Note Title

Life Factor, K_L

All AGMA bending-fatigue strength data are for 10^7 cycles of repeated stress (rather than 10^6 or $5\text{x}10^8$ cycles). Hence, we should make a correction if the life is different from 10^7 cycles. Figure 12-24 shows S-N curves for the bending-fatigue strength of steels having several different tensile strengths as defined by their Brinell hardness numbers. Curve-fitted equations are also shown for each S-N line.

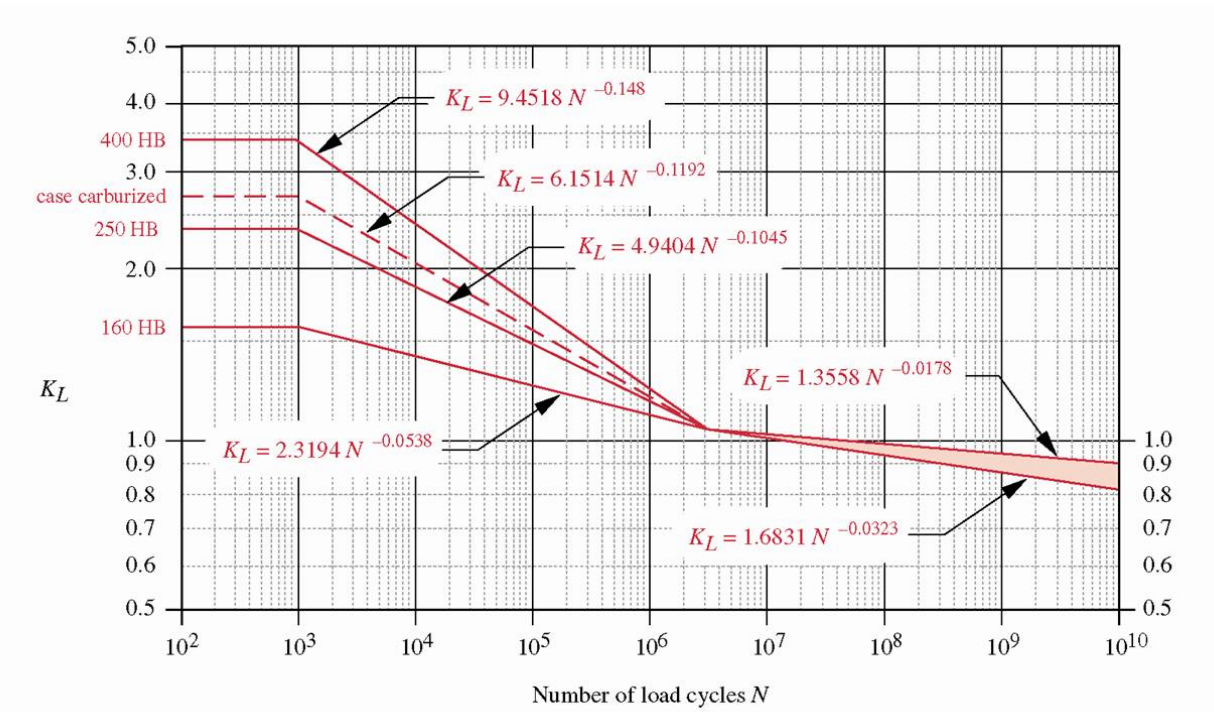


Figure 11-24

AGMA Bending Strength Life Factor K_L.

Temperature Factor, K_T

The lubricant temperature is a good measure of the temperature of gear. Designating the oil temperature (in °F) by T_F, we can find K_T for steel from the empirical relations:

$$K_T = 1 \quad \text{for} \quad T_F \le 250^\circ F$$

$$K_T = \frac{460 + T_F}{620} \quad \text{for} \quad T_F > 250^\circ F \qquad (12.24a)$$

Reliability Factor, K_R

The AGMA strength data are based on 99% reliability (i.e., on a probability of one failure in 100 samples). Hence, the reliability factor K_R is defined to be 1 for 99% reliability. For other than 99% reliability, Table 12-19 can be used to find K_R.

Bending Fatigue Strength Data are given in Table 12-20 for commonly used gear materials. Fig. 12-25 shows ranges of bending fatigue strength for steels as a function of HB.

Reliability %	K_R
90	0.85
99	1.00
99.9	1.25
99.99	1.50

Table 11-19
AGMA Factor K_R.

Material	AGMA Class	Material Designation	Heat Treatment	Minimum Surface Hardness	Bending-Fatigue Strength psi x 10^3	Bending-Fatigue Strength MPa
Steel	A1–A5		Through hardened	≤ 180 HB	25–33	170–230
			Through hardened	240 HB	31–41	210–280
			Through hardened	300 HB	36–47	250–325
			Through hardened	360 HB	40–52	280–360
			Through hardened	400 HB	42–56	290–390
			Flame or induction hardened	Type A pattern 50–54 HRC	45–55	310–380
			Flame or induction hardened	Type B pattern	22	150
			Carburized and case hardened	55–64 HRC	55–75	380–520
		AISI 4140	Nitrided	84.6 HR15N†	34–45	230–310
		AISI 4340	Nitrided	83.5 HR15N	36–47	250–325
		Nitralloy 135M	Nitrided	90.0 HR15N	38–48	260–330
		Nitralloy N	Nitrided	90.0 HR15N	40–50	280–345
		2.5% Chrome	Nitrided	87.5–90.0 15N	55–65	380–450
Cast Iron	20	Class 20	As cast		5	35
	30	Class 30	As cast	175 HB	8	69
	40	Class 40	As cast	200 HB	13	90
Nodular (ductile) Iron	A-7-a	60-40-18	Annealed	140 HB	22–33	150–230
	A-7-c	80-55-06	Quenched and tempered	180 HB	22–33	150–230
	A-7-d	100-70-03	Quenched and tempered	230 HB	27–40	180–280
	A-7-e	120-90-02	Quenched and tempered	230 HB	27–40	180–280
Malleable Iron (pearlitic)	A-8-c	45007		165 HB	10	70
	A-8-e	50005		180 HB	13	90
	A-8-f	53007		195 HB	16	110
	A-8-i	80002		240 HB	21	145
Bronze	Bronze 2	AGMA 2C	Sand cast	40 ksi min tensile strength	5.7	40
	Al/Br 3	ASTM B-148 78 alloy 954	Heat treated	90 ksi min tensile strength	23.6	160

† Rockwell 15N scale used for case-hardened materials—see Section 2-4

12
Table ~~1~~1-20
AGMA Bending-Fatigue Strengths S_{fb}' for a Selection of Gear Materials*.

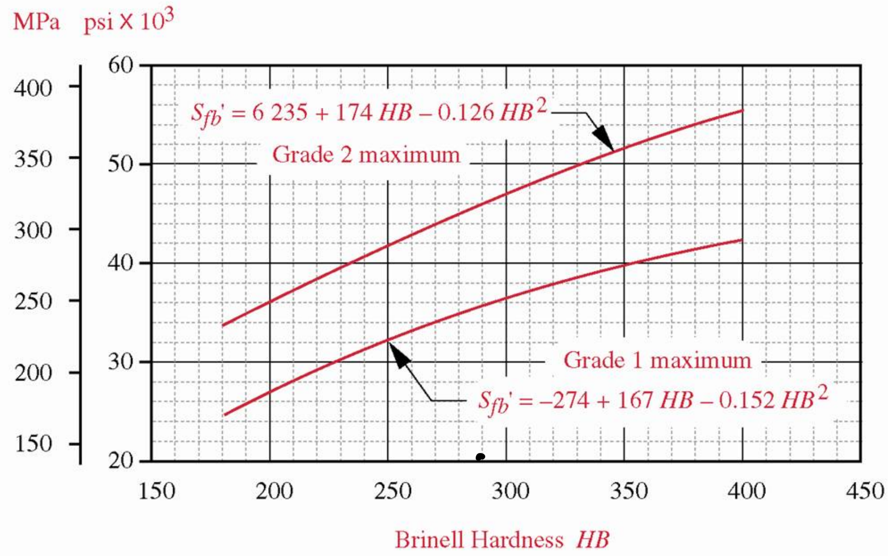


Figure 11-25

AGMA Bending-Fatigue Strengths S_{fb}' for Steels*.

AGMA Surface-Fatigue Strength for Gear Materials

The uncorrected surface-fatigue strength is designated by $S_{fc'}$ and is given in Table 12-21 and Fig. 12-27. There are four correction factors that need to be applied to $S_{fc'}$ in

Material	AGMA Class	Material Designation	Heat Treatment	Minimum Surface Hardness	Surface-Fatigue Strength psi x 10^3	Surface-Fatigue Strength MPa
Steel	A1–A5		Through hardened	≤ 180 HB	85–95	590–660
			Through hardened	240 HB	105–115	720–790
			Through hardened	300 HB	120–135	830–930
			Through hardened	360 HB	145–160	1000–1100
			Through hardened	400 HB	155–170	1100–1200
			Flame or induction hardened	50 HRC	170–190	1200–1300
			Flame or induction hardened	54 HRC	175–195	1200–1300
			Carburized and case hardened	55–64 HRC	180–225	1250–1300
		AISI 4140	Nitrided	84.6 HR15N†	155–180	1100–1250
		AISI 4340	Nitrided	83.5 HR15N	150–175	1050–1200
		Nitralloy 135M	Nitrided	90.0 HR15N	170–195	1170–1350
		Nitralloy N	Nitrided	90.0 HR15N	195–205	1340–1410
		2.5% Chrome	Nitrided	87.5 HR15N	155–172	1100–1200
		2.5% Chrome	Nitrided	90.0 HR15N	192–216	1300–1500
Cast iron	20	Class 20	As cast		50–60	340–410
	30	Class 30	As cast	175 HB	65–70	450–520
	40	Class 40	As cast	200 HB	75–85	520–590
Nodular (ductile) iron	A-7-a	60-40-18	Annealed	140 HB	77–92	530–630
	A-7-c	80-55-06	Quenched and tempered	180 HB	77–92	530–630
	A-7-d	100-70-03	Quenched and tempered	230 HB	92–112	630–770
	A-7-e	120-90-02	Quenched and tempered	230 HB	103–126	710–870
Malleable iron (pearlitic)	A-8-c	45007		165 HB	72	500
	A-8-e	50005		180 HB	78	540
	A-8-f	53007		195 HB	83	570
	A-8-i	80002		240 HB	94	650
Bronze	Bronze 2	AGMA 2C	Sand cast	40 ksi min tensile strength	30	450
	Al/Br 3	ASTM B-148 78 alloy 954	Heat-treated	90 ksi min tensile strength	65	450

† Rockwell 15N scale used for case-hardened materials—see Section 2-4

Table 11-21

AGMA Surface-Fatigue Strengths S_{fc}' for a Selection of Gear Materials*.

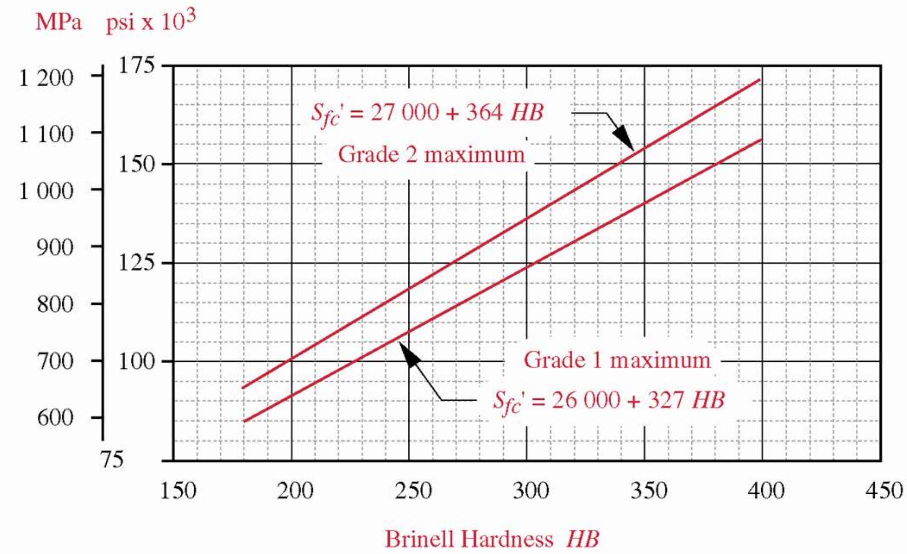


Figure 11-27
AGMA Surface-Fatigue Strengths S_{fc}' for Steels*.

order to find the corrected surface-fatigue strength S_{fc} for the application of interest:

$$S_{fc} = \frac{C_L C_H}{C_T C_R} S_{fc}' \qquad (12.25)$$

where $C_T = K_T$ **is given by Eq. 12-24a**
$C_R = K_R$ **is given in Table 12-19**

and where C_L is the surface-life factor and C_H is the hardness ratio factor. They are discussed below.

Surface-Life Factor, C_L

Since published surface-fatigue data are for a life of 10^7 cycles, C_L has been defined to be "1" for a life of 10^7 cycles. Fig. 12-26 is used to modify C_L for a life other than 10^7 cycles. Note that this figure is for steel and that the upper portion of the shaded zone is for commercial and the lower portion is for critical service applications.

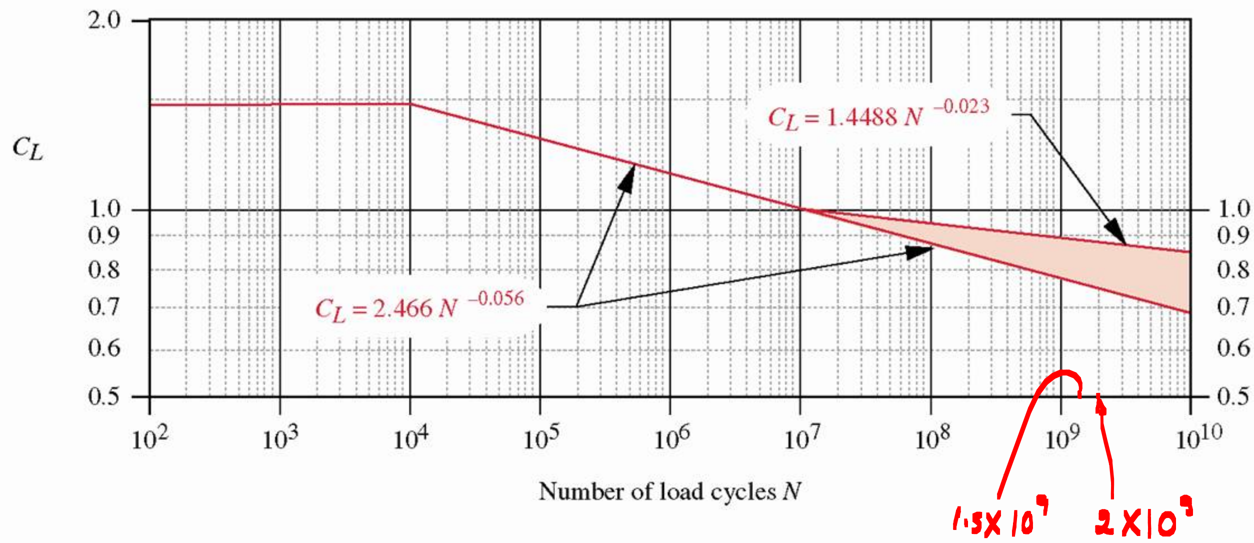


Figure 11-26*

AGMA Surface-Fatigue Strength Life Factor C_L.

Hardness Ratio Factor, C_H

This factor is a function of relative hardness of pinion and gear as well as the gear ratio. It is defined so that $C_H > 1$. Hence, it acts to increase the strength of the gear. This happens when the pinion teeth are harder than the gear teeth and thus act to work-harden the gear-tooth surfaces. C_H is only applied to gear-tooth strength, not to the pinion. For through-hardened pinions in mesh with through-hardened gears:

$$C_H = 1 + A\,(m_G - 1) \qquad (12.26a)$$

where m_G is the gear ratio and A is given by

$$\text{if } \frac{HB_p}{HB_g} < 1.2, \quad A = 0 \qquad (12.26b\&c)$$

$$\text{if } 1.2 \leq \frac{HB_p}{HB_g} \leq 1.7 \text{ then } A = 0.00898\,\frac{HB_p}{HB_g} - 0.00829$$

$$\text{if } \frac{HB_p}{HB_g} > 1.7 \text{ then } A = 0.00698$$

For surface-hardened pinion in mesh with through-hardened gears, C_H is:

$$C_H = 1 + B\,(450 - HB_g) \qquad (12.27)$$

where

$$B = 0.00075\,e^{-0.0112 R_q} \qquad (12.28)\ US$$

$$B = 0.00075\,e^{-0.052 R_q} \qquad (12.28)\ SI$$

where R_q is the root mean square (rms) surface roughness value of the pinion teeth in μ in rms (see Section 7.1, Fig. 7-2 on p. 422).

Example 5: Material Selection and Safety Factor for Spur Gears

A spur pinion transmits 15 hp at 1200 rpm. It has a pitch of 6 teeth per in, 22 full-depth teeth, and a 20º pressure angle. The gear has 60 teeth. For a face width of 2 in select suitable materials and calculate the safety factors for both bending and surface stresses in the gearset. Assume a service life of 10 years of one-shift operations.

Solution:

Selecting a trial material. In Example 4, assuming that both gears were made of steel, we calculated the applied tooth-contact stress to be

$$\sigma_c = 77.8 \text{ kpsi}$$

The gear-tooth bending stress was obtained in Ex. 3:

$$\sigma_{b_p} = 7555 \text{ psi} \qquad \sigma_{b_g} = 6611 \text{ psi}$$

We use these values to choose a trial material from Figs. 12-25 and 12-27 by estimating the surface-fatigue strengths $S_{fc'}$ and bending-fatigue strengths $S_{fb'}$.

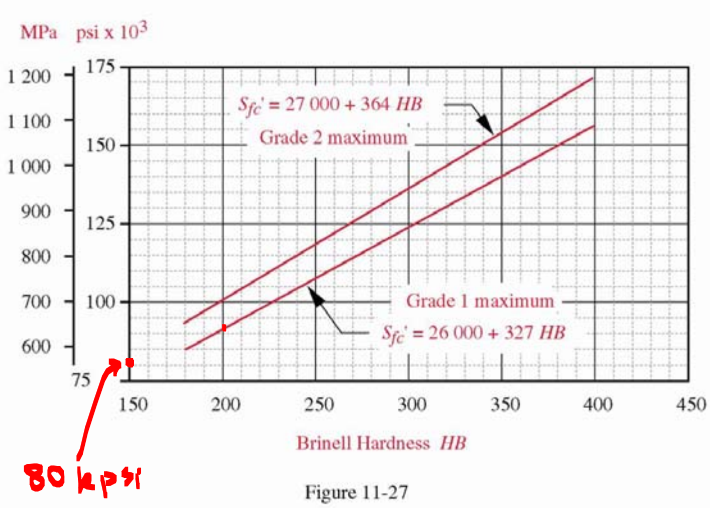


Figure 11-27
AGMA Surface-Fatigue Strengths S_{fc}' for Steels*.

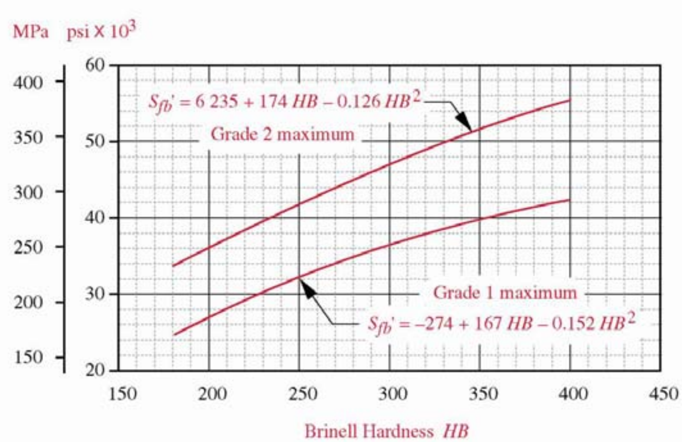


Figure 11-25
AGMA Bending-Fatigue Strengths S_{fb}' for Steels*.

We try an AGMA Grade 1 steel, through hardened to 200 HB. Then

$$S_{fb'} = -274 + 167\,HB - 0.152\,HB^2$$
$$=$$

$$S_{fc'} = 26000 + 327\ HB$$

$$=$$

Corrected bending-fatigue strength

$$S_{fb} = \frac{K_L}{K_T K_R} S_{fb'}$$

The life factor K_L is calculated from appropriate equations in Fig. 12-24.

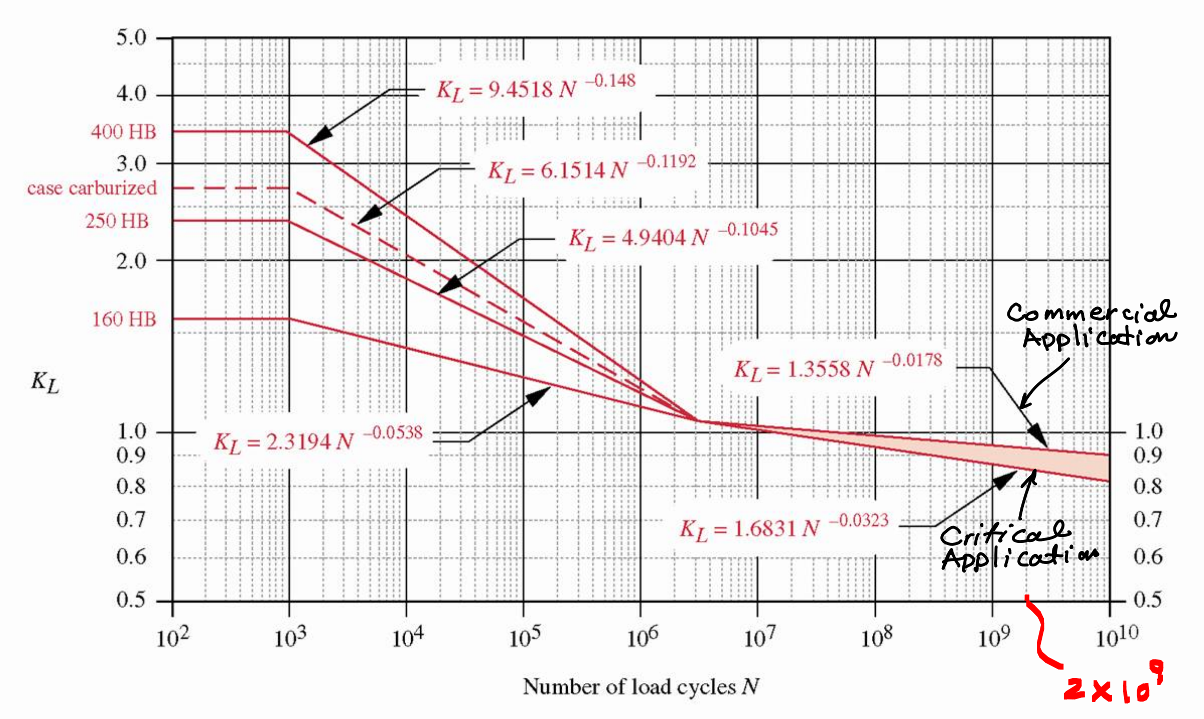


Figure 11-24

AGMA Bending Strength Life Factor K_L.

Since

$$N = 1200\ \text{rpm}\left(\frac{60\ \text{min}}{\text{hr}}\right)\left(\frac{2080\ \text{hr}}{\text{shift year}}\right)(10\ y)(1\ \text{shift}) = 1.5\times10^9\ \text{Cycles}$$

For commercial application: $K_L = 1.3558\,N^{-0.0178} = 0.9308$

Temperature Factor, $K_T = 1$

Reliability Factor, $K_R = 1$ for 99% reliability

$$\text{So } S_{fb} = \frac{\quad}{(\ )(\ )}(\qquad) = \qquad \triangleleft$$

Corrected surface-fatigue strength

$$S_{fc} = \frac{C_L C_H}{C_T C_R} S_{fc}' \qquad (12.25)$$

The life factor C_L

C_L is calculated from an appropriate equation in Fig. 12-26 based on $N = 1.5 \times 10^9$ cycles.

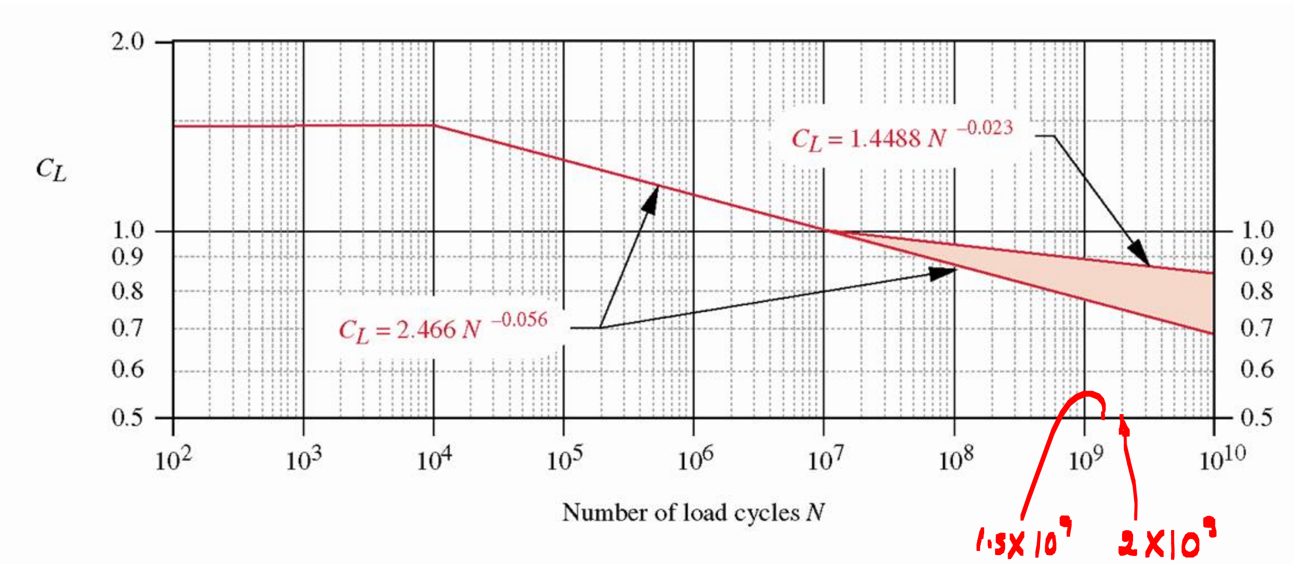


Figure 11-26*

AGMA Surface-Fatigue Strength Life Factor C_L.

Assuming commercial application:

$$C_L = 1.4488\,N^{-0.023} = (1.4488)(\qquad)^{-0.023}$$

$$=$$

Hardness ratio factor $C_H = 1$ (since gear and pinion are of the same hardness)

Temperature and reliability factors, $C_T = C_R = 1$.

$$\therefore\ S_{fc} = \frac{(\qquad)(\ )}{(\ )(\ )}(\qquad) = \qquad \triangleleft$$

Factor of Safety against tooth-bending failure

$$N_{b_{pinion}} = \frac{S_{fb}}{\sigma_{b_{pinion}}} = \underline{\hspace{2cm}} =$$

$$N_{b_{gear}} = \frac{S_{fb}}{\sigma_{b_{gear}}} = \underline{\hspace{2cm}} = 3.80$$

These are both acceptable.

Factor of Safety against tooth-surface failure

Since surface stress σ_c is related to the square root of the load, the surface fatigue safety factor is given by

$$N_{c_{pinion-gear}} = \left(\frac{S_{fc}}{\sigma_{c_{pinion}}}\right)^2 = \left(\underline{\hspace{2cm}}\right)^2 =$$

Hence, if gears are made of Grade 1, through-hardened steel, they will be safe against bending and surface fatigue failure with a reliabity of 99% we can say that they will last 10 years before pitting of the pinion begins.

Chapter 15 Screws and Fasteners

The success or failure of a design can hinge on proper selection and use of its fasteners. They have tremendous safety and economic implications because they are used in vehicles carrying people and in other machines. The frame of a large jet aircraft has millions of fasteners costing millions of dollars. Our discussion here is limited to the design and selection of conventional fasteners such as bolts, screws, nuts, etc. used in machines.

Screws are used both to hold things together as fasteners and to move loads as the so called power screws or lead screws. We will study both of these applications.

15.1 Standard Thread Forms

Thread forms are standardized in the US, Canada, and the UK. This standard is known as the Unified National Standard (UNS) series, as shown. ISO has also defined a metric series standard that has the same thread form but it is not interchangeable with UNS threads.

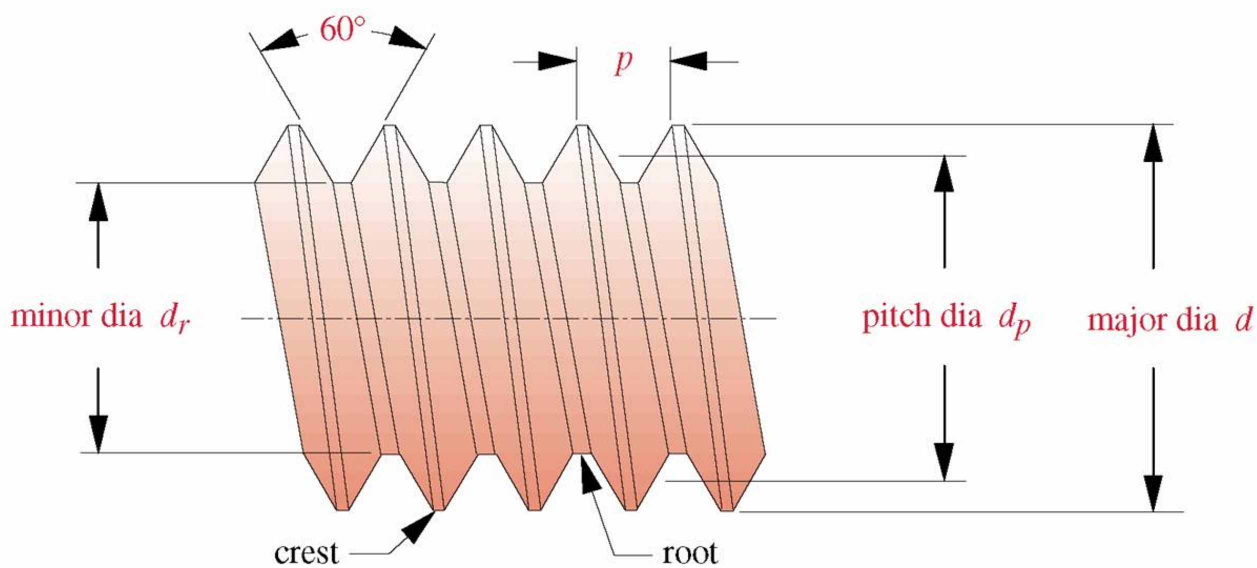


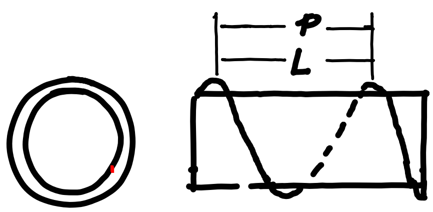
Figure [illegible]-2
Unified National and ISO Standard Thread Form.

Pitch, p is the distance from a point on one thread to the corresponding point on the next adjacent thread, measured parallel to the axis. $p = 1/N$ where N is the number of threads per inch.

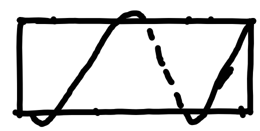
Major (nominal outside) diameter, d is the outside diameter of an imaginary cylinder that touches the screw only on the thread crest.

Minor (root) diameter, d_r is the diameter of an imaginary cylinder that cuts through the thread roots.

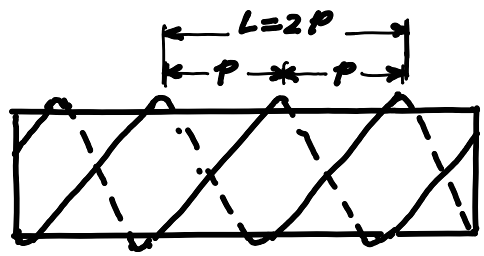
Lead, L (not shown) is the distance the nut moves parallel to the screw axis when the nut is given one turn. For a single thread, the lead is the same as the pitch, L= P, for a multiple thread screw, L = np.



Single thread – Right Hand



Single thread – Left Hand



Double thread – Left Hand

Right-Hand and Left-Hand Threads: If rotating the screw in the direction of the fingers of the right hand causes it to advance in the direction of the thumb it is said to have a right-hand thread; otherwise it is said to have a left-hand thread.

Pitch diameter, d_p is the diameter of an imaginary cylinder that cuts through the threads in such a way that the width of the threads and the grooves are equal on this cylinder.

A thread is specified with a code that defines its series, diameter, pitch, and class of fit (tolerance). For example,

$\frac{1}{4}$ - 20 UNC - 2A - LH

Nominal Outside Diameter or Major Diameter (Size)

Thread per inch

Unified Nat'l Coarse

Tolerance class (A is for external thread)

Left Hand

Other standard series of thread pitch families are: fine-pitch (UNA) and extra-fine-pitch (UNEF).

There are eight classes of thread which differ by the amount of allowance and/or tolerance. Classes IA, 2A, and 3A apply to external threads.

1. Classes 1A & 1B have liberal allowance for ease of assembly even when threads are dirty or slightly damaged

2. Classes 2A & 2B permit external threads to be plated. Used for commercial fasteners.

3. Classes 3A & 3B have small tolerances and are used when no allowance is required.

4. Classes 2 & 3 are not standardized and are used only with American products.

ISO threads are designated by the letter "M".

M 10 x 1.25

SI Thread designation

Major diameter (mm)

Pitch in mm per thread

There are six classes of fit for metric threads:

8g, 7H correspond to 1A &1B

6g, 6H correspond to 2A & 2B

5g, 5H correspond to 3A & 3B

Bolt classes are designated by "g" and nuts by "H".

Basic dimensions of the UNS threads are given in Table 15-1 and those of ISO thread are given in Table 15-2. In these tables the "Tensile Stress Area", A_t is defined by

$$A_t = \frac{\pi}{4}\left(\frac{d_p + d_r}{2}\right)^2 \qquad (15.1a)$$

where for UNS threads:

$$d_p = d - \frac{0.649519}{N} \quad , \quad d_r = d - \frac{1.299038}{N} \qquad (15.1b)$$

and for ISO threads:

$$d_p = d - 0.649519\,p \quad , \quad d_r = d - 1.299038\,p \qquad (15.1c)$$

where d is the outside diameter, N is the number of threads per inch, and p is the pitch in mm.

Testing of thread rods in tension shows that

$$\sigma_t = \frac{F}{A_t} \qquad (15.2)$$

Gage Number

		Coarse Threads—UNC			Fine Threads—UNF		
Size	Major Diameter d (in)	Threads per inch	Minor Diameter d_r (in)	Tensile Stress Area A_t (in^2)	Threads per inch	Minor Diameter d_r (in)	Tensile Stress Area A_t (in^2)
0	0.0600	–	–	–	80	0.0438	0.0018
1	0.0730	64	0.0527	0.0026	72	0.0550	0.0028
2	0.0860	56	0.0628	0.0037	64	0.0657	0.0039
3	0.0990	48	0.0719	0.0049	56	0.0758	0.0052
4	0.1120	40	0.0795	0.0060	48	0.0849	0.0066
5	0.1250	40	0.0925	0.0080	44	0.0955	0.0083
6	0.1380	32	0.0974	0.0091	40	0.1055	0.0101
8	0.1640	32	0.1234	0.0140	36	0.1279	0.0147
10	0.1900	24	0.1359	0.0175	32	0.1494	0.0200
12	0.2160	24	0.1619	0.0242	28	0.1696	0.0258
1/4	0.2500	20	0.1850	0.0318	28	0.2036	0.0364
5/16	0.3125	18	0.2403	0.0524	24	0.2584	0.0581
3/8	0.3750	16	0.2938	0.0775	24	0.3209	0.0878
7/16	0.4375	14	0.3447	0.1063	20	0.3725	0.1187
1/2	0.5000	13	0.4001	0.1419	20	0.4350	0.1600
9/16	0.5625	12	0.4542	0.1819	18	0.4903	0.2030
5/8	0.6250	11	0.5069	0.2260	18	0.5528	0.2560
3/4	0.7500	10	0.6201	0.3345	16	0.6688	0.3730
7/8	0.8750	9	0.7307	0.4617	14	0.7822	0.5095
1	1.0000	8	0.8376	0.6057	12	0.8917	0.6630
1 1/8	1.1250	7	0.9394	0.7633	12	1.0167	0.8557
1 1/4	1.2500	7	1.0644	0.9691	12	1.1417	1.0729
1 3/8	1.3750	6	1.1585	1.1549	12	1.2667	1.3147
1 1/2	1.5000	6	1.2835	1.4053	12	1.3917	1.5810
1 3/4	1.7500	5	1.4902	1.8995			
2	2.0000	4.5	1.7113	2.4982			
2 1/4	2.2500	4.5	1.9613	3.2477			
2 1/2	2.5000	4	2.1752	3.9988			
2 3/4	2.7500	4	2.4252	4.9340			
3	3.0000	4	2.6752	5.9674			
3 1/4	3.2500	4	2.9252	7.0989			
3 1/2	3.5000	4	3.1752	8.3286			
3 3/4	3.7500	4	3.4252	9.6565			
4	4.0000	4	3.6752	11.0826			

Table 14-1

Note: For d < 0.25 in, to find diameter use the formula:

13 (Gage Number) +60 = Diameter / 1000

Major Diameter d (mm)	Coarse Threads Pitch p mm	Coarse Threads Minor Diameter d_r (mm)	Coarse Threads Tensile Stress Area A_t (mm^2)	Fine Threads Pitch p mm	Fine Threads Minor Diameter d_r (mm)	Fine Threads Tensile Stress Area A_t (mm^2)
3.0	0.50	2.39	5.03			
3.5	0.60	2.76	6.78			
4.0	0.70	3.14	8.78			
5.0	0.80	4.02	14.18			
6.0	1.00	4.77	20.12			
7.0	1.00	5.77	28.86			
8.0	1.25	6.47	36.61	1.00	6.77	39.17
10.0	1.50	8.16	57.99	1.25	8.47	61.20
12.0	1.75	9.85	84.27	1.25	10.47	92.07
14.0	2.00	11.55	115.44	1.50	12.16	124.55
16.0	2.00	13.55	156.67	1.50	14.16	167.25
18.0	2.50	14.93	192.47	1.50	16.16	216.23
20.0	2.50	16.93	244.79	1.50	18.16	271.50
22.0	2.50	18.93	303.40	1.50	20.16	333.06
24.0	3.00	20.32	352.50	2.00	21.55	384.42
27.0	3.00	23.32	459.41	2.00	24.55	495.74
30.0	3.50	25.71	560.59	2.00	27.55	621.20
33.0	3.50	28.71	693.55	2.00	30.55	760.80
36.0	4.00	31.09	816.72	3.00	32.32	864.94
39.0	4.00	34.09	975.75	3.00	35.32	1028.39

Table 14-2

Principal Dimensions of ISO Metric Standard Screw Threads.

Data Calculated from Equations 14.1—See Reference 4 for More Information.