CHAPTER 14

Spur and Helical Gears

- In this chapter we are going to analyze and design Spur and Helical gears to:
  - Resist bending failure of the teeth
  - Pitting failure of tooth surfaces.
- Failure by bending will occur when the significant tooth stress equals or exceeds either the yield strength or the bending endurance strength.
- A surface failure occurs when the significant contact stress equals or exceeds the surface endurance strength.

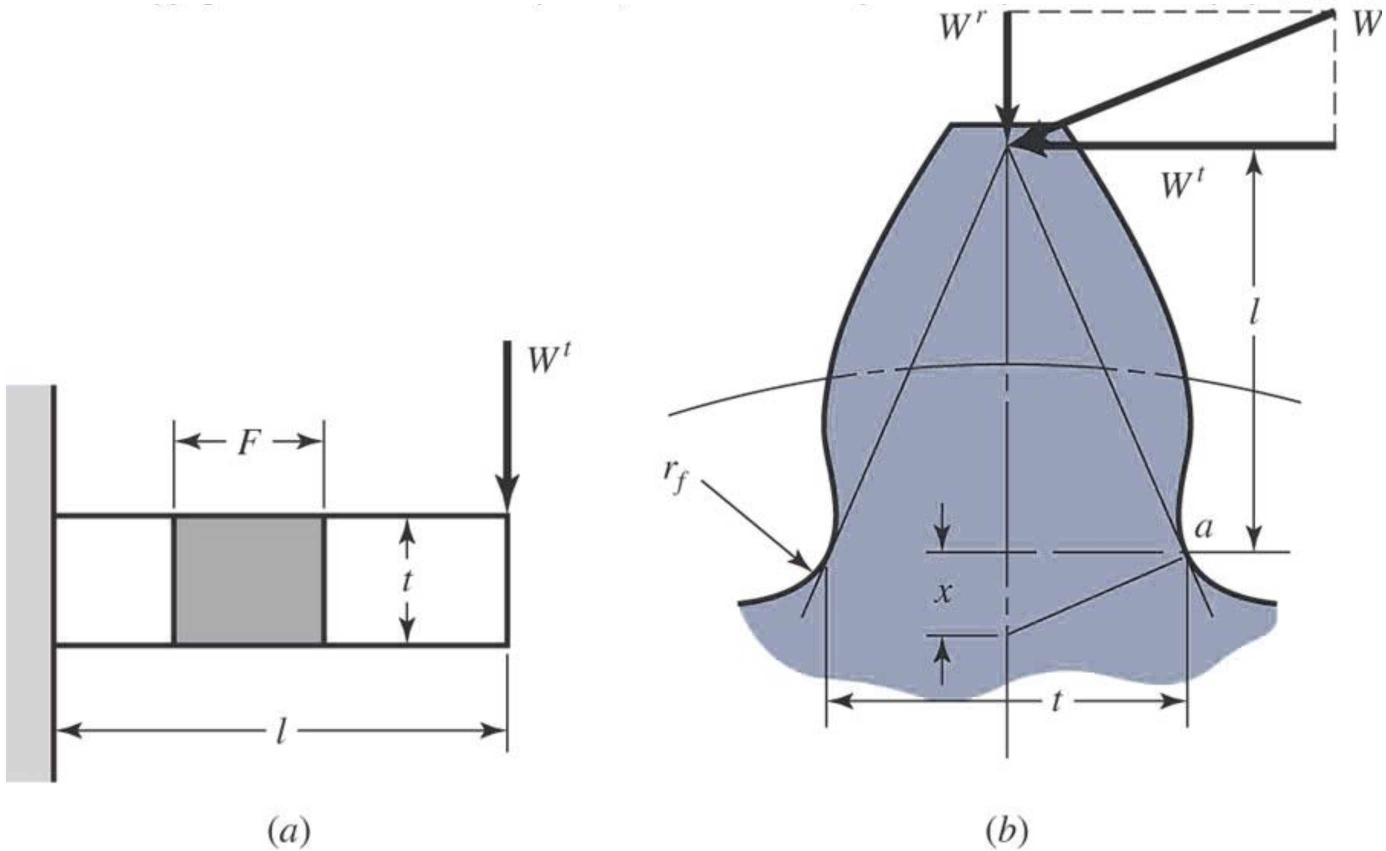
- In the first two sections a little of the history of the analyses from which current methodology developed is going to be covered.
- The American Gear Manufacturers Association (AGMA) has for many years been the responsible authority for the distribution of knowledge pertaining to the design and analysis of gearing. The methods this organization presents are in general use when strength and wear are primary considerations.

- The general AGMA approach requires a great many charts and graphs.
- Many of these charts and graphs can be omitted by choosing a single pressure angle and by using only full-depth teeth.
- This simplification reduces the complexity and makes possible a better development of the fundamentals and hence should constitute an ideal introduction to the use of the general AGMA method.

14.1 The Lewis Bending Equation

Static Effect

- An equation for estimating the bending stress in gear teeth in which the tooth form entered into the formulation was introduced by Wilfred Lewis.
- To derive the basic Lewis equation (See Fig. 14-la) which shows a cantilever of cross-sectional dimensions F and t, having a length l and a load W^t, uniformly distributed across the face width F.

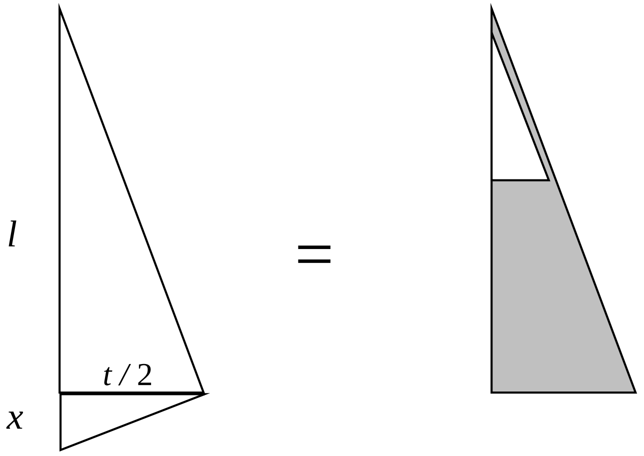


- The section modulus *I/c* is *Ft²/6*, and therefore the bending stress is:

$$\sigma = \frac{M}{I/c} = \frac{6W^t l}{Ft^2} \qquad \text{(a)}$$

- The maximum stress in a gear tooth occurs at point a as shown in figure 14-1a.
- Using the similarity of triangles, we can write:

$$\frac{x}{t/2} = \frac{t/2}{l} \Rightarrow x = \frac{t^2}{4l} \qquad \text{(B)}$$



- From equation (a): we can rewrite it as the following:

$$\sigma = \frac{6W^t l}{Ft^2} = \frac{W^t}{F}\frac{1}{t^2/6l} = \frac{W^t}{F}\frac{1}{t^2/4l}\frac{1}{4/6} \qquad \text{(C)}$$

- Substitute (b) into (c) and multiply the numerator and denominator by the circular pitch *p*, we find

$$\sigma = \frac{W^t p}{F(2/3)xp} \qquad \text{(d)}$$

- Letting $y = 2x/3p$, we have

$$\sigma = \frac{W^t}{Fpy} \qquad \text{(14-1)}$$

- Equation (14-1) is known as Lewis equation and y is called the Lewis form factor.
- Most engineers prefer to employ the diametral pitch in determining the stresses. This is done by substituting $P=\pi / p$ and $Y= \pi y$ in equation (14-1). This gives

$$\sigma = \frac{W^t P}{FY} \qquad (14\text{-}2)$$

Where

$$Y = \frac{2xP}{3} \qquad (14\text{-}3)$$

- Equation (14-2) considers only the bending of the tooth. And the effect of the radial load W^r is neglected.

- The value of *Y* can be found in table 14-2. (These values are for a normal pressure angle of 20^o, Full-Depth Teeth, and a Diametral Pitch of Unity in the Plane of Rotation)

Number of Teeth	Y	Number of Teeth	Y
12	0.245	28	0.353
13	0.261	30	0.359
14	0.277	34	0.371
15	0.290	38	0.384
16	0.296	43	0.397
17	0.303	50	0.409
18	0.309	60	0.422
19	0.314	75	0.435
20	0.322	100	0.447
21	0.328	150	0.460
22	0.331	300	0.472
24	0.337	400	0.480
26	0.346	Rack	0.485

- Eq. (14-3) also implies that
  - the teeth do not share the load
  - the greatest force is exerted at the tip of the tooth.
- the contact ratio should be somewhat greater than unity, say about 1.5, to achieve a quality gearset. If, in fact, the gears are cut with sufficient accuracy, the tip-load condition is not the worst, because another pair of teeth will be in contact when this condition occurs.

- Examination of run-in teeth will show that the heaviest loads occur near the middle of the tooth. Therefore the maximum stress probably occurs while a single pair of teeth is carrying the full load, at a point where another pair of teeth is just on the verge of coming into contact.

Dynamic Effect

- Dynamic effect occurs when a pair of gears is driven at moderate or high speed and noise is generated
- Barth first expressed the velocity factor in terms of AGMA standards and can be represented as (SI units):

$$K_\upsilon = \frac{3.05 + V}{3.05} \qquad \text{(cast iron, cast profile)} \qquad (14\text{-}6a)$$

Where *V* is the pitch-line velocity in m/s

- Barth equation is modified for cut or milled profile (see equation 14-6b)

$$K_\upsilon = \frac{6.10 + V}{6.10} \qquad \text{(cut or milled profile)} \qquad (14\text{-}6b)$$

- Later AGMA added two velocity factors for shaped and ground profiles and can be written as:

$$K_\upsilon = \frac{3.56 + \sqrt{V}}{3.56} \qquad \text{(hobbed or shaped profile)} \qquad (14\text{-}6c)$$

$$K_\upsilon = \sqrt{\frac{5.56 + \sqrt{V}}{5.56}} \qquad \text{(shaved or ground profile)} \qquad (14\text{-}6d)$$

$$K_v = \frac{600 + V}{600} \qquad \text{(cast iron, cast profile)}$$

$$K_v = \frac{1200 + V}{1200} \qquad \text{(cut or milled profile)}$$

$$K_v = \frac{50 + \sqrt{V}}{50} \qquad \text{(hobbed or shaped profile)}$$

$$K_v = \sqrt{\frac{78 + \sqrt{V}}{78}} \qquad \text{(shaved or ground profile)}$$

V is in ft/min

- By introducing the velocity factor into (14-2) gives

$$\sigma = \frac{K_{\upsilon} W^{t} P}{FY} \qquad (14\text{-}7)$$

- The metric version of this equation is

$$\sigma = \frac{K_{\upsilon} W^{t}}{FmY} \qquad (14\text{-}8)$$

- Where

F = the face width (mm)

m = the modile (mm)

W^{t} = tangential component of the load (N)

- As a general rule, spur gears should have a face width *F* from 3 to 5 times the circular pitch *p*.

- Equations (14-7) and (14-8) are important because they form the basis for AGMA approach to the bending strength of gear teeth. They are in general use for estimating the capacity of gear drives when life and reliability are not important considerations.

- The equations can be useful in obtaining a preliminary estimate of gear sizes needed for various applications.

Example 14-1

- A stock spur gear is available having a diametral pitch of 8 teeth/in, a 1.50 in face, 16 teeth, and a pressure angle of 20° with full-depth teeth. The material is AISI 1020 steel in as-rolled condition. Use a design factor of $n_d = 3$ to rate the horsepower output of the gear corresponding to a speed of 1200 rev/m and moderate applications.

Solution

- The term *moderate applications* seems to imply that the gear can be rated by using the yield strength as a criterion of failure.
- From Table A-20, we find:
- S_{ut} = 55 kpsi=379MPa and S_y = 30kpsi=207MPa
- A design factor of 3 means that the allowable bending stress is 207/3 = 69 MPa
- The pitch diameter is *N/P* = 16/8 = 2 in, so the pitch-line velocity is:

$$V = \frac{\pi d n}{12} = \frac{\pi(2)1200}{12} = 628\, ft/\min = 0.0051(628) = 3.2028 m/s$$

- Using milled profile (eq. 14-6b) for the velocity factor

$$K_\upsilon = \frac{6.1+V}{6.1} = \frac{9.3028}{6.1} = 1.525$$

- From table 14-2, for *N* =16 teeth *Y* = 0.296, thus

$$\sigma = \frac{K_\upsilon W^t}{FmY} \Rightarrow W^t = \frac{FmY\sigma_{all}}{K_\upsilon} = \frac{(38.1)(0.125*25.4)(0.296)(69)}{1.525} = 1620N$$

From equation 13 - 35,

$$H = \frac{\pi dnW^t}{60(10)^3} = \frac{\pi(2*25.4)1200*1620}{60(10)^3} = 5171.1W = (5171.1/746)\text{hp} = 6.93\text{hp}$$

Example 14-2

- Estimate the horsepower rating of the gear in the previous example (14-1) based on obtaining an infinite life in bending.

Solution

The rotating-beam endurance limit is estimated from Eq. (7-8)

$$S_e^{'} = 0.504 S_{ut} = 0.504(379) = 191 \text{MPa}$$

To obtain the surface finish Marin factor k_a, for machine surface

$$a = 2.7, b = -0.265$$

$$k_a = aS_{ut}^{b} = 2.70(191)^{-0.265} = 0.934$$

- For the size factor k_b

$$d_e = 0.808(hb)^{0.5} = 0.808(Ft)^{0.5}$$

$$F = 1.5in$$

$$t = (4l / x)^{0.5},$$

$$l = \frac{1}{P} + \frac{1.25}{P} = \frac{1}{8} + \frac{1.25}{8} = 0.281in$$

$$x = \frac{3Y}{2P} = \frac{3(0.296)}{2(8)} = 0.0555in$$

$$\therefore t = 0.25in \Rightarrow d_e = 0.495in$$

$$k_b = \left(\frac{d_e}{0.3}\right)^{-.0107} = 0.948$$

- for load factor k_c =1, for bending

$$k_d = k_e = k_f = 1$$

$$\therefore S_s = k_a k_b k_c k_d k_e k_f S_e^{'} = (0.934)(0.948)(1)(1)(1)(1)(191) = 169.12\text{MPa}$$

- Stress concentration factor K_f
- For a 20o full-depth tooth the radius of the root fillet is denoted r_f, where r_f = 0.3/P=0.3/8=0.0375 in
- From Figure A-15-6

$$\frac{r}{d} = \frac{r_f}{t} = \frac{0.0375}{0.25} = 0.15$$

Since $D/d = \infty$, we can take $D/d = 3, \Rightarrow K_t = 1.68$

From figure 7.20, $q = 0.63$

$$\therefore K_f = 1 + q(K_t - 1) = 1 + 0.63(1.68 - 1) = 1.43$$

$$\sigma_{all}\big|_o = \frac{S_e}{n_d} = \frac{169.12}{3} = 56.373\text{MPa}$$

$$\sigma_{all} = \sigma_{all}\big|_o / K_f = 39.42\text{MPa}$$

$$W^t = \frac{FmY\sigma_{all}}{K_v} = \frac{38.1(0.125*25.4)(0.296)(39.42)}{1.525} = 925.58N$$

$$H = \frac{\pi dnW^t}{60(10)^3} = 2952W = (2952/746) = 3.96\text{hp}$$

- These results should be accepted only as preliminary estimates to alert you to the nature of bending in gear teeth.

Dolan and Broghamer stress concentration formula

$$K_f = H + \left(\frac{t}{r}\right)^L + \left(\frac{t}{l}\right)^M \qquad (14\text{-}9)$$

Where

$$H = 0.34 - 0.4583662\phi$$

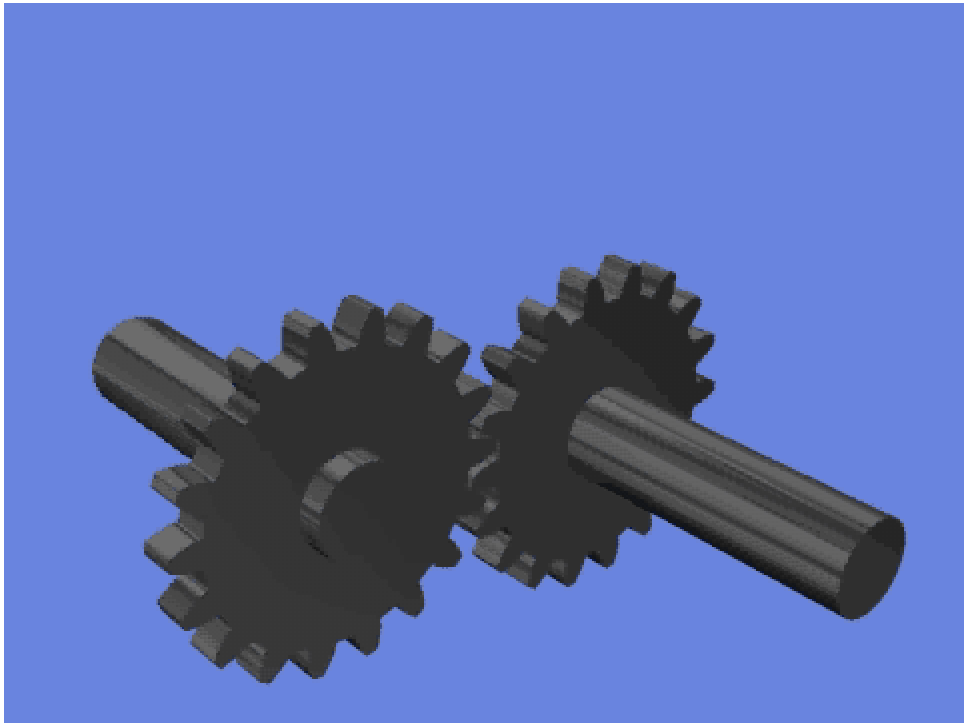
$$L = 0.316 - 0.4583662\phi$$

$$M = 0.290 + 0.4583662\phi$$

$$r = \frac{(b - r_f)^2}{(d/2) + b - r_f}$$

ϕ = pressure angle, r_f = fillet radius, b = the dedendum, d = the pitch diameter

Gear Teeth Contact



14.2 Surface Durability

- In this section, the failure of the surfaces of gear teeth, which is generally called wear is studied. To obtain an expression for the surface-contact stress, the Hertz theory is employed.
- It was shown that the contact stress between two cylinders may be computed from the equation:

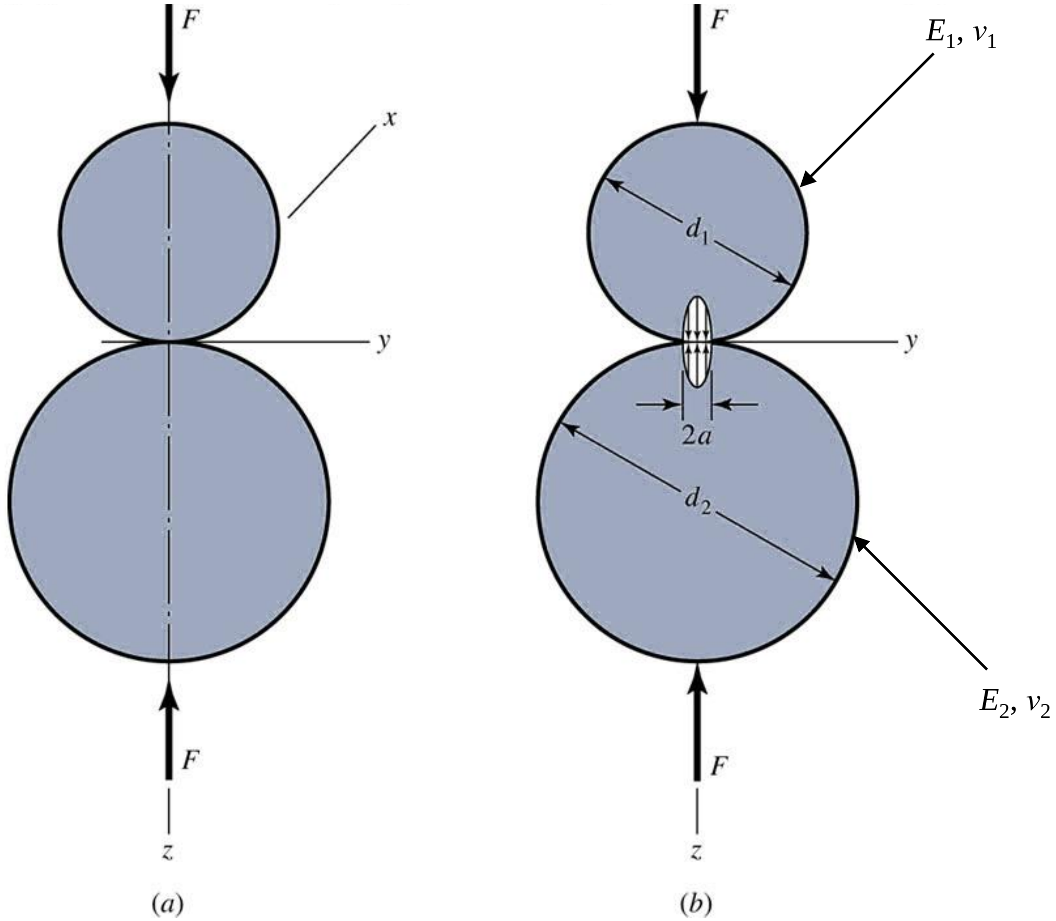
$$p_{max} = \frac{2F}{\pi b l} \qquad \text{(a)}$$

Where

p_{max} = largest surface pressure

F = force pressing the two cylinders together

l = length of cylinders



$$b = \text{half - width} = \left\{ \frac{2F}{\pi l} \frac{\left[\left(1-\nu_1^2\right)/E_1\right] + \left[\left(1-\nu_2^2\right)/E_2\right]}{\left(1/d_1\right) + \left(1/d_2\right)} \right\}^{1/2} \quad (14\text{-}10)$$

- To adapt these relations to the notation used in gearing, we replace *F* by Wt/cos(ϕ), *d* by 2*r*, and *l* by the face width *F*.
- With these changes, we can substitute the value of *b* as given by equation (14-10) in equation (a). Replacing *p*max by σ_C, the surface compressive stress (Hertzian stress) is found from the equation:

$$\sigma_C^2 = \frac{W^t}{\pi F \cos\phi} \frac{(1/r_1) + (1/r_2)}{\left[\left(1-\nu_1^2\right)/E_1\right] + \left[\left(1-\nu_2^2\right)/E_2\right]} \qquad (14\text{-}11)$$

- Where r_1 and r_2 are the instantaneous values of the radii of curvature on the pinion- and gear-tooth profiles, respectively, at the point of contact.

- By accounting for load sharing in the value of W^t used, equation (14-11) can be solved for the Hertzian stress for any or all points from the beginning to the end of tooth contact
- Noted that the first evidence of wear occurs near the pitch line. The radii of curvature of the tooth profiles at the pitch point are:

$$r_1 = \frac{d_P \sin\phi}{2} \qquad r_2 = \frac{d_G \sin\phi}{2} \qquad (14\text{-}12)$$

  where ϕ is the pressure angle and d_P and d_G are the pitch diameters of the pinion and the gear, respectively.
- In equation (14-11), that the denominator of the second group of terms contains four elastic constants, two for the pinion and two for the gear.

- As a simple means of combining and tabulating the results for various combinations of pinion and gear materials, AGMA defines an elastic coefficient C_P by the equation

$$C_P = \left[\frac{1}{\pi \left(\frac{1-\nu_P^2}{E_P} + \frac{1-\nu_G^2}{E_G} \right)} \right]^{1/2} \qquad (14\text{-}13)$$

- With this simplification, and the addition of a velocity factor K_v, equation (14-11) can be written as:

$$\sigma_C = -C_P \left[\frac{K_\upsilon W^t}{F \cos\phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{1/2} \qquad (14\text{-}14)$$

Where the sign is negative because σ_C is a compressive stress.

Example 14-3

- The pinion of Example 14-1 and 14-2 is to be mated with a 50 tooth gear manufactured of ASTM No. 50 cost iron. Using the tangential load of 382 lbf, estimate the factor of safety of the drive based on the possibility of a surface fatigue failure.

Solution

- From Table A-5:

$$E_P = 30\text{Mpsi},\ \nu_P = 0.292,\ E_G = 14.5\text{Mpsi},\ \nu_G = 0.211$$

$$C_P = \left\{ \frac{1}{\pi\left[\frac{1-(0292)^2}{30(10^6)} + \frac{1-(0211)^2}{14.5(10^6)}\right]} \right\}^{1/2} = 1817$$

- From example 14-1, the pinion pitch diameter is d_P = 2in, and for the gear d_G = 50/8 =6.25in
- Thus from equation (14-12):

$$r_1 = \frac{2\sin 20^o}{2} = 0.342\text{in},\ r_2 = \frac{6.25\sin 20^o}{2} = 1.069\text{in}$$

- The face width is given as F = 1.5 in and K_v =1.525 from example 14-1. Substituting all these values in equation (14-14) with pressure angle of 20^o, thus the contact stress is

$$\sigma_C = -C_P\left[\frac{K_\upsilon W^t}{F\cos\phi}\left(\frac{1}{r_1}+\frac{1}{r_2}\right)\right]^{1/2}$$

$$= -1817\left[\frac{1.525(380)}{1.5\cos 20}\left(\frac{1}{0.342}+\frac{1}{1.069}\right)\right]^{1/2} = -72400\text{psi}$$

- The surface endurance strength of cast iron can be estimated from

$$S_C = 0.32 H_B \text{ kpsi, for } 10^8 \text{ cycles,}$$

- For table A-24, *HB* = 262 for ASTM No. 50 cast iron, Thus $S_C = 0.32(262) = 83.8$ kpsi
- Thus the factor of safety can be found as:

$$n = \left(\frac{\text{loss of function load}}{\text{imposed load}} \right)^2 = \left(\frac{83.8}{72.4} \right)^2 = 1.35$$

14.3 AGMA Stress Equations

- Two fundamental stress equations are used in the AGMA methodology, these are:
  - The bending stress
  - The pitting resistance (contact stress).
- In AGMA terminology, these are called *stress numbers* and are designated by a lowercase letter s instead of the Greek lower case σ. The fundamental equations are:

$$\sigma = \begin{cases} W^t K_o K_\upsilon K_s \dfrac{P_d}{F} \dfrac{K_m K_B}{J} & \text{(U.S. customary units)} \\ W^t K_o K_\upsilon K_s \dfrac{1}{b m_t} \dfrac{K_H K_B}{Y_J} & \text{(SI units)} \end{cases} \qquad (14\text{-}15)$$

Where for U.S. customary units (SI units),

W^t = the tangential transmitted load, Ibf (N)

K_o = the overload factor

K_υ = the dynamic factor

K_s = the size factor

P_d = the transverse diameteral pitch

$F(b)$ = the face width of the narrower member, in (mm)

$K_m(K_H)$ = the load-distribution factor

K_B = the rim-thickness factor

$J(Y_J)$ = the geometry factor for bending strengh

(which includes root fillet stress-concentration factor K_f)

m_t: transverse metric module

- Before you try to digest the meaning of all these terms in equation (14-15), view them as advice from AGMA concerning items the designer should consider whether you follows the voluntary standard or not. These items include issues such as:

  - Transmitted load magnitude
  - Overload
  - Dynamic augmentation of transmitted load
  - Size
  - Geometry: pitch and face width
  - Distribution of load across the teeth
  - Rim support of the tooth
  - Lewis form factor and root fillet stress concentration

- The fundamental equation for pitting resistance (contact stress) is

$$\sigma_C = \begin{cases} C_P \sqrt{W^t K_o K_\upsilon K_s \dfrac{K_m}{d_P F} \dfrac{C_f}{I}} & \text{(U.S. customary units)} \\ Z_E \sqrt{W^t K_o K_\upsilon K_s \dfrac{K_H}{d_{w1} b} \dfrac{Z_R}{Z_I}} & \text{(SI units)} \end{cases} \qquad (14\text{-}16)$$

Where

$C_p (Z_E)$ = an elastic coefficient, $\sqrt{\text{lbf/in}^2}$ ($\sqrt{\text{N/mm}^2}$)

$C_f (Z_R)$ = the surface condition factor

$d_P (d_{W1})$ = the pitch diameter of the pinion, in (mm)

$I (Z_I)$ = the geometry factor for pitting resistance

The evaluation of all these factors is explained in the sections that follow.

14.4 AGMA Strength Equations

Instead of using the term *strength*, AGMA uses data termed *allowable stress numbers* and designates these by the symbols s_{at} and s_{ac}. It will be less confusing here if we continue the practice in this book of using the uppercase letter S to designate strength and the lowercase Greek letters σ and τ for stress. To make it perfectly clear we shall use the term *gear strength* as a replacement for the phrase *allowable stress numbers* as used by AGMA.

- Following this convention, values for AGMA bending strength, designated here as *St*, are to be found in Figs. 14-2, 14-3, and 14-4, and in Tables 14-3 and 14-4.

- Since AGMA strengths are not identified with other strengths such as S_{ut}, S_e, or S_y as used elsewhere in this book, their use should be restricted to gear problems.

In this approach the strengths are modified by various factors that produce limiting values of the bending stress and the contact stress.

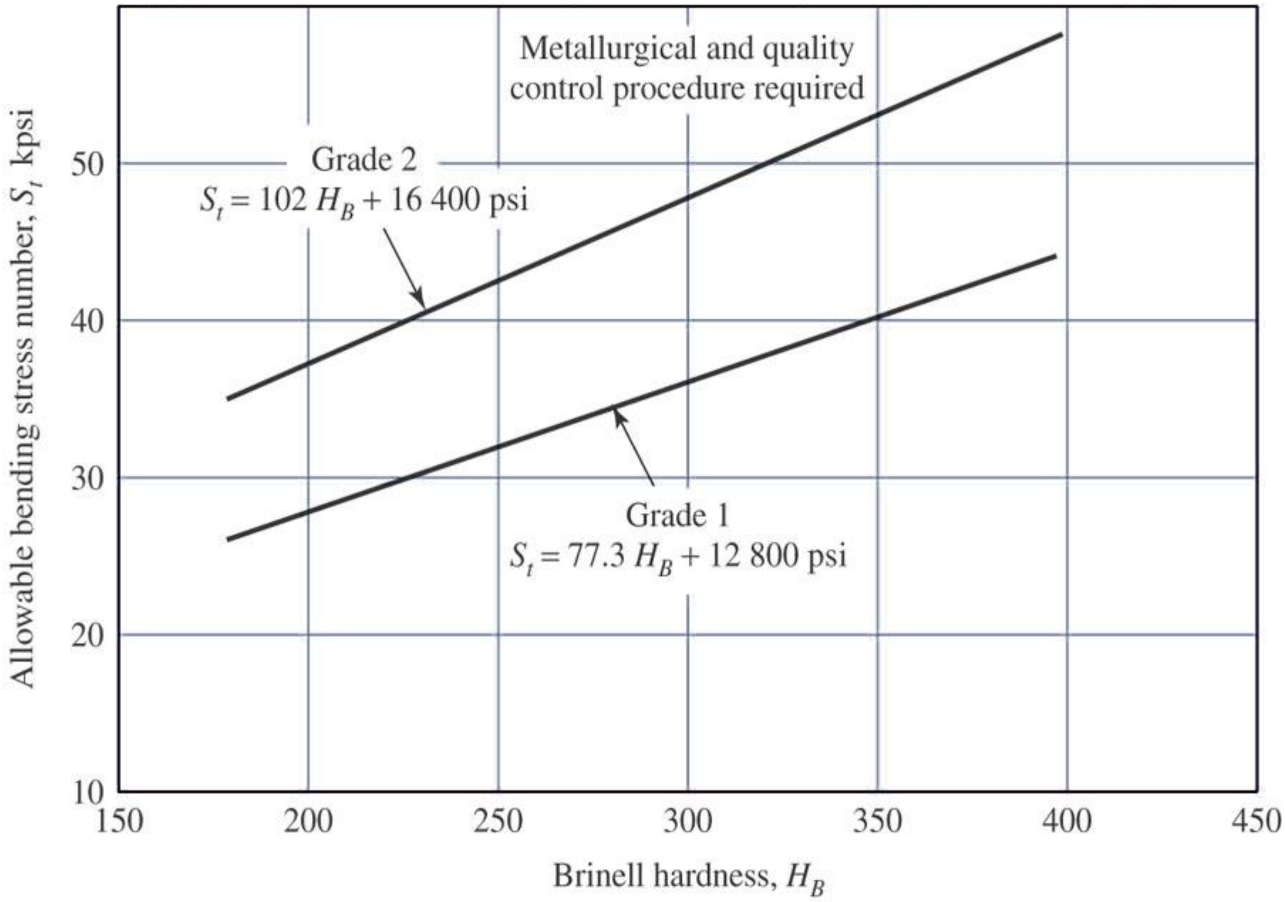


Figure 14-2: Allowable bending stress number for through-hardened steels. The SI equations are S_t , $\sigma_{FP} = 0.533H_B + 88.3$ MPa, grade 1, and $\sigma_{FP} = 0.703H_B + 113$ MPa, for grade 2.

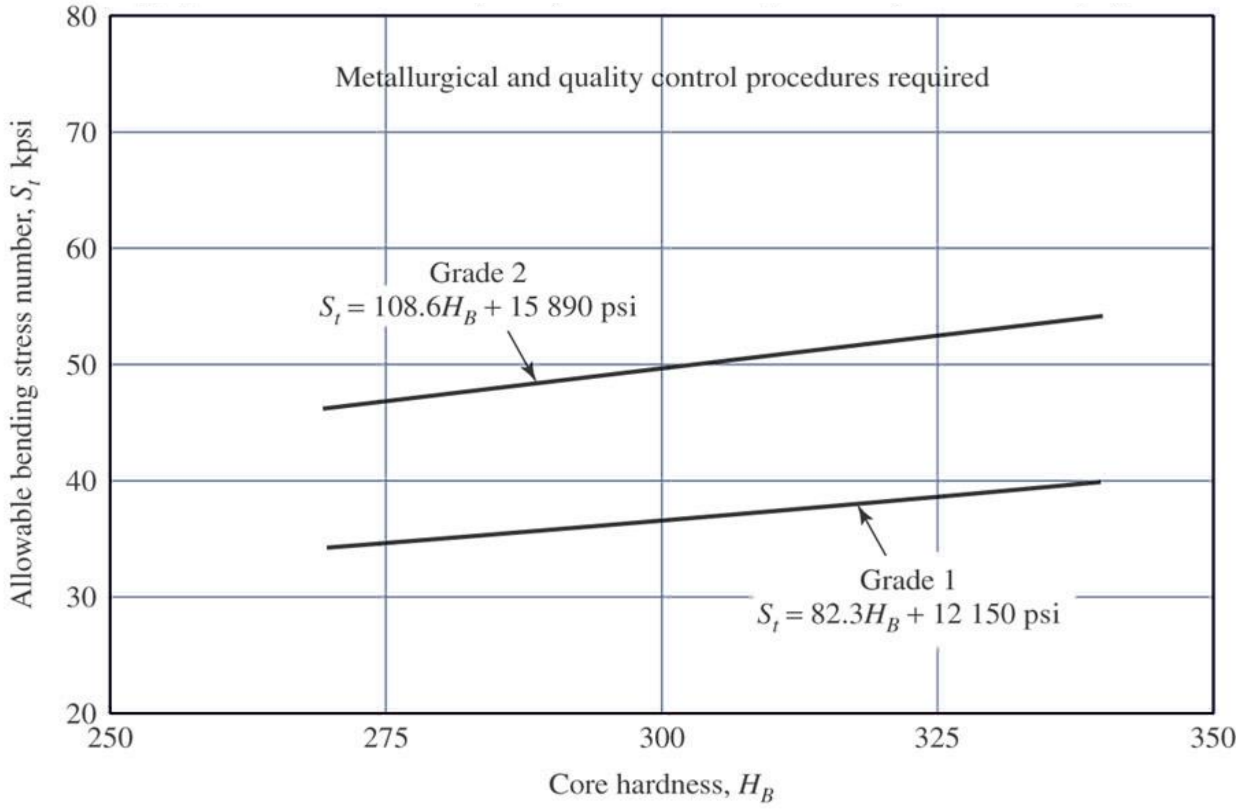


Figure 14-3: Allowable bending stress number for nitrided through-hardened steel gears (i.e., AISI4140, 4340, *St*. The SI equations are $\sigma_{FP} = 0.568H_B + 83.8$ MPa, grade 1, and $\sigma_{FP} = 0.749H_B + 110$ MPa, for grade 2.

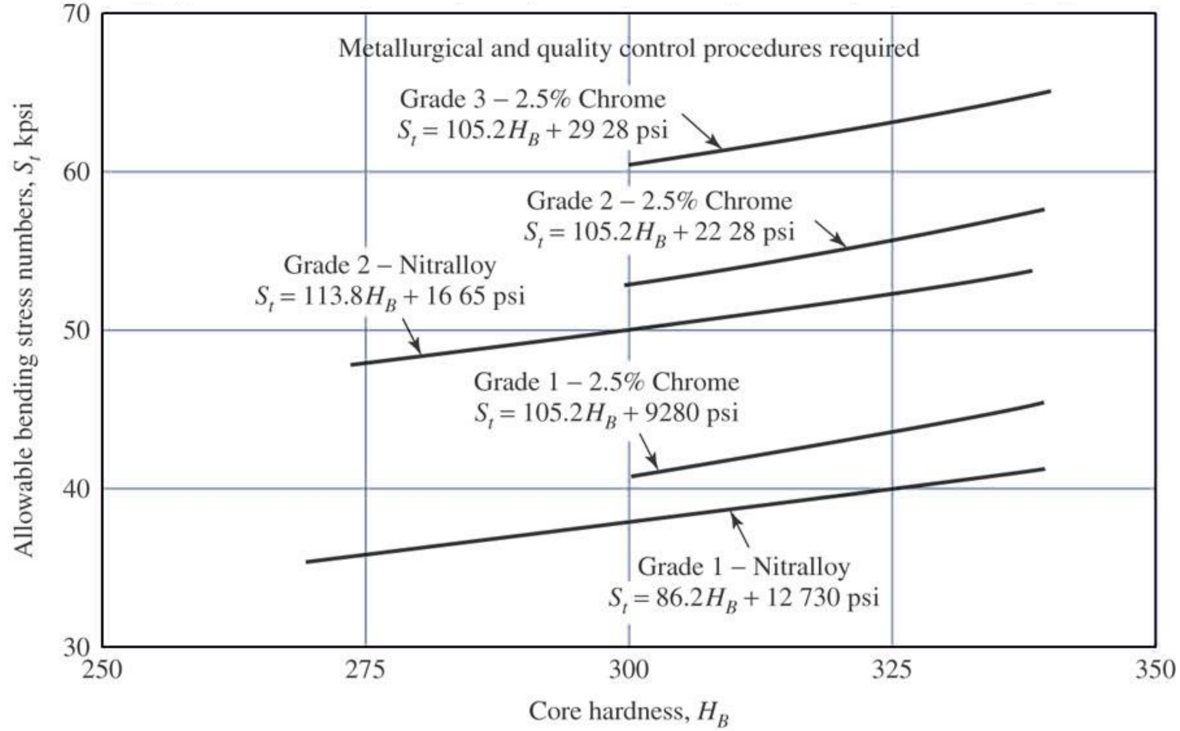


Figure 14-4: Allowable bending stress number for nitriding through-hardened steel gears *St*. The SI equations are σFP =0.594H_B + 87.76 MPa Nitralloy grade 1, σ_{FP} =0.784H_B + 114.81 MPa Nitralloy grade 2, σ_{FP} =0.7255H_B + 63.89 MPa 2.5% chrome, grade 1, σ_{FP} =0.7255H_B + 153.63 MPa 2.5% chrome, grade 2, and σ_{FP} = 0.7255H_B+ 201.91 MPa 2.5% chrome, grade 3.

Table 14–3

Repeatedly Applied Bending Strength S_t at 10^7 Cycles and 0.99 Reliability for Steel Gears

Source: ANSI/AGMA 2001-D04.

Material Designation	Heat Treatment	Minimum Surface Hardness[1]	Allowable Bending Stress Number S_t,[2] psi Grade 1	Grade 2	Grade 3
Steel[3]	Through-hardened	See Fig. 14–2	See Fig. 14–2	See Fig. 14–2	—
	Flame[4] or induction hardened[4] with type A pattern[5]	See Table 8*	45 000	55 000	—
	Flame[4] or induction hardened[4] with type B pattern[5]	See Table 8*	22 000	22 000	—
	Carburized and hardened	See Table 9*	55 000	65 000 or 70 000[6]	75 000
	Nitrided[4,7] (through-hardened steels)	83.5 HR15N	See Fig. 14–3	See Fig. 14–3	—
Nitralloy 135M, Nitralloy N, and 2.5% chrome (no aluminum)	Nitrided[4,7]	87.5 HR15N	See Fig. 14–4	See Fig. 14–4	See Fig. 14–4

Table 14–4

Repeatedly Applied Bending Strength S_t for Iron and Bronze Gears at 10^7 Cycles and 0.99 Reliability

Source: ANSI/AGMA 2001-D04.

Material	Material Designation[1]	Heat Treatment	Typical Minimum Surface Hardness[2]	Allowable Bending Stress Number, S_t,[3] psi
ASTM A48 gray cast iron	Class 20	As cast	—	5000
	Class 30	As cast	174 HB	8500
	Class 40	As cast	201 HB	13 000
ASTM A536 ductile (nodular) Iron	Grade 60–40–18	Annealed	140 HB	22 000–33 000
	Grade 80–55–06	Quenched and tempered	179 HB	22 000–33 000
	Grade 100–70–03	Quenched and tempered	229 HB	27 000–40 000
	Grade 120–90–02	Quenched and tempered	269 HB	31 000–44 000
Bronze		Sand cast	Minimum tensile strength 40 000 psi	5700
	ASTM B–148 Alloy 954	Heat treated	Minimum tensile strength 90 000 psi	23 600

- In the AGMA approach the strengths are modified by various factors that produce limiting values of the bending stress and the contact stress. Using similar notation to that in equation (1-6) we shall term the resulting modifications the allowable bending stress σ_{all} and the allowable contact stress σ_{C}, all . The equation for the allowable bending stress is:

$$\sigma_{all} = \begin{cases} \dfrac{S_t}{S_F}\dfrac{Y_N}{K_T K_R} & \text{(U.S. customary units)} \\ \dfrac{\sigma_{FP}}{S_F}\dfrac{Y_N}{Y_\theta Y_Z} & \text{(SI units)} \end{cases} \qquad (14\text{-}17)$$

Where for the U.S. customary units (SI units),

$$S_t(\sigma_{FP}) = \text{allowable bending stress, lbf/in}^2\ (\text{N/mm}^2)$$

$$Y_N = \text{the stress cycle factor for bending stress}$$

$$K_T(Y_\theta) = \text{the temperature factors}$$

$$K_R(Y_Z) = \text{the reliability factors}$$

$$S_F = \text{the AGMA factor of safety, a stress ratio}$$

- The equation for the allowable contact stress $\sigma_{C,all}$ is

$$\sigma_{C,all} = \begin{cases} \dfrac{S_C}{S_H} \dfrac{Z_N C_H}{K_T K_R} & \text{(U.S. customary units)} \\ \dfrac{\sigma_{HP}}{S_H} \dfrac{Z_N Z_W}{Y_\theta Y_Z} & \text{(SI units)} \end{cases} \qquad (14\text{-}18)$$

- Where the upper equation is in U.S customary units and the lower equation is in SI units, Also

$S_C(\sigma_{HP}) =$ allowable contact stress, lbf/in^2 (N/mm^2)

$Z_N =$ the stress cycle life factor

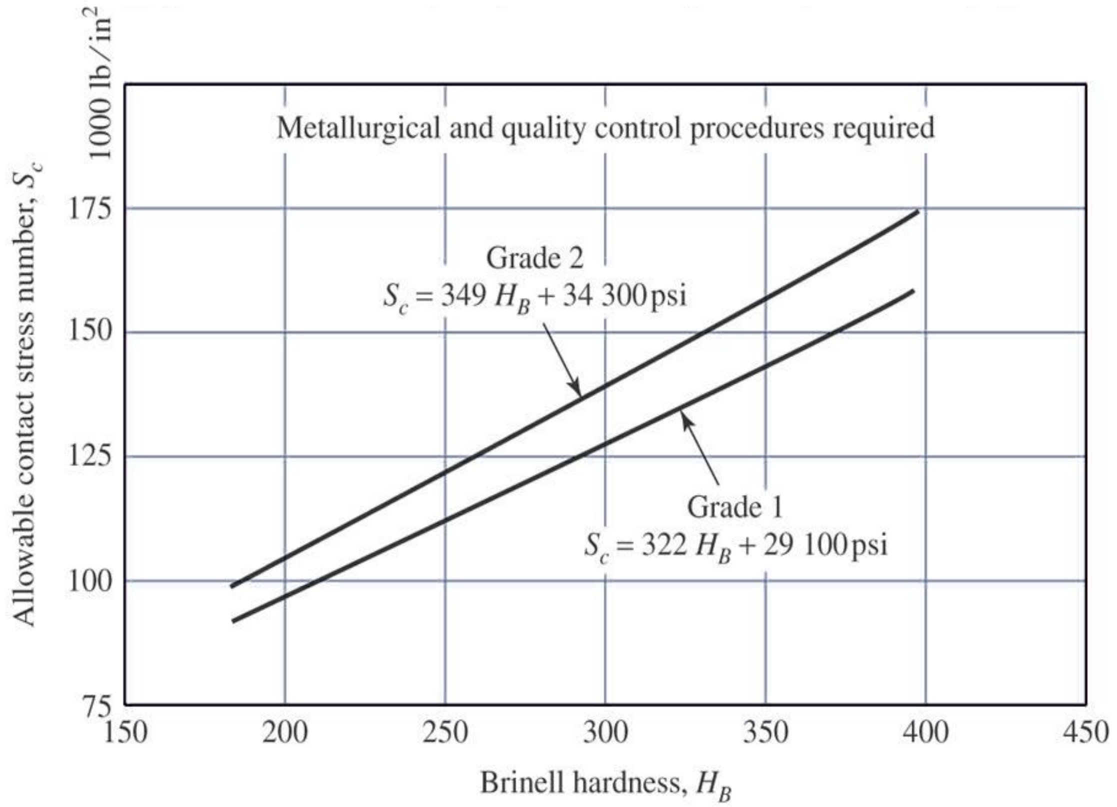
$C_H =$ the hardness ratio factors for pitting resistance

$K_T(Y_\theta) =$ the temperature factors

$K_R(Y_Z) =$ the reliability factors

$S_H =$ the AGMA factor of safety, a stress ratio

- The values for the AGMA allowable stress numbers (strengths) for bending and contact stress are for
  - Unidirectional loading
  - 10 million stress cycles
  - 99 percent reliability



- Figure 14-5: Contact-fatigue strength SC at 107 cycles and 0.99 reliablity for through-hardened steel gears. The SI equations are $\sigma_{HP} = 2.22 H_B + 200$ MPa, grade 1, and $\sigma_{HP} = 2.41 H_B + 237$ MPa, grade 2

Table 14–6

Repeatedly Applied Contact Strength S_c at 10^7 Cycles and 0.99 Reliability for Steel Gears

Source: ANSI/AGMA 2001-D04.

Material Designation	Heat Treatment	Minimum Surface Hardness[1]	Allowable Contact Stress Number,[2] S_c, psi		
			Grade 1	Grade 2	Grade 3
Steel[3]	Through hardened[4]	See Fig. 14–5	See Fig. 14–5	See Fig. 14–5	—
	Flame[5] or induction hardened[5]	50 HRC	170 000	190 000	—
		54 HRC	175 000	195 000	—
	Carburized and hardened[5]	See Table 9*	180 000	225 000	275 000
	Nitrided[5] (through hardened steels)	83.5 HR15N	150 000	163 000	175 000
		84.5 HR15N	155 000	168 000	180 000
2.5% chrome (no aluminum)	Nitrided[5]	87.5 HR15N	155 000	172 000	189 000
Nitralloy 135M	Nitrided[5]	90.0 HR15N	170 000	183 000	195 000
Nitralloy N	Nitrided[5]	90.0 HR15N	172 000	188 000	205 000
2.5% chrome (no aluminum)	Nitrided[5]	90.0 HR15N	176 000	196 000	216 000

Notes: See ANSI/AGMA 2001-D04 for references cited in notes 1–5.