Stupid Mech 325 Stupid Summary

Fuck this course

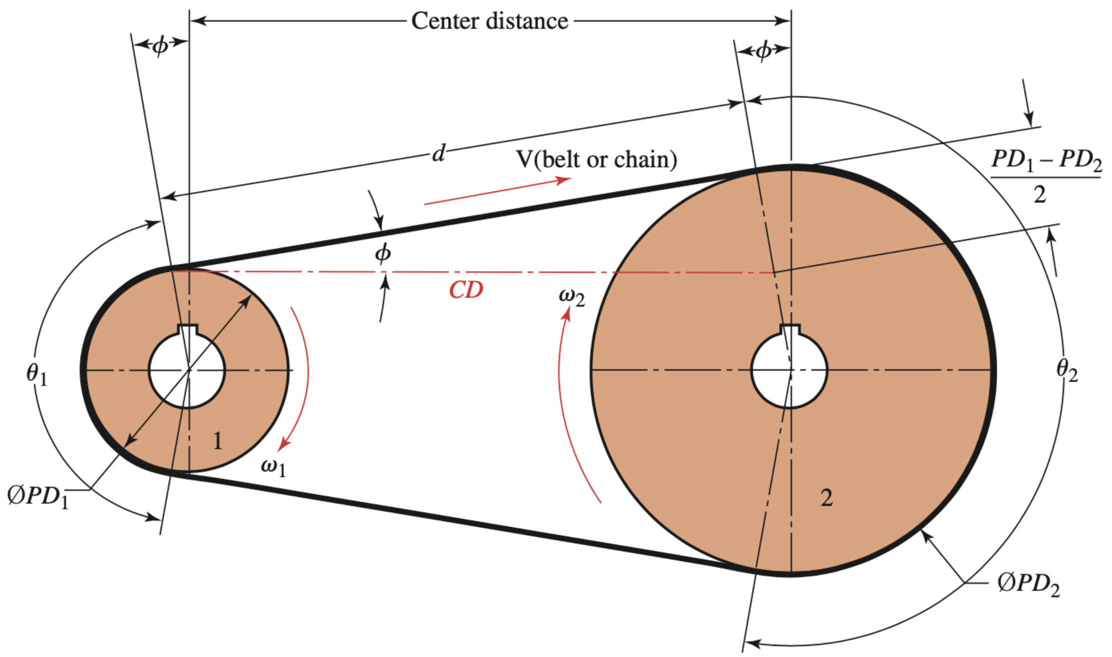
Tyler Wilson, Sarah Gauthier and Ethan Predinchuk

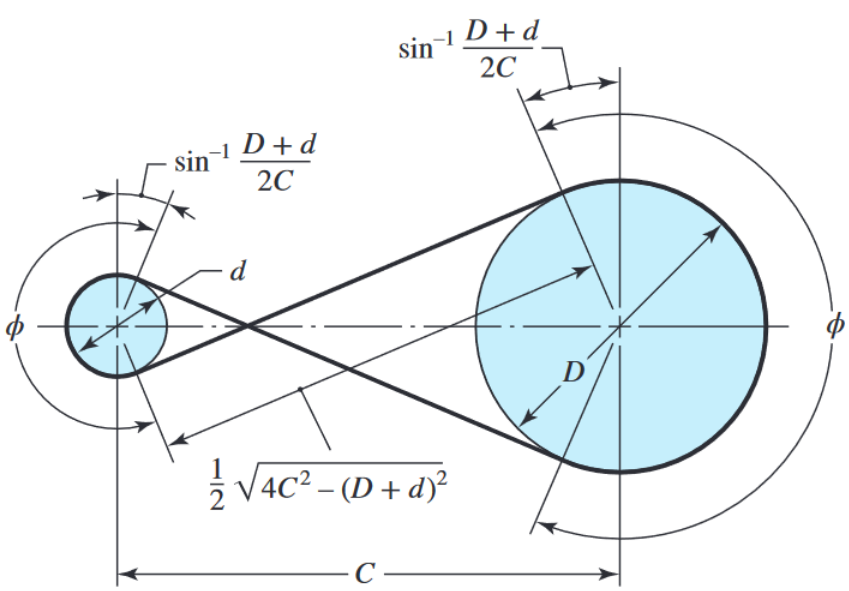
Contents

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

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

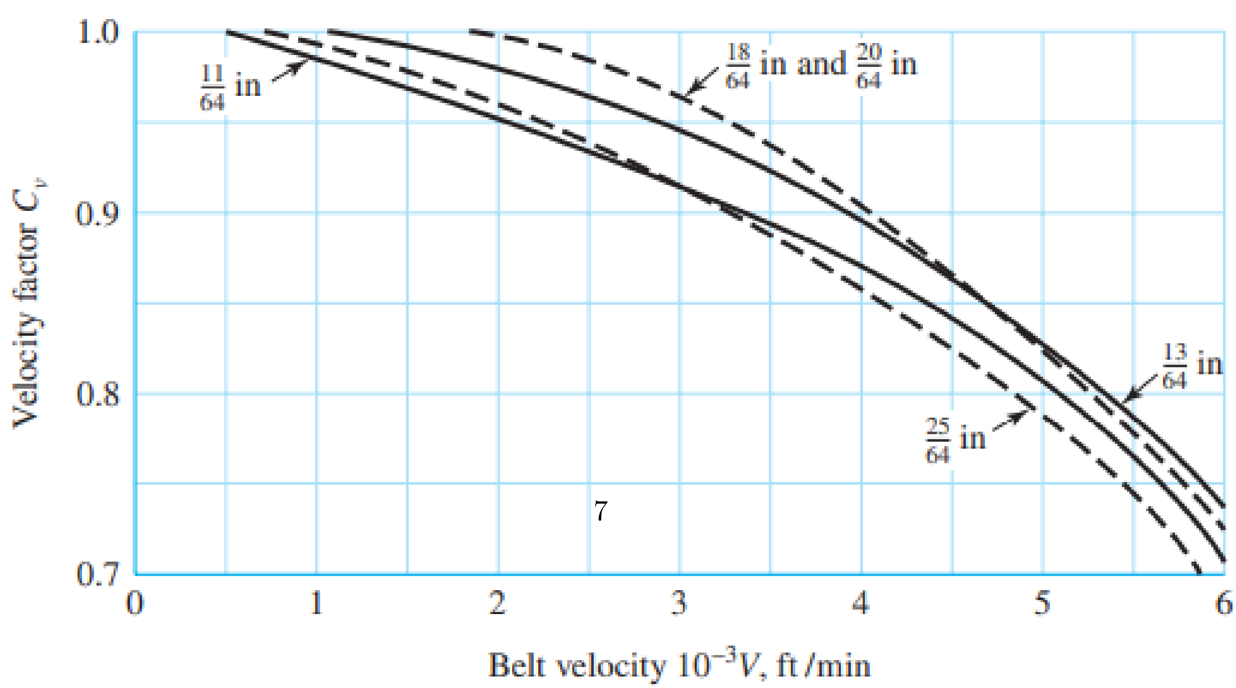
$\frac{F_1 - F_c}{F_2 - F_c} = e^{f\phi}$
$F_c = \frac{w}{32.17\,\mathrm{ft/s^2}} \left(\frac{V}{60\,\mathrm{s/min}}\right)^2$
$F_i = \frac{F_1 + F_2}{2} - F_c = \frac{T}{d} \frac{e^{f\phi}+1}{e^{f\phi}-1}$
$H = \frac{(F_1 - F_2)V}{33{,}000\,(\frac{\mathrm{ft\,lb}}{\mathrm{min}})/\mathrm{hp}}$
$w = 12\,\mathrm{in/ft}\,\gamma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



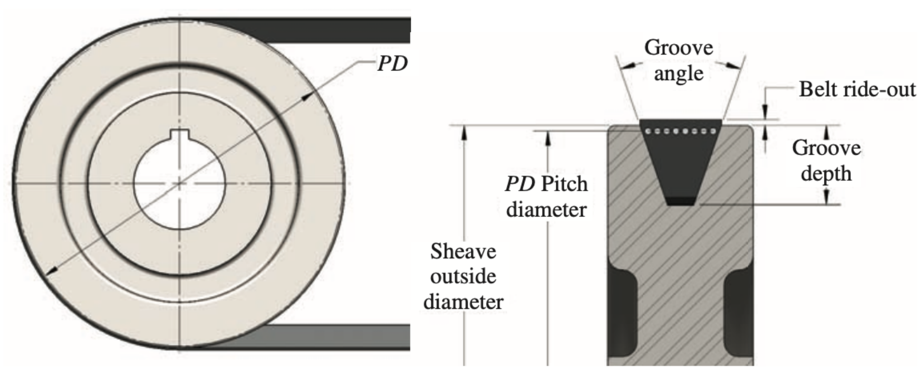
$C_v = 1$ for polyamide and urethane belts

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	AC motors: Normal torque[2] DC motors: Shunt-wound Engines: Multiple-cylinder <6 h per day	6–15 h per day	>15 h per day	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
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.

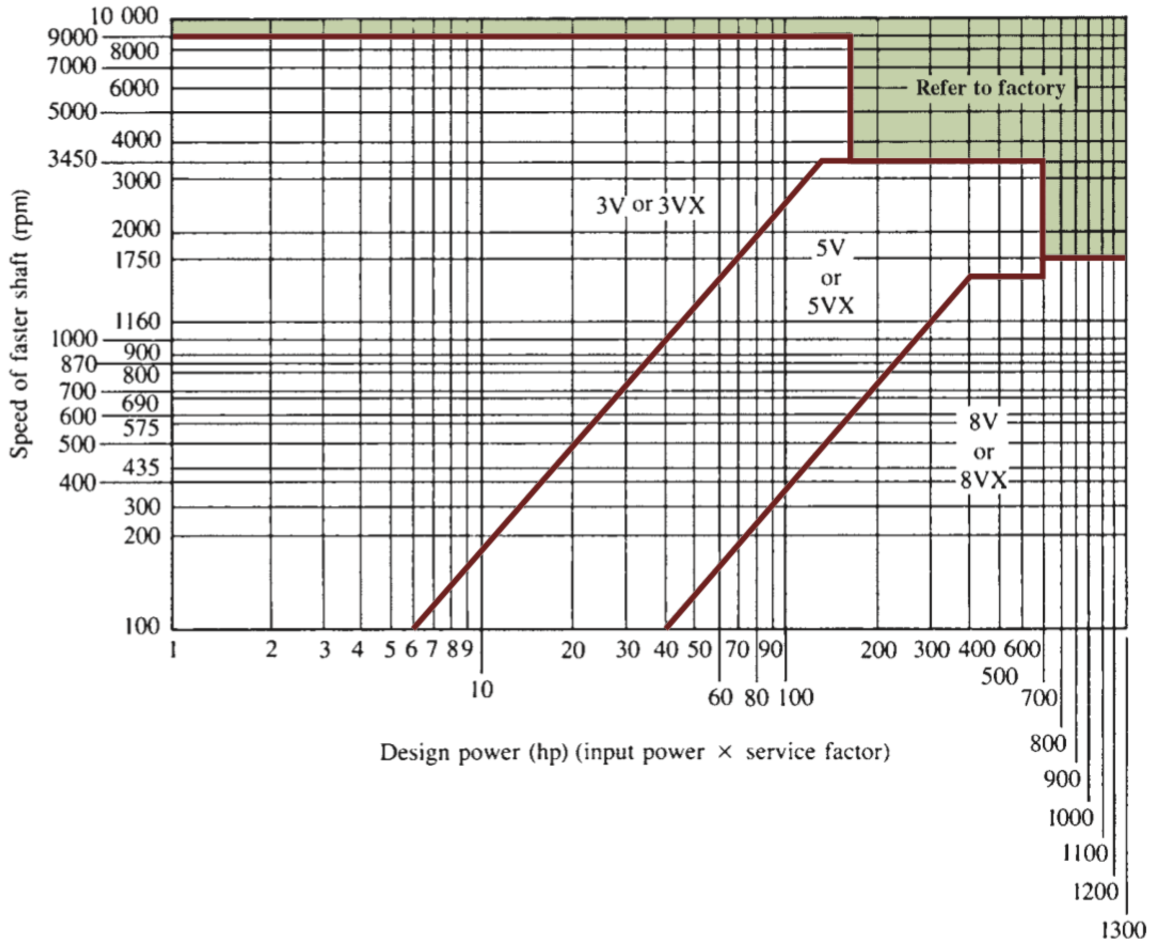
[3]Single-phase, three-phase with starting torque or breakdown torque greater than 175% of full-load torque.

(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

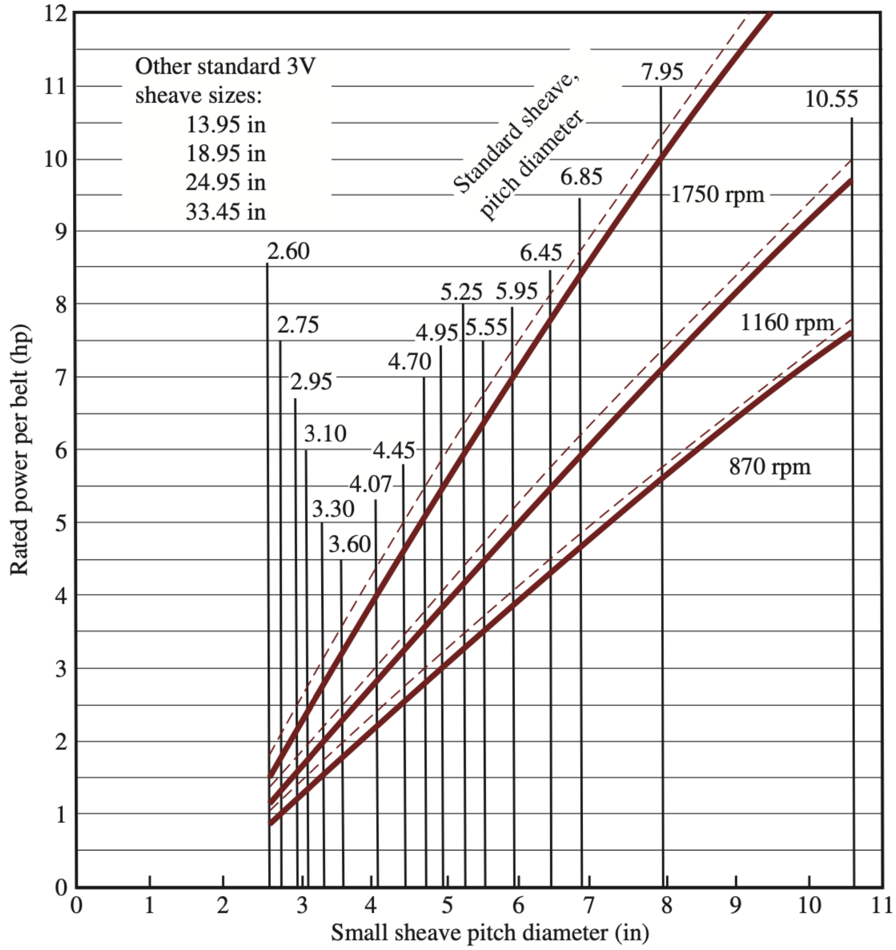
$$VR = \frac{n_1}{n_2} \text{ where } n_1 > n_2$$

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)

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

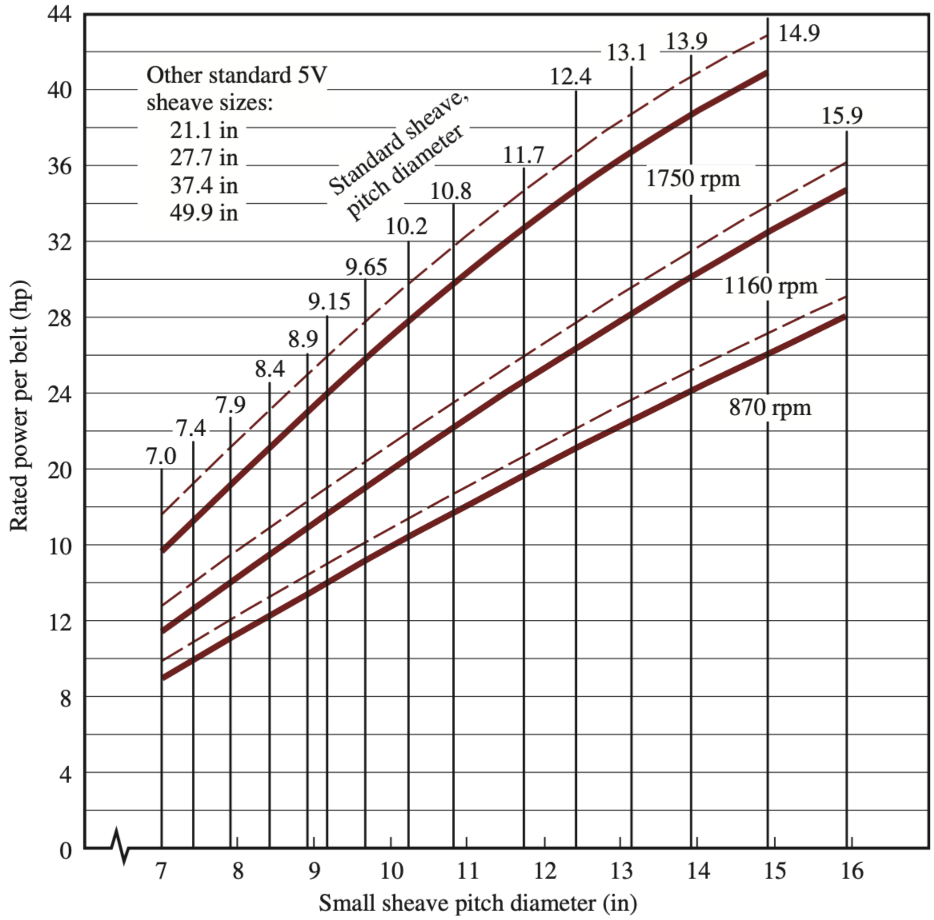
5. Select the standard sizes for the input and output sheaves

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



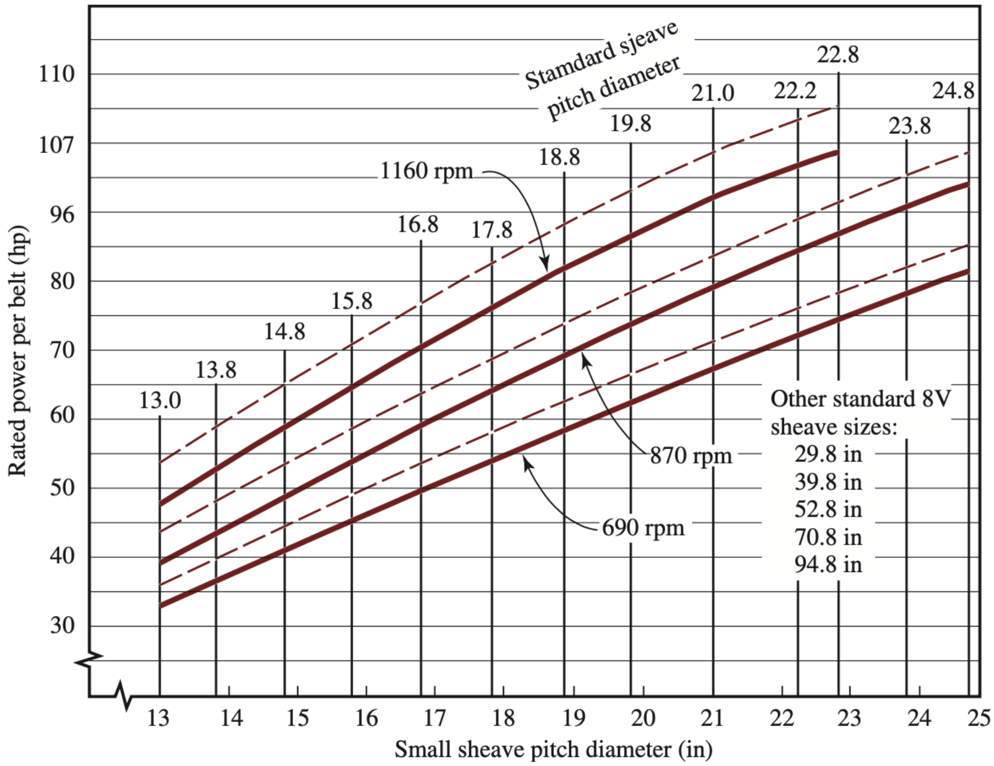
Other standard 3V
sheave sizes:
13.95 in
18.95 in
24.95 in
33.45 in
Standard sheave,
pitch diameter
2.60
2.75
2.95
3.10
3.30
3.60
4.07
4.45
4.70
4.95
5.25
5.55
5.95
6.45
6.85
7.95
10.55
1750 rpm
1160 rpm
870 rpm
Rated power per belt (hp)
Small sheave pitch diameter (in)

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

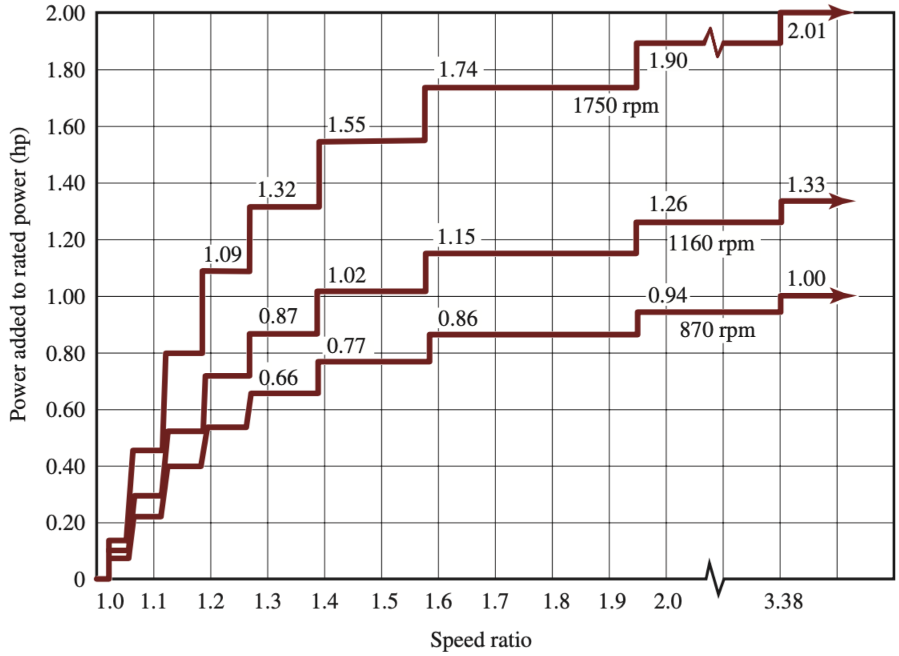
$$VR = \frac{D_2}{D_1}$$

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

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

$$B = 4L_p - 6.28(D_2 + D_1)$$
$$CD = \frac{B + \sqrt{B^2 - 32(D_2 - D_1)^2}}{16}$$

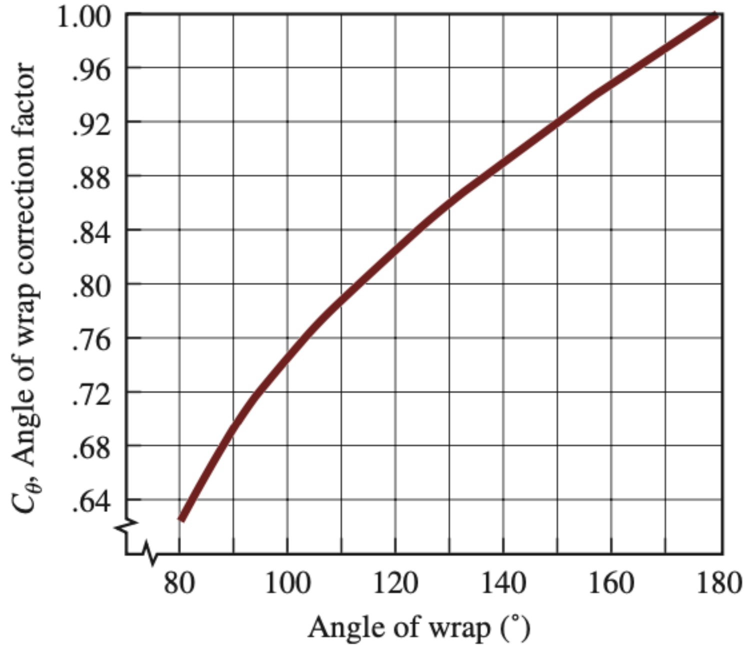
12. Compute the angle of wrap of small sheave

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

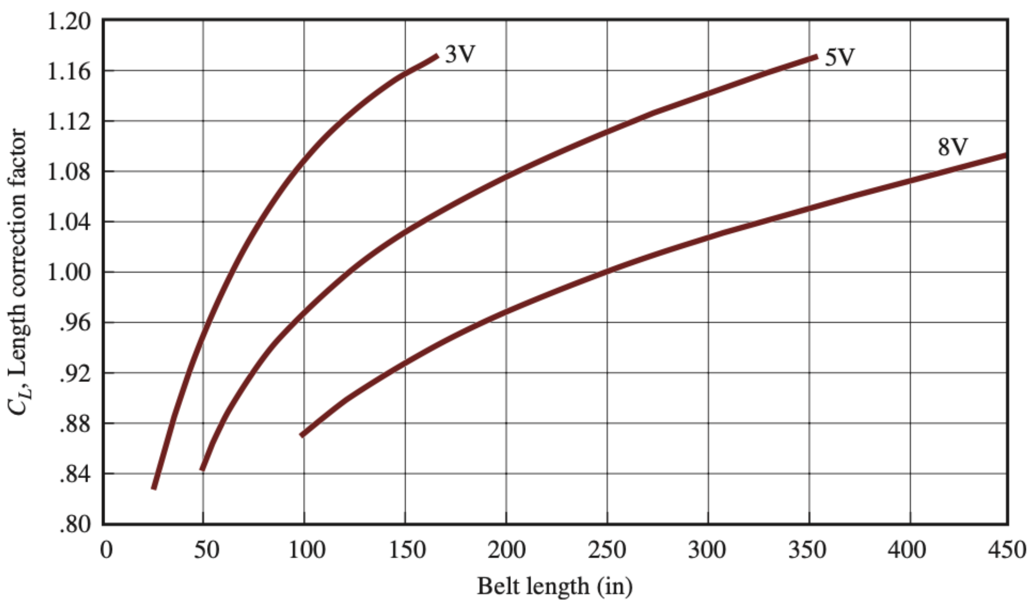
Probably won't be asked to compute the angle of wrap for the big sheave, but just in case, here is the formula:

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

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

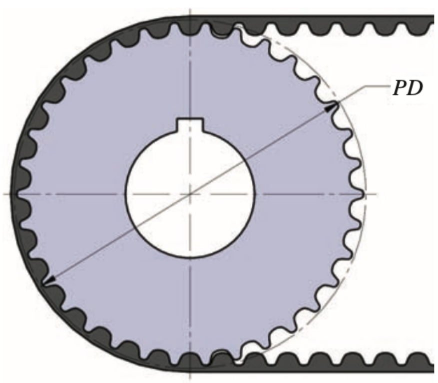
$$\text{min number of belts} = \frac{\text{design power}}{\text{corrected power rating}}$$

(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

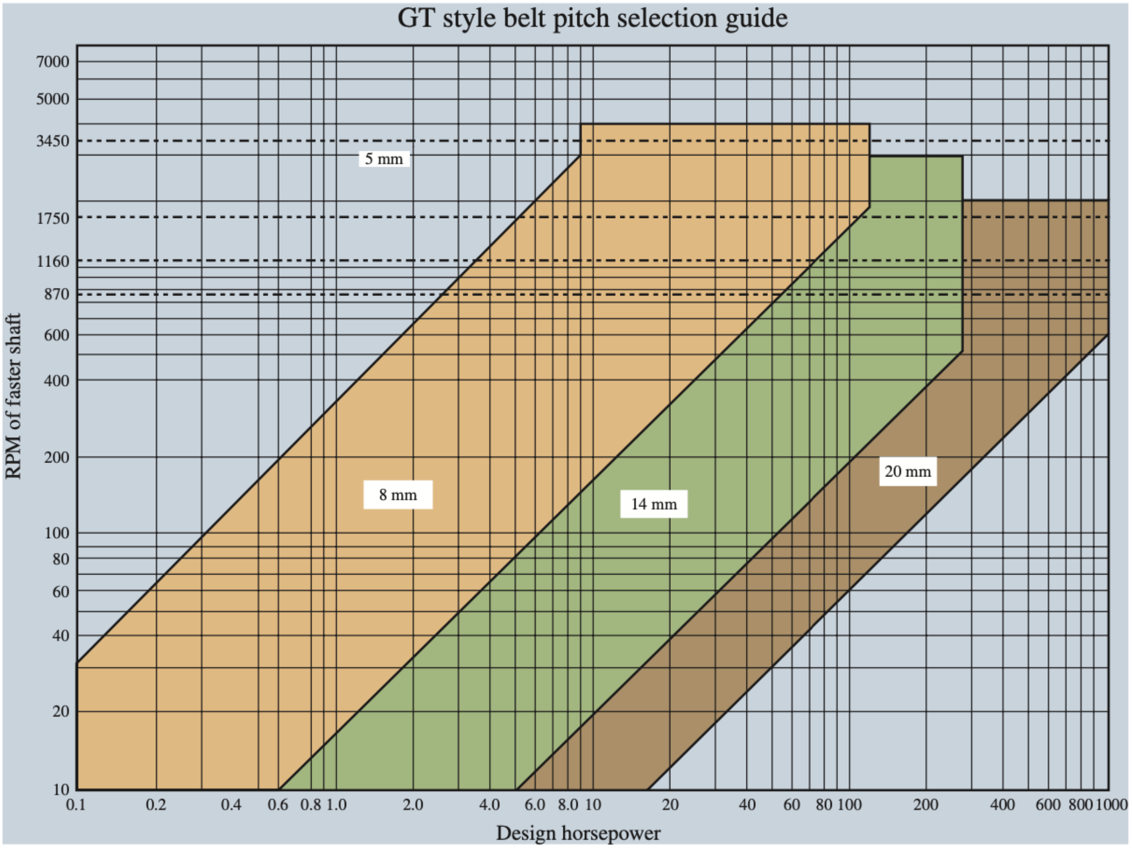
TABLE 7–8 Service Factor

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)															
Driver	Driven	Velocity ratio	920-8MGT P.L. 36.220 115 teeth	960-8MGT P.L. 37.795 120 teeth	1040-8MGT P.L. 40.945 130 teeth	1064-8MGT P.L. 41.890 133 teeth	1120-8MGT P.L. 44.094 140 teeth	1160-8MGT P.L. 45.669 145 teeth	1200-8MGT P.L. 47.244 150 teeth	1224-8MGT P.L. 48.189 153 teeth	1280-8MGT P.L. 50.394 160 teeth	1440-8MGT P.L. 56.693 180 teeth	1512-8MGT P.L. 59.528 189 teeth	1584-8MGT P.L. 62.362 198 teeth	1600-8MGT P.L. 62.992 200 teeth	1760-8MGT P.L. 69.291 220 teeth	1800-8MGT P.L. 70.866 225 teeth	2000-8MGT P.L. 78.740 250 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 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	0.25	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.

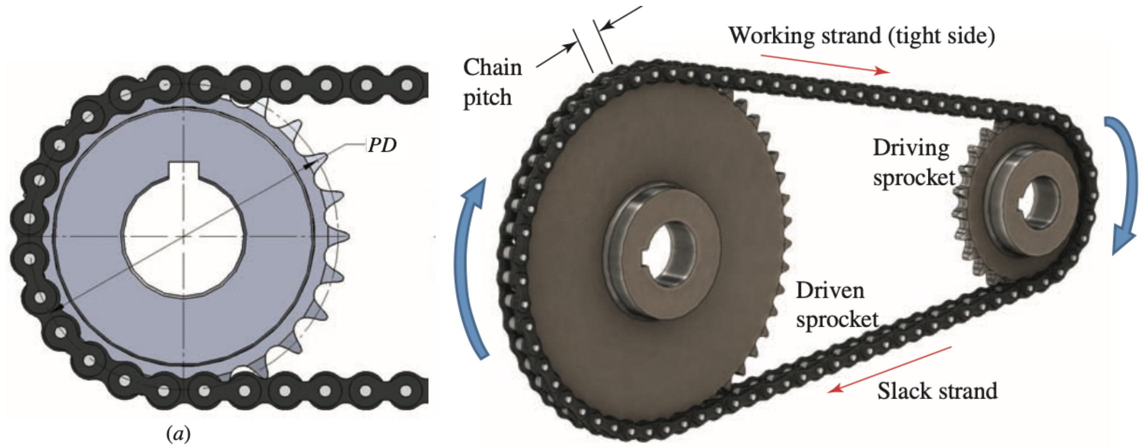
11. Calculate belt speed to ensure it does not exceed 6500 ft/min

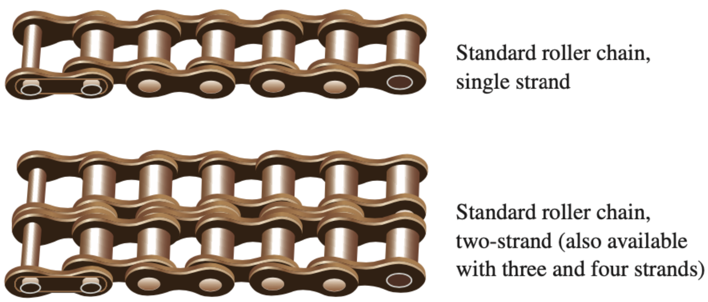
$$v_{belt} = \frac{PD_1}{2} \cdot \omega_1 \cdot 2\pi \text{ rad/rev} \cdot \frac{1 \text{ ft}}{12 \text{ in}}$$

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

1.6 Chain Drives

1.6.1 Anatomy





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

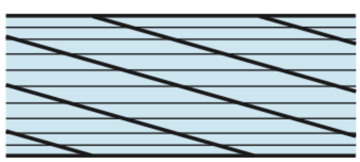
$$FS = P_{allowed}/P_{des}$$

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

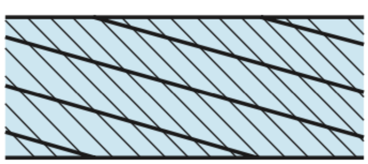
1.7 Wire Rope

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

Table 17–27 **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

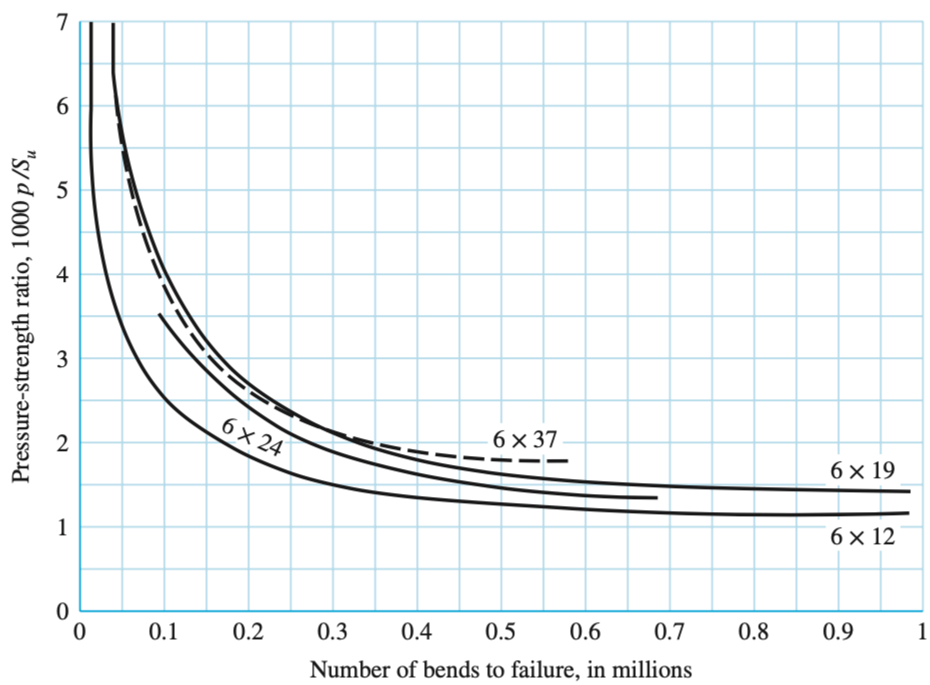
Table 17–26 **Maximum Allowable Bearing Pressures of Ropes on Sheaves (in psi)**

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

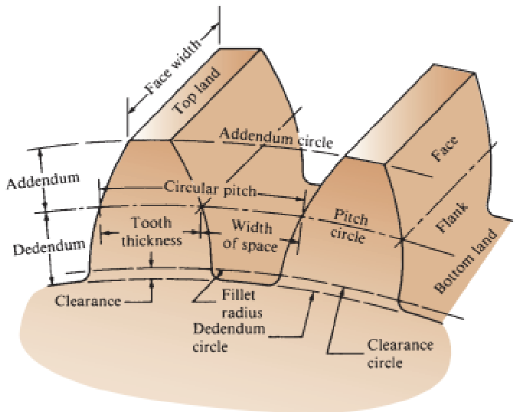
No idea how to use this but here it is in case it's relevant:

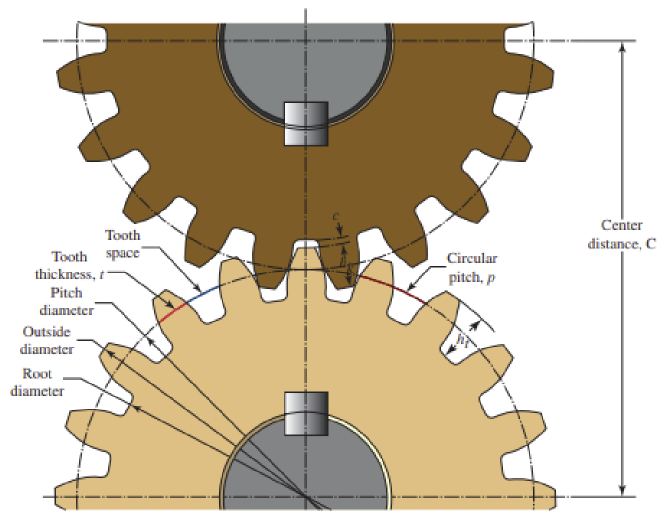


2 Gears and Shit

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 · 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:

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

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

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

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

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

$$\text{root diameter: } D_R = D - 2b$$

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

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

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

$$\text{whole depth: } h_f = a + b$$

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

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

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

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

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

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

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

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

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

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

Forces and motion:

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

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

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

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

$$
\begin{aligned}
\text{radial force: } & W_r = W_t \tan\phi \\
\text{normal force: } & W_n = \frac{W_t}{\cos\phi} = \sqrt{W_t^2 + W_r^2} \\
\text{bending stress number: } & s_t = \frac{W_t P_d}{FJ} K_O K_s K_m K_B K_v \\
\text{contact stress number: } & s_c = C_p \sqrt{\frac{W_t K_O K_s K_m K_v}{F D_p I}} \\
\text{allowable bending stress: } & s_{at} > s_t \frac{(SF) K_R}{Y_N} \\
\text{allowable contact stress: } & s_{ac} > s_c \frac{(SF) K_R}{Z_N}
\end{aligned}
$$

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.

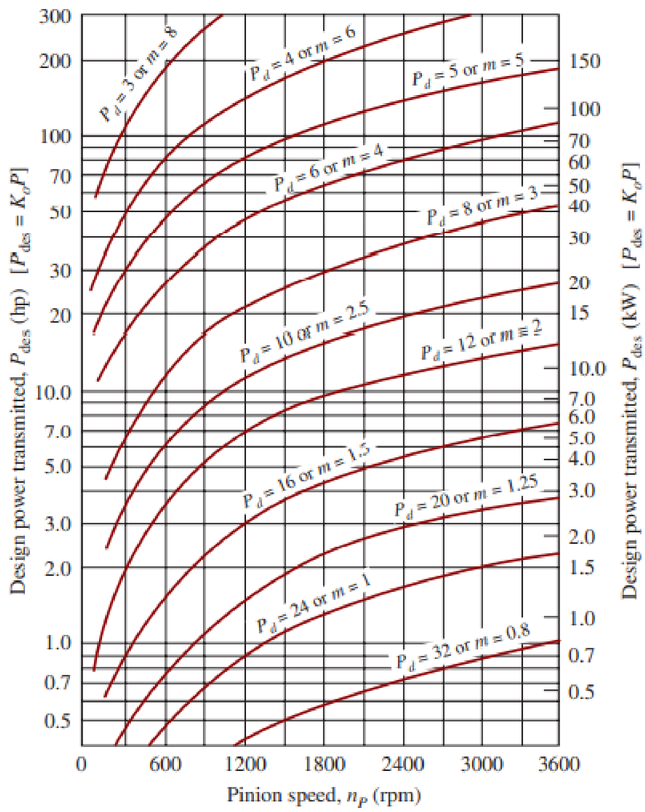
2. Use this fucking thing to find the shock

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)	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)
		Gear material and modulus of elasticity, E_G, lb/in^2 (MPa)					
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 (lb/in^2)$^{0.5}$ or (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.

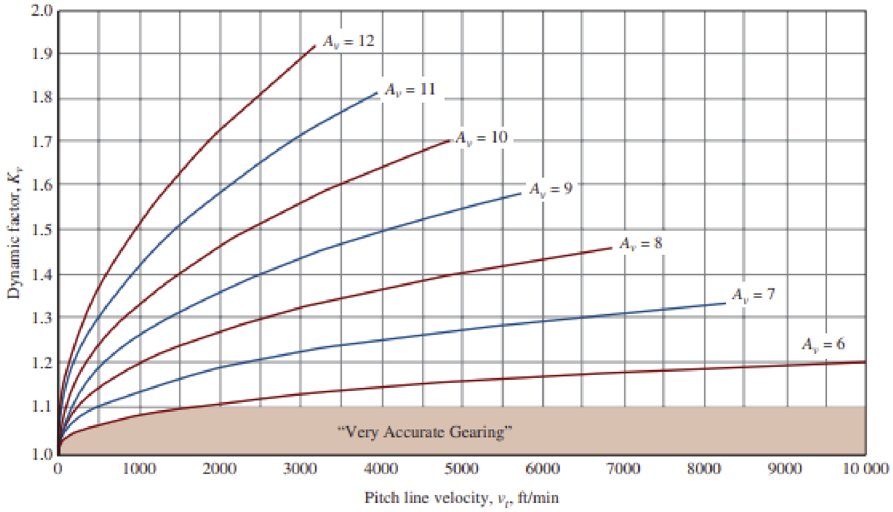
TABLE 9–5 Recommended AGMA Quality Numbers

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

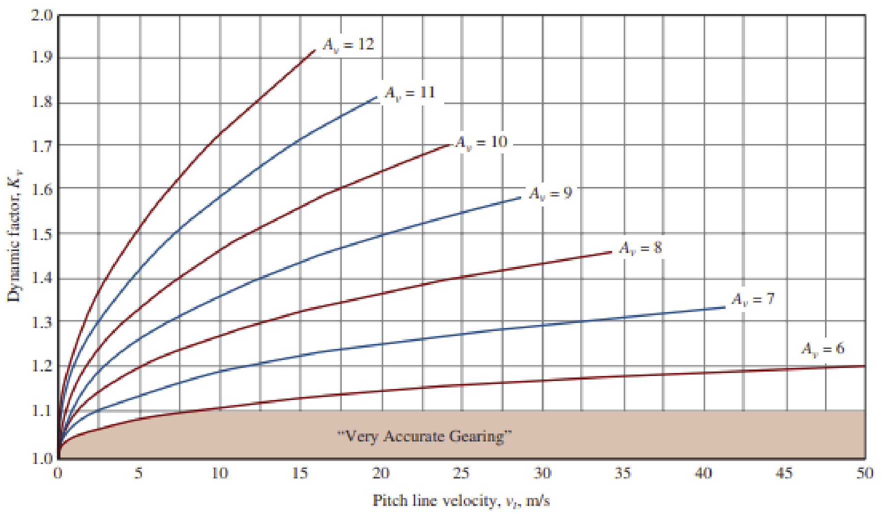
Machine tool drives and drives for other high-quality mechanical systems

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

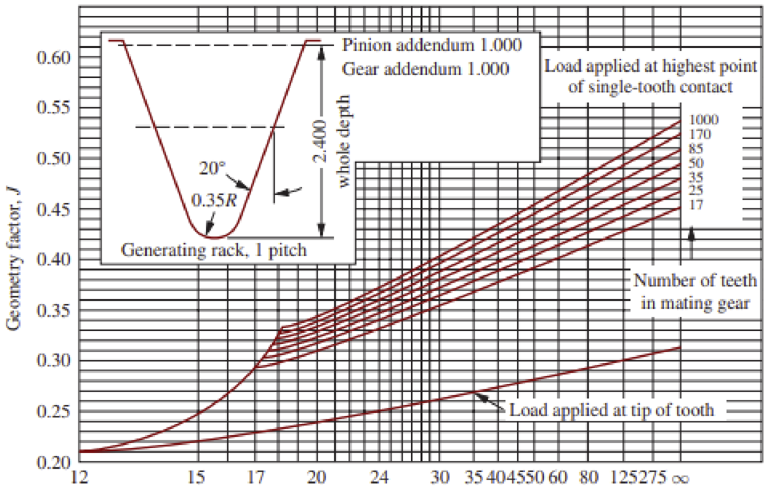


(*a*) U.S. Customary units

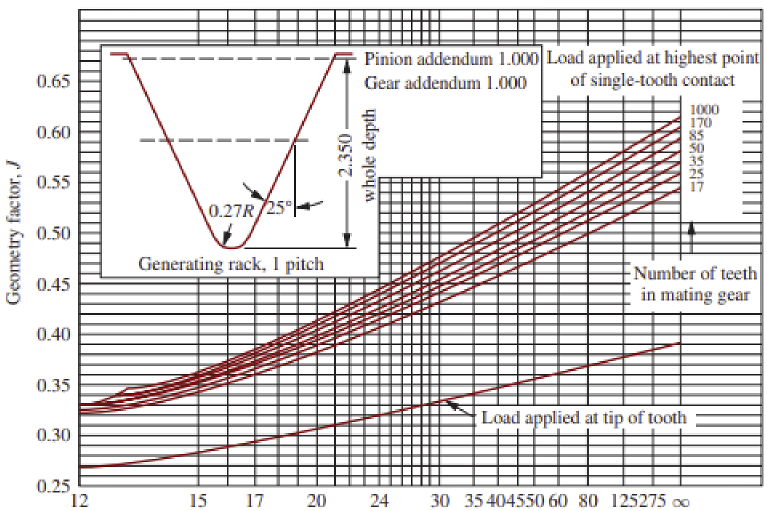


(*b*) SI Metric units

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

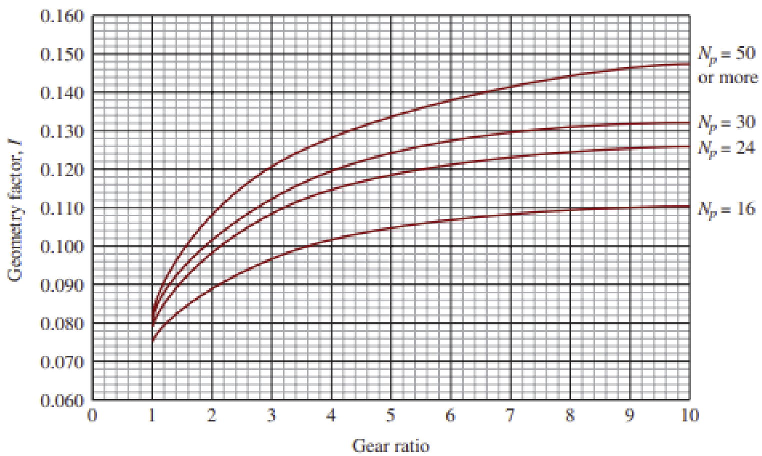


(*a*) 20° spur gear: standard addendum

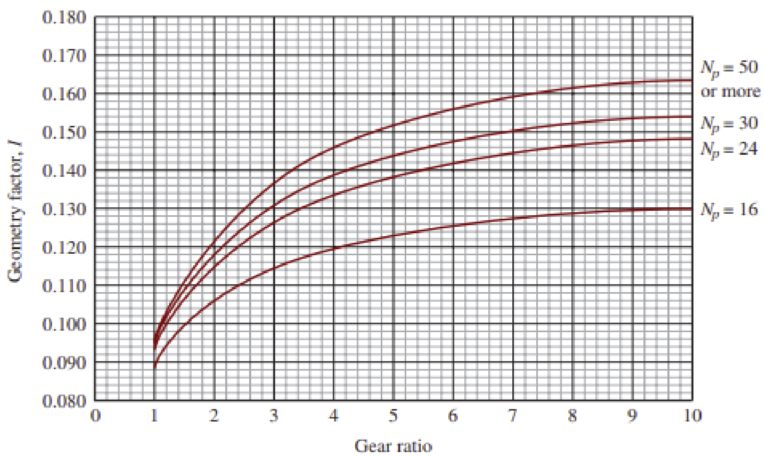


(*b*) 25° spur gear: standard addendum

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

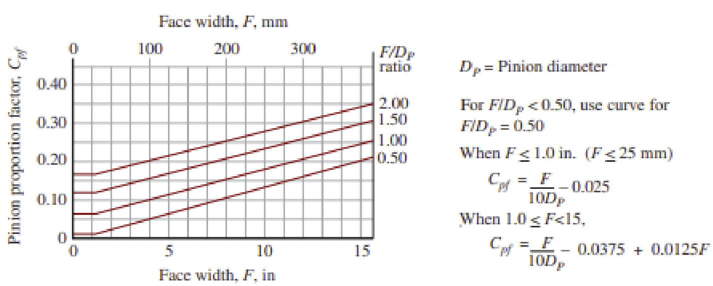


(*a*) 20° pressure angle, full-depth teeth (standard addendum = $1/P_d$)

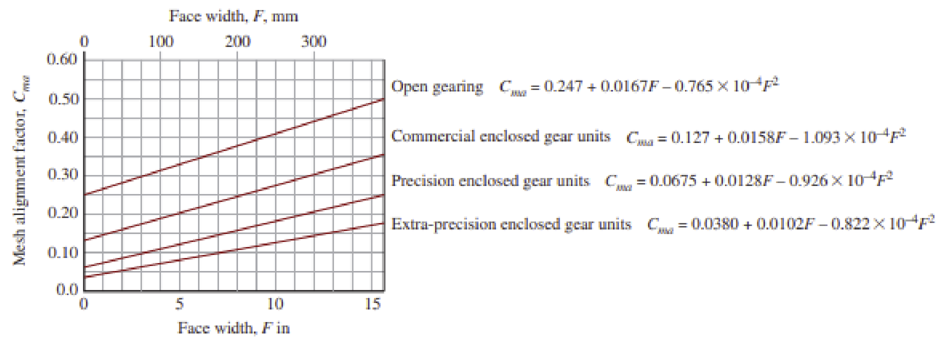


(*b*) 25° pressure angle, full-depth teeth (standard addendum = $1/P_d$)

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.



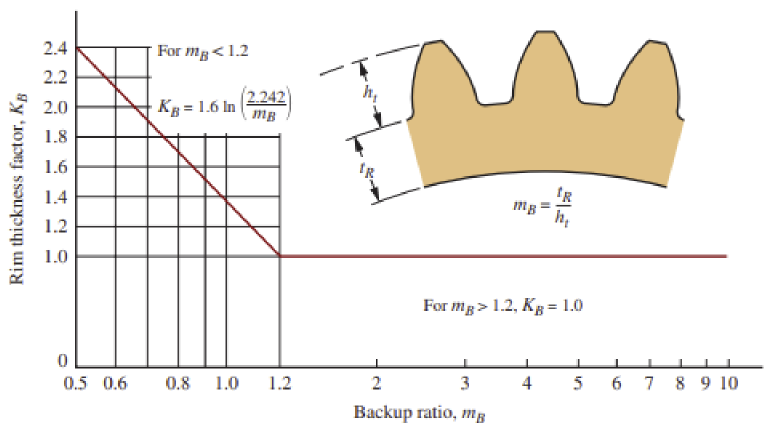
22. Compute $K_m = 1 + C_{pf} + C_{ma}$

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.

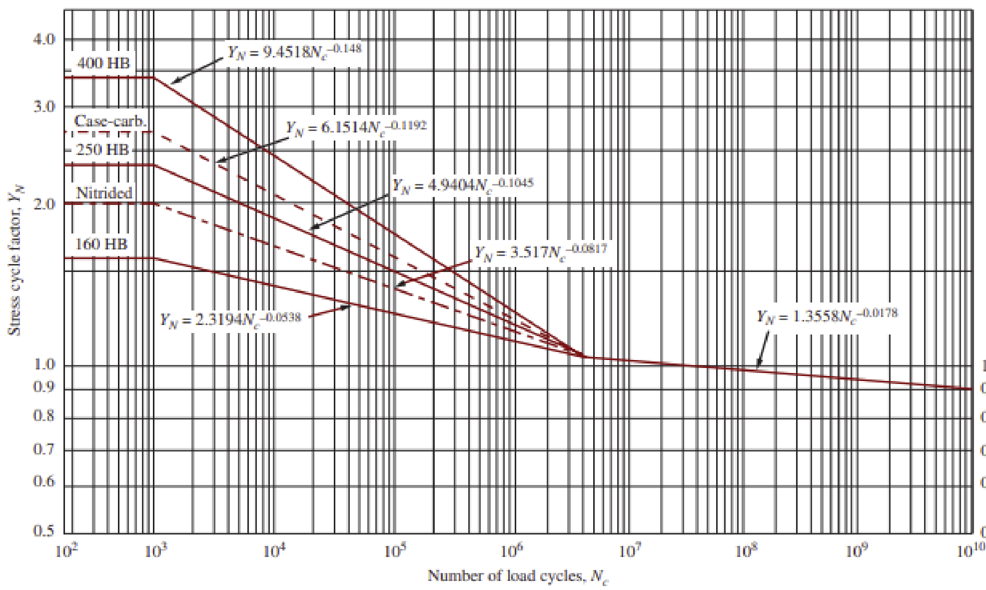
TABLE 9–12 Recommended Design Life	
Application	Design life (h)
Domestic appliances	1000–2000
Aircraft engines	1000–4000
Automotive	1500–5000
Agricultural equipment	3000–6000
Elevators, industrial fans, multipurpose gearing	8000–15 000
Electric motors, industrial blowers, general industrial machines	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.

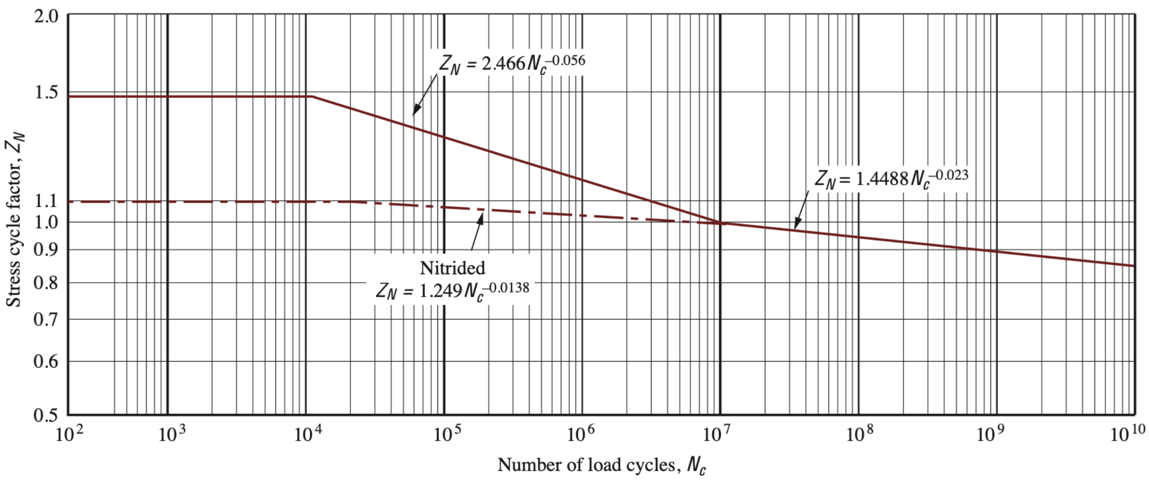
29. Find the number of loading cycles using these formulas

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

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: $6.5 \times 10^{-6}°F^{-1}$.

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)	(MPa)	Yield strength (ksi)	(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

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

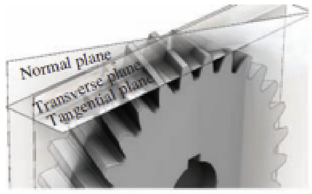
40. Anything with $HB > 400$ should use flame or induction hardening techniques over through hardening, since they can provide the strength required for high contact stress because their inside is still ductile to prevent failure. But fuck theory here's a table that does it for you

TABLE 9–8 Examples of Gear Materials

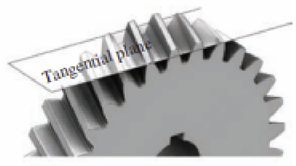
Heat treatment	Typical alloys (SAE numbers)
Through-hardened or Case-hardened by flame or induction hardening	1045, 4140, 4150, 4340, 4350
Carburizing, case hardened	1020, 4118, 4320, 4820, 8620, 9310

2.2 Helical Gears

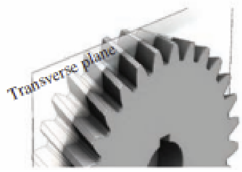
2.2.1 Anatomy



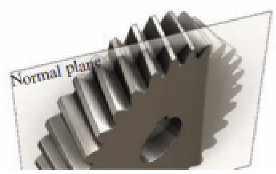
(a) The three primary planes defining helical gears



(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 $\cdot$ 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.

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

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

normal circular pitch: $p_n = p_t \cos\psi$

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

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

diametral pitch: $P_d = \dfrac{N}{D}$

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

$P_d p_t = \pi$

$P_{nd} p_n = \pi$

metric module: $m = \dfrac{D}{N}$

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

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

Forces and motion:

torque: $T = \dfrac{63000P}{n}$

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

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

radial force: $W_r = W_t \tan\phi_t$

axial force: $W_x = W_t \tan\psi$

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

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

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

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

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

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

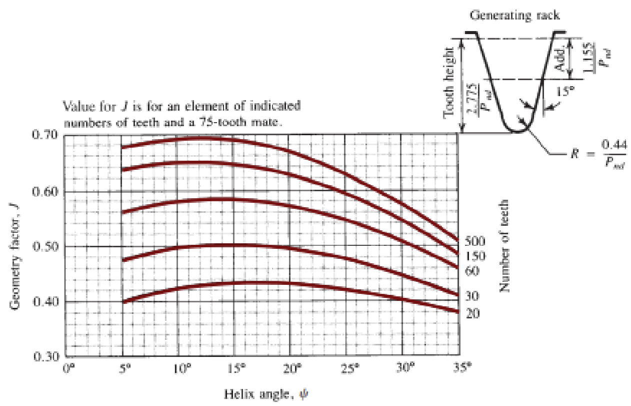
Round it however you want so that the value is convenient.

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

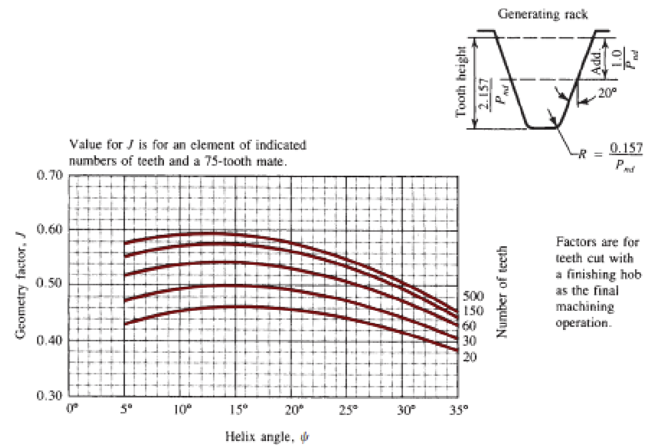
$$C = \frac{N_P + N_G}{2P_d}$$
$$v_t = \frac{\pi}{12} D_P n_P$$
$$W_t = \frac{33000P}{v_t}$$

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.

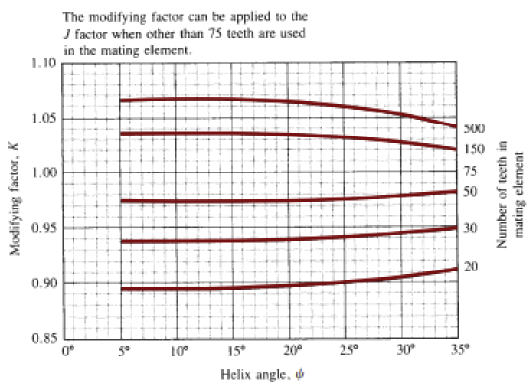
11. Choose the J_P and J_G values from one of the graphs depending on the normal pressure angle ϕ_n. (this is different from spur gears)



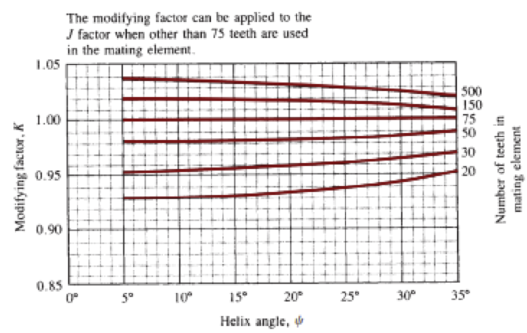
(a) Geometry factor (J) for 15° normal pressure angle and indicated addendum



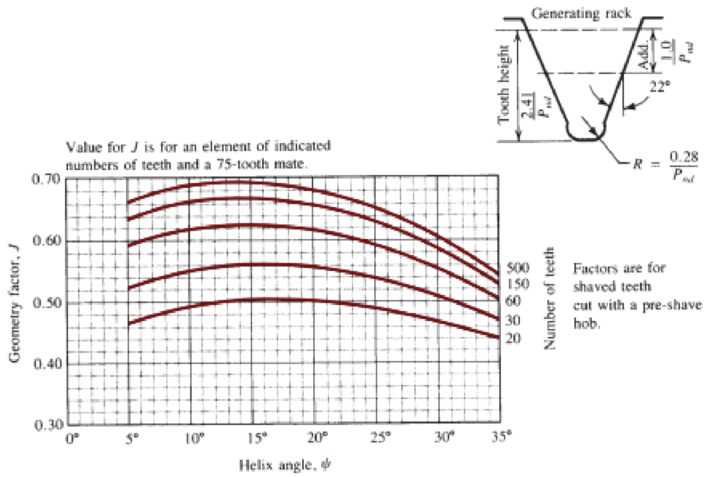
(a) Geometry factor (J) for 20° normal pressure angle, standard addendum, and finishing hob



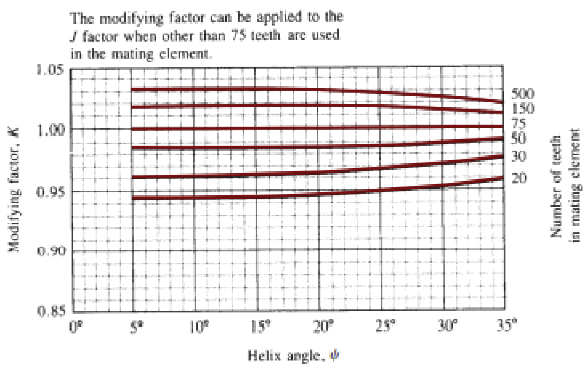
(b) J factor multipliers



(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 $\psi = 15.0°$

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 $\psi = 25.0°$

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 $\psi = 15.0°$

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 $\psi = 25.0°$

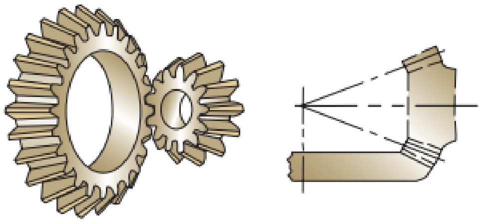
Gear teeth	Pinion teeth 12	14	17	21	26	35	55
12	0.129						
14	0.141	0.132					
17	0.155	0.146	0.135				
21	0.170	0.162	0.151	0.138			
26	0.185	0.177	0.166	0.154	0.141		
35	0.203	0.197	0.188	0.176	0.163	0.144	
55	0.227	0.223	0.216	0.207	0.196	0.178	0.148
135	0.259	0.258	0.255	0.251	0.246	0.235	0.213

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.

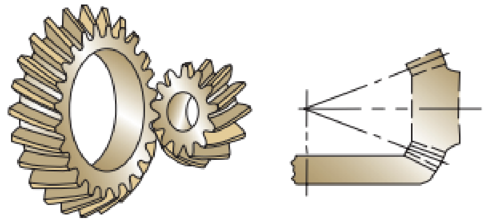
13. Compute s_{tP} and s_{tG} and follow the remaining steps in the spur gear design selection guide.

2.3 Bevel Gears

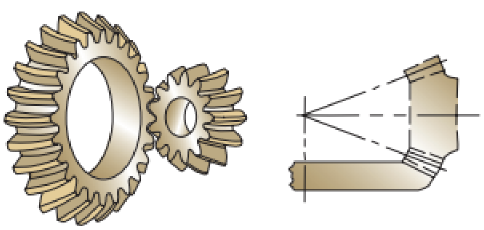
2.3.1 Anatomy



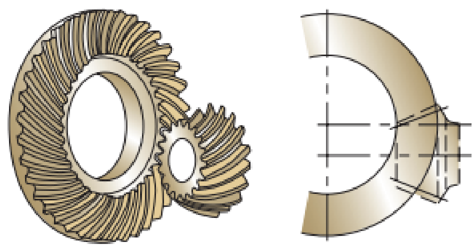
(a) Straight bevel



(b) Spiral bevel

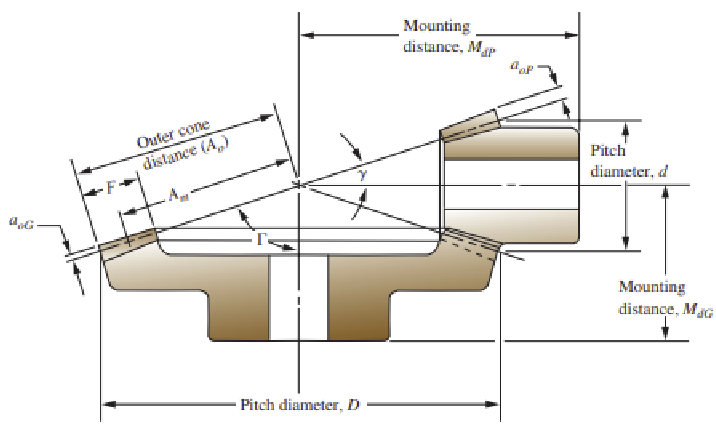


(c) Zero spiral bevel



(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.

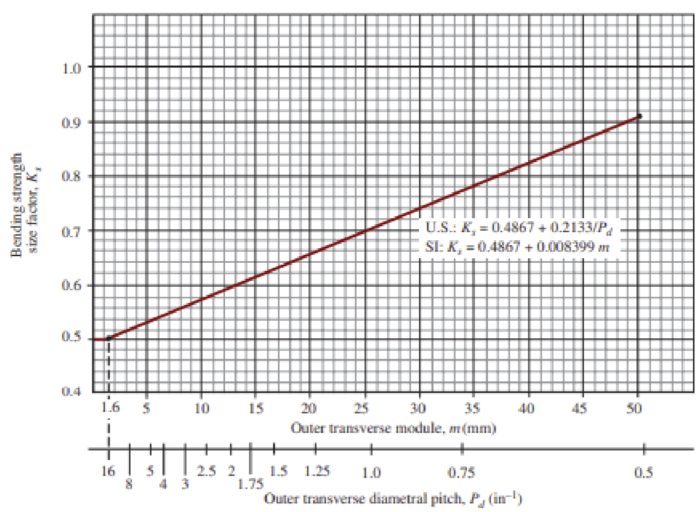
2. compute or guess any of the basic mechanical values missing, such as N or F, using the spur gear guide as reference.

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.

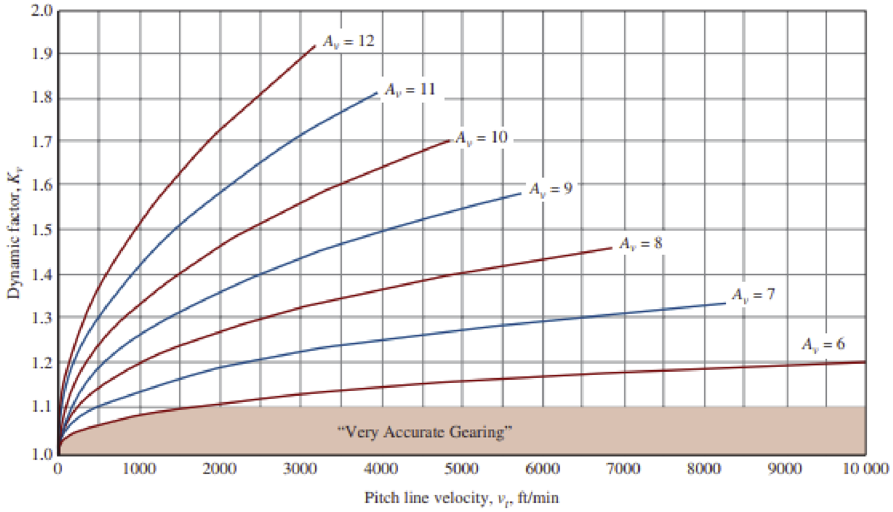
TABLE 9–5 Recommended AGMA Quality Numbers

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

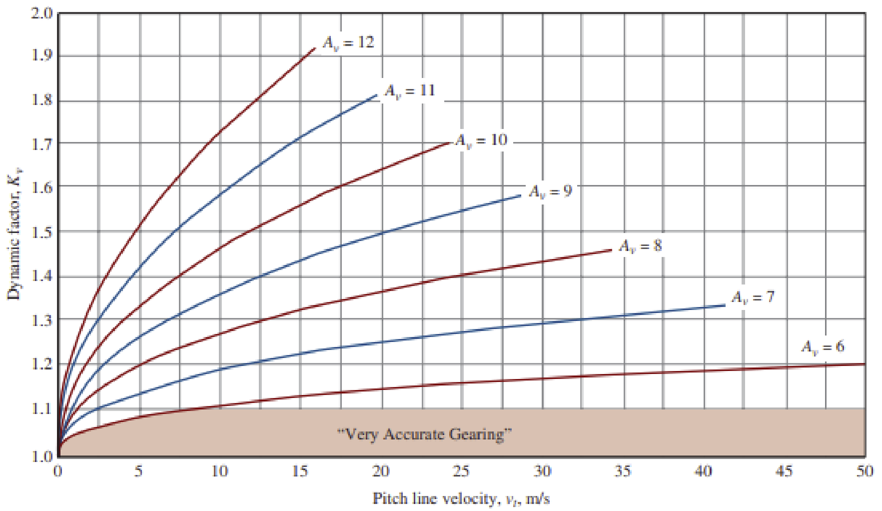
Machine tool drives and drives for other high-quality mechanical systems

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

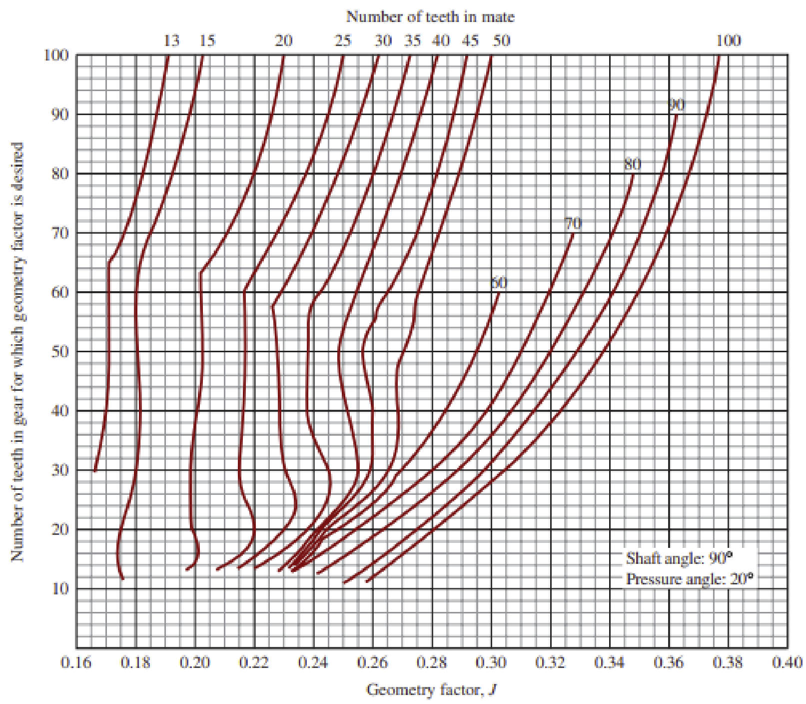


(*a*) U.S. Customary units



(*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 R	Interpretation	Reliability factors: Bending K_R	Reliability factors: 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

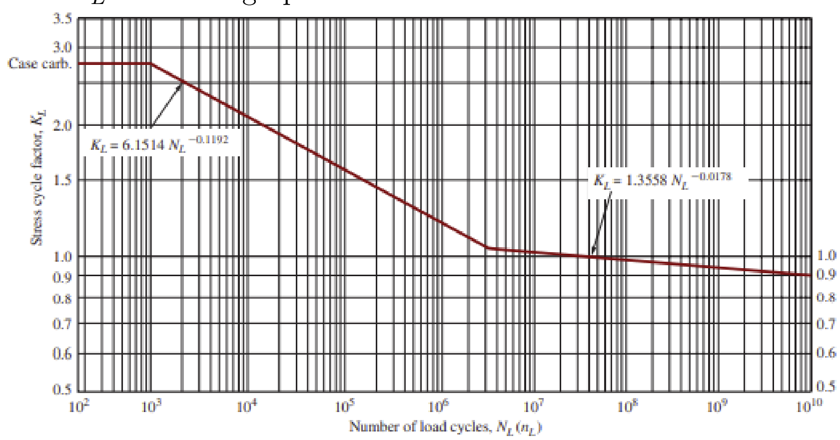
Application	Design life (h)
Domestic appliances	1000–2000
Aircraft engines	1000–4000
Automotive	1500–5000
Agricultural equipment	3000–6000
Elevators, industrial fans, multipurpose gearing	8000–15 000
Electric motors, industrial blowers, general industrial machines	20 000–30 000
Pumps and compressors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



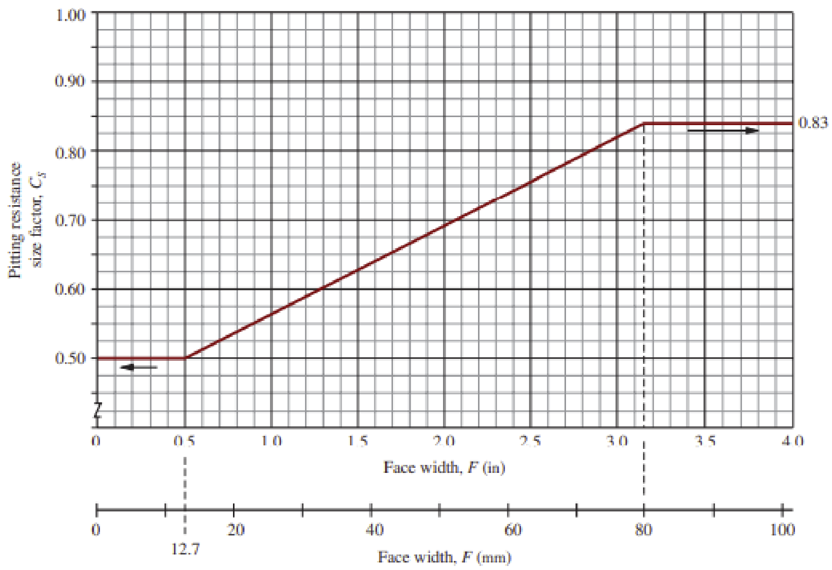
17. Take the safety factor SF to be anywhere between 1 and 1.5. We always just assume $SF = 1$ because fuck safety.

18. find the max allowable bending strength

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

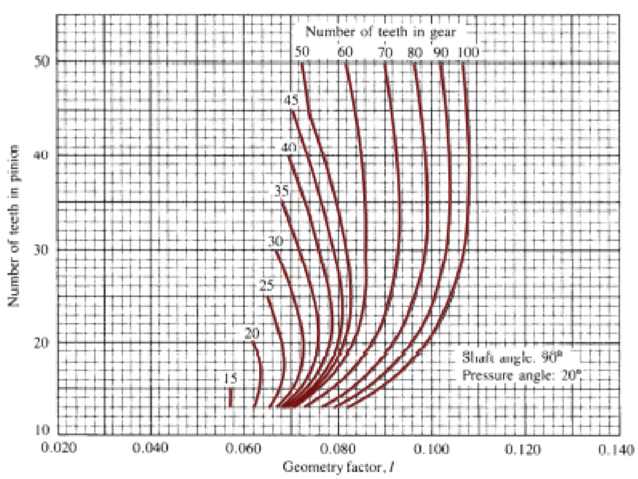
19. Take $C_p = 2300$ for steel

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.

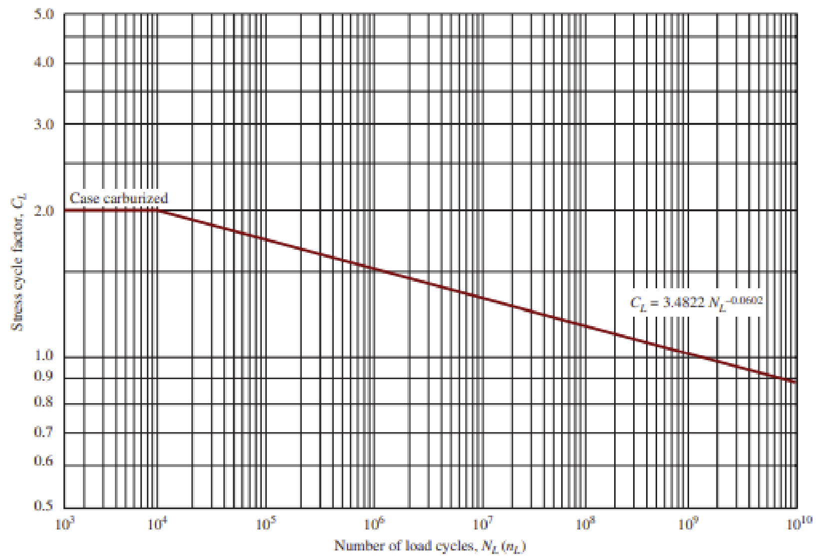
22. Get I from this graph



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



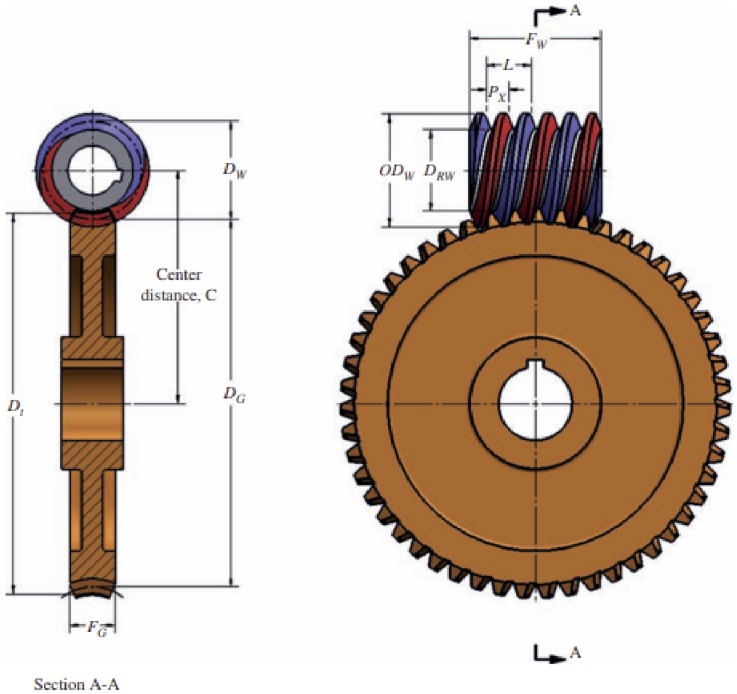
25. Get the max allowable contact stress number

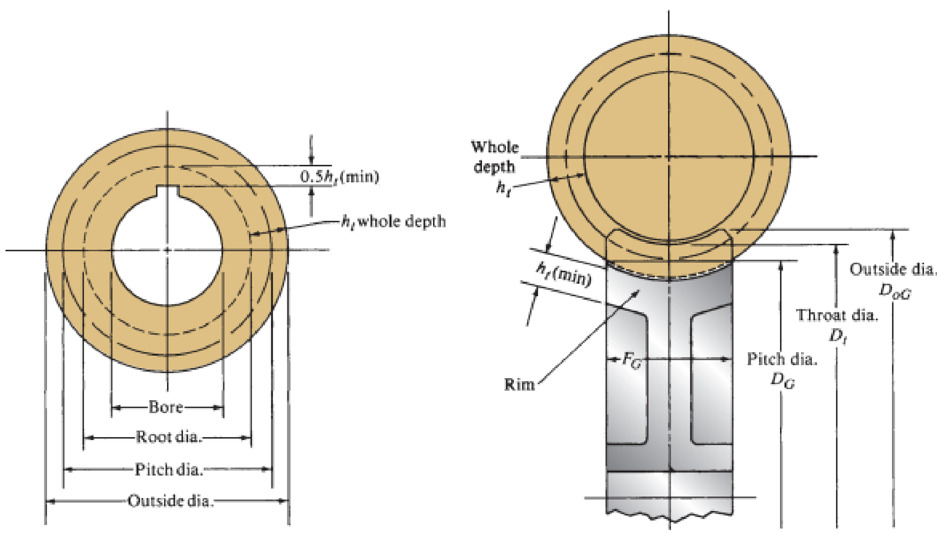
$$s_{ac} = \frac{s_c(SF)C_R}{C_L}$$

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

2.4 Worm Gears

2.4.1 Anatomy





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:

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

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

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

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

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

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

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

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

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

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

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:

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

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

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

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

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

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

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

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

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

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

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

6. Compute the forces on the gear

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

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

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

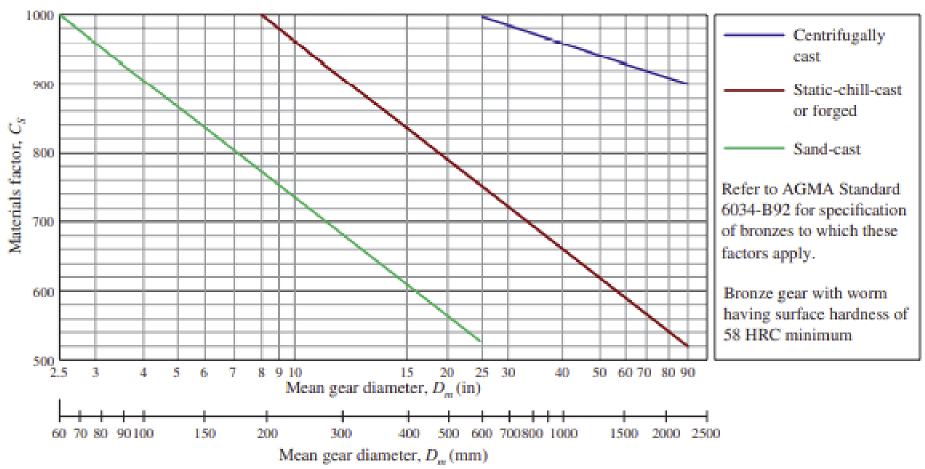
14. Compute the dynamic load

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

15. Find the stress in the gear teeth

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

16. Find C_s



For sand-cast bronze

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

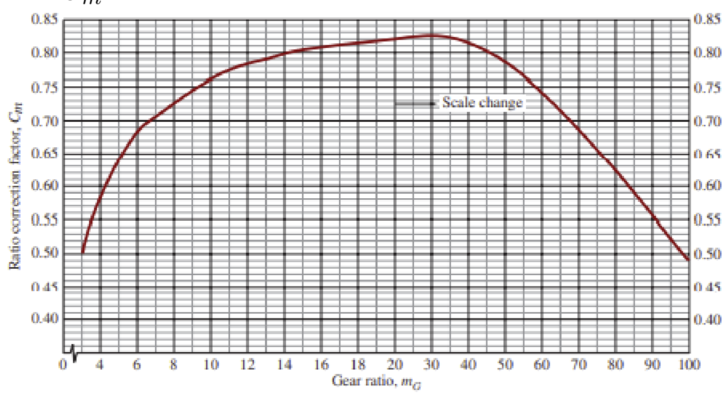
For static-chill-cast or forged bronze

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

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

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

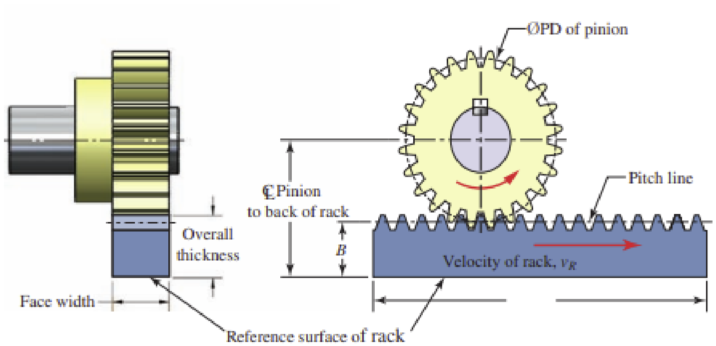
20. Find the rated tangential load

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

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 Fluid Diagrams

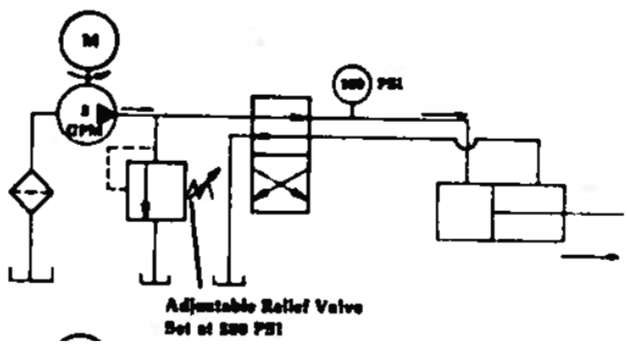
What the fuck is going on?

3.2 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 3.3.3 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:

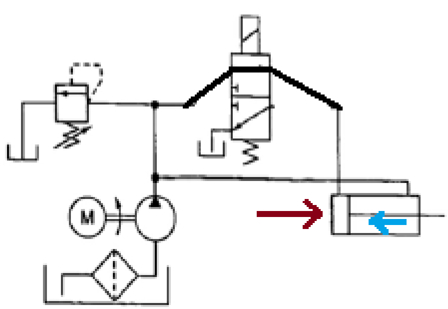


Figure 13-22 Regenerative circuit.

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.

3.3 Selecting Bores and Pumps and Shit

3.3.1 Nomenclature

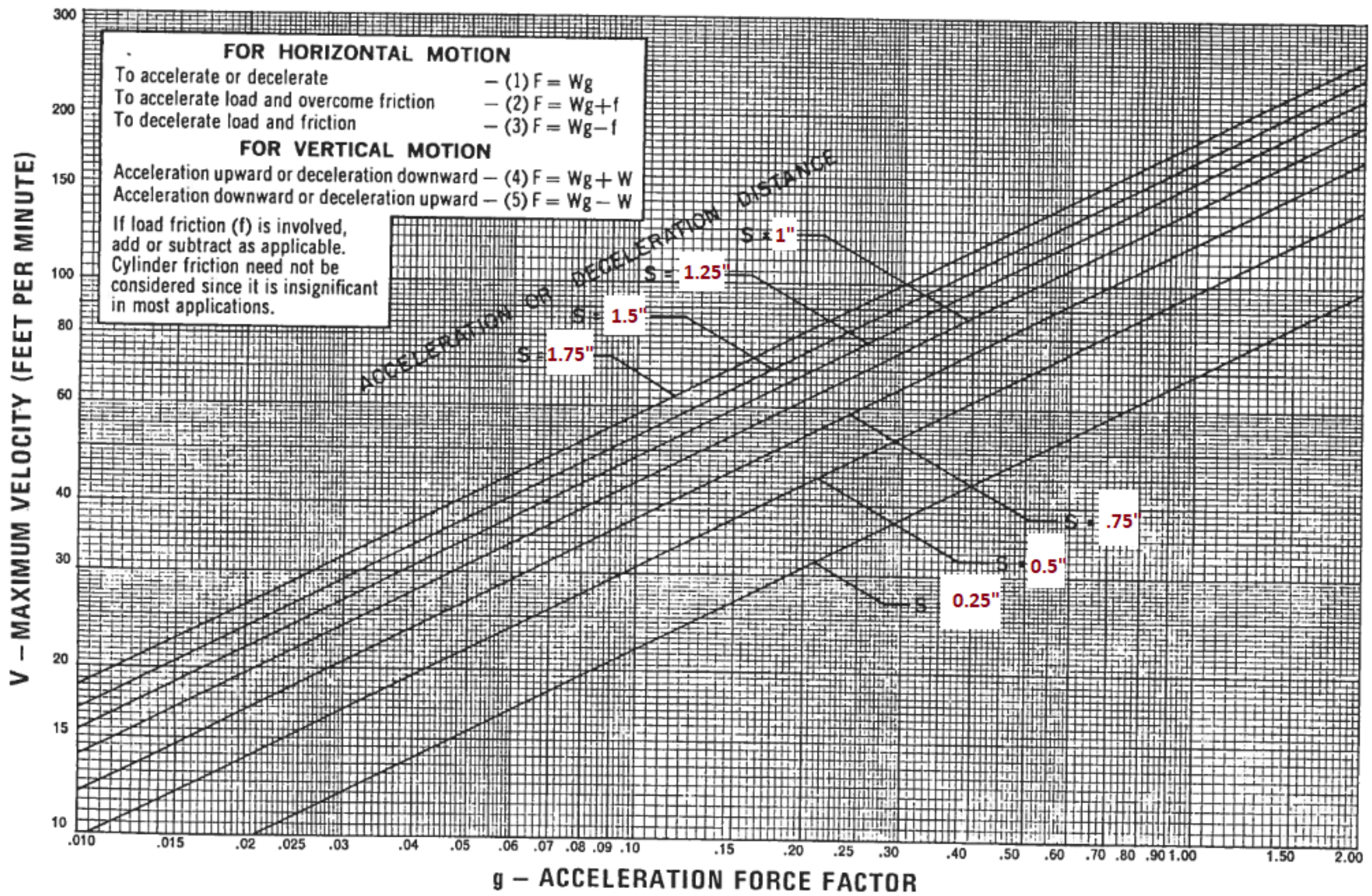
Nomenclature is not very well defined so here is what I will use:
s = distance (in)
v = speed (ft/min)
F = (total) force (lb)
F_w = force of weight (lb)
F_a = force from acceleration (lb)
F_f = force due to friction (lb)
μ = coefficient of friction
g = acceleration constant (from table. Not the same 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)

3.3.2 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:

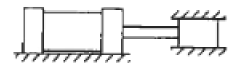
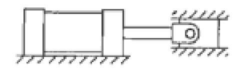
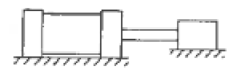
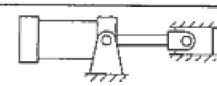
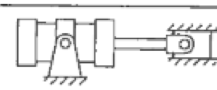
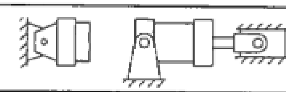
$$F_a = F_w g$$

(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

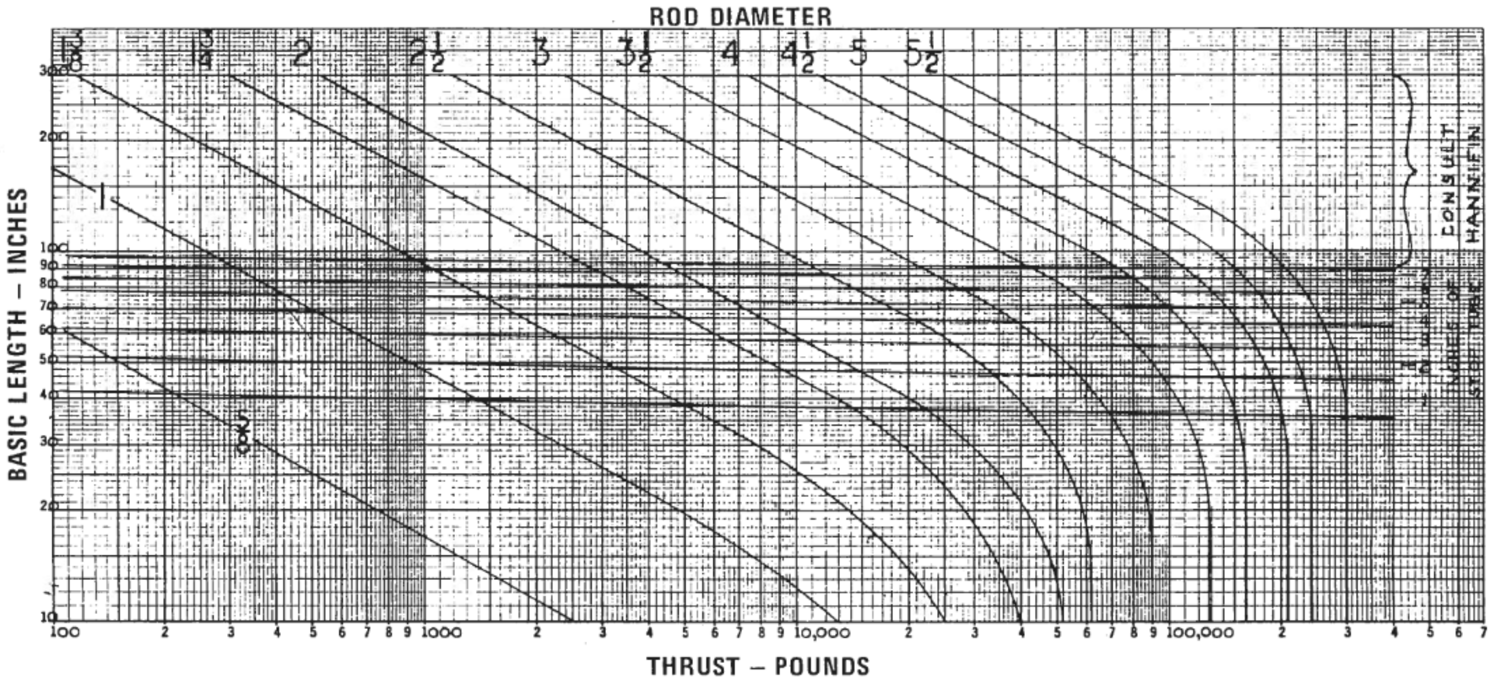
3. Compute the basic length, L_b, from the stroke length, L (specified in question) and the stroke factor, k

$$L_b = kL$$

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



3.3.3 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.

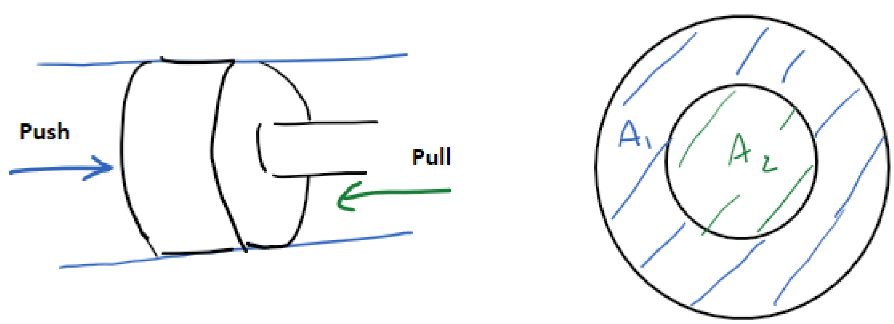


Figure 1: Piston Area and Motion

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	$F = P \cdot (A_1)$	$F = P \cdot (A_1)$

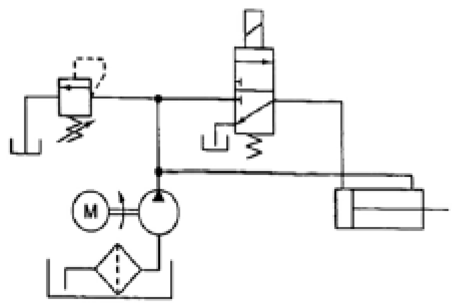


Figure 13-22 Regenerative circuit.

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;

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

3.3.4 Volume Flow Rate

6. Compute the pump capacity, Q in gpm using v in ft/min and A in in^2.

$$Q_p = v_p A_p \left(\frac{12}{231}\right) = \frac{\pi}{4} v_p d_p^2 \left(\frac{12}{231}\right)$$

	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	$Q = \frac{\pi}{4} v_{push} (\mathbf{d_{bore}^2} - \mathbf{d_{rod}^2}) \left(\frac{12}{231}\right)$	$Q = \frac{\pi}{4} v_{push} (\mathbf{d_{bore}^2} - \mathbf{d_{rod}^2}) \left(\frac{12}{231}\right)$

3.3.5 Pipe Sizing

7. 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.

8. 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	.614
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

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

CYLINDER BORE-INCHES	PISTON ROD		CYLINDER NET AREA SQ. IN.	FLUID DISPLACEMENT AT 10 FT. PER MINUTE PISTON VELOCITY		FLUID VELOCITY (IN FEET PER SECOND) THROUGH EXTRA HEAVY PIPE AT 10 F.P.M. PISTON SPEED.								
	DIA.-INCHES	AREA SQ. IN.		G.P.M.	C.F.M.	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	.....

7	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	.....
	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	.....
8	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
	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
10	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
	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
12	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
	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
14	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
	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

3.3.6 Horsepower for Motor

10. 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

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

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
	Wire Capacity: RH Amps.	15	20	30	45	65	85	100	115	130	150	175	200	230	255	285	310	335	380	475	545
	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
	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.

motor starter, conduit and wire size

3.4 Circuit Analysis Techniques

3.4.1 Flowrate

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

$$Q = Av$$

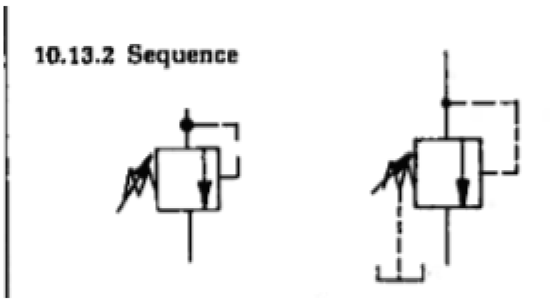
Use the area of the fluid column and the movement speed of the piston head to get Q (usually in in^3) then convert to GPM with $\frac{1\text{in}^3}{\text{s}} = \frac{60}{231}\text{GPM}$

3.4.2 Power

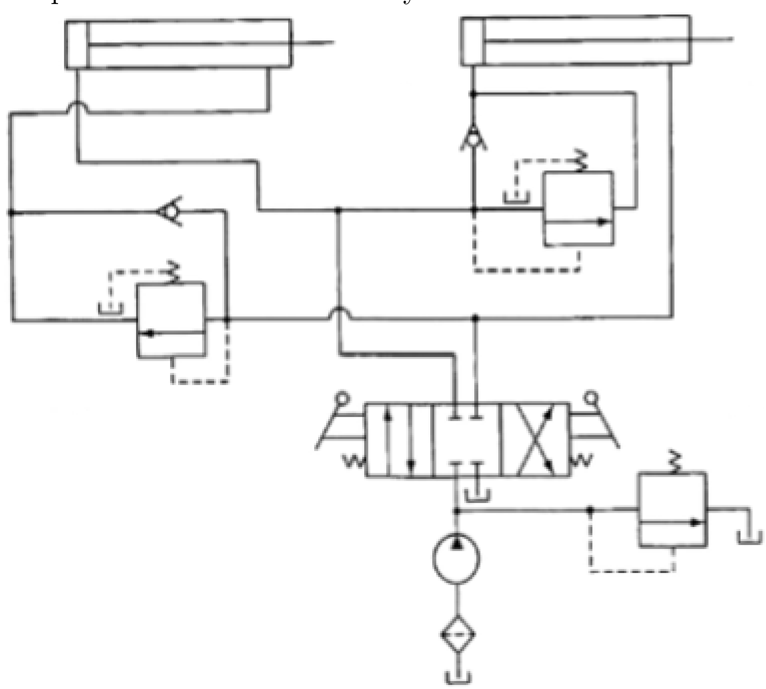
Work is pressure times volume, so power is the time derivative. Conversion factor for Q in GPM and P in psi included below.

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

3.4.3 Other Components

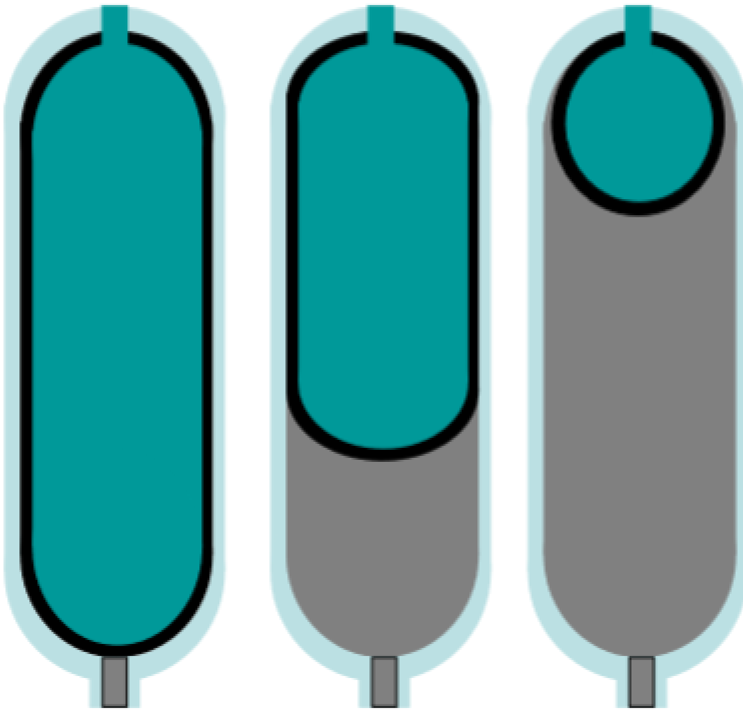


Sequence valves are circuit components which can control the order in which hydraulic elements are actuated. Sequences valves only allow fluid to flow through them once a minimum pressure has been reached to actuate them. This pressure is usually reached after a piston reaches maximum extension stops 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.