Material Selection and Heat Treatment

by
National Broach & Machine
Mt. Clemens, Michigan

Table 1 – AGMA Standards Useful in Selecting Gear Materials

Nomenclature		
110.03	Gear-Tooth Wear and Failure (USAS B6.12-1964)	Jan. 1962
Durability		
215.01	Information Sheet for Surface Durability (Pitting) of Spur, Helical, Herringbone and Bevel Gear Teeth	Sept. 1966
217.01	Information Sheet-Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears	Oct. 1965
Strength		
225.01	Information Sheet for Strength of Spur, Helical, Herringbone and Bevel Gear Teeth	Dec. 1967
Inspection		
230.01	AGMA Standard-Surface Temper Inspection Process	March 1968
Materials		
241.02	Specification for General Industrial Gear Materials—Steel (Drawn, Rolled and Forged)	Jan. 1965
242.02	Cast Iron Blanks	Sept. 1946
244.02	Nodular Iron Gear Materials	July 1965
245.01	Specification for Cast Steel Gear Materials	Jan. 1964
246.01	Recommended Procedure for Carburized Industrial Gearing	Jan. 1965
247.01	Recommended Procedure for Nitriding, Materials and Process	Jan. 1965
248.01	Recommended Procedure for Induction Hardened Gears and Pinions	Jan. 1964
249.01	Recommended Procedure for Flame Hardening	Jan. 1964

Gear materials are selected to provide the optimum combination of properties, at the lowest possible cost consistent with satisfying other requirements. Some of the important physical properties of gears are abrasion or wear resistance, toughness, static compression strength, shear strength, fatigue strength, and strength at elevated temperatures.

Because of widely varying requirements, gears are produced from a wide variety of materials. These materials include plastics such as nylon, powdered metals, brasses, bronzes, cast or ductile irons, and steels. Many types of steels, including stainless steel and tool steel, are used. Each of the materials mentioned will best satisfy some specific requirement such as corrosion resistance, extreme wear resistance, special damping qualities, ability to operate without lubrication, low cost, or producibility.

The majority of gears for automotive, aircraft, farm machinery, off-the-road equipment, and machine tool applications are produced from hardenable carbon or low-alloy steels or cast iron. Therefore, this article will cover only ferrous materials.

The choice of a gear material depends on four factors:

1. Mechanical properties
2. Metallurgical characteristics
3. Blank-forming method
4. Manufacturing process

Each of these factors must be evaluated with an eye to its effect on the three major performance criteria – durability, strength, and wear.

Mechanical Properties*

Before the optimum mechanical properties can be selected, the working stress must be determined, based on recommended allowable stresses.

This article discusses durability and strength formulas adopted by the American Gear Manufacturers Association and widely used throughout the world, Table 1. There is a fundamental relationship among most formulas, so that working and allowable stresses determined by different formulas can be used or compared.

Working stress is usually based on the fatigue or yield strength of the material. Less frequently, impact resistance, tensile strength, or brittle-fracture characteristics of the material must be considered.

For most gear trains, the limiting design consideration is profile durability (pitting resistance), gear-tooth strength (resistance to fracture), or wear. Normally, durability is the limiting consideration; but sometimes all three are of nearly equal importance.

A number of other possible modes of failure may also limit gear performance, such as scoring associated with high

*Abstracted from Machine Design, Gear Materials Design Guide, June 20, 1968.

speed and heavy loads, case crushing in carburized and hardened gearing, and micro-pitting. However, these are less likely to occur.

The AGMA gear-rating formulas for both strength and durability are basically the same for spur, helical, herringbone, and bevel gear teeth. Terms in both formulas are divided into four major groupings, associated with load, size, stress distribution, and stress.

The surface-durability or pitting-resistance formula – *AGMA 215.01, Sept. 1966, "Information Sheet for Surface Durability (Pitting) of Spur, Helical, Herringbone and Bevel Gear Teeth"* – is

$$s_c = C_p \sqrt{\underbrace{\left(\frac{W_t C_o}{C_v}\right)}_{\text{(Load)}} \underbrace{\left(\frac{C_s}{dF}\right)}_{\text{(Size)}} \underbrace{\left(\frac{C_f C_m}{I}\right)}_{\text{(Stress dist.)}}} \quad (1)$$

where

$$s_c \leqq s_{ac} \underbrace{\left(\frac{C_L C_H}{C_R C_T}\right)}_{\text{(Stress)}} \quad (2)$$

The "stress" term, Equation 2, is the most important consideration in selecting a gear material. Table 2 lists all symbols used in the strength and durability formulas. Allowable fatigue (contact) stresses, s_{ac} are shown in Table 3.

The mechanical properties for gear materials are almost always specified in terms of Brinell hardness rather than ultimate strength or test-bar properties. Location (normally the toothed portion) and number of hardness tests should also be specified. Maximum hardness is also specified in the interests of machinability and sometimes to insure a hardness difference between the mating gears; such a difference can increase wear resistance. Typical combinations are shown in Table 4.

Values of s_{ac} are based on 10^7 cycles, since most gear materials generally show the typical "knee" at or below this value. Hence, these values can be considered fatigue-strength stresses. If a gearset must operate for only a finite number of cycles, the s_{ac} values can be increased by life factor C_L, Fig. 1.

Pitting of gear teeth is considered a fatigue phenomenon. There are two kinds of pitting – initial and progressive (destructive). Corrective and non-progressive initial pitting is not considered serious and is normally to be expected until imperfections, such as high spots, are worn in.

The stresses in Table 3 are satisfactory

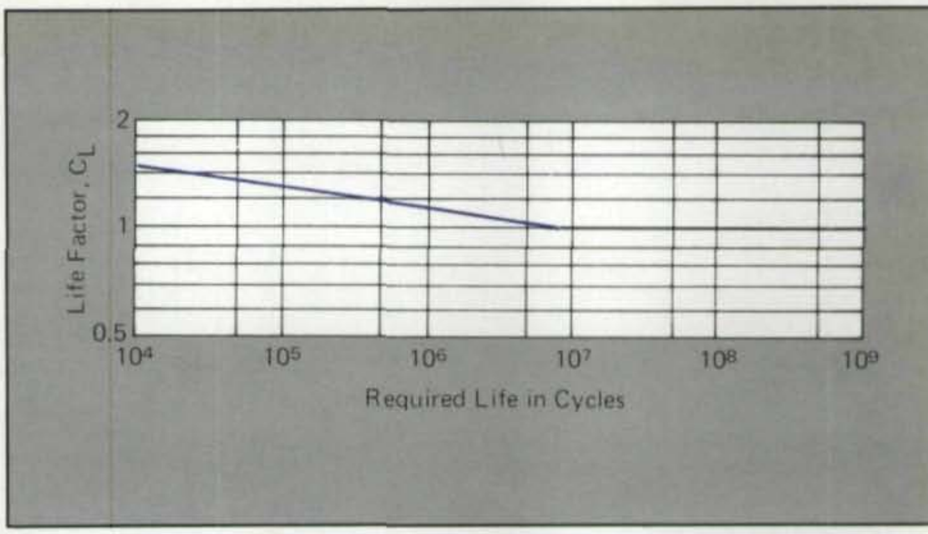


Fig. 1 – Life factor for durability rating of gears.

Table 2 – Gear Rating Factors

Factor	Strength	Durability
Load		
Transmitted load	W_t	W_t
Dynamic factor	K_v	C_v
Overload factor	K_o	C_o
Size		
Pinion pitch diameter	—	d
Net face width	F	F
Transverse diametral pitch	P_d	—
Size factor*	—	C_s
Stress Distribution		
Load-distribution factor	K_m	C_m
Geometry factor	J	I
Surface-condition factor	—	C_f
Size factor*	K_s	—
Stress		
Calculated stress	s_t	s_c
Allowable stress	s_{at}	s_{ac}
Elastic coefficient	—	C_p
Hardness-ratio factor	—	C_H
Life factor	K_L	C_L
Temperature factor	K_T	C_T
Safety factor	K_R	C_R

**Size factor is placed either in the size or stress-distribution grouping, depending upon the importance of the effect.*

Table 3 – Allowable Contact Stress (Durability)

Material	Minimum Surface Hardness	Contact Stress, s_{ac} (1000 psi)
Steel		
Through-hardened	180 Bhn	85-95
	240 Bhn	105-115
	300 Bhn	120-135
	360 Bhn	145-160
	440 Bhn	170-190
Case-carburized	55 R_c	180-200
	60 R_c	200-225
Flame or induction-hardened	50 R_c	170-190
Cast Iron		
AGMA Grade 20	—	50-60
AGMA Grade 30	175 Bhn	65-75
AGMA Grade 40	200 Bhn	75-85
Nodular Iron	165-300 Bhn	10% less than for steel with same hardness

Table 4 – Typical Gear/Pinion Hardness Combinations

Minimum Hardness (Bhn)*

Gear	180	210	225	255	270	285	300	335	350	375	55R_c	58R_c
Pinion	210	245	265	295	310	325	340	375	390	415	55R_c	58R_c

*Maximum hardness is usually 35 to 40 Bhn higher.

Table 5 – Recommended Safety Factors in Pitting

Required Reliability	Factor C_R
High	1.25+
Fewer than 1 failure in 100	1.00
Fewer than 1 failure in 3	0.80*

**At this value, plastic profile deformation might occur before pitting.*

Table 6 – Allowable Stress (Strength)

Material	Minimum Hardness	Allowable Stress, s_{at} (1000 psi) Spur, Helical, & Herringbone	Bevel**
Steel			
Normalized	140 Bhn	19-25	11
Quenched & temp.	180 Bhn	25-33	14
Quenched & temp.	300 Bhn	36-47	19
Quenched & temp.	450 Bhn	44-59	25
Case carb.	55R_c	55-65	27.5
Case carb.	60R_c	60-70	30
Ind. or flame through-hardened	54 Bhn	45-55*	
Ind. or flame through-hardened	54 Bhn	22	13.5
Nitrided AISI 4140	53R_c†	37-42*	20
Cast Iron			
AGMA Grade 20	......	5	2.7
AGMA Grade 30	175 Bhn	8.5	4.6
AGMA Grade 40	200 Bhn	13	7
Nodular Iron			
ASTM Grade 60-40-18	Annealed	15	8
ASTM Grade 80-55-06		20	11
ASTM Grade 100-70-03	Normalized	26	14
ASTM Grade 120-90-02	Quenched & temp.	30	18.5

Table 7 – Safety Factors for Fatigue and Yield Strength

Reliability	Factor K_R
Fatigue	
High	1.50+
Fewer than 1 failure in 100	1.00
Fewer than 1 failure in 3	0.70
Yield	
High	3.00+
Normal	1.33

for gears with smooth profiles operating with adequate lubrication. Some "as-hobbed" or shaped gears, or those with rough-textured profiles, should use more conservative stresses. In this case, special attention should be given to the lubricant. Procedures for inducing favorable surface stresses for increased profile durability are not yet generally used.

Except for heavily loaded gears or those subjected to unusual environments, no substantial difference in allowable contact stress exists among the normal qualities of available commercial steels. Free-machining steels may be used to obtain improved machinability or a better surface finish.

Because pitting is a fatigue phenomenon, it displays a scatter which must be allowed for by a safety factor to ensure reliability. If this factor is not included in the basic design calculations, Table 5 can be used as a guide and the s_{ac} values adjusted accordingly. It should be remembered that "failure" does not necessarily mean an immediate failure under applied load, but rather shorter life than expected.

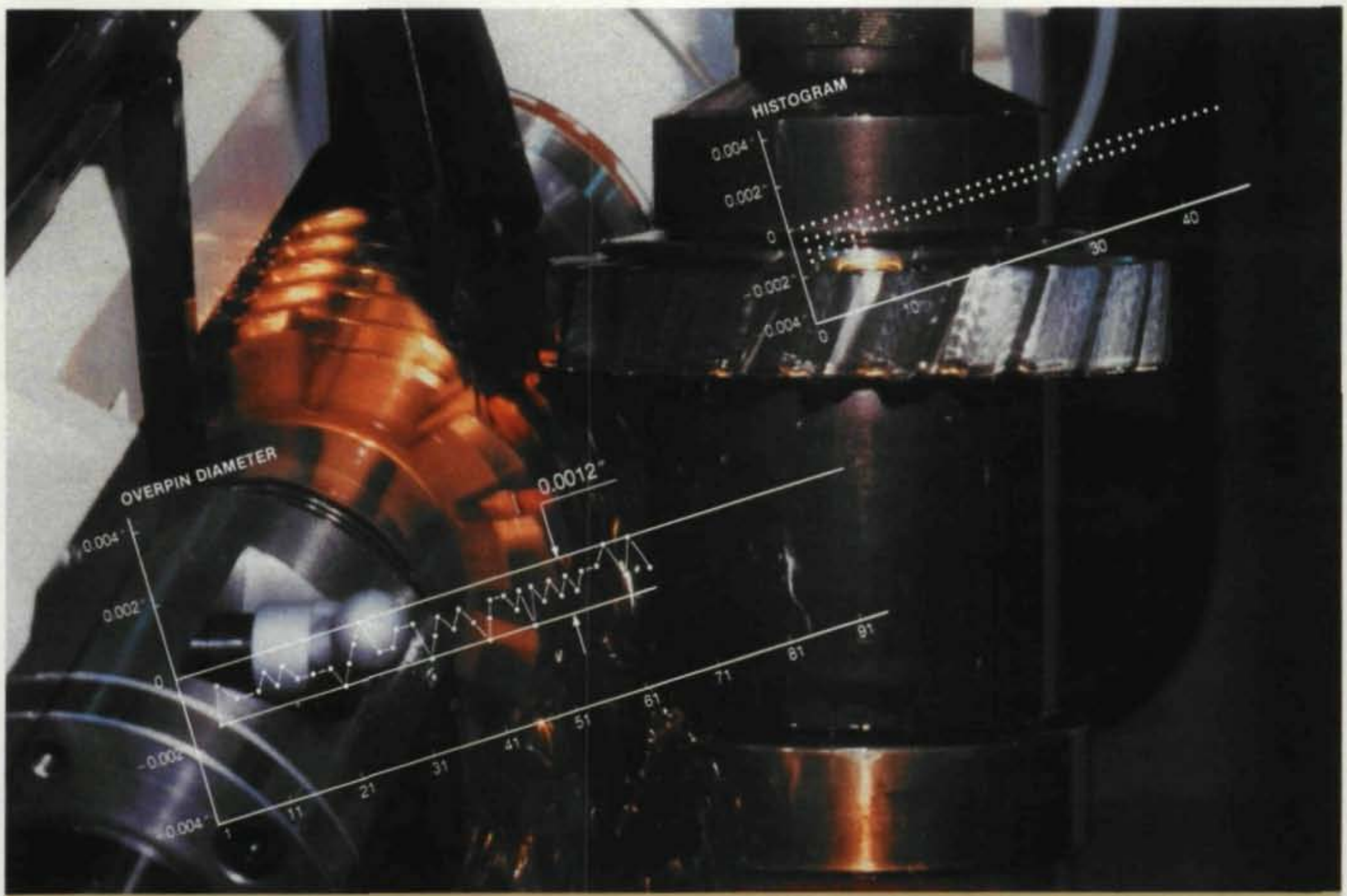
If one of the mating gear elements is considerably harder than the other, the allowable stress of the softer element can be increased under certain conditions. Normally, the gear ratio must be high – over 8:1 – and the gears large before any appreciable improvement is obtained. Typical of gears for which such a correction is normally made are those for large kilns, or for ball mills. The appropriate AGMA standards contain recommended values of the C_H factor used to rate gearsets having large differences in hardness.

Generally, temperature factor $C_T = 1$ when gears operate with oil or with gear-blank temperatures not exceeding 250F. In some instances, it is necessary to use $C_T > 1$ for carburized gears operating at oil temperatures above 180F. Tests have indicated a drop of several points R_c hardness for carburized steels subjected to 200F for 10,000 hr, as well as a 6 to 8% reduction in fatigue strength.

When gears are proportioned on the basis of allowable surface fatigue stress, surface yielding is seldom a problem. This is partially because the contact stress only increases as the square root of the transmitted load. Allowable overloads of 100% above the surface-fatigue rating are commonly specified. Greater amounts of overload are often successfully carried. However, repeated overloading can cause plastic flow, which ripples or grooves the profile or extrudes a "wire edge" at the tip of the tooth. This extruded material can affect lubrication. Excessive plastic flow of the profile can induce abrasive wear because of the rough surface texture developed.

Surface yielding does not cause immediate or catastrophic failure and the use of heavier viscosities or extreme-pressure lubricants along with reduced loading can alleviate a troublesome situation when it is encountered.

The effect of impact or brittle-fracture properties of gear materials on surface stresses need not be considered, except for the peak stresses developed by im-



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pulse or impact loading.

Tooth Strength

The tooth-strength rating formula—*AGMA 225.01, Dec. 1967, "Information Sheet for Strength of Spur, Helical, Herringbone and Bevel Gear Teeth"*—is

$$s_t = \underbrace{\left(\frac{W_t K_o}{K_v}\right)}_{\text{(Load)}} \underbrace{\left(\frac{P_d}{F}\right)}_{\text{(Size)}} \underbrace{\left(\frac{K_s K_m}{J}\right)}_{\text{(Stress dist.)}} \qquad (3)$$

where

$$s_t \leq s_{at} \underbrace{\left(\frac{K_L}{K_R K_T}\right)}_{\text{(Stress)}} \qquad (4)$$

Again, allowable stress is the most important term in selecting a material.

The fatigue and yield resistance necessary to prevent the fracture of a gear tooth depend on more complex relationships between materials properties than those for profile durability. This is true primarily because the root-radius stress varies directly with the load. Fortunately, the root stress can be calculated fairly accurately by applying standard beam or plate theories.

Some allowable fatigue stresses s_{at} used with the AGMA formula are listed in Table 6.

Allowable stress or fatigue strength is normally determined for 10^7 cycles, with adjustments for shorter finite life made by using the K_L factor given in Fig. 2. Unlike C_L, K_L depends on fatigue notch-sensitivity, which is somewhat proportional to hardness; hence the necessity for several curves for various gear hardnesses.

Experience suggests that a clearly defined knee is not always present in the gear fatigue stress/cycle plot. This phenomenon might be an inherent property, as it is with non-ferrous materials. But it is more likely a result of wear or other service-developed changes which affect either the dynamic loading, load distribution, or stress system. For conservative design, sometimes, s_{at} is reduced by the K_L factor determined from curve B in Fig. 2.

Normally the stress at the root of the tooth varies from zero to the maximum working tensile stress. If a fully reversed stress is present, as it is with reversing loads or with idler gears, the allowable stress should be 70% of the values shown in Table 6.

The quality of the material has a pronounced effect on the strength of gear teeth, more so than in the case of pitting resistance. The values shown in Table 6 are for commercially available steels. In the absence of a clear understanding of the criteria for judging quality, the lower values should be used. These are suitable for steels with a cleanliness typical of resulphurized or leaded steels.

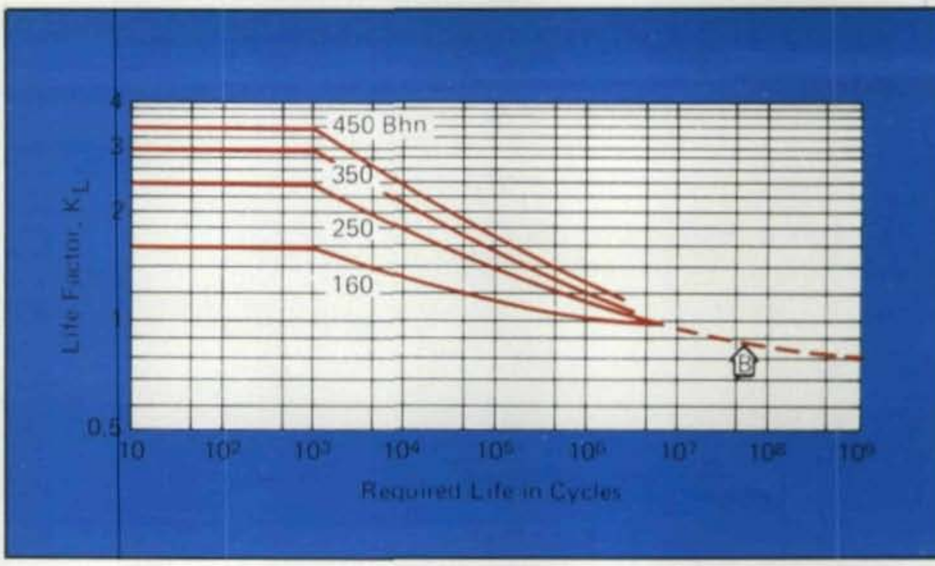

Fig. 2—Life factor for strength rating of gears.

The upper values in Table 6 would require steels produced under good melting and pouring conditions. Cast steels require adequate directional and progressive solidification. Rolled and forged steels will probably require vacuum treatment, with special instructions to prevent excessive reduction during rolling or forging; this procedure will provide a desirable direction of fiber flow lines without excessive reduction in transverse properties. Finally, heat treatment should be under rigid supervision in accurately controlled furnaces, followed by careful hardness and metallographic inspection. For high-speed gears, or for drives requiring maximum reliability, additional inspection, for example magnetic and ultrasonic inspection, is a necessity.

Fatigue: The fatigue strength of a gear tooth is significantly affected by size, surface finish, and residual stresses at the root radii. Residual stresses in a favorable direction can be developed by metallurgical processing such as case carburizing or nitriding. Under certain circumstances, flame or induction hardening, which do not develop a uniform hardness pattern on the profile or in the root area, can induce significant unfavorable stress patterns. Mechanical processing, such as shot peening, to induce favorable root stresses is used successfully.

The fatigue strength of gear teeth follows a statistical pattern, so that a safety factor should be used. Some recommended factors are listed in Table 7.

It is important to remember that a tooth fracture, unlike profile pitting, is catastrophic and cannot be repaired or alleviated by reducing load or changing lubricant. Impact loads in geared systems are commonly two or more times the rated load. Because gears have backlash and are loosely coupled, both the electrical torque and inertial energy of an electric motor can be released simultaneously, so that peak torques of two to four times the nameplate rating are not uncommon. For this reason an adequate safety factor, K_R, and life factor, K_L, are important.

Yield: Loads which develop a root stress above the yield strength of the material cause the gear tooth to bend permanently. Since gear-tooth dimensions are held to 0.001 or less, only a slight permanent bend or distortion can affect the geometric dimensions sufficient to cause interference on high dynamic loads. This will quickly result in a fatigue fracture.

The allowable yield stress for steel recommended for use with the AGMA rating formula is shown in Fig. 3. The alloy composition, heat-treatment effectiveness, and section size affect the yield strength at any ultimate strength or hardness, so that more accurate yield-strength

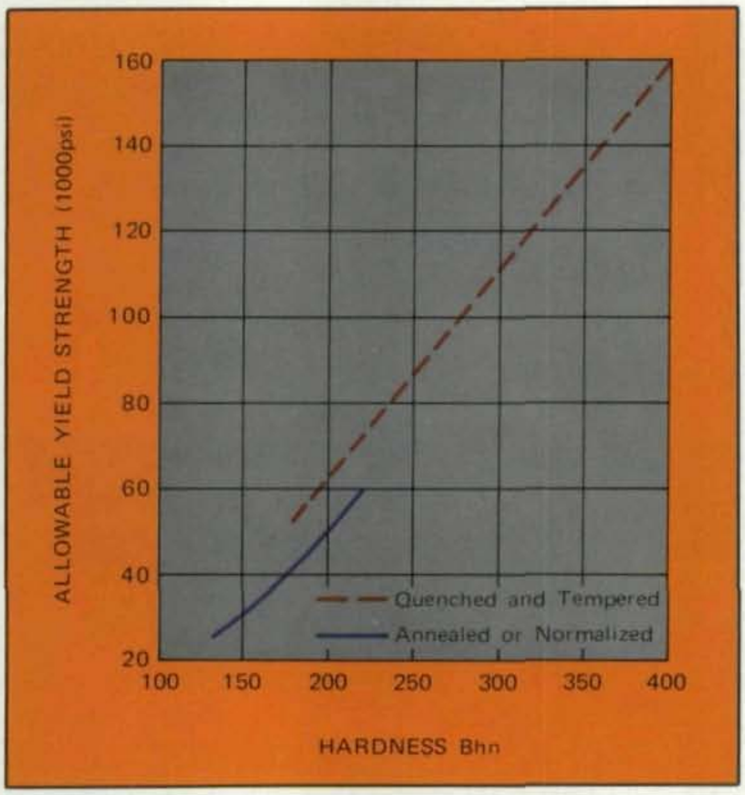

Fig. 3 – Allowable yield stress recommended for use with AGMA formulas.

data can be used if it is available.

As a general rule, when peak loads exceed 200% of the allowable endurance stress (100% overload), it is necessary to consider yield in both the design calculations and material selection. Table 7 lists appropriate safety factors for use when selecting materials for yield strength.

Operating temperatures can affect the allowable stress. According to AGMA Standards, when gears operate at oil or gear-blank temperatures not exceeding 250F temperature factor, generally $K_T = 1$. For case-carburized gears at temperatures above 160F, K_T may be found from

$$K_T = \frac{460 + T_F}{620} \tag{5}$$

where T_F = peak operating oil temperature, deg F.

Fatigue failure initiated by a bent tooth caused by loads above the yield strength is not uncommon and should not be disregarded even for surface-hardened gears. This is particularly true when the design is based on allowable fatigue stresses which have been increased by favorable induced surface stresses.

Impact: Although impact is not significant in evaluating a material for surface durability, it can be important in tooth-strength considerations.

The first step is to determine whether or not destructive impact can be developed in the gear system. Normally, if the time to develop the peak stress is a significant proportion of the natural period of vibration, the stress can be calculated from loads deduced from the oscillatory characteristics of the mass elastic system. The material must then be selected on the basis of yield strength or, if sufficient cycles are involved, fatigue strength.

In some specific applications – mostly determined by field experience – impact properties should be considered as a matter of course. One example is service at low temperatures, which are known to reduce impact strengths. Here steels containing nickel and those having fully quenched and tempered metallographic structures are desirable. Lower hardness and sometimes lower carbon content are beneficial, although these must be considered with an eye to the required size of gearing and costs. Obviously the direction of grain flow, transverse properties, and area reduction during forging must be controlled.

Brittle fractures of gear teeth occur only occasionally. Efforts to correlate material properties with brittle-fracture analysis have not yet produced results of value to the designer, although some progress is being made.

Wear

The wear in contacting teeth varies from an unmeasurable amount with fully hydrodynamic lubrication, through intermediate and boundary stages, to dry metal-to-metal contact. Hydrodynamic lubrication is present when the ratio of film thickness to surface finish exceeds approximately 1.4; boundary lubrication when the ratio is between 0.05 to 0.2; and dry lubrication when the ratio is less than 0.01. These are approximate guides based on tests with straight mineral oils.

Normally, wear is encountered only when surface finishes are rough, speeds are very low, or loads and velocities are high. There are four types of wear: adhesive, abrasive, corrosive, and fatigue.

Generally, when ferrous metals are used for gears, the wear rate decreases with increasing hardness. Under certain conditions, softer gears will polish to a fine surface finish and a good load distribution; this provides improved lubrication. In some cases, the profile will be worn to a non-involute shape; once this "wearing-in" has occurred, little wear will follow. If this beneficial wear does not occur, high hardness, obtained by quenching or by a surface-hardening source such as induction or flame hardening, carburizing, or some form of nitriding must be used.

It is not uncommon for changes to occur in the dedendum of heavily loaded gears, particularly at slow speeds. A concave surface is developed which soon stabilizes and in no way affects the load-carrying capability of the gears. This phenomenon is due to a number of reasons, one of them being the fact that the rolling and sliding are in opposite directions on the dedendum.

In some carburized and hardened gears a frosted appearance develops. This ultimately results in surface spalling, which in turn can initiate a tooth fracture. This frosting is probably due to microscopic pitting. Better surface finish

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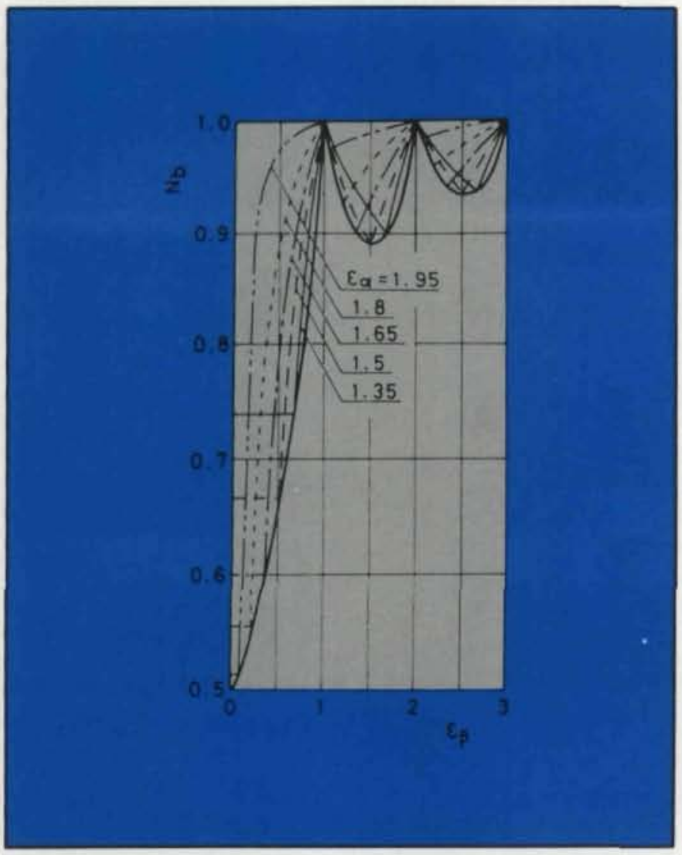

Fig. A.1 – Value of N_b

where frc (X) denotes the fraction of X. The function N_b is shown in Fig. A. 1. The dimensionless value A in equation(5) can be calculated by using N_b

$$A = \frac{L_{\min}\cos\beta_b}{b} = N_b\epsilon_\alpha \tag{A.3}$$

In the case that the face contact ratio m_F ($=\epsilon_\beta$) in AGMA 218.01 is greater than unity, the load sharing ratio m_N is defined by $m_N = F / L_{\min}$. Therefore, A is expressed as follows:

$$A = \frac{\cos\beta_b}{m_N} \tag{A.4}$$

Appendix 2

The matrix $[H_k]$ for gear $k(k = 1, 2)$ is defined by $w_{k,ij}$, which is the deflection at node j on the contact line due to a unit normal load applied to node i.

$$[H_k] = [\{W_{k,1}\}, \{W_{k,2}\}, \ldots, \{W_{k,i}\}, \ldots]$$

$$\{W_{k,i}\} = (w_{k,i1}, w_{k,i2}, \ldots, w_{k,ij}, \ldots)^T \tag{A.5}$$

$(\quad)^T$ = transposed matrix

The deflection $w_{k,ij}$ is calculated by FEM. When a pair of teeth are in mesh, the distributed load $\{P\}$ along the contact line is related to the sum of the deflection of the teeth and the relative approach due to elastic contact.

$$[H]\,\{P\} = \{w\} \tag{A.6}$$

The elements of matrices $[H]$ and $\{w\}$ are

$$H_{ij} = H_{1,ij} + H_{2,ij} + \delta_{ij}\frac{w_{P,i}}{P_i}$$
$$w_i = w_{1,i} + w_{2,i} + w_{P,i} \tag{A.7}$$

where $w_{P,i}$ is the relative approach at node i and σ_{ij} is Kronecker's delta. When some pairs of teeth I, II, . . . are in mesh matrices $[H_I]$, $[H_{II}]$, . . . are separately obtained. If the load on a pair of teeth is assumed to have little effect on the deflection of other pair of teeth, the matrix $[H]$ in equation (A.6) is diagonally constructed as follows:

$$[H] = \begin{pmatrix} [H_I] & 0 & \\ 0 & [H_{II}] & \\ & & \ldots \end{pmatrix} \tag{A.8}$$

The equation (A.6) is solved under the following conditions:

$$\Sigma P_i = P_n$$
$$w_i + \frac{s_i}{1000} = (r_{b1}\theta_1 + r_{b2}\theta_2)\cos\beta_b \qquad \text{(node in contact)} \tag{A.9}$$
$$P_i = 0 \qquad \text{(node not in contact)}$$

where s_i $[\mu m]$ is the spacing at node i caused by the effective alignment error, r_b is the radius of base cylinder and θ (rad) is the rotating angle of gear.

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MATERIAL SELECTION . . .

(continued from page 46)

and accuracy, and improved lubrication – rather than changes in material – are required to solve this problem.

Scoring

In some heavily loaded or high-speed gearing, scoring may occur under boundary film conditions. This is believed to be caused by frictional heat which reduces the lubricant protection sufficiently to allow welding and tearing of the profile.

Materials selection alone will not prevent scoring; proper lubricants and design geometry are required. This difficulty is seldom encountered in the conventional industrial gear drive. *AGMA 217.01, Oct. 1967, "AGMA Information Sheet – Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears"* provides helpful recommendations for avoiding scoring.

(This article will be continued in the September/October 1985 issue of GEAR TECHNOLOGY.)

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