velocity ratio
v_b = belt speed
CD = center distance (in)
ø = random angle that helps solve for the wrap angle (°)
θ = angle of wrap (°)
s = arc length (length of belt/chain wrap on sprocket)
d = distance (or span) (belt/chain length that is tangent to sprockets)
L_p = belt/chain total length
H_{in} = input power (hp)
P_{des} = power (hp)
P_{rated} = rated power (hp)
SF = service factor

1.3 Flat Belts

1.3.1 Nomenclature

F_1 = taut-side tension
F_2 = slack-side tension
F_c = centrifugal tension
F_i = initial tension
f = maximum coefficient of friction
T = transmitted torque
w = weight per foot (lb/ft)
V = belt speed (ft/min)
H = transmitted power (hp)
b = belt width (in)
t = belt thickness (in)
γ = specific weight (lb/in^3)
$(F_1)_a$ =largest allowable tension
F_a = allowable tension/unit width
C_P = pulley correction factor (tab. 17-4)
C_V = velocity correction factor (p. 889)
H_{nom} = nominal (rated) power
H_a = design power
K_s = service factor
n_d = design safety factor
n_f = factor of safety
n = angular velocity (rpm)

1.3.2 Formulae

$\Delta F = (F_1)_a - F_2 = \frac{2T}{d}$
$F_1 - F_2 = \frac{2T}{d}$
$F_1 = F_c + \frac{2F_i e^{f\phi}}{e^{f\phi}+1}$
$F_2 = F_c + \frac{2F_i}{e^{f\phi}+1}$

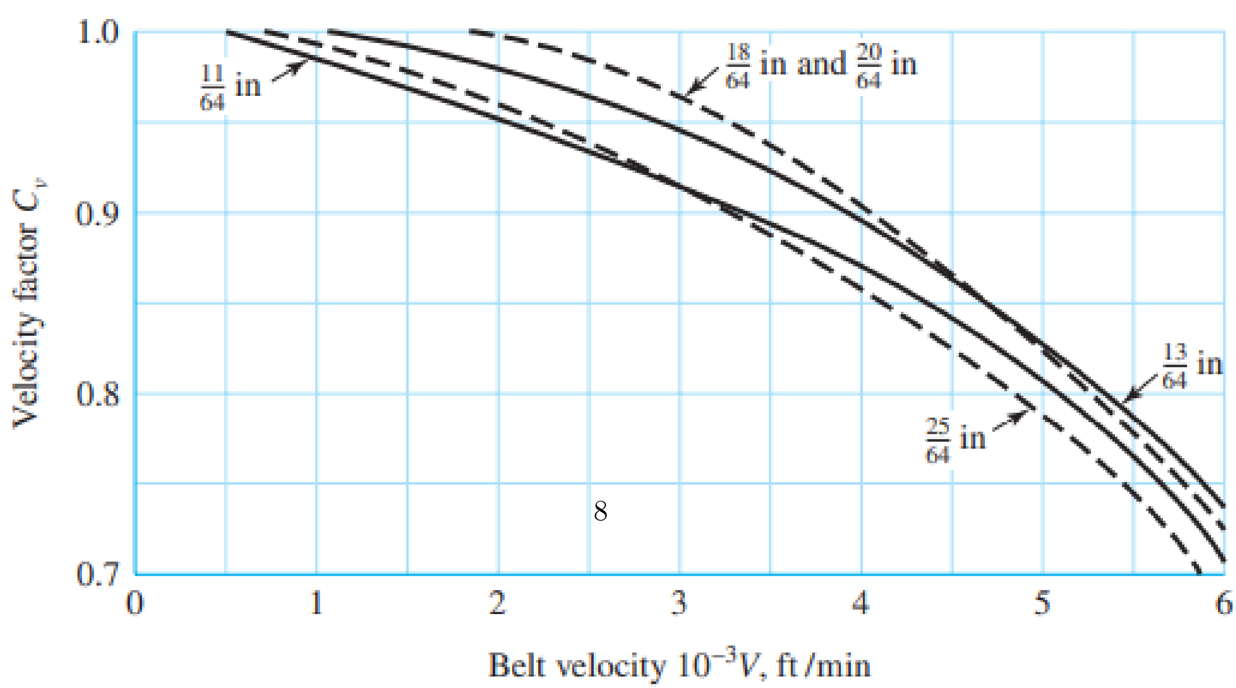
$\frac{F_1 - F_c}{F_2 - F_c} = e^{f\phi}$
$F_c = \frac{w}{32.17\,\mathrm{ft/s^2}} \left(\frac{V}{60\,\mathrm{s/min}}\right)^2$
$F_i = \frac{F_1 + F_2}{2} - F_c = \frac{T}{d} \frac{e^{f\phi}+1}{e^{f\phi}-1}$
$H = \frac{(F_1 - F_2)V}{33{,}000\,(\frac{\mathrm{ft\,lb}}{\mathrm{min}})/\mathrm{hp}}$
$w = 12\,\mathrm{in/ft}\,\gamma b t$
$(F_1)_a = b F_a C_P C_V$
$H_d = H_{\mathrm{nom}} K_s n_d$
$H_a = H_{\mathrm{nom}} K_s n_d$
$n_{\mathrm{fs}} = \frac{H_a}{H_{\mathrm{nom}} K_s}$
$T = 63025 \frac{H_{\mathrm{nom}} K_s n_d}{n} = 63025 \frac{H_d}{n}$
$f' = \frac{1}{\phi} \ln\left(\frac{(F_1)_a - F_c}{F_2 - F_c}\right)$
$\mathrm{dip} = \frac{C^2 w}{96\,\mathrm{in/ft} F_i}$
$F_1 = (F_1)_a$ at operation limit
require $f' < f$

1.3.3 Tables for Constants

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 Table 17–3	5.2[e]	0.038–0.045	0.7
	$w = 0.75$ in	$t = 0.078$		9.8[e]	0.038–0.045	0.7
	$w = 1.25$ in	$t = 0.090$		18.9[e]	0.038–0.045	0.7
	Round	$d = \frac{1}{4}$	See Table 17–3	8.3[e]	0.038–0.045	0.7
		$d = \frac{3}{8}$		18.6[e]	0.038–0.045	0.7
		$d = \frac{1}{2}$		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.



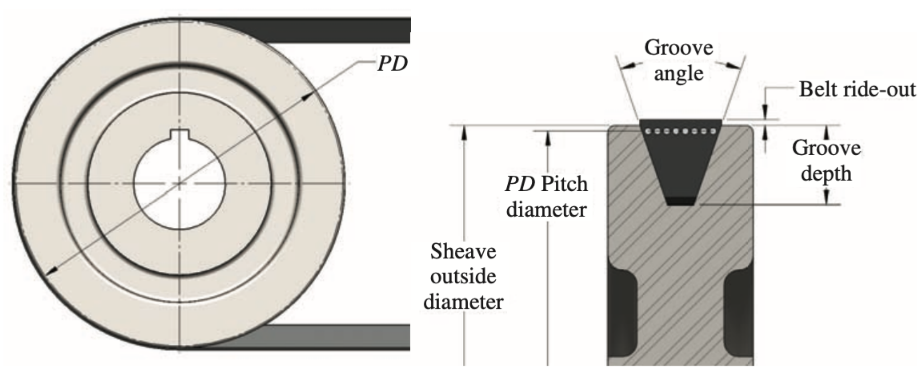
$C_v = 1$ for polyamide and urethane belts

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

1.4 V-Belt Drives

1.4.1 Anatomy



1.4.2 Design Selection

1. Compute the design power

   (a) Find the service factor based from this table:

TABLE 7–1 V-Belt Service Factors[1]

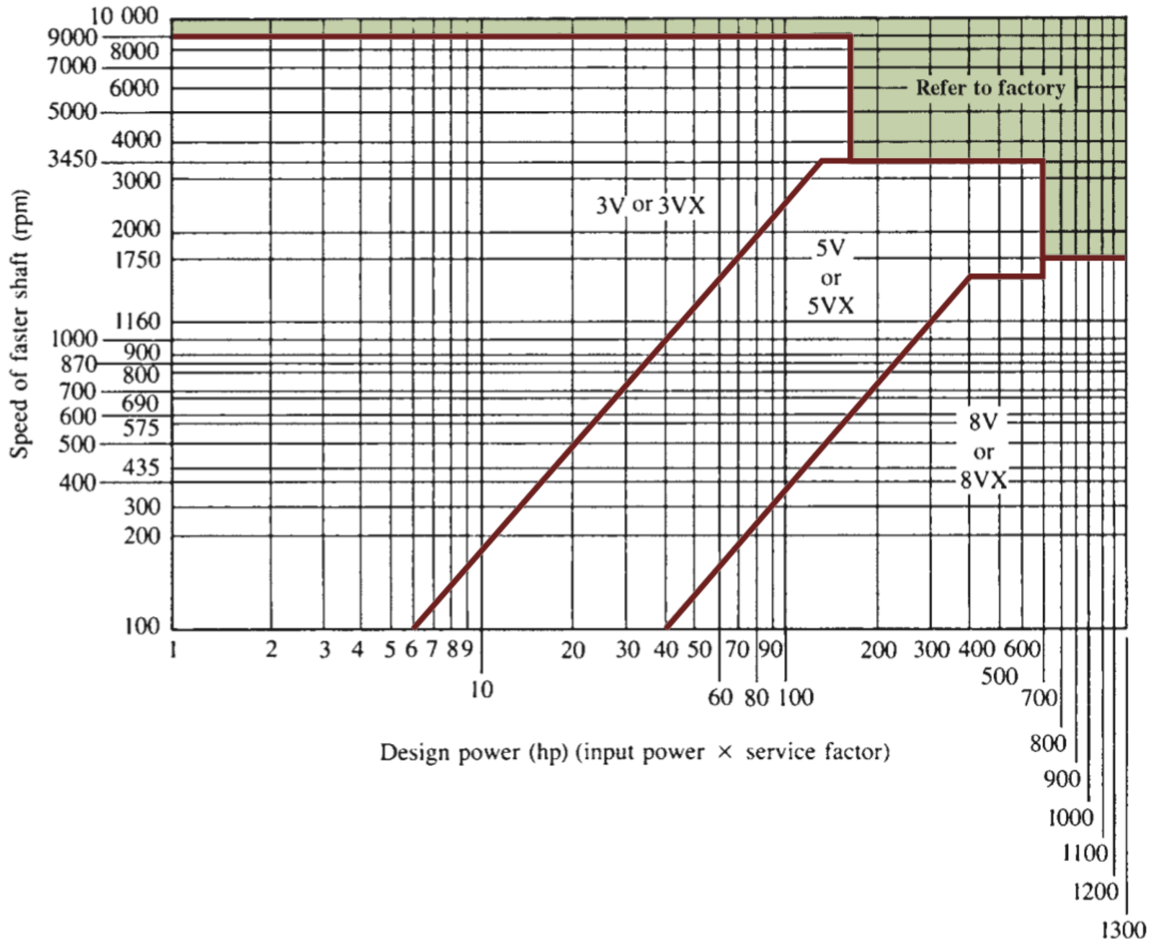
Driven machine type	Driver type: AC motors: Normal torque[2]; DC motors: Shunt-wound; Engines: Multiple-cylinder			Driver type: AC motors: High torque[3]; DC motors: Series-wound, or compound-wound; Engines: 4-cylinder or less		
	<6 h per day	6–15 h per day	>15 h per day	<6 h per day	6–15 h per day	>15 h per day
Smooth loading	1.0	1.1	1.2	1.1	1.2	1.3
Agitators, light conveyors, centrifugal pumps fans and blowers under 10 hp (7.5 kW)						
Light shock loading	1.1	1.2	1.3	1.2	1.3	1.4
Generators, machine tools mixers, fans and blowers over 10 hp (7.5 kW) gravel conveyors						
Moderate shock loading	1.2	1.3	1.4	1.4	1.5	1.6
Bucket elevators, piston pumps textile machinery, hammer mills heavy conveyors, pulverizers						
Heavy shock loading	1.3	1.4	1.5	1.5	1.6	1.8
Crushers, ball mills, hoists rubber mills, and extruders						
Machinery that can choke	2.0	2.0	2.0	2.0	2.0	2.0

[1]Factors given are for speed reducers. For speed increases, multiply listed factors by 1.2.
[2]Synchronous, split-phase, three-phase with starting torque or breakdown torque less than 175% of full-load torque.
[3]Single-phase, three-phase with starting torque or breakdown torque greater than 175% of full-load torque.

(b) Compute design power using:

$$P_{des} = H_{in} \cdot SF$$

2. Select the belt section
If at the boundary between two different types of belts, be conservative and choose the larger one



3. Compute the nominal speed ratio

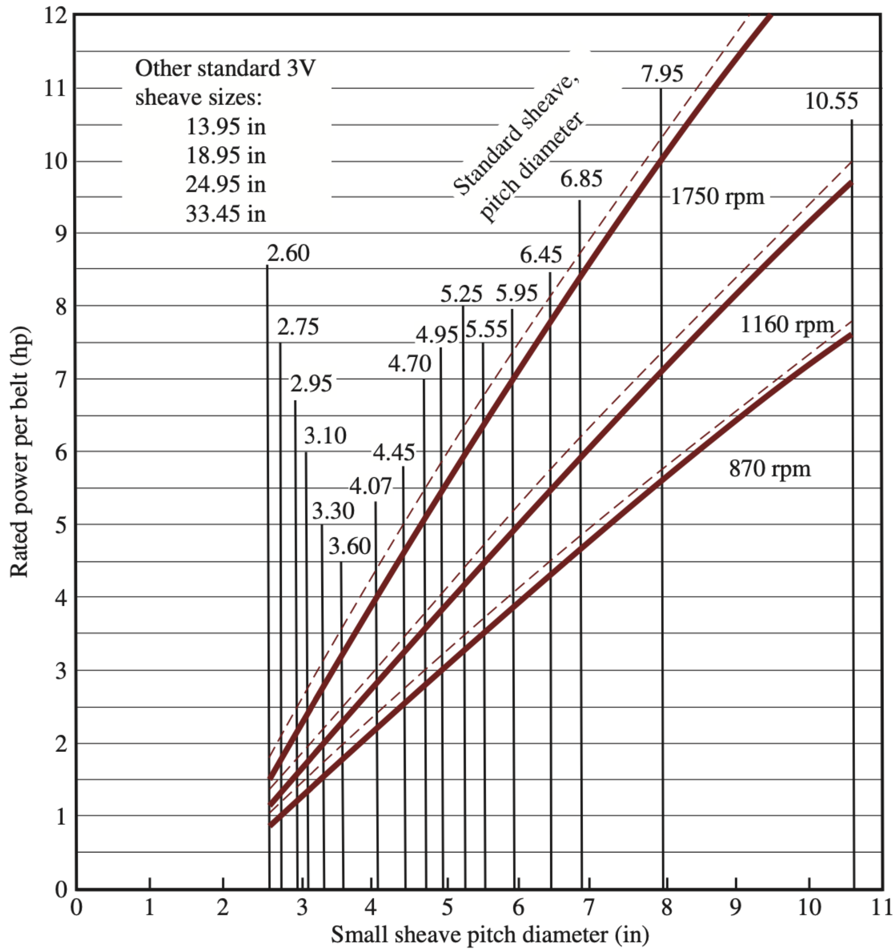
$$VR = \frac{n_1}{n_2} \text{ where } n_1 > n_2$$

4. Select the driving sheave size to produce a belt speed of 4000 ft/min (this is a standard speed we use since belt speed should not surpass 5000 ft/min, with a hard max at 6500 ft/min)

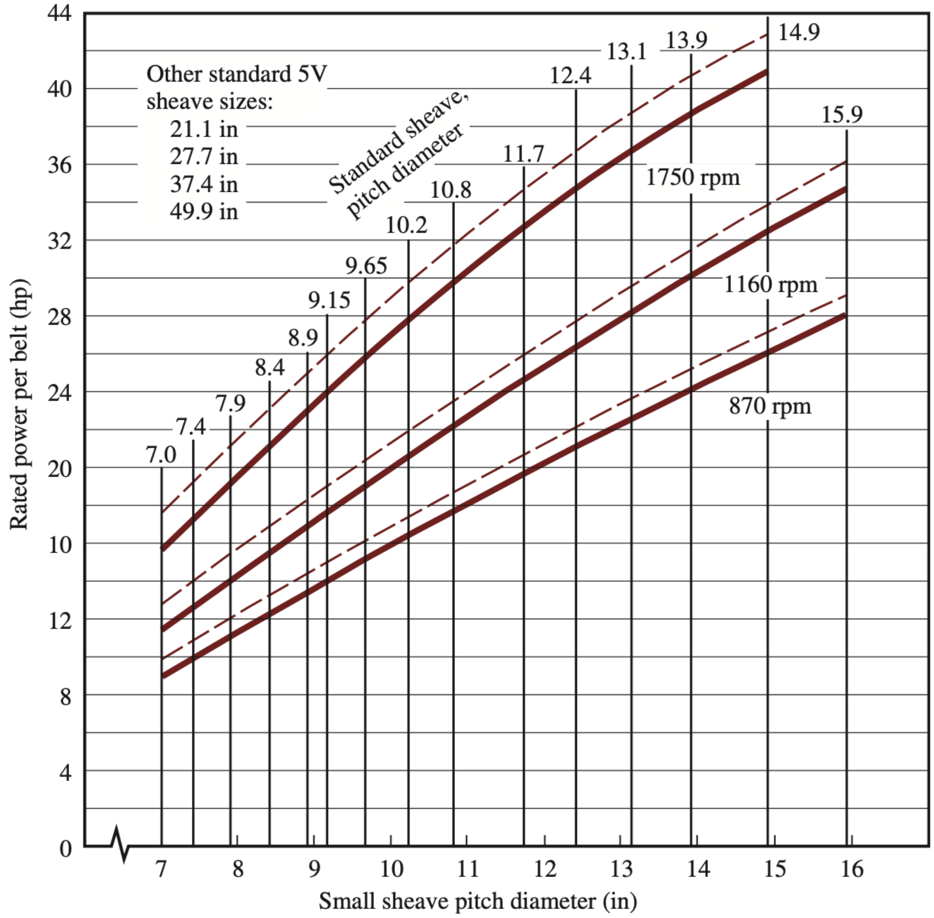
$$v_b = \frac{D_1 n_1}{2} \cdot (12 \text{ in/ft})(1 \text{ rev}/(2\pi \text{ rad}))$$

5. Select the standard sizes for the input and output sheaves

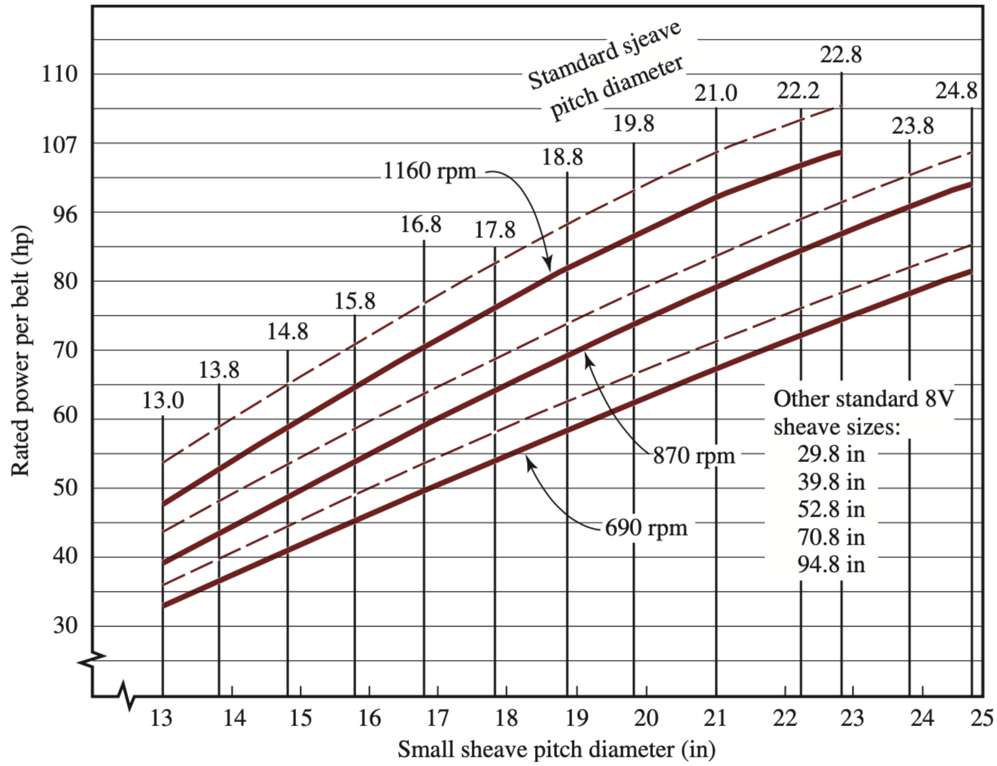
   (a) Select the closest standard size to the input sheave For 3V belts:


Other standard 3V
sheave sizes:
13.95 in
18.95 in
24.95 in
33.45 in
Standard sheave,
pitch diameter
2.60
2.75
2.95
3.10
3.30
3.60
4.07
4.45
4.70
4.95
5.25
5.55
5.95
6.45
6.85
7.95
10.55
1750 rpm
1160 rpm
870 rpm
Rated power per belt (hp)
Small sheave pitch diameter (in)

For 5V belts:


Other standard 5V
sheave sizes:
21.1 in
27.7 in
37.4 in
49.9 in
Standard sheave,
pitch diameter
7.0
7.4
7.9
8.4
8.9
9.15
9.65
10.2
10.8
11.7
12.4
13.1
13.9
14.9
15.9
1750 rpm
1160 rpm
870 rpm
Rated power per belt (hp)
Small sheave pitch diameter (in)

For 8V belts:



(b) Find the output sheave size using $D_2 = D_1 \cdot VR$ where $D_2 > D_1$

(c) Find the closest standard size to the output sheave using the same figures as above

6. Compute the actual speed ratio and belt speed

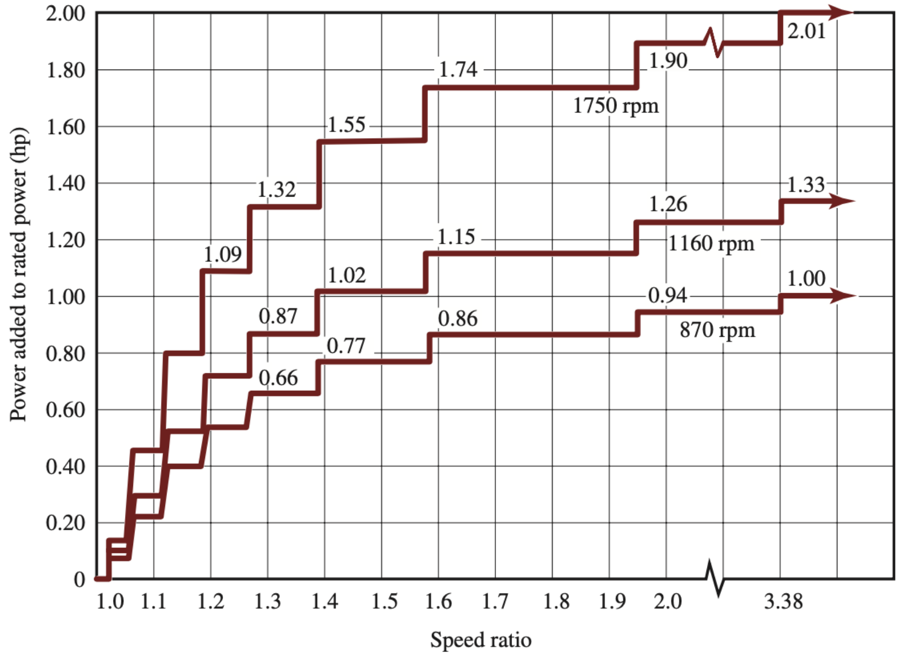
$$VR = \frac{D_2}{D_1}$$

$$v_b = \frac{D_1 n_1}{2} \cdot (12 \text{ in/ft})(1 \text{ rev}/(2\pi \text{ rad}))$$

7. Determine the rated power per belt

(a) Use the above figures to find the rated power per belt

(b) If the actual speed ratio is higher than 1, use the following table to find the power added:



(c) The total rated power per belt (P_{rated}) is the sum of both

8. Specify a trial center distance, CD, that is within the following range:

$$D_2 < CD < 3(D_2 + D_1)$$

9. Compute the required belt length

$$L_p = 2CD + 1.57(D_2 + D_1) + \frac{(D_2 - D_1)^2}{4C}$$

10. Select the closest standard belt length value from the following table:

TABLE 7–2 Standard Belt Lengths for 3V, 5V, and 8V Belts (in)

3V only	3V and 5V	3V, 5V, and 8V	5V and 8V	8V only
25	50	100	150	375
26.5	53	106	160	400
28	56	112	170	425
30	60	118	180	450
31.5	63	125	190	475
33.5	67	132	200	500
35.5	71	140	212	
37.5	75		224	
40	80		236	
42.5	85		250	
45	90		265	
47.5	95		280	
			300	
165			315	
			335	
			355	

11. Using the standard belt length, compute the actual CD. First compute B (a random constant) cause it'll help simplify the CD expression

$$B = 4L_p - 6.28(D_2 + D_1)$$

$$CD = \frac{B + \sqrt{B^2 - 32(D_2 - D_1)^2}}{16}$$

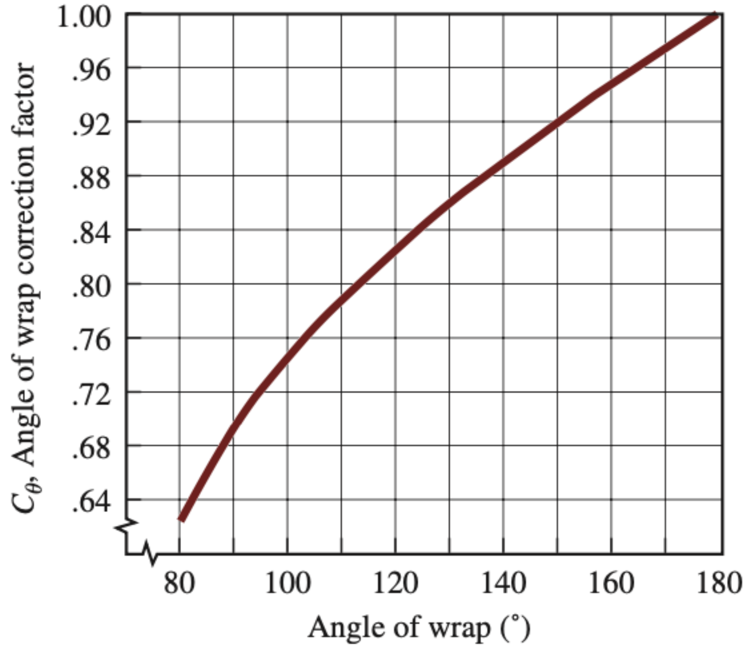
12. Compute the angle of wrap of small sheave

$$\theta_1 = 180^\circ - 2\sin^{-1}\left(\frac{D_2 - D_1}{2CD}\right)$$

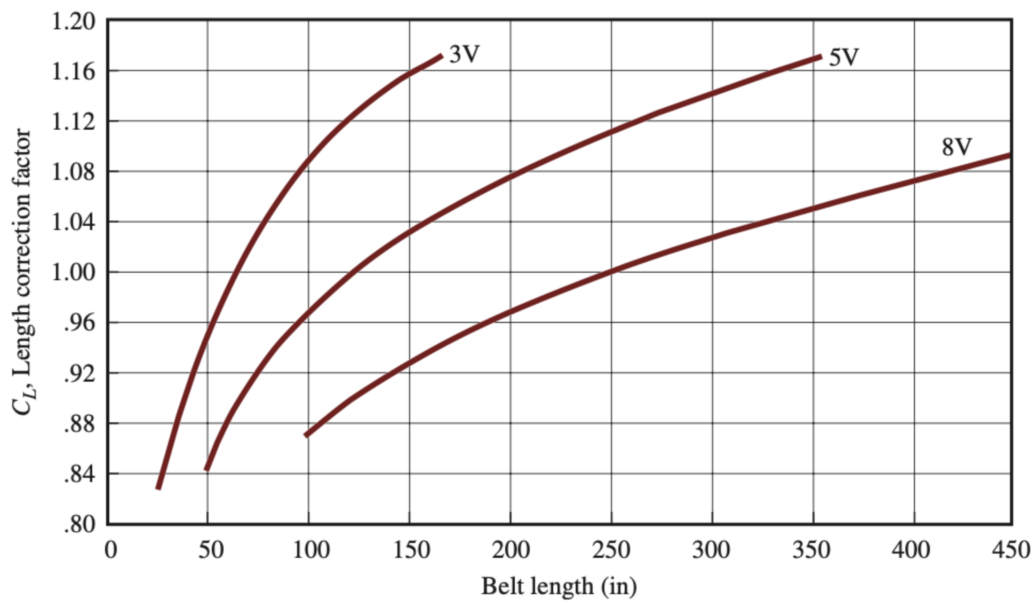
Probably won't be asked to compute the angle of wrap for the big sheave, but just in case, here is the formula:

$$\theta_2 = 180^\circ + 2\sin^{-1}\left(\frac{D_2 - D_1}{2CD}\right)$$

13. Determine the angle of wrap correction factor C_θ



14. Determine the belt length correction factor C_{L_p}



15. Determine the required number of belts

    (a) Calculate the corrected power rating $= C_\theta C_{L_p} P$

(b) Calculate the minimum number of belts required

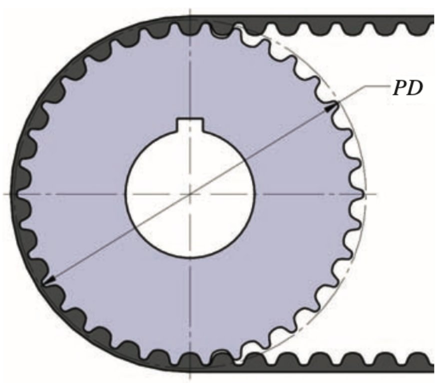
$$\text{min number of belts} = \frac{\text{design power}}{\text{corrected power rating}}$$

(c) Round up to the nearest integer

And that's it! You're doing great!!!

1.5 Synchronous Belt Drives

1.5.1 Anatomy



1.5.2 Design Selection

1. Compute the design power

   (a) Find the service factor from this table

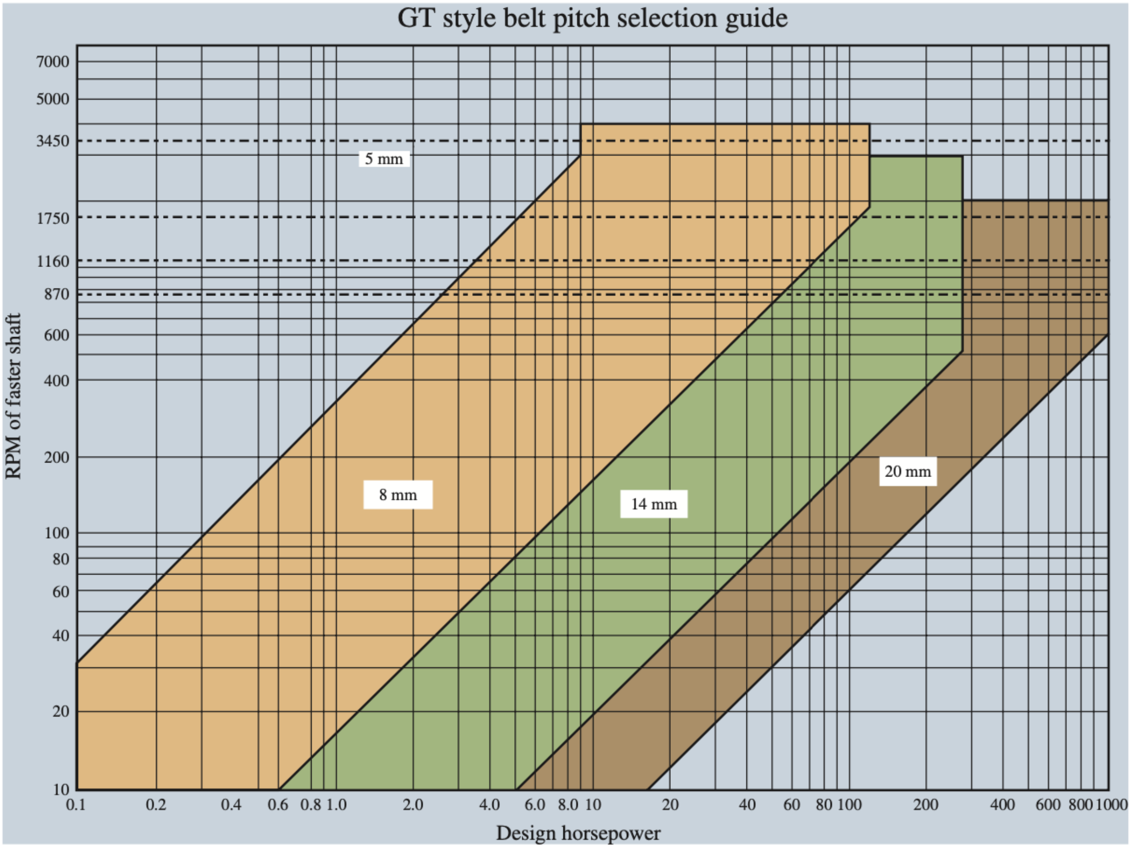
TABLE 7–8 Service Factor

DriveN machine	DriveR					
The driveN machines listed below are representative samples only. Select a driveN machine whose load characteristics most closely approximate those of the machine being considered.	**AC Motors: Normal Torque, Squirrel Cage, Synchronous, Split Phase, Inverter Controlled DC Motors: Shunt Wound Stepper Motors Engines: Multiple Cylinder Internal Combustion**			**AC Motors: High Torque, High Slip, Repulsion-Induction, Single Phase, Series Wound, Slip Ring DC Motors: Series Wound, Compound Wound Servo Motors Engines: Single Cylinder Internal Combustion, Line Shafts, Clutches**		
	Intermittent Service (Up to 8 Hours Daily or Seasonal)	**Normal Service (8–16 Hours Daily)**	**Continuous Service (16–24 Hours Daily)**	**Intermittent Service (Up to 8 Hours Daily or Seasonal)**	**Normal Service (8–16 Hours Daily)**	**Continuous Service (16–24 Hours Daily)**
Display, Dispending Equipment Instrumentation Measuring Equipment Medical Equipment Office, Projection Equipment	1.0	1.2	1.4	1.2	1.4	1.6
Appliances, Sweepers, Sewing Machines Screens, Oven Screens, Drum, Conical Woodworking Equipment (Light): Band Saws, Drills Lathes	1.1	1.3	1.5	1.3	1.5	1.7
Agitators for Liquids Conveyors: Belt, Light Package Drill Press, Lathes, Saws Laundry Machinery Wood Working Equipment (Heavy): Circular Saws, Jointers, Planers	1.2	1.4	1.6	1.6	1.8	2.0
Agitators for Semi-Liquids Compressor: Centrifugal Conveyor Belt: Ore, Coal, Sand Dough Mixers Line Shafts Machine Tools: Grinder, Shaper, Boring Mill, Milling Machines Paper Machinery (except Pulpers): Presses, Punches, Shears Printing Machinery Pumps: Centrifugal, Gear Screens: Revolving, Vibratory	1.3	1.5	1.7	1.6	1.8	2.0
Brick Machinery (except Pug Mills) Conveyor: Apron, Pan, Bucket, Elevator Extractors, Washers Fans, Centrifugal Blowers Generators & Exciters Hoists Rubber Calender, Mills, Extruders	1.4	1.6	1.8	1.8	2.0	2.2
Centrifuges Screw Conveyors Hammer Mills Paper Pulpers Textile Machinery	1.5	1.7	1.9	1.9	2.1	2.3
Blowers: Positive Displacement, Mine Fans Pulvertzers	1.6	1.8	2.0	2.0	2.2	2.4
Compressors: Reciprocating Crushers: Gyratory, Jaw, Rol Mils: Ball, Rod, Pebble, etc. Pumps: Reciprocating Saw Mill Equipment	1.7	1.9	2.1	2.1	2.3	2.5

These service factors are adequate for most belt drive applications. Note that service factors cannot be substituted for good engineering judgment. Service factors may be adjusted based upon an understanding of the severity of actual drive operating conditions.

(b) $P_{des} = P_{rated} \cdot SF$

2. Find the required pitch for the belt using this figure:



(You're probably going to get an 8mm pitch because that is all this textbook has data for...)

3. Compute the velocity ratio using $VR = \frac{n_{\text{driving}}}{n_{\text{driven}}}$

4. Select candidate combinations of driver and driven sprockets based on the VR.
You should have multiple possible combinations. List them all out, and then we will eliminate some in the next step.

TABLE 7–7 8-mm Pitch GT Drive Selection Table

Sprocket combinations			Center distance (inches)															
			920-8MGT	960-8MGT	1040-8MGT	1064-8MGT	1120-8MGT	1160-8MGT	1200-8MGT	1224-8MGT	1280-8MGT	1440-8MGT	1512-8MGT	1584-8MGT	1600-8MGT	1760-8MGT	1800-8MGT	2000-8MGT
			P.L. 36.220	P.L. 37.795	P.L. 40.945	P.L. 41.890	P.L. 44.094	P.L. 45.669	P.L. 47.244	P.L. 48.189	P.L. 50.394	P.L. 56.693	P.L. 59.528	P.L. 62.362	P.L. 62.992	P.L. 69.291	P.L. 70.866	P.L. 78.740
Driver	Driven	Velocity ratio	115 teeth	120 teeth	130 teeth	133 teeth	140 teeth	145 teeth	150 teeth	153 teeth	160 teeth	180 teeth	189 teeth	198 teeth	200 teeth	220 teeth	225 teeth	250 teeth
22	22	1.000	14.65	15.43	17.01	17.48	18.58	19.37	20.16	20.63	21.73	24.88	26.30	27.72	28.03	31.18	31.97	35.90
24	24	1.000	14.33	15.12	16.69	17.17	18.27	19.06	19.84	20.32	21.42	24.57	25.98	27.40	27.72	30.87	31.65	35.59
26	26	1.000	14.02	14.80	16.38	16.85	17.95	18.74	19.53	20.00	21.10	24.25	25.67	27.09	27.40	30.55	31.34	35.28
28	28	1.000	13.70	14.49	16.06	16.54	17.64	18.43	19.21	19.69	20.79	23.94	25.35	26.77	27.09	30.24	31.02	34.96
30	30	1.000	13.39	14.17	15.75	16.22	17.32	18.11	18.90	19.37	20.47	23.62	25.04	26.46	26.77	29.92	30.71	34.65
32	32	1.000	13.07	13.86	15.43	15.91	17.01	17.80	18.58	19.06	20.16	23.31	24.72	26.14	26.46	29.61	30.39	34.33
34	34	1.000	12.76	13.54	15.12	15.59	16.69	17.48	18.27	18.74	19.84	22.99	24.41	25.83	26.14	29.29	30.08	34.02
36	36	1.000	12.44	13.23	14.80	15.28	16.38	17.17	17.95	18.43	19.53	22.68	24.09	25.51	25.83	28.98	29.76	33.70
38	38	1.000	12.13	12.91	14.49	14.96	16.06	16.85	17.64	18.11	19.21	22.36	23.78	25.20	25.51	28.66	29.45	33.39
40	40	1.000	11.67	12.46	14.03	14.50	15.61	16.39	17.18	17.65	18.76	21.91	23.32	24.74	25.06	28.21	28.99	32.93
44	44	1.000	11.18	11.97	13.54	14.02	15.12	15.91	16.69	17.17	18.27	21.42	22.83	24.25	24.57	27.72	28.50	32.44
48	48	1.000	10.55	11.34	12.91	13.39	14.49	15.28	16.06	16.54	17.64	20.79	22.21	23.62	23.94	27.09	27.87	31.81
56	56	1.000	9.29	10.08	11.65	12.13	13.23	14.02	14.80	15.28	16.38	19.53	20.95	22.36	22.68	25.83	26.61	30.55
64	64	1.000	8.03	8.82	10.39	10.87	11.97	12.76	13.54	14.02	15.12	18.27	19.69	21.10	21.42	24.57	25.35	29.29
72	72	1.000	-	-	9.13	9.61	10.71	11.50	12.28	12.76	13.86	17.01	18.43	19.84	20.16	23.31	24.10	28.03
80	80	1.000	-	-	-	-	9.45	10.24	11.02	11.50	12.60	15.75	17.17	18.58	18.90	22.05	22.84	26.77
24	30	1.250	13.85	14.64	16.22	16.69	17.79	18.58	19.37	19.84	20.94	24.09	25.51	26.93	27.24	30.39	31.18	35.12
32	40	1.250	12.43	13.22	14.80	15.27	16.37	17.16	17.95	18.42	19.52	22.67	24.09	25.51	25.82	28.97	29.76	33.70
64	80	1.250	-	-	9.10	9.57	10.68	11.47	12.26	12.73	13.84	16.99	18.41	19.83	20.14	23.29	24.08	28.02
72	90	1.250	-	-	-	-	9.25	10.04	10.83	11.30	12.41	15.56	16.98	18.40	18.72	21.87	22.66	26.60
24	32	1.333	13.70	14.48	16.06	16.53	17.63	18.42	19.21	19.68	20.78	23.93	25.35	26.77	27.08	30.23	31.02	34.96
30	40	1.333	12.59	13.38	14.95	15.42	16.53	17.32	18.10	18.58	19.68	22.83	24.25	25.66	25.98	29.13	29.92	33.85
36	48	1.333	11.48	12.27	13.85	14.32	15.42	16.21	17.00	17.47	18.57	21.72	23.14	24.56	24.87	28.03	28.81	32.75
48	64	1.333	9.26	10.05	11.63	12.10	13.20	13.99	14.78	15.25	16.36	19.51	20.93	22.35	22.66	25.81	26.60	30.54
24	36	1.500	13.36	14.15	15.73	16.20	17.30	18.09	18.88	19.35	20.46	23.61	25.02	26.44	26.76	29.91	30.70	34.63
32	48	1.500	11.78	12.57	14.15	14.62	15.73	16.52	17.30	17.78	18.88	22.03	23.45	24.87	25.18	28.34	29.12	33.06
48	72	1.500	8.58	9.37	10.96	11.43	12.54	13.33	14.12	14.60	15.70	18.86	20.28	21.70	22.01	25.17	25.96	29.90
22	44	2.000	12.87	13.66	15.24	15.71	16.81	17.60	18.39	18.87	19.97	23.12	24.54	25.96	26.28	29.43	30.22	34.16
24	48	2.000	12.38	13.17	14.75	15.23	16.33	17.12	17.91	18.39	19.49	22.65	24.06	25.48	25.80	28.95	29.74	33.68
28	56	2.000	11.41	12.20	13.79	14.26	15.37	16.16	16.95	17.42	18.53	21.69	23.11	24.53	24.84	28.00	28.79	32.73
32	64	2.000	10.43	11.22	12.81	13.29	14.40	15.19	15.98	16.46	17.56	20.73	22.15	23.57	23.88	27.04	27.83	31.77
36	72	2.000	9.43	10.24	11.83	12.31	13.42	14.22	15.01	15.49	16.60	19.76	21.18	22.61	22.92	26.08	26.87	30.81
40	80	2.000	8.42	9.23	10.84	11.32	12.44	13.23	14.03	14.51	15.62	18.79	20.22	21.64	21.96	25.12	25.91	29.85
56	112	2.000	-	-	-	-	-	9.18	10.00	10.49	11.63	14.85	16.29	17.73	18.05	21.23	22.03	25.99
72	144	2.000	-	-	-	-	-	-	-	-	-	-	12.22	13.70	14.02	17.26	18.06	22.07
32	80	2.500	8.97	9.78	11.40	11.88	13.01	13.81	14.61	15.08	16.20	19.38	20.81	22.23	22.55	25.71	26.51	30.46
36	90	2.500	7.71	8.55	10.19	10.68	11.82	12.62	13.43	13.91	15.03	18.22	19.66	21.09	21.40	24.58	25.37	29.32
24	72	3.000	10.27	11.08	12.69	13.17	14.29	15.08	15.88	16.36	17.47	20.65	22.07	23.50	23.82	26.98	27.77	31.72
30	90	3.000	8.10	8.94	10.60	11.09	12.23	13.04	13.85	14.33	15.46	18.65	20.09	21.52	21.84	25.02	25.81	29.77
48	144	3.000	-	-	-	-	-	-	-	-	-	12.29	13.81	15.31	15.64	18.92	19.73	23.76

5. Eliminate sprockets that are not acceptable due to shaft requirements and space limitations

   (a) If the motor shaft size is given, you must ensure the driving sprocket's max bore size is bigger than the motor shaft (I think the bore is the hole in the middle of the sprocket) First you're going to want to find the brushing size for the candidate driving sprockets

TABLE 7–4 Sprockets with 8 mm Belt Pitch

Dim's all widths			20-mm Wide belt		30-mm Wide belt		50-mm Wide belt		85-mm Wide belt	
No. of teeth	Pitch dia.	Flange dia.	Sprocket number	Bushing size	Sprocket number	Bushing size	Sprocket number	Bushing size	Sprocket number	Bushing size
22	2.206	2.559	P22-8MGT-20	1108	P22-8MGT-30	1108	N/A	N/A	N/A	N/A
24	2.406	2.756	P24-8MGT-20	1108	P24-8MGT-30	1108	N/A	N/A	N/A	N/A
26	2.607	2.953	P26-8MGT-20	1108	P26-8MGT-30	1108	N/A	N/A	N/A	N/A
28	2.807	3.15	P28-8MGT-20	1108	P28-8MGT-30	1108	P28-8MGT-50	MPB	N/A	N/A
30	3.008	3.346	P30-8MGT-20	1210	P30-8MGT-30	1210	P30-8MGT-50	1210	N/A	N/A
32	3.208	3.543	P32-8MGT-20	1210	P32-8MGT-30	1210	P32-8MGT-50	1210	N/A	N/A
34	3.409	3.819	P34-8MGT-20	1610	P34-8MGT-30	1610	P34-8MGT-50	1610	P34-8MGT-85	1615
36	3.609	3.937	P36-8MGT-20	1610	P36-8MGT-30	1610	P36-8MGT-50	1610	P36-8MGT-85	1615
38	3.810	4.134	P38-8MGT-20	1610	P38-8MGT-30	1610	P38-8MGT-50	1610	P38-8MGT-85	1610
40	4.010	4.331	P40-8MGT-20	1610	P40-8MGT-30	2012	P40-8MGT-50	2012	P40-8MGT-85	2012
44	4.411	4.764	P44-8MGT-20	2012	P44-8MGT-30	2012	P44-8MGT-50	2012	P44-8MGT-85	2012
48	4.812	5.157	P48-8MGT-20	2012	P48-8MGT-30	2012	P48-8MGT-50	2012	P48-8MGT-85	2012
56	5.614	5.945	P56-8MGT-20	2012	P56-8MGT-30	2012	P56-8MGT-50	2517	P56-8MGT-85	2517
64	6.416	6.772	P64-8MGT-20	2012	P64-8MGT-30	2517	P64-8MGT-50	2517	P64-8MGT-85	2517
72	7.218	7.598	P72-8MGT-20	2012	P72-8MGT-30	2517	P72-8MGT-50	2517	P72-8MGT-85	3020
80	8.020	8.386	P80-8MGT-20	2517	P80-8MGT-30	2517	P80-8MGT-50	2517	P80-8MGT-85	3020
90	9.023	N/A	P90-8MGT-20	2517	P90-8MGT-30	2517	P90-8MGT-50	3020	P90-8MGT-85	3020
112	11.229	N/A	N/A	N/A	P112-8MGT-30	2517	P112-8MGT-50	3020	P112-8MGT-85	3020
144	14.437	N/A	N/A	N/A	P144-8MGT-30	2517	P144-8MGT-50	3020	P144-8MGT-85	3535
192	19.249	N/A	N/A	N/A	N/A	N/A	P192-8MGT-50	3020	P192-8MGT-85	3535

Then find the associated bore sizes from here:

TABLE 7–5 Taper-Lock Bushing

Bushing size	Min bore	Max bore
1008	0.500	0.875
1108	0.500	1.000
1210	0.500	1.250
1610	0.500	1.500
1615	0.500	1.500
2012	0.500	1.875
2517	0.500	2.250
3020	0.875	2.750
3525	1.188	3.250
3535	1.188	3.250
4030	1.438	3.625
4040	1.438	3.625
4535	1.938	4.250
4545	1.938	4.250
5040	2.438	4.500
6050	4.438	6.000
7060	4.938	7.000

Eliminate the sprocket combinations that have a driving sprocket max bore size that is smaller than the motor shaft diameter

(b) If a limit on the diameter of a sprocket is given, eliminate all candidates that exceed this limit.
Use table 7-4 above to find the flange diameters of the candidate sprockets.

(c) You should hopefully be left with one combination of driving/driven sprockets to use. Otherwise, just choose a random one that meets all requirements.

6. Find the pitch diameters for the selected sprockets using that same table as above (table 7-4).

7. If a range for the CD is given, use table 7-7 (the really long one posted above) to find a belt with the right sprocket sizes and a CD that falls within the right range

8. Find belt width and a new rated power from the following table:

TABLE 7–9 8M GT Style Belt Power Rating Table—30-mm Belt Width

RPM of faster shaft	Base rated horsepower for small sprocket (Number of grooves and pitch diameter, inches)															
	22 2.206	24 2.406	26 2.607	28 2.807	30 3.008	32 3.208	34 3.409	36 3.609	38 3.810	40 4.010	44 4.411	48 4.812	56 5.614	64 6.416	72 7.218	80 8.020
10	0.10	0.12	0.13	0.15	0.16	0.17	0.19	0.20	0.22	0.23	0.26	0.29	0.34	0.40	0.45	0.51
20	0.20	0.22	0.25	0.28	0.31	0.33	0.36	0.39	0.42	0.44	0.50	0.55	0.66	0.76	0.87	0.98
40	0.37	0.43	0.48	0.53	0.59	0.64	0.69	0.75	0.80	0.85	0.96	1.06	1.27	1.47	1.68	1.88
60	0.54	0.62	0.70	0.78	0.86	0.94	1.01	1.09	1.17	1.25	1.40	1.55	1.86	2.16	2.46	2.76
100	0.87	1.00	1.12	1.25	1.38	1.51	1.63	1.76	1.89	2.01	2.26	2.51	3.00	3.49	3.98	4.47
200	1.64	1.89	2.13	2.38	2.63	2.87	3.12	3.36	3.60	3.84	4.33	4.80	5.76	6.70	7.64	8.58
300	2.37	2.74	3.10	3.46	3.82	4.18	4.54	4.90	5.25	5.61	6.32	7.02	8.42	9.80	11.2	12.5
400	3.08	3.56	4.04	4.51	4.99	5.46	5.93	6.40	6.87	7.33	8.26	9.18	11.0	12.8	14.6	16.4
500	3.77	4.36	4.95	5.54	6.13	6.71	7.29	7.87	8.45	9.02	10.2	11.3	13.6	15.8	18.0	20.2
600	4.45	5.15	5.85	6.55	7.25	7.94	8.63	9.31	10.0	10.7	12.0	13.4	16.1	18.7	21.4	24.0
700	5.11	5.93	6.74	7.54	8.35	9.15	9.95	10.7	11.5	12.3	13.9	15.5	18.6	21.6	24.7	27.7
800	5.77	6.69	7.61	8.52	9.44	10.3	11.2	12.1	13.0	13.9	15.7	17.5	21.0	24.5	27.9	31.4
870	6.22	7.22	8.22	9.20	10.2	11.2	12.2	13.1	14.1	15.1	17.0	18.9	22.7	26.5	30.2	33.9
1000	7.05	8.19	9.33	10.5	11.6	12.7	13.8	14.9	16.0	17.1	19.3	21.5	25.8	30.1	34.3	38.5
1160	8.06	9.37	10.7	12.0	13.3	14.5	15.8	17.1	18.4	19.6	22.2	24.7	29.6	34.5	39.4	44.2
1200	8.31	9.66	11.0	12.3	13.7	15.0	16.3	17.6	19.0	20.3	22.9	25.4	30.6	35.6	40.6	45.6
1400	9.54	11.1	12.7	14.2	15.7	17.3	18.8	20.3	21.8	23.3	26.3	29.3	35.2	41.0	46.8	52.4
1600	10.7	12.5	14.3	16.0	17.8	19.5	21.2	23.0	24.7	26.4	29.8	33.1	39.8	46.3	52.8	59.1
1750	11.6	13.6	15.5	17.4	19.3	21.2	23.0	24.9	26.8	28.6	32.3	36.0	43.2	50.3	57.2	64.1
2000	13.1	15.3	17.5	19.6	21.8	23.9	26.0	28.1	30.2	32.3	36.5	40.6	48.7	56.7	64.5	72.1
2400	15.4	18.0	20.5	23.1	25.6	28.1	30.7	33.1	35.6	38.1	43.0	47.8	57.3	66.6	75.6	84.4
2800	17.6	20.6	23.6	26.5	29.4	32.3	35.2	38.0	40.9	43.7	49.3	54.8	65.6	76.1	86.2	96.0
3200	19.8	23.2	26.5	29.8	33.1	36.4	39.6	42.8	46.0	49.2	55.4	61.6	73.6	85.2	96.2	
3450	21.1	24.7	28.3	31.9	35.4	38.9	42.3	45.8	49.2	52.5	59.2	65.7	78.4	90.6	102.2	
4000	24.0	28.1	32.2	36.2	40.3	44.2	48.1	52.0	55.9	59.7	67.1	74.5	88.5			
4500	26.6	31.1	35.6	40.1	44.5	48.9	53.2	57.5	61.7	65.9	74.0	82.0				
5000	29.0	34.0	39.0	43.8	48.7	53.4	58.1	62.8	67.3	71.8	80.6	89.1				
5500	31.4	36.8	42.2	47.5	52.7	57.8	62.9	67.8	72.7	77.5	86.8					

TABLE 7–10 8M GT Style Belt Power Rating Table—50-mm Belt Width

RPM of faster shaft	Base rated horsepower for small sprocket (Number of grooves and pitch diameter, inches)												
	28	30	32	34	36	38	40	44	48	56	64	72	80
	2.807	3.008	3.208	3.409	3.609	3.810	4.010	4.411	4.812	5.614	6.416	7.218	8.020
10	0.25	0.28	0.30	0.33	0.35	0.38	0.40	0.45	0.50	0.59	0.69	0.78	0.88
20	0.49	0.53	0.58	0.63	0.68	0.72	0.77	0.86	0.96	1.14	1.33	1.51	1.70
40	0.93	1.02	1.11	1.21	1.30	1.39	1.48	1.66	1.84	2.20	2.56	2.92	3.27
60	1.35	1.49	1.63	1.76	1.90	2.03	2.17	2.43	2.70	3.23	3.75	4.28	4.80
100	2.18	2.40	2.62	2.84	3.06	3.28	3.50	3.93	4.36	5.22	6.08	6.92	7.77
200	4.14	4.57	4.99	5.42	5.84	6.26	6.68	7.52	8.35	10.0	11.7	13.3	14.9
300	6.02	6.65	7.27	7.90	8.52	9.14	9.75	11.0	12.2	14.6	17.0	19.4	21.8
400	7.85	8.67	9.49	10.3	11.1	11.9	12.7	14.4	16.0	19.2	22.3	25.5	28.6
500	9.63	10.7	11.7	12.7	13.7	14.7	15.7	17.7	19.7	23.6	27.5	31.4	35.2
600	11.4	12.6	13.8	15.0	16.2	17.4	18.6	20.9	23.3	28.0	32.6	37.2	41.7
700	13.1	14.5	15.9	17.3	18.7	20.1	21.4	24.2	26.9	32.3	37.6	42.9	48.2
800	14.8	16.4	18.0	19.6	21.1	22.7	24.2	27.3	30.4	36.5	42.6	48.6	54.5
870	16.0	17.7	19.4	21.1	22.8	24.5	26.2	29.5	32.9	39.5	46.0	52.5	58.9
1000	18.2	20.1	22.1	24.0	25.9	27.9	29.8	33.6	37.4	44.9	52.4	59.7	67.0
1160	20.8	23.1	25.3	27.5	29.7	32.0	34.1	38.5	42.9	51.5	60.0	68.5	76.8
1200	21.5	23.8	26.1	28.4	30.7	33.0	35.2	39.8	44.2	53.1	61.9	70.6	79.2
1400	24.7	27.4	30.0	32.7	35.3	38.0	40.6	45.8	51.0	61.2	71.3	81.3	91.2
1600	27.9	30.9	33.9	36.9	39.9	42.9	45.9	51.8	57.6	69.2	80.6	91.8	102.9
1750	30.2	33.5	36.8	40.1	43.3	46.6	49.8	56.2	62.5	75.0	87.4	99.5	111.4
2000	34.1	37.8	41.5	45.2	48.9	52.6	56.2	63.4	70.6	84.7	98.5	112.1	125.4
2400	40.2	44.6	48.9	53.3	57.6	62.0	66.2	74.7	83.1	99.7	115.8	131.5	146.8
2800	46.1	51.2	56.2	61.2	66.2	71.1	76.0	85.7	95.3	114.1	132.3	149.9	166.9
3200	51.9	57.6	63.2	68.9	74.5	80.0	85.5	96.4	107.1	128.0	148.1	167.4	
3450	55.4	61.5	67.6	73.6	79.6	85.5	91.3	102.9	114.3	136.4	157.5	177.7	
4000	63.0	70.0	76.9	83.7	90.4	97.1	103.7	116.8	129.5	154.0			
4500	69.7	77.4	85.0	92.6	100.0	107.3	114.5	128.7	142.5				
5000	76.2	84.7	92.9	101.1	109.1	117.1	124.9	140.1	154.9				
5500	82.5	91.6	100.5	109.3	117.9	126.4	134.7	150.9					

9. Find belt length correction factor (C_L).
   If you can't remember the pitch/length designation for your chosen belt, take a look back at table 7-7, they are listed there.

TABLE 7–11 8M GT Style Belt Length Correction Factor

Pitch/Length designation	No. of feeth	Correction factor	Pitch/Length designation	No. of teeth	Correction factor	Pitch/Length designation	No. of teeth	Correction factor	Pitch/Length designation	No. of teeth	Correction factor
384-8MGT	48	0.70	920-8MGT	115	1.00	1440-8MGT	180	1.10	2600-8MGT	325	1.20
480-8MGT	60	0.80	960-8MGT	120	1.00	1512-8MGT	189	1.10	2800-8MGT	350	1.20
560-8MGT	70	0.80	1040-8MGT	130	1.00	1584-8MGT	198	1.10	3048-8MGT	381	1.20
600-8MGT	75	0.80	1064-8MGT	133	1.00	1600-8MGT	200	1.10	3280-8MGT	410	1.20
640-8MGT	80	0.90	1120-8MGT	140	1.00	1760-8MGT	220	1.10	3600-8MGT	450	1.20
720-8MGT	90	0.90	1160-8MGT	145	1.00	1800-8MGT	225	1.20	4400-8MGT	550	1.20
800-8MGT	100	0.90	1200-8MGT	150	1.00	2000-8MGT	250	1.20			
840-8MGT	105	0.90	1224-8MGT	153	1.00	2200-8MGT	275	1.20			
880-8MGT	110	0.90	1280-8MGT	160	1.10	2400-8MGT	300	1.20			

10. Compute the adjusted rated power using P_{rated} found from the table

$$P_{adj} = P_{rated} \cdot C_L$$

It's fine if the value is very different than P_{des} found earlier.

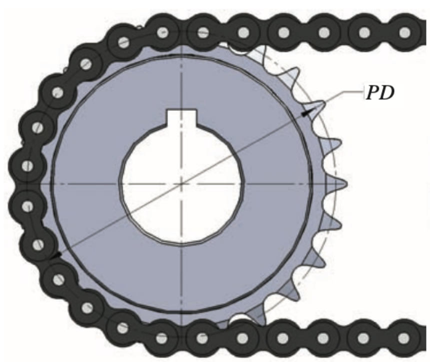
11. Calculate belt speed to ensure it does not exceed 6500 ft/min

$$v_{belt} = \frac{PD_1}{2} \cdot \omega_1 \cdot 2\pi \text{ rad/rev} \cdot \frac{1 \text{ ft}}{12 \text{ in}}$$

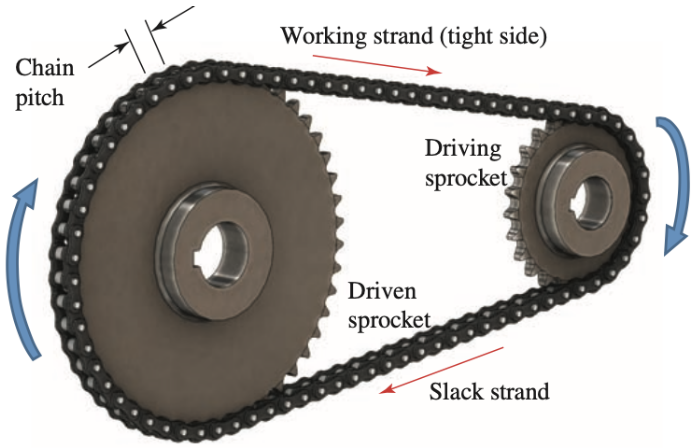
If you get an acceptable belt speed, congrats, you're done!

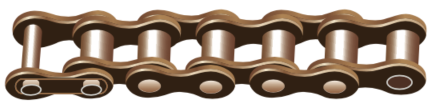
1.6 Chain Drives

1.6.1 Anatomy

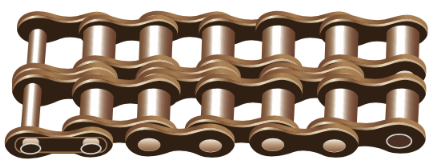


(a)





Standard roller chain, single strand



Standard roller chain, two-strand (also available with three and four strands)

1.6.2 Design Selection

Here are the US standard chains and their tensile strengths:

TABLE 7–12 U.S. Roller Chain Sizes

Chain number	ISO 10823	Pitch (in)	Average tensile strength (lb)
25	4A	1/4	925
35	6A	3/8	2100
41		1/2	2000
40	8A	1/2	3700
50	10A	5/8	6100
60	12A	3/4	8500
80	18A	1	14 500
100	20A	$1\frac{1}{4}$	24 000
120	24A	$1\frac{1}{2}$	34 000
140	28A	$1\frac{3}{4}$	46 000
160	32A	2	58 000
180	36A	$2\frac{1}{4}$	80 000
200	40A	$2\frac{1}{2}$	95 000
240	48A	3	130 000

Reference: ANS1 Standard B29.1.

If these chains are used to support a load or apply a tensile force, only 10% of the average tensile strength should be used.

1. Determine the service factor and compute the design power

   (a) Get the service factor from this table:

TABLE 7–17 Service Factors for Chain Drives

	Type of driver		
Load type	Hydraulic drive	Electric motor or turbine	Internal combustion engine with mechanical drive
Smooth Agitators; fans; generators; grinders; centrifugal pumps; rotary screens; light, uniformly loaded conveyors	1.0	1.0	1.2
Moderate shock Bucket elevators; machine tools; cranes; heavy conveyors; food mixers and grinders; ball mills; reciprocating pumps; woodworking machinery	1.2	1.3	1.4
Heavy shock Punch presses; hammer mills; boat propellers; crushers; reciprocating conveyors; rolling mills; logging hoists; dredges; printing presses	1.4	1.5	1.7

   (b) Calculate the design power using $P_{des} = SF \cdot P_{in}$

2. Compute the velocity ratio. If you're given an acceptable range for the output speed, use the middle of the range.

$$VR = \frac{n_1}{n_2} \text{ where } n_1 > n_2$$

3. Select the chain pitch (p) and number of teeth for the small sprocket (N_1) using n_1. This will also give you the rated power (P_{rated}). Refer to the following tables.
A few things to keep in mind:

   (a) You can use a multi-strand design (2, 3 or 4 strands) if you want to use a smaller drive but still transmit the same power at the same speed. To find the required power per stand, use the following power capacity factors:

   Two strands: Factor = 1.7

   Three strands: Factor = 2.5

   Four strands: Factor = 3.3

   The required power per chain is then P_{des}/factor

   (b) P_{rated} obtained from the tables must be greater than P_{des}

   (c) The minimum number of teeth in a sprocket should be 17 (unless the it is operation at $<$ 100rpm

   (d) The largest sprocket should have no more than 120 teeth, so make sure that $(N_1)(VR) < 120$ when selecting N_1

   (e) You will sadly need to use interpolation to find the rated power if n_1 isn't on the table already

   (f) The table will also give you the lubrication type

TABLE 7–14 Horsepower Ratings—Single-Strand Roller Chain No. 40

No. of teeth	0.500-in pitch										Rotational speed of small sprocket, rev/min													
	10	25	50	100	180	200	300	500	700	900	1000	1200	1400	1600	1800	2100	2500	3000	3500	4000	5000	6000	7000	8000	**9000**
11	0.06	0.14	0.27	0.52	0.91	1.00	1.48	2.42	3.34	4.25	4.70	5.60	6.49	5.57	4.66	3.70	2.85	2.17	1.72	1.41	1.01	0.77	0.61	0.50	0.00
12	0.06	0.15	0.29	0.56	0.99	1.09	1.61	2.64	3.64	4.64	5.13	6.11	7.09	6.34	5.31	4.22	3.25	2.47	1.96	1.60	1.15	0.87	0.69	0.57	0.00
13	0.07	0.16	0.31	0.61	1.07	1.19	1.75	2.86	3.95	5.02	5.56	6.62	7.68	7.15	5.99	4.76	3.66	2.79	2.21	1.81	1.29	0.98	0.78	0.00	
14	0.07	0.17	0.34	0.66	1.15	1.28	1.88	3.08	4.25	5.41	5.98	7.13	8.27	7.99	6.70	5.31	4.09	3.11	2.47	2.02	1.45	1.10	0.87	0.00	
15	0.08	0.19	0.36	0.70	1.24	1.37	2.02	3.30	4.55	5.80	6.41	7.64	8.86	8.86	7.43	5.89	4.54	3.45	2.74	2.24	1.60	1.22	0.97	0.00	
16	0.08	0.20	0.39	0.75	1.32	1.46	2.15	3.52	4.86	6.18	6.84	8.15	9.45	9.76	8.18	6.49	5.00	3.80	3.02	2.47	1.77	1.34	0.00		
17	0.09	0.21	0.41	0.80	1.40	1.55	2.29	3.74	5.16	6.57	7.27	8.66	10.04	10.69	8.96	7.11	5.48	4.17	3.31	2.71	1.94	1.47	0.00		
18	0.09	0.22	0.43	0.84	1.48	1.64	2.42	3.96	5.46	6.95	7.69	9.17	10.63	11.65	9.76	7.75	5.97	4.54	3.60	2.95	2.11	1.60	0.00		
19	0.10	0.24	0.46	0.89	1.57	1.73	2.56	4.18	5.77	7.34	8.12	9.66	11.22	12.64	10.59	8.40	6.47	4.92	3.91	3.20	2.29	0.09	0.00		
20	0.10	0.25	0.48	0.94	1.65	1.82	2.69	4.39	6.07	7.73	8.55	10.18	11.81	13.42	11.44	9.07	6.99	5.31	4.22	3.45	2.47	0.00			
21	0.11	0.26	0.51	0.98	1.73	1.91	2.83	4.61	6.37	8.11	8.98	10.69	12.40	14.10	12.30	9.76	7.52	5.72	4.54	3.71	2.65	0.00			
22	0.11	0.27	0.53	1.03	1.81	2.01	2.96	4.83	6.68	8.50	9.40	11.20	12.99	14.77	13.19	10.47	8.06	6.13	4.87	3.98	2.85	0.00			
23	0.12	0.28	0.56	1.08	1.90	2.10	3.10	5.05	6.98	8.89	9.83	11.71	13.58	15.44	14.10	11.19	8.62	6.55	5.20	4.26	3.05	0.00			
24	0.12	0.30	0.58	1.12	1.98	2.19	3.23	5.27	7.28	9.27	10.26	12.22	14.17	16.11	15.03	11.93	9.18	6.99	5.54	4.54	0.87	0.00			
25	0.13	0.31	0.60	1.17	2.06	2.28	3.36	5.49	7.59	9.66	10.69	12.73	14.76	16.78	15.98	12.68	9.76	7.43	5.89	4.82	0.00				
26	0.13	0.32	0.63	1.22	2.14	2.37	3.50	5.71	7.89	10.04	11.11	13.24	15.35	17.45	16.95	13.45	10.36	7.88	6.25	5.12	0.00				
28	0.14	0.35	0.67	1.31	2.31	2.55	3.77	6.15	8.50	10.82	11.97	14.26	16.53	18.79	18.94	15.03	11.57	8.80	6.99	5.72	0.00				
30	0.15	0.37	0.72	1.41	2.47	2.74	4.04	6.59	9.11	11.59	12.82	15.28	17.71	20.14	21.01	16.67	12.84	9.76	7.75	6.34	0.00				
32	0.16	0.40	0.77	1.50	2.64	2.92	4.31	7.03	9.71	12.38	13.66	16.30	18.89	21.48	23.14	18.37	14.14	10.76	8.54	1.41					
35	0.18	0.43	0.84	1.64	2.88	3.19	4.71	7.69	10.62	13.52	14.96	17.82	20.67	23.49	26.30	21.01	16.17	12.30	9.76	0.00					
40	0.21	0.50	0.96	1.87	3.30	3.65	5.38	8.79	12.14	15.45	17.10	20.37	23.62	26.85	30.06	25.67	19.76	15.03	0.00						
45	0.23	0.56	1.08	2.11	3.71	4.10	6.06	9.89	13.66	17.39	19.24	22.92	26.57	30.20	33.82	30.63	23.58	5.53	0.00						
		Type A							Type B										Type C						

Type A: Manual or drip lubrication
Type B: Bath or disc lubrication
Type C: Oil stream lubrication

TABLE 7–15 Horsepower Ratings—Single-Strand Roller Chain No. 60

No. of teeth	0.750-in pitch										Rotational speed of small sprocket, rev/min														
	10	25	50	100	120	200	300	400	500	600	800	1000	1200	1400	1600	1800	2000	2500	3000	3500	4000	4500	5000	5500	6000
11	0.19	0.46	0.89	1.72	2.05	3.35	4.95	6.52	8.08	9.63	12.69	15.58	11.85	9.41	7.70	6.45	5.51	3.94	3.00	2.38	1.95	1.63	1.39	1.21	0.00
12	0.21	0.50	0.97	1.88	2.24	3.66	5.40	7.12	8.82	10.51	13.85	17.15	13.51	10.72	8.77	7.35	6.28	4.49	3.42	2.71	2.22	1.86	1.59	1.38	0.00
13	0.22	0.54	1.05	2.04	2.43	3.96	5.85	7.71	9.55	11.38	15.00	18.58	15.23	12.08	9.89	8.29	7.08	5.06	3.85	3.06	2.50	2.10	1.79	0.00	
14	0.24	0.58	1.13	2.19	2.61	4.27	6.30	8.30	10.29	12.26	16.15	20.01	17.02	13.51	11.05	9.26	7.91	5.66	4.31	3.42	2.80	2.34	0.41	0.00	
15	0.26	0.62	1.21	2.35	2.80	4.57	6.75	8.90	11.02	13.13	17.31	21.44	18.87	14.98	12.26	10.27	8.77	6.28	4.77	3.79	3.10	2.60	0.00		
16	0.27	0.66	1.29	2.51	2.99	4.88	7.20	9.49	11.76	14.01	18.46	22.87	20.79	16.50	13.51	11.32	9.66	6.91	5.26	4.17	3.42	1.78	0.00		
17	0.29	0.70	1.37	2.66	3.17	5.18	7.65	10.08	12.49	14.88	19.62	24.30	22.77	18.07	14.79	12.40	10.58	7.57	5.76	4.57	3.74	0.00			
18	0.31	0.75	1.45	2.82	3.36	5.49	8.10	10.68	13.23	15.76	20.77	25.73	24.81	19.69	16.11	13.51	11.53	8.25	6.28	4.98	4.08	0.00			
19	0.33	0.79	1.53	2.98	3.55	5.79	8.55	11.27	13.96	16.63	21.92	27.16	26.91	21.35	17.48	14.65	12.50	8.95	6.81	5.40	0.20	0.00			
20	0.34	0.83	1.61	3.13	3.73	6.10	9.00	11.86	14.70	17.51	23.08	28.59	29.06	23.06	18.87	15.82	13.51	9.66	7.35	5.83	0.00				
21	0.36	0.87	1.69	3.29	3.92	6.40	9.45	12.46	15.43	18.38	24.23	30.02	31.26	24.81	20.31	17.02	14.53	10.40	7.91	6.28	0.00				
22	0.38	0.91	1.77	3.45	4.11	6.71	9.90	13.05	16.17	19.26	25.39	31.45	33.52	26.60	21.77	18.25	15.58	11.15	8.48	0.00					
23	0.40	0.95	1.85	3.61	4.29	7.01	10.35	13.64	16.90	20.13	26.54	32.88	35.84	28.44	23.28	19.51	16.66	11.92	9.07	0.00					
24	0.41	0.99	1.93	3.76	4.48	7.32	10.80	14.24	17.64	21.01	27.69	34.31	38.20	30.31	24.81	20.79	17.75	12.70	9.66	0.00					
25	0.43	1.04	2.01	3.92	4.67	7.62	11.25	14.83	18.37	21.89	28.85	35.74	40.61	32.23	26.38	22.11	18.87	13.51	10.27	0.00					
26	0.45	1.08	2.09	4.08	4.85	7.93	11.70	15.42	19.11	22.76	30.00	37.17	43.07	34.18	27.98	23.44	20.02	14.32	10.90	0.00					
28	0.48	1.16	2.26	4.39	5.23	8.54	12.60	16.61	20.58	24.51	32.31	40.03	47.68	38.20	31.26	26.20	22.37	16.01	0.00						
30	0.52	1.24	2.42	4.70	5.60	9.15	13.50	17.79	22.05	26.26	34.62	42.89	51.09	42.36	34.67	29.06	24.81	17.75	0.00						
32	0.55	1.33	2.58	5.02	5.98	9.76	14.40	18.98	23.52	28.01	36.92	45.75	54.50	46.67	38.20	32.01	27.33	19.56	0.00						
35	0.60	1.45	2.82	5.49	6.54	10.67	15.75	20.76	25.72	30.64	40.39	50.03	59.60	53.38	43.69	36.62	31.26	1.35	0.00						
40	0.69	1.66	3.22	6.27	7.47	12.20	18.00	23.73	29.39	35.02	46.16	57.18	68.12	65.22	53.38	44.74	38.20	0.00							
45	0.77	1.86	3.63	7.05	8.40	13.72	20.25	26.69	33.07	38.39	51.92	64.33	76.63	77.83	63.70	53.38	12.45	0.00							
		Type A						Type B										Type C							

Type A: Manual or drip lubrication
Type B: Bath or disc lubrication
Type C: Oil stream lubrication

TABLE 7–16 Horsepower Ratings—Single-Strand Roller Chain No. 80

No. of teeth	1.000-in pitch				Rotational speed of small sprocket, rev/min																				
	10	25	50	75	88	100	200	300	400	500	600	700	800	900	1000	1200	1400	1600	1800	2000	2500	3000	3500	4000	4500
11	0.44	1.06	2.07	3.05	3.56	4.03	7.83	11.56	15.23	18.87	22.48	26.07	27.41	22.97	19.61	14.92	11.84	9.69	8.12	6.83	4.96	3.77	3.00	2.45	0.00
12	0.48	1.16	2.26	3.33	3.88	4.39	8.54	12.61	16.82	20.59	24.53	28.44	31.23	26.17	22.35	17.00	13.49	11.04	9.25	7.90	5.65	4.30	3.41	2.79	0.00
13	0.52	1.26	2.45	3.61	4.21	4.76	9.26	13.66	18.00	22.31	26.57	30.81	35.02	29.51	25.20	19.17	15.21	12.45	10.43	8.91	6.37	4.85	3.85	3.15	
14	0.56	1.35	2.63	3.89	4.53	5.12	9.97	14.71	19.39	24.02	28.62	33.18	37.72	32.98	28.16	21.42	17.00	13.91	11.66	9.96	7.12	5.42	4.30	3.52	
15	0.60	1.45	2.82	4.16	4.86	5.49	10.68	15.76	20.77	25.74	30.66	35.55	40.41	36.58	31.23	23.76	18.85	15.43	12.93	11.04	7.90	6.01	4.77	0.00	
16	0.64	1.55	3.01	4.44	5.18	5.86	11.39	16.81	22.16	27.45	32.70	37.92	43.11	40.30	34.41	26.17	20.77	17.00	14.25	12.16	8.70	6.62	5.25	0.00	
17	0.68	1.64	3.20	4.72	5.50	6.22	12.10	17.86	23.54	29.17	34.75	40.29	45.80	44.13	37.68	28.66	22.75	18.62	15.60	13.32	9.53	7.25	0.00		
18	0.72	1.74	3.39	5.00	5.83	6.59	12.81	18.91	24.93	30.88	36.79	42.66	48.49	48.08	41.05	31.23	24.78	20.29	17.00	14.51	10.39	7.90	0.00		
19	0.76	1.84	3.57	5.28	6.15	6.95	13.53	19.96	26.31	32.60	38.84	45.03	51.19	52.15	44.52	33.87	26.88	22.00	18.44	15.74	11.26	0.36	0.00		
20	0.80	1.93	3.76	5.55	6.47	7.32	14.24	21.01	27.70	34.32	40.88	47.40	53.88	56.32	48.08	36.58	29.03	23.76	19.91	17.00	12.16	0.00			
21	0.84	2.03	3.95	5.83	6.80	7.69	14.95	22.07	29.08	36.03	42.92	49.77	56.58	60.59	51.73	39.36	31.23	25.56	21.42	18.29	13.09	0.00			
22	0.88	2.13	4.14	6.11	7.12	8.05	15.66	23.12	30.47	37.75	44.97	52.14	59.27	64.97	55.47	42.20	33.49	27.41	22.97	19.61	14.03				
23	0.92	2.22	4.33	6.39	7.45	8.42	16.37	24.17	31.85	39.46	47.01	54.51	61.97	69.38	59.30	45.11	35.80	29.30	24.55	20.97	15.00				
24	0.96	2.32	4.52	6.66	7.77	8.78	17.09	25.22	33.24	41.18	49.06	56.88	64.66	72.40	63.21	48.08	38.16	31.23	26.17	22.35	15.99				
25	1.00	2.42	4.70	6.94	8.09	9.15	17.80	26.27	34.62	42.89	51.10	59.25	67.35	75.42	67.20	51.12	40.57	33.20	27.83	23.76	8.16				
26	1.04	2.51	4.89	7.22	8.42	9.52	18.51	27.32	36.01	44.61	53.14	61.62	70.05	78.43	71.27	54.22	43.02	36.22	29.51	25.20	0.00				
28	1.12	2.71	5.27	7.77	9.06	10.25	19.93	29.42	38.78	48.04	57.23	66.36	75.44	84.47	79.65	60.59	48.08	39.36	32.98	28.16	0.00				
30	1.20	2.90	5.64	8.33	9.71	10.98	21.36	31.52	41.55	51.47	61.32	71.10	80.82	90.50	88.33	67.20	53.33	43.65	36.58	31.23					
32	1.28	3.09	6.02	8.89	10.36	11.71	22.78	33.62	44.32	54.91	65.41	75.84	86.21	96.53	97.31	74.03	58.75	48.08	40.30	5.65					
35	1.40	3.38	6.58	9.72	11.33	12.81	24.92	36.78	48.47	60.05	71.54	82.95	94.29	105.58	111.31	84.68	67.20	55.00	28.15	0.00					
40	1.61	3.87	7.53	11.11	12.95	14.64	28.48	42.03	55.40	68.63	81.76	94.80	107.77	120.67	133.51	103.46	82.10	40.16	0.00						
45	1.81	4.35	8.47	12.49	14.57	16.47	32.04	47.28	62.32	77.21	91.98	106.65	121.24	135.75	150.20	123.45	72.28	0.00							
	Type A				Type B								Type C												

Type A: Manual or drip lubrication
Type B: Bath or disc lubrication
Type C: Oil stream lubrication

4. Compute the number of teeth on the large sprocket N_2:

   $$N_2 = (N_1)(VR)$$

   Round to the nearest integer

5. Compute the actual output speed and make sure it's in the right range (if a range was given)

   $$n_2 = n_1(N_1/N_2)$$

6. Compute the pitch diameters of the sprockets

   $$PD_1 = \frac{p}{\sin{(180°/N_1)}}$$
   $$PD_2 = \frac{p}{\sin{(180°/N_2)}}$$

   Where p is the chain pitch selected in step 3

7. Specify the nominal CD. The recommended range is 30 to 50 pitches, so let's specify 40 pitches.

8. Compute the required chain length in pitches

   $$L_C = 2CD + \frac{N_2 + N_1}{2} + \frac{(N_2 - N_1)^2}{4\pi^2 CD}$$

   The chain length must be an integer multiple of the pitch, so round to the nearest integer value

9. Compute the actual CD

$$CD = \frac{1}{4}[L_C - \frac{N_2 + N_1}{2} + \sqrt{(L_C - \frac{N_2 + N_1}{2})^2 - \frac{8(N_2 - N_1)^2}{4\pi^2}}]$$

10. Compute the angle of wrap for each sprocket. The minimum angle of wrap should be 120°

$$\theta_1 = 180^\circ - 2\sin^{-1}[\frac{PD_2 - PD_1}{2CD}]$$
$$\theta_2 = 180^\circ + 2\sin^{-1}[\frac{PD_2 - PD_1}{2CD}]$$

11. Compute factor of safety

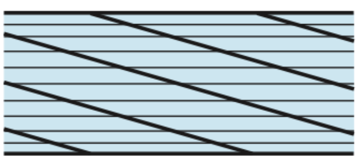
$$FS = P_{allowed}/P_{des}$$

$P_{allowed}$ is the number you got from the table in step 3 times the strand factor

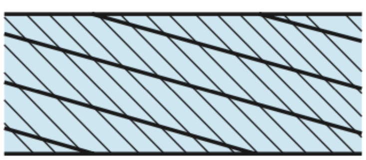
1.7 Wire Rope

1.7.1 Anatomy

There's two types of rope winding:



(*a*) Regular lay



(*b*) Lang lay

- Regular-lay ropes have the wires in the strand twisted in one direction and the strands in the rope twisted in the opposite direction
- Lang-lay ropes have the wires in the strand and the strands in the rope twisted in the same direction. Lang lay is more flexible than regular lay.

1.7.2 Nomenclature

F = tensile force on rope (lbf)
W = weight at the end of the rope (load) (lbf)
m = number of ropes supporting load
w = weight/foot supporting load (lbf/ft)

l = maximum suspended length of rope (ft)
a = maximum acceleration/deceleration (ft/s^2)
g = acceleration of gravity (32.17 ft/s^2)
p/S_u = specified life
S_u = ultimate tensile strength (psi)
D = sheave or which drum diameter (in)
d = nominal wire rope size (in)
E_r = Young's modulus (psi)
d_w = diameter of the wire (in)
A_m = metal cross-sectional area (in^2)

1.7.3 Formulae

$$\text{rope tension: } F_t = \left(\frac{W}{m} + wl\right)\left(1 + \frac{a}{g}\right)$$

$$\text{ultimate strength of wire: } S_u = \frac{2000F}{Dd}$$

$$\text{fatigue tension: } F_f = \frac{(p/S_u)S_u Dd}{2}$$

$$\text{equivalent bending load: } F_b = \frac{E_r d_w A_m}{D}$$

$$\text{fatigue factor of safety: } n_f = \frac{F_f - F_b}{F_t}$$

$$\text{factor of safety for static loading: } n_s = \frac{F_u - F_b}{F_t}$$

$$\text{bearing pressure: } P = \frac{2F}{dD}$$

1.7.4 Useful Tables

Table 17–24 **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 Plow steel Mild plow steel	$d/9$ $d/9$ $d/9$	14 14 14	100 88 76
6 × 19 standard hoisting	$1.60d^2$	$26d$–$34d$	$\frac{1}{4}$–$2\frac{3}{4}$	Monitor steel Plow steel Mild plow steel	$d/13$–$d/16$ $d/13$–$d/16$ $d/13$–$d/16$	12 12 12	106 93 80
6 × 37 special flexible	$1.55d^2$	$18d$	$\frac{1}{4}$–$3\frac{1}{2}$	Monitor steel Plow steel	$d/22$ $d/22$	11 11	100 88
8 × 19 extra flexible	$1.45d^2$	$21d$–$26d$	$\frac{1}{4}$–$1\frac{1}{2}$	Monitor steel Plow steel	$d/15$–$d/19$ $d/15$–$d/19$	10 10	92 80
7 × 7 aircraft	$1.70d^2$	—	$\frac{1}{16}$–$\frac{3}{8}$	Corrosion-resistant steel Carbon steel	— —	— —	124 124
7 × 9 aircraft	$1.75d^2$	—	$\frac{1}{8}$–$1\frac{3}{8}$	Corrosion-resistant steel Carbon steel	— —	— —	135 143
19-wire aircraft	$2.15d^2$	—	$\frac{1}{32}$–$\frac{5}{16}$	Corrosion-resistant steel Carbon steel	— —	— —	165 165

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

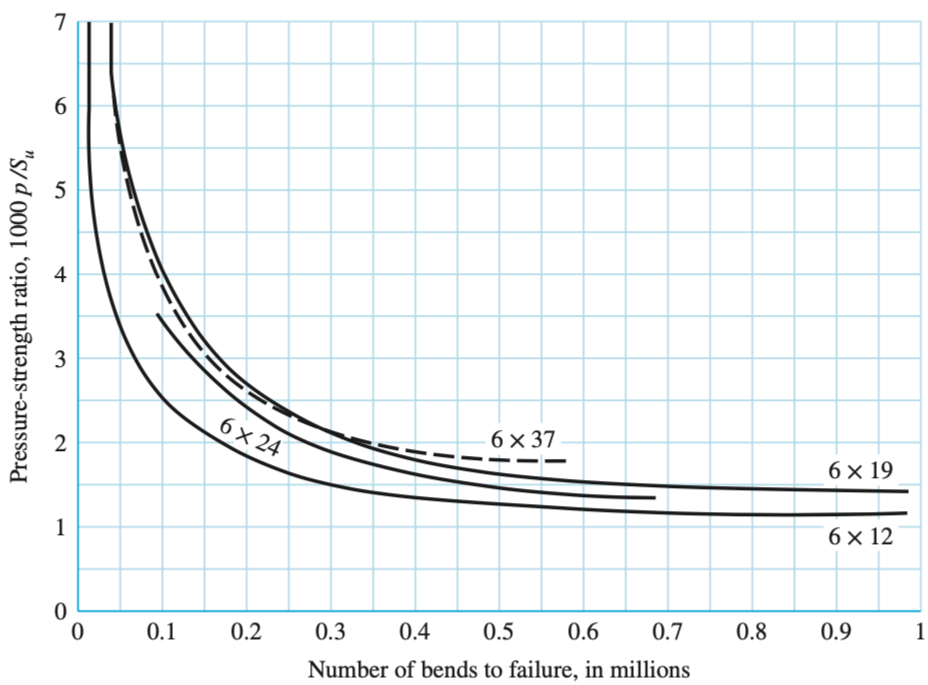
Table 17–26 **Maximum Allowable Bearing Pressures of Ropes on Sheaves (in psi)**

Rope	Sheave Material				
	Wood[a]	Cast Iron[b]	Cast Steel[c]	Chilled Cast Irons[d]	Manganese Steel[e]
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

S_u ranges:

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

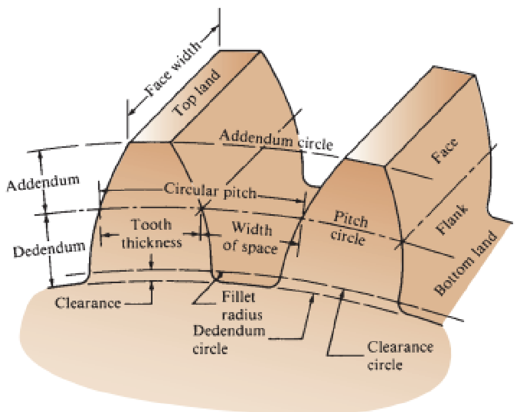
No idea how to use this but here it is in case it's relevant:

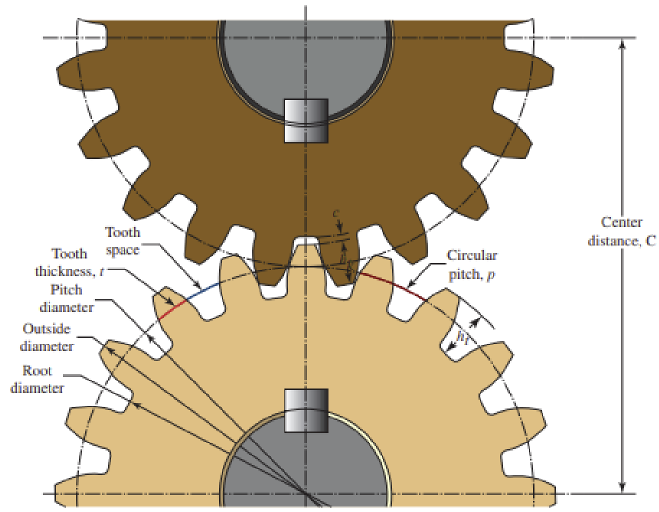


2 Gears and Shit

2.1 Spur Gears

2.1.1 Anatomy





2.1.2 Nomenclature

"pinion" is the smaller gear
"gear" is the larger gear
N_P = number of teeth on pinion (input)
N_G = number of teeth on gear (output)
p = circular pitch (in)
P_d = diametral pitch (teeth/in)
m = module (in/teeth)
D_P = pitch diameter of pinion (in)
D_G = pitch diameter of gear (in)
D_o = outside diameter (in)
D_R = root diameter (in)
D_b = base circle diameter (in)
$R = \frac{D}{2}$ = radius (in)
a = addendum (in)
b = dedendum (in)
c = clearance (in)
h_f = whole depth (in)
h_k = working depth (in)
t = tooth thickness (in)
F = face width (in)
ϕ = pressure angle
C = center distance (in)
m_f = contact ratio

m_G = gear ratio
n_P = pinion speed (input) (rpm)
n_G = gear speed (output) (rpm)
W_t = transmitted load (lbf)
W_r = radial force (lbf)
W_n = normal force (lbf)
$H = P$ = transmitted power (hp)
$V = v_t$ = pitch-line velocity (ft/min)
$F^t_{x,y}$ = transmitted force between gears x and y (lbf)
$F^r_{x,y}$ = radial force between gears x and y (lbf)
T = torque (lbf $\cdot$ in)
K_O = Overload Factor
P_{des} = design power (hp)
VR = velocity ratio
C_P = elastic coefficient
A_v = quality number
K_v = dynamic factor
J_P = bending geometry factor of the pinion
J_G = bending geometry factor of the gear
I = pitting geometry factor
C_{pf} = pinion proportion factor
C_{ma} = mesh alignment factor
K_m = load-distribution factor
K_s = size factor
K_B = rim thickness factor
FS = service factor
K_R = reliability factor
N_{cP} = number of loading cycles for pinion
N_{cG} = number of loading cycles for gear
Y_{NP} = bending stress cycle factor for pinion
Y_{NG} = bending stress cycle factor for gear
Z_{NP} = pitting resistance stress cycle factor for pinion
Z_{NG} = pitting resistance stress cycle factor for gear
s_{tP} = expected bending stress in pinion (psi)
s_{tG} = expected bending stress in gear (psi)
s_{atP} = adjusted expected bending stress in pinion (psi)
s_{atG} = adjusted expected bending stress in gear (psi)
s_c = expected contact stress
s_{cP} = adjusted expected contact stress for pinion
s_{cG} = adjusted expected contact stress for gear

2.1.3 Formulae

Geometry:

$$\text{circular pitch: } p = \frac{\pi D_P}{N_P} = \frac{\pi D_G}{N_G} = \frac{\pi}{P_d}$$

diametral pitch: $P_d = \frac{N_P}{D_P} = \frac{N_G}{D_G} = \frac{\pi}{p}$

module: $m = \frac{D_P}{N_P} = \frac{D_G}{N_G} = \frac{1}{P_d}$

gear ratio: $m_G = \frac{N_G}{N_P}$

outside diameter: $D_o = \frac{N+2}{P_d}$

root diameter: $D_R = D - 2b$

addendum: $a = \frac{1}{P_d}$

dedendum: $b = \begin{cases} \frac{1.25}{P_d} & P_d < 20 \\ \frac{1.20}{P_d} + 0.002 & P_d \geq 20 \end{cases}$

clearance: $c = \begin{cases} \frac{0.25}{P_d} & P_d < 20 \\ \frac{0.2}{P_d} + 0.002 & P_d \geq 20 \end{cases}$

whole depth: $h_f = a + b$

working depth: $h_k = 2a$

tooth thickness: $t = \frac{p}{2} = \frac{\pi}{2P_d}$

nominal face width: $F = \frac{12}{P_d}$

$$\frac{8}{P_d} < F < \frac{16}{P_d}$$

center distance: $C = \frac{D_P + D_G}{2} = \frac{N_P + N_G}{2P_d}$

base circle diameter: $D_b = \frac{N_p}{P_d}\cos\phi$

contact ratio: $m_f = \frac{\sqrt{R_{oP}^2 - R_{bP}^2} + \sqrt{R_{oG}^2 - R_{bG}^2} - C\sin\phi}{p\cos\phi}$

$$F^t_{\text{driving},x} = W_t$$

$$F^r_{x,y} = F^t_{x,y}\tan\phi$$

$$T = \frac{W_t d}{2}$$

Forces and motion:

pitch line speed(ft/min): $v_t = \frac{\pi D n}{12}$

velocity ratio: $VR = \frac{n_P}{n_G} = \frac{N_G}{N_P}$

torque: $T = \frac{63000P}{n} = \frac{W_t D}{2}$

tangential force: $W_t = \frac{33000P}{v_t} = \frac{126000P}{nD}$

$$\text{radial force: } W_r = W_t \tan\phi$$

$$\text{normal force: } W_n = \frac{W_t}{\cos\phi} = \sqrt{W_t^2 + W_r^2}$$

$$\text{bending stress number: } s_t = \frac{W_t P_d}{FJ} K_O K_s K_m K_B K_v$$

$$\text{contact stress number: } s_c = C_p \sqrt{\frac{W_t K_O K_s K_m K_v}{F D_p I}}$$

$$\text{allowable bending stress: } s_{at} > s_t \frac{(SF) K_R}{Y_N}$$

$$\text{allowable contact stress: } s_{ac} > s_c \frac{(SF) K_R}{Z_N}$$

2.1.4 Design Selection

In these 39 simple steps, you too can become a ~~masochist~~ Mechanical Engineer!

1. Find the type of shock for input and output from this random place in the textbook:

   Uniform: Electric motor or constant-speed gas turbine

   Light shock: Water turbine, variable-speed drive

   Moderate shock: Multicylinder engine

   Examples of the roughness of driven machines include the following:

   Uniform: Continuous-duty generator, paper, and film winders.

   Light shock: Fans and low-speed centrifugal pumps, liquid agitators, variable-duty generators, uniformly loaded conveyors, rotary positive displacement pumps, and metal strip processing.

   Moderate shock: High-speed centrifugal pumps, reciprocating pumps and compressors, heavy-duty conveyors, machine tool drives, concrete mixers, textile machinery, meat grinders, saws, bucket elevators, freight elevators, escalators, concrete mixers, plastics molding and processing, sewage disposal equipment, winches, and cable reels.

   Heavy shock: Rock crushers, punch press drives, pulverizers, processing mills, tumbling barrels, wood chippers, vibrating screens, railroad car dumpers, log conveyors, lumber handling equipment, metal shears, hammer mills, commercial washers, heavy-duty hoists and cranes, reciprocating feeders, dredges, rubber processing, compactors, and plastics extruders.

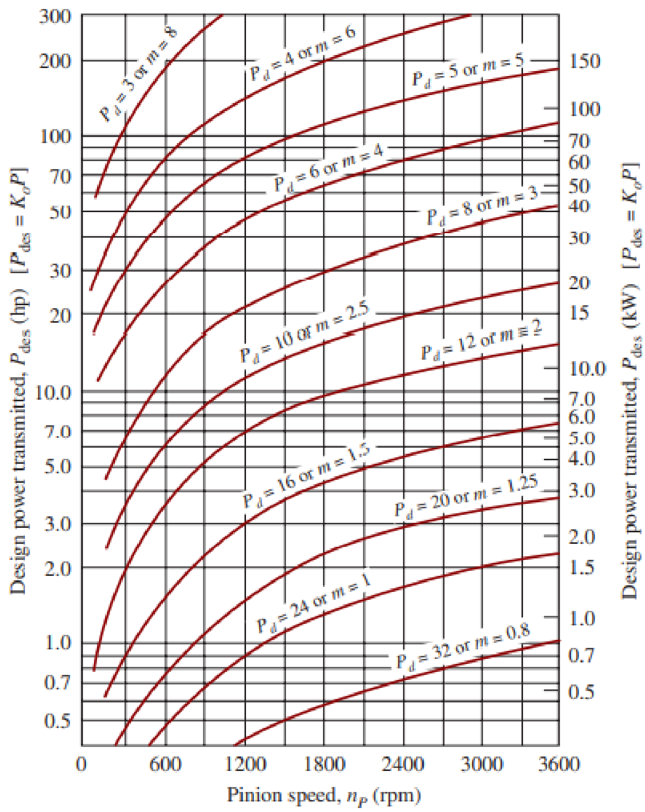
2. Use this fucking thing to find the shock

TABLE 9–1 Suggested Overload Factors, K_o

	Driven Machine			
Power source	Uniform	Light shock	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.50	1.75
Light shock	1.20	1.40	1.75	2.25
Moderate shock	1.30	1.70	2.00	2.75

3. Find $P_{\text{des}} = PK_O$

4. Find P_d:



For all curves: 20° full depth teeth;
$N_P = 24$; $N_G = 96$; $m_G = 4.00$; $F = 12/P_d$; $A_v = 11$
Steel gears, HB 300; $s_{at} = 36\,000$ psi (250 MPa); $s_{ac} = 126\,000$ psi (869 MPa)

(round to smallest number)

5. Choose N_p to be some random fucking value between 17 and 20.

6. declare what you think n_G should be based on the range in the problem statement. (assume the value is the middle of the acceptable range)

7. Get the velocity ratio

$$VR = \frac{n_P}{n_G}$$

8. Compute the number of teeth on the output gear $N_G = N_P(VR)$ (round to nearest int)

9. Compute the actual velocity ratio

$$VR = \frac{N_G}{N_P}$$

10. Compute the actual output speed

$$n_G = n_P \left(\frac{N_P}{N_G}\right)$$

Check that it's within the specified range, if not try new Np

11. Compute the diameters of the gears

$$D_P = \frac{N_P}{P_d}$$
$$D_G = \frac{N_G}{P_d}$$

12. Compute the center distance, pitch line speed, and transmitted load because why the hell not

$$C = \frac{N_P + N_G}{2P_d}$$
$$v_t = \frac{\pi}{12} D_P n_P$$
$$W_t = \frac{33000P}{v_t}$$

13. Find the face width, F. Just use the nominal one and don't question where the numbers come from.

$$\text{nominal value} = \frac{12}{P_d}$$
$$\text{lower limit} = \frac{8}{P_d} \qquad 0.5 < \frac{F}{D_p} < 2, \text{ if you are outside this range try a different value}$$
$$\text{upper limit} = \frac{16}{P_d}$$

14. Choose a material. I have no clue how to do this so just always choose steel and hope it works.
For Material and stress look at table 9-13 on page 401 in Motts

15. Find C_P based on the material. It will be 2300 for steel

TABLE 9–7 Elastic Coefficient, C_p

Pinion material	Modulus of elasticity, E_p, lb/in^2 (MPa)	Gear material and modulus of elasticity, E_G, lb/in^2 (MPa): Steel 30×10^6 (2×10^5)	Malleable iron 25×10^6 (1.7×10^5)	Nodular iron 24×10^6 (1.7×10^5)	Cast iron 22×10^6 (1.5×10^5)	Aluminum bronze 17.5×10^6 (1.2×10^5)	Tin bronze 16×10^6 (1.1×10^5)
Steel	30×10^6	2300	2180	2160	2100	1950	1900
	(2×10^5)	(191)	(181)	(179)	(174)	(162)	(158)
Mall. iron	25×10^6	2180	2090	2070	2020	1900	1850
	(1.7×10^5)	(181)	(174)	(172)	(168)	(158)	(154)
Nod. iron	24×10^6	2160	2070	2050	2000	1880	1830
	(1.7×10^5)	(179)	(172)	(170)	(166)	(156)	(152)
Cast iron	22×10^6	2100	2020	2000	1960	1850	1800
	(1.5×10^5)	(174)	(168)	(166)	(163)	(154)	(149)
Al. bronze	1.75×10^6	1950	1900	1880	1850	1750	1700
	(1.2×10^5)	(162)	(158)	(156)	(154)	(145)	(141)
Tin bronze	16×10^6	1900	1850	1830	1800	1700	1650
	(1.1×10^5)	(158)	(154)	(152)	(149)	(141)	(137)

Source: Extracted from AGMA Standard 2001-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1001 North Fairfax Street, 5th floor, Alexandria, VA 22314.

Note: Poisson's ratio = 0.30; units for C_p are $(\text{lb/in}^2)^{0.5}$ or $(\text{MPa})^{0.5}$.

16. Find the quality number A_v from the application or the pitch line speed. This table is shit so just guess what looks right.

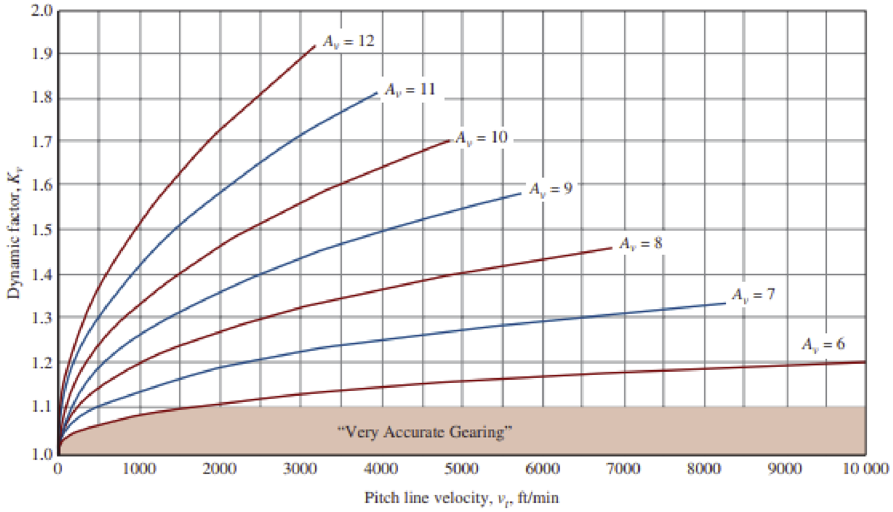
TABLE 9–5 Recommended AGMA Quality Numbers

Application	Quality number	Application	Quality number
Cement mixer drum drive	A11	Small power drill	A9
Cement kiln	A11	Clothes washing machine	A8
Steel mill drives	A11	Printing press	A7
Grain harvester	A10	Computing mechanism	A6
Cranes	A10	Automotive transmission	A6
Punch press	A10	Radar antenna drive	A5
Mining conveyor	A10	Marine propulsion drive	A5
Paper-box-making machine	A9	Aircraft engine drive	A4
Gas meter mechanism	A9	Gyroscope	A2

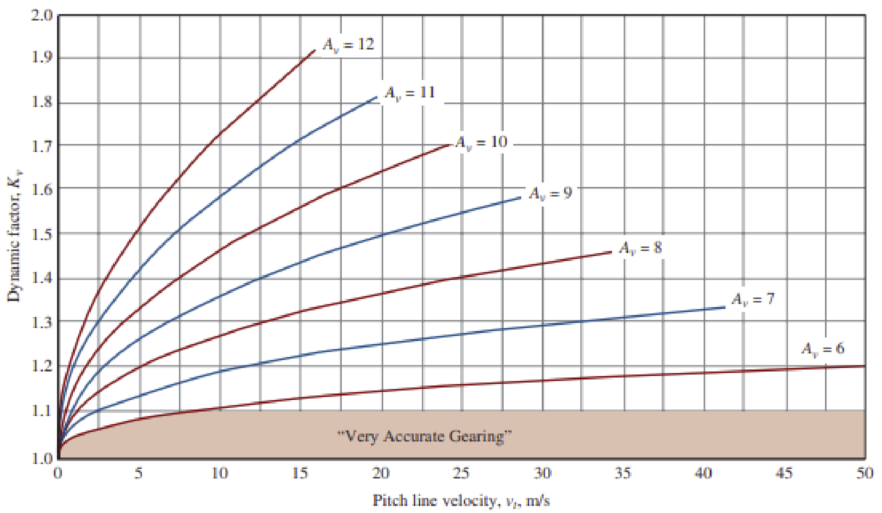
Machine tool drives and drives for other high-quality mechanical systems

Pitch line speed (fpm)	Quality number	Pitch line speed (m/s)
0–800	A10	0–4
800–2000	A8	4–11
2000–4000	A6	11–22
Over 4000	A4	Over 22

17. Find the dynamic factor K_v from this graph using A_v and v_t

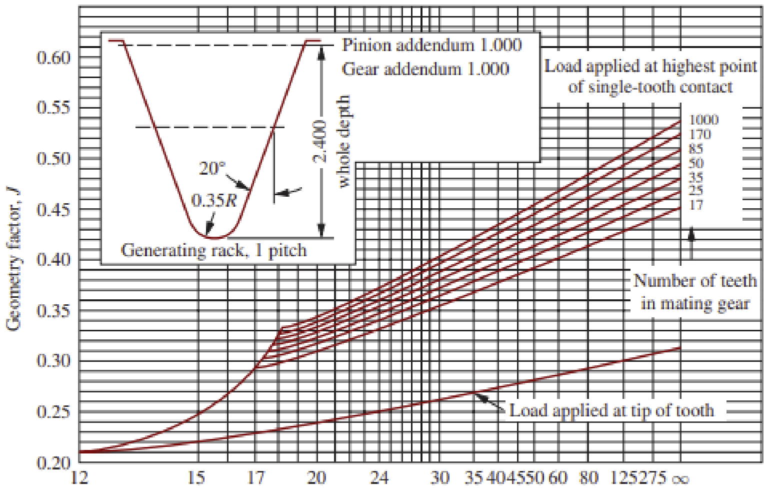


(*a*) U.S. Customary units

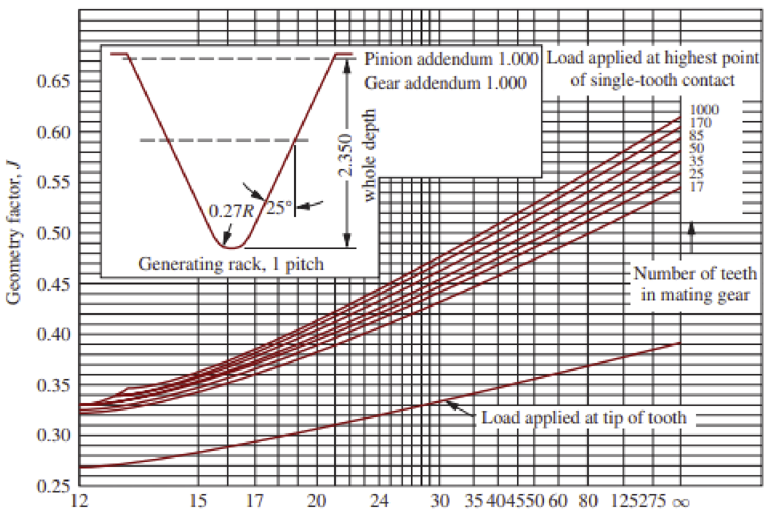


(*b*) SI Metric units

18. Choose the J_P and J_G values. Assume 20° unless otherwise specified. Why? Because fuck you, that's why.

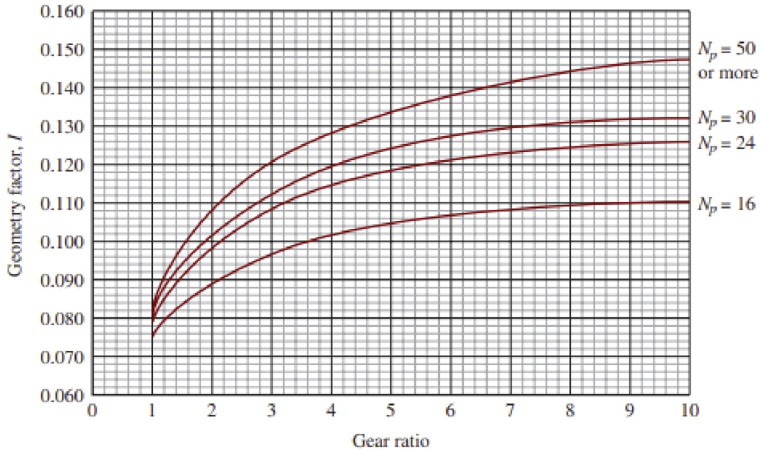


Number of teeth for which geometry factor is desired
(*a*) 20° spur gear: standard addendum

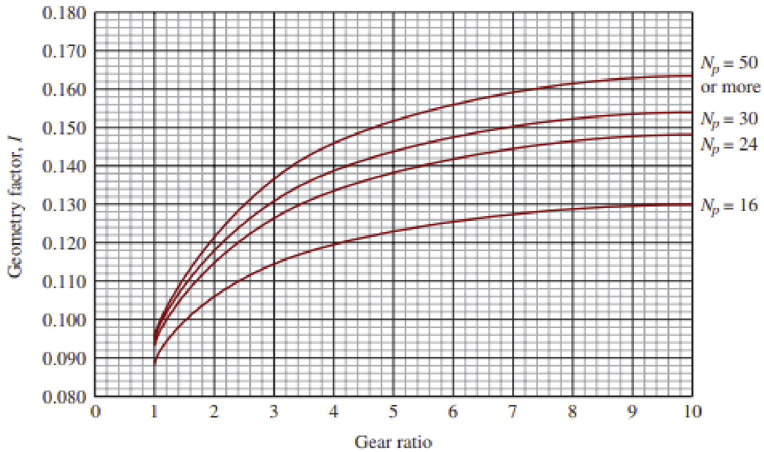


Number of teeth for which geometry factor is desired
(*b*) 25° spur gear: standard addendum

19. Choose the pitting geometry factor, I. Use the same pressure angle.

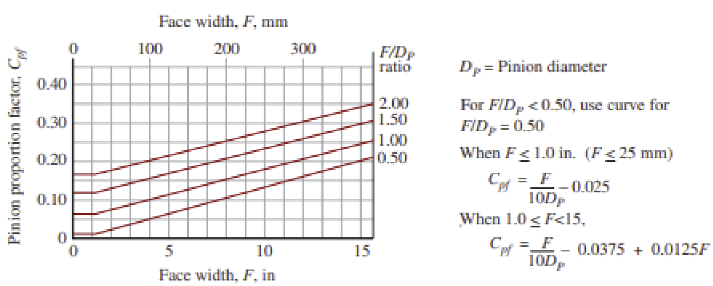


(*a*) 20° pressure angle, full-depth teeth (standard addendum = $1/P_d$)

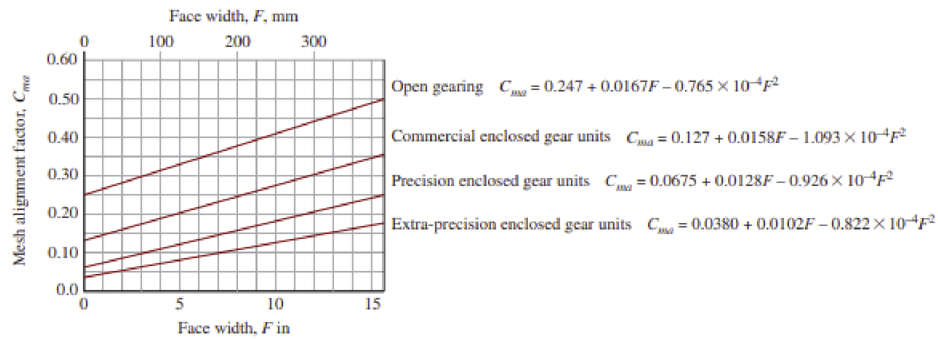


(*b*) 25° pressure angle, full-depth teeth (standard addendum = $1/P_d$)

20. Find C_{pf} from this. Use the equations if you can because the table is bad.



21. Find C_{ma} from this. Probably use commercial enclosed gear units but do whatever you feel like.

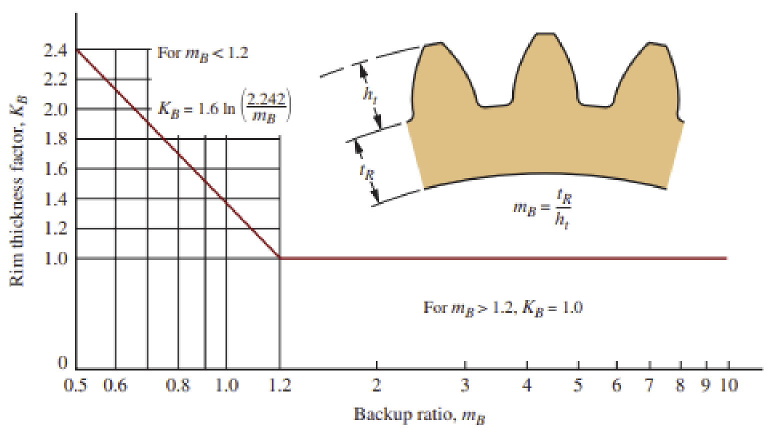


22. Compute $K_m = 1 + C_{pf} + C_{ma}$
23. Find K_s from this

TABLE 9–2 Suggested Size Factors, K_s

Diametral pitch, P_d	Metric module, m	Size factor, K_s
≥ 5	≤ 5	1.00
4	6	1.05
3	8	1.15
2	12	1.25
1.25	20	1.40

24. Find K_B. It will always be 1.00 unless it isn't. If it isn't then use this ugly picture



25. Specify a service factor, SF between 1.00 and 1.50. Usually pick 1.00 but if your data is uncertain then ramp that shit up.
26. Gander a guess at how reliable your system will be. Let's assume for most cases that you're not that shit of an Engineer and it works 99% of the time.
27. Use your rigorously calculated reliability to get K_R from yet another fucking table

TABLE 9–11 Reliability Factor, K_R	
Reliability	K_R
0.90, one failure in 10	0.85
0.99, one failure in 100	1.00
0.999, one failure in 1000	1.25
0.9999, one failure in 10 000	1.50

28. Guess what the lifetime of your machine will be. Don't worry, there's a shitty table to help you.

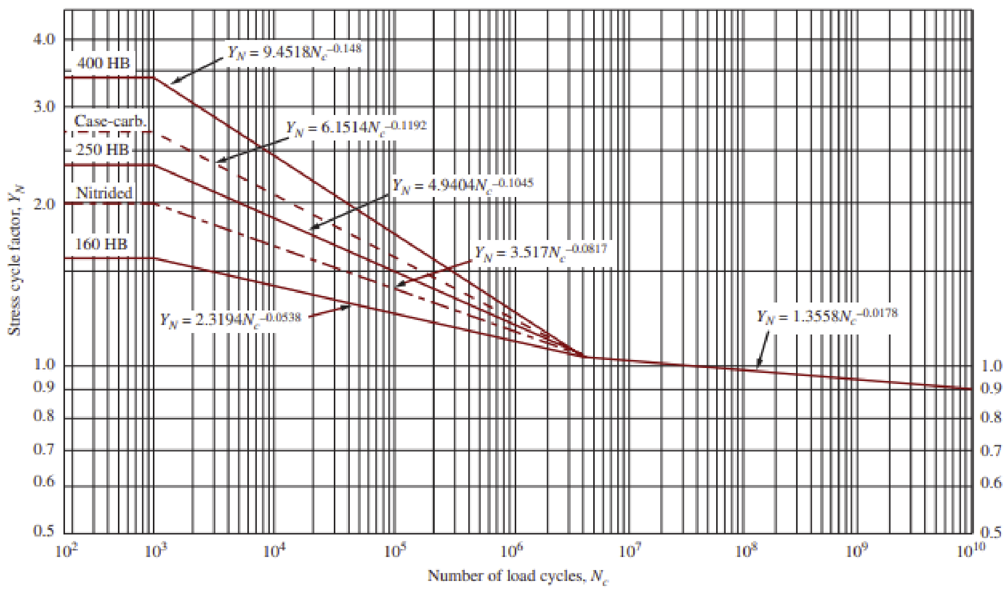
TABLE 9–12 Recommended Design Life	
Application	**Design life (h)**
Domestic appliances	1000–2000
Aircraft engines	1000–4000
Automotive	1500–5000
Agricultural equipment	3000–6000
Elevators, industrial fans, multipurpose gearing	8000–15 000
Electric motors, industrial blowers, general industrial machines	20 000–30 000
Pumps and compressors	40 000–60 000
Critical equipment in continuous 24-h operation	100 000–200 000

Source: Eugene A. Avallone and Theodore Baumeister III, eds. *Marks' Standard Handbook for Mechanical Engineers.* 9th ed. New York: McGraw-Hill, 1986.

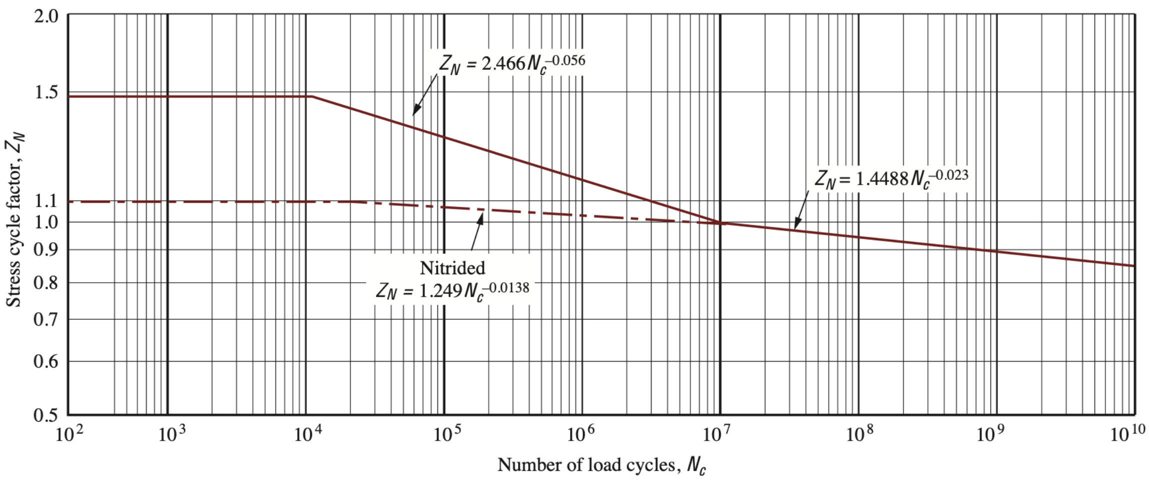
29. Find the number of loading cycles using these formulas

$$N_{cP} = (60)(\text{lifetime})n_P$$
$$N_{cG} = (60)(\text{lifetime})n_G$$

30. Use this to get Y_{NP} and Y_{NG}



31. Use this to get Z_{NP} and Z_{NG}



32. God assembled all the known constants in the universe and compiled them into this fucking formula. Now use it to get s_{tP} and s_{tG}.

$$s_{tP} = \frac{W_t P_d}{F J_P} K_O K_s K_m K_B K_v$$
$$s_{tG} = s_{tP} \frac{J_P}{J_G}$$

33. Now take that shit and do this shit

$$s_{atP} > s_{tP} \frac{K_R(SF)}{Y_{NP}}$$
$$s_{atG} > s_{tG} \frac{K_R(SF)}{Y_{NG}}$$

34. Thought you were done. Haha nope, you have to calculate this shit

$$s_c = C_P\sqrt{\frac{W_t K_O K_s K_m K_v}{F D_p I}}$$

This is the expected contact stress and will be the same for the gear and pinion

35. Now find the adjusted values of s_C

$$s_{acP} > s_c \frac{K_R(SF)}{Z_{NP}}$$
$$s_{acG} > s_c \frac{K_R(SF)}{Z_{NG}}$$

36. Compute the safety factors for the gears. Or don't, I don't care about safety.
For bending stress:

$$SF_P = \frac{s_{atP} Y_{NP}}{s_{tP} K_R} \qquad SF_G = \frac{s_{atG} Y_{NG}}{s_{tG} K_R}$$

For contact stress:

$$SF_P = \frac{s_{acP} Y_{NP}}{s_{cP} K_R} \qquad SF_G = \frac{s_{acG} Y_{NG}}{s_{cG} K_R}$$

Verify the values satisfy $1.0 < SF < 1.5$ or fudge the number so that it works.

37. The required HB for grade 1 and 2 steels is as follows. Stresses in **psi**. (As far as we're concerned, grade 1 steel is the only one that exists.) Note that first two equations are for contact stress and the last two are for bending stress. Choose the largest one for a selected grade.

$$\text{Contact: Required HB grade 1} = \frac{\frac{s_{ac}}{1000} - 29.10}{0.322}$$
$$\text{Contact: Required HB grade 2} = \frac{\frac{s_{ac}}{1000} - 34.30}{0.349}$$
$$\text{Bending: Required HB grade 1} = \frac{\frac{s_{at}}{1000} - 12.8}{0.0773}$$
$$\text{Bending: Required HB grade 2} = \frac{\frac{s_{at}}{1000} - 16.40}{0.102}$$

38. Use any of the tables below to find a material that satisfies the required HB for the gear and pinion. We should use the same material for both the gear and pinion.

APPENDIX 3 Design Properties of Carbon and Alloy Steels

Material designation (SAE number)	Condition	Tensile strength (ksi)	Tensile strength (MPa)	Yield strength (ksi)	Yield strength (MPa)	Ductility (percent elongation in 2 in)	Brinell hardness (HB)
1020	Hot-rolled	55	379	30	207	25	111
1020	Cold-drawn	61	420	51	352	15	122
1020	Annealed	60	414	43	296	38	121
1040[1]	Hot-rolled	72	496	42	290	18	144
1040	Cold-drawn	80	552	71	490	12	160
1040	OQT 1300	88	607	61	421	33	183
1040	OQT 400	113	779	87	600	19	262
1050	Hot-rolled	90	620	49	338	15	180
1050	Cold-drawn	100	690	84	579	10	200
1050	OQT 1300	96	662	61	421	30	192
1050	OQT 400	143	986	110	758	10	321
1117	Hot-rolled	65	448	40	276	33	124
1117	Cold-drawn	80	552	65	448	20	138
1117	WQT 350	89	614	50	345	22	178
1137	Hot-rolled	88	607	48	331	15	176
1137	Cold-drawn	98	676	82	565	10	196
1137	OQT 1300	87	600	60	414	28	174
1137	OQT 400	157	1083	136	938	5	352
1144[1]	Hot-rolled	94	648	51	352	15	188
1144	Cold-drawn	100	690	90	621	10	200
1144	OQT 1300	96	662	68	469	25	200
1144	OQT 400	127	876	91	627	16	277
1213	Hot-rolled	55	379	33	228	25	110
1213	Cold-drawn	75	517	58	340	10	150
12L13	Hot-rolled	57	393	34	234	22	114
12L13	Cold-drawn	70	483	60	414	10	140
1340[1]	Annealed	102	703	63	434	26	207
1340	OQT 1300	100	690	75	517	25	235
1340	OQT 1000	144	993	132	910	17	363
1340	OQT 700	221	1520	197	1360	10	444
1340	OQT 400	285	1960	234	1610	8	578
3140	Annealed	95	655	67	462	25	187
3140	OQT 1300	115	792	94	648	23	233
3140	OQT 1000	152	1050	133	920	17	311
3140	OQT 700	220	1520	200	1380	13	461
3140	OQT 400	280	1930	248	1710	11	555
4130	Annealed	81	558	52	359	28	156
4130	WQT 1300	98	676	89	614	28	202
4130	WQT 1000	143	986	132	910	16	302
4130	WQT 700	208	1430	180	1240	13	415
4130	WQT 400	234	1610	197	1360	12	461
4140[1]	Annealed	95	655	54	372	26	197
4140	OQT 1300	117	807	100	690	23	235
4140	OQT 1000	168	1160	152	1050	17	341
4140	OQT 700	231	1590	212	1460	13	461
4140	OQT 400	290	2000	251	1730	11	578
4150	Annealed	106	731	55	379	20	197
4150	OQT 1300	127	880	116	800	20	262
4150	OQT 1000	197	1360	181	1250	11	401
4150	OQT 700	247	1700	229	1580	10	495
4150	OQT 400	300	2070	248	1710	10	578

(*Continued*)

APPENDIX 3 (*Continued*)

Material designation (SAE number)	Condition	Tensile strength (ksi)	Tensile strength (MPa)	Yield strength (ksi)	Yield strength (MPa)	Ductility (percent elongation in 2 in)	Brinell hardness (HB)
4340[1]	Annealed	108	745	68	469	22	217
4340	OQT 1300	140	965	120	827	23	280
4340	OQT 1000	171	1180	158	1090	16	363
4340	OQT 700	230	1590	206	1420	12	461
4340	OQT 400	283	1950	228	1570	11	555
5140	Annealed	83	572	42	290	29	167
5140	OQT 1300	104	717	83	572	27	207
5140	OQT 1000	145	1000	130	896	18	302
5140	OQT 700	220	1520	200	1380	11	429
5140	OQT 400	276	1900	226	1560	7	534
5150	Annealed	98	676	52	359	22	197
5150	OQT 1300	116	800	102	700	22	241
5150	OQT 1000	160	1100	149	1030	15	321
5150	OQT 700	240	1650	220	1520	10	461
5150	OQT 400	312	2150	250	1720	8	601
5160	Annealed	105	724	40	276	17	197
5160	OQT 1300	115	793	100	690	23	229
5160	OQT 1000	170	1170	151	1040	14	341
5160	OQT 700	263	1810	237	1630	9	514
5160	OQT 400	322	2220	260	1790	4	627
6150[1]	Annealed	96	662	59	407	23	197
6150	OQT 1300	118	814	107	738	21	241
6150	OQT 1000	183	1260	173	1190	12	375
6150	OQT 700	247	1700	223	1540	10	495
6150	OQT 400	315	2170	270	1860	7	601
8650	Annealed	104	717	56	386	22	212
8650	OQT 1300	122	841	113	779	21	255
8650	OQT 1000	176	1210	155	1070	14	363
8650	OQT 700	240	1650	222	1530	12	495
8650	OQT 400	282	1940	250	1720	11	555
8740	Annealed	100	690	60	414	22	201
8740	OQT 1300	119	820	100	690	25	241
8740	OQT 1000	175	1210	167	1150	15	363
8740	OQT 700	228	1570	212	1460	12	461
8740	OQT 400	290	2000	240	1650	10	578
9255	Annealed	113	780	71	490	22	229
9255	Q&T 1300	130	896	102	703	21	262
9255	Q&T 1000	181	1250	160	1100	14	352
9255	Q&T 700	260	1790	240	1650	5	534
9255	Q&T 400	310	2140	287	1980	2	601

Notes: Properties common to all carbon and alloy steels:
Poisson's ratio: 0.27.
Shear modulus: 11.5×10^6 psi; 80 GPa.
Coefficient of thermal expansion: 6.5×10^{-6}°F^{-1}.
Density: 0.283 lb/in^3; 7680 kg/m^3.
Modulus of elasticity: 30×10^6 psi; 207 GPa.
[1]See Appendix 4 for graphs of properties versus heat treatment.

APPENDIX 5 Properties of Carburized Steels

Material designation (SAE number)	Condition	Core properties: Tensile strength (ksi)	Tensile strength (MPa)	Yield strength (ksi)	Yield strength (MPa)	Ductility (percent elongation in 2 in)	Brinell hardness (HB)	Case hardness (HRC)
1015	SWQT 350	106	731	60	414	15	217	62
1020	SWQT 350	129	889	72	496	11	255	62
1022	SWQT 350	135	931	75	517	14	262	62
1117	SWQT 350	125	862	66	455	10	235	65
1118	SWQT 350	144	993	90	621	13	285	61
4118	SOQT 300	143	986	93	641	17	293	62
4118	DOQT 300	126	869	63	434	21	241	62
4118	SOQT 450	138	952	89	614	17	277	56
4118	DOQT 450	120	827	63	434	22	229	56
4320	SOQT 300	218	1500	178	1230	13	429	62
4320	DOQT 300	151	1040	97	669	19	302	62
4320	SOQT 450	211	1450	173	1190	12	415	59
4320	DOQT 450	145	1000	94	648	21	293	59
4620	SOQT 300	119	820	83	572	19	277	62
4620	DOQT 300	122	841	77	531	22	248	62
4620	SOQT 450	115	793	80	552	20	248	59
4620	DOQT 450	115	793	77	531	22	235	59
4820	SOQT 300	207	1430	167	1150	13	415	61
4820	DOQT 300	204	1405	165	1140	13	415	60
4820	SOQT 450	205	1410	184	1270	13	415	57
4820	DOQT 450	196	1350	171	1180	13	401	56
8620	SOQT 300	188	1300	149	1030	11	388	64
8620	DOQT 300	133	917	83	572	20	269	64
8620	SOQT 450	167	1150	120	827	14	341	61
8620	DOQT 450	130	896	77	531	22	262	61
E9310	SOQT 300	173	1190	135	931	15	363	62
E9310	DOQT 300	174	1200	139	958	15	363	60
E9310	SOQT 450	168	1160	137	945	15	341	59
E9310	DOQT 450	169	1170	138	952	15	352	58

Notes: Properties given are for a single set of tests on 1/2-in round bars.
SWQT: single water-quenched and tempered.
SOQT: single oil-quenched and tempered.
DOQT: double oil-quenched and tempered.
300 and 450 are the tempering temperatures in °F. Steel was carburized for 8 h. Case depth ranged from 0.045 to 0.075 in.

39. Because we haven't already done enough work, let's go ahead and compute the power transmitting capacity

$$P_{\text{cap}} = \frac{s_{at} Y_N F J n_P D_P}{126000 P_d (SF) K_R K_O K_s K_m K_B K_v} = \frac{n_P F I}{126000 K_O K_s K_m K_v} \left(\frac{s_{ac} D_P Z_N}{(SF) K_R C_P} \right)^2$$

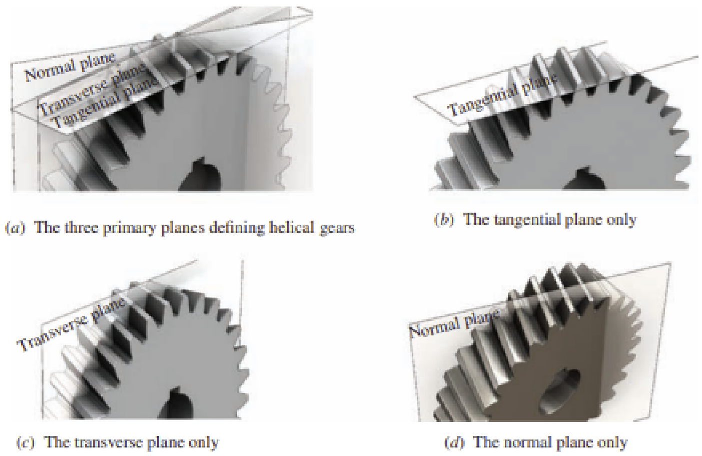
40. Anything with $HB > 400$ should use flame or induction hardening techniques over through hardening, since they can provide the strength required for high contact stress because their inside is still ductile to prevent failure. But fuck theory here's a table that does it for you

TABLE 9–8 Examples of Gear Materials

Heat treatment	Typical alloys (SAE numbers)
Through-hardened or Case-hardened by flame or induction hardening	1045, 4140, 4150, 4340, 4350
Carburizing, case hardened	1020, 4118, 4320, 4820, 8620, 9310

2.2 Helical Gears

2.2.1 Anatomy



(a) The three primary planes defining helical gears

(b) The tangential plane only

(c) The transverse plane only

(d) The normal plane only

2.2.2 Nomenclature

N = number of teeth
D = pitch diameter (in)
p_t = transverse circular pitch (in)
p_n = normal circular pitch (in)
$P_d = P_t$ = diametral pitch (teeth/in)
$P_{nd} = P_n$ = normal diametral pitch (teeth/in)
p_x = axial pitch (in)
m = metric module
m_n = normal metric module
Face Contact Ratio = number of axial pitches in the face width
F = face width (in)
ψ = helix angle
ϕ_n = normal pressure angle
ϕ_t = transverse pressure angle
T = torque (lbf · in)
W_t = transmitted load (lbf)
W_a = axial load (lbf)
W_r = radial load (lbf)

2.2.3 Formulae

I have no clue. Use any of the random fucking formulas below to get an answer.

$$\text{angle relationship: } \tan\phi_n = \tan\phi_t \cos\psi$$

$$\text{transverse circular pitch: } p_t = \frac{\pi D_P}{N_P} = \frac{\pi D_G}{N_G} = \frac{\pi}{P_d}$$

$$\text{normal circular pitch: } p_n = p_t \cos\psi$$

$$\text{axial pitch: } p_x = \frac{p_t}{\tan\psi} = \frac{\pi P_d}{\tan\psi} = m\pi$$

$$\text{Face Contact Ratio} = \frac{F}{p_x} > 2.0$$

$$\text{diametral pitch: } P_d = \frac{N}{D}$$

$$\text{normal diametral pitch: } P_{nd} = \frac{P_d}{\cos\psi}$$

$$P_d p_t = \pi$$

$$P_{nd} p_n = \pi$$

$$\text{metric module: } m = \frac{D}{N}$$

$$\text{normal metric module: } m_n = \frac{1}{P_{nd}} = \frac{\cos\psi}{P_d} = \frac{D\cos\psi}{N} = m\cos\psi$$

$$W = \frac{W_t}{\cos\phi_n \cos\psi}$$

Forces and motion:

$$\text{torque: } T = \frac{63000P}{n}$$

$$\text{pitch line speed: } v_t = \frac{\pi D n}{12}$$

$$\text{tangential force: } W_t = \frac{33000P}{v_t} = \frac{126000P}{nD}$$

$$\text{radial force: } W_r = W_t \tan\phi_t$$

$$\text{axial force: } W_x = W_t \tan\psi$$

$$\text{bending stress number: } s_t = \frac{W_t P_d}{FJ} K_O K_s K_m K_B K_v$$

$$\text{contact stress number: } s_c = C_p \sqrt{\frac{W_t K_O K_s K_m K_v}{F D_p I}}$$

$$\text{allowable bending stress: } s_{at} > s_t \frac{(SF)K_R}{Y_N}$$

$$\text{allowable contact stress: } s_{ac} > s_c \frac{(SF)K_R}{Z_N}$$

2.2.4 Design Selection

1. Find the type of shock for input and output from this random place in the textbook:

   Uniform: Electric motor or constant-speed gas turbine

   Light shock: Water turbine, variable-speed drive

   Moderate shock: Multicylinder engine

   Examples of the roughness of driven machines include the following:

   Uniform: Continuous-duty generator, paper, and film winders.

   Light shock: Fans and low-speed centrifugal pumps, liquid agitators, variable-duty generators, uniformly loaded conveyors, rotary positive displacement pumps, and metal strip processing.

   Moderate shock: High-speed centrifugal pumps, reciprocating pumps and compressors, heavy-duty conveyors, machine tool drives, concrete mixers, textile machinery, meat grinders, saws, bucket elevators, freight elevators, escalators, concrete mixers, plastics molding and processing, sewage disposal equipment, winches, and cable reels.

   Heavy shock: Rock crushers, punch press drives, pulverizers, processing mills, tumbling barrels, wood chippers, vibrating screens, railroad car dumpers, log conveyors, lumber handling equipment, metal shears, hammer mills, commercial washers, heavy-duty hoists and cranes, reciprocating feeders, dredges, rubber processing, compactors, and plastics extruders.

2. Get the value of K_O from here

   TABLE 9–1 Suggested Overload Factors, K_o

	Driven Machine			
Power source	Uniform	Light shock	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.50	1.75
Light shock	1.20	1.40	1.75	2.25
Moderate shock	1.30	1.70	2.00	2.75

3. Take a wild fucking guess for the value of P_{nd} and N_P. The one textbook example used $P_{nd} = 12$ and $N_P = 24$ so let's just use those every single time.

4. Compute P_d and p_x

   $$P_d = P_{nd} \cos \psi$$

$$p_x = \frac{\pi}{P_d \tan\psi}$$

5. Assume that n_G is given. If not then refer to the steps in the spur gear design selection guide. Use the speed ratio to get the number of teeth in the gear.

$$VR = \frac{N_G}{N_P} = \frac{n_P}{n_G}$$

6. Compute the tangential pressure angle

$$\phi_t = \arctan\left(\frac{\tan\phi_n}{\cos\psi}\right)$$

7. Compute the diameters of the gears

$$D_P = \frac{N_P}{P_d}$$
$$D_G = \frac{N_G}{P_d}$$

8. Compute the nominal face width

$$F_{\text{nom}} = 2p_x$$

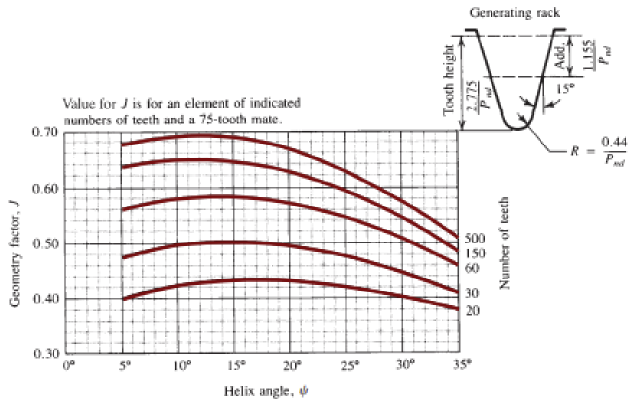
Round it however you want so that the value is convenient.

9. Compute the center distance, pitch line speed, and transmitted load

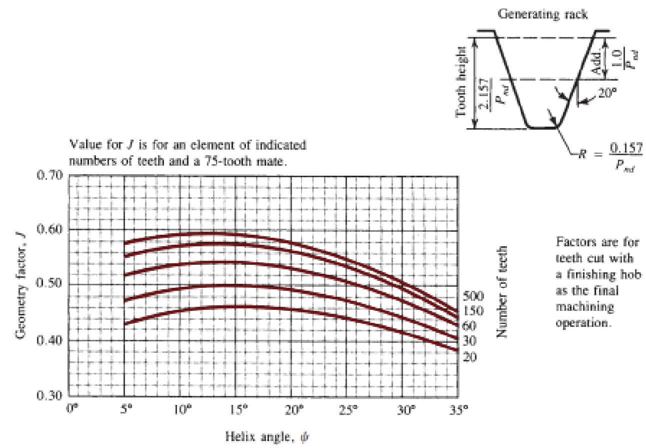
$$C = \frac{N_P + N_G}{2P_d}$$
$$v_t = \frac{\pi}{12} D_P n_P$$
$$W_t = \frac{33000P}{v_t}$$

10. Choose a material (steel) and follow the rest of the steps from the spur gear design selection to get values for $C_p, A_v, K_v, C_{pf}, C_{ma}, K_m, K_s, K_B, SF, K_R, N_c, Y_N, Z_N$.
The only different constants will be J and I which can be gotten from the following steps.

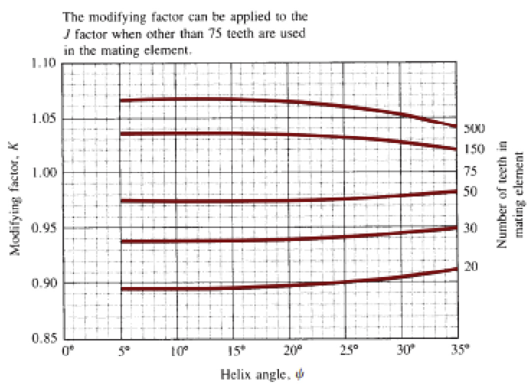
11. Choose the J_P and J_G values from one of the graphs depending on the normal pressure angle ϕ_n. (this is different from spur gears)



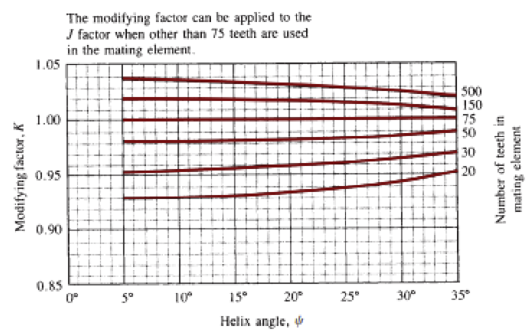
(a) Geometry factor (J) for 15° normal pressure angle and indicated addendum



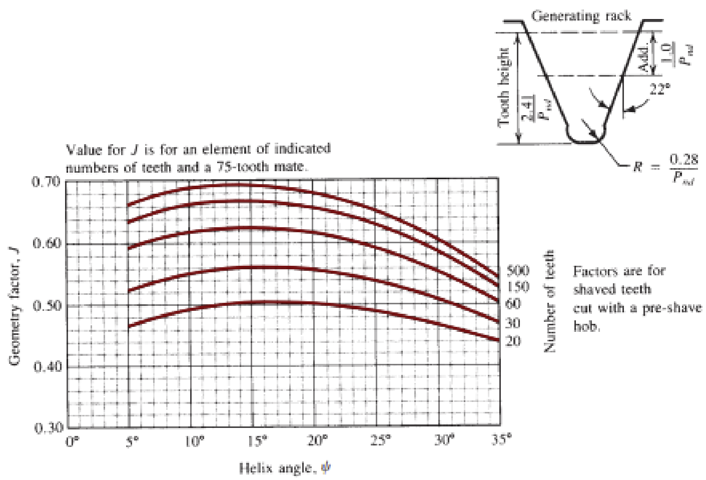
(a) Geometry factor (J) for 20° normal pressure angle, standard addendum, and finishing hob



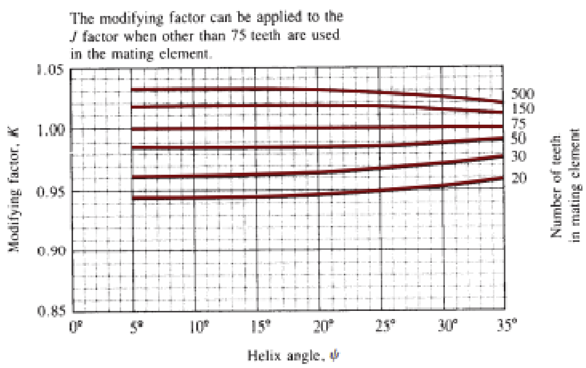
(b) J factor multipliers



(b) J factor multipliers



(a) Geometry factor (J), for 22° normal pressure angle, standard addendum, and pre-shave hob



(b) J factor multipliers

12. Choose the pitting geometry factor, I, from one of these tables.

TABLE 10–1 Geometry Factors for Pitting Resistance, *I*, for Helical Gears with 20° Normal Pressure Angle and Standard Addendum

A. Helix angle ψ = 15.0°

Gear teeth	Pinion teeth 17	21	26	35	55
17	0.124				
21	0.139	0.128			
26	0.154	0.143	0.132		
35	0.175	0.165	0.154	0.137	
55	0.204	0.196	0.187	0.171	0.143
135	0.244	0.241	0.237	0.229	0.209

B. Helix angle ψ = 25.0°

Gear teeth	Pinion teeth 14	17	21	26	35	55
14	0.123					
17	0.137	0.126				
21	0.152	0.142	0.130			
26	0.167	0.157	0.146	0.134		
35	0.187	0.178	0.168	0.156	0.138	
55	0.213	0.207	0.199	0.189	0.173	0.144
135	0.248	0.247	0.244	0.239	0.230	0.210

Source: Extracted from AGMA Standard 908-B89 (R 1999), *Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth,* with the permission of the publisher, American Gear Manufacturers Association, 1001 North Fairfax Street, 5th floor, Alexandria, VA 22314.

TABLE 10–2 Geometry Factors for Pitting Resistance, *I*, for Helical Gears with 25° Normal Pressure Angle and Standard Addendum

A. Helix angle ψ = 15.0°

Gear teeth	Pinion teeth 14	17	21	26	35	55
14	0.130					
17	0.144	0.133				
21	0.160	0.149	0.137			
26	0.175	0.165	0.153	0.140		
35	0.195	0.186	0.175	0.163	0.143	
55	0.222	0.215	0.206	0.195	0.178	0.148
135	0.257	0.255	0.251	0.246	0.236	0.214

B. Helix angle ψ = 25.0°

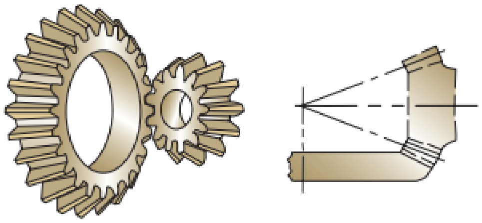
Gear teeth	Pinion teeth 12	14	17	21	26	35	55
12	0.129						
14	0.141	0.132					
17	0.155	0.146	0.135				
21	0.170	0.162	0.151	0.138			
26	0.185	0.177	0.166	0.154	0.141		
35	0.203	0.197	0.188	0.176	0.163	0.144	
55	0.227	0.223	0.216	0.207	0.196	0.178	0.148
135	0.259	0.258	0.255	0.251	0.246	0.235	0.213

Source: Extracted from AGMA Standard 908-B89, *Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth,* with the permission of the publisher, American Gear Manufacturers Association, 1001 North Fairfax Street, 5th floor, Alexandria, VA 22314.

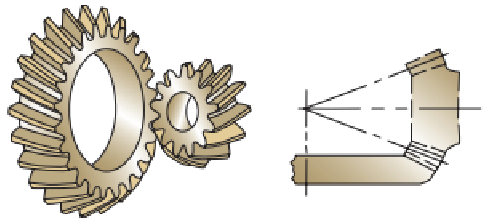
13. Compute s_{tP} and s_{tG} and follow the remaining steps in the spur gear design selection guide.

2.3 Bevel Gears

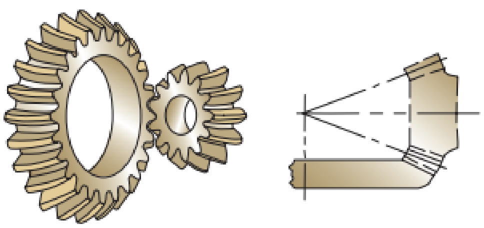
2.3.1 Anatomy



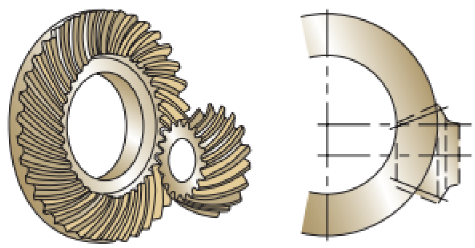
(a) Straight bevel



(b) Spiral bevel

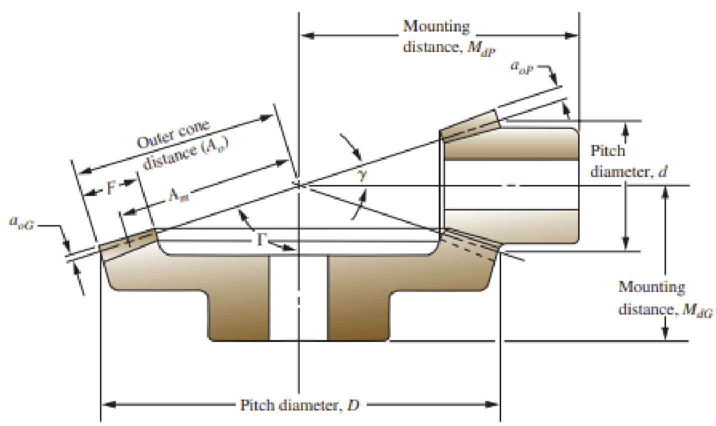


(c) Zero spiral bevel



(d) Hypoid

(e) Straight bevel gear pair



2.3.2 Formulae

N_P = number of teeth on pinion (driving)
N_G = number of teeth on gear (driven)
P_d = diametral pitch (teeth/in)
d = diameter of pinion (in)
D = diameter of gear (in)
γ = cone angle of pinion
Γ = cone angle of gear
ϕ = pressure angle
A_o = outer cone distance (in)

F = face width (in)
A_m = mean cone distance (in)
p_m = mean circular path (in)
h = mean working depth (in)
c = clearance (in)
h_m = mean whole depth (in)
c_1 = mean addendum factor
a_G = gear mean addendum (in)
a_P = pinion mean addendum (in)
b_G = gear mean dedendum (in)
b_P = pinion mean dedendum (in)
δ_G = gear dedendum angle
δ_P = pinion dedendum angle
a_oG = gear outer addendum (in)
a_oP = pinion outer addendum (in)
D_o = gear outside diameter (in)
d_o = pinion outside diameter (in)
W_t = transmitted load (lbf)
W_x = axial load (lbf)
W_r = radial load (lbf)

2.3.3 Formulae

Geometry:

$$\text{diametral pitch: } P_d = \frac{N_P}{d} = \frac{N_G}{D}$$

$$\text{gear ratio: } m_G = \frac{N_G}{N_P}$$

$$\text{pinion cone angle: } \tan\gamma = \frac{N_P}{N_G}$$

$$\text{gear cone angle: } \tan\Gamma = \frac{N_G}{N_P}$$

$$\text{outer cone distance: } A_{oG} = \frac{D}{2\sin\Gamma},\ A_{oP} = \frac{d}{2\sin\gamma}$$

$$\text{nominal face width: } F_{\text{nom}} = 0.3A_o$$

$$\text{max face width: } F_{\max} = \min\left\{\frac{A_o}{3}, \frac{10}{P_d}\right\}$$

$$\text{mean cone distance: } A_m = A_o - 0.5F$$

$$\text{mean circular pitch: } p_m = \frac{\pi A_m}{P_d A_o}$$

$$\text{mean working depth: } h = \frac{2A_m}{P_d A_o}$$

$$\text{clearance: } c = 0.125h$$

$$
\begin{aligned}
\text{mean whole depth: } & h_m = h + c \\
\text{mean addendum factor: } & c_1 = 0.21 + \frac{0.29}{m_G^2} \\
\text{gear mean addendum: } & a_G = c_1 h \\
\text{pinion mean addendum: } & a_P = h - a_G \\
\text{gear mean dedendum: } & b_G = h_m - a_G \\
\text{pinion mean dedendum: } & b_P = h_m - a_P \\
\text{gear dedendum angle: } & \tan\delta_G = \frac{b_G}{A_{mG}} \\
\text{pinion dedendum angle: } & \tan\delta_P = \frac{b_P}{A_{mP}} \\
\text{gear outer addendum: } & a_{oG} = a_G + 0.5F\tan\delta_P \\
\text{pinion outer addendum: } & a_{oP} = a_P + 0.5F\tan\delta_G \\
\text{gear outside diameter: } & D_o = D + 2a_{oG}\cos\Gamma \\
\text{pinion outside diameter: } & d_o = d + 2a_{oP}\cos\gamma
\end{aligned}
$$

Forces and motion:

$$
\begin{aligned}
\text{pitch line speed: } & v_t = \frac{\pi D n_G}{12} = \frac{\pi d n_P}{12} \\
\text{torque: } & T = \frac{63000P}{n} \\
\text{mean radius: } & r_m = \frac{d}{2} - \frac{F}{2}\sin\gamma \\
& R_m = \frac{D}{2} - \frac{F}{2}\sin\Gamma \\
\text{transmitted load: } & W_t = \frac{T_P}{r_m} = \frac{T_G}{R_m} \\
\text{radial load: } & W_r = W_t\tan\phi\cos\Gamma = W_t\tan\phi\cos\gamma \\
\text{axial load: } & W_x = W_t\tan\phi\sin\Gamma = W_t\tan\phi\sin\gamma \\
\text{also transmitted load: } & W_t = \frac{126000P}{Dn} = \frac{33000P}{v_t}
\end{aligned}
$$

You may notice that the formulas for transmitted load are different. I don't know why. Use the top one for force analysis and the bottom one for design selection.

2.3.4 Force Analysis

1. Find the transmitted, axial, and radial loads, W_t, W_r, W_x.

2. Draw a free body diagram of the forces acting on the gear. Include $\vec{W}$, the two forces on the bearing and the torque on the shaft (these directions will all be arbitrary except for $\vec{W}$). The force $\vec{W}$ will act at a distance of R_m away from the axle of the gear.
   W_t will point in the direction of the motion of the gear at that point
   W_r will point toward the shaft

W_a will point in the direction of the angular velocity vector of the gear (using the right hand rule with the rotation of the gear)

3. Find the sum of moments about one of the bearings and the sum of forces to solve for the unknowns.
   Recall

$$\vec{M} = \vec{r} \times \vec{F}$$

   Note that the axial force of the bearings will only take place on one bearing. Choose which bearing to select for either compressive or tensile force.

2.3.5 Design Selection

1. Find the shock and get K_O from the spur gear guide.

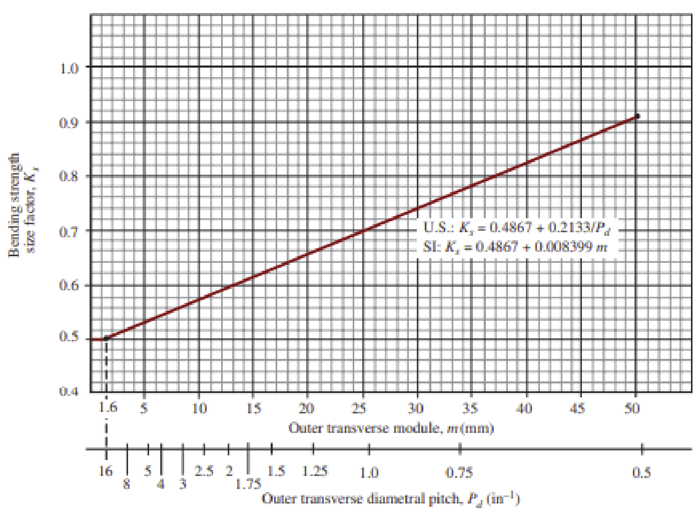
2. compute or guess any of the basic mechanical values missing, such as N or F, using the spur gear guide as reference.

3. Compute v_t and W_t

$$v_t = \frac{\pi D n_G}{12}$$
$$W_t = \frac{33000P}{v_t}$$

4. Find the size factor K_s from this equation or the table

$$K_s = \begin{cases} 0.5 & P_d \geq 16 \\ 0.4867 + \frac{0.2133}{P_d} & P_d < 16 \end{cases}$$



5. Get K_{mb} where

   - $K_{mb} = 1$ for both gears straddle mounted

- $K_{mb} = 1.1$ for one gear straddle mounted
- $K_{mb} = 1.25$ for neither gear straddle mounted

6. Compute $K_m = K_{mb} + 0.0036F^2$
7. Find the quality number A_v from the application or the pitch line speed. This table is shit so just guess what looks right.

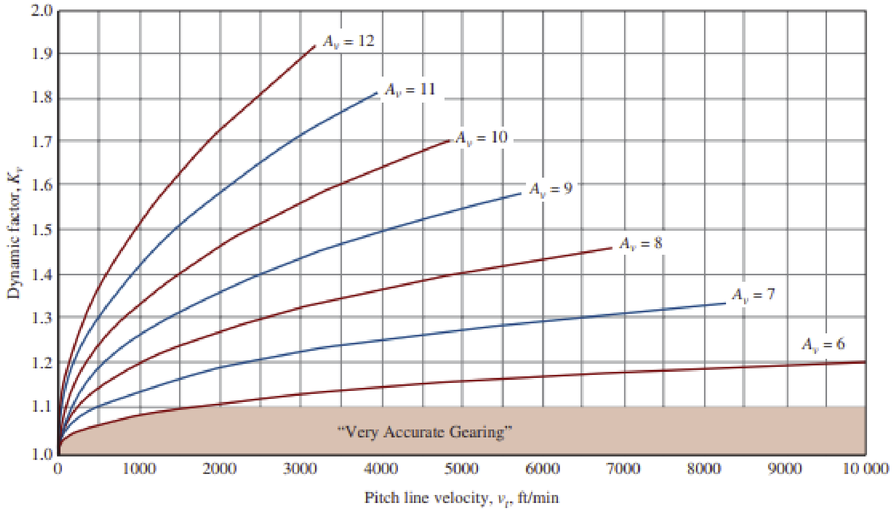
TABLE 9–5 Recommended AGMA Quality Numbers

Application	Quality number	Application	Quality number
Cement mixer drum drive	A11	Small power drill	A9
Cement kiln	A11	Clothes washing machine	A8
Steel mill drives	A11	Printing press	A7
Grain harvester	A10	Computing mechanism	A6
Cranes	A10	Automotive transmission	A6
Punch press	A10	Radar antenna drive	A5
Mining conveyor	A10	Marine propulsion drive	A5
Paper-box-making machine	A9	Aircraft engine drive	A4
Gas meter mechanism	A9	Gyroscope	A2

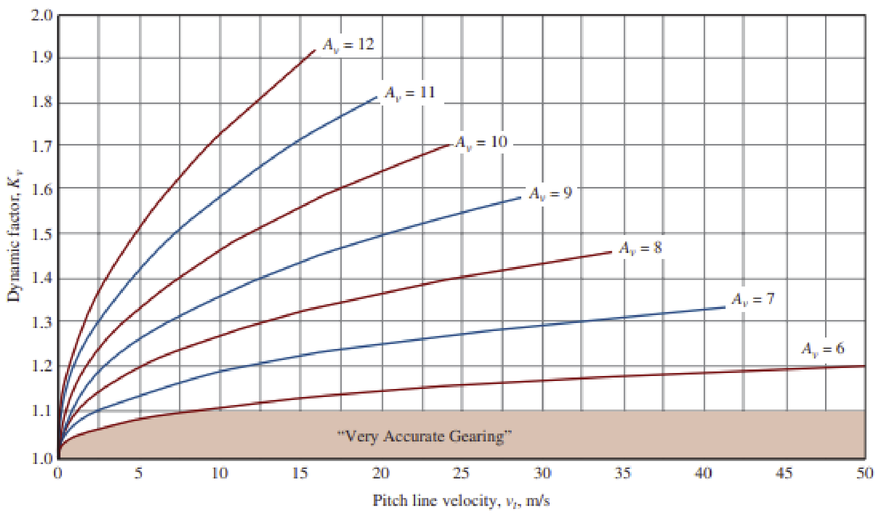
Machine tool drives and drives for other high-quality mechanical systems

Pitch line speed (fpm)	Quality number	Pitch line speed (m/s)
0–800	A10	0–4
800–2000	A8	4–11
2000–4000	A6	11–22
Over 4000	A4	Over 22

8. Get K_v from here

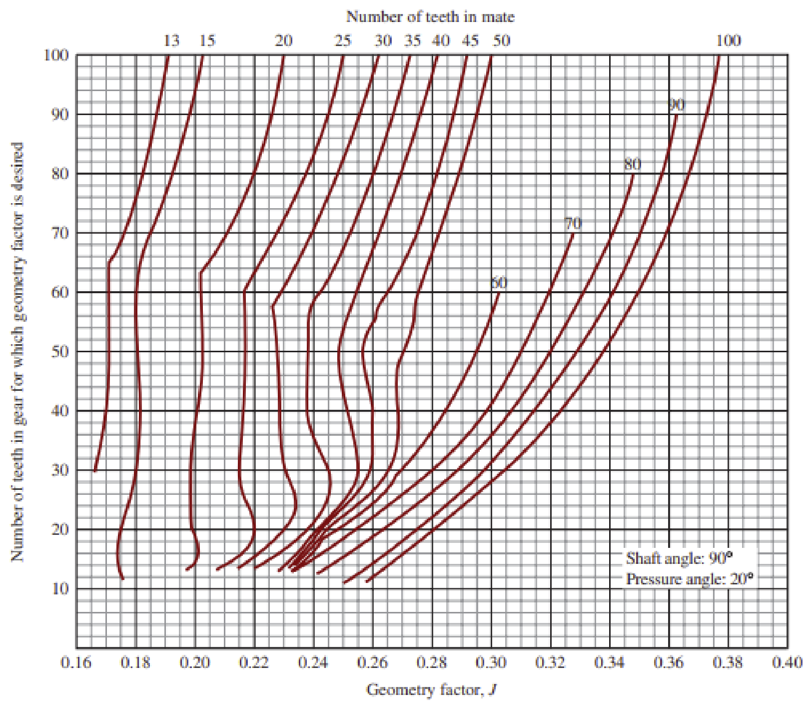


(*a*) U.S. Customary units



(*b*) SI Metric units

9. Get J from the mutant octopus graph you see below



10. Calculate the bending stress number

$$s_t = \frac{W_t P_d K_O K_s K_m K_v}{FJ}$$

11. Specify a service factor, SF between 1.00 and 1.50. Usually pick 1.00 but if your data is uncertain then ramp that shit up.

12. Gander a guess at how reliable your system will be. Let's assume for most cases that you're not that shit of an Engineer and it works 99% of the time.

13. Use your rigorously calculated reliability to get K_R and C_R from this table

TABLE 10–3 Reliability Factors for Allowable Bending and Contact Stresses

		Reliability factors	
Reliability R	Interpretation	Bending K_R	Contact C_R
0.9	Fewer than one failure in 10	0.85	0.92
0.99	Fewer than one failure in 100	1.00	1.00
0.999	Fewer than one failure in 1000	1.25	1.12
0.9999	Fewer than one failure in 10 000	1.50	1.22

Source: Adapted from AGMA 2003-C10, *Rating the Pitting Resistance and Bending Strength of Generated Straight Bevel, Zerol Bevel and Spiral Bevel Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1001 North Fairfax Street, 5th Floor, Alexandria, VA.

14. Guess what the lifetime of your machine will be. Don't worry, there's a shitty table to help you.

TABLE 9–12 Recommended Design Life

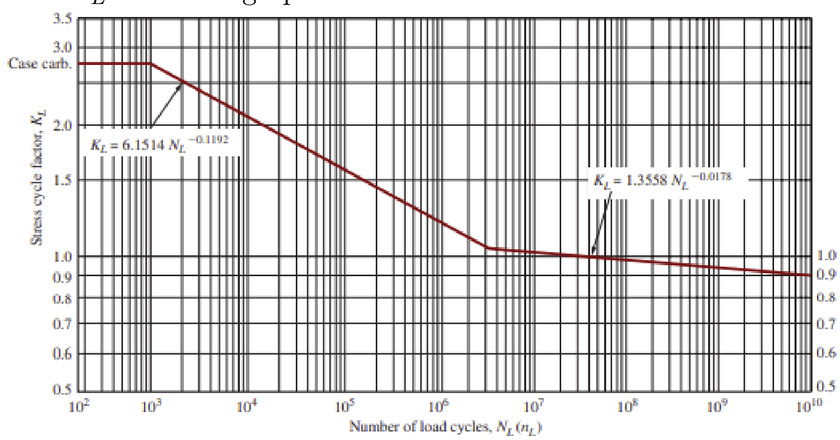
Application	Design life (h)
Domestic appliances	1000–2000
Aircraft engines	1000–4000
Automotive	1500–5000
Agricultural equipment	3000–6000
Elevators, industrial fans, multipurpose gearing	8000–15 000
Electric motors, industrial blowers, general industrial machines	20 000–30 000
Pumps and compressors	40 000–60 000
Critical equipment in continuous 24-h operation	100 000–200 000

Source: Eugene A. Avallone and Theodore Baumeister III, eds. *Marks' Standard Handbook for Mechanical Engineers.* 9th ed. New York: McGraw-Hill, 1986.

15. Find the number of loading cycles using these formulas

$$N_{LP} = (60)(\text{lifetime})n_P$$
$$N_{LG} = (60)(\text{lifetime})n_G$$

16. Find K_L from this graph



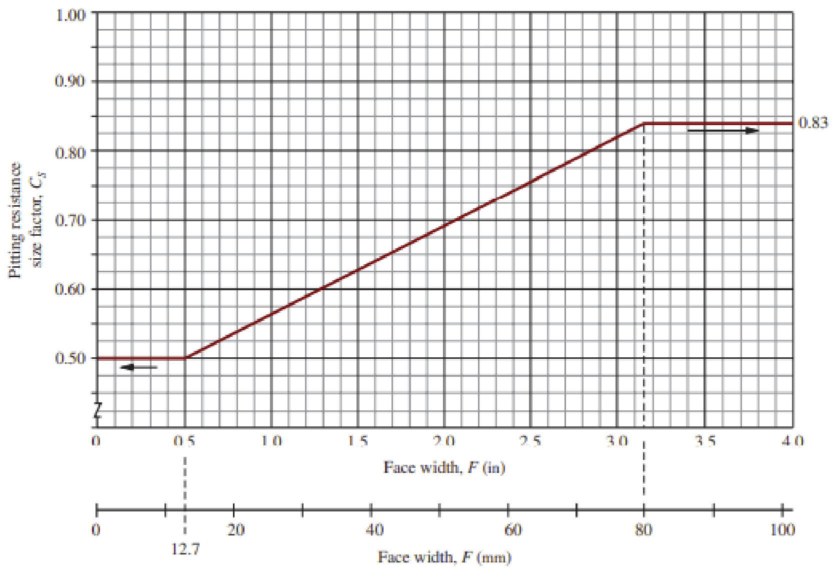
17. Take the safety factor SF to be anywhere between 1 and 1.5. We always just assume $SF = 1$ because fuck safety.

18. find the max allowable bending strength

$$s_{at} = \frac{s_t(SF)K_R}{K_L}$$

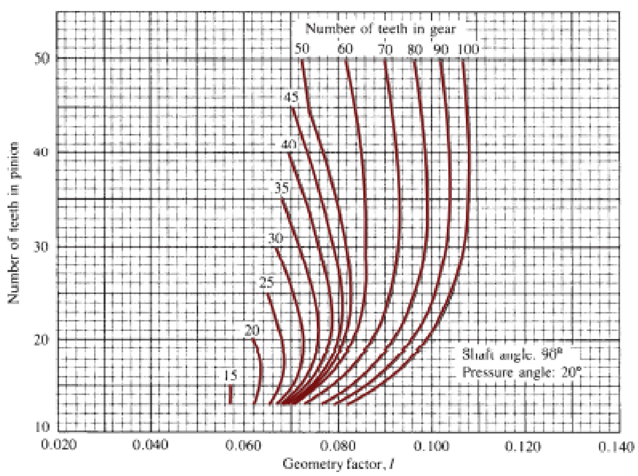
19. Take $C_p = 2300$ for steel

20. Compute $C_s = 0.125F + 0.4375$



21. Get $C_{xc} = 1.5$ for properly crowned teeth (what we usually assume)
    $C_{xc} = 2$ for non-crowned teeth.

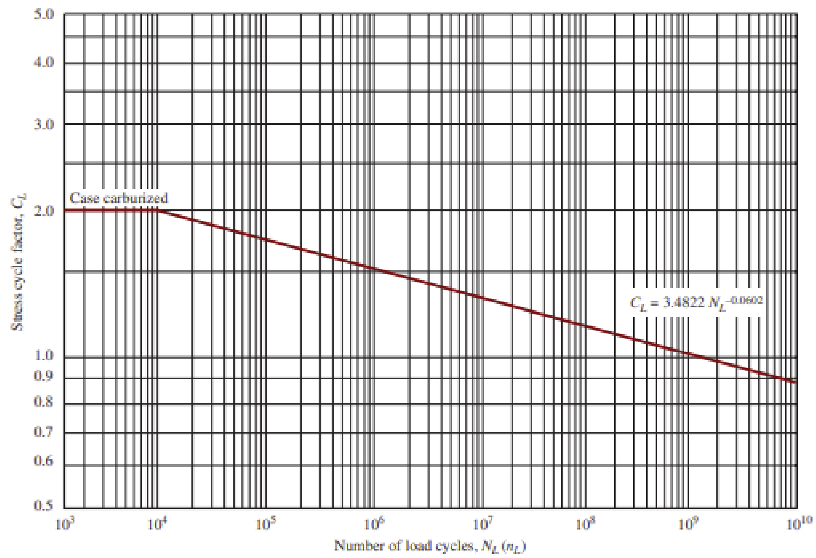
22. Get I from this graph



23. Compute the contact stress number

$$s_c = C_p \sqrt{\frac{W_t K_O K_m K_v C_s C_{xc}}{F D_p I}}$$

24. Get C_L from here



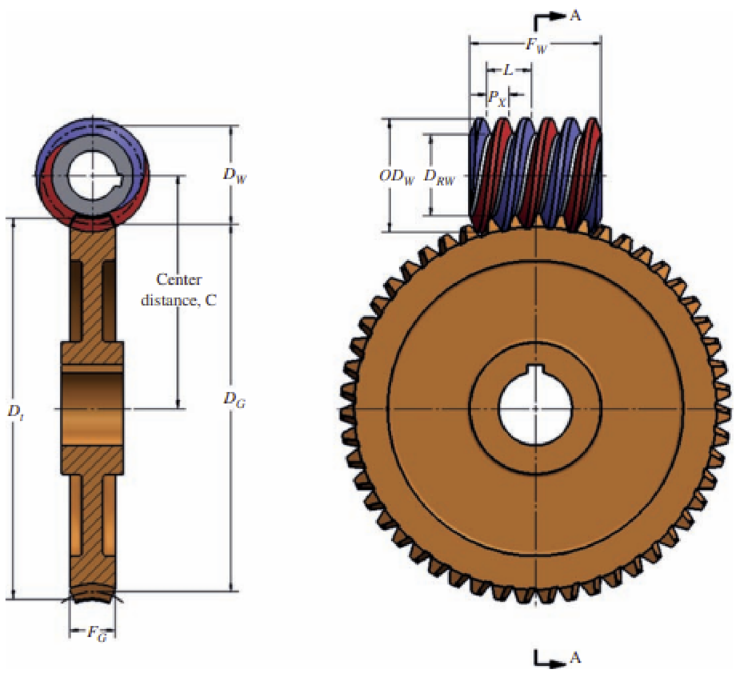
25. Get the max allowable contact stress number

$$s_{ac} = \frac{s_c(SF)C_R}{C_L}$$

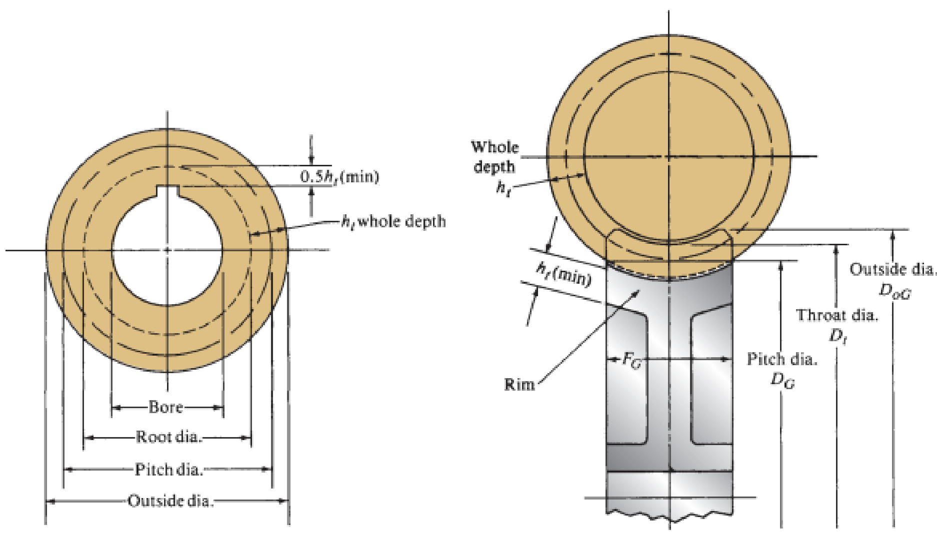
26. Follow the remaining steps for material selection from spur gear guide.

2.4 Worm Gears

2.4.1 Anatomy



Section A-A



2.4.2 Nomenclature

N_G = number of teeth on the gear
N_W = number of worm threads
D_G = pitch diameter of the gear (in)
D_W = pitch diameter of the worm (in)

p = circular pitch (in)
P_d = diametral pitch (teeth/in)
m = module
L = lead (in): axial distance if the worm completes one revolution
P_x = axial pitch
λ = lead angle
C = center distance (in)
ϕ_n = normal pressure angle
ϕ_t = transverse pressure angle
a = addendum (in)
h_t = whole depth (in)
h_k = working depth (in)
b = dedendum (in)
D_{rW} = root diameter of worm (in)
D_{oW} = outside diameter of worm (in)
D_{rG} = root diameter of gear (in)
D_t = throat diameter of gear (in)
F_G = face width of wormgear (in)
F_W = face length of worm (in)
n_W = speed of worm (rpm)
n_G = speed of gear (rpm)
v_{tW} = pitch line speed for worm (ft/min)
v_{tG} = pitch line speed for gear (ft/min)
VR = velocity ratio

2.4.3 Formulae

Geometry:

$$\begin{aligned}
\text{circular pitch: } & p = \frac{\pi D_G}{N_G} = \pi m \\
\text{diametral pitch: } & P_d = \frac{N_G}{D_G} \\
& P_d p = \pi \\
\text{module: } & m = \frac{D}{N} \\
\text{axial pitch: } & P_x = p \\
\text{lead: } & L = N_W P_x \\
\text{lead angle: } & \tan\lambda = \frac{L}{\pi D_W} \\
\text{center distance: } & C = \frac{D_W + D_G}{2} = \frac{N_W + N_G}{2P_d} \\
\text{angle relationship: } & \tan\phi_n = \tan\phi_t \cos\lambda \\
\text{addendum: } & a = 0.3183 P_x = \frac{1}{P_d}
\end{aligned}$$

$$\text{whole depth: } h_t = 0.6866 P_x = \frac{2.157}{P_d}$$

$$\text{working depth: } h_k = 2a$$

$$\text{dedendum: } b = h_t - a$$

$$\text{root diameter of worm: } D_{rW} = D_W - 2b$$

$$\text{outside diameter of worm: } D_{oW} = D_W + 2a = D_W + h_k$$

$$\text{root diameter of gear: } D_{rG} = D_G - 2b$$

$$\text{throat diameter of gear: } D_t = D_G + 2a$$

$$\text{face width of wormgear: } F_G = \sqrt{D_{oW}^2 - D_{W^2}} = 2p = \frac{2\pi}{P_d} \approx \frac{6}{P_d}$$

$$\text{face length of worm: } F_W = 2\sqrt{\left(\frac{D_t}{2}\right)^2 - \left(\frac{D_G}{2-a}\right)^2}$$

Speed:

$$\text{pitch line speed for worm: } v_{tW} = \frac{\pi D_W n_W}{12} \text{ or } v_{tW} = \frac{\pi D_W n_W}{60000} \text{ m/s}$$

$$\text{pitch line speed for gear: } v_{tG} = \frac{\pi D_G n_G}{12} \text{ or } v_{tG} = \frac{\pi D_G n_G}{60000} \text{ m/s}$$

$$\text{velocity ratio: } VR = \frac{n_W}{n_G} = \frac{N_G}{N_W}$$

$$\text{sliding speed: } v_s = \frac{v_{tG}}{\sin\lambda} = \frac{v_{tW}}{\cos\lambda}$$

Forces:

$$\text{force relationship: } W_{tG} = W_{xW},\ W_{xG} = W_{tW},\ W_{rG} = W_{rW}$$

$$\text{output torque: } T_o = \frac{63000 P_o}{n_G} = \frac{W_{tG} D_G}{2}$$

$$\text{transmitted force: } W_{tG} = \frac{2T_o}{D_G}$$

$$\text{axial force: } W_{xG} = W_{tG}\frac{\cos\phi_n \sin\lambda + \mu\cos\lambda}{\cos\phi_n\cos\lambda - \mu\sin\lambda}$$

$$\text{radial force: } W_{rG} = \frac{W_{tG}\sin\phi_n}{\cos\phi_n\cos\lambda - \mu\sin\lambda}$$

$$\text{friction force: } W_f = \frac{\mu W_{tG}}{\cos\lambda\cos\phi_n - \mu\sin\lambda}$$

$$\text{power loss due to friction: } P_L = \frac{v_s W_f}{33000}$$

$$\text{input power: } P_i = P_o + P_L$$

$$\text{efficiency: } \eta = \frac{P_o}{P_i} = \frac{\cos\phi_n - \mu\tan\lambda}{\cos\phi_n + \frac{\mu}{\tan\lambda}}$$

2.4.4 Design Selection

1. Compute the lead and lead angle

$$p = \frac{\pi}{P_d} = \frac{\pi D_G}{N_G}$$
$$P_x = p$$
$$L = N_W P_x$$
$$\lambda = \arctan\left(\frac{L}{\pi D_W}\right)$$

2. Compute the center distance

$$C = \frac{D_G + D_W}{2}$$

3. Compute the pitch line speed of the gear

$$v_{tG} = \frac{\pi D_G n_G}{12}$$

4. Compute the sliding speed

$$v_s = \frac{v_{tG}}{\sin\lambda}$$

5. Find the coefficient of friction

$$\mu = \begin{cases} 0.15 & v_s = 0 \\ 0.124e^{-0.074v_s^{0.645}} & 0 < v_s < 10 \\ 0.103e^{-0.11v_s^{0.45}} + 0.012 & v_s > 10 \end{cases}$$

6. Compute the forces on the gear

$$T_o = \frac{63000P_o}{n_G} = \frac{W_{tG}D_G}{2}$$
$$W_{tG} = \frac{2T_o}{D_G}$$
$$W_{xG} = W_{tG}\frac{\cos\phi_n \sin\lambda + \mu\cos\lambda}{\cos\phi_n\cos\lambda - \mu\sin\lambda}$$
$$W_{rG} = \frac{W_{tG}\sin\phi_n}{\cos\phi_n\cos\lambda - \mu\sin\lambda}$$

7. Compute the friction force

$$W_f = \frac{\mu W_{tG}}{\cos\lambda\cos\phi_n - \mu\sin\lambda}$$

8. Compute the power loss due to friction

$$P_L = \frac{v_s W_f}{33000}$$

9. Compute the input power $P_i = P_o + P_L$

10. Compute the efficiency

$$\eta = \frac{P_o}{P_i} = \frac{\cos\phi_n - \mu\tan\lambda}{\cos\phi_n + \frac{\mu}{\tan\lambda}}$$

11. Find the Lewis form factor y

TABLE 10-5 Approximate Lewis Form Factor for Wormgear Teeth

ϕ_n	y
$14\frac{1}{2}°$	0.100
20°	0.125
25°	0.150
30°	0.175

12. Find the normal circular pitch

$$p_n = p\cos\lambda = \frac{\pi\cos\lambda}{P_d}$$

13. Compute K_v

$$K_v = \frac{1200}{1200 + v_{tG}}$$

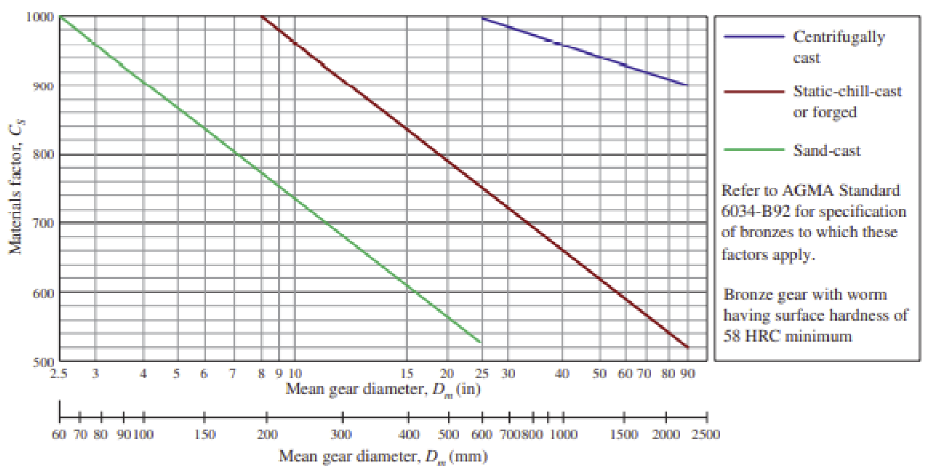
14. Compute the dynamic load

$$W_d = \frac{W_{tG}}{K_v}$$

15. Find the stress in the gear teeth

$$\sigma = \frac{W_d}{yFp_n}$$

16. Find C_s



For sand-cast bronze

$$C_s = \begin{cases} 1189.636 - 476.545\log_{10}(D_G) & D_G > 2.5 \\ 1000 & D_G < 2.5 \end{cases}$$

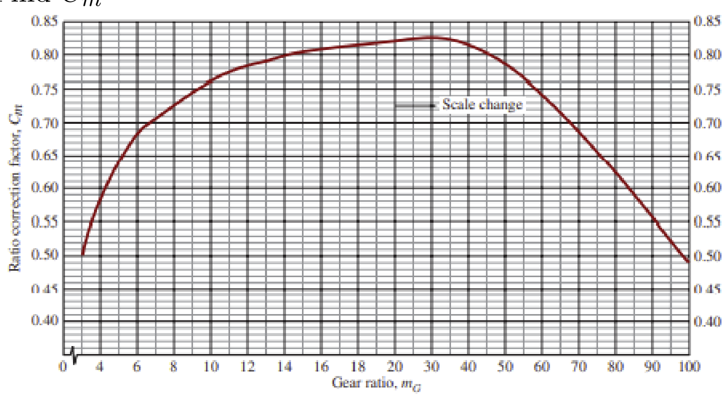
For static-chill-cast or forged bronze

$$C_s = \begin{cases} 1411.651 - 455.825\log_{10}(D_G) & \\ 1000 & D_G < 8 \end{cases}$$

For centrifugally cast bronze

$$C_s = \begin{cases} 1251.291 - 179.75\log_{10}(D_G) & \\ 1000 & D_G < 25 \end{cases}$$

17. Find C_m



$$C_m = \begin{cases} 0.02\sqrt{-m_G^2 + 40m_G - 76} + 0.46 & 6 < m_G < 20 \\ 0.0107\sqrt{-m_G^2 + 56m_G + 5146} & 20 < m_G < 76 \\ 1.1483 - 0.00658m_G & m_G > 76 \end{cases}$$

18. Find C_v

$$C_v = \begin{cases} 0.659e^{-0.0011v_s} & 0 < v_s < 700 \\ 13.31v_s^{-0.571} & 700 < v_s < 3000 \\ 65.52v_s^{-0.774} & v_s > 3000 \end{cases}$$

19. Find F_e

$$F_e = \begin{cases} F & F < \frac{D_W}{3} \\ \frac{D_W}{3} & F > \frac{D_W}{3} \end{cases}$$

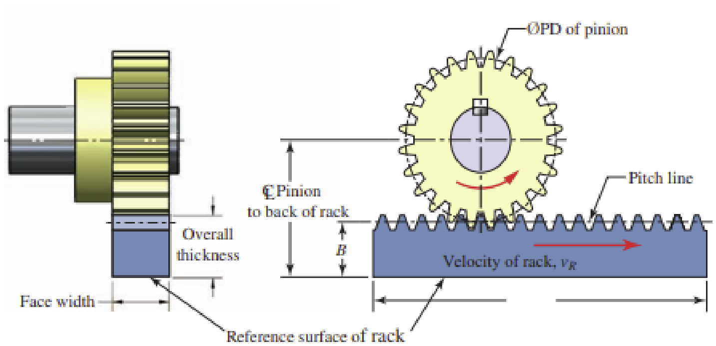
20. Find the rated tangential load

$$W_{tR} = C_s D_G^{0.8} F_e C_m C_v$$

21. Check if the design is satisfactory to resist pitting:
If $W_{tR} > W_{tG}$ then the design is satisfactory

2.5 Rack and Pinion

2.5.1 Anatomy



2.5.2 Nomenclature

P_d = diametral pitch (teeth/in)
N_p = number of teeth on the pinion
D_p = pitch diameter (in)
n_P = angular speed of the pinion (rpm)
v_t = pitch line velocity of the pinion
B = distance from pitch line to back (in) (tab. 8-10)
$B - C$ = distance from back of the rack to the pinion centerline (in)
V_{rack} = speed of rack (ft/min)
s_{rack} = distance rack travels (ft)
t = time (s)
θ_p = number of revolutions of the pinion (rev)

2.5.3 Formulae

$$\text{pitch line speed: } v_t = \frac{D_P n_P}{2}$$

$$\text{speed of rack: } V_{\text{rack}} = \frac{\pi D_P n_P}{12}$$

$$\text{distance rack travels: } s_{\text{rack}} = \frac{D_P \theta_P}{2}$$

2.5.4 Analysis Method

1. Find pitch diameter D_p

$$D_p = \frac{N}{P_d}$$

2. Find distance from pitch line to back B from the table

TABLE 8–10 Example rack specifications

Diametral pitch	Pitch line to back (*B*)	Overall thickness	Face width	Nominal length [ft]
64	0.109	0.125	0.125	2
48	0.104	0.125	0.125	2
32	0.156	0.187	0.187	4
24	0.208	0.250	0.25	4
20	0.450	0.500	0.5	6
16	0.688	0.750	0.75	6
12	0.917	1.000	1	6
10	1.150	1.250	1.25	6
8	1.375	1.500	1.5	6
6	1.333	1.500	2	6
5	1.300	1.500	2.5	6
4	1.750	2.000	3.5	6

3. Find distance from back of the rack to the pinion centerline $B-C$

$$B - C = B + \frac{D_p}{2}$$

4. Find the velocity of the rack V_{rack}

$$V_{\text{rack}} = \left(\frac{\pi}{6}\right)\left(\frac{D_p n_p}{2}\right)$$

5. Find the time it takes to move the rack some distance

$$t = 60\left(\frac{s_{\text{rack}}}{V_{\text{rack}}}\right)$$

6. Find the number of revolutions required to move the rack that far

$$\theta_p = \left(\frac{6}{\pi}\right)\left(\frac{2s_{\text{rack}}}{D_p}\right)$$

2.6 Gear Trains

$$\text{train value: } TV = (VR)_1(VR)_2\cdots = \frac{N_{\text{output}}}{N_{\text{input}}} = \frac{n_{\text{input}}}{n_{\text{output}}}$$

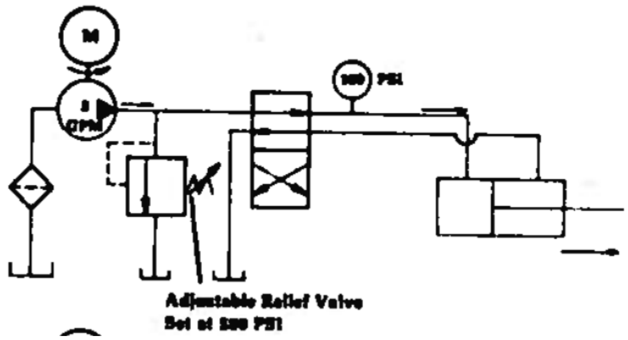
3 Fucking Fluids

3.1 Regenerative vs Non-Regenerative Circuits

There may be a question about the suitability of regenerative vs non regenerative circuits. They have their own advantages and disadvantages.

Summary: A regen circuit allows for more control over the ratio of force and speed for push and pull cycles. A non-regen circuit will always have a weaker/faster retraction cycle, which is ideal for application where pushing is the desired goal (hydraulic press). However if greater force is required for retraction, the regen circuit allows for balancing of push/pull forces, at the expense of greater complexity. see Section 3.2.3 for calculations.

Details: We examine a basic non-regenerative circuit first:



Usually the pump is at a fixed GPM flow rate and the system has a relief valve which regulates pressure to a constant value. The cycle above is in a state of pushing the piston and the fluid on the other side of the piston head provides no resistance to the pushing, so the full force of pressure times bore area is being applied. However when the piston retracts, it moves faster due to the rod occupying some of the bore volume on that side, with the same flow rate entering. So the result is a faster retraction at a lower force than the push cycle. No matter what bore and rod are selected there will be an imbalance in the cycles.

The regenerative circuit in contrast is able to produce equalized cycles:

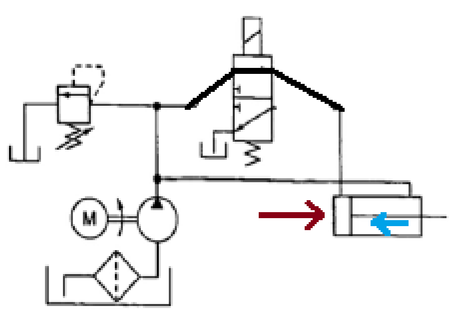


Figure 13-22 Regenerative circuit.

The push cycle is met by equal pressure on the other side. The force developed on the left in red is higher because the pressure area is larger, while the blue side is lower due to the rod. The net force developed is the pressure times the rod cross section area. The fluid will also recycle *regeneratively* through the system on the push cycle, allowing for more movement at a given volume of pump flow. On the pull cycle it operates the same as a non-regen circuit. The net effect is that the ratio between the rod area and the (bore - rod) areas determines the ratio of movement velocity and force developed. If the ratio is one, then both the push and retraction speed/force are the same.

3.2 Selecting Bores and Pumps and Shit

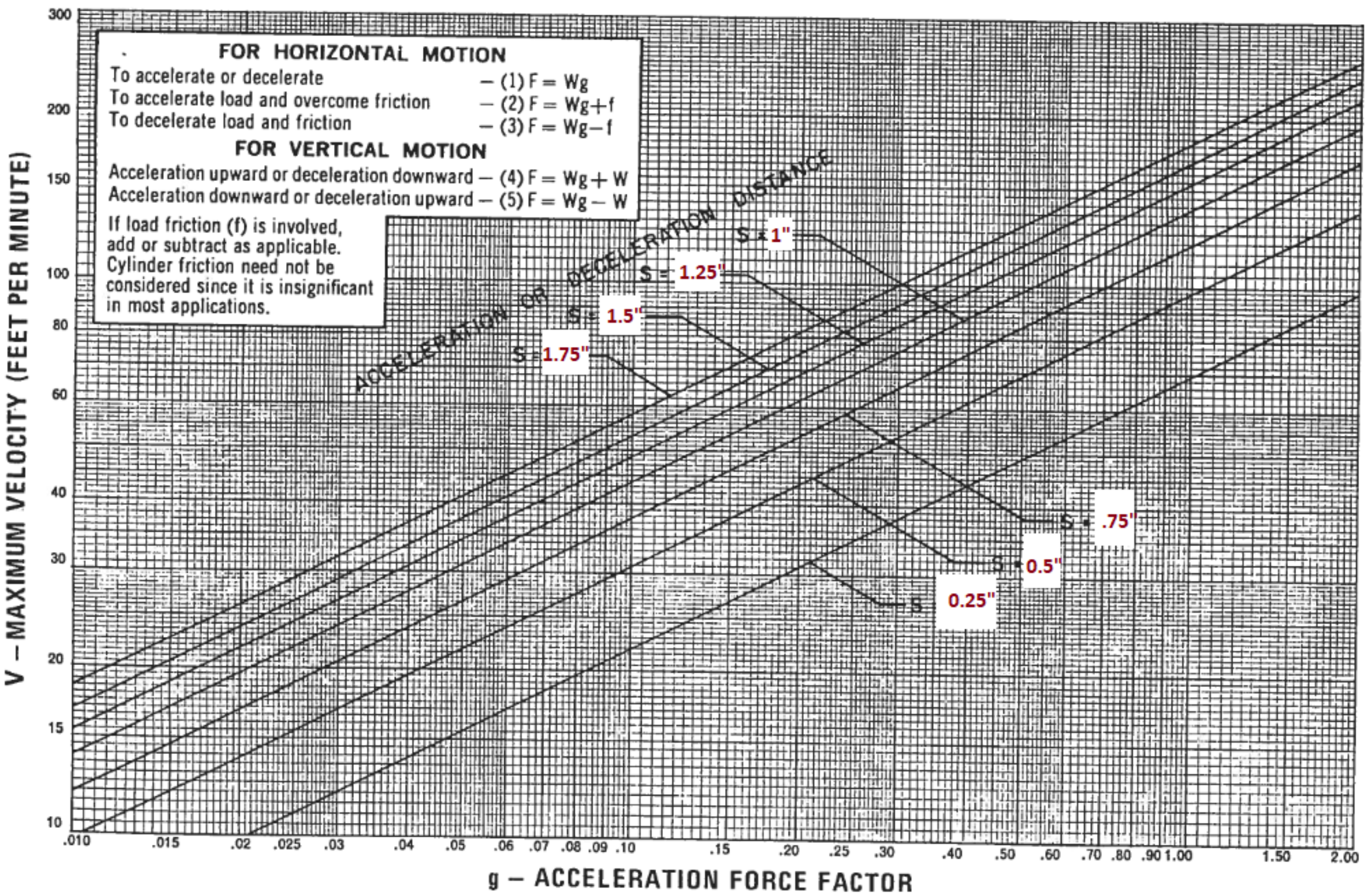
3.2.1 Nomenclature

Nomenclature is not very well defined so here is what I will use:
s = distance (in)
v = speed (ft/min)
F = (total) force (lb)
F_w = force of weight (lb)
F_a = force from acceleration (lb)
F_f = force due to friction (lb)
μ = coefficient of friction
g = acceleration constant (from table. Not the same g as gravity)
p = pressure (psi)
P = power (hp)
L = stroke length (in)
L_b = basic length (in)
k = stroke factor
Q = flow rate (gpm)

3.2.2 Minimum Rod Diameter

1. Find the force acting on the piston (rod? The thingy that goes back and forward).
   (a) Do some fucking free body analysis to find the relevant forces and velocities if you have a somewhat complicated system
   (b) Find the force due to the weight of the load
   (c) Find the force due to friction (if coefficient of friction is given, otherwise ignore that shit) Remember that the force will need to overcome friction to move so it is additive.
   (d) If you are given that the rod has to approach some velocity, v, in some distance, s then we use the worst shitty photocopy table you've ever seen, using s (diagonals) and v (y-axis) to get g (x-axis).
   (e) Compute g using the following equation or the table:

$$g = \left(\frac{v^2}{s}\right)(0.0000517) \text{ where } v \text{ is in ft/min and } s \text{ is in inches}$$



(f) Use g to compute the force due to the acceleration:

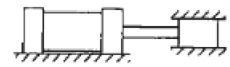
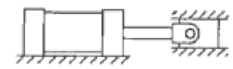
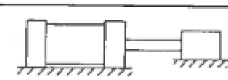
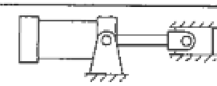
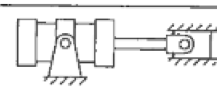
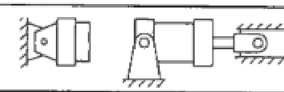
$$F_a = F_w g$$

(g) Sum the forces to get the total force (or thrust)

$$F = F_w + F_f + F_a$$

2. The question should tell/show you how your pump is mounted. Use that to determine the stroke factor, k from this table

piston rod — stroke selection table

RECOMMENDED MOUNTING STYLES FOR MAXIMUM STROKE AND THRUST LOADS	ROD END CONNECTION	CASE		STROKE FACTOR
CLASS 1 – GROUPS 1 OR 3 Long stroke cylinders for thrust loads should be mounted using a heavy-duty mounting style at one end, firmly fixed and aligned to take the principle force. Additional mounting should be specified at the opposite end, which should be used for alignment and support. An intermediate support may also be desirable for long stroke cylinders mounted horizontally. Machine mounting pads can be adjustable for support mountings to achieve proper alignment.	FIXED AND RIGIDLY GUIDED.	I		.50
	PIVOTED AND RIGIDLY GUIDED	II		.70
	SUPPORTED BUT NOT RIGIDLY GUIDED	III		2.00
CLASS 2 – GROUP 2 Style – Trunnion on Head	PIVOTED AND RIGIDLY GUIDED	IV		1.00
Style – Intermediate Trunnion	PIVOTED AND RIGIDLY GUIDED	V		1.50
Style – Trunnion on Cap or Style – Clevis on Cap	PIVOTED AND RIGIDLY GUIDED	VI		2.00

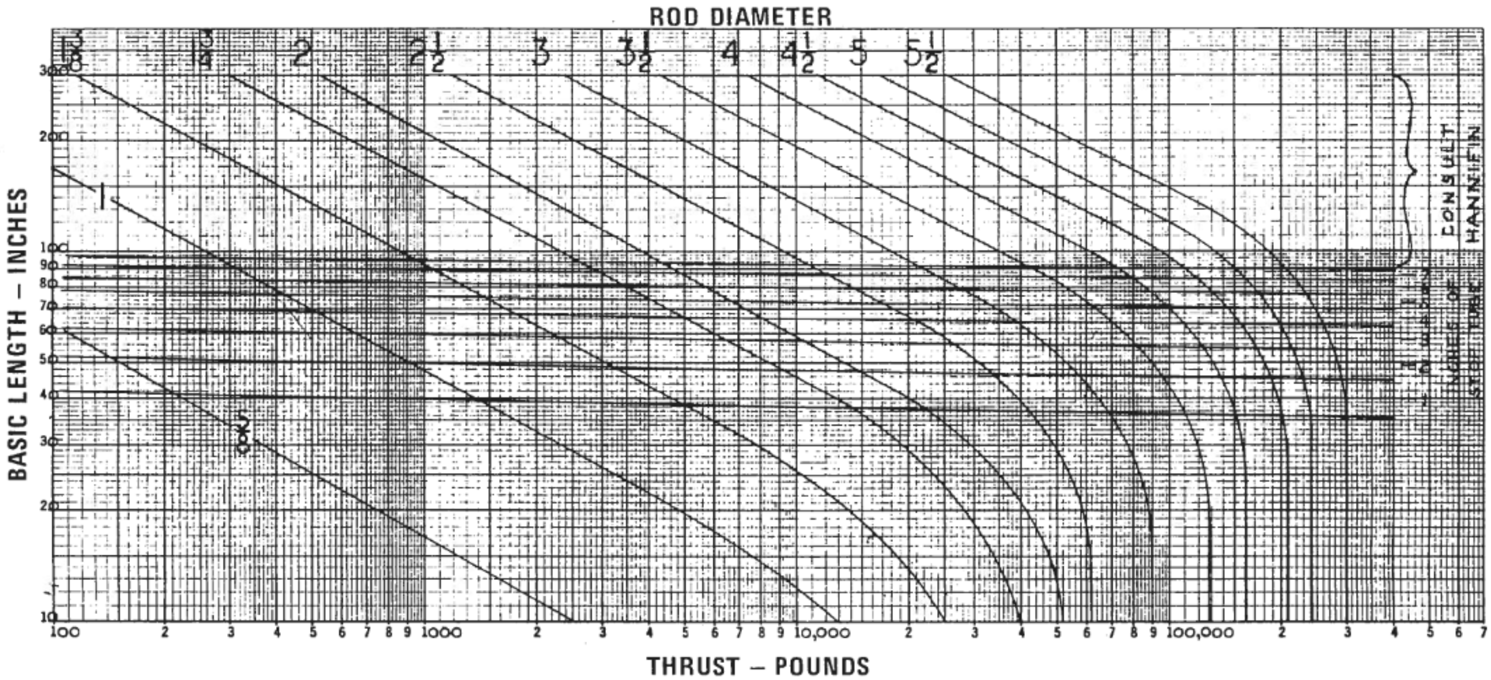
3. Compute the basic length, L_b, from the stroke length, L (specified in question) and the stroke factor, k

$$L_b = kL$$

4. Use the basic length and the total force (thrust) to choose the rod diameter. Find your point and round up to the nearest size. Because you may not be able to read this fucking awful table, the sizes are as follows:
$\frac{5}{8}$", 1", $1\frac{3}{8}$", $1\frac{3}{4}$", 2", $2\frac{1}{2}$", 3", $3\frac{1}{2}$", 4", $4\frac{1}{2}$", 5", $5\frac{1}{2}$"

   Note that this will give a minimum diameter, but the design requirements when it comes to balancing push and retraction or using regenerative circuits may require a larger bore diameter.

PISTON ROD — STROKE SELECTION GRAPH



3.2.3 Bore Size, Rod Size, Working Pressure

5. Use the pressure and the total force to choose a bore size from this table or calculate using formulas below. (Note that for pull applications, you must subtract force from the rod diameter)

theoretical push and pull forces for cylinders

CUSTOMARY U.S. UNITS

Cyl. Bore or Piston Rod. Dia. (in.)	Cyl. Bore Size (ϕ mm)	Area (sq. in.)	CYLINDER PUSH STROKE FORCE IN POUNDS AT VARIOUS PRESSURES (PSI)										Displacement per inch of Stroke (gallons)
			50	80	100	500	750	1000	1500	2000	2500	3000	
5/8	15.9	.307	15	25	31	154	230	307	461	614	768	921	.0013
1	25.4	.785	39	65	79	392	588	785	1,177	1,570	1,962	2,355	.0034
1-3/8	34.9	1.490	75	119	149	745	1,118	1,490	2,235	2,980	3,725	4,470	.0065
1-1/2	38.1	1.767	88	142	177	885	1,325	1,770	2,651	3,540	4,425	5,310	.00765
1-3/4	44.5	2.410	121	193	241	1,205	1,808	2,410	3,615	4,820	6,025	7,230	.0104
2	50.8	3.140	157	251	314	1,570	2,357	3,140	4,713	6,280	7,850	9,420	.0136
2-1/2	63.5	4.910	245	393	491	2,455	3,682	4,910	7,364	9,820	12,275	14,730	.0213
3	76.2	7.070	354	566	707	3,535	3,502	7,070	10,604	14,140	17,675	21,210	.0306
3-1/4	82.6	8.300	415	664	830	4,150	6,225	8,300	12,450	16,600	20,750	24,900	.0359
3-1/2	88.9	9.620	481	770	962	4,810	7,215	9,620	14,430	19,240	24,050	28,860	.0416
4	101.6	12.570	628	1,006	1,257	6,285	9,428	12,570	18,856	25,140	31,425	37,710	.0544
5	127.0	19.640	982	1,571	1,964	9,820	14,730	19,640	29,460	39,280	49,100	58,920	.0850
5-1/2	139.7	23.760	1,188	1,901	2,376	11,880	17,820	23,760	35,640	47,520	59,400	71,280	.1028
6	152.4	28.270	1,414	2,262	2,827	14,135	21,203	28,270	42,406	56,540	70,675	84,810	.1224
7	177.8	38.490	1,924	3,079	3,849	19,245	28,868	38,490	57,736	76,980	96,225	115,470	.1666
8	203.2	50.270	2,513	4,022	5,027	25,135	37,703	50,270	75,406	100,540	125,675	150,810	.2176
8-1/2	215.9	56.750	2,838	4,540	5,675	28,375	42,563	56,750	85,125	113,500	142,875	170,250	.2455
10	254.0	78.540	3,927	6,283	7,854	39,270	58,905	78,540	117,810	157,080	196,350	235,620	.3400
12	304.8	113.100	5,655	9,048	11,310	56,550	84,825	113,100	169,650	226,200	282,750	339,300	.4896

table b-1

NOTE: Deduct Force of Poston Rod Size from Bore Size for Pull Applications.

SI (METRIC) UNITS

Cyl. Bore or Piston Rod Dia. (in.)	Size in MM	Area in Sq. MM	CYLINDER PUSH FORCE IN NEWTONS AT VARIOUS PRESSURES IN BARS										Displacement for 1 MM of Stroke (Cu. MM)
			4	6.3	10	16	25	40	63	100	160	200	
5/8	15.87	197.9	79	125	198	317	495	792	1247	1979	3167	3959	197.9
1	25.40	506.7	203	319	507	811	1267	2027	3192	5067	8107	10134	506.7
1-3/8	34.93	958.0	383	604	958	1533	2395	3832	6035	9580	15328	19160	958.0
1-1/2	38.10	1140.1	456	718	1140	1824	2850	4560	7183	11401	18242	22802	1140.1
1-3/4	44.45	1551.8	621	978	1552	2483	3879	6207	9776	15518	24829	31036	1551.8
2	50.80	2026.9	811	1277	2027	3243	5067	8107	12769	20268	32429	40537	2026.9
2-1/2	63.50	3166.9	1267	1995	3167	5067	7917	12668	19952	31669	50671	63339	3166.9
3	76.20	4560.4	1824	2873	4560	7297	11401	18242	28730	45604	72966	91208	4560.4
3-1/4	82.55	5352.1	2141	3372	5352	8563	13380	21408	33718	53521	85634	107042	5352.1
3-1/2	88.90	6207.2	2483	3911	6207	9931	15518	24829	39105	62072	99315	124144	6207.2
4	101.60	8107.3	3243	5108	8107	12972	20268	32429	51076	81073	129717	162147	8107.3
5	127.00	12667.7	5067	7981	12668	20268	31669	50671	79807	126677	202683	253354	12667.7
5-1/2	139.70	15327.9	6131	9657	15328	24525	38320	61312	96566	153279	245247	306559	15327.9
6	152.40	18241.5	7297	11492	18242	29186	45604	72966	114922	182415	291864	364830	18241.5
7	177.80	24828.1	9931	15642	24829	39726	62072	99315	156421	248287	397260	496574	24828.7
8	203.20	32429.4	12972	20430	32429	51887	81073	129717	204305	324294	518870	648587	32429.4
8-1/2	215.90	36609.7	14644	23064	36610	58576	91524	146439	230641	366097	585755	732194	36609.7
10	254.00	50670.9	20268	31923	50671	81073	126677	202683	319226	506709	810734	1013417	50670.9
12	304.80	72966.0	29186	45968	72966	116746	182415	291864	459686	729660	1167457	1459321	72966.0

table b-2

REF. $1\ \#_f = 4.448$ NEWTONS (N)
1 BAR = 14.504 PSI

Alternatively, you can apply the principle of $F = PA$ to get rod and bore diameter to force relations.

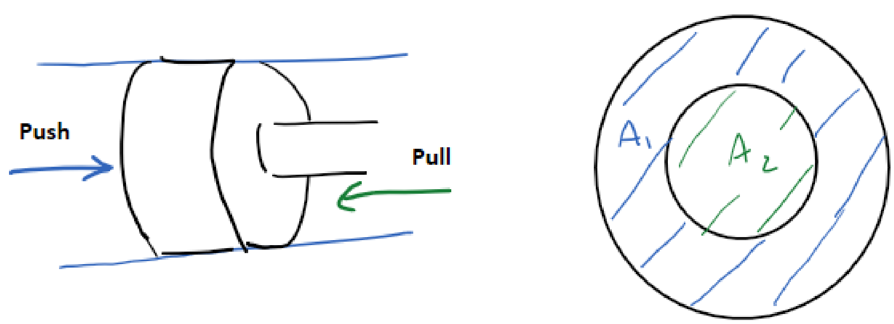


Figure 1: Piston Area and Motion

From the diagram above we can relate the areas shown to the forces developed, depending on if the circuit is *regenerative* or not. The advantage to a regenerative circuit is that both the push and pull forces can be made equal by making $A_1 = A_2$

Solve for any push pull force ratio constraints using standardized rod and bore sizes from the tables, making sure to keep a minimum rod size larger than the buckling requirement from earlier. The relation between force in lbs and pressure in psi is given below, use it to size parts or determine a working pressure required.

$$A_1 = \frac{\pi}{4}(d_{bore}^2 - d_{rod}^2)$$
$$A_2 = \frac{\pi}{4}d_{rod}^2$$

	Regular	Regenerative
Push	$F = P \cdot (A_1 + A_2)$	$F = P \cdot (A_2)$
Pull	$F = P \cdot (A_1)$	$F = P \cdot (A_1)$

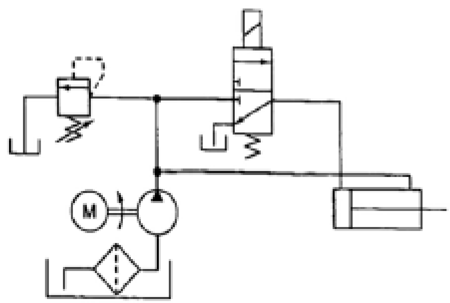


Figure 13-22 Regenerative circuit.

If using a regular push piston with known pressure, the required bore diameter can be found:

$$d_{bore} = \sqrt{\frac{4F}{\pi P}}$$

and then round the value up to the nearest standard size.
You can then compute the rated force using the diameter you chose;

$$F = \frac{\pi}{4}Pd_{bore}^2$$

3.2.4 Volume Flow Rate

6. Convert v to ft/min if not already
7. Compute the pump capacity, Q in gpm using v in ft/min and A in in^2.

$$Q_p = v_pA_p\left(\frac{12}{231}\right) = \frac{\pi}{4}v_pd_p^2\left(\frac{12}{231}\right)$$

	Regular	Regenerative
Push	$Q = \frac{\pi}{4} v_{push} \mathbf{d_{bore}}^2 \left(\frac{12}{231}\right)$	$Q = \frac{\pi}{4} v_{pull} (\mathbf{d_{rod}}^2) \left(\frac{12}{231}\right)$
Pull	$Q = \frac{\pi}{4} v_{push} (\mathbf{d^2_{bore}} - \mathbf{d^2_{rod}}) \left(\frac{12}{231}\right)$	$Q = \frac{\pi}{4} v_{push} (\mathbf{d^2_{bore}} - \mathbf{d^2_{rod}}) \left(\frac{12}{231}\right)$

3.2.5 Pipe Sizing

8. Compute Pipe Diameter

   There are two methods to select pipe sizing. We always use a maximum pipeflow velocity of 15 ft/s so we should find minimum pipe size for this constraint:

   $$d_p = \sqrt{\frac{4Q}{\pi v}\left(\frac{231}{720}\right)}$$

   where $v = 15$ft/s and Q is in GPM. Round this value up to the nearest standard size from the table to select a pipe size.

9. Method 1: Using the table from Fluid Power Basics: The working pressure determines the schedule, 40 - Low, 80 - Medium, 160 - High. Just assume Schedule 80 saying that proper schedule is dependent on working pressure and material selection.

   Get the pipe diameter in the blue box under Schedule 80 that is larger than the minimum calculated in the last step. The nominal pipe size and Schedule can be specified for the pipe.

HYDRAULIC PIPE SPECIFICATIONS (Dimensions)

Nominal Size	Pipe OD	INSIDE DIAMETER Schedule 40	Schedule 80	Schedule 160
⅛	.405	.269	.215	
¼	.540	.364	.302	
⅜	.675	.493	.423	
½	.840	.622	.546	.466
¾	1.050	.824	.742	.587
1	1.315	1.049	.957	.815
1¼	1.660	1.380	1.278	1.160
1½	1.900	1.610	1.500	1.338
2	2.375	2.067	1.939	1.689
2½	2.875	2.469	2.323	2.125
3	3.500	3.068	2.900	2.624

10. Method 2: Alternatively try selecting a pipe size from this table:

CYLINDER BORE-INCHES	PISTON ROD DIA.-INCHES	PISTON ROD AREA SQ. IN.	CYLINDER NET AREA SQ. IN.	FLUID DISPLACEMENT AT 10 FT. PER MINUTE PISTON VELOCITY G.P.M.	C.F.M.	FLUID VELOCITY (IN FEET PER SECOND) THROUGH EXTRA HEAVY PIPE AT 10 F.P.M. PISTON SPEED. 1/4	3/8	1/2	3/4	1	1-1/4	1-1/2	2	2-1/2
1	0	0	0.785	0.41	0.054	1.82	0.92	0.56	0.30	0.183	0.102	0.074	0.045	.....
	1/2	0.196	0.589	0.30	0.041	1.33	0.68	0.41	0.21	0.134	0.075	0.055	0.033	.....
	5/8	0.307	0.478	0.16	0.033	0.71	0.36	0.22	0.12	0.071	0.040	0.029	0.017	.....
1½	0	0	1.77	0.92	0.123	4.09	2.09	1.259	0.680	0.410	0.230	0.167	0.100	.....
	5/8	0.307	1.46	0.76	0.101	3.38	1.73	1.040	0.562	0.338	0.190	0.138	0.082	.....
	1	0.785	0.98	0.51	0.068	2.27	1.16	0.699	0.378	0.228	0.128	0.093	0.055	.....
2	0	0	3.14	1.63	0.218	7.27	3.71	2.238	1.209	0.728	0.408	0.296	0.177	.....
	5/8	0.307	2.84	1.48	0.197	6.56	3.35	2.019	1.091	0.657	0.368	0.267	0.160	.....
	1	0.785	2.36	1.23	0.164	5.45	2.79	1.678	0.907	0.546	0.306	0.222	0.133	.....
	1-3/8	1.485	1.66	0.86	0.115	3.84	1.96	1.180	0.638	0.384	0.215	0.156	0.094	.....
2½	0	0	4.91	2.55	0.341	11.36	5.80	3.496	1.890	1.138	0.638	0.463	0.277	.....
	5/8	0.307	4.60	2.39	0.319	10.65	5.44	3.278	1.771	1.067	0.598	0.434	0.260	.....
	1	0.785	4.12	2.14	0.286	9.54	4.87	2.937	1.587	0.956	0.536	0.389	0.233	.....
	1-3/8	1.485	3.42	1.78	0.237	7.93	4.05	2.439	1.318	0.794	0.445	0.323	0.193	.....
	1-3/4	2.405	2.50	1.30	0.174	5.96	2.96	1.783	0.963	0.580	0.325	0.236	0.141	.....
3¼	0	0	8.30	4.31	0.576	19.20	9.81	5.909	3.193	1.923	1.078	0.783	0.468	.....
	1	0.785	7.51	3.90	0.521	17.38	8.88	5.349	2.891	1.741	0.976	0.708	0.424	.....
	1-3/8	1.485	6.81	3.54	0.473	15.77	8.05	4.851	2.622	1.579	0.885	0.642	0.384	.....
	1-3/4	2.405	5.89	3.06	0.409	13.64	6.96	4.196	2.268	1.366	0.765	0.556	0.333	.....
	2	3.142	5.15	2.68	0.357	11.93	6.09	3.671	1.984	1.195	0.670	0.486	0.291	.....
4	0	0	12.57	6.53	0.872	29.09	14.85	8.95	4.84	2.91	1.63	1.19	0.709	.....
	1	0.785	11.78	6.12	0.818	27.27	13.93	8.39	4.54	2.73	1.53	1.11	0.665	.....
	1-3/8	1.485	11.08	5.76	0.769	25.65	13.10	7.89	4.27	2.57	1.44	1.05	0.625	.....
	1-3/4	2.405	10.16	5.28	0.705	23.52	12.01	7.24	3.91	2.36	1.32	0.96	0.574	.....
	2	3.142	9.42	4.89	0.654	21.82	11.14	6.71	3.63	2.19	1.22	0.89	0.532	.....
	2-1/2	4.909	7.66	3.98	0.532	17.73	9.05	5.45	2.95	1.78	1.00	0.72	0.432	.....
5	0	0	19.64	10.20	1.363	45.45	23.21	13.99	7.56	4.55	2.55	1.85	1.108	.....
	1	0.785	18.85	9.79	1.308	43.64	22.28	13.43	7.26	4.37	2.45	1.78	1.064	.....
	1-3/8	1.485	18.15	9.43	1.260	42.01	21.45	12.93	6.99	4.21	2.36	1.71	1.024	.....
	1-3/4	2.405	17.23	8.95	1.196	39.88	20.37	12.27	6.63	3.99	2.24	1.63	0.973	.....
	2	3.142	16.49	8.57	1.144	38.18	19.50	11.75	6.35	3.82	2.14	1.56	0.931	.....
	2-1/2	4.909	14.73	7.65	1.022	34.09	17.41	10.49	5.67	3.41	1.91	1.39	0.831	.....
	3	7.069	12.57	6.53	0.872	29.09	14.85	8.95	4.84	2.91	1.63	1.19	0.709	.....
	3-1/2	9.621	10.01	5.21	0.695	23.18	11.84	7.13	3.86	2.32	1.30	0.95	0.565	.....
6	0	0	28.27	14.69	1.962	65.45	33.42	20.14	10.88	6.55	3.67	2.67	1.596	.....
	1-3/8	1.485	26.79	13.92	1.859	62.01	31.67	19.08	10.31	6.21	3.48	2.53	1.512	.....
	1-3/4	2.405	25.87	13.44	1.795	59.88	30.58	18.43	9.96	5.60	3.36	2.44	1.460	.....
	2	3.142	25.13	13.06	1.744	58.18	29.71	17.90	9.67	5.83	3.27	2.37	1.418	.....
	2-1/2	4.909	23.37	12.14	1.622	54.1	27.6	16.64	8.99	5.42	3.04	2.20	1.32	.....
	3	7.069	21.21	11.02	1.472	49.1	25.1	15.10	8.16	4.92	2.76	2.00	1.20	.....
	3-1/2	9.621	18.65	9.69	1.294	43.2	22.1	13.29	7.18	4.32	2.42	1.76	1.05	.....
	4	12.566	15.71	8.16	1.090	36.4	18.6	11.19	6.05	3.64	2.04	1.48	0.89	.....

	0	0	38.49	20.00	2.671	89.1	45.5	27.41	14.81	8.92	5.00	3.63	2.17	.....
	1-3/8	1.485	37.00	19.22	2.568	85.7	43.7	26.35	14.24	8.58	4.81	3.49	2.09	.....
	1-3/4	2.405	36.08	18.74	2.504	83.5	42.7	25.70	13.89	8.36	4.69	3.40	2.04	.....
	2	3.142	35.34	18.36	2.453	81.8	41.8	25.17	13.60	8.19	4.59	3.33	2.00	.....
	2-1/2	4.909	33.58	17.44	2.330	77.7	39.7	23.92	12.92	7.78	4.36	3.17	1.90	.....
7	3	7.069	31.42	16.32	2.181	72.7	37.1	22.38	12.09	7.28	4.08	2.96	1.77	.....
	3-1/2	9.621	28.86	14.99	2.003	66.8	34.1	20.56	11.11	6.69	3.75	2.72	1.63	.....
	4	12.566	25.92	13.47	1.799	60.0	30.6	18.46	9.98	6.01	3.37	2.45	1.46	.....
	4-1/2	15.904	22.58	11.73	1.567	52.3	26.7	16.08	8.69	5.23	2.93	2.12	1.28	.....
	5	19.635	18.85	9.79	1.308	43.6	22.3	13.43	7.26	4.37	2.45	1.78	1.06	.....
	0	0	50.27	26.12	3.489	116.4	59.4	35.80	19.35	11.65	6.53	4.74	2.84	1.977
	1-3/8	1.485	48.78	25.34	3.385	112.9	57.7	34.74	18.78	11.31	6.34	4.60	2.75	1.918
	1-3/4	2.405	47.86	24.86	3.321	110.8	56.6	34.09	18.42	11.09	6.22	4.51	2.70	1.882
	2	3.142	47.12	24.48	3.270	109.1	55.7	33.56	18.14	10.92	6.12	4.45	2.66	1.853
	2-1/2	4.909	45.36	23.57	3.149	105.0	53.61	32.31	17.46	10.51	5.892	4.278	2.560	1.784
8	3	7.069	43.20	22.44	2.998	100.0	51.06	30.77	16.63	10.01	5.612	4.074	2.438	1.699
	3-1/2	9.621	40.65	21.12	2.821	94.1	48.04	28.95	15.65	9.42	5.279	3.834	2.294	1.598
	4	12.566	37.70	19.59	2.616	87.3	44.56	26.85	14.51	8.74	4.897	3.556	2.128	1.483
	4-1/2	15.904	34.36	17.85	2.385	79.5	40.62	24.47	13.23	8.20	4.464	3.241	1.939	1.351
	5	19.635	30.63	15.91	2.126	70.9	36.21	21.82	11.79	7.10	3.979	2.889	1.729	1.205
	5-1/2	23.758	26.51	13.77	1.840	61.4	31.33	18.88	10.20	6.15	3.444	2.500	1.496	1.043
	0	0	78.54	40.80	5.451	181.8	92.84	55.94	30.23	18.21	10.203	7.408	4.433	3.089
	1-3/4	2.405	76.14	39.56	5.284	176.2	89.99	54.23	29.31	17.65	9.890	7.181	4.297	2.994
	2	3.142	75.40	39.17	5.233	174.5	89.12	53.70	29.02	17.48	9.795	7.112	4.255	2.965
	2-1/2	4.909	73.63	38.25	5.110	170.4	87.03	52.44	28.34	17.07	9.565	6.945	4.156	2.896
	3	7.069	71.47	37.13	4.960	165.4	84.48	50.91	27.51	16.57	9.284	6.741	4.034	2.811
	3-1/2	9.621	68.92	35.80	4.783	159.5	81.47	49.09	26.53	15.98	8.953	6.501	3.890	2.710
10	4	12.566	65.97	34.27	4.578	152.7	77.98	46.99	25.39	15.29	8.570	6.223	3.724	2.595
	4-1/2	15.904	62.64	32.54	4.347	145.0	74.04	44.61	24.11	14.52	8.137	5.908	3.535	2.463
	5	19.635	58.91	30.60	4.088	136.4	69.63	41.96	22.67	13.65	7.652	5.556	3.325	2.317
	5-1/2	23.758	54.78	28.46	3.802	126.8	64.75	39.02	21.09	12.70	7.116	5.167	3.092	2.154
	6	28.274	50.27	26.12	3.489	116.4	59.42	35.80	19.35	11.65	6.530	4.741	2.837	1.977
	6-1/2	33.183	45.36	23.57	3.148	105.0	53.6	32.31	17.46	10.52	5.89	4.278	2.560	1.784
	7	38.485	40.06	20.81	2.780	92.7	47.4	28.53	15.42	9.29	5.20	3.778	2.261	1.575
	0	0	113.10	58.76	7.849	261.8	133.7	80.55	43.53	26.22	14.69	10.668	6.383	4.448
	2	3.142	109.96	57.12	7.631	254.5	130.0	78.32	42.32	25.49	14.28	10.371	6.206	4.324
	2-1/2	4.909	108.19	56.21	7.508	250.4	127.9	77.06	41.64	25.08	14.05	10.205	6.106	4.255
	3	7.069	106.03	55.08	7.359	245.4	125.3	75.52	40.81	24.58	13.77	10.001	5.984	4.170
	3-1/2	9.621	103.48	53.76	7.182	239.5	122.3	73.70	39.83	23.99	13.44	9.760	5.840	4.069
	4	12.566	100.53	52.23	6.977	232.7	118.8	71.60	38.70	23.30	13.06	9.482	5.674	3.954
	4-1/2	15.904	97.19	50.49	6.745	225.0	114.9	69.23	37.41	22.53	12.63	9.168	5.486	3.822
12	5	19.635	93.46	48.55	6.486	216.4	110.5	66.57	35.98	21.67	12.14	8.816	5.275	3.676
	5-1/2	23.758	89.34	46.41	6.200	206.8	105.6	63.63	34.39	20.71	11.61	8.427	5.042	3.513
	6	28.274	84.82	44.06	5.887	196.4	100.3	60.42	32.65	19.66	11.02	8.001	4.787	3.336
	6-1/2	33.183	79.92	41.52	5.547	185.0	94.5	56.92	30.76	18.53	10.38	7.538	4.510	3.143
	7	38.485	74.61	38.77	5.179	172.7	88.2	53.14	28.72	17.30	9.69	7.038	4.211	2.934
	7-1/2	44.179	68.92	35.80	4.783	159.5	81.5	49.09	26.53	15.98	8.95	6.501	3.890	2.710
	8	50.266	62.83	32.64	4.360	145.4	74.3	44.75	24.19	14.57	8.16	5.926	3.546	2.471
	8-1/2	56.745	56.35	29.27	3.911	130.5	66.6	40.14	21.69	13.06	7.32	5.315	3.181	2.216
	0	0	153.94	79.97	10.683	356.3	182.0	109.6	59.25	35.68	20.00	14.52	8.688	6.054
	2-1/2	4.909	149.03	77.42	10.343	345.0	176.2	106.2	57.36	34.55	19.36	14.06	8.411	5.861
	3	7.069	146.87	76.30	10.193	340.0	173.6	104.6	56.53	34.05	19.08	13.85	8.289	5.776
14	3-1/2	9.621	144.32	74.97	10.016	334.1	170.6	102.8	55.55	33.45	18.75	13.61	8.145	5.676
	4	12.566	141.37	73.44	9.811	327.3	167.1	100.7	54.42	32.77	18.37	13.33	7.979	5.560
	4-1/2	15.904	138.03	71.71	9.579	319.5	163.2	98.3	53.13	32.00	17.93	13.02	7.791	5.428
	5	19.635	134.30	69.77	9.320	310.9	158.8	95.7	51.70	31.13	17.45	12.67	7.580	5.282
	5-1/2	23.758	130.18	67.63	9.035	301.3	153.9	92.7	50.11	30.18	16.91	12.28	7.347	5.120

3.2.6 Horsepower for Motor

11. Compute the required horsepower. Or use a fucking table. Same difference

$$HP = \frac{Q \cdot P}{1714 \cdot 0.85}$$

Note that Q is the flow rate of the stroke, P is the pressure, 0.85 is the assumed efficiency, and 1714 is a conversion factor between GPM and HP.

ELECTRIC MOTOR HORSEPOWER REQUIRED TO DRIVE A HYDRAULIC PUMP

GPM	100 PSI	200 PSI	250 PSI	300 PSI	400 PSI	500 PSI	750 PSI	1000 PSI	1250 PSI	1500 PSI	2000 PSI	2500 PSI	3000 PSI
1/2	.04	.07	.09	.11	.14	.18	.26	.35	.44	.53	.70	.88	1.10
1	.07	.14	.18	.21	.28	.35	.52	.70	.88	1.05	1.40	1.76	1.92
1-1/2	.10	.21	.26	.31	.41	.52	.77	1.03	1.29	1.55	2.06	2.58	3.09
2	.14	.28	.35	.42	.56	.70	1.04	1.40	1.76	2.10	2.80	3.53	4.20
2-1/2	.17	.34	.43	.51	.69	.86	1.29	1.72	2.15	2.58	3.44	4.30	5.14
3	.21	.42	.53	.63	.84	1.05	1.56	2.10	2.64	3.15	4.20	5.28	6.30
3-1/2	.24	.48	.60	.72	.96	1.20	1.80	2.40	3.00	3.60	4.80	6.00	7.20
4	.28	.56	.70	.84	1.12	1.40	2.08	2.80	3.52	4.20	5.60	7.04	8.40
5	.35	.70	.88	1.05	1.40	1.75	2.60	3.50	4.40	5.25	7.00	8.80	10.50
6	.42	.84	1.05	1.26	1.68	2.10	3.12	4.20	5.28	6.30	8.40	10.56	12.60
7	.49	.98	1.23	1.47	1.96	2.45	3.64	4.90	6.16	7.35	9.80	12.32	14.70
8	.56	1.12	1.40	1.68	2.24	2.80	4.16	5.60	7.04	8.40	11.20	14.08	16.80
9	.62	1.24	1.55	1.86	2.48	3.10	4.65	6.18	7.73	9.28	12.40	15.56	18.58
10	.70	1.40	1.75	2.10	2.80	3.50	5.20	7.00	8.80	10.50	14.00	17.60	21.00
11	.77	1.54	1.93	2.31	3.08	3.85	5.72	7.70	9.68	11.50	15.40	19.36	23.10
12	.84	1.68	2.10	2.52	3.36	4.20	6.24	8.40	10.50	12.60	16.80	21.00	25.20
13	.89	1.78	2.23	2.67	3.56	4.45	6.68	8.92	11.20	13.40	17.80	22.40	26.72
14	.96	1.92	2.40	2.88	3.84	4.80	7.20	9.60	12.00	14.40	19.20	24.00	28.80
15	1.05	2.10	2.63	3.15	4.20	5.25	7.80	10.50	13.20	15.70	21.00	26.40	31.50
16	1.10	2.20	2.75	3.30	4.40	5.50	8.25	11.00	13.80	16.50	22.00	27.60	33.00
17	1.17	2.34	2.93	3.51	4.68	5.85	8.78	11.70	14.60	17.60	23.40	29.20	35.10
18	1.26	2.52	3.15	3.78	5.04	6.30	9.35	12.60	15.80	18.90	25.20	31.60	37.80
19	1.30	2.60	3.25	3.90	5.20	6.50	9.75	13.00	16.30	19.50	26.00	32.60	39.00
20	1.40	2.80	3.50	4.20	5.60	7.00	10.40	14.00	17.60	21.00	28.00	35.20	42.00
25	1.75	3.50	4.38	5.25	7.00	8.75	13.10	17.50	21.90	26.20	35.00	43.80	52.50
30	2.10	4.20	5.25	6.30	8.40	10.50	15.60	21.00	26.40	31.50	42.00	52.80	63.00
35	2.45	4.90	6.13	7.35	9.80	12.20	18.40	24.50	30.60	36.70	49.00	61.20	73.50
40	2.80	5.60	7.00	8.40	11.20	14.00	20.80	28.00	35.20	42.00	56.00	70.40	84.00
45	3.15	6.30	7.87	9.45	12.60	15.80	23.60	31.50	39.40	47.30	63.00	78.80	94.50
50	3.50	7.00	8.75	10.50	14.00	17.50	26.00	35.00	44.00	52.50	70.00	88.00	105.00
55	3.85	7.70	9.63	11.60	15.40	19.30	28.60	38.50	48.40	57.80	77.00	96.80	115.50
60	4.20	8.40	10.50	12.60	16.80	21.00	31.20	42.00	52.80	63.00	84.00	105.60	126.00
65	4.55	9.10	11.40	13.60	18.20	22.80	33.80	45.50	57.20	68.20	90.00	114.40	136.50

12. Based on your required horsepower, choose a motor to use. Three phase power is for industrial applications and would be available in that setting.

motor starter, conduit and wire size

3 PHASE MOTOR STARTERS

1/2 TO 20 H.P. — MOTOR H.P. 3 ø	1/2		3/4		1		1-1/2		2		3		5		7-1/2		10		15		20	
VOLTAGE	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440
Nema Starter Size	00	00	00	00	00	00	00	00	0	00	0	0	1	0	1	1	2	1	2	2	3	2
⊕ Full Load Current	2.0	1.0	2.8	1.4	3.5	1.8	5.0	2.5	6.5	3.3	9.0	4.5	15	7.5	22	11	28	14	40	20	52	26
Fuses – Amps Std. N.E.C.	15	15	15	15	15	15	15	15	20	15	25	15	40	20	60	30	70	35	100	50	150	70
Fuses – Amps Dual Element	5	4	5	4	8	4	8	5	12	8	15	10	25	15	30	20	45	25	60	30	80	40
Circuit Breaker Max. Amps.	15	15	15	15	15	15	15	15	15	15	20	15	30	15	50	20	50	30	70	40	100	50
Minimum Wire Sizes R, RW, T, TW	14	14	14	14	14	14	14	14	14	14	14	14	12	14	10	14	8	12	6	10	4	8
Minimum Wire Sizes RH	14	14	14	14	14	14	14	14	14	14	14	14	12	14	10	14	8·	12	6	10	6	8

Always specify voltage and frequency.

3 PHASE MOTOR STARTERS

25 TO 200 H.P. — MOTOR H.P. 3 ø	25		30		40		50		60		75		100		125		150		200	
VOLTAGE	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440
Nema Starter Size	3	2	3	3	4	3	4	3	5	4	5	4	5	4	6	5	6	5	6	5
⊕ Full Load Current	64	32	78	39	104	52	125	63	150	75	185	93	246	123	310	155	360	180	480	240
Fuses – Amps Std. N.E.C.	175	80	200	100	300	150	350	175	400	200	500	250	600	350	–	400	–	450	–	600
Fuses – Amps Dual Element	100	50	125	60	175	80	200	100	225	125	300	150	400	200	450	250	600	300	–	400
Circuit Breaker Max. Amps.	125	50	100	70	175	100	200	125	225	125	300	150	400	200	–	250	–	300	–	400
Minimum Wire Sizes R, RW, T, TW	3	8	1	6	00	4	000	3	0000	2	300	0	500	000	–	0000	–	300	–	500
Minimum Wire Sizes RH	4	8	3	6	1	6	00	4	000	3	0000	1	350	00	–	000	–	0000	–	350

Always specify voltage and frequency.

SINGLE PHASE MOTOR STARTERS

1/6 TO 5 H.P. — MOTOR H.P. 1 ø	1/6		1/4		1/3		1/2		3/4		1		1-1/2		2		3		5	
VOLTAGE	115	230	115	230	115	230	115	230	115	230	115	230	115	230	115	230	115	230	115	230
⊕ Full Load Current	4.4	2.2	5.8	2.9	7.2	3.6	9.8	4.9	13.8	6.9	16	8	20	10	24	12	34	17	56	28
Fuses – Amps. Std. N.E.C.	15	15	20	15	25	15	30	15	45	25	50	25	60	30	80	40	100	60	–	90
Circuit Breaker Max. Amps.	15	15	15	15	15	15	30	15	40	20	40	20	50	30	70	30	100	50	–	70
Min. Wire Sizes – R, RH, RW, T, TW	14	14	14	14	14	14	14	14	12	14	12	14	10	14	10	14	6	10	–	8

WIRE & CONDUIT SIZES

WIRE SIZE AWG or MCM	14	12	10	8	6	4	3	2	1	0	00	000	0000	250	300	350	400	500	750	1000
MAXIMUM 3 WIRES IN CONDUIT — Wire Capacity R-RW-T-TW Amps.	15	20	30	40	55	70	80	95	110	125	145	165	195	215	240	260	280	320	400	455
MAXIMUM 3 WIRES IN CONDUIT — Wire Capacity RH Amps.	15	20	30	45	65	85	100	115	130	150	175	200	230	255	285	310	335	380	475	545
MAXIMUM 3 WIRES IN CONDUIT — CONDUIT SIZE – Inches	1/2	1/2	3/4	3/4	1	1-1/4	1-1/4	1-1/4	1-1/2	2	2	2	2-1/2	2-1/2	2-1/2	3	3	3	3-1/2	4
Volts Drop Per Ampere Per 100 Ft. – 80% P.F. 1 Phase Volts	.4762	.3125	.1961	.1250	.0833	.0538	.0431	.0370	.0323	.0269	.0222	.0190	.0161	.0147	.0131	.0121	.0115	.0101	.0086	.0081
Volts Drop Per Ampere Per 100 Ft. – 80% P.F. 3 Phase Volts	.4167	.2632	.1677	.1087	.0714	.0463	.0379	.0323	.0278	.0231	.0196	.0163	.0139	.0128	.0114	.0106	.0091	.0088	.0066	.0061

Capacity of conductors in conduit based on room temperature of 30° C. (86° F.)

⊕ The full load currents shown are average values.

3.3 Circuit Analysis Techniques

3.3.1 Flowrate

The following are some handy fluid relations to use when computing flow rates and power:

$$Q = Av$$

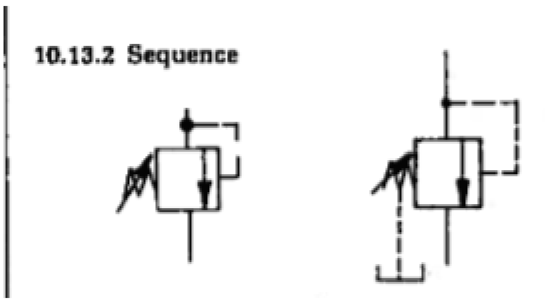
Use the area of the fluid column and the movement speed of the piston head to get Q (usually in in^3) then convert to GPM with $\frac{1\text{in}^3}{\text{s}} = \frac{60}{231}\text{GPM}$

3.3.2 Power

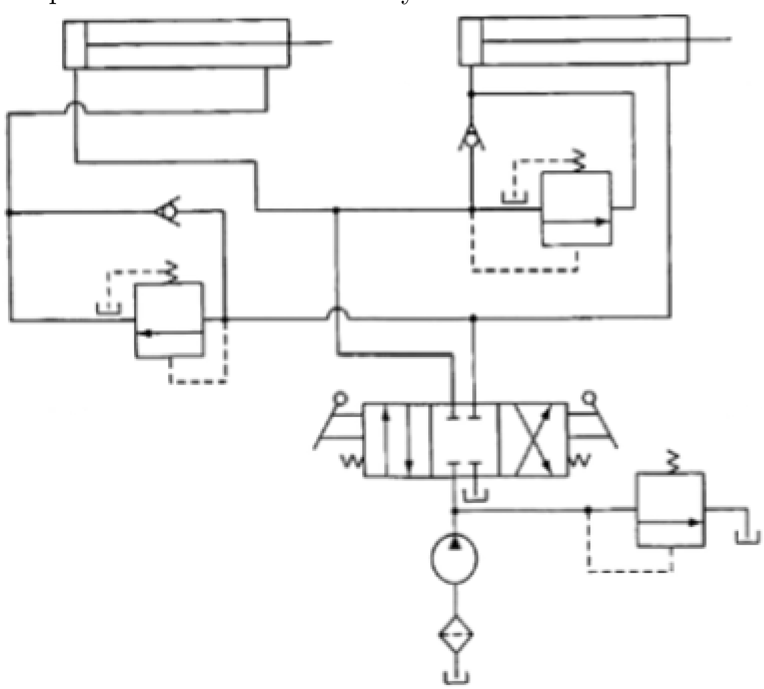
Work is pressure times volume, so power is the time derivative. Conversion factor for Q in GPM and P in psi included below.

$$\textbf{Power} = P\frac{dV}{dt} = \frac{PQ}{1714}$$

3.3.3 Other Components

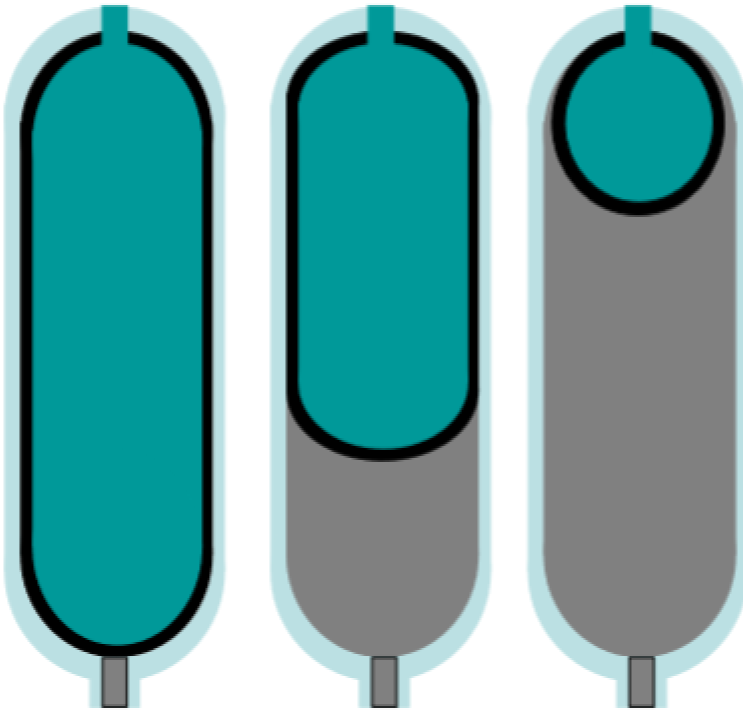


Sequence valves are circuit components which can control the order in which hydraulic elements are actuated. Sequences valves only allow fluid to flow through them once a minimum pressure has been reached to actuate them. This pressure is usually reached after a piston reaches maximum extension stops doing work, causing the pressure in the circuit to reach max pressure actuating the sequence valve. This allows cylinders to be turned on and off in a specific sequential order.



In this example of a circuit utilizing sequence valves, when the left side of the valve is used such that both lines are straight, the left cylinder extends then the right one as the path to extend the left cylinder is directly connected while the flow control valve blocks the right. Vice versa on the other side the right cylinder retracts then the left one.

Accumulators



Accumulators act as the hydraulic equivalent of capacitors. Using air pressure they can store pressure created by a pump to provide pressure during intermittent outages or to dampen shocks in a system (like how a capacitor can be used to filter signals). The calculations of it are beyond what we do in this course.

4 Bearings, Bushings, and Other Shit That Spins

4.1 Boundary-Lubricated Bearings (Bushings)

4.1.1 Anatomy

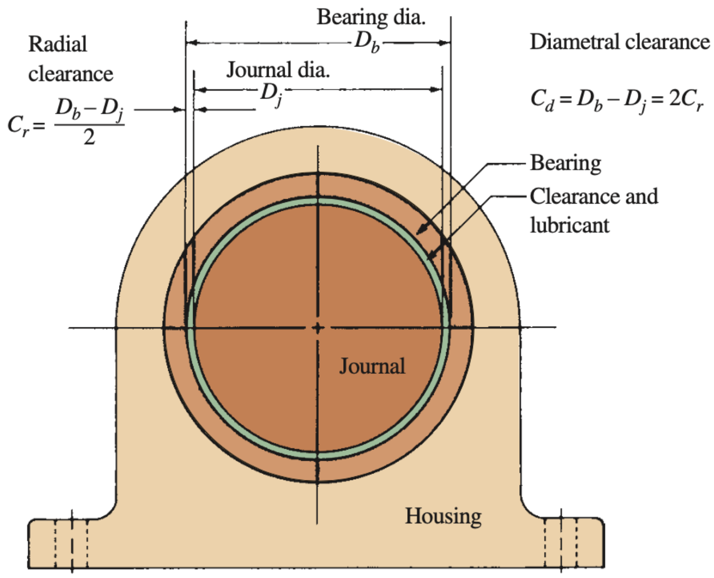


FIGURE 16–1 Bearing geometry

4.1.2 Design Selection

1. Get random factors from these tables based on your bearing type/material,

Table 12–9 Wear Factors in U.S. Customary Units*

Bushing Material	Wear Factor K	Limiting PV
Oiles 800	$3(10^{-10})$	18 000
Oiles 500	$0.6(10^{-10})$	46 700
Polyactal copolymer	$50(10^{-10})$	5 000
Polyactal homopolymer	$60(10^{-10})$	3 000
66 nylon	$200(10^{-10})$	2 000
66 nylon + 15% PTFE	$13(10^{-10})$	7 000
+ 15% PTFE + 30% glass	$16(10^{-10})$	10 000
+ 2.5% MoS_2	$200(10^{-10})$	2 000
6 nylon	$200(10^{-10})$	2 000
Polycarbonate + 15% PTFE	$75(10^{-10})$	7 000
Sintered bronze	$102(10^{-10})$	8 500
Phenol + 25% glass fiber	$8(10^{-10})$	11 500

*dim[K] = in^3 · min/(lbf · ft · h), dim [PV] = psi · ft/min.

Source: Data from Oiles America Corp., Plymouth, MI 48170.

Table 12–10 Coefficients of Friction

Type	Bearing	f_s
Placetic	Oiles 80	0.05
Composite	Drymet ST	0.03
	Toughmet	0.05
Met	Cermet M	0.05
	Oiles 2000	0.03
	Oiles 300	0.03
	Oiles 500SP	0.03

2. Calculate minimum bearing length L using the following equation:

$$L \geq \frac{720 f_s n_d F N}{J \hbar_{CR}(T_f - T_\infty)}$$

where f_s is coefficient of friction,
n_d is the design factor,
F is the radial load,
N is the angular speed of the bearing,
T_f is the lubricant temperature/max temperature
T_∞ is the ambient temperature

For some reason, we just randomly use $\hbar_{CR} = 2.7$ and $J = 778$ unless some other values are

given.
Also, use $T_\infty = 70$ if no value is given cause that's what they use in the textbook.

3. Choose a bushing from this table that exceeds the minimum length. Make sure the inner diameter (ID) is bigger than the shaft diameter if that is given.

Table 12–12 Available Bushing Sizes (in inches) of One Manufacturer*

		L													
ID	**OD**	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5
$\frac{1}{2}$	$\frac{3}{4}$	•	•	•	•	•									
$\frac{5}{8}$	$\frac{7}{8}$		•	•		•		•							
$\frac{3}{4}$	$1\frac{1}{8}$		•	•		•		•							
$\frac{7}{8}$	$1\frac{1}{4}$			•		•	•	•							
1	$1\frac{3}{8}$			•		•	•	•	•	•					
1	$1\frac{1}{2}$			•		•		•		•					
$1\frac{1}{4}$	$1\frac{5}{8}$					•	•	•	•	•					
$1\frac{1}{2}$	2					•	•	•	•	•					
$1\frac{3}{4}$	$2\frac{1}{4}$						•	•	•	•	•	•	•	•	
2	$2\frac{1}{2}$							•		•	•	•			
$2\frac{1}{4}$	$2\frac{3}{4}$							•		•	•	•			
$2\frac{1}{2}$	3							•		•		•			
$2\frac{3}{4}$	$3\frac{3}{8}$							•		•	•	•			
3	$3\frac{5}{8}$									•	•	•	•		
$3\frac{1}{2}$	$4\frac{1}{8}$									•		•		•	
4	$4\frac{3}{4}$									•		•		•	
$4\frac{1}{2}$	$5\frac{3}{8}$											•		•	•
5	6											•		•	•

*In a display such as this a manufacturer is likely to show catolog numbers where the • appears.

4. Make sure the bushing's length-to-inner diameter (ID) ratio falls within this range:

$$0.5 \geq L/D \geq 2$$

If it doesn't, choose another bushing that does.

5. To make sure your bearing is satisfactory, we will calculate random values and make sure this fall within the acceptable ranges (assuming your bushing is an Oiles 500 SP type):

Table 12–11 Oiles 500 SP (SPBN · SPWN) Service Range and Properties

Service Range	Units	Allowable
Characteristic pressure P_{max}	psi	<3560
Velocity V_{max}	ft/min	<100
PV product	(psi)(ft/min)	<46 700
Temperature T	°F	<300
Properties	**Test Method, Units**	**Value**
Tensile strength	(ASTM E8) psi	>110 000
Elongation	(ASTM E8) %	>12
Compressive strength	(ASTM E9) psi	49 770
Brinell hardness	(ASTM E10) HB	>210
Coefficient of thermal expansion	(10^{-5})°C	>1.6
Specific gravity		8.2

6. Calculate the characteristic pressure (in psi) and make sure it's in the right range:

$$P_{max} = \frac{4}{\pi}\frac{n_d F}{DL}$$

where F is the radial load in lbf,
n_d is the angular speed of the bearing in rpm,
$D = ID$ is the inner bearing diameter in inches,
L is the length of the bearing in inches

7. Calculate the nominal pressure (in psi):

$$P = \frac{n_d F}{DL}$$

8. Calculate the velocity (in ft/min) and make sure it is less than V_{max}:

$$V = \frac{\pi DN}{12}$$

9. Calculate PV by multiplying P and V. Make sure it's in the acceptable range.

10. If a maximum wear value was given in the question, calculate the linear wear w.

$$w = \frac{K n_d F N t}{3L}$$

where K is the wear factor,
t is the time in hours

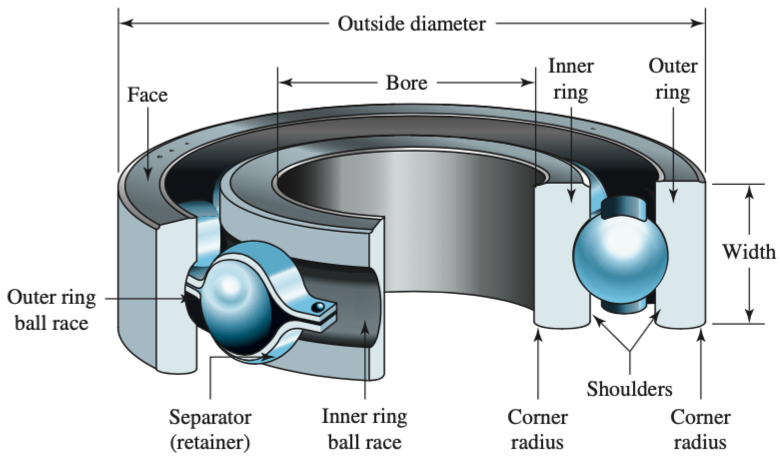
11. If all values are within acceptable ranges, congrats you have selected an acceptable bushing!

4.2 Ball and Cylindrical Roller Bearings

Types of bearings that fall under this category are:

- Deep-groove ball bearings
- Angular deep-groove ball bearings
- Cylindrical roller bearings

4.2.1 Anatomy



4.2.2 Design Selection - Radial Load Only

1. Find the angular speed n_D (in rpm) if it is not already given to you.
   (a) Find the transmitted torque from a free-body-diagram using $T = Fd$. It will be the torque that causes the shaft to rotate along its axis.
   (b) If the design horsepower is given, use the following equation:

$$n_D = \frac{63025H}{T} \text{ where } H \text{ is the design horsepower in hp,}$$
$$T \text{ is the torque transmitted in lbf} \cdot \text{in}$$

2. Find the radial load by taking the magnitude of the radial force components:

$$F_R = \sqrt{F_x^2 + F_y^2}$$

3. Calculate the bearing reliability of each individual bearing (if the total reliability of the ensemble is given):

$$R_i = \sqrt[n]{R_{tot}}$$
$$\text{where } R_i \text{ is the individual bearing reliability,}$$
$$R_{tot} \text{ is the total bearing reliability,}$$
$$n \text{ is the number of bearings in the assembly}$$

4. Get bearing design life $\mathscr{L}_D$ for your application from this table (if not given in the question):

Table 11–4 **Bearing-Life Recommendations for Various Classes of Machinery**

Type of Application	Life, kh
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines for 8-h service that are not always fully utilized	14–20
Machines for 8-h service that are fully utilized	20–30
Machines for continuous 24-h service	50–60
Machines for continuous 24-h service where reliability is of extreme importance	100–200

5. Calculate the multiple of rating life for each bearing in the assembly:

$$x_D = \frac{L_D}{L_{10}} = \frac{60\mathscr{L}_D n_D}{L_{10}}$$

where L_D is bearing design life in number of revolutions,
L_{10} is the rating life,
$\mathscr{L}_D$ is the design life in hours,
n_D is the angular speed of the bearing in rpm

The Weibull parameters used will depend on which manufacturer's bearings we are using.

- Timken (Manufacturer 1) is common for tapered roller bearings
- SKF (Manufacturer 2) is common for ball and straight roller bearings

		Weibull Parameters Rating Lives		
Manufacturer	**Rating Life, Revolutions**	X_0	θ	b
1	$90(10^6)$	0	4.48	1.5
2	$1(10^6)$	0.02	4.459	1.483

6. Get application factor from this table (if not given in the question):

Table 11–5 **Load-Application Factors**

Type of Application	Load Factor
Precision gearing	1.0–1.1
Commercial gearing	1.1–1.3
Applications with poor bearing seals	1.2
Machinery with no impact	1.0–1.2
Machinery with light impact	1.2–1.5
Machinery with moderate impact	1.5–3.0

7. Calculate the load rating C_{10} for the radial load:

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)[\ln(1/R_D)]^{1/b}} \right]^{1/a}$$

- $a = 3$ for ball bearings
- $a = 10/3$ for roller bearings (cylindrical and tapered)

8. Select a bearing from one of these tables that has a C_{10} value greater than the calculated one.

- For deep-groove and angular-contact ball bearings:

Table 11–2 **Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings**

Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm		Load Ratings, kN Deep Groove		Load Ratings, kN Angular Contact	
				d_S	d_H	C_{10}	C_0	C_{10}	C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

- For cylindrical roller bearings:

Table 11–3 **Dimensions and Basic Load Ratings for Cylindrical Roller Bearings**

	02-Series				03-Series			
Bore, mm	OD, mm	Width, mm	Load Rating, kN		OD, mm	Width, mm	Load Rating, kN	
			C_{10}	C_0			C_{10}	C_0
25	52	15	16.8	8.8	62	17	28.6	15.0
30	62	16	22.4	12.0	72	19	36.9	20.0
35	72	17	31.9	17.6	80	21	44.6	27.1
40	80	18	41.8	24.0	90	23	56.1	32.5
45	85	19	44.0	25.5	100	25	72.1	45.4
50	90	20	45.7	27.5	110	27	88.0	52.0
55	100	21	56.1	34.0	120	29	102	67.2
60	110	22	64.4	43.1	130	31	123	76.5
65	120	23	76.5	51.2	140	33	138	85.0
70	125	24	79.2	51.2	150	35	151	102
75	130	25	93.1	63.2	160	37	183	125
80	140	26	106	69.4	170	39	190	125
85	150	28	119	78.3	180	41	212	149
90	160	30	142	100	190	43	242	160
95	170	32	165	112	200	45	264	189
100	180	34	183	125	215	47	303	220
110	200	38	229	167	240	50	391	304
120	215	40	260	183	260	55	457	340
130	230	40	270	193	280	58	539	408
140	250	42	319	240	300	62	682	454
150	270	45	446	260	320	65	781	502

9. We're done!

4.2.3 Design Selection - Radial and Thrust Loads

1. Determine the rotation factor V:
   - $V = 1$ when the inner ring rotates
   - $V = 1.2$ when the outer ring rotates
2. If we do not yet have the C_0 and C_{10} value for the bearing, we will have to make assumptions:
   - Assume $F_a/(VF_r) > e$ (we will check if this is actually true later)
   - Choose random values for the X_2 and Y_2 factors. It's best to just choose middle values on the table, so let's go with $X_2 = 0.56$ and $Y_2 = 1.63$.

Table 11–1 Equivalent Radial Load Factors for Ball Bearings

		$F_a/(VF_r) \le e$		$F_a/(VF_r) > e$	
F_a/C_0	e	X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

3. Using our assumptions, calculate the equivalent load using the following equation:

$$F_e = X_i V F_r + Y_i F_a$$

where F_r is the radial load,

F_a is the axial/thrust load

4. Calculate the load rating C_{10} for the equivalent load:

$$C_{10} = a_f F_e \left[\frac{x_D}{x_0 + (\theta - x_0)[\ln(1/R_D)]^{1/b}} \right]^{1/a}$$

- $a = 3$ for ball bearings
- $a = 10/3$ for roller bearings (cylindrical and tapered)

5. Select a bearing that has a C_{10} value that exceeds the calculated one. Refer to tables 11-12 and 11-13 in **step 8** of the radial section right above.
6. Find the C_0 value for the chosen bearing.
7. Calculate F_a/C_0 where F_a is the axial load.
8. Look back at table 11-1 in **step 2** of this section and find the closest value of F_a/C_0. Get e from this.
9. Check if $F_a/(VF_r) > e$. If so, we will be looking for Y_2 values in table 11-1 from **step 2**. If not, we will be looking for Y_1 values.

10. Interpolate to find Y_2.
    - The two closest F_a/C_0 values in the table will be the x-points. The two associated Y_2 values will be the y-points.
    - Calculate the slope: $m = \frac{y_1 - y_2}{x_1 - x_2}$
    - Plug in a point into $y = mx + b$ and solve for b
    - Finally, plug in your value of F_a/C_0 as x to get Y_2 as y
11. Recalculate F_e with the new Y_2 value:

    $$F_e = X_i V F_r + Y_i F_a$$

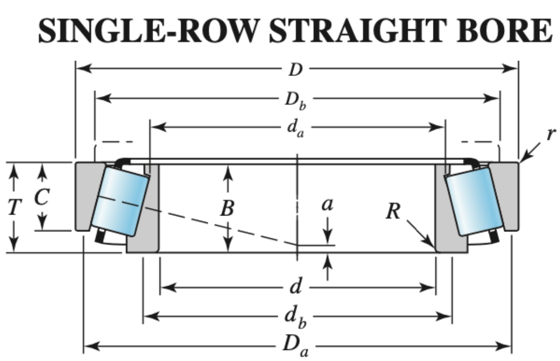
12. Calculate the new C_{10}. The calculation for C_{10} only changes in F_e so we can do the following calculation:

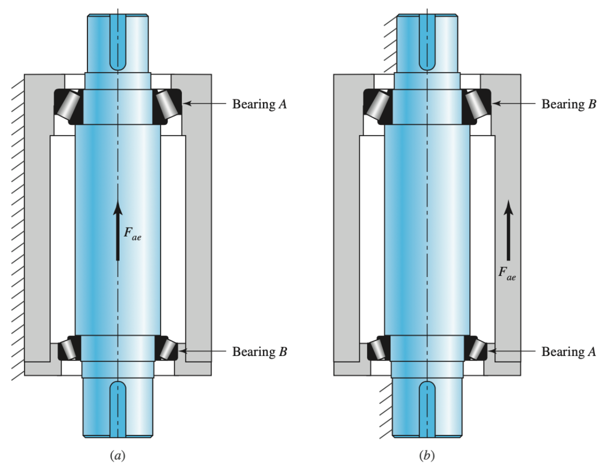
    $$\text{new } C_{10} = \frac{\text{new } F_e}{\text{old } F_e}(\text{old } C_{10})$$

13. Select a bearing that has a C_{10} value that exceeds the calculated one. Refer to tables 11-2 or 11-3 from **step 8** of the previous section.
14. Repeat **steps 6 to 13** until you select the same bearing from the table twice in a row.

4.3 Tapered Roller Bearings

4.3.1 Anatomy





4.3.2 Design Selection

1. Calculate the radial and axial loads on each bearing
    - $F_{rA} = \sqrt{R_{xA}^2 + R_{yA}^2}$
    - $F_{rB} = \sqrt{R_{xB}^2 + R_{yB}^2}$

- Assume bearing A carries the axial load: $F_{ae} = F_a$

2. Calculate the induced loads for each bearings. For our initial calculation we will assume $K_A = K_B = 1.5$.

$$F_{iA} = \frac{0.47F_{rA}}{K_A}$$
$$F_{iB} = \frac{0.47F_{rB}}{K_B}$$

3. Calculate the equivalent loads.

$$\text{If} \quad F_{iA} \leq (F_{iB} + F_{ae}) \quad \begin{cases} F_{eA} = 0.4F_{rA} + K_A(F_{iB} + F_{ae}) \\ F_{eB} = F_{rB} \end{cases}$$

$$\text{If} \quad F_{iA} > (F_{iB} + F_{ae}) \quad \begin{cases} F_{eB} = 0.4F_{rB} + K_B(F_{iA} - F_{ae}) \\ F_{eA} = F_{rA} \end{cases}$$

4. Get bearing design life $\mathscr{L}_D$ for your application from this table (if not given in the question):

Table 11–4 Bearing-Life Recommendations for Various Classes of Machinery

Type of Application	Life, kh
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines for 8-h service that are not always fully utilized	14–20
Machines for 8-h service that are fully utilized	20–30
Machines for continuous 24-h service	50–60
Machines for continuous 24-h service where reliability is of extreme importance	100–200

5. Calculate the multiple of rating life for each bearing in the assembly:

$$x_D = \frac{L_D}{L_{10}} = \frac{60\mathscr{L}_D n_D}{L_{10}}$$

where L_D is bearing design life in number of revolutions,
L_{10} is the rating life,
$\mathscr{L}_D$ is the design life in hours,
n_D is the angular speed of the bearing in rpm

The Weibull parameters used will depend on which manufacturer's bearings we are using.

- Timken (Manufacturer 1) is common for tapered roller bearings
- SKF (Manufacturer 2) is common for ball and straight roller bearings

		Weibull Parameters Rating Lives		
Manufacturer	Rating Life, Revolutions	X_0	θ	b
1	$90(10^6)$	0	4.48	1.5
2	$1(10^6)$	0.02	4.459	1.483

6. Calculate the bearing reliability of each individual bearing (if the total reliability of the ensemble is given):

$$R_i = \sqrt[n]{R_{tot}}$$

where R_i is the individual bearing reliability,
R_{tot} is the total bearing reliability,
n is the number of bearings in the assembly

7. For each bearing, calculate the load rating C_{10} for the equivalent load:

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)[\ln(1/R_D)]^{1/b}} \right]^{1/a}$$

- $a = 10/3$ for tapered roller bearings

8. Select tentative tapered roller bearings from the following table that have one-row radial values that exceed the calculated C_{10} values:

									cone				cup			
bore	outside diameter	width	rating at 500 rpm for 3000 hours L_{10}		factor	eff. load center	part numbers		max shaft fillet radius	width	backing shoulder diameters		max housing fillet radius	width	backing shoulder diameters	
			one-row radial	thrust			cone	cup								
d	**D**	**T**	**N** **lbf**	**N** **lbf**	**K**	**a**②			**R**①	**B**	**d_b**	**d_a**	**r**①	**C**	**D_b**	**D_a**
25.000 0.9843	**52.000** 2.0472	**16.250** 0.6398	**8190** 1840	**5260** 1180	**1.56**	**–3.6** –0.14	**◆30205**	**◆30205**	**1.0** 0.04	**15.000** 0.5906	**30.5** 1.20	**29.0** 1.14	**1.0** 0.04	**13.000** 0.5118	**46.0** 1.81	**48.5** 1.91
25.000 0.9843	**52.000** 2.0472	**19.250** 0.7579	**9520** 2140	**9510** 2140	**1.00**	**–3.0** –0.12	**◆32205-B**	**◆32205-B**	**1.0** 0.04	**18.000** 0.7087	**34.0** 1.34	**31.0** 1.22	**1.0** 0.04	**15.000** 0.5906	**43.5** 1.71	**49.5** 1.95
25.000 0.9843	**52.000** 2.0472	**22.000** 0.8661	**13200** 2980	**7960** 1790	**1.66**	**–7.6** –0.30	**◆33205**	**◆33205**	**1.0** 0.04	**22.000** 0.8661	**34.0** 1.34	**30.5** 1.20	**1.0** 0.04	**18.000** 0.7087	**44.5** 1.75	**49.0** 1.93
25.000 0.9843	**62.000** 2.4409	**18.250** 0.7185	**13000** 2930	**6680** 1500	**1.95**	**–5.1** –0.20	**◆30305**	**◆30305**	**1.5** 0.06	**17.000** 0.6693	**32.5** 1.28	**30.0** 1.18	**1.5** 0.06	**15.000** 0.5906	**55.0** 2.17	**57.0** 2.24
25.000 0.9843	**62.000** 2.4409	**25.250** 0.9941	**17400** 3910	**8930** 2010	**1.95**	**–9.7** –0.38	**◆32305**	**◆32305**	**1.5** 0.06	**24.000** 0.9449	**35.0** 1.38	**31.5** 1.24	**1.5** 0.06	**20.000** 0.7874	**54.0** 2.13	**57.0** 2.24
25.159 0.9905	**50.005** 1.9687	**13.495** 0.5313	**6990** 1570	**4810** 1080	**1.45**	**–2.8** –0.11	**07096**	**07196**	**1.5** 0.06	**14.260** 0.5614	**31.5** 1.24	**29.5** 1.16	**1.0** 0.04	**9.525** 0.3750	**44.5** 1.75	**47.0** 1.85
25.400 1.0000	**50.005** 1.9687	**13.495** 0.5313	**6990** 1570	**4810** 1080	**1.45**	**–2.8** –0.11	**07100**	**07196**	**1.0** 0.04	**14.260** 0.5614	**30.5** 1.20	**29.5** 1.16	**1.0** 0.04	**9.525** 0.3750	**44.5** 1.75	**47.0** 1.85
25.400 1.0000	**50.005** 1.9687	**13.495** 0.5313	**6990** 1570	**4810** 1080	**1.45**	**–2.8** –0.11	**07100-S**	**07196**	**1.5** 0.06	**14.260** 0.5614	**31.5** 1.24	**29.5** 1.16	**1.0** 0.04	**9.525** 0.3750	**44.5** 1.75	**47.0** 1.85
25.400 1.0000	**50.292** 1.9800	**14.224** 0.5600	**7210** 1620	**4620** 1040	**1.56**	**–3.3** –0.13	**L44642**	**L44610**	**3.5** 0.14	**14.732** 0.5800	**36.0** 1.42	**29.5** 1.16	**1.3** 0.05	**10.668** 0.4200	**44.5** 1.75	**47.0** 1.85
25.400 1.0000	**50.292** 1.9800	**14.224** 0.5600	**7210** 1620	**4620** 1040	**1.56**	**–3.3** –0.13	**L44643**	**L44610**	**1.3** 0.05	**14.732** 0.5800	**31.5** 1.24	**29.5** 1.16	**1.3** 0.05	**10.668** 0.4200	**44.5** 1.75	**47.0** 1.85
25.400 1.0000	**51.994** 2.0470	**15.011** 0.5910	**6990** 1570	**4810** 1080	**1.45**	**–2.8** –0.11	**07100**	**07204**	**1.0** 0.04	**14.260** 0.5614	**30.5** 1.20	**29.5** 1.16	**1.3** 0.05	**12.700** 0.5000	**45.0** 1.77	**48.0** 1.89
25.400 1.0000	**56.896** 2.2400	**19.368** 0.7625	**10900** 2450	**5740** 1290	**1.90**	**–6.9** –0.27	**1780**	**1729**	**0.8** 0.03	**19.837** 0.7810	**30.5** 1.20	**30.0** 1.18	**1.3** 0.05	**15.875** 0.6250	**49.0** 1.93	**51.0** 2.01
25.400 1.0000	**57.150** 2.2500	**19.431** 0.7650	**11700** 2620	**10900** 2450	**1.07**	**–3.0** –0.12	**M84548**	**M84510**	**1.5** 0.06	**19.431** 0.7650	**36.0** 1.42	**33.0** 1.30	**1.5** 0.06	**14.732** 0.5800	**48.5** 1.91	**54.0** 2.13
25.400 1.0000	**58.738** 2.3125	**19.050** 0.7500	**11600** 2610	**6560** 1470	**1.77**	**–5.8** –0.23	**1986**	**1932**	**1.3** 0.05	**19.355** 0.7620	**32.5** 1.28	**30.5** 1.20	**1.3** 0.05	**15.080** 0.5937	**52.0** 2.05	**54.0** 2.13
25.400 1.0000	**59.530** 2.3437	**23.368** 0.9200	**13900** 3140	**13000** 2930	**1.07**	**–5.1** –0.20	**M84249**	**M84210**	**0.8** 0.03	**23.114** 0.9100	**36.0** 1.42	**32.5** 1.27	**1.5** 0.06	**18.288** 0.7200	**49.5** 1.95	**56.0** 2.20
25.400 1.0000	**60.325** 2.3750	**19.842** 0.7812	**11000** 2480	**6550** 1470	**1.69**	**–5.1** –0.20	**15578**	**15523**	**1.3** 0.05	**17.462** 0.6875	**32.5** 1.28	**30.5** 1.20	**1.5** 0.06	**15.875** 0.6250	**51.0** 2.01	**54.0** 2.13
25.400 1.0000	**61.912** 2.4375	**19.050** 0.7500	**12100** 2730	**7280** 1640	**1.67**	**–5.8** –0.23	**15101**	**15243**	**0.8** 0.03	**20.638** 0.8125	**32.5** 1.28	**31.5** 1.24	**2.0** 0.08	**14.288** 0.5625	**54.0** 2.13	**58.0** 2.28
25.400 1.0000	**62.000** 2.4409	**19.050** 0.7500	**12100** 2730	**7280** 1640	**1.67**	**–5.8** –0.23	**15100**	**15245**	**3.5** 0.14	**20.638** 0.8125	**38.0** 1.50	**31.5** 1.24	**1.3** 0.05	**14.288** 0.5625	**55.0** 2.17	**58.0** 2.28
25.400 1.0000	**62.000** 2.4409	**19.050** 0.7500	**12100** 2730	**7280** 1640	**1.67**	**–5.8** –0.23	**15101**	**15245**	**0.8** 0.03	**20.638** 0.8125	**32.5** 1.28	**31.5** 1.24	**1.3** 0.05	**14.288** 0.5625	**55.0** 2.17	**58.0** 2.28

									cone				cup			
bore	outside diameter	width	rating at 500 rpm for 3000 hours L_{10}		fac-tor	eff. load center	part numbers		max shaft fillet radius	width	backing shoulder diameters		max hous-ing fillet radius	width	backing shoulder diameters	
			one-row radial	thrust			cone	cup								
d	D	T	N lbf	N lbf	K	a②			R①	B	d_b	d_a	r①	C	D_b	D_a
25.400 1.0000	62.000 2.4409	19.050 0.7500	12100 2730	7280 1640	1.67	–5.8 –0.23	15102	15245	1.5 0.06	20.638 0.8125	34.0 1.34	31.5 1.24	1.3 0.05	14.288 0.5625	55.0 2.17	58.0 2.28
25.400 1.0000	62.000 2.4409	20.638 0.8125	12100 2730	7280 1640	1.67	–5.8 –0.23	15101	15244	0.8 0.03	20.638 0.8125	32.5 1.28	31.5 1.24	1.3 0.05	15.875 0.6250	55.0 2.17	58.0 2.28
25.400 1.0000	63.500 2.5000	20.638 0.8125	12100 2730	7280 1640	1.67	–5.8 –0.23	15101	15250	0.8 0.03	20.638 0.8125	32.5 1.28	31.5 1.24	1.3 0.05	15.875 0.6250	56.0 2.20	59.0 2.32
25.400 1.0000	63.500 2.5000	20.638 0.8125	12100 2730	7280 1640	1.67	–5.8 –0.23	15101	15250X	0.8 0.03	20.638 0.8125	32.5 1.28	31.5 1.24	1.5 0.06	15.875 0.6250	55.0 2.17	59.0 2.32
25.400 1.0000	64.292 2.5312	21.433 0.8438	14500 3250	13500 3040	1.07	–3.3 –0.13	M86643	M86610	1.5 0.06	21.433 0.8438	38.0 1.50	36.5 1.44	1.5 0.06	16.670 0.6563	54.0 2.13	61.0 2.40
25.400 1.0000	65.088 2.5625	22.225 0.8750	13100 2950	16400 3690	0.80	–2.3 –0.09	23100	23256	1.5 0.06	21.463 0.8450	39.0 1.54	34.5 1.36	1.5 0.06	15.875 0.6250	53.0 2.09	63.0 2.48
25.400 1.0000	66.421 2.6150	23.812 0.9375	18400 4140	8000 1800	2.30	–9.4 –0.37	2687	2631	1.3 0.05	25.433 1.0013	33.5 1.32	31.5 1.24	1.3 0.05	19.050 0.7500	58.0 2.28	60.0 2.36
25.400 1.0000	68.262 2.6875	22.225 0.8750	15300 3440	10900 2450	1.40	–5.1 –0.20	02473	02420	0.8 0.03	22.225 0.8750	34.5 1.36	33.5 1.32	1.5 0.06	17.462 0.6875	59.0 2.32	63.0 2.48
25.400 1.0000	72.233 2.8438	25.400 1.0000	18400 4140	17200 3870	1.07	–4.6 –0.18	HM88630	HM88610	0.8 0.03	25.400 1.0000	39.5 1.56	39.5 1.56	2.3 0.09	19.842 0.7812	60.0 2.36	69.0 2.72
25.400 1.0000	72.626 2.8593	30.162 1.1875	22700 5110	13000 2910	1.76	–10.2 –0.40	3189	3120	0.8 0.03	29.997 1.1810	35.5 1.40	35.0 1.38	3.3 0.13	23.812 0.9375	61.0 2.40	67.0 2.64
26.157 1.0298	62.000 2.4409	19.050 0.7500	12100 2730	7280 1640	1.67	–5.8 –0.23	15103	15245	0.8 0.03	20.638 0.8125	33.0 1.30	32.5 1.28	1.3 0.05	14.288 0.5625	55.0 2.17	58.0 2.28
26.162 1.0300	63.100 2.4843	23.812 0.9375	18400 4140	8000 1800	2.30	–9.4 –0.37	2682	2630	1.5 0.06	25.433 1.0013	34.5 1.36	32.0 1.26	0.8 0.03	19.050 0.7500	57.0 2.24	59.0 2.32
26.162 1.0300	66.421 2.6150	23.812 0.9375	18400 4140	8000 1800	2.30	–9.4 –0.37	2682	2631	1.5 0.06	25.433 1.0013	34.5 1.36	32.0 1.26	1.3 0.05	19.050 0.7500	58.0 2.28	60.0 2.36
26.975 1.0620	58.738 2.3125	19.050 0.7500	11600 2610	6560 1470	1.77	–5.8 –0.23	1987	1932	0.8 0.03	19.355 0.7620	32.5 1.28	31.5 1.24	1.3 0.05	15.080 0.5937	52.0 2.05	54.0 2.13
† 26.988 † 1.0625	50.292 1.9800	14.224 0.5600	7210 1620	4620 1040	1.56	–3.3 –0.13	L44649	L44610	3.5 0.14	14.732 0.5800	37.5 1.48	31.0 1.22	1.3 0.05	10.668 0.4200	44.5 1.75	47.0 1.85
† 26.988 † 1.0625	60.325 2.3750	19.842 0.7812	11000 2480	6550 1470	1.69	–5.1 –0.20	15580	15523	3.5 0.14	17.462 0.6875	38.5 1.52	32.0 1.26	1.5 0.06	15.875 0.6250	51.0 2.01	54.0 2.13
† 26.988 † 1.0625	62.000 2.4409	19.050 0.7500	12100 2730	7280 1640	1.67	–5.8 –0.23	15106	15245	0.8 0.03	20.638 0.8125	33.5 1.32	33.0 1.30	1.3 0.05	14.288 0.5625	55.0 2.17	58.0 2.28
† 26.988 † 1.0625	66.421 2.6150	23.812 0.9375	18400 4140	8000 1800	2.30	–9.4 –0.37	2688	2631	1.5 0.06	25.433 1.0013	35.0 1.38	33.0 1.30	1.3 0.05	19.050 0.7500	58.0 2.28	60.0 2.36
28.575 1.1250	56.896 2.2400	19.845 0.7813	11600 2610	6560 1470	1.77	–5.8 –0.23	1985	1930	0.8 0.03	19.355 0.7620	34.0 1.34	33.5 1.32	0.8 0.03	15.875 0.6250	51.0 2.01	54.0 2.11
28.575 1.1250	57.150 2.2500	17.462 0.6875	11000 2480	6550 1470	1.69	–5.1 –0.20	15590	15520	3.5 0.14	17.462 0.6875	39.5 1.56	33.5 1.32	1.5 0.06	13.495 0.5313	51.0 2.01	53.0 2.09
28.575 1.1250	58.738 2.3125	19.050 0.7500	11600 2610	6560 1470	1.77	–5.8 –0.23	1985	1932	0.8 0.03	19.355 0.7620	34.0 1.34	33.5 1.32	1.3 0.05	15.080 0.5937	52.0 2.05	54.0 2.13
28.575 1.1250	58.738 2.3125	19.050 0.7500	11600 2610	6560 1470	1.77	–5.8 –0.23	1988	1932	3.5 0.14	19.355 0.7620	39.5 1.56	33.5 1.32	1.3 0.05	15.080 0.5937	52.0 2.05	54.0 2.13
28.575 1.1250	60.325 2.3750	19.842 0.7812	11000 2480	6550 1470	1.69	–5.1 –0.20	15590	15523	3.5 0.14	17.462 0.6875	39.5 1.56	33.5 1.32	1.5 0.06	15.875 0.6250	51.0 2.01	54.0 2.13
28.575 1.1250	60.325 2.3750	19.845 0.7813	11600 2610	6560 1470	1.77	–5.8 –0.23	1985	1931	0.5 0.03	19.355 0.7620	34.0 1.34	33.5 1.32	1.3 0.05	15.875 0.6250	52.0 2.05	55.0 2.17

9. Obtain new K_A and K_B values and repeat **steps 2 to 8** until you select the same bearing from the table twice in a row.

Note that the calculation for C_{10} only changes in F_e so we can use the following to calculate the new C_{10} value in every iteration:

$$\text{new } C_{10} = \frac{\text{new } F_e}{\text{old } F_e}(\text{old } C_{10})$$

4.4 Shafts and Keys

4.4.1 Design Selection

1. You should be given some shaft with length dimensions and certain gears/sheaves attached as well as bearing locations.
   Find the torque from each sheave/gear from the power (P) and angular speed (n)

   $$T = \frac{63000P}{n}$$

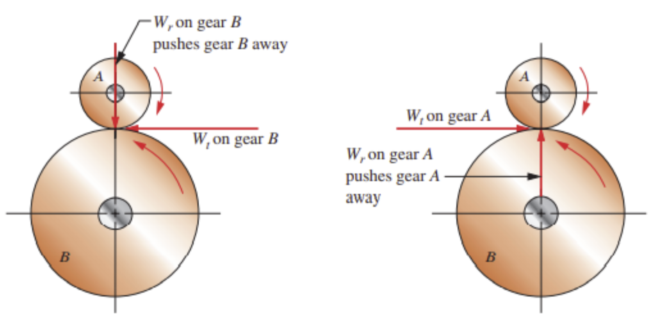
   Verify that the sum of torques is equal to 0

2. Compute the force from each sheave/gear

   (a) For spur gears:

   $$\text{tangential force: } W_t = \frac{2T}{D}$$
   $$\text{radial force: } W_r = W_t \tan\phi$$

   D = pitch diameter
   ϕ = pressure angle
   The direction will be given by

   (*a*) ***Action forces***—gear *A* drives gear *B*. Forces exerted on gear *B* by gear *A*.

   (*b*) ***Reaction forces.*** Forces exerted on gear *A* by gear *B*.

   (b) For helical gears:

   $$\text{tangential force: } W_t = \frac{2T}{D}$$
   $$\text{radial force: } W_r = W_t\frac{\tan\phi_n}{\cos\psi}$$
   $$\text{axial force: } W_x = W_t \tan\psi$$

   ϕ_n = normal pressure angle
   ψ = helix angle

(c) Bevel gears:

$$
\begin{aligned}
\text{tangential force: } W_t &= \frac{2T}{D} \\
\text{radial load: } W_r &= W_t \tan\phi \cos\Gamma = W_t \tan\phi \cos\gamma \\
\text{axial load: } W_x &= W_t \tan\phi \sin\Gamma = W_t \tan\phi \sin\gamma
\end{aligned}
$$

$\phi =$ pressure angle
$\gamma =$ cone angle of pinion
$\Gamma =$ cone angle of gear

(d) Wormgears:
Refer to wormgear section. I ain't writing that out again

(e) Chain sprockets:

$$F_{\text{shaft}} = \frac{2T}{D}$$

(f) V-Belt sheaves:

$$
\begin{aligned}
&\frac{F_1}{F_2} = k \text{ (assume } k = 5 \text{ if not given)} \\
&F_{\text{shaft}} = \frac{2T}{D}\frac{k+1}{k-1}
\end{aligned}
$$

$F_1 =$ tight side tension
$F_2 =$ slack side tension

(g) Flat belt pulleys:

$$
\begin{aligned}
&\frac{F_1}{F_2} = k \text{ (assume } k = 3 \text{ if not given)} \\
&F_{\text{shaft}} = \frac{2T}{D}\frac{k+1}{k-1}
\end{aligned}
$$

$F_1 =$ tight side tension
$F_2 =$ slack side tension

3. Take the forces you just calculated and draw a free body diagram of the shaft (recommended to break every force up into x and y components)

4. Calculate the reaction forces at the bearings
Set up the following equations to use:

$$
\begin{aligned}
&\sum F_x = 0 \\
&\sum F_y = 0 \\
&\sum M_x = 0 \\
&\sum M_y = 0
\end{aligned}
$$

$$\sum F_z = 0 \text{ (if axial load)}$$

Note that bearings will only have an axial force component if there's an axial force from a gear.

5. Draw the shear diagrams for the horizontal (x) and vertical forces (y)

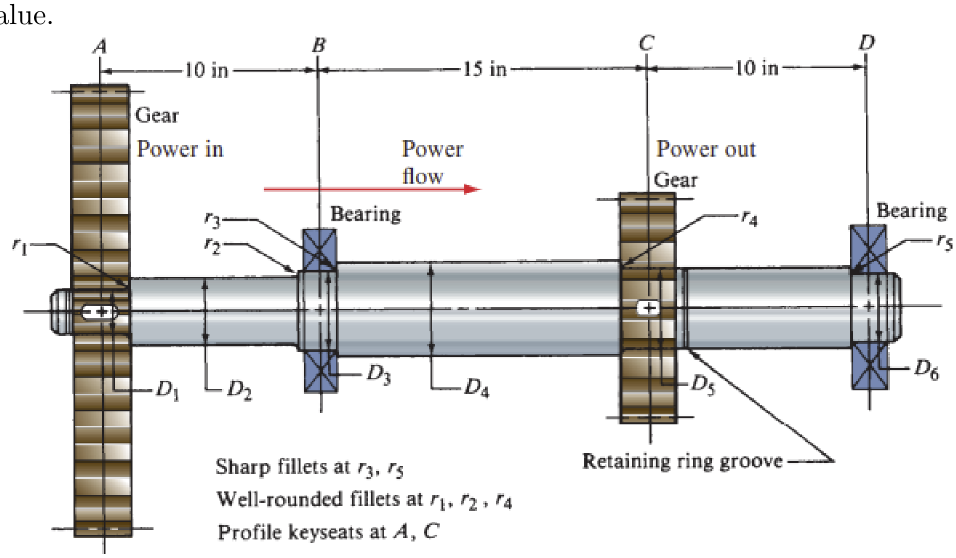
6. Draw the bending moment diagrams for the horizontal (x) and the vertical (y) moments
Recall bending moment is the integral (area) of the shear force diagram

7. Draw the torque diagram for the shaft (exact same idea as how to draw shear diagram)
Note that we will only have torque gains/drops across gears/pinions and not bearings

8. Compute the resultant shear and moment at each point of interest (where there's a component) along the shaft

$$V = \sqrt{V_x^2 + V_y^2}$$
$$M = \sqrt{M_x^2 + M_y^2}$$

9. Find the stress concentration factors (K_t).
Whenever there's a change in diameter or a component mounted we want to find the K_t value.



We will require K_t values for the following cases (usually corresponds to 3 K_t values for each component accounting for left, center, and right mounting of the component)

(a) Keyseats:
This corresponds to how the component (gear, pinion, etc.) is fastened in the center (usually a little slot the key slips into to make it turn). This is the center of A and C in the diagram.
For moving parts we have:
- $K_t = 2$ (profile keyseat)

- $K_t = 1.6$ (sled runner keyseat)

We always assume profile unless told otherwise.

(b) Shoulder fillets:
This is how sharp the fillet is when we have a change in diameter.

- $K_t = 2.5$ (sharp fillet)
- $K_t = 1.5$ (well-rounded fillet)

Components such as gears and sheaves can almost always be assumed to have well-rounded fillets. This corresponds to the right of A and the left of C on the diagram.

(c) Fitting bearings:
A bearing will sit on a shoulder and so the diameter of the left, right, and center will all be different.

- Small diameter side:
  Use a well-rounded fillet: $K_t = 1.5$
- Bearing seat (middle):
  Assume bearing is press fit: $K_t = 1$
- Large diameter side:
  Use a sharp fillet: $K_t = 2.5$

(d) Retaining rings:
These will be placed on the small diameter side of components such as gears/pinions to keep them in place.

- $K_t = 3$ (retaining ring)

These are located on the diagram at the left of A and the right of C.

10. Specify a material for the shaft and find tensile strength, s_u, and yield strength, s_y.
Tables for material properties can be found at the back of Motts or somewhere in this guide in the gears section.

11. Specify a material factor, C_m

Steel type	C_m
Wrought steel:	$C_m = 1.00$
Cast steel:	$C_m = 0.80$
Powdered steel:	$C_m = 0.76$
Malleable cast iron:	$C_m = 0.80$
Gray cast iron:	$C_m = 0.70$
Ductile cast iron:	$C_m = 0.66$

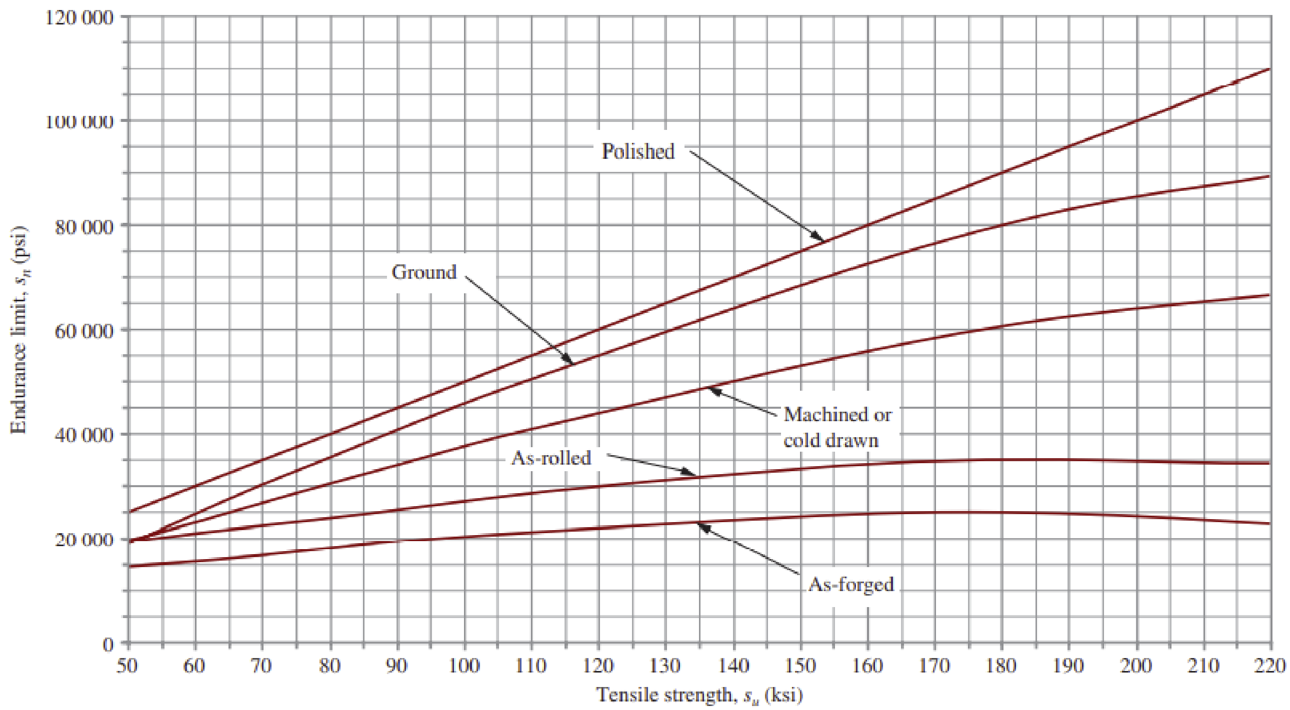
If not specified, assume Wrought steel and $C_m = 1$

12. Apply a type-of-stress factor, C_{st}

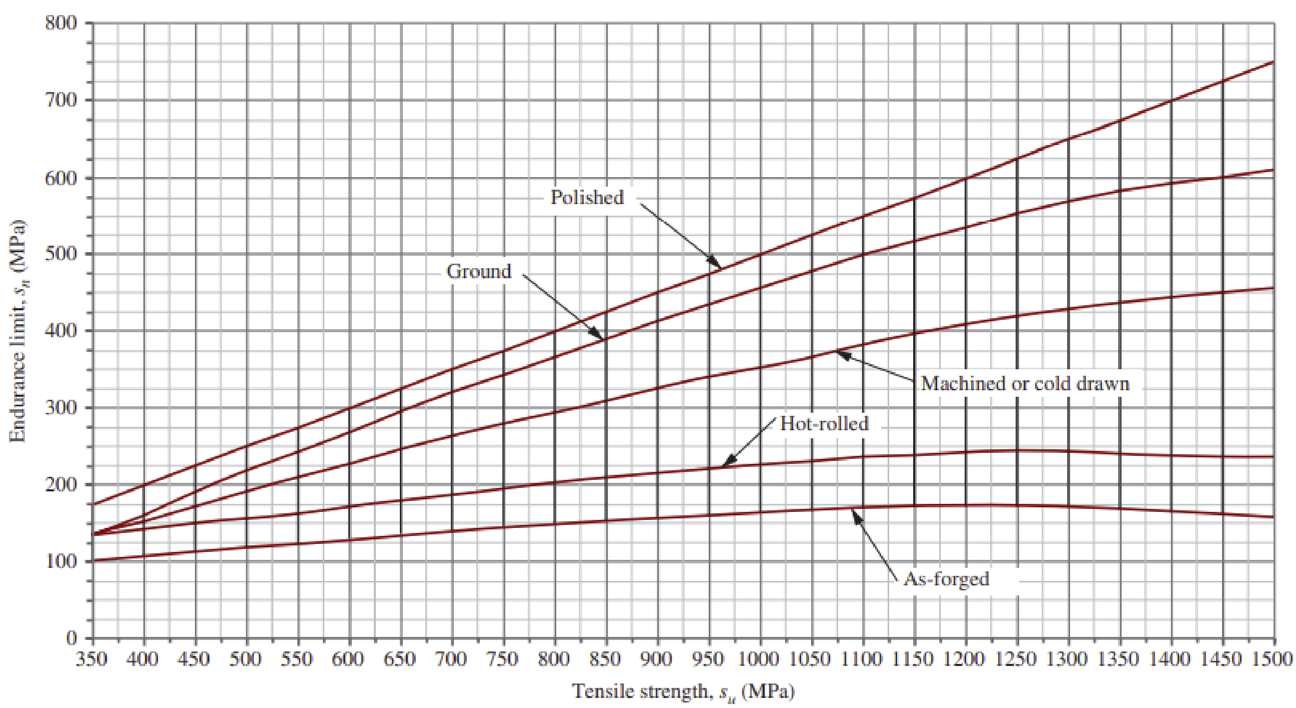
- $C_{st} = 1$ (bending stress)
- $C_{st} = 0.8$ (axial tension)

In almost all cases we will have bending stress so $C_{st} = 1$

13. Based on the material manufacturing (should be given) get s_n from the graph (if not specified assume cold drawn)



(*a*) U.S. customary units



(*b*) SI metric units

14. Get the relaibility factor C_R, from this table

TABLE 5–3 Approximate Reliability Factors, C_R

Desired reliability	C_R
0.50	1.0
0.90	0.90
0.99	0.81
0.999	0.75

A safe guess is 99% reliability so $C_R = 0.81$

15. Guess a size factor C_s.
We will usually want to guess an initial size factor of around $C_s = 0.8$ which corresponds to a shaft diameter of around 2". This will be adjusted later in further iterations if needed

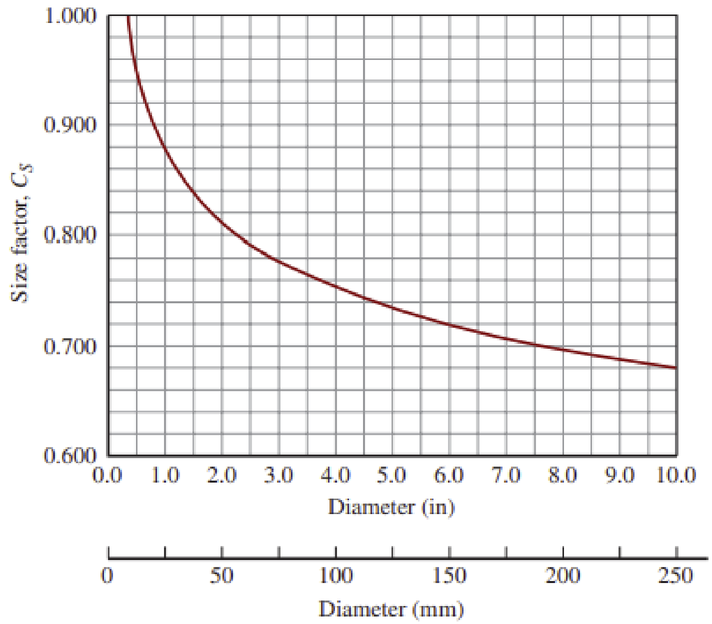


TABLE 5–4 Size Factors

U.S. customary units	
Size range	**For D in inches**
$D \leq 0.30$	$C_s = 1.0$
$0.30 < D \leq 2.0$	$C_s = (D/0.3)^{-0.11}$
$2.0 < D < 10.0$	$C_s = 0.859 - 0.021\,25D$
SI units	
Size range	**For D in mm**
$D \leq 7.62$	$C_s = 1.0$
$7.62 < D \leq 50$	$C_s = (D/7.62)^{-0.11}$
$50 < D < 250$	$C_s = 0.859 - 0.000\,837D$

16. Compute the estimated actual endurance limit s_n'

$$s_n' = s_n C_m C_{st} C_R C_s$$

17. Compute the minimum shaft diameter for each part (i.e. to the left, center, and right of a component. Any place where you may have a different K_t value).
Use this equation if there is torque or bending moment

$$D = \left(\frac{32N}{\pi}\sqrt{\left(\frac{K_t M}{s_n'}\right)^2 + \frac{3}{4}\left(\frac{T}{s_y}\right)^2}\right)^{1/3}$$

Use this equation for if there is no torque or bending (only shear), or if it might exceed the diameter above

$$D = \sqrt{\frac{2.94 K_t V N}{s_n'}}$$

N = design (safety) factor
M = bending moment
T = torque
V = shear force

Take a moment to remember that there are people out there who are proud of you.

18. Multiply any diameters corresponding to retaining rings (with $K_t = 3$) by 1.06

$$D_{\text{retaining ring}} = 1.06D$$

19. Some of these diameters will correspond to the same locations along the shaft. Identify the most restrictive diameters to get a minimum diameter size for each part of the shaft.

20. At this point you may want to consider a 2nd iteration, making a better guess for C_s in step 15. This is recommended if your shaft diameters are significantly different from your initial guess of around 2”.

21. Find the C value for the bearing:

   (a) Determine the load life L_2 based on the usage description, basically the total number of revolutions over the expected lifetime:

$$L_2 = (\#\text{ rpm}) \cdot (\frac{60\text{min}}{\text{hr}})(\frac{\#\text{ hours}}{\text{day}})(\frac{\#\text{ days}}{\text{month}})(\frac{12\text{month}}{\text{year}})(\#\text{ years})$$

   (b) If the reliability of the bearing is not 90%, then you have to multiply $L_{10} = 10^6$ by a constant to adjust. Find the reliability constant from the table below:

TABLE 14–6 Life Adjustment Factors for Reliability, C_R

Reliability (%)	C_R	Life designation
90	1.0	L_{10}
95	0.62	L_5
96	0.53	L_4
97	0.44	L_3
98	0.33	L_2
99	0.21	L_1

   and your new L_1 value will be:

$$L_1 = C_R \cdot 10^6$$

   (c) Bearing Load/Life relationship, use $k = 3.00$ for ball bearings and $k = 3.33$ for tapered. L_1 is the catalog value of 1×10^6 revs for SKF bearings and most manufacturers. P_2 is the actual radial force experienced on the shaft:

$$\frac{L_2}{L_1} = (\frac{C}{P_2})^k \qquad\qquad C = P_2(\frac{L_2}{C_R \cdot 10^6})^{\frac{1}{k}}$$

22. Use the C value and the minimum diameter of the shaft at the bearing to select a suitable bearing from the table

TABLE 14–3 Dimensions for Single Row, Deep-groove Ball Bearings

Bearing number	Nominal bearing dimensions						Basic load ratings				Maximum fillet radius r_{max}[1]		Minimum shaft shoulder diameter, S		Maximum housing shoulder diameter, H		Bearing mass	
	Bore, d		Outside dia., D		Width, B		Static, C_o		Dynamic, C									
	mm	in	mm	in	mm	in	kN	lb_f	kN	lb_f	mm	in	mm	in	mm	in	kg	lb_m
6000	**10**	0.3937	26	1.0236	8	0.3150	1.96	441	4.62	1039	0.3	0.012	12	0.472	24	0.945	0.019	0.042
6200	**10**	0.3937	30	1.1811	9	0.3543	2.36	531	5.07	1140	0.6	0.024	14	0.551	26	1.024	0.032	0.071
6300	**10**	0.3937	35	1.3780	11	0.4331	8.06	1812	3.40	764	0.6	0.024	14	0.551	31	1.220	0.053	0.117
6001	**12**	0.4724	28	1.1024	8	0.3150	2.36	531	5.07	1140	0.3	0.012	14	0.551	26	1.024	0.022	0.049
6201	**12**	0.4724	32	1.2598	10	0.3937	3.10	697	6.89	1549	0.6	0.024	16	0.630	28	1.102	0.037	0.082
6301	**12**	0.4724	37	1.4567	12	0.4724	4.15	933	9.75	2192	1.0	0.039	17	0.669	32	1.260	0.060	0.132
6002	**15**	0.5906	32	1.2598	9	0.3543	2.85	641	5.59	1257	0.3	0.012	17	0.669	30	1.181	0.030	0.066
6202	**15**	0.5906	35	1.3780	11	0.4331	3.75	843	7.80	1754	0.6	0.024	19	0.748	31	1.220	0.045	0.099
6302	**15**	0.5906	42	1.6535	13	0.5118	5.40	1214	11.40	2563	1.0	0.039	20	0.787	37	1.457	0.082	0.181
6003	**17**	0.6693	35	1.3780	10	0.3937	3.25	731	6.05	1360	0.3	0.012	19	0.748	33	1.299	0.039	0.086
6203	**17**	0.6693	40	1.5748	12	0.4724	4.75	1068	9.56	2149	0.6	0.024	21	0.827	36	1.417	0.065	0.143
6303	**17**	0.6693	47	1.8504	14	0.5512	6.55	1473	13.50	3035	1.0	0.039	22	0.866	42	1.654	0.120	0.265
6004	**20**	0.7874	42	1.6535	12	0.4724	5.00	1124	9.36	2104	0.6	0.024	24	0.945	38	1.496	0.069	0.152
6204	**20**	0.7874	47	1.8504	14	0.5512	6.55	1473	12.70	2855	1.0	0.039	25	0.984	42	1.654	0.110	0.243
6304	**20**	0.7874	52	2.0472	15	0.5906	7.80	1754	15.90	3575	1.0	0.039	27	1.063	45	1.772	0.140	0.309
6005	**25**	0.9843	47	1.8504	12	0.4724	6.55	1473	11.20	2518	0.6	0.024	29	1.142	43	1.693	0.080	0.176
6205	**25**	0.9843	52	2.0472	15	0.5906	7.80	1754	14.00	3147	1.0	0.039	30	1.181	47	1.850	0.130	0.287
6305	**25**	0.9843	62	2.4409	17	0.6693	11.60	2608	22.50	5058	1.0	0.039	32	1.260	55	2.165	0.230	0.507
6006	**30**	1.1811	55	2.1654	13	0.5118	8.30	1866	13.30	2990	1.0	0.039	35	1.378	50	1.969	0.160	0.353
6206	**30**	1.1811	62	2.4409	16	0.6299	11.2	2518	19.5	4384	1.0	0.039	35	1.378	57	2.244	0.200	0.441
6306	**30**	1.1811	72	2.8346	19	0.7480	16.0	3597	28.1	6317	1.0	0.039	37	1.457	65	2.559	0.350	0.772
6007	**35**	1.3780	62	2.4409	14	0.5512	10.2	2293	15.9	3575	1.0	0.039	40	1.575	57	2.244	0.160	0.353
6207	**35**	1.3780	72	2.8346	17	0.6693	15.3	3440	25.5	5733	1.0	0.039	42	1.654	65	2.559	0.290	0.639
6307	**35**	1.3780	80	3.1496	21	0.8268	19.0	4272	33.2	7464	1.5	0.059	43	1.693	72	2.835	0.460	1.014
6008	**40**	1.5748	68	2.6772	15	0.5906	11.6	2608	16.8	3777	1.0	0.039	45	1.772	63	2.480	0.190	0.419
6208	**40**	1.5748	80	3.1496	18	0.7087	19.0	4272	30.7	6902	1.0	0.039	47	1.850	73	2.874	0.370	0.816
6308	**40**	1.5748	90	3.5433	23	0.9055	24.0	5396	41.0	9218	1.5	0.059	48	1.890	82	3.228	0.630	1.389

TABLE 14–3 Dimensions for Single Row, Deep-groove Ball Bearings (*continued*)

Bearing number	Nominal bearing dimensions						Basic load ratings				Maximum fillet radius r_{max}[1]		Minimum shaft shoulder diameter, S		Maximum housing shoulder diameter, H		Bearing mass	
	Bore, d		Outside dia., D		Width, B		Static, C_o		Dynamic, C									
	mm	in	mm	in	mm	in	kN	lb_f	kN	lb_f	mm	in	mm	in	mm	in	kg	lb_m
6009	**45**	1.7717	75	2.9528	16	0.6299	14.6	3282	20.8	4676	1.0	0.039	50	1.969	70	2.756	0.250	0.551
6209	**45**	1.7717	85	3.3465	19	0.7480	21.6	4856	33.2	7464	1.0	0.039	52	2.047	78	3.071	0.410	0.904
6309	**45**	1.7717	100	3.9370	25	0.9843	31.5	7082	52.7	11 848	1.5	0.059	53	2.087	92	3.622	0.830	1.830
6010	**50**	1.9685	80	3.1496	16	0.6299	16.0	3597	21.6	4856	1.0	0.039	55	2.165	75	2.953	0.260	0.573
6210	**50**	1.9685	90	3.5433	20	0.7874	23.2	5216	35.1	7891	1.0	0.039	57	2.244	83	3.268	0.460	1.014
6310	**50**	1.9685	110	4.3307	27	1.0630	38.0	8543	61.8	13 894	2.0	0.079	59	2.323	101	3.976	1.050	2.315
6011	**55**	2.1654	90	3.5433	18	0.7087	21.2	4766	28.1	6317	1.0	0.039	62	2.441	83	3.268	0.390	0.860
6211	**55**	2.1654	100	3.9370	21	0.8268	29.0	6520	43.6	9802	1.5	0.059	63	2.480	92	3.622	0.610	1.345
6311	**55**	2.1654	120	4.7244	29	1.1417	45.0	10 117	71.5	16 075	2.0	0.079	64	2.520	111	4.370	1.350	2.977
6012	**60**	2.3622	95	3.7402	18	0.7087	23.2	5216	29.6	6655	1.0	0.039	67	2.638	88	3.465	0.420	0.926
6212	**60**	2.3622	110	4.3307	22	0.8661	32.5	7307	47.5	10 679	1.5	0.059	68	2.677	102	4.016	0.780	1.720
6312	**60**	2.3622	130	5.1181	31	1.2205	52.0	11 691	81.9	18 413	2.0	0.079	71	2.795	119	4.685	1.700	3.749
6013	**65**	2.5591	100	3.9370	18	0.7087	25.0	5621	30.7	6902	1.0	0.039	72	2.835	93	3.661	0.440	0.970
6213	**65**	2.5591	120	4.7244	23	0.9055	40.5	9105	55.9	12 567	1.5	0.059	73	2.874	112	4.409	0.990	2.183
6313	**65**	2.5591	140	5.5118	33	1.2992	60.0	13 489	92.3	20 751	2.0	0.079	76	2.992	129	5.079	2.100	4.631
6014	**70**	2.7559	110	4.3307	20	0.7874	31.0	6969	37.7	8476	1.0	0.039	77	3.031	103	4.055	0.600	1.323
6214	**70**	2.7559	125	4.9213	24	0.9449	45.0	10 117	60.5	13 602	1.5	0.059	78	3.071	117	4.606	1.050	2.315
6314	**70**	2.7559	150	5.9055	35	1.3780	68.0	15 288	104.0	23 381	2.0	0.079	81	3.189	139	5.472	2.500	5.513
6015	**75**	2.9528	115	4.5276	20	0.7874	33.5	7531	39.7	8925	1.0	0.039	82	3.228	108	4.252	0.640	1.411
6215	**75**	2.9528	130	5.1181	25	0.9843	49.0	11 016	66.3	14 906	1.5	0.059	83	3.268	122	4.803	1.200	2.646
6315	**75**	2.9528	160	6.2992	37	1.4567	76.5	17 199	114.0	25 629	2.0	0.079	86	3.386	149	5.866	3.000	6.615
6016	**80**	3.1496	125	4.9213	22	0.8661	40.0	8993	47.5	10 679	1.0	0.039	87	3.425	118	4.646	0.850	1.874
6216	**80**	3.1496	140	5.5118	26	1.0236	55.0	12 365	70.2	15 782	2.0	0.079	89	3.504	131	5.157	1.400	3.087
6316	**80**	3.1496	170	6.6929	39	1.5354	86.5	19 447	124.0	27 878	2.0	0.079	91	3.583	159	6.260	3.600	7.938

TABLE 14–3 Dimensions for Single Row, Deep-groove Ball Bearings (*continued*)

Bearing number	Nominal bearing dimensions						Basic load ratings				Maximum fillet radius r_{max}[1]		Minimum shaft shoulder diameter, S		Maximum housing shoulder diameter, H		Bearing mass	
	Bore, d		Outside dia., D		Width, B		Static, C_o		Dynamic, C									
	mm	in	mm	in	mm	in	kN	lb_f	kN	lb_f	mm	in	mm	in	mm	in	kg	lb_m
6017	**85**	3.3465	130	5.1181	22	0.8661	43.0	9667	49.4	11 106	1.0	0.039	92	3.622	123	4.843	0.890	1.962
6217	**85**	3.3465	150	5.9055	28	1.1024	64.0	14 388	83.2	18 705	2.0	0.079	94	3.701	141	5.551	1.800	3.969
6317	**85**	3.3465	180	7.0866	41	1.6142	96.5	21 695	133.0	29 901	2.5	0.098	98	3.858	167	6.575	4.250	9.371
6018	**90**	3.5433	140	5.5118	24	0.9449	50.0	11 241	58.5	13 152	1.5	0.059	98	3.858	132	5.197	1.150	2.536
6218	**90**	3.5433	160	6.2992	30	1.1811	73.5	16 524	95.6	21 493	2.0	0.079	99	3.898	151	5.945	2.150	4.741
6318	**90**	3.5433	190	7.4803	43	1.6929	108.0	24 281	143.0	32 149	2.5	0.098	103	4.055	177	6.969	4.900	10.805
6019	**95**	3.7402	145	5.7087	24	0.9449	54.0	12 140	60.5	13 602	1.5	0.059	103	4.055	137	5.394	1.200	2.646
6219	**95**	3.7402	170	6.6929	32	1.2598	81.5	18 323	108.0	24 281	2.0	0.079	106	4.173	159	6.260	2.600	5.733
6319	**95**	3.7402	200	7.8740	45	1.7717	118.0	26 529	153.0	34 397	2.5	0.098	108	4.252	187	7.362	5.650	12.458
6020	**100**	3.9370	150	5.9055	24	0.9449	54.0	12 140	60.5	13 602	1.5	0.059	108	4.252	142	5.591	1.250	2.756
6220	**100**	3.9370	180	7.0866	34	1.3386	93.0	20 908	124.0	27 878	2.0	0.079	111	4.370	169	6.654	3.150	6.946
6320	**100**	3.9370	215	8.4646	47	1.8504	140.0	31 475	174.0	39 119	2.5	0.098	113	4.449	202	7.953	7.000	15.435
6021	**105**	4.1339	160	6.2992	26	1.0236	65.5	14 726	72.8	16 367	2.0	0.079	114	4.488	151	5.945	1.600	3.528
6221	**105**	4.1339	190	7.4803	36	1.4173	104.0	23 381	133.0	29 901	2.0	0.079	116	4.567	179	7.047	3.700	8.159
6321	**105**	4.1339	225	8.8583	49	1.9291	153.0	34 397	182.0	40 917	2.5	0.098	118	4.646	212	8.346	8.250	18.191
6022	**110**	4.3307	170	6.6929	28	1.1024	73.5	16 524	81.9	18 413	2.0	0.079	119	4.685	161	6.339	1.950	4.300
6222	**110**	4.3307	200	7.8740	38	1.4961	118.0	26 529	143.0	32 149	2.0	0.079	121	4.764	189	7.441	4.350	9.592
6322	**110**	4.3307	240	9.4488	50	1.9685	180.0	40 468	203.0	45 638	2.5	0.098	123	4.843	227	8.937	9.550	21.058
6024	**120**	4.7244	180	7.0866	28	1.1024	80.0	17 986	85.2	19 155	2.0	0.079	129	5.079	171	6.732	2.050	4.520
6224	**120**	4.7244	215	8.4646	40	1.5748	118.0	26 529	146.0	32 824	2.0	0.079	131	5.157	204	8.031	5.150	11.356
6324	**120**	4.7244	260	10.2362	55	2.1654	186.0	41 817	208.0	46 763	2.5	0.098	133	5.236	247	9.724	14.500	31.973
6026	**130**	5.1181	200	7.8740	33	1.2992	100.0	22 482	106.0	23 831	2.0	0.079	139	5.472	191	7.520	3.150	6.946
6226	**130**	5.1181	230	9.0551	40	1.5748	132.0	29 676	156.0	35 072	2.5	0.098	143	5.630	217	8.543	5.800	12.789
6326	**130**	5.1181	280	11.0236	58	2.2835	216.0	48 561	229.0	51 484	3.0	0.118	146	5.748	264	10.394	18.000	39.690

[1]Maximum fillet on shaft shoulder that will clear radius on bearing race

23. Take the bearing you chose from the table and use its bore diameter to identify the minimum shaft diameter

TABLE 15–5 Shaft and Housing Fits for Bearings

A. Shaft fits

Bearing bore			ISO tolerance grade	Shaft diameter		Limits of fit	
Nominal (mm)	Maximum (in)	Minimum (in)		Maximum (in)	Minimum (in)	Minimum (in)	Maximum (in)
10	0.3937	0.3934	j5	0.3939	0.3936	0.0001L	0.0005T
12	0.4724	0.4721	j5	0.4726	0.4723	0.0001L	0.0005T
15	0.5906	0.5903	j5	0.5908	0.5905	0.0001L	0.0005T
17	0.6693	0.6690	j5	0.6695	0.6692	0.0001L	0.0005T
20	0.7874	0.7870	k5	0.7878	0.7875	0.0001T	0.0008T
25	0.9843	0.9839	k5	0.9847	0.9844	0.0001T	0.0008T
30	1.1811	1.1807	k5	1.1815	1.1812	0.0001T	0.0008T
35	1.3780	1.3775	k5	1.3785	1.3781	0.0001T	0.0010T
40	1.5748	1.5743	k5	1.5753	1.5749	0.0001T	0.0010T
45	1.7717	1.7712	k5	1.7722	1.7718	0.0001T	0.0010T
50	1.9685	1.9680	k5	1.9690	1.9686	0.0001T	0.0010T
55	2.1654	2.1648	k5	2.1660	2.1655	0.0001T	0.0012T
60	2.3622	2.3616	k5	2.3628	2.3623	0.0001T	0.0012T
65	2.5591	2.5585	k5	2.5597	2.5592	0.0001T	0.0012T
70	2.7559	2.7553	k5	2.7565	2.7560	0.0001T	0.0012T
75	2.9528	2.9522	k5	2.9534	2.9529	0.0001T	0.0012T
80	3.1496	3.1490	k5	3.1502	3.1497	0.0001T	0.0012T
85	3.3465	3.3457	k5	3.3472	3.3466	0.0001T	0.0015T
90	3.5433	3.5425	k5	3.5440	3.5434	0.0001T	0.0015T
95	3.7402	3.7394	k5	3.7409	3.7403	0.0001T	0.0015T
100	3.9370	3.9362	k5	3.9377	3.9371	0.0001T	0.0015T
105	4.1339	4.1331	m5	4.1350	4.1344	0.0005T	0.0019T
110	4.3307	4.3299	m5	4.3318	4.3312	0.0005T	0.0019T
115	4.5276	4.5268	m5	4.5287	4.5281	0.0005T	0.0019T
120	4.7244	4.7236	m5	4.7255	4.7249	0.0005T	0.0019T
125	4.9213	4.9203	m5	4.9226	4.9219	0.0006T	0.0023T
130	5.1181	5.1171	m5	5.1194	5.1187	0.0006T	0.0023T
140	5.5118	5.5108	m5	5.5131	5.5124	0.0006T	0.0023T
150	5.9055	5.9045	m6	5.9071	5.9061	0.0006T	0.0026T
160	6.2992	6.2982	m6	6.3008	6.2998	0.0006T	0.0026T
170	6.6929	6.6919	m6	6.6945	6.6935	0.0006T	0.0026T
180	7.0866	7.0856	m6	7.0882	7.0872	0.0006T	0.0026T
190	7.4803	7.4791	m6	7.4821	7.4810	0.0007T	0.0030T
200	7.8740	7.8728	m6	7.8758	7.8747	0.0007T	0.0030T

B. Housing fits

Bearing *OD*			ISO tolerance grade	Housing bore		Limits of fit	
Nominal (mm)	Maximum (in)	Minimum (in)		Maximum (in)	Minimum (in)	Minimum (in)	Maximum (in)
30	1.1811	1.1807	H8	1.1811	1.1824	0	0.0017L
32	1.2598	1.2594	H8	1.2598	1.2613	0	0.0019L
35	1.3780	1.3776	H8	1.3780	1.3795	0	0.0019L

TABLE 15–5 Shaft and Housing Fits for Bearings (*continued*)

B. Housing fits

Bearing *OD*			ISO tolerance grade	Housing bore		Limits of fit	
Nominal (mm)	Maximum (in)	Minimum (in)		Maximum (in)	Minimum (in)	Minimum (in)	Maximum (in)
37	1.4567	1.4563	H8	1.4567	1.4582	0	0.0019L
40	1.5748	1.5744	H8	1.5748	1.5763	0	0.0019L
42	1.6535	1.6531	H8	1.6535	1.6550	0	0.0019L
47	1.8504	1.8500	H8	1.8504	1.8519	0	0.0019L
52	2.0472	2.0467	H8	2.0472	2.0490	0	0.0023L
62	2.4409	2.4404	H8	2.4409	2.4427	0	0.0023L
72	2.8346	2.8341	H8	2.8346	2.8364	0	0.0023L
80	3.1496	3.1491	H8	3.1496	3.1514	0	0.0023L
85	3.3465	3.3459	H8	3.3465	3.3486	0	0.0027L
90	3.5433	3.5427	H8	3.5433	3.5454	0	0.0027L
100	3.9370	3.9364	H8	3.9370	3.9391	0	0.0027L
110	4.3307	4.3301	H8	4.3307	4.3328	0	0.0027L
120	4.7244	4.7238	H8	4.7244	4.7265	0	0.0027L
125	4.9213	4.9206	H8	4.9213	4.9238	0	0.0032L
130	5.1181	5.1174	H8	5.1181	5.1206	0	0.0032L
140	5.5118	5.5111	H8	5.5118	5.5143	0	0.0032L
150	5.9055	5.9048	H8	5.9055	5.9080	0	0.0032L
160	6.2992	6.2982	H8	6.2992	6.3017	0	0.0035L
170	6.6929	6.6919	H8	6.6929	6.6954	0	0.0035L
180	7.0866	7.0856	H8	7.0866	7.0891	0	0.0035L
190	7.4803	7.4791	H8	7.4803	7.4831	0	0.0040L
200	7.8740	7.8728	H8	7.8740	7.8768	0	0.0040L
215	8.4646	8.4634	H8	8.4646	8.4674	0	0.0040L
225	8.8583	8.8571	H8	8.8583	8.8611	0	0.0040L
230	9.0551	9.0539	H8	9.0551	9.0579	0	0.0040L
240	9.4488	9.4476	H8	9.4488	9.4516	0	0.0040L
250	9.8425	9.8413	H8	9.8425	9.8453	0	0.0040L
260	10.2362	10.2348	H8	10.2362	10.2394	0	0.0046L
270	10.6299	10.6285	H8	10.6299	10.6331	0	0.0046L
280	11.0236	11.0222	H8	11.0236	11.0268	0	0.0046L
290	11.4173	11.4159	H8	11.4173	11.4205	0	0.0046L
300	11.8110	11.8096	H8	11.8110	11.8142	0	0.0046L
310	12.2047	12.2033	H8	12.2047	12.2079	0	0.0046L
320	12.5984	12.5968	H8	12.5984	12.6019	0	0.0051L
340	13.3858	13.3842	H8	13.3858	13.3893	0	0.0051L
360	14.1732	14.1716	H8	14.1732	14.1767	0	0.0051L
380	14.9606	14.9590	H8	14.9606	14.9641	0	0.0051L
400	15.7480	15.7464	H8	15.7480	15.7515	0	0.0051L
420	16.5354	16.5336	H8	16.5354	16.5392	0	0.0056L

Note: L = loose; and T = tight.

24. Use the selected bearing shaft diameter to round up any necessary minimum shaft diameters that should be the same or larger than the bearing shaft

25. Round up the shaft diameters (except for the bearing one you chose) to the nearest standard size

TABLE A2–1 Preferred Basic Sizes

Fractional (in)				Decimal (in)			SI metric (mm)	
1/64	0.015 625	5	5.000	0.010	2.00	8.50	**1.0**	**40**
1/32	0.031 25	$5\frac{1}{4}$	5.250	0.012	2.20	9.00	1.1	45
1/16	0.0625	$5\frac{1}{2}$	5.500	0.016	2.40	9.50	**1.2**	**50**
3/32	0.093 75	$5\frac{3}{4}$	5.750	0.020	2.60	10.00	1.4	55
1/8	0.1250	6	6.000	0.025	2.80	10.50	**1.6**	**60**
5/32	0.156 25	$6\frac{1}{2}$	6.500	0.032	3.00	11.00	1.8	70
3/16	0.1875	7	7.000	0.040	3.20	11.50	**2.0**	**80**
1/4	0.2500	$7\frac{1}{2}$	7.500	0.05	3.40	12.00	2.2	90
5/16	0.3125	8	8.000	0.06	3.60	12.50	**2.5**	**100**
3/8	0.3750	$8\frac{1}{2}$	8.500	0.08	3.80	13.00	2.8	110
7/16	0.4375	9	9.000	0.10	4.00	13.50	**3.0**	**120**
1/2	0.5000	$9\frac{1}{2}$	9.500	0.12	4.20	14.00	3.5	140
9/16	0.5625	10	10.000	0.16	4.40	14.50	**4.0**	**160**
5/8	0.6250	$10\frac{1}{2}$	10.500	0.20	4.60	15.00	4.5	180
11/16	0.6875	11	11.000	0.24	4.80	15.50	**5.0**	**200**
3/4	0.7500	$11\frac{1}{2}$	11.500	0.30	5.00	16.00	5.5	220
7/8	0.8750	12	12.000	0.40	5.20	16.50	**6**	**250**
1	1.000	$12\frac{1}{2}$	12.500	0.50	5.40	17.00	7	280
$1\frac{1}{4}$	1.250	13	13.000	0.60	5.60	17.50	**8**	**300**
$1\frac{1}{2}$	1.500	$13\frac{1}{2}$	13.500	0.80	5.80	18.00	9	350
$1\frac{3}{4}$	1.750	14	14.000	1.00	6.00	18.50	**10**	**400**
2	2.000	$14\frac{1}{2}$	14.500	1.20	6.50	19.00	11	450
$2\frac{1}{4}$	2.250	15	15.000	1.40	7.00	19.50	**12**	**500**
$2\frac{1}{2}$	2.500	$15\frac{1}{2}$	15.500	1.60	7.50	20.00	14	550
$2\frac{3}{4}$	2.750	16	16.000	1.80	8.00		**16**	**600**
3	3.000	$16\frac{1}{2}$	16.500				18	700
$3\frac{1}{4}$	3.250	17	17.000				**20**	**800**
$3\frac{1}{2}$	3.500	$17\frac{1}{2}$	17.500				22	900
$3\frac{3}{4}$	3.750	18	18.000				**25**	**1000**
4	4.000	$18\frac{1}{2}$	18.500				28	
$4\frac{1}{4}$	4.250	19	19.000				**30**	
$4\frac{1}{2}$	4.500	$19\frac{1}{2}$	19.500				35	
$4\frac{3}{4}$	4.750	20	20.000					

26. Select a material for the keys and get the yield strength, s_y.
SAE 1018 Carbon steel is a pretty sexy choice. I'd recommend that one.
But for the easiest time calculating key length make sure that the key material is weaker than the shaft material.

TABLE 11–4 Examples of Materials Used for Keys

Material designation	Tensile strength s_u (ksi)	Tensile strength s_u (MPa)	Yield strength s_y (ksi)	Yield strength s_y (MPa)
Carbon steels (SAE)				
1018	64	441	54	372
1035	72	496	39.5	272
1045	91	627	77	531
1095	140	965	83	572
Alloy steels (SAE)				
4140	102	703	90	621
8630	100	690	95	655
Stainless steels (SAE)				
303	90	621	35	241
304	85	586	35	241
316	85	586	35	241
416	75	517	40	276
Aluminum				
6061	18	124	12	83

Source: Adapted from Internet site 20.

Note: Strength properties typical, not guaranteed.

27. Select a key size using the shaft diameter in the spot of the component

TABLE 11–1 Key Size vs. Shaft Diameter

U.S. inch sizes				SI metric sizes			
Nominal shaft diameter		Key dimensions		Nominal shaft diameter		Key dimensions	
Over (in)	to-including (in)	Width, W (in)	Height, H (in)	Over (mm)	to-including (mm)	Width, W (mm)	Height, H (mm)
0.3125	0.4375	0.09375	0.09375	6	8	2	2
0.4375	0.5625	0.1250	0.1250	8	10	3	3
0.5625	0.875	0.1875	0.1875	10	12	4	4
0.875	1.250	0.2500	0.2500	12	17	5	5
1.250	1.375	0.3125	0.3125	17	22	6	6
1.375	1.75	0.375	0.375	22	30	8	7
1.75	2.25	0.500	0.500	30	38	10	8
2.25	2.75	0.625	0.625	38	44	12	8
2.75	3.25	0.750	0.750	44	50	14	9
3.25	3.75	0.875	0.875	50	58	16	10
3.75	4.50	1.00	1.00	58	65	18	11
4.50	5.50	1.25	1.25	65	75	20	12
5.50	6.50	1.50	1.50	75	85	22	14
6.50	7.50	1.75	1.50	85	95	25	14
7.50	9.00	2.00	1.50	95	110	28	16
9.00	11.00	2.50	1.75	110	130	32	18
11.00	13.00	3.00	2.00	130	150	36	20
13.00	15.00	3.50	2.50	150	170	40	22
15.00	18.00	4.00	3.00	170	200	45	25
18.00	22.00	5.00	3.50	200	230	50	28
22.00	26.00	6.00	4.00	230	260	56	32
26.00	30.00	7.00	5.00	260	290	63	32
				290	330	70	36
				330	380	80	40
				380	440	90	45
				440	500	100	50

Note: Key sizes above the horizontal line are square; others are rectangular.

28. Compute the minimum required length of the key
    (a) If the material of the key is weakest and you use a square key then use

$$L_{\min} = \frac{4TN}{DWs_y}$$

T = torque at that point on the shaft
D = shaft diameter
N = safety factor. Assume 3 if not given
W = the width of the key

    (b) If a rectangular key is used or the shaft is the weakest material, use the largest value between the following equations

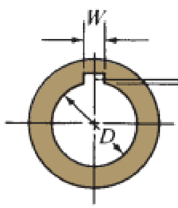
$$L_{\min} = \frac{4TN}{DHs_y}$$

$$L_{\min} = \frac{4TN}{DWs_y}$$

29. Check that your key is a suitable length. Anything larger than 2" may be a little large and you should consider increasing your shaft diameter to compensate. 1-1.5 inches is a good number of inches. The prof suggested under that 1 inch is best in his shaft design video. Or you could say fuck it and not care.

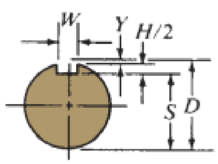
30. Round the key length up to the nearest standard size

31. Compute the measurements for the keyseat and the keyway in the hub



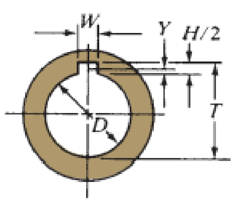
(*a*) Chordal height

$$Y = \frac{D - \sqrt{D^2 - W^2}}{2}$$



(*b*) Depth of shaft keyseat

$$S = D - Y - \frac{H}{2} = \frac{D - H + \sqrt{D^2 - W^2}}{2}$$



(*c*) Depth of hub keyseat

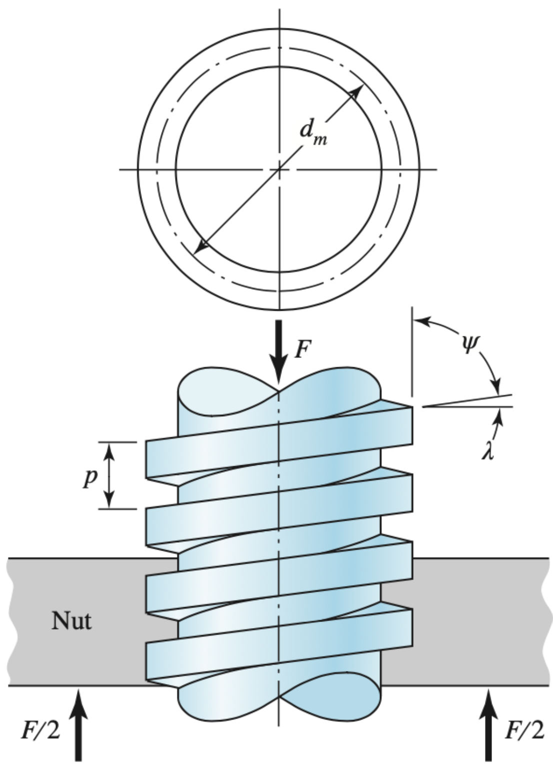
$$T = D - Y + \frac{H}{2} + C = \frac{D + H + \sqrt{D^2 - W^2}}{2} + C$$

Symbols
- C = Allowance
  - $+0.005$-in clearance for parallel keys
  - -0.020-in interference for taper keys
- D = Nominal shaft or bore dia., in
- H = Nominal key height, in
- W = Nominal key width, in
- Y = Chordal height, in

5 Springies, Bolties, Screwies (Boing Boing)

5.1 Power Screws

5.1.1 Anatomy



5.1.2 Design Selection

1. Calculate the required tensile stress area A_t in in^2:

$$A_t = \frac{F}{\sigma_d}$$

where F is the load applied to the screw in lbs,
σ_d is maximum tensile stress in psi

2. Select a screw from one of the following tables (based on your screw type) that would provide a tensile stress area A_t greater or equal to the one calculated above.
Take note of the rated A_s.

TABLE 17–1 Preferred Acme Screw Threads

Nominal major diameter, D (in)	Threads per in, n	Pitch, $p = 1/n$ (in)	Minimum minor diameter, D_r (in)	Minimum pitch diameter, D_p (in)	Tensile stress area, A_t (in^2)	Shear stress area, A_s (in^2)[a]
1/4	16	0.0625	0.1618	0.2043	0.026 32	0.3355
5/16	14	0.0714	0.2140	0.2614	0.044 38	0.4344
3/8	12	0.0833	0.2632	0.3161	0.065 89	0.5276
7/16	12	0.0833	0.3253	0.3783	0.097 20	0.6396
1/2	10	0.1000	0.3594	0.4306	0.1225	0.7278
5/8	8	0.1250	0.4570	0.5408	0.1955	0.9180
3/4	6	0.1667	0.5371	0.6424	0.2732	1.084
7/8	6	0.1667	0.6615	0.7663	0.4003	1.313
1	5	0.2000	0.7509	0.8726	0.5175	1.493
$1\frac{1}{8}$	5	0.2000	0.8753	0.9967	0.6881	1.722
$1\frac{1}{4}$	5	0.2000	0.9998	1.1210	0.8831	1.952
$1\frac{3}{8}$	4	0.2500	1.0719	1.2188	1.030	2.110
$1\frac{1}{2}$	4	0.2500	1.1965	1.3429	1.266	2.341
$1\frac{3}{4}$	4	0.2500	1.4456	1.5916	1.811	2.803
2	4	0.2500	1.6948	1.8402	2.454	3.262
$2\frac{1}{4}$	3	0.3333	1.8572	2.0450	2.982	3.610
$2\frac{1}{2}$	3	0.3333	2.1065	2.2939	3.802	4.075
$2\frac{3}{4}$	3	0.3333	2.3558	2.5427	4.711	4.538
3	2	0.5000	2.4326	2.7044	5.181	4.757
$3\frac{1}{2}$	2	0.5000	2.9314	3.2026	7.388	5.700
4	2	0.5000	3.4302	3.7008	9.985	6.640
$4\frac{1}{2}$	2	0.5000	3.9291	4.1991	12.972	7.577
5	2	0.5000	4.4281	4.6973	16.351	8.511

[a]Per inch of length of engagement.

TABLE 17–1M Examples of Power Screws with Metric Trapezoidal Screw Thread

ISO thread system—External threads

Major diameter, D (mm)	Pitch, p (mm)	Pitch diameter, D_p (mm)	Minor diameter, D_r (mm)	Tensile stress area (mm^2)
8	1.5	7.25	6.2	35.52
10	2	9.0	7.5	53.46
12	3	10.5	8.5	70.88
14	3	12.5	10.5	103.9
16	3	14.5	12.5	143.1
20	4	18.0	15.5	220.4
22	5	19.5	16.5	254.5
24	5	21.5	18.5	314.2
28	5	25.5	22.5	452.4
30	6	27.0	23.0	490.9
32	6	29.0	33.0	754.8
36	6	33.0	29.0	754.8
40	7	36.5	32.0	921.3
46	8	42.0	37.0	1225
50	8	46.0	41.0	1486
55	9	50.5	45.0	1791
60	9	55.5	50.0	2185
70	10	65.0	59.0	3019
80	10	75.0	69.0	4072
90	12	84.0	77.0	5090
100	12	94.0	87.0	6433
120	14	113.0	104.0	9246
125	14	122.0	109.0	10 477

3. Calculate the required length h of the nut/yoke in inches to maintain shear stress below the maximum shear stress.

   (a) Calculate the required shear area A_s in in^2 and call it:

$$A_s = \frac{F}{\tau_d}$$

where F is the load applied to the screw in lbs,

τ_d is the maximum shear stress in psi

(b) the required length h is:

$$h = A_{sc} \frac{1 \text{ in}}{A_{sr}}$$

where A_{sc} is the calculate shear area,

A_{sr} is the rated shear area obtained from the table

- Round up to the nearest 1/4th of an inch. For example, if you obtain 1.10 in, round up to 1.25 in. If you obtain 1.58, round up to 1.75 in, etc.

4. Compute the lead angle λ using the following equation:

$$\lambda = \tan^{-1} \frac{p}{\pi D_p}$$

where p is the pitch (obtained from the table) in inches,

D_p is the pitch diameter (obtained from the table) in inches

5. Determine the raising and lowering torque (T_u and T_d respectively, in lb $\cdot$ in).

(a) The equation to compute the raising torque is:

$$T_u = \frac{FD_p}{2} \left[\frac{\cos\phi \tan\lambda + f}{\cos\phi - f \tan\lambda} \right]$$

where λ is the lead angle,

f is the friction coefficient,

F is the load applied to the screw in lbs,

D_p is the pitch diameter (obtained from the table) in inches,

ϕ is the thread angle

- Square threads: $\phi = 0°$
- Acme threads: $\phi = 14.5°$
- Trapezoidal threads: $\phi = 15°$

(b) The equation to compute the lowering torque is:

$$T_d = \frac{FD_p}{2} \left[\frac{f - \cos\phi \tan\lambda}{\cos\phi + f \tan\lambda} \right]$$

(c) If a collar friction is given, then the formulas for T_u and T_d are changed:

$$T_u = \frac{FD_p}{2} \left[\frac{\cos\phi \tan\lambda + f}{\cos\phi - f \tan\lambda} \right] + f_c F R_c$$

$$T_d = \frac{FD_p}{2} \left[\frac{f - \cos\phi \tan\lambda}{\cos\phi + f \tan\lambda} \right] + f_c F R_c$$

where f_c is the collar friction,

R_C is the collar friction radius

Note that if there is a roller bearing then you can assume that the collar friction is zero (unless stated otherwise).

6. Compute the efficiency e of the power screw:

$$e = \frac{Fp}{2\pi T_u}$$

where F is the load applied to the screw in lbs,
p is the pitch in inches,
T_u is the raising torque in $\text{lb} \cdot \text{in}$

7. Calculate the maximum speed you can raise a load given an input power in hp.

   (a) Find the rotational speed of the screw (in rpm) using:

   $$n = \frac{63000P}{T}$$

   where P is the input power in hp,
   T is the raising torque in $\text{lb} \cdot \text{in}$

   (b) Convert to linear speed (in in/s) using the following relation:

   $$V = \left(\frac{n \text{ rev}}{\text{min}}\right)\left(\frac{p \text{ in}}{\text{rev}}\right)\left(\frac{\text{min}}{60 \text{ sec}}\right)$$

   where n is the rotational speed in rpm,
   p is the pitch in in

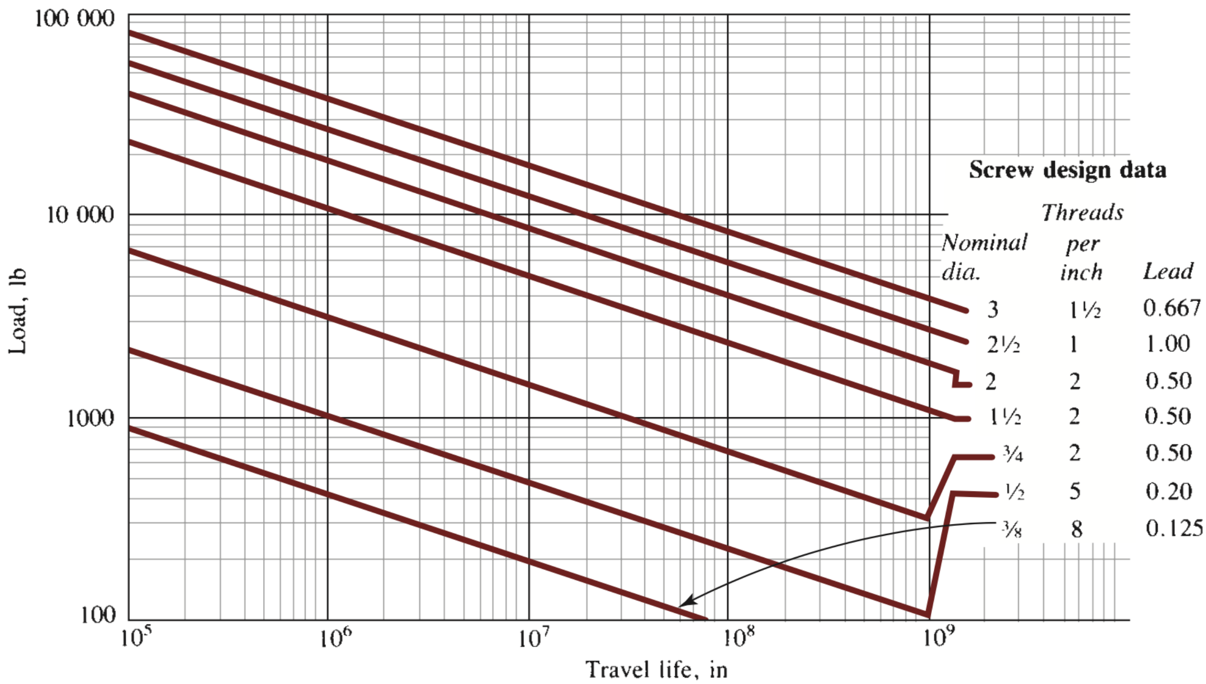
5.2 Ball Screws

5.2.1 Design Selection

1. Compute the travel life in inches:

$$\text{Travel Life} = \left(\frac{\text{distance (in)}}{\text{cycle}}\right)\left(\frac{\text{\# of cycles}}{\text{hour}}\right)\left(\frac{\text{24 hours}}{\text{day}}\right)\left(\frac{\text{365 days}}{\text{year}}\right)(\text{\# of years})$$

2. Select a ball screw based on load and travel life:



3. Compute the torque (in lb $\cdot$ in required to drive the screw using the following equation:

$$T_u = 0.177FL$$

where F is the applied load in lbs,
L is the lead (from the graph above) in inches

4. Compute the power required (in hp) given a travel speed:

   (a) Calculate the rotational speed required (in rpm):

$$n = (V)\left(\frac{1 \text{ rev}}{L}\right)\left(\frac{60 \text{ sec}}{\text{min}}\right)$$

where V is the travel speed in in/s,
L is the lead in inches

   (b) The power required (in hp) is the following:

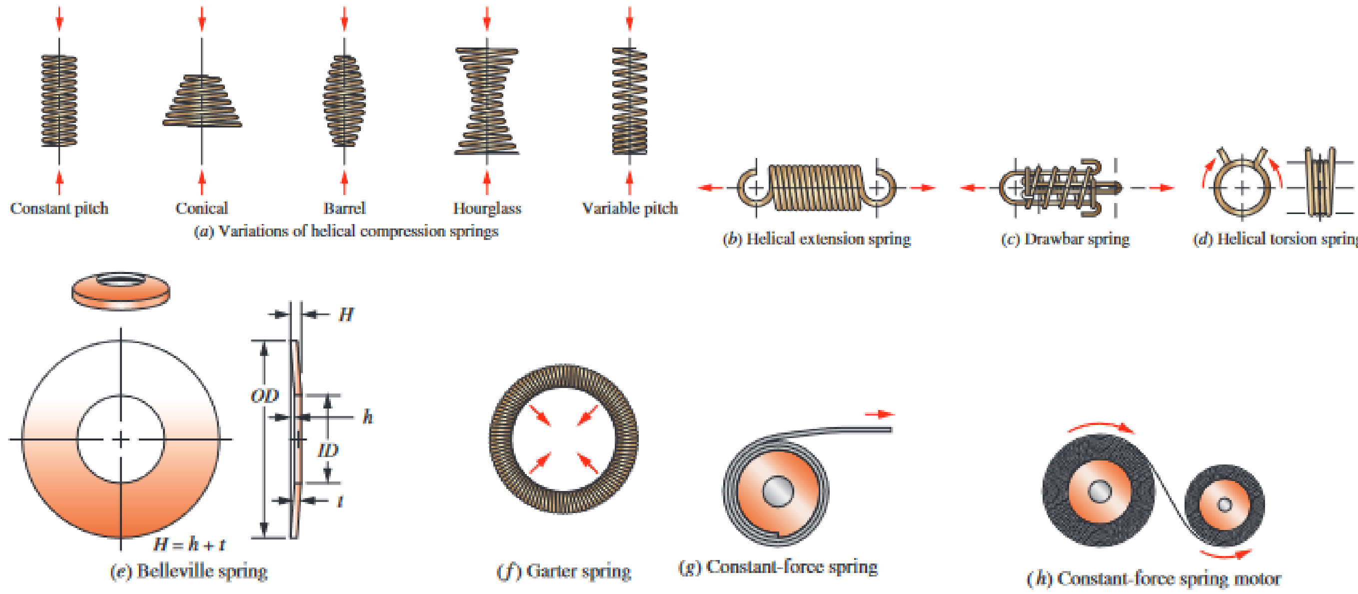
$$P = \frac{Tn}{63000}$$

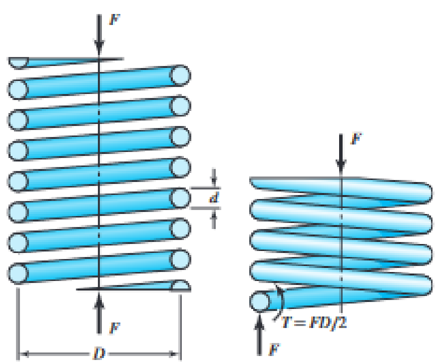
where T is the required torque in lb $\cdot$ in,
n is the rotational speed in rpm

5.3 Springs

This section is best for Static Helical Springs, Chapter 10 of shigley has pretty good depth on all spring types if we get something else

5.3.1 Anatomy

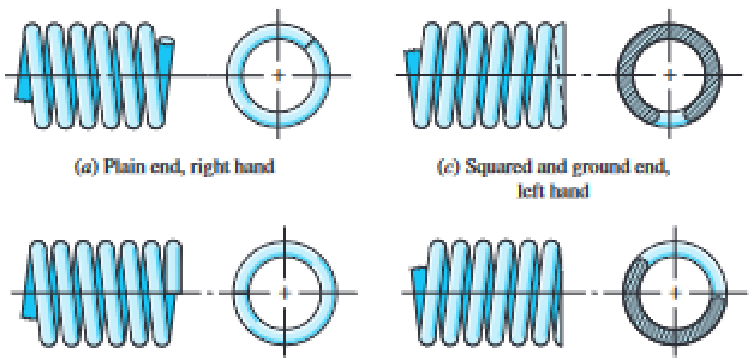




$D =$ Mean Diameter (Distance to centre of wire)
$d =$ Wire Diameter (Distance to centre of wire)
Outer Diameter $= OD = D + d$
Inner Diameter $= ID = D - d$
Max Shear Stress $= \tau_{max} = \frac{Tr}{J} + \frac{F}{A}$
This one is the above one with all subs built in: $\tau = \frac{8FD}{\pi d^3} + \frac{4F}{\pi d^2}$
Spring index (best between 4 and 12) $= C = \frac{D}{d}$
Changing that bitch again: $\tau = K_s \frac{8FD}{\pi d^3}$
How could you forget K_s is stress-correction factor: $K_s = \frac{2C+1}{2C}$
This shit is driving me up the Wahl factor: $K_W = \frac{4C-1}{4C-4} + \frac{0.615}{C}$
But of course we prefer Bergstässer Factor: $K_B = \frac{4C+2}{4C-3}$
Why do we need to account for Curvature factor? Its supposed to be curved: $K_C = \frac{K_B}{K_s} = \frac{2C(4C+2)}{(4C-3)(2C+1)}$
Most of those don't matter just use: $\tau = K_B \frac{8FD}{\pi d^3}$

Oh shit we got these using Castigliano's, wish I paid more attention in 360

$N =$ Number of coils
$N_a =$ Number of active coils
Ignore this one apparently: $U = \frac{T^2 l}{2GJ} + \frac{F^2 l}{2AG}$
This one for sure though (also $N = N_a$ idk why): $U = \frac{4F^2D^3N}{d^4G} + \frac{2F^2DN}{d^2G}$
Total deflection: $= \frac{\partial U}{\partial F} = \frac{8FD^3N}{d^4G} + \frac{4FDN}{d^2G} = \frac{8FD^3N}{d^4G}(1 + \frac{1}{2C^2}) \approx \frac{8FD^3N}{d^4G}$
Spring constant: $k = \frac{F}{y} \approx \frac{d^4G}{8D^3N}$



(*a*) Plain end, right hand

(*c*) Squared and ground end, left hand

(*b*) Squared or closed end, right hand

(*d*) Plain end, ground, left hand

Table 10–1 Formulas for the Dimensional Characteristics of Compression Springs (N_a = Number of Active Coils)

		Type of Spring Ends		
Term	**Plain**	**Plain and Ground**	**Squared or Closed**	**Squared and Ground**
End coils, N_e	0	1	2	2
Total coils, N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length, L_0	$pN_a + d$	$p(N_a + 1)$	$pN_a + 3d$	$pN_a + 2d$
Solid length, L_s	$d(N_t + 1)$	dN_t	$d(N_t + 1)$	dN_t
Pitch, p	$(L_0 - d)/N_a$	$L_0/(N_a + 1)$	$(L_0 - 3d)/N_a$	$(L_0 - 2d)/N_a$

Critical deflection: $y_{cr} = L_0 C_1'[1 - (1 - \frac{C_2'}{\lambda_{eff}^2})^{1/2}$

Table 10–2 End-Condition Constants α for Helical Compression Springs*

End Condition	Constant α
Spring supported between flat parallel surfaces (fixed ends)	0.5
One end supported by flat surface perpendicular to spring axis (fixed); other end pivoted (hinged)	0.707
Both ends pivoted (hinged)	1
One end clamped; other end free	2

*Ends supported by flat surfaces must be squared and ground.

Effective slenderness ratio: $\lambda_{eff} = \frac{\alpha L_0}{D}$

Book really said "random elastic constants": $C_1' = \frac{E}{2(E-G)}$, $C_2' = \frac{2\pi^2(E-G)}{2G+E}$

Absolute stability occurs if: $L_0 < \frac{\pi D}{\alpha}[\frac{2(E-G)}{2G+E}]^{1/2}$, But for steels: $L_O < 2.63\frac{D}{\alpha}$

Table 10–3 High-Carbon and Alloy Spring Steels

Name of Material	Similar Specifications	Description
Music wire, 0.80–0.95*C*	UNS G10850 AISI 1085 ASTM A228-51	This is the best, toughest, and most widely used of all spring materials for small springs. It has the highest tensile strength and can withstand higher stresses under repeated loading than any other spring material. Available in diameters 0.12 to 3 mm (0.005 to 0.125 in). Do not use above 120°C (250°F) or at subzero temperatures.
Oil-tempered wire, 0.60–0.70*C*	UNS G10650 AISI 1065 ASTM 229-41	This general-purpose spring steel is used for many types of coil springs where the cost of music wire is prohibitive and in sizes larger than available in music wire. Not for shock or impact loading. Available in diameters 3 to 12 mm (0.125 to 0.5000 in), but larger and smaller sizes may be obtained. Not for use above 180°C (350°F) or at subzero temperatures.
Hard-drawn wire, 0.60–0.70*C*	UNS G10660 AISI 1066 ASTM A227-47	This is the cheapest general-purpose spring steel and should be used only where life, accuracy, and deflection are not too important. Available in diameters 0.8 to 12 mm (0.031 to 0.500 in). Not for use above 120°C (250°F) or at subzero temperatures.
Chrome-vanadium	UNS G61500 AISI 6150 ASTM 231-41	This is the most popular alloy spring steel for conditions involving higher stresses than can be used with the high-carbon steels and for use where fatigue resistance and long endurance are needed. Also good for shock and impact loads. Widely used for aircraft-engine valve springs and for temperatures to 220°C (425°F). Available in annealed or pretempered sizes 0.8 to 12 mm (0.031 to 0.500 in) in diameter.
Chrome-silicon	UNS G92540 AISI 9254	This alloy is an excellent material for highly stressed springs that require long life and are subjected to shock loading. Rockwell hardnesses of C50 to C53 are quite common, and the material may be used up to 250°C (475°F). Available from 0.8 to 12 mm (0.031 to 0.500 in) in diameter.

Table 10–4 Constants A and m of $S_{ut} = A/d^m$ for Estimating Minimum Tensile Strength of Common Spring Wires

Material	ASTM No.	Exponent m	Diameter, in	A, kpsi · in^m	Diameter, mm	A, MPa · mm^m	Relative Cost of Wire
Music wire*	A228	0.145	0.004–0.256	201	0.10–6.5	2211	2.6
OQ&T wire†	A229	0.187	0.020–0.500	147	0.5–12.7	1855	1.3
Hard-drawn wire‡	A227	0.190	0.028–0.500	140	0.7–12.7	1783	1.0
Chrome-vanadium wire§	A232	0.168	0.032–0.437	169	0.8–11.1	2005	3.1
Chrome-silicon wire‖	A401	0.108	0.063–0.375	202	1.6–9.5	1974	4.0
302 Stainless wire#	A313	0.146	0.013–0.10	169	0.3–2.5	1867	7.6–11
		0.263	0.10–0.20	128	2.5–5	2065	
		0.478	0.20–0.40	90	5–10	2911	
Phosphor-bronze wire**	B159	0	0.004–0.022	145	0.1–0.6	1000	8.0
		0.028	0.022–0.075	121	0.6–2	913	
		0.064	0.075–0.30	110	2–7.5	932	

*Surface is smooth, free of defects, and has a bright, lustrous finish.
†Has a slight heat-treating scale which must be removed before plating.
‡Surface is smooth and bright with no visible marks.
§Aircraft-quality tempered wire, can also be obtained annealed.
‖Tempered to Rockwell C49, but may be obtained untempered.
#Type 302 stainless steel.
**Temper CA510.

Ultimate Tensile Strength: $S_{ut} = \frac{A}{d^m}$
Torsional yield strength: $S_{sy} = 0.577 S_y$ same as $0.35 S_{ut} \leq S_{sy} \leq 0.52 S_{ut}$

Table 10–5 Mechanical Properties of Some Spring Wires

Material	Elastic Limit, Percent of S_{ut} Tension	Elastic Limit, Percent of S_{ut} Torsion	Diameter d, in	E Mpsi	E GPa	G Mpsi	G GPa
Music wire A228	65–75	45–60	<0.032	29.5	203.4	12.0	82.7
			0.033–0.063	29.0	200	11.85	81.7
			0.064–0.125	28.5	196.5	11.75	81.0
			>0.125	28.0	193	11.6	80.0
HD spring A227	60–70	45–55	<0.032	28.8	198.6	11.7	80.7
			0.033–0.063	28.7	197.9	11.6	80.0
			0.064–0.125	28.6	197.2	11.5	79.3
			>0.125	28.5	196.5	11.4	78.6
Oil tempered A239	85–90	45–50		28.5	196.5	11.2	77.2
Valve spring A230	85–90	50–60		29.5	203.4	11.2	77.2
Chrome-vanadium A231	88–93	65–75		29.5	203.4	11.2	77.2
A232	88–93			29.5	203.4	11.2	77.2
Chrome-silicon A401	85–93	65–75		29.5	203.4	11.2	77.2
Stainless steel							
A313*	65–75	45–55		28	193	10	69.0
17-7PH	75–80	55–60		29.5	208.4	11	75.8
414	65–70	42–55		29	200	11.2	77.2
420	65–75	45–55		29	200	11.2	77.2
431	72–76	50–55		30	206	11.5	79.3
Phosphor-bronze B159	75–80	45–50		15	103.4	6	41.4
Beryllium-copper B197	70	50		17	117.2	6.5	44.8
Inconel alloy X-750	65–70	40–45		31	213.7	11.2	77.2

*Also includes 302, 304, and 316.

Table 10–6 Maximum Allowable Torsional Stresses for Helical Compression Springs in Static Applications

Material	Maximum Percent of Tensile Strength: Before Set Removed (includes K_W or K_B)	Maximum Percent of Tensile Strength: After Set Removed (includes K_s)
Music wire and cold-drawn carbon steel	45	60–70
Hardened and tempered carbon and low-alloy steel	50	65–75
Austenitic stainless steels	35	55–65
Nonferrous alloys	35	55–65

$S_{sy} = \tau_{all} = 0.56 S_{ut}$

Table A–28 **Decimal Equivalents of Wire and Sheet-Metal Gauges* (All Sizes Are Given in Inches)**

Name of Gauge:	American or Brown & Sharpe	Birmingham or Stubs Iron Wire	United States Standard†	Manufacturers Standard	Steel Wire or Washburn & Moen	Music Wire	Stubs Steel Wire	Twist Drill
Principal Use:	**Nonferrous Sheet, Wire, and Rod**	**Tubing, Ferrous Strip, Flat Wire, and Spring Steel**	**Ferrous Sheet and Plate, 480 lbf/ft^2**	**Ferrous Sheet**	**Ferrous Wire Except Music Wire**	**Music Wire**	**Steel Drill Rod**	**Twist Drills and Drill Steel**
7/0			0.500		0.490			
6/0	0.580 0		0.468 75		0.461 5	0.004		
5/0	0.516 5		0.437 5		0.430 5	0.005		
4/0	0.460 0	0.454	0.406 25		0.393 8	0.006		
3/0	0.409 6	0.425	0.375		0.362 5	0.007		
2/0	0.364 8	0.380	0.343 75		0.331 0	0.008		
0	0.324 9	0.340	0.312 5		0.306 5	0.009		
1	0.289 3	0.300	0.281 25		0.283 0	0.010	0.227	0.228 0
2	0.257 6	0.284	0.265 625		0.262 5	0.011	0.219	0.221 0
3	0.229 4	0.259	0.25	0.239 1	0.243 7	0.012	0.212	0.213 0
4	0.204 3	0.238	0.234 375	0.224 2	0.225 3	0.013	0.207	0.209 0
5	0.181 9	0.220	0.218 75	0.209 2	0.207 0	0.014	0.204	0.205 5
6	0.162 0	0.203	0.203 125	0.194 3	0.192 0	0.016	0.201	0.204 0
7	0.144 3	0.180	0.187 5	0.179 3	0.177 0	0.018	0.199	0.201 0
8	0.128 5	0.165	0.171 875	0.164 4	0.162 0	0.020	0.197	0.199 0
9	0.114 4	0.148	0.156 25	0.149 5	0.148 3	0.022	0.194	0.196 0
10	0.101 9	0.134	0.140 625	0.134 5	0.135 0	0.024	0.191	0.193 5
11	0.090 74	0.120	0.125	0.119 6	0.120 5	0.026	0.188	0.191 0
12	0.080 81	0.109	0.109 357	0.104 6	0.105 5	0.029	0.185	0.189 0
13	0.071 96	0.095	0.093 75	0.089 7	0.091 5	0.031	0.182	0.185 0
14	0.064 08	0.083	0.078 125	0.074 7	0.080 0	0.033	0.180	0.182 0
15	0.057 07	0.072	0.070 312 5	0.067 3	0.072 0	0.035	0.178	0.180 0
16	0.050 82	0.065	0.062 5	0.059 8	0.062 5	0.037	0.175	0.177 0
17	0.045 26	0.058	0.056 25	0.053 8	0.054 0	0.039	0.172	0.173 0

Table A–28 Decimal Equivalents of Wire and Sheet-Metal Gauges* (All Sizes Are Given in Inches) ***(Continued)***

Name of Gauge:	American or Brown & Sharpe	Birmingham or Stubs Iron Wire	United States Standard†	Manu-facturers Standard	Steel Wire or Washburn & Moen	Music Wire	Stubs Steel Wire	Twist Drill
Principal Use:	**Nonferrous Sheet, Wire, and Rod**	**Tubing, Ferrous Strip, Flat Wire, and Spring Steel**	**Ferrous Sheet and Plate, 480 lbf/ft^3**	**Ferrous Sheet**	**Ferrous Wire Except Music Wire**	**Music Wire**	**Steel Drill Rod**	**Twist Drills and Drill Steel**
18	0.040 30	0.049	0.05	0.047 8	0.047 5	0.041	0.168	0.169 5
19	0.035 89	0.042	0.043 75	0.041 8	0.041 0	0.043	0.164	0.166 0
20	0.031 96	0.035	0.037 5	0.035 9	0.034 8	0.045	0.161	0.161 0
21	0.028 46	0.032	0.034 375	0.032 9	0.031 7	0.047	0.157	0.159 0
22	0.025 35	0.028	0.031 25	0.029 9	0.028 6	0.049	0.155	0.157 0
23	0.022 57	0.025	0.028 125	0.026 9	0.025 8	0.051	0.153	0.154 0
24	0.020 10	0.022	0.025	0.023 9	0.023 0	0.055	0.151	0.152 0
25	0.017 90	0.020	0.021 875	0.020 9	0.020 4	0.059	0.148	0.149 5
26	0.015 94	0.018	0.018 75	0.017 9	0.018 1	0.063	0.146	0.147 0
27	0.014 20	0.016	0.017 187 5	0.016 4	0.017 3	0.067	0.143	0.144 0
28	0.012 64	0.014	0.015 625	0.014 9	0.016 2	0.071	0.139	0.140 5
29	0.011 26	0.013	0.014 062 5	0.013 5	0.015 0	0.075	0.134	0.136 0
30	0.010 03	0.012	0.012 5	0.012 0	0.014 0	0.080	0.127	0.128 5
31	0.008 928	0.010	0.010 937 5	0.010 5	0.013 2	0.085	0.120	0.120 0
32	0.007 950	0.009	0.010 156 25	0.009 7	0.012 8	0.090	0.115	0.116 0
33	0.007 080	0.008	0.009 375	0.009 0	0.011 8	0.095	0.112	0.113 0
34	0.006 305	0.007	0.008 593 75	0.008 2	0.010 4		0.110	0.111 0
35	0.005 615	0.005	0.007 812 5	0.007 5	0.009 5		0.108	0.110 0
36	0.005 000	0.004	0.007 031 25	0.006 7	0.009 0		0.106	0.106 5
37	0.004 453		0.006 640 625	0.006 4	0.008 5		0.103	0.104 0
38	0.003 965		0.006 25	0.006 0	0.008 0		0.101	0.101 5
39	0.003 531				0.007 5		0.099	0.099 5
40	0.003 145				0.007 0		0.097	0.098 0

*Specify sheet, wire, and plate by stating the gauge number, the gauge name, and the decimal equivalent in parentheses.

†Reflects present average and weights of sheet steel.

$\frac{\partial^2 u}{\partial x^2} = \frac{W}{kgl^2}\frac{\partial^2 u}{\partial t^2} // \; g =$ acceleration from gravity,$l =$ length of spring between plates,$W =$ weight of spring

Fundamental Frequencies: $\omega = m\pi\sqrt{\frac{kg}{W}}$, $m = 1, 2, 3, ...$

Both ends always contact plates: $f = \frac{1}{2}\sqrt{\frac{kg}{W}}$

One end always contacts plate: $f = \frac{1}{4}\sqrt{\frac{kg}{W}}$ (also works if being driven by a sine wave machine

$W = AL\gamma = \frac{\pi^2 d^2 D N_a \gamma}{4}$ Where γ is specific weight

Always make sure f_{cr} is 15-20 times higher than operating frequency to avoid resonance

I don't understand the next two lines

Unpeened: $S_{sa} = 35kpsi(241MPa)$, $S_{sm} = 55kpsi(379MPa)$

Peened: $S_{sa} = 57.5kpsi(398MPa)$, $S_{sm} = 77.5kpsi(534MPa)$

Goodman Criteria: $S_{se} = \frac{S_{sa}}{1-\frac{S_{sm}}{S_{su}}}$

Gerber Criteria: $S_{se} = \frac{S_{sa}}{1-(\frac{S_{sm}}{S_{su}})^2}$

Sines Failure Criterion: $S_{su} = 0.67S_{ut}$

$F_a = \frac{F_{max}-F_{min}}{2}$

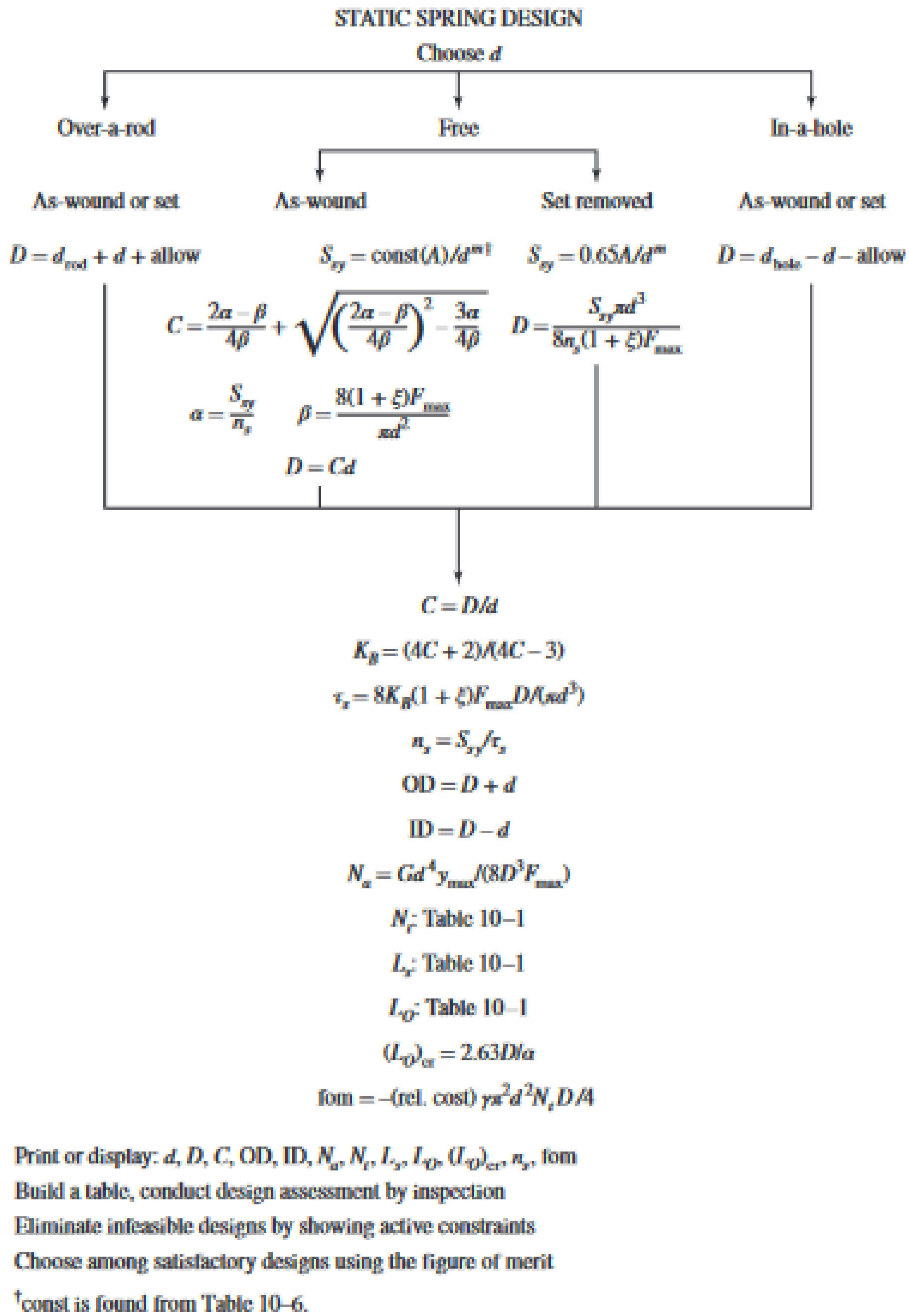
$F_m = \frac{F_{max}+F_{min}}{2}$

$\tau_a = K_B\frac{8F_aD}{\pi d^3}$

$\tau_m = K_B \frac{8F_m D}{\pi d^3}$

5.3.2 Design for Static Service

Design requirements: $4 \le C \le 12, 3 \le N_a \le 15, \xi \ge 0.15, n_s \ge 1.2$ Note n_s is the safety factor at solid height fom $= -$(relative material cost$\frac{\gamma\pi^2 d^2 N_t D}{4}$



Recall: $\tau = \frac{S_{sy}}{n_s} = K_B \frac{8F_s D}{\pi d^3} = \frac{4C+2}{4C-3}[\frac{8(1+\xi)F_{max}C}{\pi d^2}$

Let: $\alpha = \frac{S_{sy}}{n_s}$ and $\beta = \frac{8(1+\xi)F_{max}}{\pi d^2}$

$\therefore\ C = \frac{2\alpha-\beta}{4\beta} + \sqrt{(\frac{2\alpha-\beta}{4\beta})^2 - \frac{3\alpha}{4\beta}}$ this gives two answers, always pick the larger

5.3.3 Iterate wire diameter

- Gather needed info:
  Based on the wire you need, use these tables to pick an A, m E and G value (you can assume $d > 0.064in$ I think):

Table 10–4 Constants A and m of $S_{ut} = A/d^m$ for Estimating Minimum Tensile Strength of Common Spring Wires

Material	ASTM No.	Exponent m	Diameter, in	A, kpsi · in^m	Diameter, mm	A, MPa · mm^m	Relative Cost of Wire
Music wire[*]	A228	0.145	0.004–0.256	201	0.10–6.5	2211	2.6
OQ&T wire[†]	A229	0.187	0.020–0.500	147	0.5–12.7	1855	1.3
Hard-drawn wire[‡]	A227	0.190	0.028–0.500	140	0.7–12.7	1783	1.0
Chrome-vanadium wire[§]	A232	0.168	0.032–0.437	169	0.8–11.1	2005	3.1
Chrome-silicon wire[‖]	A401	0.108	0.063–0.375	202	1.6–9.5	1974	4.0
302 Stainless wire[#]	A313	0.146	0.013–0.10	169	0.3–2.5	1867	7.6–11
		0.263	0.10–0.20	128	2.5–5	2065	
		0.478	0.20–0.40	90	5–10	2911	
Phosphor-bronze wire[**]	B159	0	0.004–0.022	145	0.1–0.6	1000	8.0
		0.028	0.022–0.075	121	0.6–2	913	
		0.064	0.075–0.30	110	2–7.5	932	

[*]Surface is smooth, free of defects, and has a bright, lustrous finish.
[†]Has a slight heat-treating scale which must be removed before plating.
[‡]Surface is smooth and bright with no visible marks.
[§]Aircraft-quality tempered wire, can also be obtained annealed.
[‖]Tempered to Rockwell C49, but may be obtained untempered.
[#]Type 302 stainless steel.
[**]Temper CA510.

Table 10–5 Mechanical Properties of Some Spring Wires

Material	Elastic Limit, Percent of S_{ut} Tension	Elastic Limit, Percent of S_{ut} Torsion	Diameter d, in	E Mpsi	E GPa	G Mpsi	G GPa
Music wire A228	65–75	45–60	<0.032	29.5	203.4	12.0	82.7
			0.033–0.063	29.0	200	11.85	81.7
			0.064–0.125	28.5	196.5	11.75	81.0
			>0.125	28.0	193	11.6	80.0
HD spring A227	60–70	45–55	<0.032	28.8	198.6	11.7	80.7
			0.033–0.063	28.7	197.9	11.6	80.0
			0.064–0.125	28.6	197.2	11.5	79.3
			>0.125	28.5	196.5	11.4	78.6
Oil tempered A239	85–90	45–50		28.5	196.5	11.2	77.2
Valve spring A230	85–90	50–60		29.5	203.4	11.2	77.2
Chrome-vanadium A231	88–93	65–75		29.5	203.4	11.2	77.2
A232	88–93			29.5	203.4	11.2	77.2
Chrome-silicon A401	85–93	65–75		29.5	203.4	11.2	77.2
Stainless steel							
A313*	65–75	45–55		28	193	10	69.0
17-7PH	75–80	55–60		29.5	208.4	11	75.8
414	65–70	42–55		29	200	11.2	77.2
420	65–75	45–55		29	200	11.2	77.2
431	72–76	50–55		30	206	11.5	79.3
Phosphor-bronze B159	75–80	45–50		15	103.4	6	41.4
Beryllium-copper B197	70	50		17	117.2	6.5	44.8
Inconel alloy X-750	65–70	40–45		31	213.7	11.2	77.2

*Also includes 302, 304, and 316.

- random constants
  $n_s = 1.2$ at solid height as a start, you will check this again later
  $\xi = 0.15$

- get the percentage of S_{sy} you are allowed from this table:

Table 10–6 **Maximum Allowable Torsional Stresses for Helical Compression Springs in Static Applications**

	Maximum Percent of Tensile Strength	
Material	**Before Set Removed (includes K_W or K_B)**	**After Set Removed (includes K_s)**
Music wire and cold-drawn carbon steel	45	60–70
Hardened and tempered carbon and low-alloy steel	50	65–75
Austenitic stainless steels	35	55–65
Nonferrous alloys	35	55–65

- Choose a range of wire diameters (d) from Table A-28 that you think will work:

Table A–28 **Decimal Equivalents of Wire and Sheet-Metal Gauges* (All Sizes Are Given in Inches)**

Name of Gauge:	American or Brown & Sharpe	Birmingham or Stubs Iron Wire	United States Standard†	Manu-facturers Standard	Steel Wire or Washburn & Moen	Music Wire	Stubs Steel Wire	Twist Drill
Principal Use:	**Nonferrous Sheet, Wire, and Rod**	**Tubing, Ferrous Strip, Flat Wire, and Spring Steel**	**Ferrous Sheet and Plate, 480 lbf/ft^3**	**Ferrous Sheet**	**Ferrous Wire Except Music Wire**	**Music Wire**	**Steel Drill Rod**	**Twist Drills and Drill Steel**
7/0			0.500		0.490			
6/0	0.580 0		0.468 75		0.461 5	0.004		
5/0	0.516 5		0.437 5		0.430 5	0.005		
4/0	0.460 0	0.454	0.406 25		0.393 8	0.006		
3/0	0.409 6	0.425	0.375		0.362 5	0.007		
2/0	0.364 8	0.380	0.343 75		0.331 0	0.008		
0	0.324 9	0.340	0.312 5		0.306 5	0.009		
1	0.289 3	0.300	0.281 25		0.283 0	0.010	0.227	0.228 0
2	0.257 6	0.284	0.265 625		0.262 5	0.011	0.219	0.221 0
3	0.229 4	0.259	0.25	0.239 1	0.243 7	0.012	0.212	0.213 0
4	0.204 3	0.238	0.234 375	0.224 2	0.225 3	0.013	0.207	0.209 0
5	0.181 9	0.220	0.218 75	0.209 2	0.207 0	0.014	0.204	0.205 5
6	0.162 0	0.203	0.203 125	0.194 3	0.192 0	0.016	0.201	0.204 0
7	0.144 3	0.180	0.187 5	0.179 3	0.177 0	0.018	0.199	0.201 0
8	0.128 5	0.165	0.171 875	0.164 4	0.162 0	0.020	0.197	0.199 0
9	0.114 4	0.148	0.156 25	0.149 5	0.148 3	0.022	0.194	0.196 0
10	0.101 9	0.134	0.140 625	0.134 5	0.135 0	0.024	0.191	0.193 5
11	0.090 74	0.120	0.125	0.119 6	0.120 5	0.026	0.188	0.191 0
12	0.080 81	0.109	0.109 357	0.104 6	0.105 5	0.029	0.185	0.189 0
13	0.071 96	0.095	0.093 75	0.089 7	0.091 5	0.031	0.182	0.185 0
14	0.064 08	0.083	0.078 125	0.074 7	0.080 0	0.033	0.180	0.182 0
15	0.057 07	0.072	0.070 312 5	0.067 3	0.072 0	0.035	0.178	0.180 0
16	0.050 82	0.065	0.062 5	0.059 8	0.062 5	0.037	0.175	0.177 0
17	0.045 26	0.058	0.056 25	0.053 8	0.054 0	0.039	0.172	0.173 0

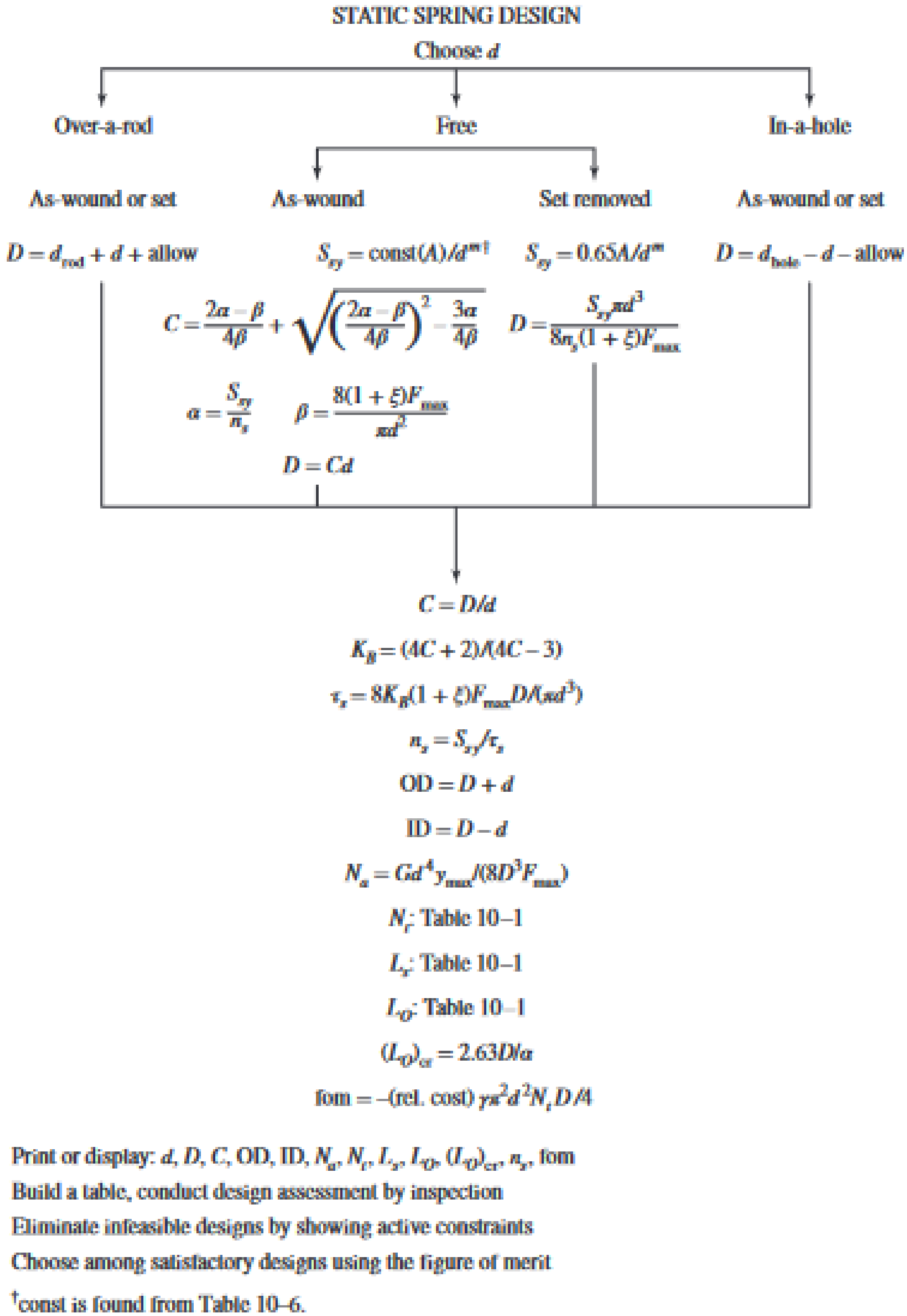
Table A–28 **Decimal Equivalents of Wire and Sheet-Metal Gauges* (All Sizes Are Given in Inches)** ***(Continued)***

Name of Gauge:	American or Brown & Sharpe	Birmingham or Stubs Iron Wire	United States Standard†	Manufacturers Standard	Steel Wire or Washburn & Moen	Music Wire	Stubs Steel Wire	Twist Drill
Principal Use:	Nonferrous Sheet, Wire, and Rod	Tubing, Ferrous Strip, Flat Wire, and Spring Steel	Ferrous Sheet and Plate, 480 lbf/ft^3	Ferrous Sheet	Ferrous Wire Except Music Wire	Music Wire	Steel Drill Rod	Twist Drills and Drill Steel
18	0.040 30	0.049	0.05	0.047 8	0.047 5	0.041	0.168	0.169 5
19	0.035 89	0.042	0.043 75	0.041 8	0.041 0	0.043	0.164	0.166 0
20	0.031 96	0.035	0.037 5	0.035 9	0.034 8	0.045	0.161	0.161 0
21	0.028 46	0.032	0.034 375	0.032 9	0.031 7	0.047	0.157	0.159 0
22	0.025 35	0.028	0.031 25	0.029 9	0.028 6	0.049	0.155	0.157 0
23	0.022 57	0.025	0.028 125	0.026 9	0.025 8	0.051	0.153	0.154 0
24	0.020 10	0.022	0.025	0.023 9	0.023 0	0.055	0.151	0.152 0
25	0.017 90	0.020	0.021 875	0.020 9	0.020 4	0.059	0.148	0.149 5
26	0.015 94	0.018	0.018 75	0.017 9	0.018 1	0.063	0.146	0.147 0
27	0.014 20	0.016	0.017 187 5	0.016 4	0.017 3	0.067	0.143	0.144 0
28	0.012 64	0.014	0.015 625	0.014 9	0.016 2	0.071	0.139	0.140 5
29	0.011 26	0.013	0.014 062 5	0.013 5	0.015 0	0.075	0.134	0.136 0
30	0.010 03	0.012	0.012 5	0.012 0	0.014 0	0.080	0.127	0.128 5
31	0.008 928	0.010	0.010 937 5	0.010 5	0.013 2	0.085	0.120	0.120 0
32	0.007 950	0.009	0.010 156 25	0.009 7	0.012 8	0.090	0.115	0.116 0
33	0.007 080	0.008	0.009 375	0.009 0	0.011 8	0.095	0.112	0.113 0
34	0.006 305	0.007	0.008 593 75	0.008 2	0.010 4		0.110	0.111 0
35	0.005 615	0.005	0.007 812 5	0.007 5	0.009 5		0.108	0.110 0
36	0.005 000	0.004	0.007 031 25	0.006 7	0.009 0		0.106	0.106 5
37	0.004 453		0.006 640 625	0.006 4	0.008 5		0.103	0.104 0
38	0.003 965		0.006 25	0.006 0	0.008 0		0.101	0.101 5
39	0.003 531				0.007 5		0.099	0.099 5
40	0.003 145				0.007 0		0.097	0.098 0

*Specify sheet, wire, and plate by stating the gauge number, the gauge name, and the decimal equivalent in parentheses.

†Reflects present average and weights of sheet steel.

- You are going to use this flow chart for every d and decide which one is best (note Free as-wound springs have a different formula for C):



- You should end up with a table very similar to this:

d	0.063	0.067	0.071	0.075	0.080	0.085	0.090	0.095
D	0.391	0.479	0.578	0.688	0.843	1.017	1.211	1.427
C	6.205	7.153	8.143	9.178	10.53	11.96	13.46	15.02
OD	0.454	0.546	0.649	0.763	0.923	1.102	1.301	1.522
N_a	39.1	26.9	19.3	14.2	10.1	7.3	5.4	4.1
L_s	2.587	1.936	1.513	1.219	0.964	0.790	0.668	0.581
L_0	4.887	4.236	3.813	3.519	3.264	3.090	2.968	2.881
$(L_0)_{cr}$	2.06	2.52	3.04	3.62	4.43	5.35	6.37	7.51
n_s	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2
fom	−0.409	−0.399	−0.398	−0.404	−0.417	−0.438	−0.467	−0.505

- Now eliminate options based on the conditions:
  Design requirements: $4 \leq C \leq 12$, $3 \leq N_a \leq 15$, $\xi \geq 0.15$, $n_s \geq 1.2$ Note n_s is the safety factor at solid height

- Hopefully you only have a few left at most, use the figure of merit (fom) to decide which one is best. Biggest number is best

5.3.4 Calculating based on C (seems better):

- Find an A and m value that fits the needs of the wire for the question, this is mostly material dependent. Note W&M wire is hand drawn:

Table 10–4 **Constants A and m of $S_{ut} = A/d^m$ for Estimating Minimum Tensile Strength of Common Spring Wires**

Material	ASTM No.	Exponent m	Diameter, in	A, kpsi · in^m	Diameter, mm	A, MPa · mm^m	Relative Cost of Wire
Music wire*	A228	0.145	0.004–0.256	201	0.10–6.5	2211	2.6
OQ&T wire†	A229	0.187	0.020–0.500	147	0.5–12.7	1855	1.3
Hard-drawn wire‡	A227	0.190	0.028–0.500	140	0.7–12.7	1783	1.0
Chrome-vanadium wire§	A232	0.168	0.032–0.437	169	0.8–11.1	2005	3.1
Chrome-silicon wire‖	A401	0.108	0.063–0.375	202	1.6–9.5	1974	4.0
302 Stainless wire#	A313	0.146	0.013–0.10	169	0.3–2.5	1867	7.6–11
		0.263	0.10–0.20	128	2.5–5	2065	
		0.478	0.20–0.40	90	5–10	2911	
Phosphor-bronze wire**	B159	0	0.004–0.022	145	0.1–0.6	1000	8.0
		0.028	0.022–0.075	121	0.6–2	913	
		0.064	0.075–0.30	110	2–7.5	932	

*Surface is smooth, free of defects, and has a bright, lustrous finish.
†Has a slight heat-treating scale which must be removed before plating.
‡Surface is smooth and bright with no visible marks.
§Aircraft-quality tempered wire, can also be obtained annealed.
‖Tempered to Rockwell C49, but may be obtained untempered.
#Type 302 stainless steel.
**Temper CA510.

- Use this table to get a percentage of S_ut you are allowed

Table 10–6 **Maximum Allowable Torsional Stresses for Helical Compression Springs in Static Applications**

	Maximum Percent of Tensile Strength	
Material	**Before Set Removed (includes K_W or K_B)**	**After Set Removed (includes K_s)**
Music wire and cold-drawn carbon steel	45	60–70
Hardened and tempered carbon and low-alloy steel	50	65–75
Austenitic stainless steels	35	55–65
Nonferrous alloys	35	55–65

- find S_{sy}
  $S_{sy} =$ percent from above $\cdot S_{ut} =$ percent $\cdot \frac{A}{d^m}$

- Solve for d using C = 10 since we just guess C and the example used 10:
  $K_B = \frac{4C+2}{4C-3}$
  $d = (0.163 K_B \frac{C}{A})^{1/(2-m)}$

- Use the wire that most closely fits what you calculated from table A-28:

- check safety factor:
  $n_s = 7.363 \frac{Ad^{2-m}}{K_B C}$
  If it is way over or way under 1.2 you can try a different d close to what you calculated, or don't they'll probably mark it right either way

- Find the spring rate k:
  $k = \frac{F}{y}$

- Find the number of active coils:
  $N_a = \frac{dG}{8kC^3}$
  if this is outside the condition $3 \leq N_a \leq 15$ adjust C while keeping it within $4 \leq C \leq 12$, if you can't do this you're fuck try another d maybe idk

- Find number of coils (N_t) from this table:

Table 10–1 Formulas for the Dimensional Characteristics of Compression Springs (N_a = Number of Active Coils)

	Type of Spring Ends			
Term	Plain	Plain and Ground	Squared or Closed	Squared and Ground
End coils, N_e	0	1	2	2
Total coils, N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length, L_0	$pN_a + d$	$p(N_a + 1)$	$pN_a + 3d$	$pN_a + 2d$
Solid length, L_s	$d(N_t + 1)$	dN_t	$d(N_t + 1)$	dN_t
Pitch, p	$(L_0 - d)/N_a$	$L_0/(N_a + 1)$	$(L_0 - 3d)/N_a$	$(L_0 - 2d)/N_a$

- Calculate deflection length:
  $y_s = \frac{F_{max}}{k}$

- Use Table 1 above to solid length, free length and then solve for mean coil diameter:
  $D = Cd$
  $OD = D + d$ Remember some of these forms may be different for different boundary type of

springs, refer to the flow chart above to confirm formulas

- Calculate the alpha condition:
  $\alpha < 2.63\frac{D}{L_0}$

- Use this table to see how you can support the spring:

Table 10–2 End-Condition Constants α for Helical Compression Springs*

End Condition	Constant α
Spring supported between flat parallel surfaces (fixed ends)	0.5
One end supported by flat surface perpendicular to spring axis (fixed); other end pivoted (hinged)	0.707
Both ends pivoted (hinged)	1
One end clamped; other end free	2

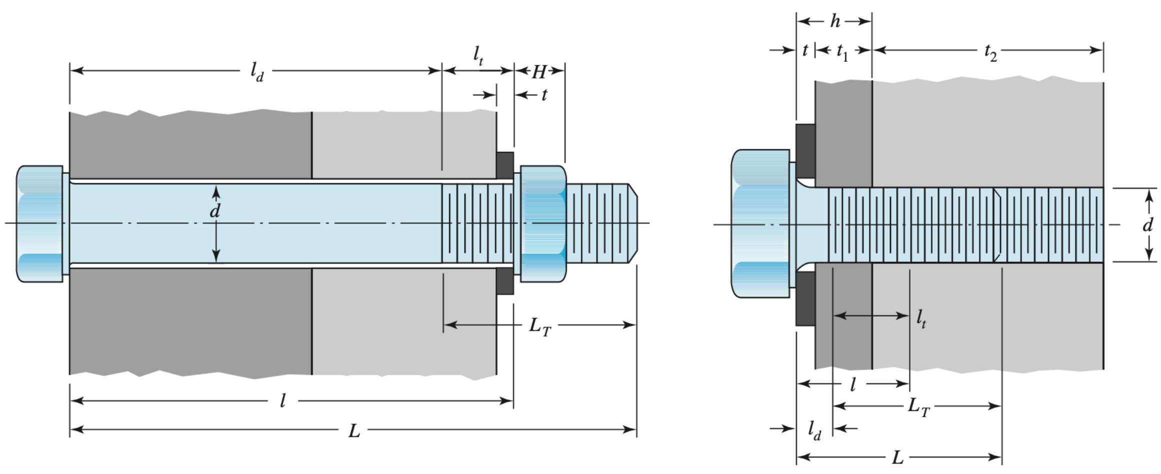
*Ends supported by flat surfaces must be squared and ground.

5.4 Fasteners/Bolts

5.4.1 Design Selection

1. Determine a suitable length for the bolt:

   (a) Compute the grip length.



(a) (b)

- If your bolt arrangement is like figure (a):

  l = thickness of all material squeezed between face of bolt and face of nut

- If your bolt arrangement is like figure (b):

$$l = \begin{cases} h + t_2/2, & t_2 < d \\ h + d/2 & t_2 \geq d \end{cases}$$

(b) Compute the fastener/bolt length (round up to nearest 1/4th inch):

- For figure (a):

$$L > l + H$$

- for figure (b):

$$L > h + 1.5d$$

2. Compute the bolt stiffness k_b:

(a) Determine the threaded length L_T:

- If your units are in inches:

$$L_T = \begin{cases} 2d + \frac{1}{4} \text{ in} & L \leq 6 \text{ in} \\ 2d + \frac{1}{2} \text{ in} & L > 6 \text{ in} \end{cases}$$

- If your units are in mm:

$$L_T = \begin{cases} 2d + 6 \text{ mm} & L \leq 125 \text{ mm}, d \leq 48 \text{ mm} \\ 2d + 12 \text{ mm} & 125 < L \leq 200 \text{ mm} \\ 2d + 25 \text{ mm} & L > 200 \text{ mm} \end{cases}$$

(b) Compute the length of the unthreaded portion in grip l_d:

$$l_d = L - L_T$$

(c) Compute the length of the threaded portion in grip l_t:

$$l_t = l - l_d$$

(d) Compute the area of the unthreaded portion A_d:

$$A_d = \pi d^2/4$$

(e) Find the area of the threaded portion A_t from one of the following tables (it is the tensile-stress area A_t):

Table 8–1 Diameters and Areas of Coarse-Pitch and Fine-Pitch Metric Threads*

	Coarse-Pitch Series			Fine-Pitch Series		
Nominal Major Diameter d mm	Pitch p mm	Tensile-Stress Area A_t mm^2	Minor-Diameter Area A_r mm^2	Pitch p mm	Tensile-Stress Area A_t mm^2	Minor-Diameter Area A_r mm^2
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470
110				2	9180	9080

Table 8–2 **Diameters and Area of Unified Screw Threads UNC and UNF***

		Coarse Series—UNC			Fine Series—UNF		
Size Designation	Nominal Major Diameter in	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	0.373	0.351
$\frac{7}{8}$	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
$1\frac{1}{4}$	1.2500	7	0.969	0.890	12	1.073	1.024
$1\frac{1}{2}$	1.5000	6	1.405	1.294	12	1.581	1.521

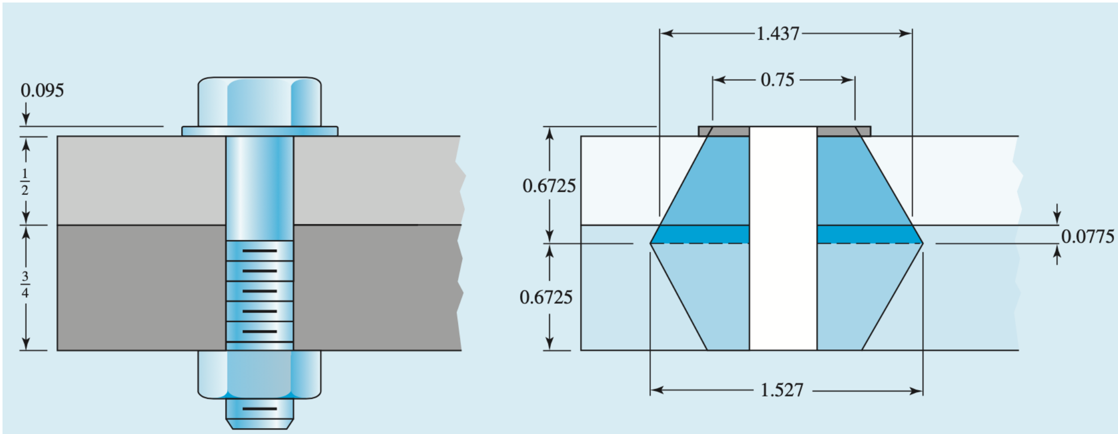
(f) Compute the bolt stiffness k_b (in lbf/in or N/m):

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}$$

where E is Young's Modulus

3. Determine the stiffness of the members:

(a) Make a diagram like the one below. The angle at which the line goes down is assumed to be 30°. The line will start at a diameter of 1.5 times the nominal diameter of the bolt apart. The lines will start on the top washer (or if there is none then the top of the upper plate) and will end either on the bottom of the lower plate or if there is a second washer then into it as well. The lines will continue to separate until they reach the middle of the height of the region that they are allowed in. After that, they will start to converge again.



(b) For each section of the diagram, (i.e. top washer, top plate, bottom plate, bottom washer etc.) calculate the spring rate k using the following expression:

$$k = \frac{0.5774\pi E d}{\ln \dfrac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

where E is Young's Modulus of the material,
d is the inner diameter of the section member
for plates, it is usually the nominal diameter of the bolt
for washers, it is their inner diameter
D is the shortest distance between two separating lines in the member's region,
t is the thickness of the section

Note that if two members have the same material and the inner diameter can they can be treated as one member where the thickness is the sum of their individual thicknesses.

Also note that if the lines go from separating to converging in the middle of a member's region, then you have to treat the section where they are separating as a different member from the section where they are converging. This is because they will have a different D value.

(c) Once you have calculated all the spring rates for the individual members you can add them with the following equation:

$$\frac{1}{k_t} = \frac{1}{k_1} + \cdots + \frac{1}{k_n}$$

(d) If the top half and bottom half are the same, then you can calculate the spring rate of the top half with:

$$\frac{1}{k_{top}} = \frac{1}{k_1} + \cdots + \frac{1}{k_n}$$

and the total spring rate of the joint will be:

$$k_t = \frac{k_{top}}{2}$$

5.4.2 Factor of Safety and Preload Calculations

1. Determine the torque to reach a given preload

   (a) Calculate P, the load per bolt.

   $$P = \frac{P_{tot}}{N}$$

   Where P_{tot} is the total load in the tension joint,
   N is the number of bolts

   (b) Find the proof strength S_p, the minimum tensile strength S_{ut} and the endurance strength of the bolt S_e from the following tables:

Table 8–9 **SAE Specifications for Steel Bolts**

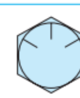
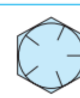
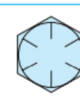
SAE Grade No.	Size Range Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
1	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low or medium carbon	
2	$\frac{1}{4}$–$\frac{3}{4}$ $\frac{7}{8}$–$1\frac{1}{2}$	55 33	74 60	57 36	Low or medium carbon	
4	$\frac{1}{4}$–$1\frac{1}{2}$	65	115	100	Medium carbon, cold-drawn	
5	$\frac{1}{4}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Medium carbon, Q&T	
5.2	$\frac{1}{4}$–1	85	120	92	Low-carbon martensite, Q&T	
7	$\frac{1}{4}$–$1\frac{1}{2}$	105	133	115	Medium-carbon alloy, Q&T	
8	$\frac{1}{4}$–$1\frac{1}{2}$	120	150	130	Medium-carbon alloy, Q&T	
8.2	$\frac{1}{4}$–1	120	150	130	Low-carbon martensite, Q&T	

Table 8–10 ASTM Specifications for Steel Bolts

ASTM Designation No.	Size Range, Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
A307	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low carbon	
A325, type 1	$\frac{1}{2}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Medium carbon, Q&T	
A325, type 2	$\frac{1}{2}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Low-carbon, martensite, Q&T	
A325, type 3	$\frac{1}{2}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Weathering steel, Q&T	
A354, grade BC	$\frac{1}{4}$–$2\frac{1}{2}$ $2\frac{3}{4}$–4	105 95	125 115	109 99	Alloy steel, Q&T	
A354, grade BD	$\frac{1}{4}$–4	120	150	130	Alloy steel, Q&T	
A449	$\frac{1}{4}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$ $1\frac{3}{4}$–3	85 74 55	120 105 90	92 81 58	Medium-carbon, Q&T	
A490, type 1	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Alloy steel, Q&T	
A490, type 3	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Weathering steel, Q&T	

Table 8–11 Metric Mechanical-Property Classes for Steel Bolts, Screws, and Studs

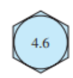
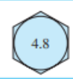
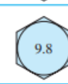
Property Class	Size Range, Inclusive	Minimum Proof Strength,* MPa	Minimum Tensile Strength,* MPa	Minimum Yield Strength,* MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	
4.8	M1.6–M16	310	420	340	Low or medium carbon	
5.8	M5–M24	380	520	420	Low or medium carbon	
8.8	M16–M36	600	830	660	Medium carbon, Q&T	
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	

Table 8–17 **Fully Corrected Endurance Strengths for Bolts and Screws with Rolled Threads***

Grade or Class	Size Range	Endurance Strength
SAE 5	$\frac{1}{4}$–1 in	18.6 kpsi
	$1\frac{1}{8}$–$1\frac{1}{2}$ in	16.3 kpsi
SAE 7	$\frac{1}{4}$–$1\frac{1}{2}$ in	20.6 kpsi
SAE 8	$\frac{1}{4}$–$1\frac{1}{2}$ in	23.2 kpsi
ISO 8.8	M16–M36	129 MPa
ISO 9.8	M1.6–M16	140 MPa
ISO 10.9	M5–M36	162 MPa
ISO 12.9	M1.6–M36	190 MPa

(c) Determine the initial bolt tension/preload force:
- If given there you are done.
- If said to be X% of proof strength, then the preload force F_i is

$$F_i = \frac{X}{100} A_t S_p$$

where A_t is the tensile-stress area given in Table 8-2
S_p is the proof strength of the bolt

- If the question asks you to choose then from Shigley's recommendation:

For nonpermanent connections, reused fasteners: $F_i = .75 A_t S_p$
For permanent connections: $F_i = .90 A_t S_p$

(d) Choose a torque factor K from the table

Bolt Condition	K
Nonplated, black finish	0.30
Zinc-plated	0.20
Lubricated	0.18
Cadmium-plated	0.16
With Bowman Anti-Seize	0.12
With Bowman-Grip nuts	0.09

(e) Compute the torque from the given formula:

$$T = K F_i d$$

where T is the K is the torque factor,
F_i is the initial bolt tension,
d is the nominal diameter of the bolt

- Note that according to the homework solutions a torque of around 1000 lb-in is "VERY high" for a wrench

2. Compute overload/load factor of safety

(a) Calculate the stiffness constant of the joint:

$$C = \frac{k_b}{k_b + k_m}$$

Where k_b is the bolt spring rate,
k_m is the member spring rate

Note that the stiffness constant of the joint represents what percent of the load is picked up by the bolt compared to the members.

(b) Calculate the load factor:

$$n_L = \frac{S_P A_t - F_i}{CP}$$

Where S_p is the proof strength of the bolt in Pa or psi,
A_t is the tensile-stress area given in Table 8-1 or 8-2 above in m^2 or in^2,
F_i is the initial bolt tension in N or lb,
C is the stiffness constant,
P is the tension load per bolt in N or lb

Note that to convert from mm^2 to m^2, multiply by 10^6

3. Compute bolt yield/yielding factor of safety:

(a) The tensile stress in the bolt is:

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t}$$

Where A_t is the tensile-stress area given in Table 8-1 or 8-2 above in m^2 or in^2,
F_i is the initial bolt tension in N or lb,
C is the stiffness constant,
P is the tension load per bolt in N or lb

(b) The static yielding factor of safety is:

$$n_p = \frac{S_p}{\sigma_b} = \frac{S_p A_t}{CP + F_i}$$

4. Compute the joint separation factor of safety:

$$n_0 = \frac{F_i}{P(1 - C)}$$

5. Compute the Goodman criteria fatigue factor of safety:

(a) Compute the alternating stress:

$$\sigma_a = \frac{C(P_{max} - P_{min})}{2A_t}$$

(b) Compute the midrange stress:

$$\sigma_m = \frac{C(P_{max} + P_{min})}{2A_t} + \frac{F_i}{A_t}$$

(c) Compute the initial stress:

$$\sigma_i = \frac{F_i}{A_t}$$

(d) Compute the fatigue factor of safety:

$$n_f = \frac{S_e(S_{ut} - \sigma_i)}{S_{ut}\sigma_a + S_e(\sigma_m - \sigma_i)}$$

Where S_e is the endurance strength,
S_{ut} is the tensile strength,
σ_i is the initial stress,
σ_m is the midrange stress,
σ_a is the alternating stress

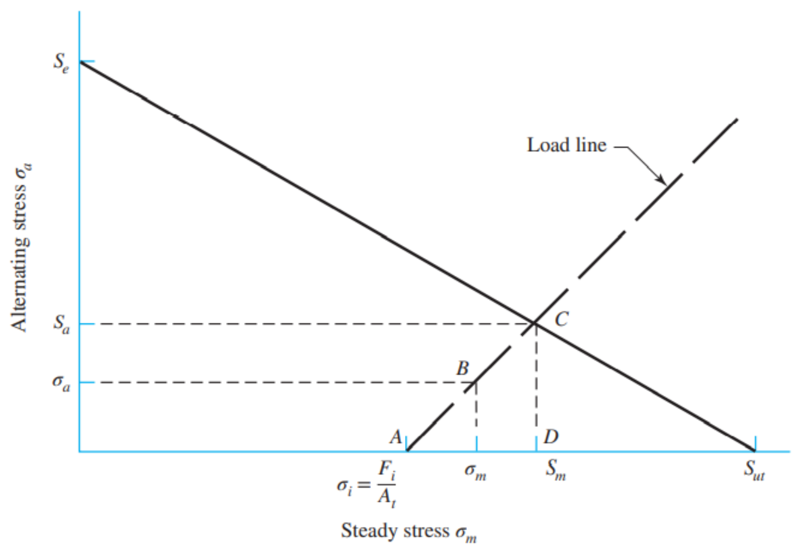
(e) If the external load is a repeated load i.e. (from 0 to P) then:

$$\sigma_a = \frac{CP}{2A_t}$$

$$\sigma_m = \frac{CP}{2A_t} + \frac{F_i}{A_t} = \sigma_a + \sigma_i$$

$$n_f = \frac{S_e(S_{ut} - \sigma_i)}{\sigma_a(S_{ut} + S_e)}$$

(f) If fatigue failure diagram is needed, it is shown below:



- The load line is given by the equation:

$$S_a = \frac{\sigma_a}{\sigma_m - \sigma_i}(S_m - \sigma_i)$$

- Then the Goodman line a.k.a. the line going from S_e to S_{ut} is given by the equation:

$$S_a = S_e - \frac{S_e}{S_{ut}} S_m$$

- Note that even though the graph has axes σ_a v.s. σ_m, it is essentially a S_a v.s. S_m graph. So just plot S_a as a function of S_m.

6. If any of the factor safeties are low or below 1, then the grade, size, or quantity of bolts can be increased to increase the safety factors.