Stupid Mech 325 Stupid Summary

Fuck this course

The New Testament

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Contents

1 Bearings, Bushings, and Other Shit That Spins

1.1 Boundary-Lubricated Bearings (Bushings)

1.1.1 Anatomy

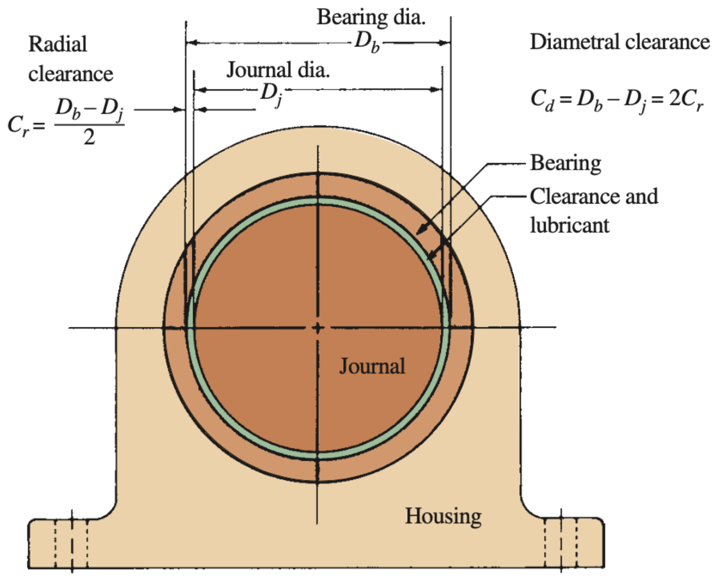


FIGURE 16–1 Bearing geometry

1.1.2 Design Selection

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

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

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

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

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

Table 12–10 Coefficients of Friction

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

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

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

where f_s is coefficient of friction,

n_d is the design factor,

F is the radial load,

N is the angular speed of the bearing,

T_f is the lubricant temperature/max temperature

T_∞ is the ambient temperature

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

7. Calculate the nominal pressure (in psi):

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

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

$$V = \frac{\pi D N}{12}$$

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

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

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

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

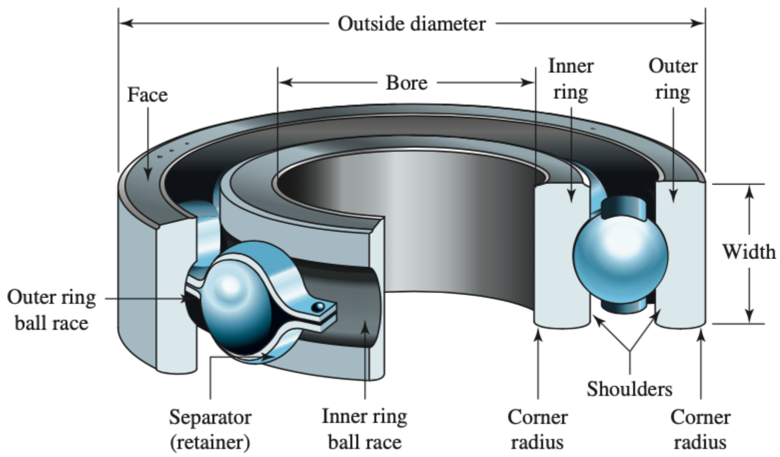
11. If all values are within acceptable ranges, congrats you have selected an acceptable bushing!

1.2 Ball and Cylindrical Roller Bearings

Types of bearings that fall under this category are:

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

1.2.1 Anatomy



1.2.2 Design Selection - Radial Load Only

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

- For cylindrical roller bearings:

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

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

9. We're done!

1.2.3 Design Selection - Radial and Thrust Loads

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

Table 11–1 Equivalent Radial Load Factors for Ball Bearings

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

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

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

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

where F_r is the radial load,

F_a is the axial/thrust load

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

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

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

5. Select a bearing that has a C_{10} value that exceeds the calculated one. Refer to tables 11-12 and 11-13 in **step 8** of the radial section right above.

6. Find the C_0 value for the chosen bearing.

7. Calculate F_a/C_0 where F_a is the axial load.

8. Look back at table 11-1 in **step 2** of this section and find the closest value of F_a/C_0. Get e from this.

9. Check if $F_a/(VF_r) > e$. If so, we will be looking for Y_2 values in table 11-1 from **step 2**. If not, we will be looking for Y_1 values.

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

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

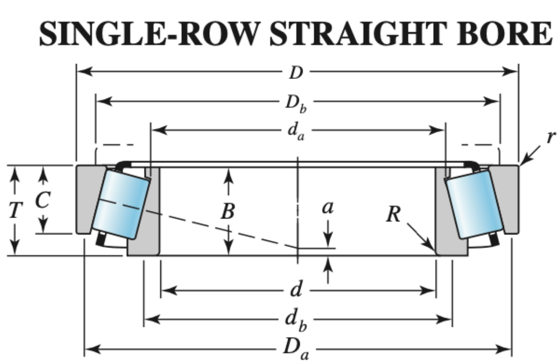
12. Calculate the new C_{10}. The calculation for C_{10} only changes in F_e so we can do the following calculation:

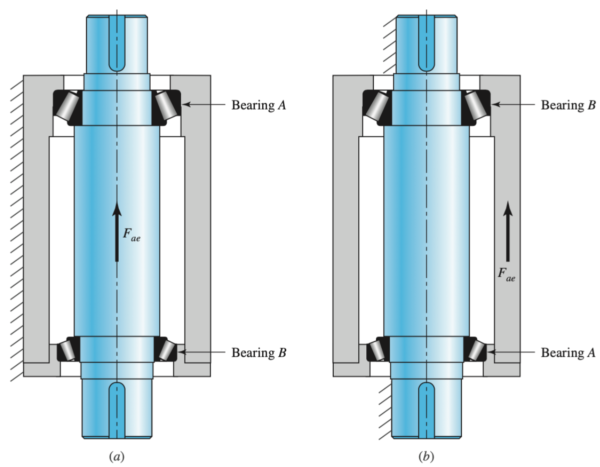
    $$\text{new } C_{10} = \frac{\text{new } F_e}{\text{old } F_e}(\text{old } C_{10})$$

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

1.3 Tapered Roller Bearings

1.3.1 Anatomy





1.3.2 Design Selection

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

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

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

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

3. Calculate the equivalent loads.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

bore d	outside diameter D	width T	rating at 500 rpm for 3000 hours L_{10} one-row radial N lbf	rating at 500 rpm for 3000 hours L_{10} thrust N lbf	factor K	eff. load center a②	part numbers cone	part numbers cup	cone: max shaft fillet radius R①	cone: width B	cone: backing shoulder diameters d_b	cone: backing shoulder diameters d_a	cup: max housing fillet radius r①	cup: width C	cup: backing shoulder diameters D_b	cup: backing shoulder diameters D_a
25.000	**52.000**	**16.250**	**8190**	**5260**	**1.56**	**–3.6**	**◆30205**	**◆30205**	**1.0**	**15.000**	**30.5**	**29.0**	**1.0**	**13.000**	**46.0**	**48.5**
0.9843	2.0472	0.6398	1840	1180		–0.14			0.04	0.5906	1.20	1.14	0.04	0.5118	1.81	1.91
25.000	**52.000**	**19.250**	**9520**	**9510**	**1.00**	**–3.0**	**◆32205-B**	**◆32205-B**	**1.0**	**18.000**	**34.0**	**31.0**	**1.0**	**15.000**	**43.5**	**49.5**
0.9843	2.0472	0.7579	2140	2140		–0.12			0.04	0.7087	1.34	1.22	0.04	0.5906	1.71	1.95
25.000	**52.000**	**22.000**	**13200**	**7960**	**1.66**	**–7.6**	**◆33205**	**◆33205**	**1.0**	**22.000**	**34.0**	**30.5**	**1.0**	**18.000**	**44.5**	**49.0**
0.9843	2.0472	0.8661	2980	1790		–0.30			0.04	0.8661	1.34	1.20	0.04	0.7087	1.75	1.93
25.000	**62.000**	**18.250**	**13000**	**6680**	**1.95**	**–5.1**	**◆30305**	**◆30305**	**1.5**	**17.000**	**32.5**	**30.0**	**1.5**	**15.000**	**55.0**	**57.0**
0.9843	2.4409	0.7185	2930	1500		–0.20			0.06	0.6693	1.28	1.18	0.06	0.5906	2.17	2.24
25.000	**62.000**	**25.250**	**17400**	**8930**	**1.95**	**–9.7**	**◆32305**	**◆32305**	**1.5**	**24.000**	**35.0**	**31.5**	**1.5**	**20.000**	**54.0**	**57.0**
0.9843	2.4409	0.9941	3910	2010		–0.38			0.06	0.9449	1.38	1.24	0.06	0.7874	2.13	2.24
25.159	**50.005**	**13.495**	**6990**	**4810**	**1.45**	**–2.8**	**07096**	**07196**	**1.5**	**14.260**	**31.5**	**29.5**	**1.0**	**9.525**	**44.5**	**47.0**
0.9905	1.9687	0.5313	1570	1080		–0.11			0.06	0.5614	1.24	1.16	0.04	0.3750	1.75	1.85
25.400	**50.005**	**13.495**	**6990**	**4810**	**1.45**	**–2.8**	**07100**	**07196**	**1.0**	**14.260**	**30.5**	**29.5**	**1.0**	**9.525**	**44.5**	**47.0**
1.0000	1.9687	0.5313	1570	1080		–0.11			0.04	0.5614	1.20	1.16	0.04	0.3750	1.75	1.85
25.400	**50.005**	**13.495**	**6990**	**4810**	**1.45**	**–2.8**	**07100-S**	**07196**	**1.5**	**14.260**	**31.5**	**29.5**	**1.0**	**9.525**	**44.5**	**47.0**
1.0000	1.9687	0.5313	1570	1080		–0.11			0.06	0.5614	1.24	1.16	0.04	0.3750	1.75	1.85
25.400	**50.292**	**14.224**	**7210**	**4620**	**1.56**	**–3.3**	**L44642**	**L44610**	**3.5**	**14.732**	**36.0**	**29.5**	**1.3**	**10.668**	**44.5**	**47.0**
1.0000	1.9800	0.5600	1620	1040		–0.13			0.14	0.5800	1.42	1.16	0.05	0.4200	1.75	1.85
25.400	**50.292**	**14.224**	**7210**	**4620**	**1.56**	**–3.3**	**L44643**	**L44610**	**1.3**	**14.732**	**31.5**	**29.5**	**1.3**	**10.668**	**44.5**	**47.0**
1.0000	1.9800	0.5600	1620	1040		–0.13			0.05	0.5800	1.24	1.16	0.05	0.4200	1.75	1.85
25.400	**51.994**	**15.011**	**6990**	**4810**	**1.45**	**–2.8**	**07100**	**07204**	**1.0**	**14.260**	**30.5**	**29.5**	**1.3**	**12.700**	**45.0**	**48.0**
1.0000	2.0470	0.5910	1570	1080		–0.11			0.04	0.5614	1.20	1.16	0.05	0.5000	1.77	1.89
25.400	**56.896**	**19.368**	**10900**	**5740**	**1.90**	**–6.9**	**1780**	**1729**	**0.8**	**19.837**	**30.5**	**30.0**	**1.3**	**15.875**	**49.0**	**51.0**
1.0000	2.2400	0.7625	2450	1290		–0.27			0.03	0.7810	1.20	1.18	0.05	0.6250	1.93	2.01
25.400	**57.150**	**19.431**	**11700**	**10900**	**1.07**	**–3.0**	**M84548**	**M84510**	**1.5**	**19.431**	**36.0**	**33.0**	**1.5**	**14.732**	**48.5**	**54.0**
1.0000	2.2500	0.7650	2620	2450		–0.12			0.06	0.7650	1.42	1.30	0.06	0.5800	1.91	2.13
25.400	**58.738**	**19.050**	**11600**	**6560**	**1.77**	**–5.8**	**1986**	**1932**	**1.3**	**19.355**	**32.5**	**30.5**	**1.3**	**15.080**	**52.0**	**54.0**
1.0000	2.3125	0.7500	2610	1470		–0.23			0.05	0.7620	1.28	1.20	0.05	0.5937	2.05	2.13
25.400	**59.530**	**23.368**	**13900**	**13000**	**1.07**	**–5.1**	**M84249**	**M84210**	**0.8**	**23.114**	**36.0**	**32.5**	**1.5**	**18.288**	**49.5**	**56.0**
1.0000	2.3437	0.9200	3140	2930		–0.20			0.03	0.9100	1.42	1.27	0.06	0.7200	1.95	2.20
25.400	**60.325**	**19.842**	**11000**	**6550**	**1.69**	**–5.1**	**15578**	**15523**	**1.3**	**17.462**	**32.5**	**30.5**	**1.5**	**15.875**	**51.0**	**54.0**
1.0000	2.3750	0.7812	2480	1470		–0.20			0.05	0.6875	1.28	1.20	0.06	0.6250	2.01	2.13
25.400	**61.912**	**19.050**	**12100**	**7280**	**1.67**	**–5.8**	**15101**	**15243**	**0.8**	**20.638**	**32.5**	**31.5**	**2.0**	**14.288**	**54.0**	**58.0**
1.0000	2.4375	0.7500	2730	1640		–0.23			0.03	0.8125	1.28	1.24	0.08	0.5625	2.13	2.28
25.400	**62.000**	**19.050**	**12100**	**7280**	**1.67**	**–5.8**	**15100**	**15245**	**3.5**	**20.638**	**38.0**	**31.5**	**1.3**	**14.288**	**55.0**	**58.0**
1.0000	2.4409	0.7500	2730	1640		–0.23			0.14	0.8125	1.50	1.24	0.05	0.5625	2.17	2.28
25.400	**62.000**	**19.050**	**12100**	**7280**	**1.67**	**–5.8**	**15101**	**15245**	**0.8**	**20.638**	**32.5**	**31.5**	**1.3**	**14.288**	**55.0**	**58.0**
1.0000	2.4409	0.7500	2730	1640		–0.23			0.03	0.8125	1.28	1.24	0.05	0.5625	2.17	2.28

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

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

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

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

1.4 Shafts and Keys

1.4.1 Design Selection

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

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

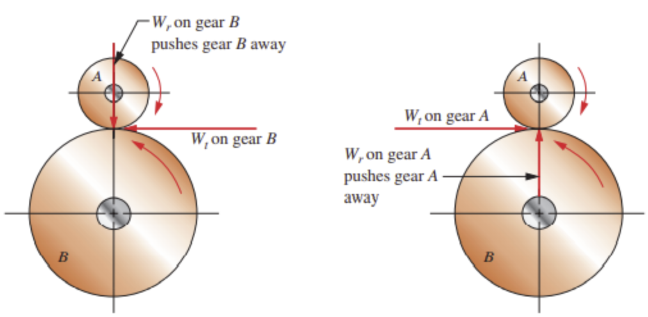
   Verify that the sum of torques is equal to 0

2. Compute the force from each sheave/gear

   (a) For spur gears:

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

   $D =$ pitch diameter
   $\phi =$ pressure angle
   The direction will be given by

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

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

   (b) For helical gears:

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

   $\phi_n =$ normal pressure angle
   $\psi =$ helix angle

(c) Bevel gears:

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

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

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

(e) Chain sprockets:

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

(f) V-Belt sheaves:

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

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

(g) Flat belt pulleys:

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

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

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

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

$$\sum F_x = 0$$
$$\sum F_y = 0$$
$$\sum M_x = 0$$
$$\sum M_y = 0$$

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

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

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

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

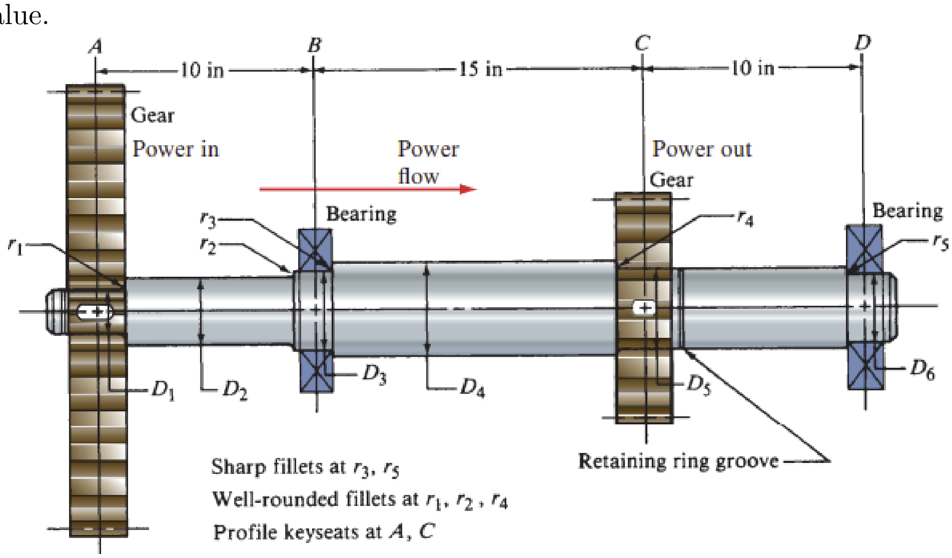
7. Draw the torque diagram for the shaft (exact same idea as how to draw shear diagram)
   Note that we will only have torque gains/drops across gears/pinions and not bearings

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

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

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

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



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

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

- $K_t = 2$ (profile keyseat)

- $K_t = 1.6$ (sled runner keyseat)

We always assume profile unless told otherwise.

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

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

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

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

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

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

- $K_t = 3$ (retaining ring)

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

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

11. Specify a material factor, C_m

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

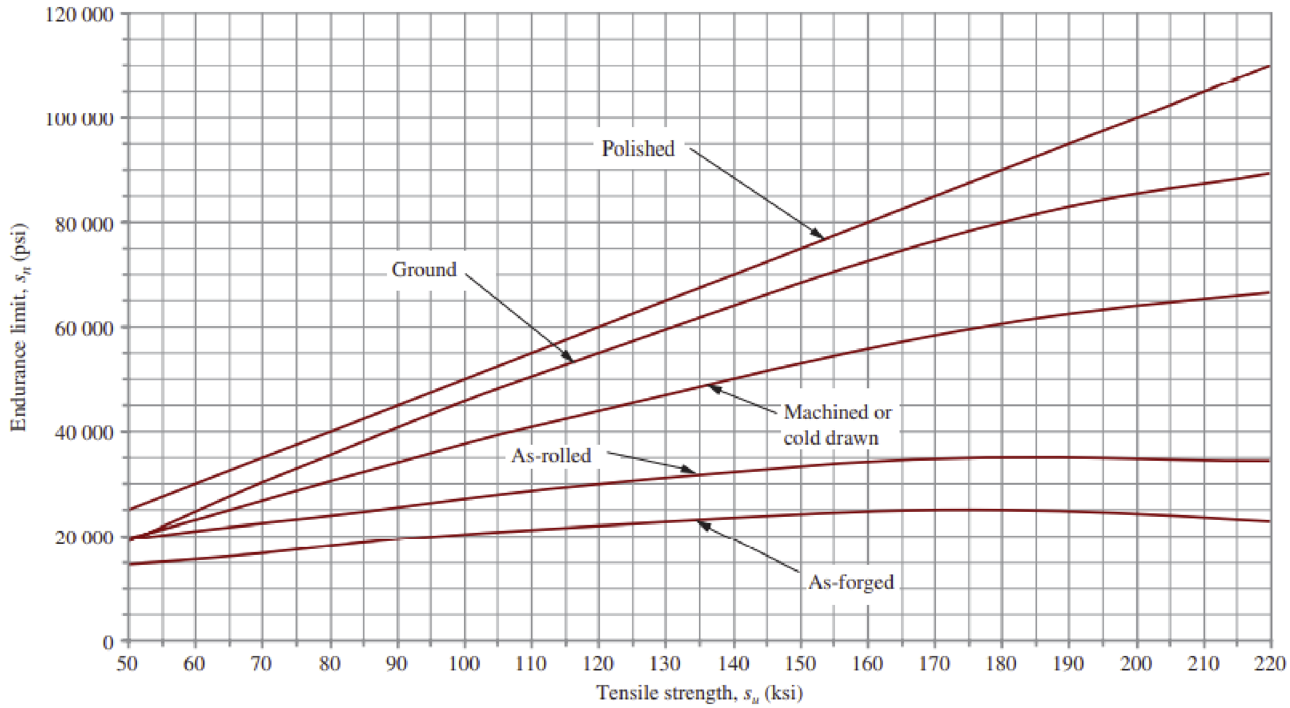
If not specified, assume Wrought steel and $C_m = 1$

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

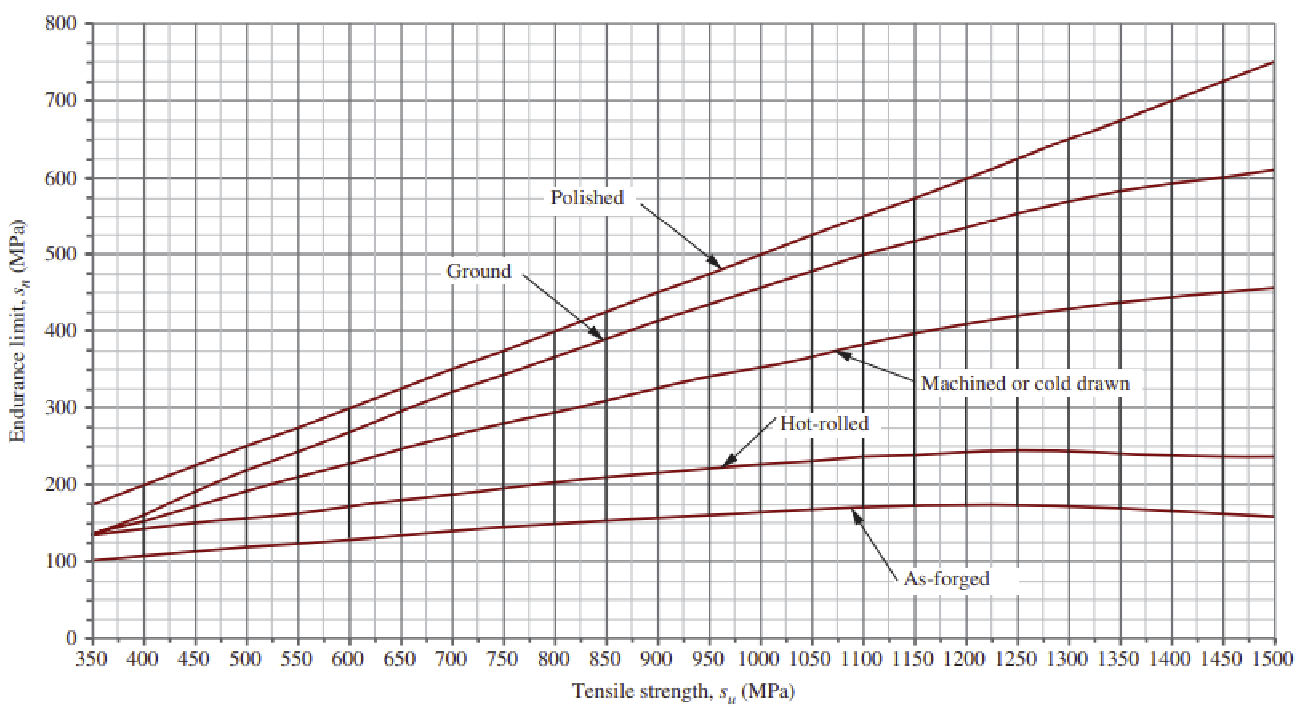
- $C_{st} = 1$ (bending stress)
- $C_{st} = 0.8$ (axial tension)

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

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



(*a*) U.S. customary units



(*b*) SI metric units

14. Get the relaibility factor C_R, from this table

TABLE 5–3 Approximate Reliability Factors, C_R

Desired reliability	C_R
0.50	1.0
0.90	0.90
0.99	0.81
0.999	0.75

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

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

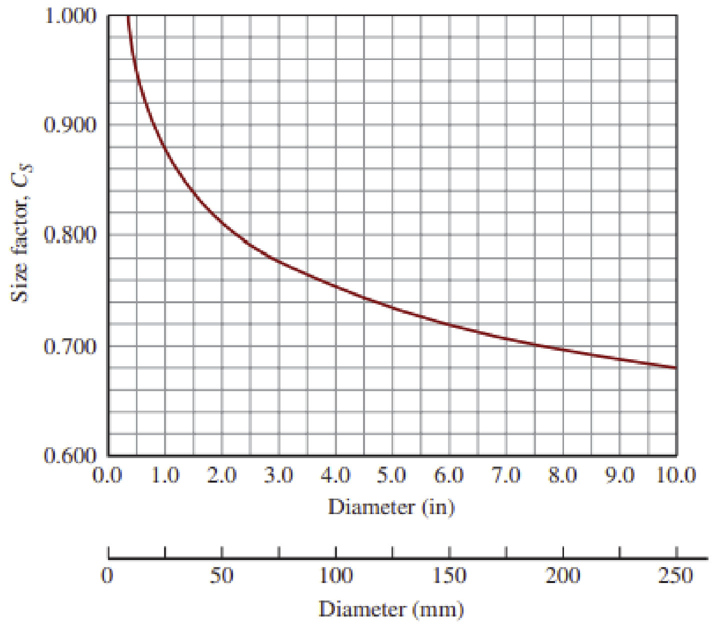


TABLE 5–4 Size Factors

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

16. Compute the estimated actual endurance limit s'_n

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

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

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

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

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

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

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

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

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

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

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

21. Find the C value for the bearing:

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

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

    (b) If the reliability is not 90%, then you have to

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

    $$\frac{L_2}{L_1} = (\frac{C_{10}}{P_2})^k \qquad\qquad C_{10} = P_2(\frac{L_2}{1 \times 10^6})^{\frac{1}{k}}$$

22. Use the C_{10} value and the minimum diameter of the shaft at the bearing to select a suitable bearing from the table

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

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

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

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

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

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

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

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

TABLE 15–5 Shaft and Housing Fits for Bearings

A. Shaft fits

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

B. Housing fits

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

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

B. Housing fits

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

Note: L = loose; and T = tight.

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

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

TABLE A2–1 Preferred Basic Sizes

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

26. Select a material for the keys and get the yield strength, s_y.
SAE 1018 Carbon steel is a pretty sexy choice. I'd recommend that one.

TABLE 11–4 Examples of Materials Used for Keys

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

Source: Adapted from Internet site 20.

Note: Strength properties typical, not guaranteed.

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

TABLE 11–1 Key Size vs. Shaft Diameter

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

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

28. Compute the minimum required length of the key

   (a) If the material of the key is weakest and you use a square key then use

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

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

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

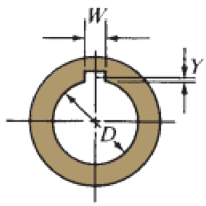
$$L_{\text{min}} = \frac{4TN}{DHs_y}$$

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

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

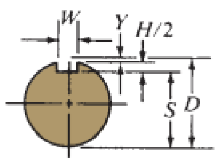
30. Round the key length up to the nearest standard size

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



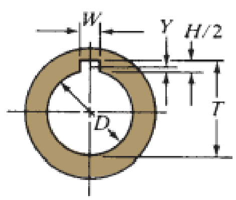
(a) Chordal height

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



(b) Depth of shaft keyseat

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



(c) Depth of hub keyseat

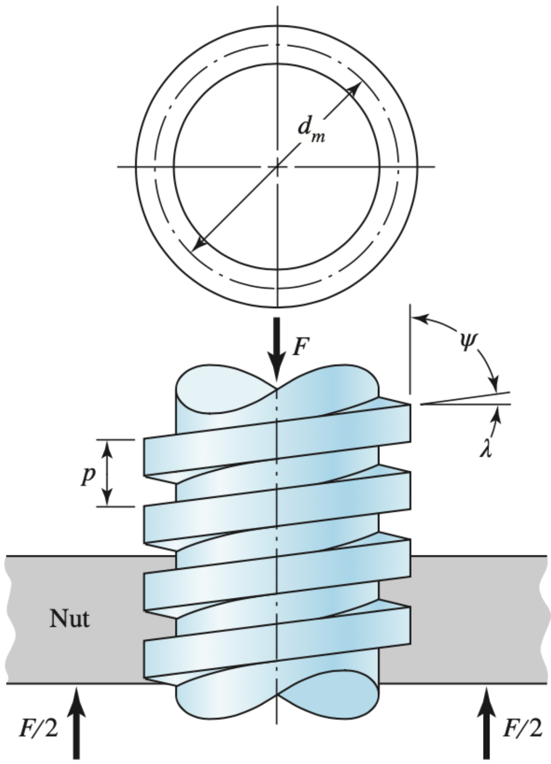
$$T = D - Y + \frac{H}{2} + C = \frac{D + H + \sqrt{D^2 - W^2}}{2} + C$$

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

2 Springies, Bolties, Screwies (Boing Boing)

2.1 Power Screws

2.1.1 Anatomy



2.1.2 Design Selection

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

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

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

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

TABLE 17–1 Preferred Acme Screw Threads

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

[a]Per inch of length of engagement.

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

ISO thread system—External threads

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

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

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

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

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

τ_d is the maximum shear stress in psi

(b) the required length h is:

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

where A_{sc} is the calculate shear area,
A_{sr} is the rated shear area obtained from the table

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

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

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

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

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

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

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

where λ is the lead angle,
f is the friction coefficient,
F is the load applied to the screw in lbs,
D_p is the pitch diameter (obtained from the table) in inches,
ϕ is the thread angle

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

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

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

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

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

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

where f_c is the collar friction,
R_C is the collar friction radius

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

6. Compute the efficiency e of the power screw:

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

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

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

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

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

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

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

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

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

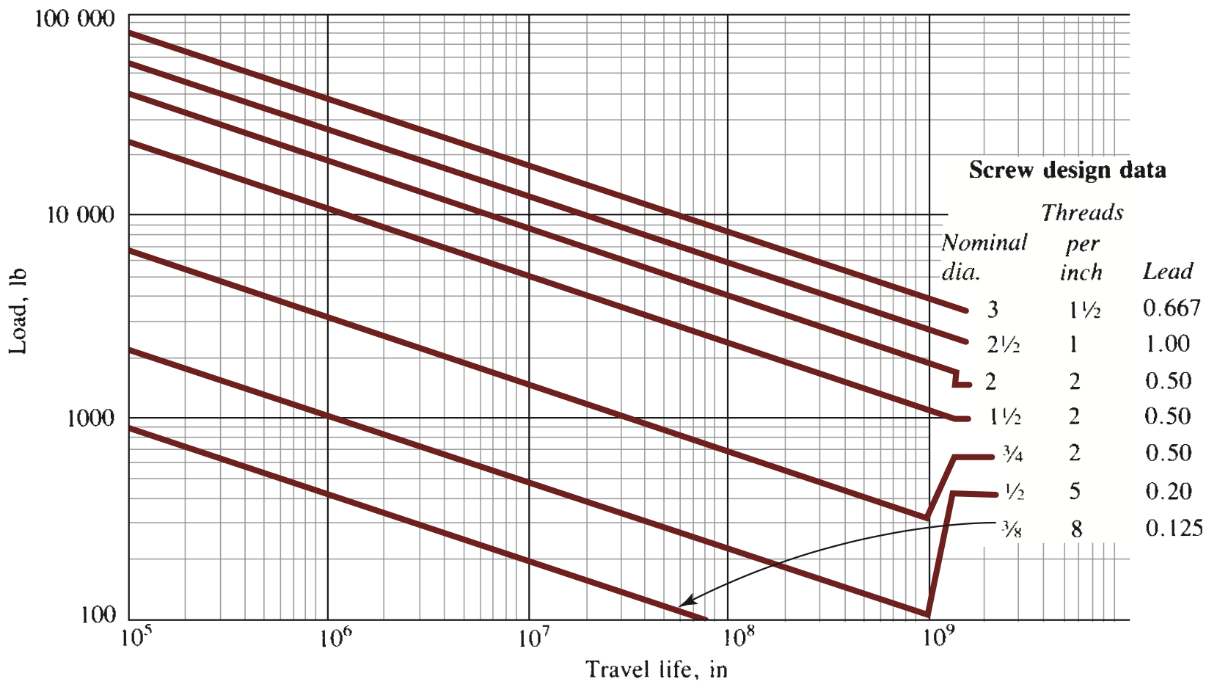
2.2 Ball Screws

2.2.1 Design Selection

1. Compute the travel life in inches:

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

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



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

$$T_u = 0.177FL$$

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

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

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

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

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

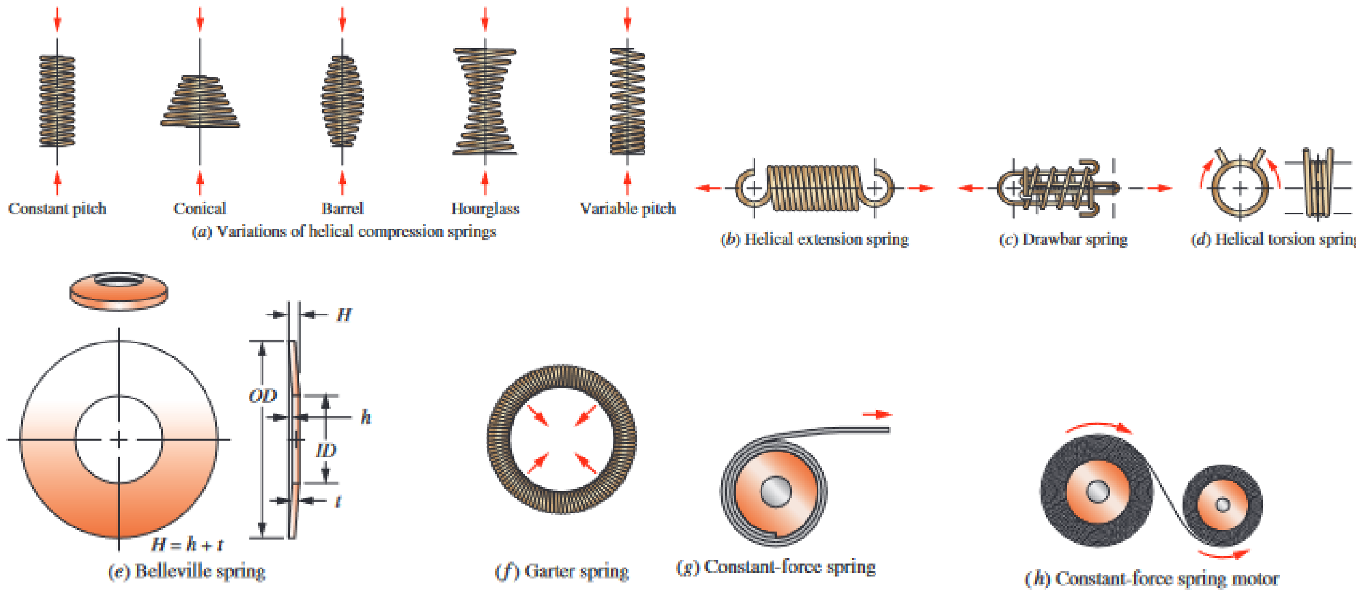
   (b) The power required (in hp) is the following:

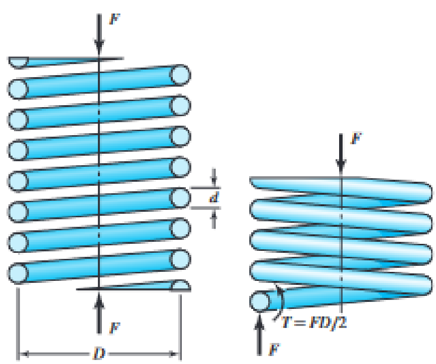
   $$P = \frac{Tn}{63000}$$

   where T is the required torque in lb · in,
   n is the rotational speed in rpm

2.3 Springs

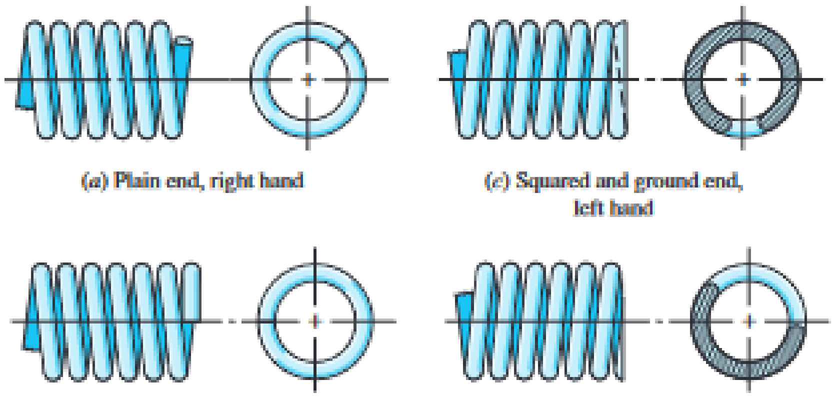
2.3.1 Anatomy





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

Oh shit we got these using Castigliano's, wish I paid more attention in 360
N = Number of coils
N_a = Number of active coils
Ignore this one apparently: $U = \frac{T^2 l}{2GJ} + \frac{F^2 l}{2AG}$
This one for sure though (also $N = N_a$ idk why): $U = \frac{4F^2 D^3 N}{d^4 G} + \frac{2F^2 DN}{d^2 G}$
Total deflection: $= \frac{\partial U}{\partial F} = \frac{8FD^3 N}{d^4 G} + \frac{4FDN}{d^2 G} = \frac{8FD^3 N}{d^4 G}(1 + \frac{1}{2C^2}) \approx \frac{8FD^3 N}{d^4 G}$
Spring constant: $k = \frac{F}{y} \approx \frac{d^4 G}{8D^3 N}$



(*a*) Plain end, right hand

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

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

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

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

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

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

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

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

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

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

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

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

Table 10–3 High-Carbon and Alloy Spring Steels

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

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

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

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

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

Table 10–5 Mechanical Properties of Some Spring Wires

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

*Also includes 302, 304, and 316.

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

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

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

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

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

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

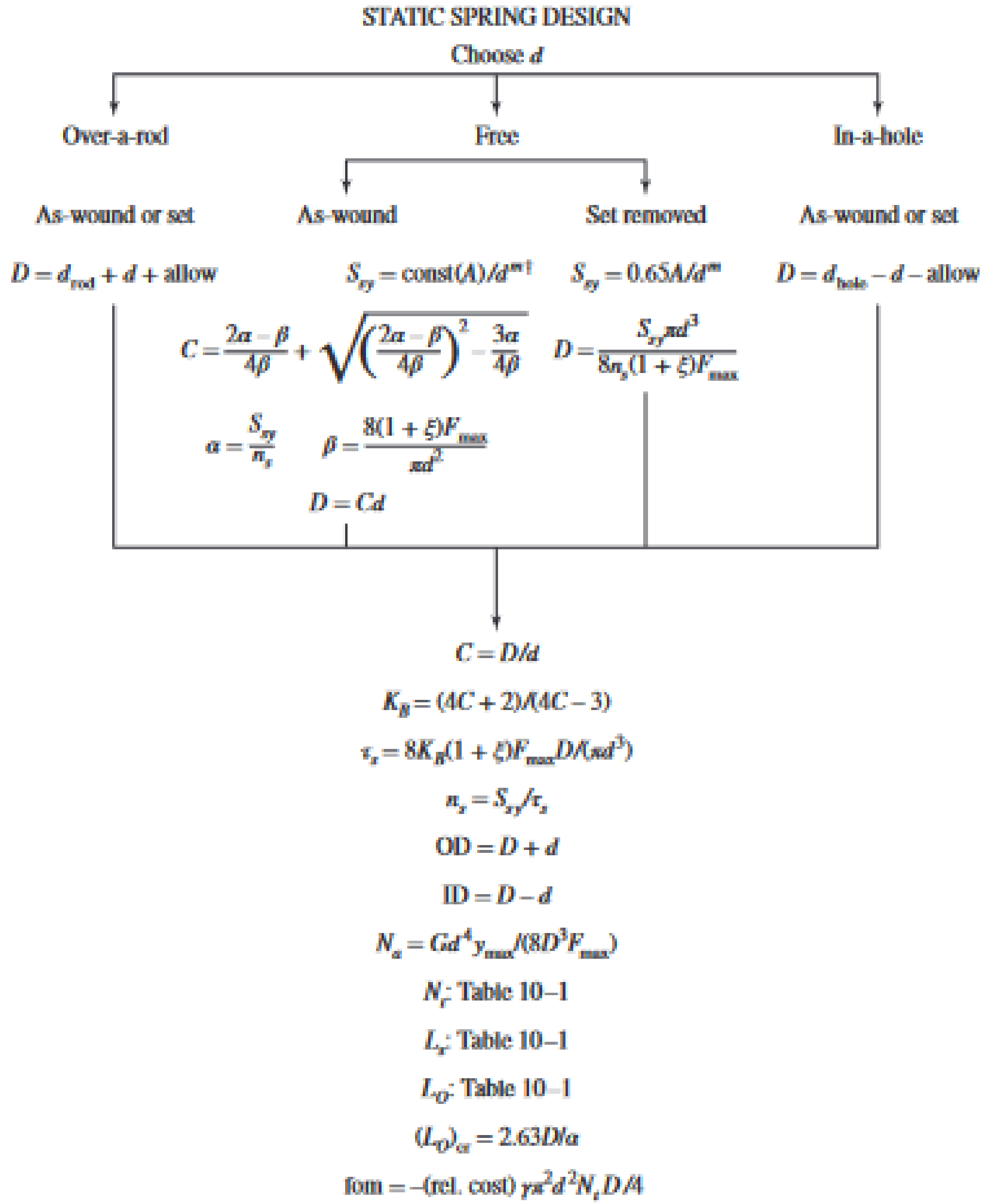
Name of Gauge:	American or Brown & Sharpe	Birmingham or Stubs Iron Wire	United States Standard[†]	Manufacturers Standard	Steel Wire or Washburn & Moen	Music Wire	Stubs Steel Wire	Twist Drill
Principal Use:	**Nonferrous Sheet, Wire, and Rod**	**Tubing, Ferrous Strip, Flat Wire, and Spring Steel**	**Ferrous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

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

[†]Reflects present average and weights of sheet steel.

2.3.2 Design for Static Service

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



Print or display: d, D, C, OD, ID, N_a, N_t, L_s, L_O, $(L_O)_{cr}$, n_s, fom

Build a table, conduct design assessment by inspection

Eliminate infeasible designs by showing active constraints

Choose among satisfactory designs using the figure of merit

†const is found from Table 10–6.

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

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

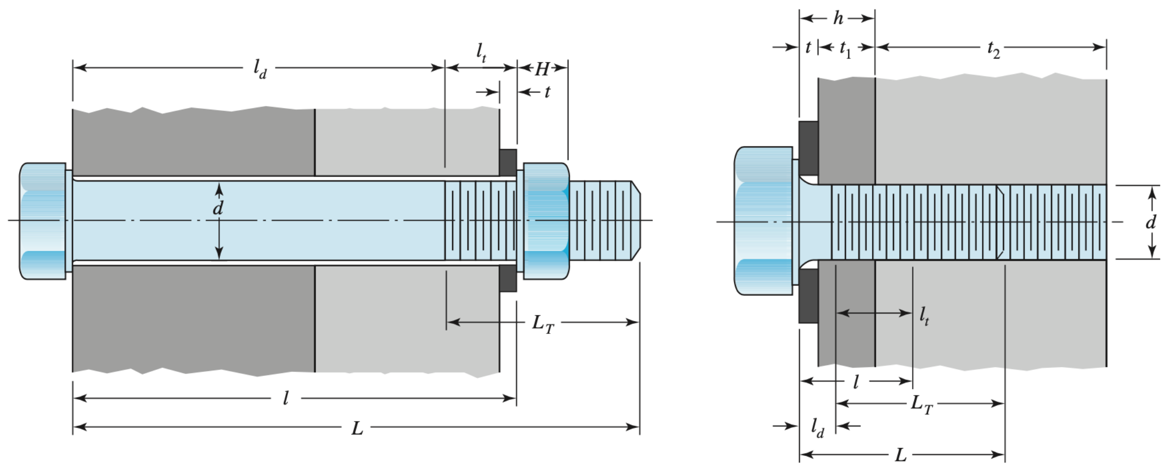
$\therefore\ C = \frac{2\alpha-\beta}{4\beta} + \sqrt{(\frac{2\alpha-\beta}{4\beta})^2 - \frac{3\alpha}{4\beta}}$

2.4 Fasteners/Bolts

2.4.1 Design Selection

1. Determine a suitable length for the bolt:

   (a) Compute the grip length.



(a) (b)

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

$$l = \text{thickness of all material squeezed between face of bolt and face of nut}$$

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

$$l = \begin{cases} h + t_2/2, & t_2 < d \\ h + d/2 & t_2 \geq d \end{cases}$$

   (b) Compute the fastener/bolt length (round up to nearest 1/4th inch):

- For figure (a):

$$L > l + H$$

- for figure (b):

$$L > h + 1.5d$$

2. Compute the bolt stiffness k_b:

   (a) Determine the threaded length L_T:

- If your units are in inches:

$$L_T = \begin{cases} 2d + \frac{1}{4} \text{ in} & L \leq 6 \text{ in} \\ 2d + \frac{1}{2} \text{ in} & L > 6 \text{ in} \end{cases}$$

- If your units are in mm:

$$L_T = \begin{cases} 2d + 6 \text{ mm} & L \leq 125 \text{ mm}, d \leq 48 \text{ mm} \\ 2d + 12 \text{ mm} & 125 < L \leq 200 \text{ mm} \\ 2d + 25 \text{ mm} & L > 200 \text{ mm} \end{cases}$$

(b) Compute the length of the unthreaded portion in grip l_d:

$$l_d = L - L_T$$

(c) Compute the length of the threaded portion in grip l_t:

$$l_t = l - l_d$$

(d) Compute the area of the unthreaded portion A_d:

$$A_d = \pi d^2/4$$

(e) Find the area of the threaded portion A_t from one of the following tables (it is the tensile-stress area A_t):

Table 8–1 **Diameters and Areas of Coarse-Pitch and Fine-Pitch Metric Threads***

	Coarse-Pitch Series			Fine-Pitch Series		
Nominal Major Diameter d mm	Pitch p mm	Tensile-Stress Area A_t mm^2	Minor-Diameter Area A_r mm^2	Pitch p mm	Tensile-Stress Area A_t mm^2	Minor-Diameter Area A_r mm^2
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470
110				2	9180	9080

Table 8–2 **Diameters and Area of Unified Screw Threads UNC and UNF***

		Coarse Series—UNC			Fine Series—UNF		
Size Designation	Nominal Major Diameter in	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	0.373	0.351
$\frac{7}{8}$	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
$1\frac{1}{4}$	1.2500	7	0.969	0.890	12	1.073	1.024
$1\frac{1}{2}$	1.5000	6	1.405	1.294	12	1.581	1.521

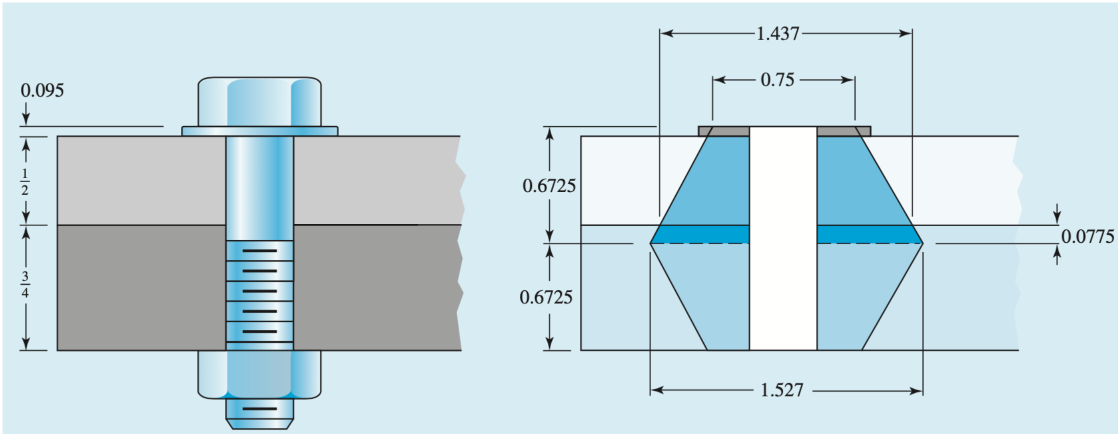
(f) Compute the bolt stiffness k_b (in lbf/in or N/m):

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}$$

where E is Young's Modulus

3. Determine the stiffness of the members:

(a) Make a diagram like the one below. The angle at which the line goes down is assumed to be 30°. The line will start at a diameter of 1.5 times the nominal diameter of the bolt apart. The lines will start on the top washer (or if there is none then the top of the upper plate) and will end either on the bottom of the lower plate or if there is a second washer then into it as well. The lines will continue to separate until they reach the middle of the height of the region that they are allowed in. After that, they will start to converge again.



(b) For each section of the diagram, (i.e. top washer, top plate, bottom plate, bottom washer etc.) calculate the spring rate k using the following expression:

$$k = \frac{0.5774\pi Ed}{\ln \dfrac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

where E is Young's Modulus of the material,
d is the inner diameter of the section member
for plates, it is usually the nominal diameter of the bolt
for washers, it is their inner diameter
D is the shortest distance between two separating lines in the member's region,
t is the thickness of the section

Note that if two members have the same material and the inner diameter can they can be treated as one member where the thickness is the sum of their individual thicknesses.

Also note that if the lines go from separating to converging in the middle of a member's region, then you have to treat the section where they are separating as a different member from the section where they are converging. This is because they will have a different D value.

(c) Once you have calculated all the spring rates for the individual members you can add them with the following equation:

$$\frac{1}{k_t} = \frac{1}{k_1} + \cdots + \frac{1}{k_n}$$

(d) If the top half and bottom half are the same, then you can calculate the spring rate of the top half with:

$$\frac{1}{k_{top}} = \frac{1}{k_1} + \cdots + \frac{1}{k_n}$$

and the total spring rate of the joint will be:

$$k_t = \frac{k_{top}}{2}$$

2.4.2 Factor of Safety and Preload Calculations

1. Determine the torque to reach a given preload

   (a) Calculate P, the load per bolt.

   $$P = \frac{P_{tot}}{N}$$

   Where P_{tot} is the total load in the tension joint,
   N is the number of bolts

   (b) Find the proof strength S_p, the minimum tensile strength S_{ut} and the endurance strength of the bolt S_e from the following tables:

Table 8–9 **SAE Specifications for Steel Bolts**

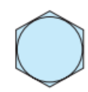
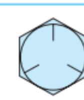
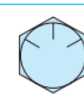
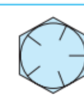
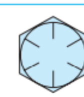
SAE Grade No.	Size Range Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
1	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low or medium carbon	
2	$\frac{1}{4}$–$\frac{3}{4}$ $\frac{7}{8}$–$1\frac{1}{2}$	55 33	74 60	57 36	Low or medium carbon	
4	$\frac{1}{4}$–$1\frac{1}{2}$	65	115	100	Medium carbon, cold-drawn	
5	$\frac{1}{4}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Medium carbon, Q&T	
5.2	$\frac{1}{4}$–1	85	120	92	Low-carbon martensite, Q&T	
7	$\frac{1}{4}$–$1\frac{1}{2}$	105	133	115	Medium-carbon alloy, Q&T	
8	$\frac{1}{4}$–$1\frac{1}{2}$	120	150	130	Medium-carbon alloy, Q&T	
8.2	$\frac{1}{4}$–1	120	150	130	Low-carbon martensite, Q&T	

Table 8–10 ASTM Specifications for Steel Bolts

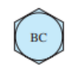
ASTM Desig-nation No.	Size Range, Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
A307	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low carbon	
A325, type 1	$\frac{1}{2}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Medium carbon, Q&T	
A325, type 2	$\frac{1}{2}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Low-carbon, martensite, Q&T	
A325, type 3	$\frac{1}{2}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Weathering steel, Q&T	
A354, grade BC	$\frac{1}{4}$–$2\frac{1}{2}$ $2\frac{3}{4}$–4	105 95	125 115	109 99	Alloy steel, Q&T	
A354, grade BD	$\frac{1}{4}$–4	120	150	130	Alloy steel, Q&T	
A449	$\frac{1}{4}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$ $1\frac{3}{4}$–3	85 74 55	120 105 90	92 81 58	Medium-carbon, Q&T	
A490, type 1	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Alloy steel, Q&T	
A490, type 3	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Weathering steel, Q&T	

Table 8–11 Metric Mechanical-Property Classes for Steel Bolts, Screws, and Studs

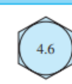
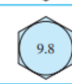
Property Class	Size Range, Inclusive	Minimum Proof Strength,* MPa	Minimum Tensile Strength,* MPa	Minimum Yield Strength,* MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	
4.8	M1.6–M16	310	420	340	Low or medium carbon	
5.8	M5–M24	380	520	420	Low or medium carbon	
8.8	M16–M36	600	830	660	Medium carbon, Q&T	
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	

Table 8–17 **Fully Corrected Endurance Strengths for Bolts and Screws with Rolled Threads***

Grade or Class	Size Range	Endurance Strength
SAE 5	$\frac{1}{4}$–1 in	18.6 kpsi
	$1\frac{1}{8}$–$1\frac{1}{2}$ in	16.3 kpsi
SAE 7	$\frac{1}{4}$–$1\frac{1}{2}$ in	20.6 kpsi
SAE 8	$\frac{1}{4}$–$1\frac{1}{2}$ in	23.2 kpsi
ISO 8.8	M16–M36	129 MPa
ISO 9.8	M1.6–M16	140 MPa
ISO 10.9	M5–M36	162 MPa
ISO 12.9	M1.6–M36	190 MPa

(c) Determine the initial bolt tension/preload force:

- If given there you are done.
- If said to be X% of proof strength, then the preload force F_i is

$$F_i = \frac{X}{100} A_t S_p$$

where A_t is the tensile-stress area given in Table 8-2
S_p is the proof strength of the bolt

- If the question asks you to choose then from Shigley's recommendation:

For nonpermanent connections, reused fasteners: $F_i = .75 A_t S_p$
For permanent connections: $F_i = .90 A_t S_p$

(d) Choose a torque factor K from the table

Bolt Condition	K
Nonplated, black finish	0.30
Zinc-plated	0.20
Lubricated	0.18
Cadmium-plated	0.16
With Bowman Anti-Seize	0.12
With Bowman-Grip nuts	0.09

(e) Compute the torque from the given formula:

$$T = K F_i d$$

where T is the K is the torque factor,
F_i is the initial bolt tension,
d is the nominal diameter of the bolt

- Note that according to the homework solutions a torque of around 1000 lb-in is "VERY high" for a wrench

2. Compute overload/load factor of safety

(a) Calculate the stiffness constant of the joint:

$$C = \frac{k_b}{k_b + k_m}$$

Where k_b is the bolt spring rate,
k_m is the member spring rate

Note that the stiffness constant of the joint represents what percent of the load is picked up by the bolt compared to the members.

(b) Calculate the load factor:

$$n_L = \frac{S_P A_t - F_i}{CP}$$

Where S_p is the proof strength of the bolt in Pa or psi,
A_t is the tensile-stress area given in Table 8-1 or 8-2 above in m^2 or in^2,
F_i is the initial bolt tension in N or lb,
C is the stiffness constant,
P is the tension load per bolt in N or lb

Note that to convert from mm^2 to m^2, multiply by 10^6

3. Compute bolt yield/yielding factor of safety:

(a) The tensile stress in the bolt is:

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t}$$

Where A_t is the tensile-stress area given in Table 8-1 or 8-2 above in m^2 or in^2,
F_i is the initial bolt tension in N or lb,
C is the stiffness constant,
P is the tension load per bolt in N or lb

(b) The static yielding factor of safety is:

$$n_p = \frac{S_p}{\sigma_b} = \frac{S_p A_t}{CP + F_i}$$

4. Compute the joint separation factor of safety:

$$n_0 = \frac{F_i}{P(1 - C)}$$

5. Compute the Goodman criteria fatigue factor of safety:

(a) Compute the alternating stress:

$$\sigma_a = \frac{C(P_{max} - P_{min})}{2A_t}$$

(b) Compute the midrange stress:

$$\sigma_m = \frac{C(P_{max} + P_{min})}{2A_t} + \frac{F_i}{A_t}$$

(c) Compute the initial stress:

$$\sigma_i = \frac{F_i}{A_t}$$

(d) Compute the fatigue factor of safety:

$$n_f = \frac{S_e(S_{ut} - \sigma_i)}{S_{ut}\sigma_a + S_e(\sigma_m - \sigma_i)}$$

Where S_e is the endurance strength,
S_{ut} is the tensile strength,
σ_i is the initial stress,
σ_m is the midrange stress,
σ_a is the alternating stress

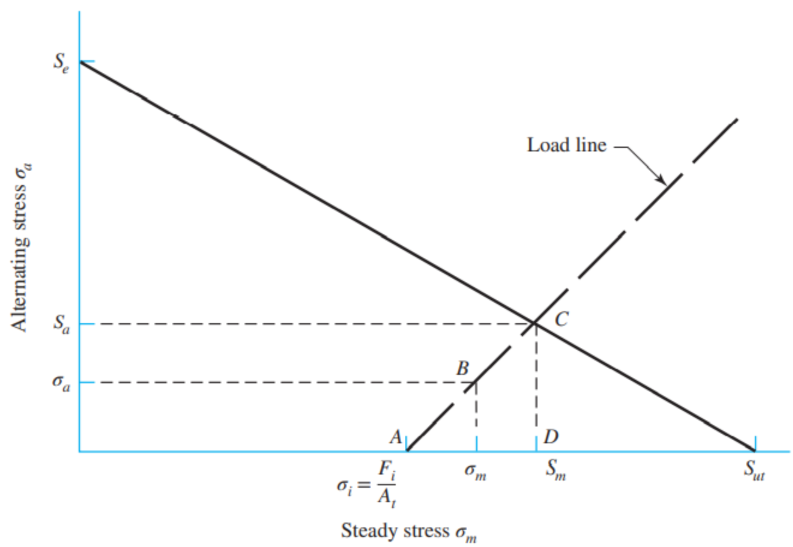
(e) If the external load is a repeated load i.e. (from 0 to P) then:

$$\sigma_a = \frac{CP}{2A_t}$$

$$\sigma_m = \frac{CP}{2A_t} + \frac{F_i}{A_t} = \sigma_a + \sigma_i$$

$$n_f = \frac{S_e(S_{ut} - \sigma_i)}{\sigma_a(S_{ut} + S_e)}$$

(f) If fatigue failure diagram is needed, it is shown below:



- The load line is given by the equation:

$$S_a = \frac{\sigma_a}{\sigma_m - \sigma_i}(S_m - \sigma_i)$$

- Then the Goodman line a.k.a. the line going from S_e to S_{ut} is given by the equation:

$$S_a = S_e - \frac{S_e}{S_{ut}} S_m$$

- Note that even though the graph has axes σ_a v.s. σ_m, it is essentially a S_a v.s. S_m graph. So just plot S_a as a function of S_m.

6. If any of the factor safeties are low or below 1, then the grade, size, or quantity of bolts can be increased to increase the safety factors.