ERRATA January 1996

ANSI/AGMA 2101–C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*

The following editorial correction has been made to ANSI/AGMA 2101–C95 *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.*

This change, discovered after publication, has not been made in the printing of this document. The change is shown below.

Users of ANSI/AGMA 2101–C95 are encouraged to cut out this sticker and insert it in the Standard. The equation can be placed over the existing equations.

Page 10:

$$P_{az} = \frac{\omega_1 b}{1.91 \times 10^7} \frac{Z_I}{K_o K_v K_s K_H Z_R} \left(\frac{d_{w1}}{Z_E} \frac{\sigma_{HP} Z_N Z_W}{S_H \; Y_\theta Y_Z} \right)^2 \quad \text{...(5)}$$

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ANSI/AGMA 2101–C95
Metric Edition of ANSI/AGMA 2001–C95

AMERICAN NATIONAL STANDARD

Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth

ANSI/AGMA 2101–C95

AGMA STANDARD

Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth

ANSI/AGMA 2101-C95, Metric Edition of ANSI/AGMA 2001-C95
(Revision of ANSI/AGMA 2101-B88)

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Approved January 12, 1995
American National Standards Institute, Inc.

Abstract

This standard specifies a method for rating the pitting resistance and bending strength of spur and helical involute gear pairs. A detailed discussion of factors influencing gear survival and calculation methods are provided.

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Foreword

[The foreword, footnotes, and appendices in this document are provided for informational purposes only and are not to be construed to be a part of AGMA Standard 2101–C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.*]

This standard presents general formulas for rating the pitting resistance and bending strength of spur and helical involute gear teeth using ISO symbology and SI units, and supersedes AGMA 2001–B88.

The purpose of this standard is to establish a common base for rating various types of gears for differing applications, and to encourage the maximum practical degree of uniformity and consistency between rating practices within the gear industry. It provides the basis from which more detailed AGMA application standards are developed, and provides a basis for calculation of approximate ratings in the absence of such standards.

The formulas presented in this standard contain factors whose values vary significantly depending on application, system effects, gear accuracy, manufacturing practice, and definition of gear failure. Proper evaluation of these factors is essential for realistic ratings. This standard is intended for use by the experienced gear designer capable of selecting reasonable values for rating factors and aware of the performance of similar designs through test results or operating experience.

In AGMA 218.01 the values for Life Factor, Z_N and Y_N, Dynamic Factor, K_v, and Load Distribution Factor, K_H, were revised. Values for factors assigned in standards prior to that were not applicable to 218.01 nor were the values assigned in 218.01 applicable to previous standards.

The detailed information on the Geometry Factors, Z_I and Y_J, were removed from ANSI/AGMA 2001–B88, the revision of AGMA 218.01. This material was amplified and moved to AGMA 908–B89, *Geometry Factors for Determining the Pitting Resistance and Bending Strength for Spur, Helical and Herringbone Gear Teeth.* The values of Z_I and Z_J have not been changed from previous Standards.

In ANSI/AGMA 2001–B88 the Allowable Stress Number section was expanded. Metallurgical quality factors for steel materials were defined, establishing minimum quality control requirements and allowable stress numbers for various steel quality grades. Additional higher allowable stress numbers for carburized gears were added when made with high quality steel. A new rim thickness factor, K_B, was introduced to reduce allowable bending loads on gears with thin rims. Material on scuffing (scoring) resistance was added as an annex. ANSI/AGMA 2001–B88 was first drafted in January, 1986, approved by the AGMA Membership in May 1988, and approved as an American National Standard on September 30, 1988.

This standard, ANSI/AGMA 2101–C95, is a revision of the rating method described in its superseded publications. The changes include: the Miner's rule annex was removed; the analytical method for load distribution factors, K_H, was revised and placed in an annex; nitrided allowable stress numbers were expanded to cover three grades; nitrided stress cycle factors were introduced; through hardened allowable stresses were revised; application factor was replaced by overload factor; safety factors S_H and S_F were introduced; and, life factor was replaced by stress cycle factor and its use with service factor redefined.

Caution: The dynamic factor has been redefined as the reciprocal of that used in previous AGMA standards and is relocated to the denominator of the power equation.

This AGMA Standard and related publications are based on typical or average data, conditions, or applications. The Association intends to continue working to update this Standard and to incorporate in future revisions the latest acceptable technology from domestic and international sources.

The first draft of this revision was made in February, 1992. This version was approved by the AGMA Membership in July, 1994. It was approved as an American National Standard on January 12, 1995.

Suggestions for the improvement of this Standard will be welcome. They should be sent to the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314.

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American National Standard –

Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth

1 Scope

1.1 Rating formulas

This standard provides a method by which different gear designs can be theoretically rated and compared. It is not intended to assure the performance of assembled gear drive systems.

These fundamental rating formulas are applicable for rating the pitting resistance and bending strength of internal and external spur and helical involute gear teeth operating on parallel axes. The formulas evaluate gear tooth capacity as influenced by the major factors which affect gear tooth pitting and gear tooth fracture at the fillet radius.

The knowledge and judgment required to evaluate the various rating factors come from years of accumulated experience in designing, manufacturing, and operating gear units. Empirical factors given in this standard are general in nature. AGMA application standards may use other empirical factors that are more closely suited to the particular field of application. This standard is intended for use by the experienced gear designer, capable of selecting reasonable values for the factors. It is not intended for use by the engineering public at large.

1.2 Exceptions

The formulas of this standard are not applicable to other types of gear tooth deterioration such as plastic yielding, wear, case crushing and welding. They are also not applicable when vibratory conditions exceed the limits specified for the normal operation of the gears (see ANSI/AGMA 6000–A88, *Specification for Measurement of Lateral Vibration on Gear Units*).

The formulas of this standard are not applicable when any of the following conditions exist:

- Damaged gear teeth.
- Spur gears with transverse contact ratio, ε_a, less than 1.0.
- Spur or helical gears with transverse contact ratio, ε_a, greater than 2.0.
- Interference exists between tips of teeth and root fillets.
- Teeth are pointed.
- Backlash is zero.
- Undercut exists in an area above the theoretical start of active profile. The effect of this undercut is to move the highest point of single tooth contact, negating the assumption of this calculation method. However, the reduction in tooth root thickness due to protuberance below the active profile is handled correctly by this method.
- The root profiles are stepped or irregular. The Y_J factor calculation uses the stress correction factors developed by Dolan and Broghamer [19]. These factors may not be valid for root forms which are not smooth curves. For root profiles which are stepped or irregular, other stress correction factors may be more appropriate.
- Where root fillets of the gear teeth are produced by a process other than generating.
- The helix angle at the standard (reference) diameter* is greater than 50 degrees.

Scuffing criteria are not included in the body of the standard, but a method to evaluate scuffing risk is included as annex A. This information is provided for evaluation by users of this standard, with the intent to include a scuffing evaluation method in a future version of this standard.

Design considerations to prevent fractures emanating from stress risers on the tooth profile, tip

[ ] Numbers in brackets refer to the reference number listed in annex G, bibliography.
* Refer to ANSI/AGMA 1012–F90 for further discussion of standard (reference) diameters.

chipping, and failures of the gear blank through the web or hub should be analyzed by general machine design methods.

1.3 References

The following standards contain provisions which, through reference in this text, constitute provisions of this American National Standard. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this American National Standard are encouraged to investigate the possibility of applying the most recent editions of the standards indicated:

AGMA 246.02A, *Recommended Procedure for Carburized Aerospace Gearing.*

AGMA 427.01, *Information Sheet – Systems Considerations for Critical Service Gear Drives.*

AGMA 908–B89, *Information Sheet – Geometry Factors for Determining the Pitting Resistance and Bending Strength for Spur, Helical and Herringbone Gear Teeth.*

AMS 2300G, *Cleanliness, Premium Aircraft–Quality, Magnetic Particle Inspection Procedure.*

AMS 2301G, *Cleanliness, Aircraft–Quality Steel Magnetic Particle Inspection Procedure.*

ANSI/AGMA 1012–F90, *Gear Nomenclature, Definitions of Terms with Symbols.*

ANSI/AGMA 2000–A88, *Gear Classification and Inspection Handbook – Tolerances and Measuring Methods for Unassembled Spur and Helical Gears (Including Metric Equivalents).*

ANSI/AGMA 2004–B89, *Gear Materials and Heat Treatment Manual.*

ANSI/AGMA 2007–B92, *Surface Temper Etch Inspection After Grinding.*

ANSI/AGMA 6000–A88, *Specification for Measurement of Lateral Vibration on Gear Units.*

ANSI/AGMA 6033–A88, *Standard for Marine Propulsion Gear Units, Part 1, Materials.*

ANSI/AGMA 9005–D94, *Industrial Gear Lubrication.*

ASTM A48–93a, *Specification for Gray Iron Castings.*

ASTM A388–91, *Practice for Ultrasonic Examination of Heavy Steel Forgings.*

ASTM A534–90, *Specification for Carburizing Steels for Anti–friction Bearings.*

ASTM A535–85 (1992), *Specification for Special Quality Ball and Roller Bearing Steel.*

ASTM A536–84 (1993), *Specification for Ductile Iron Castings.*

ASTM A609–91, *Practice for Castings, Carbon, Low Alloy and Martensitic Stainless Steel, Ultrasonic Examination Thereof.*

ASTM A866–92, *Specification for Medium Carbon Anti–friction Bearing Steel.*

ASTM B148–93, *Specification for Aluminum Bronze Castings.*

ASTM E112–88, *Test Methods for Determining the Average Grain Size.*

ASTM E428–92, *Practice for Fabrication and Control of Steel Reference Blocks used in Ultrasonic Inspection.*

ASTM E709–91, *Practice for Magnetic Particle Examination.*

2 Definitions and symbols

2.1 Definitions

The terms used, wherever applicable, conform to ANSI/AGMA 1012–F90 and reference [2].

2.2 Symbols

The symbols used in the pitting resistance and bending strength formulas are shown in table 1.

NOTE: The symbols and definitions used in this standard may differ from other AGMA standards. The user should not assume that familiar symbols can be used without a careful study of these definitions.

Table 1 – Symbols used in gear rating equations

Symbol	Description	Units	First Used	Ref. Clause
a	Operating center distance	mm	Eq 2	5.1.1
b	Net face width of narrowest member	mm	Eq 1	5.1.1
C_G	Gear ratio factor	—	Eq 6	5.1.4
C_{SF}	Service factor for pitting resistance	—	Eq 29	10
d_{w1}	Operating pitch diameter of pinion	mm	Eq 1	5.1.1
E_1	Modulus of elasticity for pinion	N/mm^2	Eq 30	12
E_2	Modulus of elasticity for gear	N/mm^2	Eq 30	12
F_d	Incremental dynamic tooth load	N	Eq 20	8.1
F_{max}	Maximum peak tangential load	N	Eq 45	16
F_t	Transmitted tangential load	N	Eq 18	7.1
H_{B1}	Brinell hardness of pinion	HB	Eq 32	14.1
H_{B2}	Brinell hardness of gear	HB	Eq 32	14.1
h_{cmin}	Minimum total case depth for external nitrided gear teeth	mm	Eq 44	16.1
h_{emax}	Maximum effective case depth	mm	Eq 43	16.2
h_{emin}	Minimum effective case depth for external carburized and induction hardened gear teeth	mm	Eq 42	16.1
h_t	Gear tooth whole depth	mm	Eq 17	5.2.5
K	Contact load factor for pitting resistance	N/mm^2	Eq 6	5.1.4
K_{az}	Allowable contact load factor	N/mm^2	Eq 9	5.1.4
K_B	Rim thickness factor	—	Eq 10	5.2.5
K_f	Stress correction factor	—	Eq 45	16.4
K_H	Load distribution factor	—	Eq 1	15.1
K_{He}	Mesh alignment correction factor	—	Eq 37	15.3
K_{Hma}	Mesh alignment factor	—	Eq 37	15.3
K_{Hmc}	Lead correction factor	—	Eq 37	15.3
K_{Hpf}	Pinion proportion factor	—	Eq 37	15.3
K_{Hpm}	Pinion proportion modifier	—	Eq 37	15.3
K_{Hs}	Load distribution factor under overload conditions	—	Eq 45	16.4
$K_{H\alpha}$	Transverse load distribution factor	—	Eq 35	15.2
$K_{H\beta}$	Face load distribution factor	—	Eq 35	15.3
K_o	Overload factor	—	Eq 1	9
K_s	Size factor	—	Eq 1	20
K_{SF}	Service factor for bending strength	—	Eq 29	10.
K_v	Dynamic factor	—	Eq 1	5.2.1
K_y	Yield strength factor	—	Eq 45	16.5
L	Life	hours	Eq 47	17.1
m_B	Back–up ratio	—	Eq 17	5.2.5
m_t	Transverse metric module	mm	Eq 10	5.2.1
m_n	Normal metric module, nominal	mm	Eq 11	5.2.1
n_L	Number of load cycles	—	Fig 17	17
P	Transmitted power	kW	Eq 18	7.1

(continued)

Table 1 *(continued)*

Symbol	Description	Units	First Used	Ref. Clause
P_a	Allowable transmitted power for gear set	kW	Eq 29	10
P_{ay}	Allowable transmitted power for bending strength	kW	Eq 14	5.2.3
P_{ayu}	Allowable transmitted power for bending strength at unity service factor	kW	Eq 28	10
P_{az}	Allowable transmitted power for pitting resistance	kW	Eq 5	5.1.3
P_{azu}	Allowable transmitted power for pitting resistance at unity service factor	kW	Eq 27	10
p_x	Axial pitch	mm	Eq 11	5.2.1
Q_v	Transmission accuracy level number	—	Eq 21	8.3.2
q	Number of contacts per revolution	—	Eq 47	17.1
R_{z1}	Pinion surface finish	µm	Eq 34	14.2
S	Bearing span	mm	Fig 6	15.3
S_1	Pinion offset	mm	Fig 6	15.3
S_F	Safety factor – bending	—	Eq 13	11
S_H	Safety factor – pitting	—	Eq 4	11
s_{an}	Normal tooth thickness at the top land of gear	mm	Eq 43	16.1
T	Transmitted pinion torque	Nm	Eq 18	7.1
t_R	Gear rim thickness	mm	Eq 17	5.2.5
U_{ay}	Allowable unit load for bending strength	N/mm^2	Eq 16	5.2.4
U_c	Core hardness coefficient	—	Eq 44	16.1
U_H	Hardening process factor	—	Eq 42	16.1
U_L	Unit load for bending strength	N/mm^2	Eq 15	5.2.4
u	Gear ratio (never less than 1.0)	—	Eq 2	5.1.1
V_{pA}	Absolute value of pitch variation	µm	Eq 21	8.3.2
v_t	Pitch line velocity at operating pitch diameter	m/s	Eq 18	7.1
$v_{t\max}$	Pitch line velocity maximum at operating pitch diameter	m/s	Eq 26	8.3.2
ν_1	Poisson's ratio for pinion	—	Eq 30	12
ν_2	Poisson's ratio for gear	—	Eq 30	12
Y_J	Geometry factor for bending strength	—	Eq 10	6.2
Y_N	Stress cycle life factor for bending strength	—	Eq 13	17
Y_Z	Reliability factor	—	Eq 4	18
Y_θ	Temperature factor	—	Eq 4	19
z_i	Adjusted number of pinion or gear teeth	—	Eq 21	8.3.2
z_1	Number of teeth in pinion	—	Eq 7	5.1.4
z_2	Number of teeth in gear	—	Eq 7	5.1.4
Z_I	Geometry factor for pitting resistance	—	Eq 1	6.1
Z_E	Elastic coefficient	[N/mm^2]$^{0.5}$	Eq 1	12.
Z_N	Stress cycle life factor for pitting resistance	—	Eq 4	17.
Z_R	Surface condition factor for pitting resistance	—	Eq 1	13.
Z_W	Hardness ratio factor for pitting resistance	—	Eq 4	17.
α_{pt}	Operating transverse pressure angle	—	Eq 42	16.1
β	Helix angle at standard pitch diameter	—	Eq 11	5.2.1

(continued)

Table 1 *(concluded)*

Symbol	Description	Units	First Used	Ref. Clause
β_{mb}	Base helix angle	—	Eq 42	16.1
σ_F	Bending stress number	N/mm^2	Eq 10	5.2.1
σ_H	Contact stress number	N/mm^2	Eq 1	5.1.1
σ_{FP}	Allowable bending stress number	N/mm^2	Eq 13	5.2.2
σ_{HP}	Allowable contact stress number	N/mm^2	Eq 4	5.1.2
σ_s	Allowable yield stress number	N/mm^2	Eq 45	16.4
ω	Speed	rpm	Eq 47	17.1
ω_1	Pinion speed	rpm	Eq 5	5.1.3

3 Application

3.1 Rating practices

Pitting resistance and bending strength rating practices for a particular field of gearing may be established by selecting proper values for the factors used in the general formulas of clause 5.

Where applicable AGMA application standards exist, they should be used in preference to this standard. Consult AGMA Headquarters for current list of applicable standards. Where no applicable AGMA application standard exists, numerical values may be estimated for the factors in the general formulas, and the approximate pitting resistance and bending strength ratings calculated.

3.2 Implied accuracy

Where empirical values for rating factors are given by curves, curve fitting equations are provided to facilitate computer programming. The constants and coefficients used in curve fitting often have significant digits in excess of those inferred by the reliability of the empirical data. Experimental data from actual gear unit measurements are seldom repeatable within a plus or minus 10 percent band. Calculated gear ratings are intended to be conservative, but the scatter in actual results may exceed 20 percent.

3.3 Testing

The preferred method to predict overall system performance is to test a proposed new design. Where sufficient experience is available from similar designs, satisfactory results can be obtained by extrapolation of previous tests or field data.

NOTE: When suitable test results or field data are not available, values for the rating factors should be chosen conservatively.

3.4 Manufacturing quality

Rating factors should be evaluated on the basis of the expected variation of component parts in the production run. The formulas of this standard are only valid for appropriate material quality and geometric quality that conforms to the manufacturing tolerances. Defects such as surface cracks, grinding temper, or tooth root steps may invalidate calculations of pitting resistance and bending strength.

3.4.1 Geometric quality

The rating formulas of this standard are only valid if the gear tooth and gear element support accuracies assumed in the calculations are actually achieved in manufacture (see clause 8).

Gear tooth accuracy considerations include: involute profile, tooth alignment (lead), tooth spacing and tooth finish.

Gear element support considerations include: gear case bore alignment, bearing eccentricities and shaft runouts.

3.4.2 Metallurgy

The allowable stress numbers, σ_{HP} and σ_{FP}, included herein are a function of melting, casting, forging and heat treating practice. Hardness, tensile strength, microstructure and cleanliness are some criteria for determining allowable stress numbers. Allowable stress numbers in this standard are based on 10^7 cycles, 99 percent reliability and unidirectional loading.

The allowable stresses are only valid for materials and conditions listed in this standard (see clause 16). For example, materials such as aluminum or stainless steel may encounter lubrication problems that invalidate calculations of pitting resistance and bending strength.

Variations in microstructure account for some variation in gear capacity. Higher levels of cleanliness and better metallurgical control permit the use of higher allowable stress numbers. Conversely, lower metallurgical quality levels require the use of lower allowable stress numbers.

3.4.3 Residual stress

Any material having a case–core relationship is likely to have residual stresses. If properly managed, these stresses should be compressive at the surface and should enhance the bending strength performance of the gear teeth. Shot peening, case carburizing, nitriding, and induction hardening are common methods of inducing compressive pre–stress in the surface of the gear teeth.

Grinding the tooth surface after heat treatment may reduce the residual compressive stresses. Grinding the tooth surface and root fillet area may introduce tensile stresses and possibly cracks in these areas if incorrectly done. Care must be taken to avoid excessive reduction in hardness and changes in microstructure during the grinding process.

3.5 Lubrication

The ratings determined by these formulas are only valid when the gear teeth are operated with a lubricant of proper viscosity for the load, gear tooth surface finish, temperature, and pitch line velocity.

Lubricant recommendations are given in ANSI/AGMA 9005–D94, *Industrial Gear Lubrication.*

3.5.1 Oil film thickness

Field results and laboratory tests have shown that pitting resistance of gear teeth can also be affected by elastohydrodynamic (EHD) oil film thickness, see [9] and [18]. This appears to be a nonlinear relationship where a small change in film thickness in the critical range makes a large change in pitting resistance. Oil film thickness depends on viscosity, load, temperature, and pitch line velocity. Annex A provides a method to estimate EHD film thickness. This standard does not provide a method to estimate the minimum film thickness required.

Lubrication problems are not common in industrial gears in the speed range of 5 to 50 m/s, but show up from time to time in aerospace gearing and in marine gearing. This may be due to high temperatures, inadequate additive package in the oil, size of the pinion, inadequate oil viscosity, or tooth finish characteristics.

The ratings are valid only for those lubrication conditions which allow the gears to operate without experiencing appreciable wear.

3.5.2 Low operating speeds

The design of slower gears, from a lubrication standpoint, should be based on application requirements such as hours of life, degree of reliability needed, and acceptable increase in noise and vibration as the gear teeth wear or deform Field experience and test stand experience can be used to select design parameters and lubricant criteria to meet the application.

Slower speed gears, with pitch line velocities less than 0.5 m/s, require special design consideration to avoid premature failure due to inadequate lubrication.

At low surface speeds [below 0.5 m/s pitch line velocity or 20 rpm input speed] the gear designer may expect some pitting and wear to occur during the gear life when using these rating practices for other than surface hardened gearing. Methods and limits for determining acceptable wear at low speeds should be based on the field or test experience of the manufacturer. The rating of gear teeth due to wear is not covered by this standard.

Slow speed gears, with pitch line velocities greater than 0.5 m/s but less than 5 m/s frequently require special design considerations, even when the lubricants used conform to ANSI/AGMA 9005–D94 recommendations. (ANSI/AGMA 9005–D94 does not, at present, cover the complexities of elastohydrodynamic oil film thickness and its relation to load rating).

3.6 Temperature extremes

3.6.1 Cold temperature operation

When operating temperatures result in gear temperatures below 0°C, special care must be given to select materials which will have adequate impact properties at the operating temperature. Consideration should be given to:

– Low temperature Charpy specification.

– Fracture appearance transition or nil ductility temperature specification.

– Reduce carbon content to less than 0.4 percent.

– Use of higher nickel alloy steels.

– Using heating elements to increase lubricant and gear temperatures.

3.6.2 Hot temperatures

Consideration must be given to the loss of hardness and strength of some materials due to the tempering effect of gear blank temperatures over 150°C.

3.7 Oscillatory motion

The formulas in this standard are only valid for gears that rotate in one direction, or gears that reverse direction with several rotations between reversals, provided that adequate consideration is given to the dynamic loads that are developed during reversals. The formulas are not valid for applications such as robotics or yaw drives where gears are subjected to small oscillatory motion.

3.8 Non–uniform loading

Non–uniform loading may require the use of Miner's Rule for analysis (see 7.2).

3.9 Other considerations

In addition to the factors considered in this standard which influence pitting resistance and bending strength, other interrelated factors can affect overall transmission performance. The following factors are particularly significant.

3.9.1 Service damaged teeth

The formulas of this standard are only valid for undamaged gear teeth. Deterioration such as plastic deformation, pitting, micropitting, wear, or scuffing invalidate calculations of pitting resistance and bending strength.

3.9.2 Misalignment and deflection of foundations

Many gear systems depend on external supports such as machinery foundations to maintain alignment of the gear mesh. If these supports are initially misaligned, or are allowed to become misaligned during operation through elastic or thermal deflection, overall gear system performance will be adversely affected.

3.9.3 Deflection due to external loads

Deflection of gear supporting housings, shafts, and bearings due to external overhung, transverse, and thrust loads affects tooth contact across the mesh. Since deflection varies with load, it is difficult to obtain good tooth contact at different loads. Generally, deflection due to external loads reduces capacity.

3.9.4 System dynamics

The dynamic response of the system results in additional gear tooth loads due to the relative accelerations of the connected masses of the driver and the driven equipment. The overload factor, K_o, is intended to account for the operating characteristics of the driving and driven equipment. It must be recognized, however, that if the operating roughness of the driver, gearbox, or driven equipment causes an excitation with a frequency that is near to one of the system's major natural frequencies, resonant vibrations may cause severe overloads which may be several times higher than the nominal load. For critical service applications, it is recommended that a vibration analysis be performed. This analysis must include the total system of driver, gearbox, driven equipment, couplings, mounting conditions, and sources of excitation. Natural frequencies, mode shapes, and the dynamic response amplitudes should be calculated. The responsibility for the vibration analysis of the system rests with the purchaser of the gearing. For more information, refer to AGMA 427.01, *Information Sheet – Systems Considerations for Critical Service Gear Drives*.

3.9.5 Corrosion

Corrosion of the gear tooth surface can have a significant detrimental effect on the bending strength and pitting resistance of the teeth. Quantification of the effect of corrosion on gear teeth is beyond the scope of this standard.

4 Criteria for tooth capacity

4.1 Relationship of pitting resistance and bending strength ratings

There are two major differences between the pitting resistance and the bending strength ratings. Pitting is a function of the Hertzian contact (compressive) stresses between two cylinders and is proportional to the square root of the applied tooth load. Bending strength is measured in terms of the bending (tensile) stress in a cantilever plate and is directly proportional to this same load. The difference in nature of the stresses induced in the tooth surface areas and at the tooth root is reflected in a corresponding difference in allowable limits of contact and bending stress numbers for identical materials and load intensities.

The analysis of the load and stress modifying factors is similar in each case, so many of these factors have identical numerical values.

The term "gear failure" is itself subjective and a source of considerable disagreement. One observer's failure may be another observer's wearing–in. For a more complete discussion, see ANSI/AGMA 110.04, *Nomenclature of Gear Tooth Failure Modes* [3].

4.2 Pitting resistance

The pitting of gear teeth is considered to be a fatigue phenomenon. Initial pitting and progressive pitting are illustrated and discussed in ANSI/AGMA 110.04.

In most industrial practice non–progressive initial pitting is not deemed serious. Initial pitting is characterized by small pits which do not extend over the entire face width or profile height of the affected teeth. The definition of acceptable initial pitting varies widely with gear application. Initial pitting occurs in localized, overstressed areas. It tends to redistribute the load by progressively removing high contact spots. Generally, when the load has been reduced or redistributed, the pitting stops.

The aim of the pitting resistance formula is to determine a load rating at which progressive pitting of the teeth does not occur during their design life. The ratings for pitting resistance are based on the formulas developed by Hertz for contact pressure between two curved surfaces, modified for the effect of load sharing between adjacent teeth.

4.3 Surface conditions not covered by this standard

Conditions such as micropitting, electric discharge pitting, wear and scuffing are not rated by this standard but could be a problem. See ANSI/AGMA 110.04 for more information.

4.3.1 Micropitting

Micropitting is one type of gear tooth surface fatigue. It is characterized by very small pits on the surface of the material, usually less than 20 μm deep, that give the gear tooth the appearance of being frosted or grey in color. This deterioration of the surface of the material is generally thought to occur because of excessive Hertzian stresses due to influences from gear loading, material and its heat treatment, the type of lubricant, and degree of lubrication.

Micropitting is most frequently observed on surface hardened gear teeth, although it can develop on through hardened gear teeth as well. Gear sets operating at moderate pitchline velocities, 4 to 10 m/s are commonly affected, but micropitting has been seen on gear sets running at other velocities as well. Micropitting generally occurs in the dedendum of a speed reducing pinion, but it can develop anywhere along the active profile of a tooth.

4.3.2 Electric discharge pitting

Electric discharge pitting is not a gear tooth rating problem, however, it is a distressed condition of the tooth surface. To the naked eye, the tooth surface may not be distinguishable from micropitting as the gear teeth exhibit the same so–called "frosted" appearance. It is caused by either static or stray electricity conducted through the gear mesh due to inappropriate electrical grounding or inappropriate gear motor isolation. If neglected, gear failure can occur.

4.3.3 Wear capacity of gears

The wear resistance of mating gears can be a dictating performance limitation, particularly in low speed, heavily loaded gears. Gear wear is a difficult phenomenon to predict analytically.

Wear may occur when the oil film that separates the contacting surfaces of mating gear teeth is not adequate (see annex A).

Wear in low speed applications may be tolerable. Wear in high speed applications could be catastrophic where the magnitude of dynamic loading that can occur from nonconjugate gear tooth action is excessive.

4.3.4 Scuffing

Scuffing is severe adhesive wear on the flanks of gear teeth. The adhesive wear is a welding and tearing of the metal surface by the flank of the mating gear. It occurs when the oil film thickness is small enough to allow the flanks of the gear teeth to contact and slide against each other.

Scuffing is not a fatigue phenomenon and it may occur instantaneously. Annex A provides a method of evaluating the risk of a gear set scuffing. This risk is a function of oil viscosity and additives, operating bulk temperature of gear blanks, sliding velocity, surface roughness of teeth, gear materials and heat treatments, and surface pressure.

4.4 Bending strength

The bending strength of gear teeth is a fatigue phenomenon related to the resistance to cracking at the tooth root fillet in external gears and at the critical section in internal gears. Typical cracks and fractures are illustrated in ANSI/AGMA 110.04.

The basic theory employed in this analysis assumes the gear tooth to be rigidly fixed at its base. If the rim supporting the gear tooth is thin relative to the size of the tooth and the gear pitch diameter, another critical stress may occur not at the fillet but in the root area.

The rim thickness factor, K_B, adjusts the calculated bending stress number for thin rimmed gears.

The user should ensure that the gear blank construction is representative of the basic theory embodied in this standard. Gear blank design is beyond the scope of this standard (see 5.2.5).

The bending strength ratings determined by this standard are based on plate theory modified to consider:

– The compressive stress at tooth roots caused by the radial component of tooth loading.

– Non–uniform moment distribution resulting from the inclined angle of the load lines on the teeth.

– Stress concentrations at the tooth root fillets.

– The load sharing between adjacent teeth in contact.

The intent of the AGMA strength rating formula is to determine the load which can be transmitted for the design life of the gear drive without causing root fillet cracking.

Occasionally, wear, surface fatigue, or plastic flow may limit bending strength due to stress concentrations around large, sharp cornered pits or wear steps on the tooth surface.

5 Fundamental rating formulas

5.1 Pitting resistance

5.1.1 Fundamental formula

The contact stress number formula for gear teeth is:

$$\sigma_H = Z_E \sqrt{F_t\, K_o\, K_v\, K_s\, \frac{K_H}{d_{w1} b}\, \frac{Z_R}{Z_I}} \qquad ...(1)$$

where

σ_H is contact stress number, N/mm^2;

Z_E is elastic coefficient, [N/mm^2]$^{0.5}$ (see clause 12);

F_t is transmitted tangential load, N (see clause 7);

K_o is overload factor (see clause 9);

K_v is dynamic factor (see clause 8);

K_s is size factor (see clause 20);

K_H is load distribution factor (see clause15);

Z_R is surface condition factor for pitting resistance (see clause 13);

b is net face width of narrowest member, mm;

Z_I is geometry factor for pitting resistance (see clause 6);

d_{w1} is operating pitch diameter of pinion, mm.

$$d_{w1} = \frac{2a}{u+1} \text{ for external gears} \qquad ...(2)$$

$$d_{w1} = \frac{2a}{u-1} \text{ for internal gears} \qquad ...(3)$$

where

a is operating center distance, mm;

u is gear ratio (never less than 1.0).

5.1.2 Allowable contact stress number

The relation of calculated contact stress number to allowable contact stress number is:

$$\sigma_H \le \frac{\sigma_{HP}}{S_H}\frac{Z_N}{Y_\theta}\frac{Z_W}{Y_Z} \qquad \text{...(4)}$$

where

σ_{HP} is allowable contact stress number, N/mm^2 (see clause 16);

Z_N is stress cycle factor for pitting resistance (see clause 17);

Z_W is hardness ratio factor for pitting resistance (see clause 14);

S_H is safety factor for pitting (see clause 11);

Y_θ is temperature factor (see clause 19);

Y_Z is reliability factor (see clause 18).

5.1.3 Pitting resistance power rating

The pitting resistance power rating is:

$$P_{az} = \frac{\omega_1 b}{1.91\times 10^7}\frac{Z_I}{K_o K_v K_s K_H Z_R}\left(\frac{d_{w1}\,\sigma_{HP}}{Z_E S_H}\frac{Z_N Z_W}{Y_\theta Y_Z}\right)^2 \qquad \text{...(5)}$$

where

P_{az} is allowable transmitted power for pitting resistance, kW;

ω_1 is pinion speed, rpm.

CAUTION: The ratings of both pinion and gear teeth must be calculated to evaluate differences in material properties and the number of tooth contact cycles under load. The pitting resistance power rating is based on the lowest value of the product $\sigma_{HP}\, Z_N\, Z_W$ for each of the mating gears.

5.1.4 Contact load factor, K

In some industries, pitting resistance is rated in terms of K factor.

$$K = \frac{F_t}{d_{w1} b}\frac{1}{C_G} \qquad \text{...(6)}$$

where

K is contact load factor for pitting resistance, N/mm^2;

C_G is gear ratio factor.

$$C_G = \frac{u}{u+1} \text{ or } \frac{z_2}{z_2+z_1} \text{ for external gears} \qquad \text{...(7)}$$

and

$$C_G = \frac{u}{u-1} \text{ or } \frac{z_2}{z_2-z_1} \text{ for internal gears} \qquad \text{...(8)}$$

where

z_2 is number of teeth in gear;

z_1 is number of teeth in pinion.

In terms of this standard, the allowable K factor is defined as:

$$K_{az} = \frac{Z_I}{K_o K_v K_s K_H Z_R C_G}\left(\frac{\sigma_{HP}}{Z_E S_H}\frac{Z_N Z_W}{Y_\theta Y_Z}\right)^2 \qquad \text{...(9)}$$

K_{az} is allowable contact load factor, N/mm^2.

The allowable contact load factor, K_{az}, is the lowest of the ratings calculated using the different values of σ_{HP}, Z_W and Z_N for pinion and gear.

5.2 Bending strength

5.2.1 Fundamental formula

The fundamental formula for bending stress number in a gear tooth is:

$$\sigma_F = F_t\, K_o\, K_v\, K_s \frac{1}{b m_t}\frac{K_H K_B}{Y_J} \qquad \text{...(10)}$$

where

σ_F is bending stress number, N/mm^2;

K_B is rim thickness factor (see 5.2.5);

Y_J is geometry factor for bending strength (see clause 6);

m_t is transverse metric module, mm.

$$m_t = \frac{p_x \tan\beta}{\pi} = \frac{m_n}{\cos\beta} \qquad \text{...(11)}$$

where

m_n is normal metric module, mm;

p_x is axial pitch, mm;

β is helix angle at standard pitch diameter.

$$\beta = \arcsin\left(\frac{\pi\, m_n}{p_x}\right) \qquad \text{...(12)}$$

5.2.2 Allowable bending stress number

The relation of calculated bending stress number to allowable bending stress number is:

$$\sigma_F \le \frac{\sigma_{FP} Y_N}{S_F Y_\theta Y_Z} \qquad \text{...(13)}$$

where

σ_{FP} is allowable bending stress number, N/mm^2 (see clause 16);

Y_N is stress cycle factor for bending strength (see clause17);

S_F is safety factor for bending strength (see clause 11).

5.2.3 Bending strength power rating

The bending strength power rating is:

$$P_{ay} = \frac{\omega_1 d_{w1}}{1.91 \times 10^7 K_o K_v} \frac{b m_t}{K_s} \frac{Y_J}{K_H K_B} \frac{\sigma_{FP} Y_N}{S_F Y_\theta Y_Z} \quad \text{...(14)}$$

where

P_{ay} is allowable transmitted power for bending strength, kW.

CAUTION: The ratings of both pinion and gear teeth must be calculated to evaluate differences in geometry factors, number of load cycles, and material properties. The bending strength power rating is based on the lowest value of the term

$\frac{\sigma_{FP} Y_N Y_J}{K_B}$ for each of the mating gears.

5.2.4 Unit load, U_L

In some industries, bending strength is rated in terms of unit load.

$$U_L = \frac{F_t}{b m_n} \quad \text{...(15)}$$

where

U_L is unit load for bending strength, N/mm^2.

In terms of this standard the allowable unit load is defined as:

$$U_{ay} = \frac{Y_J}{\cos\beta \; K_o K_v K_s K_H K_B} \frac{\sigma_{FP} \; Y_N}{Y_\theta Y_Z S_F} \quad \text{...(16)}$$

where

U_{ay} is allowable unit load for bending strength, N/mm^2.

The allowable unit load, U_{ay}, is the lowest of the ratings calculated using the different values of σ_{FP}, K_B, Y_N and Y_J for pinion and gear.

5.2.5 Rim thickness factor, K_B

Where the rim thickness is not sufficient to provide full support for the tooth root, the location of bending fatigue failure may be through the gear rim, rather than at the root fillet. Published data [5] suggest the use of a stress modifying factor, K_B, in this case.

The rim thickness factor, K_B, is not sufficiently conservative for components with hoop stresses, notches or keyways. This data is based on external gears with smooth bores and no notches or keyways.

The rim thickness factor, K_B, adjusts the calculated bending stress number for thin rimmed gears. It is a function of the backup ratio, m_B, (see annex B).

$$m_B = \frac{t_R}{h_t} \quad \text{...(17)}$$

where

t_R is gear rim thickness below the tooth root, mm;

h_t is gear tooth whole depth, mm.

The effects of webs and stiffeners can be an improvement but are not accounted for in annex B. The effect of tapered rims has not been investigated. When previous experience or detailed analysis justifies, lower values of K_B may be used.

K_B is applied in addition to the 0.70 reverse loading factor where it is applicable (see 16.2).

6 Geometry factors, Z_I and Y_J

6.1 Pitting resistance geometry factor, Z_I

The geometry factor, Z_I, evaluates the radii of curvature of the contacting tooth profiles based on tooth geometry. These radii are used to evaluate the Hertzian contact stress in the tooth flank. Effects of modified tooth proportions and load sharing are considered.

6.2 Bending strength geometry factor, Y_J

The geometry factor, Y_J, evaluates the shape of the tooth, the position at which the most damaging load is applied, and the sharing of the load between oblique lines of contact in helical gears. Both the tangential (bending) and radial (compressive) components of the tooth load are included.

6.3 Calculation method

It is recommended that the geometry factors, Z_I and Y_J, be determined by AGMA 908-B89, *Information Sheet – Geometry Factors for Determining the Pitting Resistance and Bending Strength for Spur, Helical and Herringbone Gear Teeth.* It includes tables for some common tooth forms and the analytical method for involute gears with generated root fillets.

7 Transmitted tangential load, F_t

In most gear applications the torque is not constant. Therefore, the transmitted tangential load will vary. To obtain values of the operating tangential load, the designer should use the values of power and speed at which the driven device will perform. F_t represents the tooth load due to the driven apparatus.

Overload factor, K_o (see clause 9), and dynamic factor, K_v (see clause 8), are included in the rating formulas (see clause 5) to account for loads in excess of F_t.

7.1 Uniform load

If the rating is calculated on the basis of uniform load, the transmitted tangential load is:

$$F_t = \frac{1000P}{v_t} = \frac{2000T}{d_{w1}} = \frac{1.91 \times 10^7 P}{\omega_1 d_{w1}} \quad ...(18)$$

where

- P is transmitted power, kW;
- T is transmitted pinion torque, Nm;
- v_t is pitch line velocity at operating pitch diameter, m/s.

$$v_t = \frac{\pi\ \omega_1 d_{w1}}{60\ 000} \quad ...(19)$$

7.2 Non-uniform load

When the transmitted load is not uniform, consideration should be given not only to the peak load and its anticipated number of cycles, but also to intermediate loads and their numbers of cycles. This type of load is often considered a duty cycle and may be represented by a load spectrum. In such cases, the cumulative fatigue effect of the duty cycle is considered in rating the gear set. A method of calculating the effect of the loads under these conditions, such as Miner's Rule, is given in ISO TR 10495 [1].

8 Dynamic factor, K_v

CAUTION: Dynamic factor, K_v, has been redefined as the reciprocal of that used in previous AGMA standards. It is now greater than 1.0. In earlier AGMA standards it was less than 1.0.

8.1 Dynamic factor considerations

Dynamic factor, K_v, accounts for internally generated gear tooth loads which are induced by non-conjugate meshing action of the gear teeth. Even if the input torque and speed are constant, significant vibration of the gear masses, and therefore dynamic tooth forces, can exist. These forces result from the relative accelerations between the gears as they vibrate in response to an excitation known as "transmission error". Ideally, a gear set would have a uniform velocity ratio between the input and output rotation. Transmission error is defined as the departure from uniform relative angular motion of the pair of meshing gears. It is influenced by all the deviations from the ideal gear tooth form and ideal spacing.

The dynamic factor relates the total tooth load including internal dynamic effects to the transmitted tangential tooth load.

$$K_v = \frac{F_d + F_t}{F_t} \quad ...(20)$$

where

- F_d is incremental dynamic tooth load due to the dynamic response of the gear pair to the transmission error excitation, not including the transmitted tangential load, N.

8.1.1 Excitation

The transmission error is influenced by:

- Manufacturing variations including spacing, profile, lead, and runout.
- Gear mesh stiffness variation as the gear teeth pass through the meshing cycle. This source of excitation is especially pronounced in spur gears without profile modification. Spur gears with properly designed profile modification, and helical gears with axial contact ratios larger than 1.0 have a smaller stiffness variation.
- Transmitted load. Since elastic deflections are load dependent, gear tooth profile modifications can be designed to give a uniform velocity ratio only for one load magnitude. Loads different from the design load will give increased transmission error.
- Dynamic unbalance of the gears and shafts.
- Excessive wear and plastic deformation of the gear tooth profiles that increase the amount of transmission error.
- Shaft alignment. Gear tooth alignment is influenced by load and thermal deformations of the gears, shafts, bearings and housings, and by manufacturing variations.
- Tooth friction induced excitation.

8.1.2 Dynamic response

The dynamic tooth forces are influenced by:

- Mass of the gears, shafts, and other major internal components.
- Stiffness of the gear teeth, gear blanks, shafts, bearings, and gear housing.
- Damping. The principal source of coulomb or viscous damping is the shaft bearings. Generally oil film bearings provide greater damping than rolling element bearings. Other sources of damping include the hysteresis of the gear shafts, and viscous damping at sliding interfaces and shaft couplings.

8.2 Resonance

When an excitation frequency coincides with a natural frequency, the resonant response is limited only by the damping, and high dynamic loads may result. The dynamic factor, K_v, does not apply to resonance.

8.2.1 Gear pair resonance

If a particular frequency of the transmission error excitation is close to the natural frequency of the gear spring–mass system, or some multiple of the natural frequency such as 2 or 3, a resonant vibration may cause high dynamic tooth forces due to large relative displacements of the gear masses. The dynamic factor, K_v, does not account for gear pair resonance, and operation in this regime is to be avoided.

8.2.2 Gear blank resonance

Gear blanks may have natural frequencies within the operating speed range. If the gear blank is excited by a frequency which is close to one of its natural frequencies, the resonant deflections may cause high dynamic tooth loads. This occurs more frequently in high speed, light weight gear blanks, but can also occur in other thin rimmed or thin webbed blanks. The dynamic factor, K_v, does not account for gear blank resonance. A separate investigation is recommended when these conditions occur.

8.2.3 System resonance

The gearbox is one component of a system comprised of a power source, gearbox, driven equipment, and interconnecting shafts and couplings. The dynamic response of this system depends on the distribution of the masses, stiffness, and damping. In certain cases, a system may possess a torsional natural frequency close to an excitation frequency associated with an operating speed. Under these resonant conditions, the dynamic gear tooth loads may be very high, and operation near a system resonance is to be avoided. The dynamic factor, K_v, does not include considerations of the dynamic tooth loads due to torsional vibration of the gear system. These loads must be included with other externally applied forces in the overload factor, K_o. For critical drives, a separate dynamic analysis of the entire system is recommended.

8.2.4 Shaft critical speeds

Due to the high bending stiffness of gear shafts, the natural frequencies of lateral vibration of the gear shafts are usually much higher than the operating speeds. For high speed gears, however, it is recommended that the shaft critical speeds be analyzed to ensure that they are well removed from the operating speed range. The dynamic factor, K_v, does not account for the dynamic tooth loads due to this mode of vibration.

8.2.5 Nonlinear resonance

Large cyclical variation in gear mesh stiffness and impact loads may lead to additional regions of resonance and instability. This is primarily a problem with lightly–loaded, lightly–damped spur gears which do not have profile modifications.

8.3 Approximate dynamic factor, K_v

Figure 1 shows dynamic factors which can be used in the absence of specific knowledge of the dynamic loads. The curves of figure 1 and the equations given are based on empirical data, and do not account for resonance.

Due to the approximate nature of the empirical curves and the lack of measured tolerance values at the design stage of the job, the dynamic factor curve should be selected based on experience with the manufacturing methods and operating considerations of the design.

Choice of curves Q_v = 5 through Q_v = 11 and "very accurate gearing" should be based on transmission error. When transmission error is not available, it is reasonable to refer to the pitch accuracy, and to some extent profile accuracy, as a representative value to determine the dynamic factor. "Q_v" is related to the transmission accuracy grade number. Due to the approximation mentioned above, slight variation from the selected "Q_v" value is not considered significant to the gearset rating.

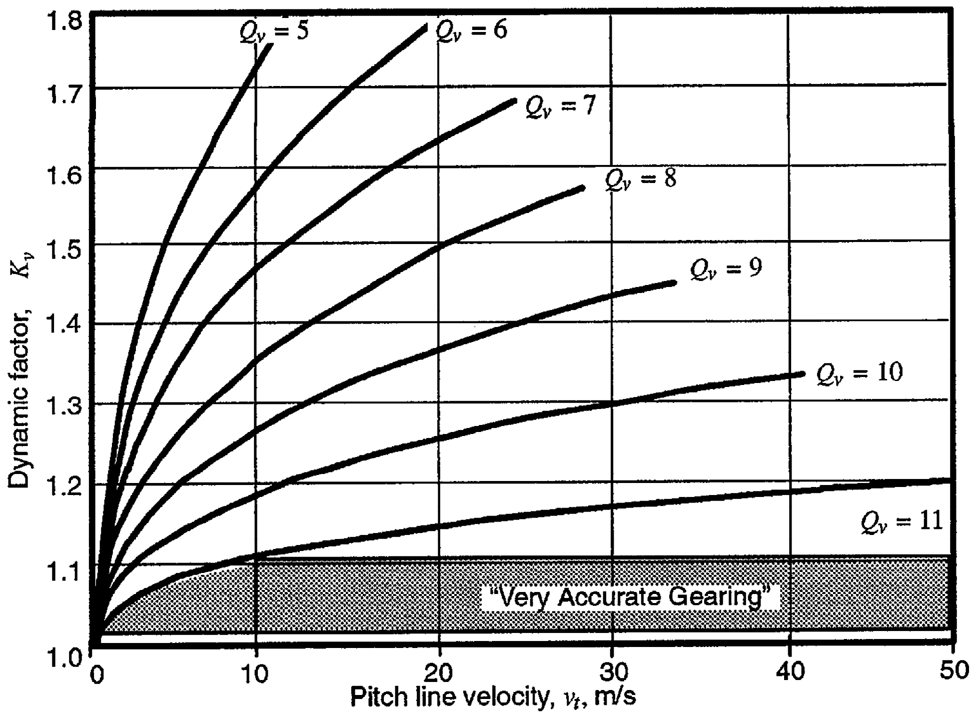


Figure 1 – Dynamic factor, K_v

8.3.1 Very accurate gearing

Where gearing is manufactured using process controls which provide tooth accuracies which correspond to "very accurate gearing", or where the design and manufacturing techniques ensure a low transmission error which is equivalent to this accuracy, values of K_v between 1.02 and 1.11 may be used, depending on the specifier's experience with similar applications and the degree of accuracy actually achieved.

To use these values, the gearing must be maintained in accurate alignment and adequately lubricated so that its accuracy is maintained under the operating conditions.

8.3.2 Curves labeled $Q_v = 5$ through $Q_v = 11$

The empirical curves of figure 1 are generated by the following equations for integer values of Q_v, such that $5 \leq Q_v \leq 11$. Q_v is related to the transmission accuracy grade number.

The dynamic load is influenced by many factors, (see 8.1.1). The dynamic factor can be expressed as a function of Q_v. Q_v can be approximated by using the pitch variation of the pinion or gear member (whichever is greater) by the following formula, rounded to the next lower integer.

$$Q_v = 0.5048 \ln(z_i) + 1.144 \ln(m_n) - 2.852 \ln(V_{pA}) + 13.664 \quad \text{...(21)}$$

where

ln is natural log, $\log_e$;

z_i is adjusted number of pinion or gear teeth.

z_i is $z_1/\cos\beta$ or $z_2/\cos\beta$ ...(22)

whichever results in the lower value of Q_v.

z_i must be between 6 and 1200 or 10 000 m_n, whichever is smaller.

m_n must be between 1.25 and 50 in equation 21.

V_{pA} is absolute value of pitch variation in micrometers (1 µm = 0.001 mm). This is defined in ANSI/AGMA 2000-A88.

Q_v can also be estimated as the appropriate quality number for the expected pitch and profile variations in accordance with ANSI/AGMA 2000-A88.

The profile accuracy for the gearing must be consistent with the pitch accuracy.

Curves may be extrapolated beyond the end points shown in figure 1 based on experience and careful consideration of the factors influencing dynamic load. For purposes of calculation, Eq 26 defines the end points of the curves in figure 1.

$$K_v = \left(\frac{A + \sqrt{200 v_t}}{A}\right)^B \quad ...(23)$$

where

$A = 50 + 56\,(1.0 - B)$ for $5 \le Q_v \le 11$ ...(24)

$B = 0.25\,(12 - Q_v)^{0.667}$ ...(25)

The maximum recommended pitch line velocity for a given grade Q_v is determined:

$$v_{t\,\max} = \frac{[A + (Q_v - 3)]^2}{200} \quad ...(26)$$

where

$v_{t\,\max}$ is maximum pitch line velocity at operating pitch diameter (end point of K_v curves on figure 1), m/s.

8.4 Other values

With specific knowledge of the influencing factors listed in 8.1 and 8.2, and by using a comprehensive dynamic analysis, other dynamic factors can be used for specific applications.

8.5 Unity dynamic factor

When the known dynamic loads (from analysis or experience) are added to the nominal transmitted load, then the dynamic factor can be unity.

9 Overload factor, K_o

The overload factor is intended to make allowance for all externally applied loads in excess of the nominal tangential load, F_t, for a particular application. Overload factors can only be established after considerable field experience is gained in a particular application.

For an overload factor of unity, this rating method includes the capacity to sustain a limited number of up to 200% momentary overload cycles (typically less than four starts in 8 hours, with a peak not exceeding one second duration). Higher or more frequent momentary overloads shall be considered separately.

In determining the overload factor, consideration should be given to the fact that many prime movers and driven equipment, individually or in combination, develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment. There are many possible sources of overload which should be considered. Some of these are: system vibrations, acceleration torques, overspeeds, variations in system operation, split path load sharing among multiple prime movers, and changes in process load conditions.

10 Service factor

The service factor has been used in previous AGMA standards to include the combined effects of overload, reliability, life, and other application related factors. This standard provides a means to account for: variations in load (with overload factor), statistical variations in S–N data (with reliability factor), and the number of design stress cycles (with stress cycle factor).

The AGMA service factor as traditionally used in gear applications depends on experience acquired in each specific application. Product application standards can be a good source for the appropriate value of service factor (see annex C for a more detailed discussion of application analysis).

Equations 27 and 28 are used to establish power ratings for unity service factor to which established service factors may be applied using equation 29. When this is done, the stress cycle factor is calculated using the number of cycles equivalent to a specific number of hours at a specific speed, to establish power rating for unity service factor.

From equation 5:

$$P_{azu} = \frac{\omega_1 b}{1.91 \times 10^7} \frac{Z_I}{K_v K_s K_H Z_R} \left(\frac{d_{w1} \sigma_{HP} Z_N Z_W}{Z_E Y_\theta}\right)^2 \quad ...(27)$$

and from equation 14:

$$P_{ayu} = \frac{\omega_1 d_{w1}}{1.91 \times 10^7 K_v} \frac{b\ m_t}{K_s} \frac{Y_J}{K_H K_B} \frac{\sigma_{FP} Y_N}{Y_\theta} \quad ...(28)$$

where

P_{azu} is allowable transmitted power for pitting resistance at unity service factor ($C_{SF} = 1.0$);

P_{ayu} is allowable transmitted power for bending strength at unity service factor ($K_{SF} = 1.0$);

CAUTION: Both pinion and gear teeth must be checked to account for the differences in material properties, geometry factors, and the number of cycles under load. Therefore, the power rating for unity service factor should be based on the lowest values of the expressions for each of the mating gears.

$\sigma_{HP}\ Z_N\ Z_W$ for pitting resistance

$\frac{\sigma_{HP}\,Y_N\,Y_J}{K_B}$ for bending strength

The allowable transmitted power for the gear set, P_a, is determined:

$$P_a = \text{the lesser of } \frac{P_{azu}}{C_{SF}} \text{ and } \frac{P_{ayu}}{K_{SF}} \quad \text{...(29)}$$

where

C_{SF} is service factor for pitting resistance;

K_{SF} is service factor for bending strength.

11 Safety factors, S_H and S_F

When K_o and Y_Z are used for applying ratings an additional safety factor should be considered to allow for safety and economic risk considerations along with other unquantifiable aspects of the specific design and application (variations in manufacturing, analysis, etc.).

The term "factor of safety" has historically been used in mechanical design to describe a general derating factor to limit the design stress in proportion to the material strength. A safety factor is intended to account for uncertainties or statistical variations in:

- Design analysis;
- Material characteristics;
- Manufacturing tolerances.

Safety factor also must consider human safety risk and the economic consequences of failure. The greater the uncertainties or consequences of these considerations, the higher the safety factor should be. As the extent of these factors become known with more certainty, the value of the safety factor can be more accurately determined. For example, a product such as an automobile transmission which is subjected to full size, full load prototype testing and rigorous quality control of dimensions, materials and processes during manufacture, could have a less conservative safety factor than a hoist made in small quantities to normal commercial practices.

As design practices become more comprehensive, some influence factors have been removed from the unknown area of "safety factor" and introduced as predictable portions of the design method.

Safety factors must be established from a thorough analysis of the service experience with a particular application. A minimum safety factor is normally established for the designer by specific agreement between manufacturer and purchaser. When specific service experience is not available, a thorough analytical investigation should be made.

12 Elastic coefficient, Z_E

The elastic coefficient, Z_E, is defined by the following equation:

$$Z_E = \sqrt{\frac{1}{\pi\left[\left(\frac{1-\nu_1^2}{E_1}\right)+\left(\frac{1-\nu_2^2}{E_2}\right)\right]}} \quad \text{...(30)}$$

where

Z_E is elastic coefficient, $[\text{N/mm}^2]^{0.5}$;

ν_1 and ν_2 is Poisson's ratio for pinion and gear, respectively;

E_1 and E_2 is Modulus of elasticity for pinion and gear, respectively, N/mm^2.

For example, Z_E equals 190 $[\text{N/mm}^2]^{0.5}$, for a steel pinion and gear with ν=0.3 and E=2.05 × 10^5 N/mm^2 for both members.

13 Surface condition factor, Z_R

The surface condition factor, Z_R, used only in the pitting resistance formula, depends on:

- Surface finish as affected by, but not limited to, cutting, shaving, lapping, grinding, shot peening;
- Residual stress;
- Plasticity effects (work hardening).

Standard surface condition factors for gear teeth have not yet been established for cases where there is a detrimental surface finish effect. In such cases, some surface finish factor greater than unity should be used.

The surface condition factor can be taken as unity provided the appropriate surface condition is achieved.

14 Hardness ratio factor, Z_W

The hardness ratio factor, Z_W, depends upon:

- Gear ratio;
- Surface finish of pinion;
- Hardness of pinion and gear.

14.1 Through hardened gears

When the pinion is substantially harder than the gear, the work hardening effect increases the gear capacity. Typical values of Z_W are shown in figure 2. These values are applied to the gear only, not to the pinion.

The values from figure 2 can be calculated as follows:

$$Z_W = 1.0 + A\,(u - 1.0) \quad ...(31)$$

where

$$A = 0.008\,98 \left[\frac{H_{B1}}{H_{B2}}\right] - 0.008\,29 \quad ...(32)$$

H_{B2} is gear Brinell hardness number, HB;

H_{B1} is pinion Brinell hardness number HB.

This equation is valid for the range

$1.2 \le H_{B1}/H_{B2} \le 1.7$

For $H_{B1}/H_{B2} < 1.2$, $A = 0.0$

$H_{B1}/H_{B2} > 1.7$, $A = 0.006\,98$

14.2 Surface hardened/through hardened values

When surface hardened pinions (48 HRC or harder) are run with through hardened gears (180 to 400 HB), a work hardening effect is achieved. The Z_W factor varies with the surface finish of the pinion, R_{z1}, and the mating gear hardness.

Typical values are shown in figure 3, or can be calculated as follows:

$$Z_W = 1.0 + B\,(450 - H_{B2}) \quad ...(33)$$

where

$$B = 0.000\,75\,(e)^{-0.448(R_{z1})} \quad ...(34)$$

e is base of natural or Napierian logarithms = 2.718 28

R_{z1} is surface finish of pinion, micrometers, R_a.

15 Load distribution factor, K_H

The load distribution factor modifies the rating equations to reflect the non–uniform distribution of the load along the lines of contact. The amount of non–uniformity of the load distribution is caused by, and is dependent upon, the following influences:

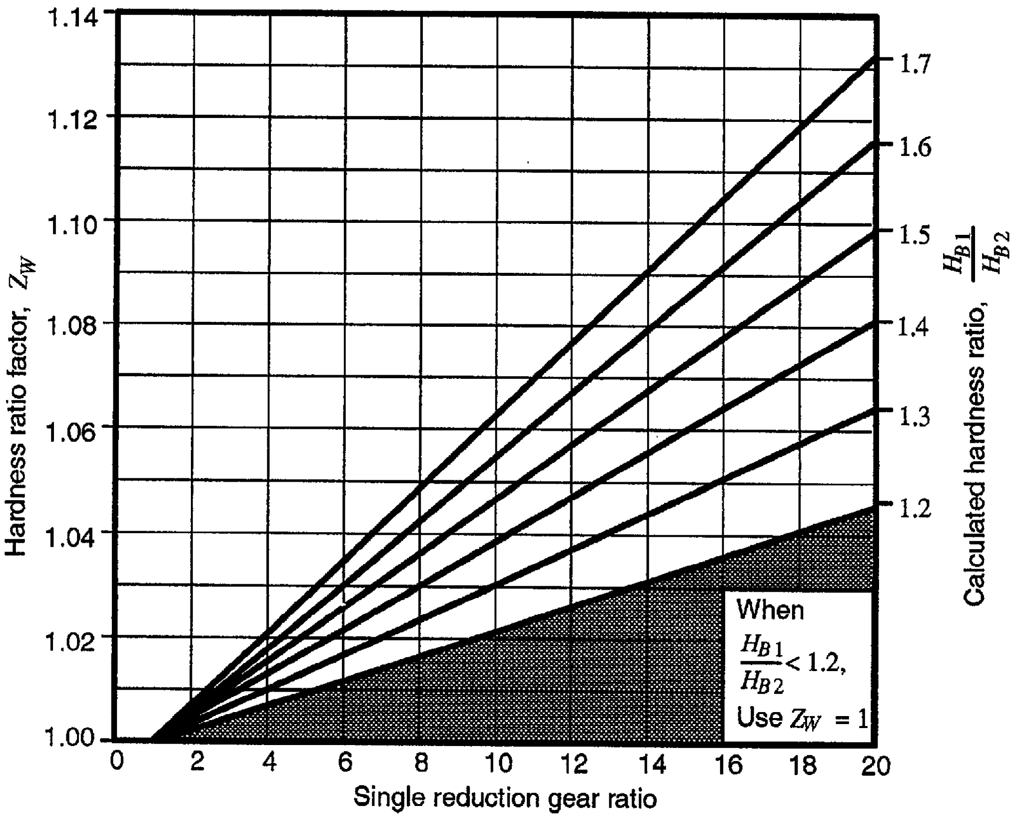


Figure 2 – Hardness ratio factor, Z_W (through hardened)

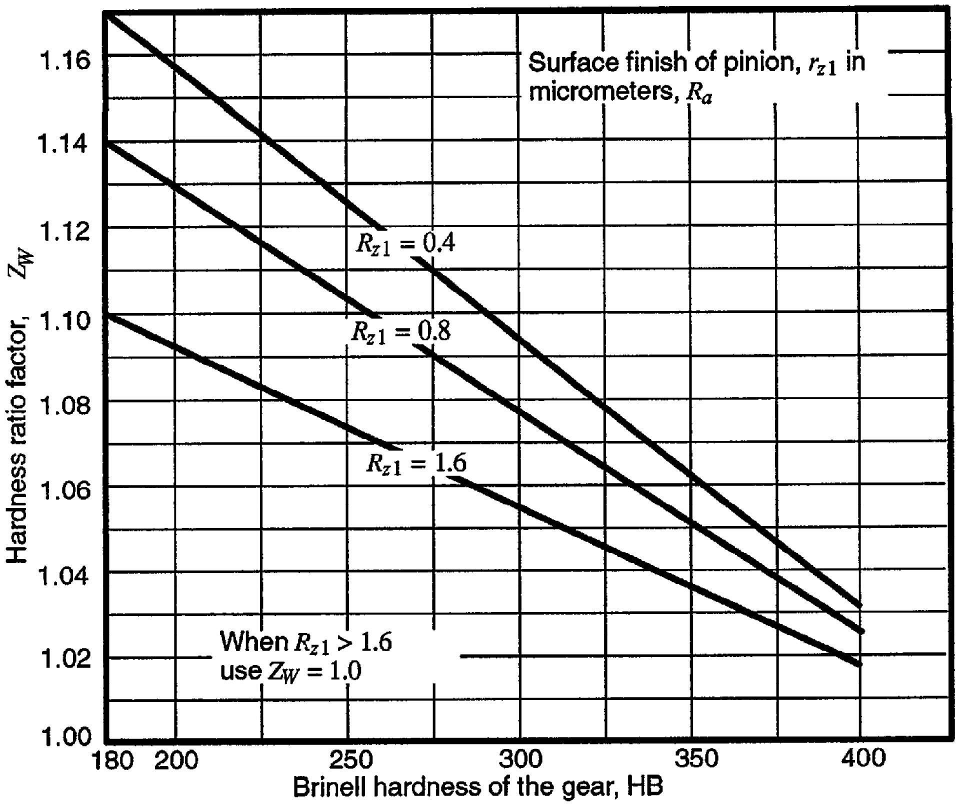


Figure 3 – Hardness ratio factor, Z_W (surface hardened pinions)

Manufacturing variation of gears

– Lead, profile, spacing and runout of both the pinion and the gear.

– Tooth crowning and end relief.

Assembly variations of installed gears

– Alignment of the axes of rotation of the pitch cylinders of the pinion and gear as influenced by housing accuracy and concentricity of the bearings.

Deflections due to applied loads

– Elastic deflections of the pinion and gear teeth.

– Elastic deflections of the pinion and gear bodies.

– Elastic deflections of shafts, bearings, housings and foundations that support the gear elements.

– Displacements of the pinion or gear due to clearance in the bearings.

Distortions due to thermal and centrifugal effects

– Thermal expansion and distortion of the gears due to temperature gradients.

– Temperature gradients in the housing causing nonparallel shafts.

– Centrifugal distortion of the gears due to high speeds.

15.1 Values for load distribution factor, K_H

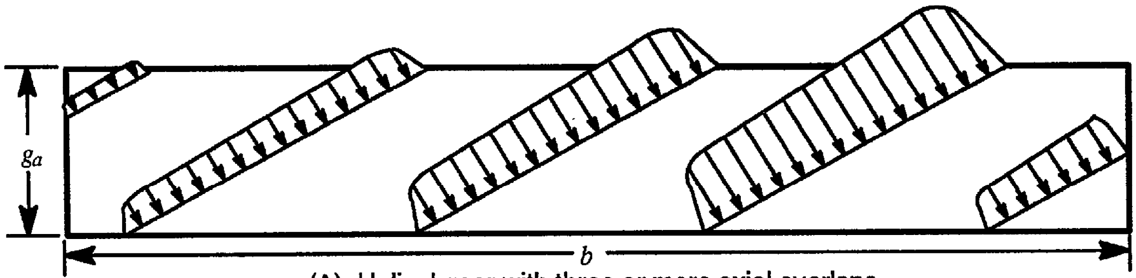
The load distribution factor is defined as: the peak load intensity divided by the average, or uniformly distributed, load intensity; i.e., the ratio of peak to mean loading. Its magnitude is affected by two components:

$K_{H\beta}$ is face load distribution factor;

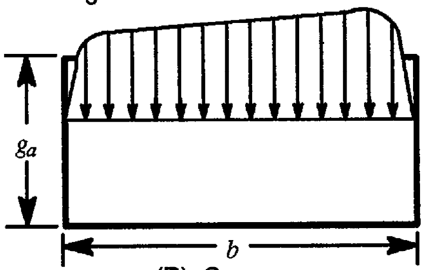
$K_{H\alpha}$ is transverse load distribution factor.

$K_{H\beta}$ and $K_{H\alpha}$ can be interrelated depending on the form of the instantaneous contact line in the plane of action as shown by figure 4. In functional equation form,

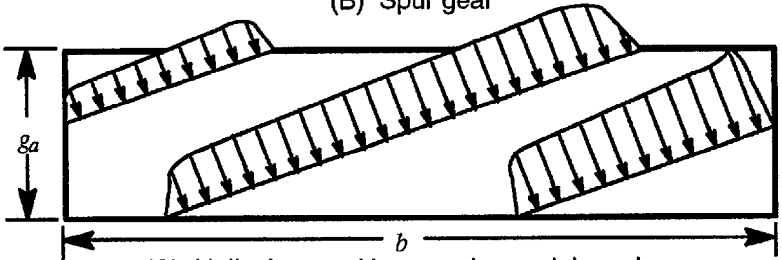
$$K_H = f(K_{H\beta}, K_{H\alpha}) \quad \text{...(35)}$$



(A) Helical gear with three or more axial overlaps



(B) Spur gear



(C) Helical gear with two or less axial overlaps

Figure 4 – Instantaneous contact lines in the plane of action

For helical gears, having three or more axial overlaps, the face load distribution factor, $K_{H\beta}$, accounts for the non–uniformity of load sharing between instantaneous contact lines across the entire face width encompassing all teeth in contact. It is affected primarily by the correctness of pinion and gear leads. Gradual lead deviation (such as results from helix error, misalignment, or pinion deflection), regular patterns of undulation, or random irregularities in lead, are examples of causes of non–uniform load sharing among the contact surfaces of mating teeth across the face width (see figure 4(A)).

For spur gears, where instantaneous contact lines are parallel to the axes, $K_{H\beta}$ is affected primarily by lead and parallelism (see figure 4(B)). In this case, $K_{H\alpha}$ is affected by the transverse contact ratio.

For helical gears having two or less axial overlaps, the interaction of lead and profile effects are so difficult to separate that, for practical purposes, the load distribution subfactors, $K_{H\beta}$ and $K_{H\alpha}$, can be considered as one factor that reflects the ratio of the peak to mean load intensity along the total length of the instantaneous contact lines (see figure 4(C)).

15.2 Transverse load distribution factor, $K_{H\alpha}$

The transverse load distribution factor accounts for the non–uniform distribution of load among the gear teeth which share the load. It is affected primarily by the correctness of the profiles of mating teeth: i.e., profile modification or profile error or both.

Standard procedures to evaluate the influence of $K_{H\alpha}$ have not been established. Therefore, evaluation of the numeric value of the transverse load distribution factor is beyond the scope of this standard and it can be assumed to be unity. Equation 35 therefore, can be modified to:

$$K_H = K_{H\beta} \quad \text{...(36)}$$

15.3 Face load distribution factor, $K_{H\beta}$

The face load distribution factor accounts for the non–uniform distribution of load across the gearing face width. The magnitude of the face load distribution factor is defined as the peak load intensity divided by the average load intensity across the face width.

This factor can be determined empirically or analytically. This standard provides an empirical method only, but includes a theoretical discussion for analytical analysis in annex D. Either method can be used, but when using the analytical approach, the calculated load capacity of the gears should be compared with past experience since it may be necessary to re-evaluate other rating factors to arrive at a rating consistent with past experience.

The empirical method requires a minimum amount of information. This method is recommended for relatively stiff gear designs which meet the following requirements:

- Net face width to pinion pitch diameter ratio, b/d_{w1}, ≤ 2.0. (For double helical gears the gap is not included in the face width).
- The gear elements are mounted between bearings (see following paragraph for overhung gears).
- Face width up to 1020 mm.
- Contact across full face width of narrowest member when loaded.

CAUTION: If $b/d_{w1} > 2.4 - 0.29K$ where K = the contact load factor (see Eq 6), the value of $K_{H\beta}$ determined by the empirical method may not be sufficiently conservative. In this case, it may be necessary to modify the lead or profile of the gears to arrive at a satisfactory result. The empirical method shall not be used when analyzing the effect of a momentary overload. See 16.3.

When gear elements are overhung, consideration must be given to shaft deflections and bearing clearances. Shafts and bearings must be stiff enough to support the bending moments caused by the gear forces to the extent that resultant deflections do not adversely affect the gear contact. Bearing clearances affect the gear contact in the same way as offset straddle mounted pinions. However, gear elements with their overhang to the same support side can compound the effect. This effect is addressed by the pinion proportion modifying factor, K_{Hpm}. When deflections or bearing clearances exceed reasonable limits, as determined by test or experience, an analytical method must be used to establish the face load distribution factor.

When the gap in a double helical gear set is other than the gap required for tooth manufacture, for example in a nested design, each helix should be treated as a single helical set.

Designs which have high crowns to centralize tooth contact under deflected conditions may not use this method.

This method will give results similar to those obtained in previous AGMA standards. Designs falling outside the above b/d_{w1} ranges require special consideration.

For relatively stiff gear designs having gears mounted between bearings (not overhung) and relatively free from externally caused deflections, the following approximate method may be used:

$$K_{H\beta} = 1.0 + K_{Hmc}(K_{Hpf}\, K_{Hpm} + K_{Hma}\, K_{He}) \quad \ldots(37)$$

where

K_{Hmc} = lead correction factor;

K_{Hpf} = pinion proportion factor;

K_{Hpm} = pinion proportion modifier;

K_{Hma} = mesh alignment factor;

K_{He} = mesh alignment correction factor.

The lead correction factor, K_{Hmc}, modifies peak load intensity when crowning or lead modification is applied.

K_{Hmc} = 1.0 for gear with unmodified leads;

K_{Hmc} = 0.8 for gear with leads properly modified by crowning or lead correction.

NOTE: For wide face gears, when methods for careful lead matching or lead corrections to compensate for deflection are employed, it may be desirable to use an analytical approach to determine the load distribution factor.

The pinion proportion factor, K_{Hpf}, accounts for deflections due to load. These deflections are normally higher for wide face widths or higher b/d_{w1} ratios. The pinion proportion factor can be obtained from figure 5.

For double helical gearing, the pinion proportion factor should be evaluated by considering b to be the net face width.

The values for K_{Hpf} as shown in figure 5 can be determined by the following equations:

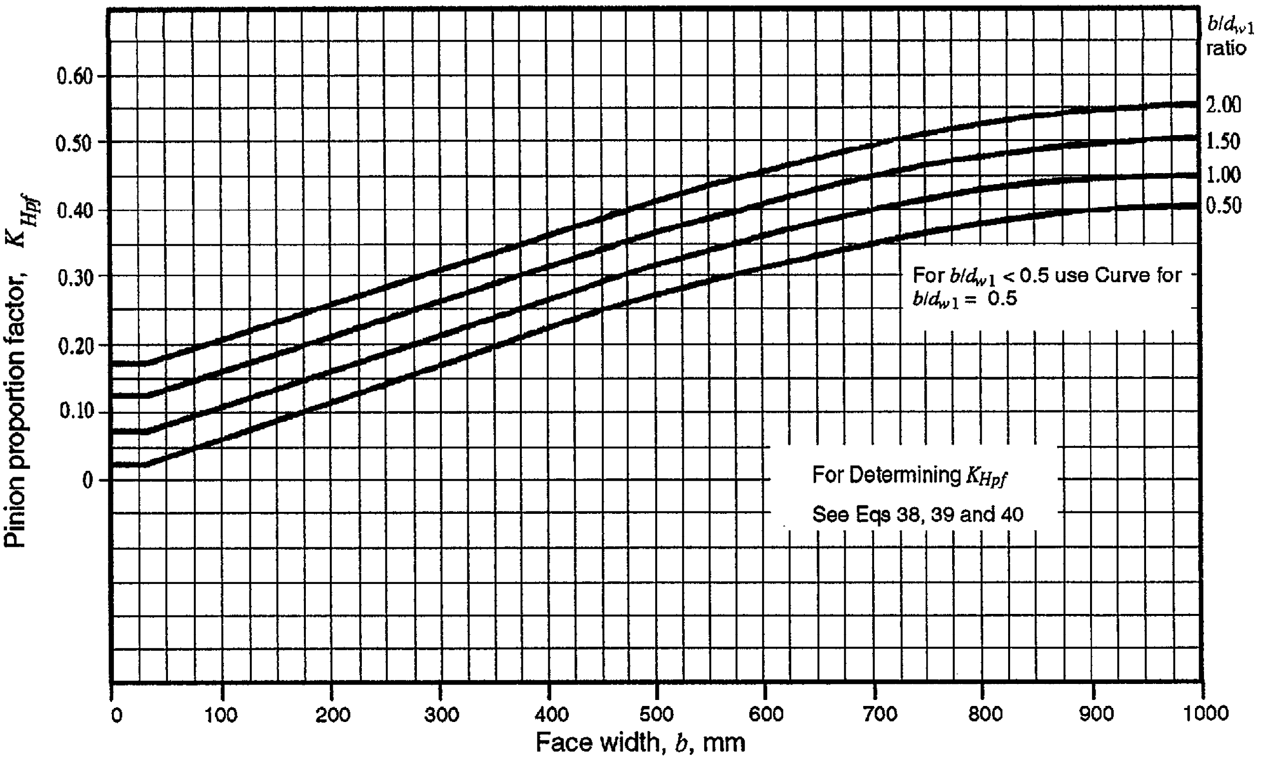


Figure 5 – Pinion proportion factor, K_{Hpf}

when $b \leq 25$

$$K_{Hpf} = \frac{b}{(10)d_{w1}} - 0.025 \qquad ...(38)$$

when $25 < b \leq 432$

$$K_{Hpf} = \frac{b}{(10)d_{w1}} - 0.0375 + 0.000\ 492b \qquad ...(39)$$

when $432 < b \leq 1020$

$$K_{Hpf} = \frac{b}{(10)d_{w1}} - 0.1109 + 0.000\ 815b - 0.000\ 000\ 353b^2 \qquad ...(40)$$

NOTE: For values of $\frac{b}{(10)\ d_{w1}}$ less than 0.05, use 0.05 for this value in equations 38, 39 or 40.

The pinion proportion modifier, K_{Hpm}, alters K_{Hpf}, based on the location of the pinion relative to its bearing centerline.

K_{Hpm} = 1.0 for straddle mounted pinions with $(S_1/S) < 0.175$;

K_{Hpm} = 1.1 for straddle mounted pinions with $(S_1/S) \geq 0.175$.

where

S_1 is the offset of the pinion; i.e, the distance from the bearing span centerline to the pinion mid–face, mm (see figure 6);

S is the bearing span; i.e, the distance between the bearing center lines, mm (see figure 6).

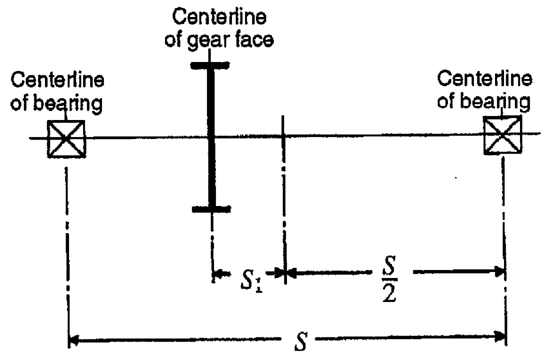


Figure 6 – Evaluation of S and S_1

The mesh alignment factor, K_{Hma}, accounts for the misalignment of the axes of rotation of the pitch cylinders of the mating gear elements from all causes other than elastic deformations. The value for the mesh alignment factor can be obtained from figure 7. The four curves of figure 7 provide representative values for K_{Hma} based on the accuracy of gearing and misalignment effects which can be expected for the four classes of gearing shown.

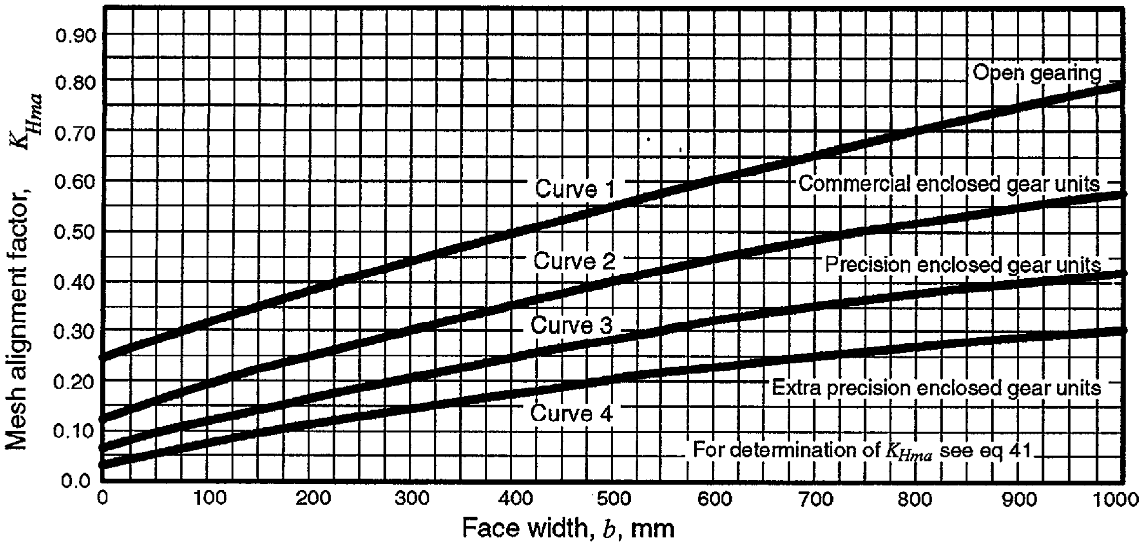


Figure 7 – Mesh alignment factor, K_{Hma}

For double helical gearing, the mesh alignment factor should be evaluated by considering b to be one half of the net face width.

The values for the four curves of figure 7 are defined as follows:

$$K_{Hma} = A + B\,(b) + C\,(b)^2 \quad ...(41)$$

See table 2 for values of A, B and C.

The mesh alignment correction factor is used to modify the mesh alignment factor when the manufacturing or assembly techniques improve the effective mesh alignment. The following values are suggested for the mesh alignment correction factor:

K_{He} = 0.80 when the gearing is adjusted at assembly;
= 0.80 when the compatibility of the gearing is improved by lapping;
= 1.0 for all other conditions.

When gears are lapped and mountings are adjusted at assembly, the suggested value of K_{He} is 0.80.

16 Allowable stress numbers, σ_{HP} and σ_{FP}

The allowable stress numbers for gear materials vary with items such as material composition, cleanliness, residual stress, microstructure, quality, heat treatment, and processing practices. For materials other than steel, a range is shown, and the lower values should be used for general design purposes.

Allowable stress numbers in this standard (tables 3 through 6) are determined or estimated from laboratory tests and accumulated field experiences. They are based on unity overload factor, 10 million stress cycles, unidirectional loading and 99 percent reliability. The allowable stress numbers are designated as σ_{HP} and σ_{FP}, for pitting resistance and bending strength. For service life other than 10 million cycles, the allowable stress numbers are adjusted by the use of stress cycle factors (see clause 17).

Table 2 – Empirical constants; A, B, and C

Curve	A	B	C
Curve 1 Open gearing	2.47×10^{-1}	0.657×10^{-3}	-1.186×10^{-7}
Curve 2 Commercial enclosed gear units	1.27×10^{-1}	0.622×10^{-3}	-1.69×10^{-7}
Curve 3 Precision enclosed gear units	0.675×10^{-1}	0.504×10^{-3}	-1.44×10^{-7}
Curve 4 Extra precision enclosed gear units	0.380×10^{-1}	0.402×10^{-3}	-1.27×10^{-7}

Allowable stress numbers for steel gears are established by specific quality control requirements for each material type and grade. All requirements for the quality grade must be met in order to use the stress values for that grade. This can be accomplished by specifically certifying each requirement where necessary, or by establishing practices and procedures to obtain the requirements on a production basis. It is not the intent of this standard that all requirements for quality grades be certified, but that practices and procedures be established for their compliance on a production basis. Intermediate values are not classified since the effect of deviations from the quality standards cannot be evaluated easily. When justified by testing or experience, higher stress levels for any given grade may be used. The allowable stress numbers are shown in tables 3 through 6, and figures 8 through 11.

The grade cleanliness requirements apply only to those portions of the gear material where the teeth will be located, to a distance below the finished tip diameter of at least two times the tooth depth. On external gears this portion of the gear blank normally will be less than 25 percent of the radius.

Through hardened gears specified above 400 HB may vary widely in endurance strength, depending on the transformation characteristics of the steel, heat treating technique used and the size and shape of the part. The successful use of through hardened parts above 400 HB depends upon experimentally developing a satisfactory technique for heat treating which will develop both high hardness and high fatigue strength.

16.1 Guide for case depth of surface hardened gears

Surface hardened gear teeth require adequate case depth to resist the subsurface shear stresses developed by tooth contact loads and the tooth root fillet tensile stresses, but depths must not be so great as to result in brittle teeth tips and high residual tensile stress in the core.

For gearing requiring maximum performance, especially large sizes, coarse pitches, and high contact stresses, detailed studies must be made of application, loading, and manufacturing procedures to determine the desirable gradients of hardness, strength, and internal residual stresses throughout the tooth.

Table 3 – Allowable contact stress number, σ_{HP}, for steel gears

Material designation	Heat treatment	Minimum surface hardness[1)]	Allowable contact stress number[2)], σ_{HP} N/mm^2		
			Grade 1	Grade 2	Grade 3
Steel[3)]	Through hardened[4)]	see figure 8	see figure 8	see figure 8	—
	Flame[5)] or induction hardened[5)]	50 HRC	1170	1310	—
		54 HRC	1205	1345	—
	Carburized & hardened[5)]	see table 9	1240	1550	1895
	Nitrided[5)] (through hardened steels)	83.5 HR15N	1035	1125	1205
		84.5 HR15N	1070	1160	1240
2.5% Chrome (no aluminum)	Nitrided[5)]	87.5 HR15N	1070	1185	1305
Nitralloy 135M	Nitrided[5)]	90.0 HR15N	1170	1260	1345
Nitralloy N	Nitrided[5)]	90.0 HR15N	1185	1300	1415
2.5% Chrome (no aluminum)	Nitrided[5)]	90.0 HR15N	1215	1350	1490

NOTES
1)Hardness to be equivalent to that at the start of active profile in the center of the face width.
2)See tables 7 through 10 for major metallurgical factors for each stress grade of steel gears.
3)The steel selected must be compatible with the heat treatment process selected and hardness required.
4)These materials must be annealed or normalized as a minimum.
5)The allowable stress numbers indicated may be used with the case depths prescribed in 16.1.

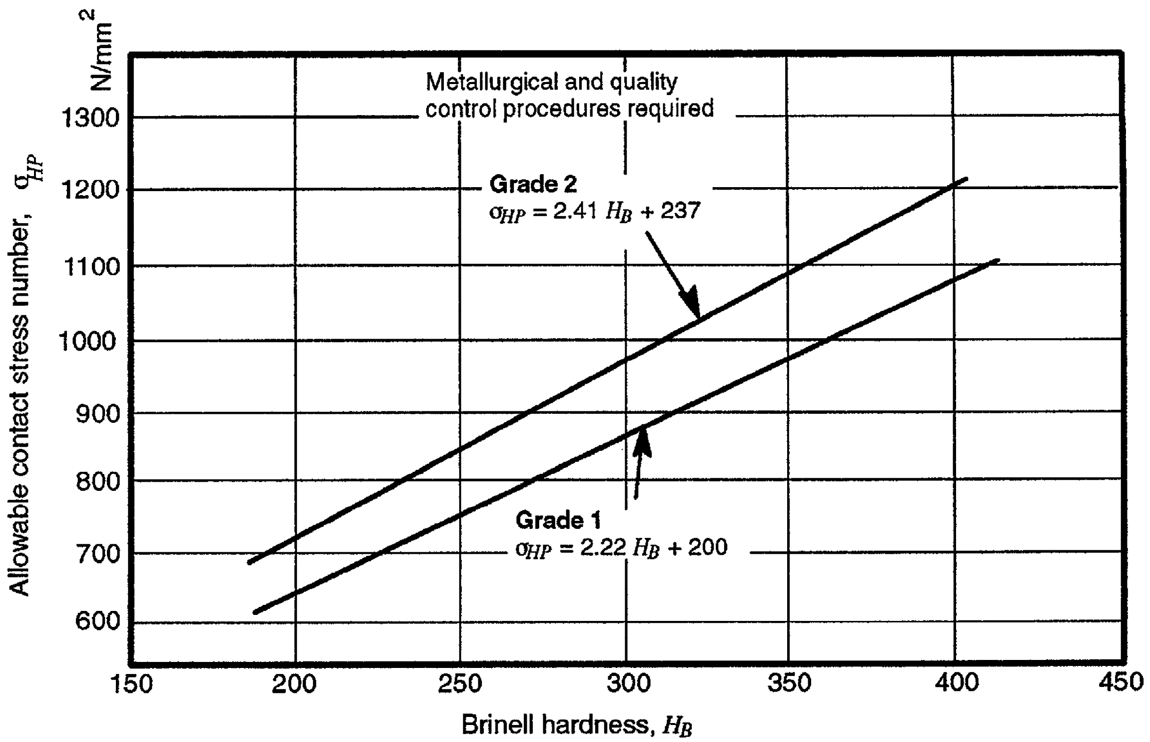


Figure 8 – Allowable contact stress number for through hardened steel gears, σ_{HP}

Table 4 – Allowable bending stress number, σ_{FP}, for steel gears

Material designation	Heat treatment	Minimum surface hardness[1)]	Allowable bending stress number[2)], σ_{FP} N/mm^2		
			Grade 1	Grade 2	Grade 3
Steel[3)]	Through hardened	see figure 9	see figure 9	see figure 9	—
	Flame[4)] or induction hardened[4)] with type A pattern[5)]	see table 8	310	380	—
	Flame[4)] or induction hardened[4)] with type B pattern[5)]	see table 8	150	150	—
	Carburized & hardened[4)]	see table 9	380	450 or 485[6)]	515
	Nitrided[4) 7)] (through hardened steels)	83.5 HR15N	see figure 10	see figure 10	—
Nitralloy 135M, Nitralloy N and 2.5% Chrome (no aluminum)	Nitrided[4) 7)]	87.5 HR15N	see figure 11	see figure 11	see figure 11

NOTES

1)Hardness to be equivalent to that at the root diameter in the center of the tooth space and face width.

2)See tables 7 through 10 for major metallurgical factors for each stress grade of steel gears.

3)The steel selected must be compatible with the heat treatment process selected and hardness required.

4)The allowable stress numbers indicated may be used with the case depths prescribed in 16.1.

5)See figure 12 for type A and type B hardness patterns.

6)If bainite and microcracks are limited to Grade 3 levels, 485 N/mm^2 may be used.

7)The overload capacity of nitrided gears is low. Since the shape of the effective S–N curve is flat, the sensitivity to shock should be investigated before proceeding with the design. [7]

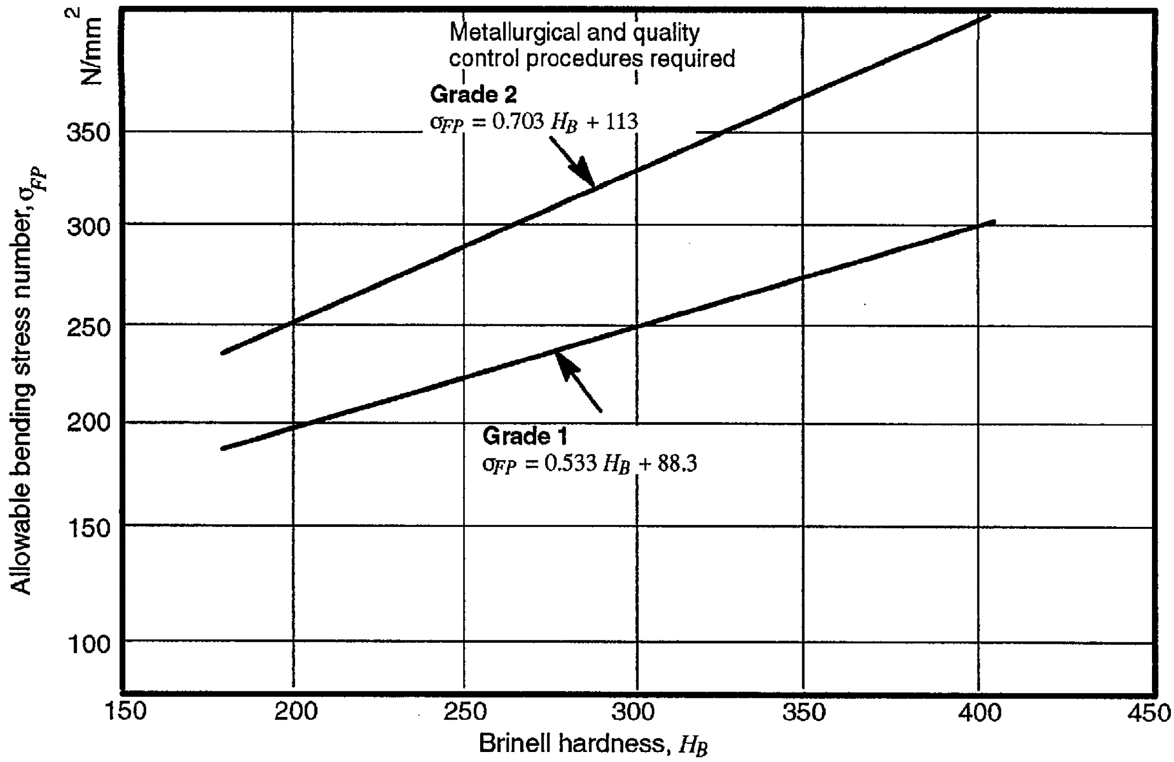


Figure 9 – Allowable bending stress number for through hardened steel gears, σ_{FP}

Table 5 – Allowable contact stress number, σ_{HP}, for iron and bronze gears

Material	Material designation[1)]	Heat treatment	Typical minimum surface hardness[2)]	Allowable contact stress number[3)] σ_{HP} N/mm²
ASTM A48 Gray cast iron	Class 20	As cast	—	345 – 415
	Class 30	As cast	174 HB	450 – 520
	Class 40	As cast	201 HB	520 – 585
ASTM A536 Ductile (nodular) iron	Grade 60–40–18	Annealed	140 HB	530 – 635
	Grade 80–55–06	Quenched & tempered	179 HB	530 – 635
	Grade 100–70–03	Quenched & tempered	229 HB	635 – 770
	Grade 120–90–02	Quenched & tempered	269 HB	710 – 870
Bronze		Sand cast	Minimum tensile strength 275 N/mm²	205
	ASTM B–148 Alloy 954	Heat treated	Minimum tensile strength 620 N/mm²	450

NOTES

1) See ANSI/AGMA 2004–B89, *Gear Materials and Heat Treatment Manual.*

2) Hardness to be equivalent to that at the start of active profile in the center of the face width.

3) The lower values should be used for general design purposes. The upper values may be used when:

– High quality material is used.
– Section size and design allow maximum response to heat treatment.
– Proper quality control is effected by adequate inspection.
– Operating experience justifies their use.

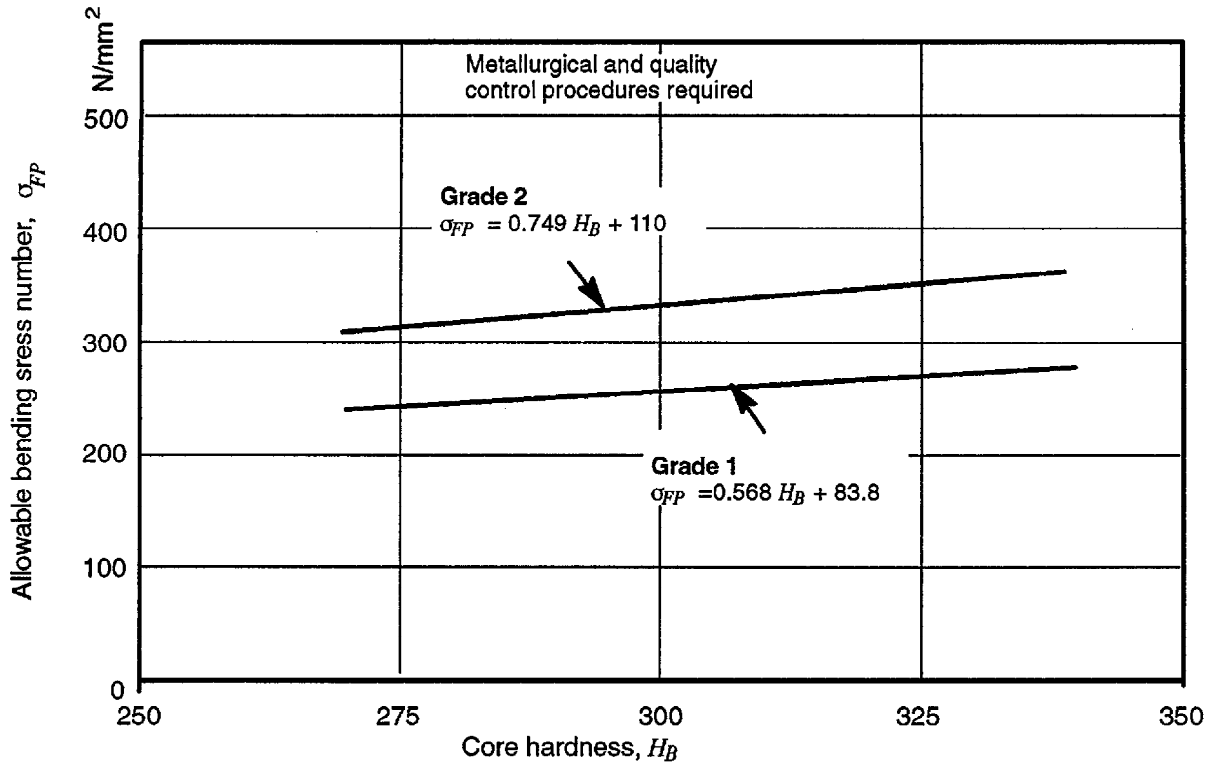


Figure 10 – Allowable bending stress numbers for nitrided through hardened steel gears (i.e., AISI 4140, AISI 4340), σ_{FP}

Table 6 – Allowable bending stress number, σ_{FP}, for iron and bronze gears

Material	Material Designation[1)]	Heat Treatment	Typical Minimum Surface Hardness[2)]	Allowable Bending Stress Number[3)] σ_{FP} N/mm²
ASTM A48 Gray cast iron	Class 20	As cast	—	34.5
	Class 30	As cast	174 HB	59
	Class 40	As cast	201 HB	90
ASTM A536 Ductile (nodular) iron	Grade 60–40–18	Annealed	140 HB	150 – 230
	Grade 80–55–06	Quenched & tempered	179 HB	150 – 230
	Grade 100–70–03	Quenched & tempered	229 HB	185 – 275
	Grade 120–90–02	Quenched & tempered	269 HB	215 – 305
Bronze		Sand cast	Minimum tensile strength 275 N/mm²	39.5
	ASTM B–148 Alloy 954	Heat treated	Minimum tensile strength 620 N/mm²	165

NOTES

1) See ANSI/AGMA 2004–B89, *Gear Materials and Heat Treatment Manual.*

2) Measured hardness to be equivalent to that which would be measured at the root diameter in the center of the tooth space and face width.

3) The lower values should be used for general design purposes. The upper values may be used when:
- High quality material is used.
- Section size and design allow maximum response to heat treatment.
- Proper quality control is effected by adequate inspection.
- Operating experience justifies their use.

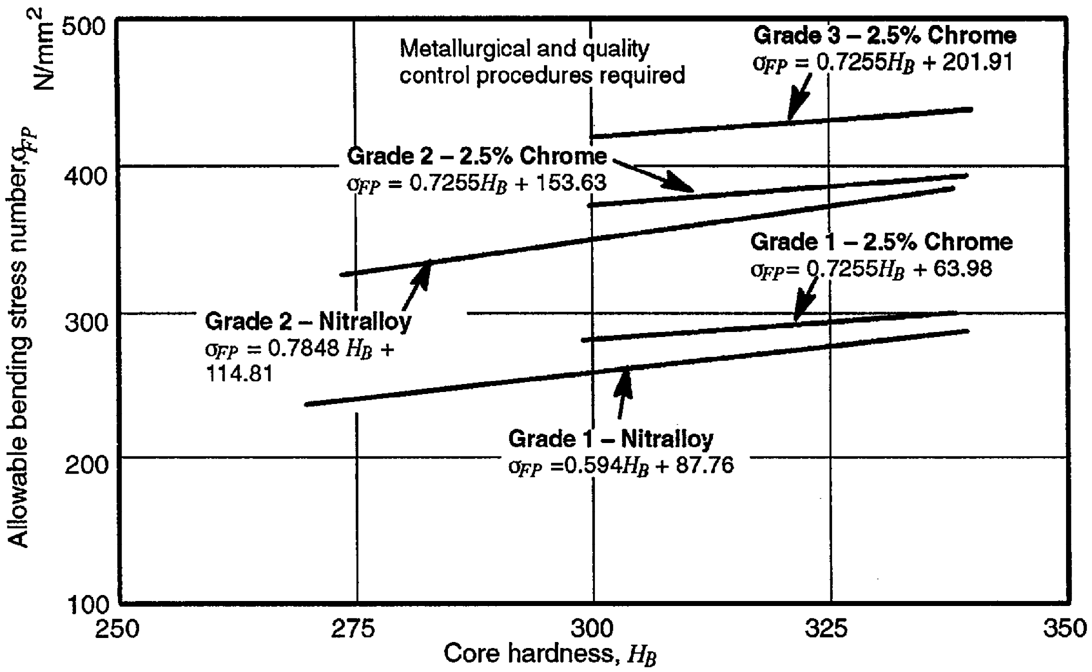


Figure 11 – Allowable bending stress numbers for nitriding steel gears, σ_{FP}

Table 7 – Major metallurgical factors affecting the allowable contact stress number, σ_{HP}, and allowable bending stress number, σ_{FP}, of through hardened steel gears[1) 2) 3)]

Metallurgical factor	Grade 1	Grade 2
ASTM E112 grain size	Predominantly 5 or finer	Predominantly 5 or finer
Upper transformation products which primarily include bainite and fine pearlite.[4)]	Not specified	Max controlling section, mm (see annex F) / Max upper transformation products at 400X to 254 incl — 10% Over 254 — 20% No blocky ferrite (due to improper austenization)
Decarburization and stock removal	Not specified	None apparent at 400X, stock removal sufficient to remove any decarburization.
Specified hardness at surface, σ_{HP} only	See figure 8	See figure 8
Specified hardness at root, σ_{FP} only	See figure 9	See figure 9
Cleanliness[5)]	Not specified	AMS 2301 or ASTM A866 for wrought steel (certification not required). Castings are permissible with primarily round (Type 1) sulfide inclusions
Sulfur	Not specified	0.025% maximum for wrought 0.040% maximum for castings

NOTES

1)See table 3 for values of σ_{HP} and table 4 for values of σ_{FP}. Criteria for grades 1 & 2 apply to both stress numbers unless otherwise specified in the metallurgical factor column.

2)All criteria in any given grade must be met to qualify for the stress number in that grade.

3Unless otherwise specified, proper process control with periodic verification is an acceptable method to meet these requirements (see clause 16).

4)The microstructure requirements apply only to those portions of the gear material where the teeth will be located to a depth equal to that of 1.2 times the tooth depth.

5)The grade cleanliness requirements apply only to those portions of the gear material where the teeth will be located to a distance below the finished tip diameter of at least two times the tooth depth. On external gears, this portion of the gear blank normally will be less than 25 percent of the radius.

CAUTION: For cold service, below 0°C, see 3.6.1.

Table 8 – Major metallurgical factors affecting the allowable contact stress number, σ_{HP}, and allowable bending stress number, σ_{FP}, of flame or induction hardened steel gears[1) 2) 3)]

<table>
<tr><th>Metallurgical factor</th><th>Grade 1</th><th colspan="2">Grade 2</th></tr>
<tr><td>ASTM E112 grain size</td><td>Predominantly 5 or finer</td><td colspan="2">Predominantly 5 or finer</td></tr>
<tr><td>Material composition</td><td>Not specified</td><td colspan="2">Medium carbon alloy steel</td></tr>
<tr><td>Prior structure</td><td>Not specified</td><td colspan="2">Quenched and tempered</td></tr>
<tr><td>Material form</td><td>Not specified</td><td colspan="2">Forgings and wrought steel; castings with magnetic particle inspection of gear tooth area</td></tr>
<tr><td>Cleanliness[4)]</td><td>Not specified</td><td colspan="2">AMS 2301 or ASTM A866 for wrought steel (certification not required); castings are permissible with primarily round (type 1) sulfide inclusions.</td></tr>
<tr><td>Sulfur content</td><td>Not specified</td><td colspan="2">0.025% maximum for wrought
0.040% maximum for castings</td></tr>
<tr><td>Core hardness, center of tooth at root diameter, σ_{HP} only</td><td>Not specified</td><td colspan="2">28 HRC minimum</td></tr>
<tr><td>Core hardness, center of tooth at root diameter, σ_{FP} only</td><td>Not specified</td><td colspan="2">Type A – 28 HRC minimum
Type B – not specified</td></tr>
<tr><td>Non–martensitic transformation products in hardened zone</td><td>Limited by effect on specified hardness</td><td colspan="2">10% maximum, no free ferrite</td></tr>
<tr><td>Surface hardness, σ_{HP} only</td><td>See table 3</td><td colspan="2">See table 3</td></tr>
<tr><td>Surface hardness at root, σ_{FP} only</td><td>Type A – see table 4
Type B – not specified</td><td colspan="2">Type A – See table 4
Type B – not specified</td></tr>
<tr><td>Hardness pattern (see figure 12), σ_{FP} only</td><td>As required per table 4</td><td colspan="2">Type A – Contour pattern with a ductile core
Type B – not specified</td></tr>
<tr><td rowspan="2">Magnetic particle (method per ASTM E709 on teeth)[5)]</td><td rowspan="2">Not specified</td><td>Module
m_n</td><td>Maximum indication, mm</td></tr>
<tr><td>≤ 2.5
>2.5 to < 8
≥ 8</td><td>1.6
2.4
3.2</td></tr>
</table>

NOTES

1) See table 3 for values of σ_{HP} and table 4 for values of σ_{FP}. Criteria for grades 1 & 2 apply to both stress numbers unless otherwise specified in the metallurgical factor column.

2) All criteria in any given grade must be met to qualify for the stress number in that grade.

3) Unless otherwise specified, proper process control with periodic verification is an acceptable method to meet these requirements (see clause 16).

4) The grade cleanliness requirements apply only to those portions of the gear material where the teeth will be located to a distance below the finished tip diameter of at least two times the tooth depth. On external gears, this portion of the gear blank normally will be less than 25 percent of the radius.

5) No cracks, bursts, seams or laps are permitted in the tooth area of finished gears, regardless of grade. Limits: maximum of one indication per 25 mm of face width and maximum of five in one tooth flank. No indications allowed below 1/2 working depth of tooth. Indications smaller than 0.40 mm are not considered. Removal of defects which exceed the stated limits is acceptable provided the integrity of the gear is not compromised.

Table 9 – Major metallurgical factors affecting the allowable contact stress number, σ_{HP}, and allowable bending stress number, σ_{FP}, of carburized and hardened steel gears[1) 2) 3)]

Metallurgical factor[4) 5)]	Grade 1	Grade 2	Grade 3
Surface hardness (HRC or equivalent on representative surface)	55–64 HRC	58–64 HRC	58–64 HRC
Case hardness	55–64 HRC or equivalent	58–64 HRC or equivalent	58–64 HRC or equivalent
Limit of carbides in case	Semicontinuous	Acceptable per AGMA 246.02A or ANSI/AGMA 6033–A88	Acceptable per light discontinuous micro per AGMA 246.02A or ANSI/AGMA 6033–A88
Tempering	Recommended	Required	Required
Surface temper (per ANSI/AGMA 2007–B92 with swab technique permitted), σ_{HP} only	Not specified	Class FB3	Class FB2
Cleanliness[6)]	Not specified	AMS 2301 or ASTM A534 for wrought steel (certification not required); castings are permissible which have primarily round (type 1) sulphide inclusions. Magnetic particle in the final product to grade 3 levels may be substitute in lieu of AMS 2301	AMS 2300 or ASTM A535 (certification required)
Ultrasonic inspection (UT)	Not specified	Specified for wrought per ASTM A388 and castings per ASTM A609[7)] recommended but not required. Suggested for large diameter parts to detect flaws before the expense of machining	Specified for wrought per ASTM A388. Castings not applicable[7)]
Magnetic particle (method per ASTM E709 on teeth)[8)]	Not specified	Module m_n – Maximum indication, mm ≤ 2.5 – 1.6 > 2.5 to < 8 – 2.4 ≥8 – 3.2	Module m_n – Maximum indication, mm ≤ 2.5 – 0.8 > 2.5 to < 8 – 1.6 ≥8 – 2.4
Decarburization in case (to 0.127 mm depth), σ_{HP} only	Not specified (hardness must be met)	No partial decarb. apparent at 400X, except in unground roots	No partial decarb. apparent at 400X, except in unground roots
Decarburization in case (to 0.127 mm depth), σ_{FP} only	Not specified		
Surface carbon in case	0.60 – 1.10%	0.60 – 1.10%	0.60 – 1.00%
Minimum effective case depth at root radius, or on representative coupon, σ_{FP} only	Not specified	50% of minimum specified case at 1/2 tooth height recommended	66% of minimum specified case at 1/2 tooth height recommended
Microcracks in case (cracks across more than one platelet)[9]	Not specified	Not specified	10 maximum per 0.065 mm² field at 400X
Secondary transformation products, (upper bainite) in case along flank above root, or on representative coupon, to 0.25 mm deep, σ_{HP} only	Not specified	5% maximum at 400X	Trace at 400X
Secondary transformation products, (upper bainite) in case along flank above root, or on representative coupon, to 0.25 mm deep, σ_{FP} only	Not specified	10% maximum at 400X	5% maximum at 400X
Intergranular oxidation (IGO) applicable to unground surface. Determined by metallographic inspection of unetched coupon, if used. Limits in inches to be based on case depth as follows:	Not specified	Case depth, mm – IGO, mm <0.76 – 0.018 $0.76 \le h_e < 1.50$ – 0.025 $1.50 \le h_e < 2.25$ – 0.038 $2.25 \le h_e < 3.00$ – 0.051 ≥3.00 – 0.061	Case depth, mm – IGO, mm <0.76 – 0.013 $0.76 \le h_e < 1.50$ – 0.020 $1.50 \le h_e < 2.25$ – 0.020 $2.25 \le h_e < 3.00$ – 0.025 ≥3.00 – 0.031

(continued)

Table 9 *(concluded)*

Metallurgical factor[4) 5)]	Grade 1	Grade 2	Grade 3
Intergranular oxidation (IGO) applicable to unground surface. Determined by metallographic inspection of unetched coupon, if used. Limits in inches to be based on case depth as follows:	Not specified	If excessive salvage is allowed by controlled shotpeening, with the agreement of the customer.	
Maximum retained austenite in case (determined metallographically)[10)]	Not specified	30% maximum	30% maximum
Hardenability band	Not specified	According to H–Band requirements. Recommended but not required	According to upper half of H–Band requirements
Core hardness (at center of tooth at root diameter or on representative coupon), σ_{HP} only[11)]	Not specified	21 HRC minimum	21 HRC minimum
Core hardness (at center of tooth at root diameter or on representative coupon), σ_{FP} only[11)]	21 HRC minimum	25 HRC minimum	30 HRC minimum[12)]
ASTM E112 grain size	Predominantly 5 or finer	Predominantly 5 or finer	Predominantly 5 or finer
Sulfur content	Not specified	0.040% maximum	0.015% maximum
Material form	Not specified	Not specified	Steel forgings and bar stock[13)]
Shot peening, σ_{FP} only	Not specified	Recommended if the root is ground	Required in tooth root area

NOTES

1) See table 3 for values of σ_{HP}, and table 4 for values of σ_{FP}. Criteria for grades 1, 2, and 3 apply to both stress numbers unless otherwise specified in the metallurgical factor column.

2) All criteria in any given grade must be met to qualify for the stress number in that grade.

3) Unless otherwise specified, proper process control with periodic verification is an acceptable method to meet these requirements (see clause 16).

4) Microstructure, microhardness and core hardness considerations may be determined from test coupons. Test coupons shall be from the same alloy steel (not necessarily same heat) as the production parts. Coupon should be sized to produce a similar cooling rate to that obtained in the gear teeth of the actual gear. Coupon proportions of minimum diameter 6 m_n and minimum length 12 m_n are used in ISO 6336–5. Microhardness is to be measured on the test coupon at a depth not more than 0.76 mm below the depth corresponding to the finished tooth surface.

5) For low temperature service, below 0°C, consider low temperature Charpy V–notch impact strength, fracture appearance transition temperature (FATT) requirements and use of nickel alloy steel. Consideration must be given to the loss of hardness and strength of some materials due to the tempering effect of temperatures over 175°C.

6) The grade cleanliness requirements apply only to those portions of the gear material where the teeth will be located to a distance below the finished tip diameter of at least two times the tooth depth. On external gears, this portion of the gear blank normally will be less than 25 percent of the radius.

7) Specified for wrought gearing per ASTM A388, using either the back reflection or reference block technique. Use a 3.18 mm FBH (8–0400) per ASTM E428 (also described in ANSI/AGMA 6033–A88). A distance amplitude correction curve is not intended. Inspection is from the O.D. to mid–radius and a 360 degree scan is required. Other UT specifications which ensure the same quality level are permitted. Specified for cast gears (Grade 2 only) per ASTM A609 Level 1 in Zone 1 (OD to 25 mm below roots) and Level 2 in Zone 2 (remainder of rim) using 3.18 mm FBH; or approved equivalent using back reflection technique (also described in ANSI/AGMA 6033–A88).

8) No cracks, bursts, seams or laps are permitted in the tooth area of finished gears, regardless of grade. Limits: maximum of one indication per 25 mm of face width and maximum of five in one tooth flank. No indications allowed below 1/2 working depth of tooth. Indications smaller than 0.40 mm are not considered. Removal of defects which exceed the stated limits is acceptable provided the integrity of the gear is not compromised.

9) Maximum limit of microcracks for Grade 3 gearing may be difficult to achieve with sub–zero treatment to transform retained austenite level to 30% max.

10) Sub–zero treatment, if required, should be preceded by tempering at 150°C minimum, to minimize formation of microcracks, followed by retempering. The purpose of the sub–zero treatment should be to pick up an additional one to two Rockwell 'C' hardness points. Sub–zero treatment should not be employed to transform large amounts of retained austenite (e.g., 50%) to gain dramatic improvements in hardness, even with prior tempering, or microcracking may occur.

11) Core hardness requirements for pitting resistance and bending strength are considered independently. The allowable stress numbers are established for the grade selected based on hardness. Because higher contact stresses are allowed for carburized and hardened gears, the resulting higher bending stresses must also be accommodated. Therefore, for gearing of this type, higher core hardnesses are specified for the bending strength. The gear rating may be limited by either pitting resistance or bending strength for the selected grade and its core hardness requirement.

12) Minimum hardness of 30 HRC for grade 3 may be difficult to achieve on gears coarser than 4.23 m_n. Therefore, a minimum hardness of 25 HRC is acceptable in such cases.

13) Requires a minimum reduction of 7 to 1 for strand or continuous cast barstock; or minimum reduction of 4 to 1 for forged gears.

Table 10 – Major metallurgical factors affecting the allowable contact stress number, σ_{HP}, and allowable bending stress number, σ_{FP}, for nitrided steel gears[1) 2) 3)]

Metallurgical factor	Grade 1	Grade 2	Grade 3
ASTM E112 grain size	Predominantly 4 or finer	Predominantly 5 or finer	Predominantly 5 or finer
Hardenability	H–Band	H–Band	H–Band
Sulfur content	0.040% maximum	0.025% maximum	0.015% maximum
Cleanliness[4)]	Not specified	AMS 2301 or ASTM A866 (certification not required)	AMS 2300 or ASTM A866 certification required
Surface hardness	Per table 3	Per table 3	Per table 3
Core hardness	28 HRC minimum	28 HRC minimum	32 HRC minimum
White layer (maximum)	0.025 mm	0.020 mm	0.012 mm
Upper transformation products which primarily include bainite and fine pearlite[5)]	Not specified	Max controlling section, mm (see annex F) — Max upper transformation products @ 400X to 254 inc. — 10% Over 254 — 20% No blocky ferrite (due to improper austenization)	Trace at 400X
Ultrasonic inspection	Not specified	Not specified	Specified for wrought per ASTM A388[6)]
Magnetic particle (method per ASTM E709 on teeth)[7)]	Not specified	Not specified	Module m_n — Maximum indication, mm ≤2.5 — 0.8 > 2.5 to < 8 — 1.6 ≥8 — 2.4
Grinding burns	Not specified	See note 8	See note 8

NOTES

1) See table 3 for values of σ_{HP}, and table 4 for values of σ_{FP}.

2) All criteria in any given grade must be met to qualify for the stress number in that grade.

3) Unless otherwise specified, proper process control with periodic verification is an acceptable method to meet these requirements (see clause 16).

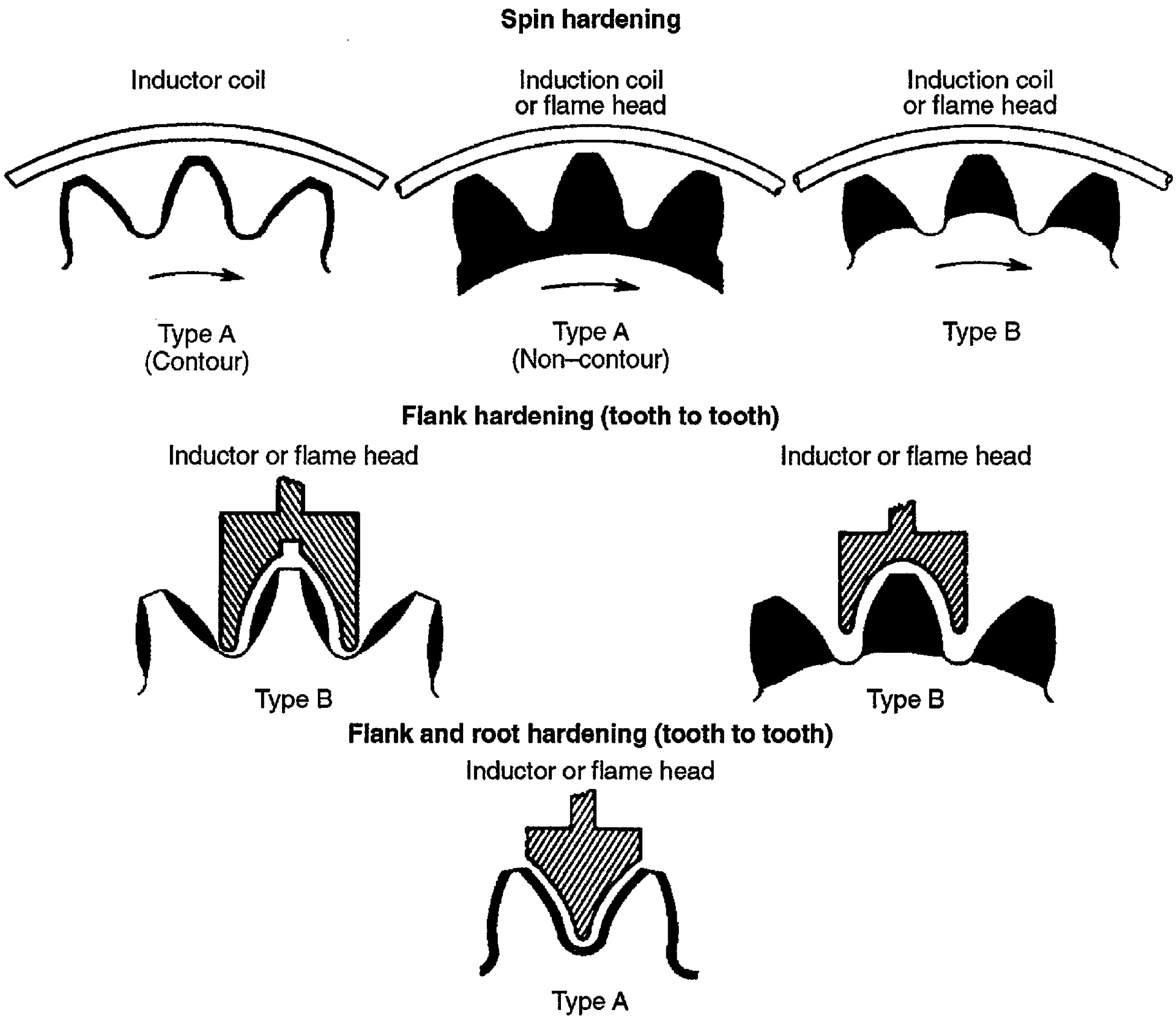
4) The grade cleanliness requirements apply only to those portions of the gear material where the teeth will be located to a distance below the finished tip diameter of at least two times the tooth depth. On external gears, this portion of the gear blank normally will be less than 25 percent of the radius.

5) The microstructure requirements apply only to those portions of the gear material where the teeth will be located to a depth equal to that of 1.2 times the tooth depth.

6) Specified for wrought gearing per ASTM A388, using either the back reflection or reference block technique. Use a 3.18 mm FBH (8–0400) per ASTM E428 (also described in ANSI/AGMA 6033–A88). A distance amplitude correction curve is not intended. Inspection is from the O.D. to mid–radius and a 360 degree scan is required. Other UT specifications which ensure the same quality level are permitted. Specified for cast gears (Grade 2 only) per ASTM A609 Level 1 in Zone 1 (OD to 25 mm below roots) and Level 2 in Zone 2 (remainder of rim) using 3.18 mm FBH; or approved equivalent using back reflection technique (also described in ANSI/AGMA 6033–A88).

7) No cracks, bursts, seams or laps are permitted in the tooth area of finished gears, regardless of grade. Limits: maximum of one indication per 25 mm of face width and maximum of five in one tooth flank. No indications allowed below 1/2 working depth of tooth. Indications smaller than 0.40 mm are not considered. Removal of defects which exceed the stated limits is acceptable provided the integrity of the gear is not compromised.

8) Grinding burns are possible on nitrided materials. The normal inspection methods, as defined in ANSI/AGMA 2007–B92, are not applicable to nitriding. Care must be taken when grinding nitrided surfaces to ensure that no harmful surface conditions are produced in the grinding process



NOTE: Type A indicates flanks and roots are hardened, contour or non-contour pattern. Type B indicates only hardening of flanks extending to the form diameter.

Figure 12 – Variations in hardening pattern obtainable on gear teeth with flame or induction hardening

The effective case depth for carburized and hardened gears is defined as the depth below the surface at which the Rockwell 'C' hardness, HRC, has dropped to 50 HRC or equivalent.

The effective case depth for induction and flame hardened gears is defined as the depth below the surface at which the hardness is equivalent to 10 Rockwell 'C' points below the specified minimum surface hardness.

A guide for minimum effective case depth, $h_{e\,\min}$, at the pitch line for carburized and induction hardened external (not internal) teeth based on the depth of maximum shear from contact loading is given by the formula:

$$h_{e\,\min} = \frac{\sigma_H d_{w1} \sin \alpha_{pt}}{U_H \cos \beta_{mb}} C_G \qquad \text{...(42) [6]}$$

where

$h_{e\,\min}$ is minimum effective case depth at pitchline, mm;

σ_H is contact stress number N/mm^2. The maximum value recommended is 1400 N/mm^2 for this equation;

α_{pt} is operating transverse pressure angle;

U_H is hardening process factor, N/mm^2;

= 4.4 × 10^4 N/mm^2 for carburized and hardened;

= 3 × 10^4 N/mm^2 for tooth–to–tooth induction hardened;

β_{mb} is base helix angle.

Another guideline for determining case depth is shown in figure 13. These case depths have had a long history of successful use on carburized gears. They are not based on equation 42.

Care should be exercised when choosing case depth, such that adequate case depths prevail at the tooth root fillet, and that tooth tips are not over hardened and brittle. A suggested value of maximum effective case depth at the pitch line, $h_{e\,\max}$, is:

$$h_{e\,\max} = \text{the lesser of } 0.4\, m_n \text{ or } 0.56\, s_{an} \qquad \text{...(43)}$$

where

$h_{e\,\max}$ is suggested maximum effective case depth at pitchline, mm;

s_{an} is normal tooth thickness at the top land of the gear in question, mm.

If $h_{e\,\min}$ from Eq 42 (with heat treat tolerance considered) exceeds $h_{e\,\max}$, a careful review of the proposed design is required. Changing the profile shift, lowering the operating pressure angle, or using a larger module will increase $h_{e\,\max}$.

For nitrided gears, case depth is specified as total case depth and is defined as the depth below the surface at which the hardness has dropped to 110 percent of the core hardness.

A guide for minimum case depth for nitrided external (not internal) teeth based on the depth of maximum shear from contact loading is given by the formula:

$$h_{c\,\min} = \frac{U_c \sigma_H d_{w1} \sin\alpha_{pt}}{1.14 \times 10^5 \cos\beta_{mb}} C_G \qquad \text{...(44)}$$

where

$h_{c\min}$ is minimum total case depth for nitrided gears, mm;

U_c is core hardness coefficient, from figure 14.

If the value of $h_{c\min}$ from Eq 44 is less than the value for normal case depth from figure 15, then the minimum value from figure 15 should be used.

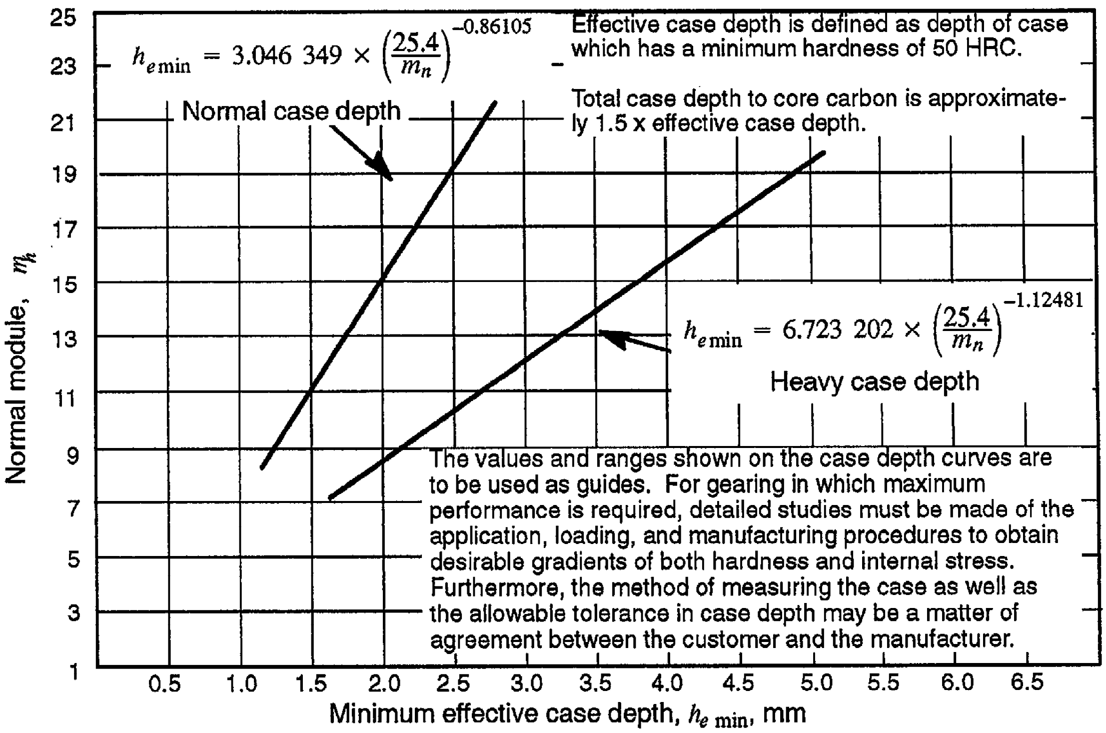


Figure 13 – Minimum effective case depth for carburized gears, $h_{e\,\min}$

$U_c = a + b\sigma_H + c\sigma_H^2 + d\sigma_H^3$

Curve No.	a	b	c	d
1	-5.542 507	1.425 472×10^{-2}	-8.658 576×10^{-6}	2.080 616×10^{-9}
2	-7.998 668	1.773 498×10^{-2}	-1.104 475×10^{-5}	2.577 360×10^{-9}
3	-9.391 585	2.009 680×10^{-2}	-1.230 944×10^{-5}	2.766 216×10^{-9}
4	-16.02 132	3.247 710×10^{-2}	-2.035 296×10^{-5}	4.474 037×10^{-9}
5	-40.64 486	8.094 566×10^{-2}	-5.221 782×10^{-5}	1.139 691×10^{-8}

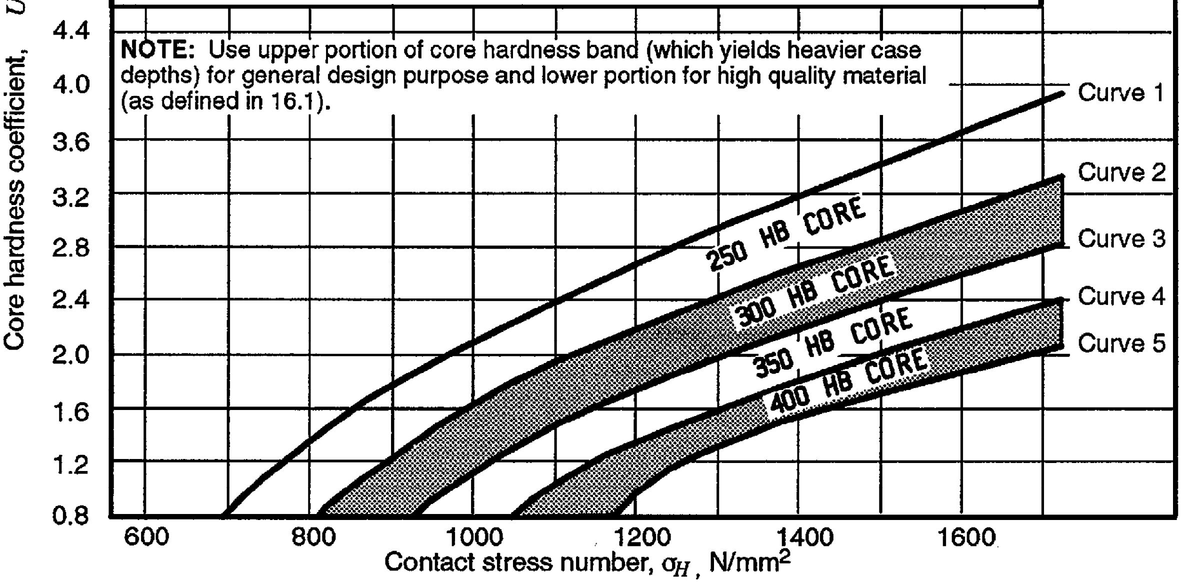


Figure 14 – Core hardness coefficient, U_c

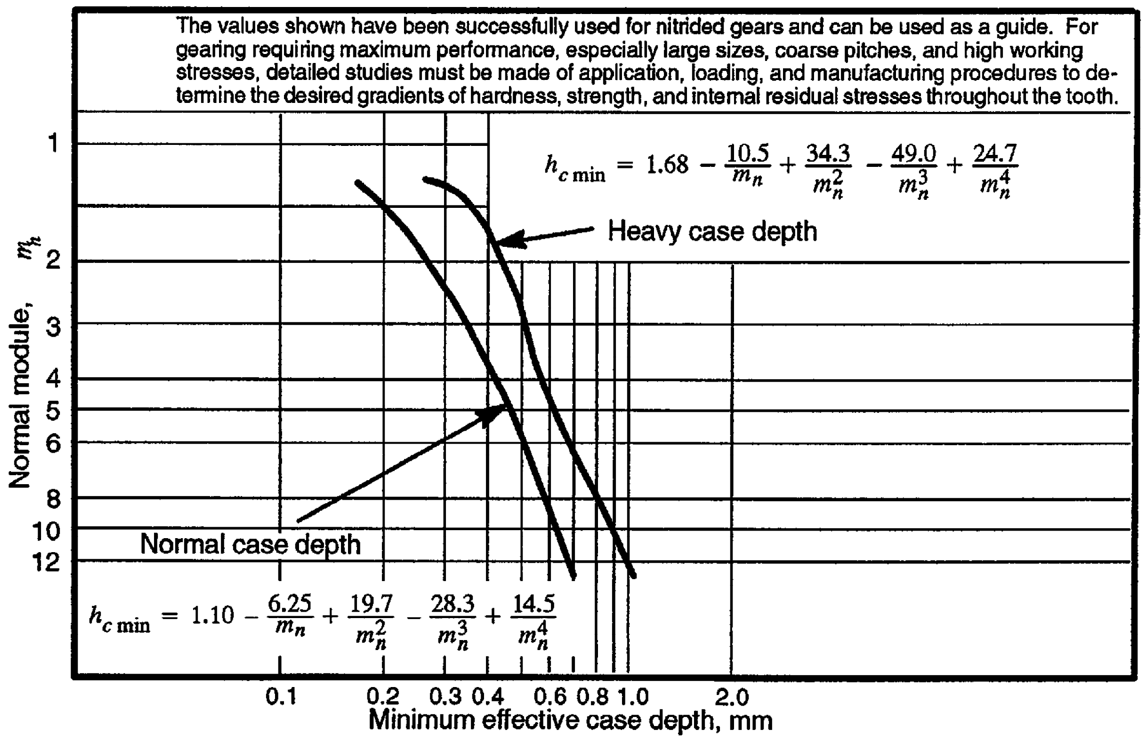


Figure 15 – Minimum total case depth for nitrided gears, $h_{c\,\min}$

16.2 Reverse loading

Use 70 percent of the σ_{FP} values for idler gears and other gears where the teeth are completely reverse loaded on every cycle.

16.3 Momentary overload

When the gear is subjected to infrequent (less than 100 cycles during the design life) momentary high overloads approaching yield, the maximum allowable stress is determined by the allowable yield properties rather than the bending fatigue strength of the material. This stress is designated as σ_s. Figure 16 shows suggested values for allowable yield strength for through hardened steel. For case hardened gears, the core hardness should be used in conjunction with figure 16. In these cases, the design should be checked to make certain that the teeth are not permanently deformed. When yield is the governing stress, the stress correction factor, K_f, is considered ineffective for ductile materials; hence, the stress correction factor can be taken as unity.

A momentary overload can cause an unusual face load distribution factor which will be influenced by the gear blank configuration and its bearing support. Special consideration, such as an approach similar to annex D, must be given to this condition when analyzing overloads. The empirical method of 15.3 shall not be used.

16.4 Yield strength

For through hardened gears up to 400 HB the factor K_y, shown in equation 45, can be applied to the yield strength of the material. These values must be applied at the maximum peak load to which the gears are subjected.

$$\sigma_s K_y \geq \frac{F_{\max} K_{Hs}}{b m_t Y_J K_f} \qquad \ldots(45)$$

where

σ_s is allowable yield strength number from figure 16, N/mm²;

K_y is yield strength factor from the following tabulation;

Requirements of application	K_y
Conservative practice	0.50
Industrial practice	0.75

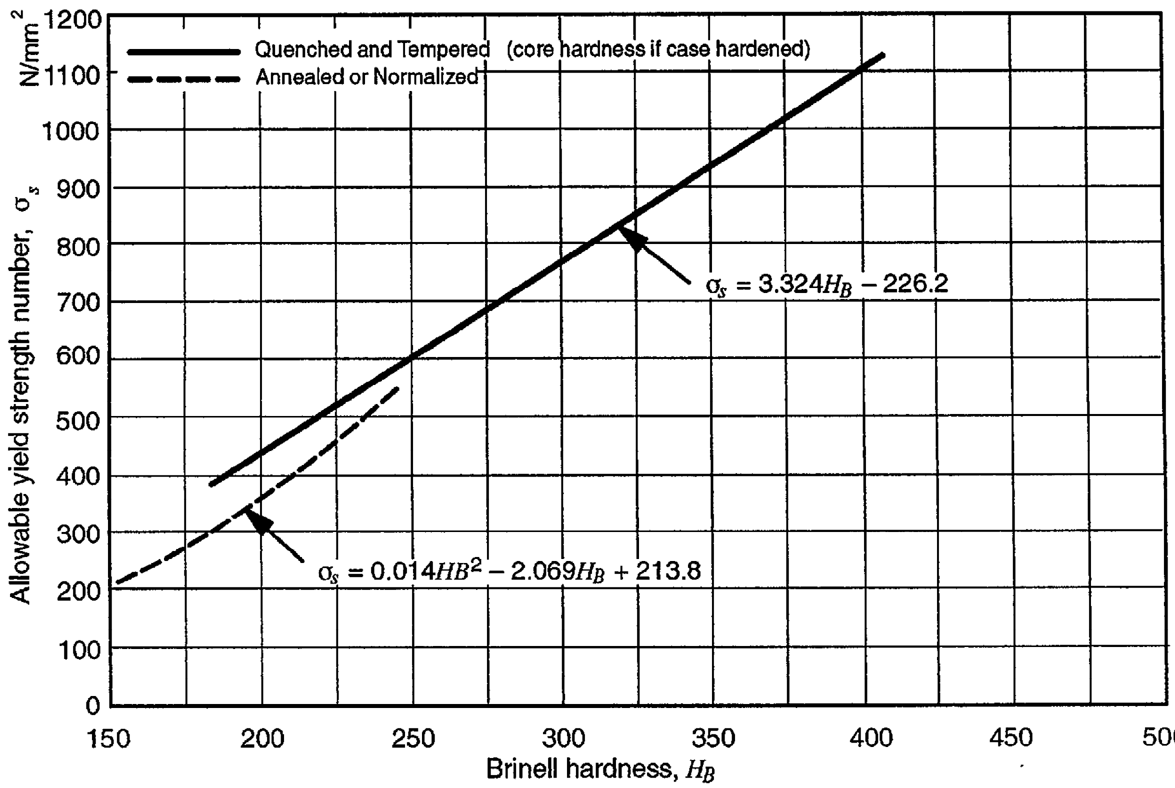


Figure 16 – Allowable yield strength number for steel gears, σ_s

F_{max} is maximum peak tangential load, N;

K_f is stress correction factor (see AGMA 908–B89);

K_{Hs} is load distribution factor under overload conditions.

CAUTION: This equation is based on a ductile material. For purposes of this standard, a material is considered ductile if the tensile elongation of the core material is at least 10%. For non–ductile materials, the effects of stress concentration should be considered.

For a straddle mounted gear or pinion in an enclosed drive, K_{Hs} may be calculated from equation 46.

$$K_{Hs} = 0.000\ 567 b + 1.07 \qquad \text{...(46)}$$

Equation 46 has been used as a design guide; a more detailed approach may give more accurate results.

For a case hardened gear, the analysis of allowable yield properties should include a stress calculation through a cross section of the material. In lieu of a cross section analysis, the use of material core hardness values can be used. For additional information, see [20].

17 Stress cycle factors, Z_N and Y_N

The stress cycle factors, Z_N and Y_N, adjust the allowable stress numbers for the required number of cycles of operation. For the purpose of this standard, n_L, the number of stress cycles is defined as the number of mesh contacts, under load, of the gear tooth being analyzed. AGMA allowable stress numbers are established for 10^7 unidirectional tooth load cycles at 99 percent reliability. The stress cycle factor adjusts the allowable stress numbers for design lives other than 10^7 cycles.

The stress cycle factor accounts for the S–N characteristics of the gear material as well as for the gradual increased tooth stress which may occur from tooth wear, resulting in increased dynamic effects and from shifting load distributions which may occur during the design life of the gearing.

When using a service factor, the determination of Z_N and Y_N shall be in accordance with clause 10.

17.1 Load cycles

When evaluating gearing, it is important to know how many stress cycles the individual gears will experience during the intended life of the equipment. Some machines will run twenty four hours per day and operate for twenty or more years. Other machines have gears that have a stress cycle equivalent to a few hours. The gear designer should design for the stress cycles that are appropriate for the application. The number of stress cycles, n_L, is used to determine the stress cycle factor as follows:

$$n_L = 60\, L \omega q \qquad \text{...(47)}$$

where

n_L is the number of stress cycles;

L is life (hours);

ω is speed (rpm);

q is number of contacts per revolution.

17.2 Stress cycle factors for steel gears

At the present time there is insufficient data to provide accurate stress cycle curves for all types of gears and gear applications. Experience, however, suggests stress cycle curves for pitting resistance and bending strength of steel gears as shown in figures 17 and 18. These figures do not include data for stainless steel gears. The shaded zones on the figures represent the influence of such items as pitch line velocity, material cleanliness, ductility and fracture toughness. The upper portion is for general applications. The lower portion is typically used for critical service where pitting and tooth wear must be minimal and low vibration levels are required.

17.3 Localized yielding

If the product of $\sigma_{FP}\, Y_N$ exceeds the allowable yield stress, σ_s, of figure 16, localized yielding of the teeth may occur. In some applications this is not acceptable. In others where profile and motion transmission accuracies are not critical, this may be acceptable for limited life.

The use of this standard at bending stress levels above those permissible for 10^4 cycles requires careful analysis. Stresses in this range may exceed the elastic limit of the gear tooth in bending stress. Depending on the material and the load imposed, a single stress cycle above the level limit at $< 10^4$ cycles could result in plastic yielding of the gear tooth.

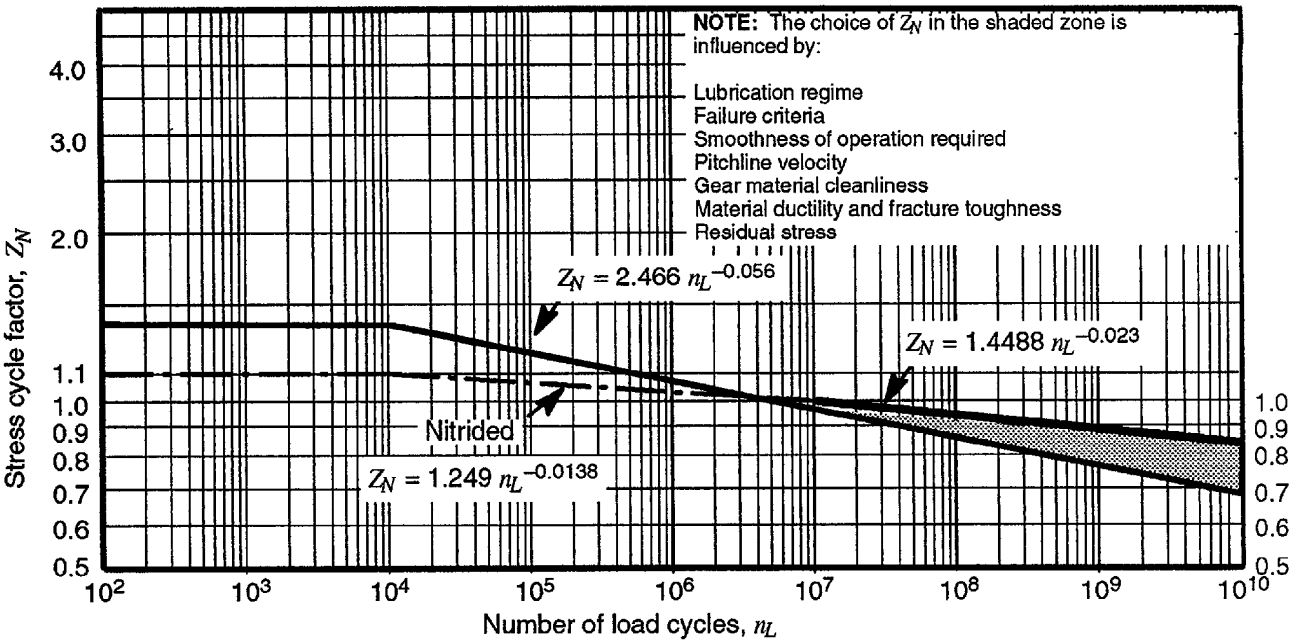


Figure 17 – Pitting resistance stress cycle factor, Z_N

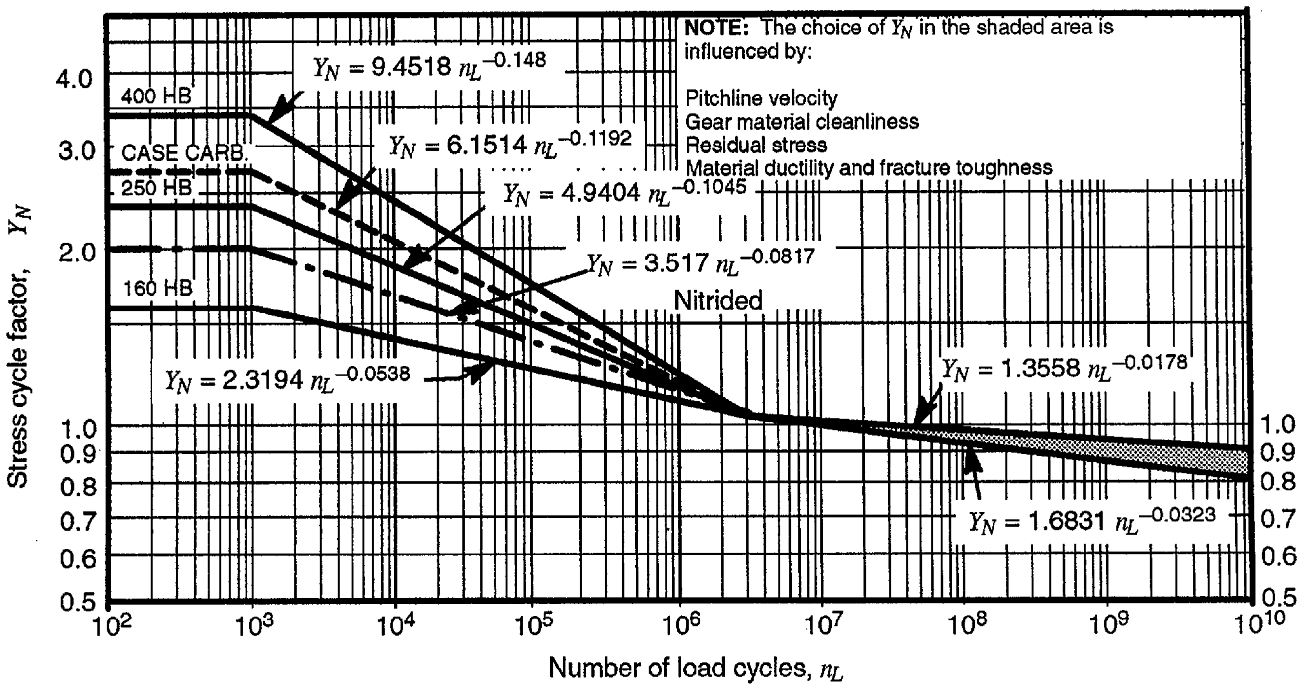


Figure 18 – Bending strength stress cycle factor, Y_N

18 Reliability factor, Y_Z

The reliability factors account for the effect of the normal statistical distribution of failures found in materials testing. The allowable stress numbers given in tables 3 through 6 are based upon a statistical probability of one failure in 100 at 10^7 cycles. Table 11 contains reliability factors which may be used to modify these allowable stresses to change that probability. These numbers are based upon data developed for bending and pitting failure by the U.S. Navy. Other values may be used if specific data is available.

Table 11 – Reliability factors, Y_Z

Requirements of application	Y_Z[1)]
Fewer than one failure in 10 000	1.50
Fewer than one failure in 1000	1.25
Fewer than one failure in 100	1.00
Fewer than one failure in 10	0.85[2)]
Fewer than one failure in 2	0.70[2) 3)]

NOTES
[1)]Tooth breakage is sometimes considered a greater hazard than pitting. In such cases a greater value of Y_Z is selected for bending.
[2)]At this value plastic flow might occur rather than pitting.
[3)]From test data extrapolation.

When strength rating is based on yield strength, σ_s, the values of K_y from 16.4 should be used instead of Y_Z.

19 Temperature factor, Y_θ

19.1 Moderate and low temperature operation

The temperature factor is generally taken as unity when gears operate with temperatures of oil or gear blank not exceeding 120°C. When operating temperatures result in gear blank temperatures below 0°C, special care must be given, see 3.6.1.

19.2 High temperature operation

When operating at oil or gear blank temperature above 120°C, Y_θ is given a value greater than 1.0 to allow for the effect of temperature on oil film and material properties.

Consideration must be given to the loss of hardness and strength of some materials due to the tempering effect of temperatures over 150°C.

20 Size factor, K_s

20.1 Size factor

The size factor reflects non-uniformity of material properties. It depends primarily on:

- Tooth size
- Diameter of parts
- Ratio of tooth size to diameter of part
- Face width
- Area of stress pattern
- Ratio of case depth to tooth size
- Hardenability and heat treatment of materials

Standard size factors for gear teeth have not yet been established for cases where there is a detrimental size effect. In such cases, some size factor greater than unity should be used.

20.2 Values for size factor

The size factor may be taken as unity for most gears, provided a proper choice of steel is made for the size of the part and its heat treatment and hardening process.

Annex A
(informative)
Method for evaluating the risk of scuffing and wear

This annex is for information only and should not be construed to be a part of ANSI/AGMA 2101–C95, *Fundamental rating factors and calculation methods for involute spur and helical gear teeth.*

A.1 Purpose

A standard method for evaluating the risk of scuffing and wear of spur and helical gearsets is described. The term scoring has been used in the past in the USA, while the term scuffing is used in Europe and in ISO standards to describe the severe form of adhesive wear which involves welding and tearing of the surfaces of gear teeth. To be consistent with current usage, the term scuffing is used in this annex where scuffing is defined as the localized damage caused by the occurrence of solid–phase welding between sliding surfaces. It is accompanied by the transfer of metal from one surface to another due to welding and tearing, and may occur in any sliding and rolling contact where the oil film is too thin to separate the surfaces. Its symptoms are a microscopically rough, matte, torn surface. Surface analysis that shows transfer of metal from one surface to the other is proof of scuffing.

> **NOTE:** The term scoring implies scratching and is defined as the formation of scratches in the direction of sliding. The word scoring is used in this annex to describe the clean–cut, parallel scratches in the direction of sliding that occur on gear teeth due to abrasion or micro–cutting caused by abrasives in the lubricant, or loose or attached, work–hardened particles.

The scuffing risk evaluation is based on Blok's [1] critical temperature hypothesis, and the wear risk evaluation is based on Dowson and Higginson's [2,3,4,5] equation for elastohydrodynamic (EHD) film thickness.

This annex is a supplement to ANSI/AGMA 2101–C95. It has been introduced to enable field experience data to be accumulated and evaluated relative to the calculation methods given. Such data will serve to enhance the future development of improved methods for evaluating scuffing and wear risk.

A.1.1 Lubrication regime

The specific film thickness, λ, defined as the ratio of the central film thickness to composite surface roughness, is a useful measure of the lubrication regime. For $\lambda > 1$, the contact between the tooth surfaces has an intermittent character with a percentage of contact time that decreases gradually as λ increases.

As an approximate guide, $\lambda > 2$ indicates full EHD lubrication, while $\lambda < 1$ indicates partial EHD or boundary lubrication regimes.

The specific film thickness cannot be used to predict the probability of scuffing, since thin EHD films are a necessary but insufficient condition for scuffing to occur. However, a thin film together with a high contact temperature suggests a high probability of scuffing in the absence of extreme pressure (EP) additives.

A.1.2 Mechanism of scuffing and wear

When gear teeth are completely separated by a thick film of lubricant, there is no contact between the asperities of the tooth surfaces, and usually there is no scuffing or wear.

For thinner elastohydrodynamic films, the degree of asperity contact increases and abrasive wear, adhesive wear or scuffing becomes possible. Abrasive wear may occur due to the rubbing action of the gear teeth or the presence of abrasive particles in the lubricant. Adhesive wear occurs by localized welding and subsequent detachment and transfer of particles from one or both of the gears. Abrasive or adhesive wear may not be harmful if it is mild and it subsides with time, as in a normal break–in process. Scuffing on the other hand, is a severe form of adhesive wear that can result in catastrophic damage to the gear teeth. The basic mechanism of scuffing is not clearly understood, but there is general agreement that it is caused by intense frictional heat that is generated by the combination of high sliding velocity and intense surface pressure. Scuffing occurs under thin film, boundary–lubricated conditions where the phenomenon is controlled by physical and chemical properties of the lubricant, oxide films and gear tooth materials.

A.1.3 Flash temperature and probability of scuffing

Blok's [1] contact temperature theory states that scuffing will occur in gear teeth that are sliding under boundary–lubricated conditions, when the maximum contact temperature of the gear teeth reaches a critical magnitude. The contact temperature is conceived as the sum of two components: the flash temperature and the bulk temperature.

Usually, the meshing position most critical in regards to scuffing is either in one of the two extreme end regions of the contact path or near the points of single tooth contact.

Prediction of the probability of scuffing is possible by comparing the calculated contact temperature with the permissible level of scuffing temperature. The limiting scuffing temperature can be recalculated from any gear scuffing test, or can be provided by field investigations.

For non-additive mineral oils, each combination of oil and rubbing materials has a critical scuffing temperature which is constant regardless of the operating conditions. It is believed that the critical scuffing temperature is not constant for synthetic and high-additive EP lubricants and it must be determined from tests which closely simulate the operating condition of the gearset.

A.1.4 Elastohydrodynamic lubrication and probability of wear

Dowson and Higginson [2,3,4,5] developed an equation for the central EHD film thickness which accounts for the exponential increase of the lubricant viscosity with pressure, tooth geometry, velocity of the gear teeth, elastic properties of the materials and the transmitted load. The film thickness determines the operating regime of the gearset and has been found to be a useful index of the wear related distress probability. Wellauer and Holloway [6], also, found that the specific film thickness could be correlated with the probability of tooth surface distress.

A.2 Symbols and units

The symbols used in this annex are shown in table A.1.

NOTE: The symbols and definitions used in this annex may differ from other AGMA standards.

Table A.1 – Symbols and units used in annex A

Symbol	Description	Units	First Used	Clause
a	Operating center distance	mm	Eq A.4	A.3.1
B_M	Thermal contact coefficient	N/[mm s$^{0.5}$K]	Eq A.60	A.7.1
B_{M1}	Thermal contact coefficient of pinion	N/[mm$^{0.5}$ m$^{0.5}$ s$^{0.5}$ K]	Eq A.66	A.7.4
B_{M2}	Thermal contact coefficient of gear	N/[mm$^{0.5}$ m$^{0.5}$ s$^{0.5}$ K]	Eq A.66	A.7.4
b	Face width	mm	Eq A.30	A.3.4
b_H	Semi-width of Hertzian contact band	mm	Eq A.58	A.6
$C_1 \ldots C_6$	Distances along line of action	mm	Eq A.17	A.3.2
C_D	Combined derating factor	—	Eq A.49	A.4
c_{M1}	Specific heat per unit mass, pinion	J/[kg K]	—	A.7.4
c_{M2}	Specific heat per unit mass, gear	J/[kg K]	—	A.7.4
D_i	Internal gear inside diameter	mm	Eq A.18	A.3.2
E_1, E_2	Modulus of elasticity (pinion, gear)	N/mm^2	Eq A.59	A.6
E_r	Reduced modulus of elasticity	N/mm^2	Eq A.58	A.6
F_t	Actual tangential load	N	Eq A.50	A.4
$(F_t)_{nom}$	Nominal tangential load	N	Eq A.48	A.4
F_{wn}	Normal operating load	N	Eq A.51	A.4
G	Materials parameter	—	Eq A.76	A.12
H_c	Dimensionless central film thickness	—	Eq A.76	A.12
h_c	Central film thickness	mm	Eq A.80	A.12
K	Flash temperature constant	—	Eq A.60	A.7.1
K_H	Load distribution factor	—	Eq A.49	A.4
K_o	Overload factor	—	Eq A.49	A.4
K_v	Dynamic factor	—	Eq A.49	A.4
L	Filter cutoff wavelength	mm	Eq A.81	A.12

(continued)

Table A.1 *(continued)*

Symbol	Description	Units	First Used	Clause
L_{min}	Minimum contact length	mm	Eq A.32	A.3.4
m_n	Normal module	mm	Eq A.2	A.3.1
n_1	Pinion speed	rpm	Eq A.41	A.4
n_a	Fractional part of ε_β	—	Eq A.32	A.3.4
n_r	Fractional part of ε_α	—	Eq A.32	A.3.4
P	Transmitted power	kW	Eq A.48	A.4
p_{bn}	Normal base pitch	mm	Eq A.10	A.3.1
p_{bt}	Transverse base pitch	mm	Eq A.9	A.3.1
p_x	Axial pitch	mm	Eq A.11	A.3.1
R_a	Arithmetic mean value for pinion and gear roughness	μm	Eq A.62	A.7.3.2
r_1	Standard pitch radius of pinion	mm	Eq A.2	A.3.1
r_2	Standard pitch radius of gear	mm	Eq A.3	A.3.1
r_{a1}	Outside radius of pinion	mm	Eq A.15	A.3.1
r_{a2}	Outside radius of gear	mm	Eq A.16	A.3.1
r_{b1}	Base radius of pinion	mm	Eq A.6	A.3.1
r_{b2}	Base radius of gear	mm	Eq A.7	A.3.1
r_{w1}	Operating pitch radius of pinion	mm	Eq A.4	A.3.1
S_B	Safety factor	—	Eq A.75	A.10.5
$S_{B\min}$	Minimum demand safety factor	—	Eq A.75	A.10.5
U	Speed parameter	—	Eq A.76	A.12
u	Gear ratio (always $\geq$1.0)	—	Eq A.1	A.3.1
v_e	Entraining velocity	m/s	Eq A.47	A.4
v_{r1}	Rolling velocity of pinion	m/s	Eq A.44	A.4
v_{r2}	Rolling velocity of gear	m/s	Eq A.45	A.4
v_s	Sliding velocity	m/s	Eq A.46	A.4
v_t	Operating pitch line velocity	m/s	Eq A.43	A.4
W	Load parameter	—	Eq A.76	A.12
w_n	Normal unit load	N/mm	Eq A.53	A.4
w_t	Transverse unit load	N/mm	Eq A.52	A.4
X_G	Geometry factor	—	Eq A.61	A.7.2
X_M	Thermal elastic factor	$KN^{-0.75}s^{0.5}m^{-0.5}mm$	Eq A.61	A.7.2
X_W	Welding factor	—	Eq A.74	A.10.3
X_Γ	Load sharing factor	—	Eq A.54	A.5
Z	Active length of line of action	mm	Eq A.23	A.3.2
z_1	Number teeth in pinion	—	Eq A.1	A.3.1
z_2	Number teeth in gear (positive)	—	Eq A.1	A.3.1
α	Pressure–viscosity coefficient	mm²/N	Eq A.77	A.12
α_{a1}	Transverse tip pressure angle pinion	—	Eq A.15	A.3.1
α_{a2}	Transverse tip pressure angle gear	—	Eq A.16	A.3.1
α_n	Normal pressure angle	—	Eq A.5	A.3.1
α_t	Transverse generating pressure angle	—	Eq A.5	A.3.1
α_{wn}	Normal operating pressure angle	—	Eq A.14	A.3.1
α_{wt}	Transverse operating pressure angle	—	Eq A.8	A.3.1
α_y	Pressure angle at arbitrary point	—	Eq A.24	A.3.2
β	Helix angle	—	Eq A.2	A.3.1

(continued)

Table A.1 *(concluded)*

Symbol	Description	Units	First Used	Clause
β_b	Base helix angle	—	Eq A.12	A.3.1
β_w	Operating helix angle	—	Eq A.13	A.3.1
Γ_y	Parameter on line of action	—	Eq A.24	A.3.3
$\Gamma_A ... \Gamma_E$	Parameter of points A ... E	—	Eq A.25	A.3.3
ε	Pinion roll angle	—	Eq A.36	A.3.3
$\varepsilon_1 ... \varepsilon_5$	Pinion roll angle at points 1 ... 5	—	Eq A.35	A.3.5
ε_α	Transverse contact ratio	—	Eq A.29	A.3.4
ε_β	Axial contact ratio	—	Eq A.30	A.3.4
η_{oil}	Dynamic viscosity of the oil at oil temperature	$N/mm^2.s$	Eq A.65	A.7.3.2
λ	Specific film thickness	—	Eq A.81	A.12
λ_M	Heat conductivity	N/s.K	—	A.7.4
λ_{M1}	Heat conductivity, pinion	N/s.K	—	A.7.4
λ_{M2}	Heat conductivity, gear	N/s.K	—	A.7.4
μ_m	Mean coefficient of friction	—	Eq A.60	A.7.1
μ_o	Absolute viscosity	cP	Eq A.78	A.12
ν_1, ν_2	Poisson's ratio (pinion, gear)	—	Eq A.59	A.6
ν_{40}	Kinematic viscosity at 40°C	mm^2/s	Eq A.73	A.10.1
ρ_1, ρ_2	Transverse radius of curvature(pinion, gear)	mm	Eq A.36	A.3.6
ρ_M	Density	kg/m^3	—	A.7.4
ρ_{M1}	Density, pinion	kg/m^3	—	A.7.4
ρ_{M2}	Density, gear	kg/m^3	—	A.7.4
ρ_n	Normal relative radius of curvature	mm	Eq A.40	A.3.6
ρ_r	Transverse relative radius of curvature	mm	Eq A.38	A.3.6
ρ_{rc}	Transverse relative radius of curvature at pitch point	mm	Eq A.39	A.3.6
σ	Composite surface roughness	µm	Eq A.81	A.12
σ_1, σ_2	Surface roughness, rms (pinion, gear)	µm	Eq A.63	A.7.3
θ_B	Contact temperature	°C	Eq A.71	A.9.1
$\theta_{B\max}$	Maximum contact temperature	°C	Eq A.72	A.9.2
θ_{fl}	Flash temperature	°C	Eq A.60	A.7.1
$\theta_{fl\max}$	Maximum flash temperature	°C	Eq A.70	A.7.6
$\theta_{fl\max,\,test}$	Maximum flash temperature of test gears	°C	Eq A.74	A.10.3
θ_M	Bulk temperature	°C	Eq A.70	A.8
$\theta_{M,\,test}$	Bulk temperature of test gears	°C	Eq A.74	A.10.3
θ_{oil}	Oil temperature	°C	Eq A.70	A.8.1
θ_S	Scuffing temperature	°C	Eq A.73	A.10.1
ω_1, ω_2	Angular velocity of pinion, gear	rad/s	Eq A.41	A.4

A.3 Gear geometry

This clause gives equations for gear geometry used to determine flash temperature and EHD film thickness. The following equations apply to both spur and helical gears; spur gearing is taken as a particular case with zero helix angle. Where double signs are used (e.g. ±), the upper sign applies to external gears and the lower sign to internal gears (see table A.1).

A.3.1 Basic gear geometry

Gear ratio

$$u = \frac{z_2}{z_1} \qquad \text{...(A.1)}$$

Standard pitch radii

$$r_1 = \frac{z_1 m_n}{2 \cos \beta} \qquad \text{...(A.2)}$$

$$r_2 = r_1 \, u \qquad \text{...(A.3)}$$

Operating pitch radius of pinion

$$r_{w1} = \frac{a}{u \pm 1} \qquad \text{...(A.4)}$$

Transverse generating pressure angle

$$\alpha_t = \arctan\left(\frac{\tan\alpha_n}{\cos\beta}\right) \quad \text{...(A.5)}$$

Base radii

$$r_{b1} = r_1 \cos\alpha_t \quad \text{...(A.6)}$$
$$r_{b2} = r_1 u \quad \text{...(A.7)}$$

Transverse operating pressure angle

$$\alpha_{wt} = \arccos\left(\frac{r_{b2} \pm r_{b1}}{a}\right) \quad \text{...(A.8)}$$

Transverse base pitch

$$p_{bt} = \frac{2\pi r_{b1}}{z_1} \quad \text{...(A.9)}$$

Normal base pitch

$$p_{bn} = \pi m_n \cos\alpha_n \quad \text{...(A.10)}$$

Axial pitch

$$p_x = \frac{\pi m_n}{\sin\beta} \quad \text{...(A.11)}$$

Base helix angle

$$\beta_b = \arccos\left(\frac{p_{bn}}{p_{bt}}\right) \quad \text{...(A.12)}$$

Operating helix angle

$$\beta_w = \arctan\left(\frac{\tan\beta_b}{\cos\alpha_{wt}}\right) \quad \text{...(A.13)}$$

Normal operating pressure angle

$$\alpha_{wn} = \arcsin(\cos\beta_b \sin\alpha_{wt}) \quad \text{...(A.14)}$$

Tip pressure angles

$$\alpha_{a1} = \arccos\left(\frac{r_{b1}}{r_{a1}}\right) \quad \text{...(A.15)}$$

$$\alpha_{a2} = \arccos\left(\frac{r_{b2}}{r_{a2}}\right) \quad \text{...(A.16)}$$

A.3.2 Distances along the line of action

Figure A.1 is the line of action shown in a transverse plane. Distances C_i are measured from the interference point of the pinion along the line of action. Distance C_1 locates the pinion start of the active profile (SAP) and distance C_5 locates the pinion end of the active profile (EAP). The lowest and highest point of single-tooth-pair contact (LPSTC and HPSTC) are located by distances C_2 and C_4 respectively. Distance $C3$ locates the operating pitch point.

$$C_6 = a \sin\alpha_{wt} \quad \text{...(A.17)}$$

$$C_1 = \pm\left[C_6 - \left(r_{a2}^2 - r_{b2}^2\right)^{0.5}\right] \quad \text{...(A.18)}$$

NOTE: For internal gears $r_{a2} = D_i/2$

$$C_3 = \frac{C_6}{u \pm 1} \quad \text{...(A.19)}$$
$$C_4 = C_1 + p_{bt} \quad \text{...(A.20)}$$

$$C_5 = \left(r_{a1}^2 - r_{b1}^2\right)^{0.5} \quad \text{...(A.21)}$$
$$C_2 = C_5 - p_{bt} \quad \text{...(A.22)}$$
$$Z = C_5 - C_1 \quad \text{...(A.23)}$$

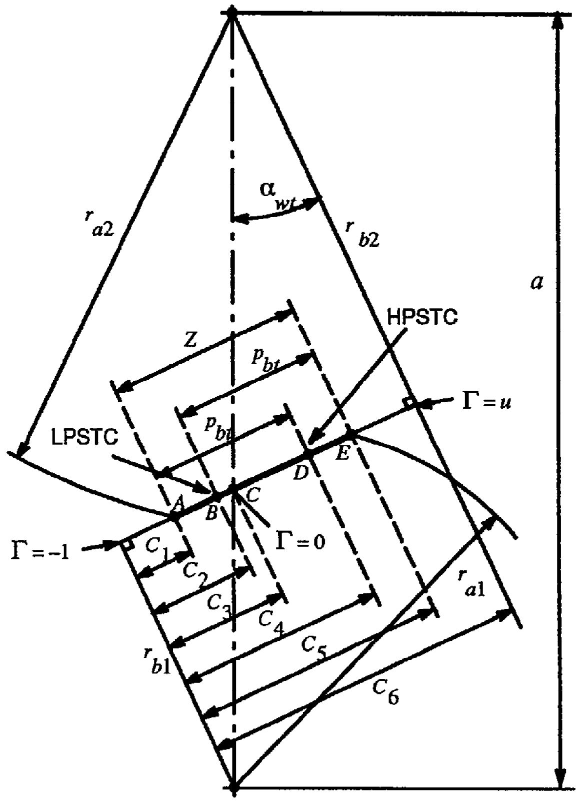


Figure A.1 – Distances along the line of action

A.3.3 Parameter along the line of action

The parameter, Γ_y, is defined as a dimensionless linear co-ordinate in the transverse plane on the line of action (see figure A.1), determined by:

Γ_y is -1 at the interference point of the pinion;

Γ_y is 0 at the pitch point

Γ_y is u at C_6.

for an arbitrary point on the line of action

$$\Gamma_y = \frac{\tan\alpha_y}{\tan\alpha_{wt}} - 1 \quad \text{...(A.24)}$$

for special points on the line of action

At the start of pinion root contact with gear tip (SAP)

$$\Gamma_A = \frac{-z_2}{z_1}\left(\frac{\tan\alpha_{a2}}{\tan\alpha_{wt}} - 1\right) \quad \text{...(A.25)}$$

At the lowest point of transverse single contact (LPSTC)

$$\Gamma_B = \Gamma_E - \frac{2\pi}{z_1 \tan\alpha_{wt}} \quad \text{...(A.26)}$$

At the highest point of transverse single contact (HPSTC)

$$\Gamma_D = \Gamma_A + \frac{2\pi}{z_1 \tan \alpha_{wt}} \quad \text{...(A.27)}$$

At the pinion tip end of active profile (EAP)

$$\Gamma_E = \frac{\tan \alpha_{a1}}{\tan \alpha_{wt}} - 1 \quad \text{...(A.28)}$$

A.3.4 Contact ratios

Transverse contact ratio

$$\varepsilon_\alpha = \frac{1}{2\pi}\left[z_2\left(\tan \alpha_{a2} - \tan \alpha_{wt}\right) \pm z_1\left(\tan\alpha_{a1} - \tan \alpha_{wt}\right)\right] \quad \text{...(A.29)}$$

n_r is fractional part of ε_α.

Axial contact ratio

for helical gears

$$\varepsilon_\beta = \frac{b \sin \beta}{\pi m_n} \quad \text{...(A.30)}$$

n_a is fractional part of ε_β

for spur gears

$$\varepsilon_\beta = 0.0 \quad \text{...(A.31)}$$

Minimum contact length

for helical gears, case 1, where $(1 - n_r) \geq n_a$

$$L_{\min} = \frac{\left(\varepsilon_\beta b\right) - \left(n_a n_r p_x\right)}{\cos \beta_b} \quad \text{...(A.32)}$$

for helical gears, case 2, where $(1 - n_r) < n_a$

$$L_{\min} = \frac{\varepsilon_\beta b - (1 - n_a)(1 - n_r)p_x}{\cos \beta_b} \quad \text{...(A.33)}$$

for spur gears

$$L_{\min} = b \quad \text{...(A.34)}$$

A.3.5 Roll angles

The pinion roll angles corresponding to the five specific points along the line of action shown in figure A.1 are given by:

$$\varepsilon_i = \frac{C_i}{r_{b1}} \quad \text{...(A.35)}$$

where

i is 1, 2, 3, 4, 5

A.3.6 Profile radii of curvature

Transverse radii of curvature

Figure A.2 shows the transverse radii of curvature, ρ_1 and ρ_2, of the gear tooth profiles at a general contact point defined by the distance Γ_y along the line of action.

$$\rho_1 = \frac{a \sin \alpha_{wt}}{u \pm 1}\left(1 \pm \Gamma_y\right) \quad \text{...(A.36)}$$

$$\rho_2 = \frac{a \sin \alpha_{wt}}{u \pm 1}\left(u \mp \Gamma_y\right) \quad \text{...(A.37)}$$

Transverse relative radius of curvature

$$\rho_r = \frac{\rho_1 \rho_2}{\rho_2 \pm \rho_1} \quad \text{...(A.38)}$$

Normal relative radius of curvature

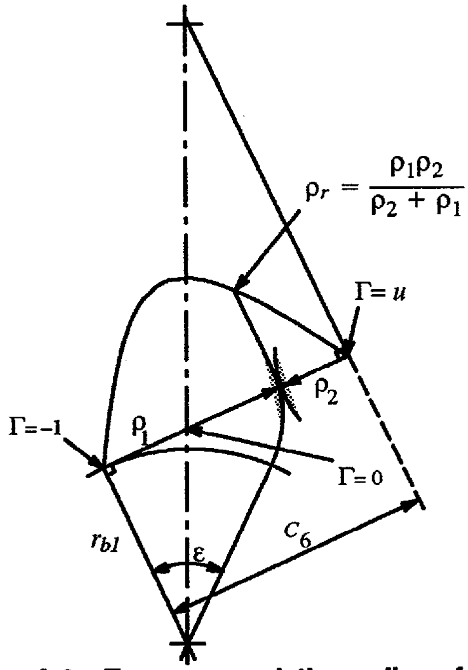


Figure A.2 – Transverse relative radius of curvature

The normal relative radius of curvature at the pitch point is given by:

$$\rho_{rc} = \frac{u}{(1 + u)^2}\, a\left[\frac{\sin \alpha_{wt}}{\cos \beta_b}\right] \quad \text{...(A.39)}$$

$$\rho_n = \frac{\rho_r}{\cos \beta_b} \quad \text{...(A.40)}$$

ρ_n is the equivalent radius of a cylinder that represents the gear pair curvatures in contact along the line of action.

A.4 Gear tooth velocities and loads

Rotational (angular) velocities

$$\omega_1 = \frac{\pi n_1}{30} \quad \text{...(A.41)}$$

$$\omega_2 = \frac{\omega_1}{u} \quad \text{...(A.42)}$$

Operating pitch line velocity

$$v_t = \frac{\pi n_1 a}{60(u \pm 1)} \quad \text{...(A.43)}$$

Rolling velocities

$v_{r1} = \omega_1 \rho_1$ of pinion ...(A.44)

$v_{r2} = \omega_2 \rho_2$ of gear ...(A.45)

Sliding velocity (absolute value)

$$v_s = | v_{r1} - v_{r2} | \quad \text{...(A.46)}$$

Entraining velocity (absolute value)

$$v_e = | v_{r1} + v_{r2} | \quad \text{...(A.47)}$$

Nominal tangential load

$$(F_t)_{\text{nom}} = \frac{1000P}{v_t} \quad \text{...(A.48)}$$

Combined derating factor

$$C_D = K_o K_H K_v \quad \text{...(A.49)}$$

Actual tangential load

$$F_t = (F_t)_{\text{nom}} \, C_D \quad \text{...(A.50)}$$

Normal operating load

$$F_{wn} = \frac{F_t}{\cos\alpha_{wn} \cos\beta_w} \quad \text{...(A.51)}$$

Transverse unit load

$$w_t = \frac{F_t}{L_{\min}} \quad \text{...(A.52)}$$

Normal unit load

$$w_n = \frac{F_{wn}}{L_{\min}} \quad \text{...(A.53)}$$

A.5 Load sharing factor

The load sharing factor accounts for load sharing between succeeding pairs of teeth as influenced by profile modification, and whether the pinion or gear is the driving member. Dynamic tooth forces due to relative displacements of the pinion and gear are considered separately with the dynamic factor. By convention, the load sharing factor is represented by a polygonal function on the line of action with magnitude equal to 1.0 between points B and D (see figure A.3).

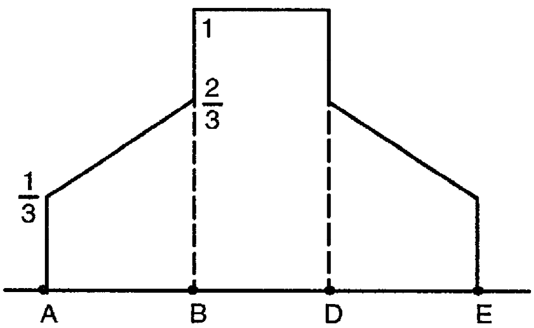


Figure A.3 – Unmodified profiles

The load sharing factor is strongly influenced by profile modification of the tooth flanks of both gears. On the other hand, profile modifications are chosen such that the load sharing follows a desired function. The following equations give the load sharing factor for unmodified tooth profiles, and for three typical cases of profile modifications.

For unmodified tooth profiles

$$\begin{aligned} X_\Gamma &= \frac{1}{3} + \frac{1}{3}\left(\frac{\Gamma_y - \Gamma_A}{\Gamma_B - \Gamma_A}\right) && \text{for } \Gamma_A \le \Gamma_y < \Gamma_B \\ X_\Gamma &= 1 && \text{for } \Gamma_B \le \Gamma_y \le \Gamma_D \\ X_\Gamma &= \frac{1}{3} + \frac{1}{3}\left(\frac{\Gamma_E - \Gamma_y}{\Gamma_E - \Gamma_D}\right) && \text{for } \Gamma_D < \Gamma_y \le \Gamma_E \end{aligned} \quad \text{...(A.54)}$$

For modified tooth profiles

If adequate tip and root relief is designed for high load capacity, and if the pinion drives the gear (see figure A.4):

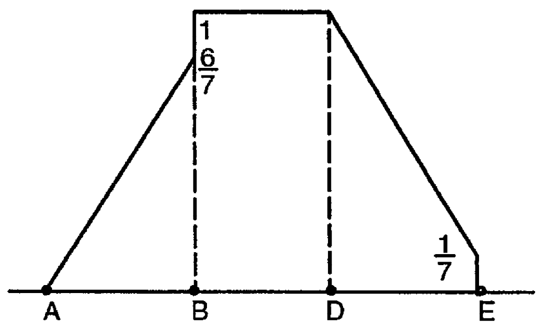


Figure A.4 – Pinion driving

$$\begin{aligned} X_\Gamma &= \frac{6}{7}\left(\frac{\Gamma_y - \Gamma_A}{\Gamma_B - \Gamma_A}\right) && \text{for } \Gamma_A \le \Gamma_y < \Gamma_B \\ X_\Gamma &= 1 && \text{for } \Gamma_B \le \Gamma_y \le \Gamma_D \\ X_\Gamma &= \frac{1}{7} + \frac{6}{7}\left(\frac{\Gamma_E - \Gamma_y}{\Gamma_E - \Gamma_D}\right) && \text{for } \Gamma_D < \Gamma_y \le \Gamma_E \end{aligned} \quad \text{...(A.55)}$$

If adequate tip and root relief is designed for high load capacity, and if the pinion is driven by the gear (see figure A.5):

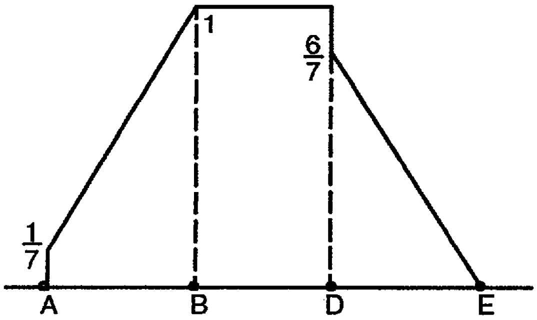


Figure A.5 – Gear driving

$$X_\Gamma = \frac{1}{7} + \frac{6}{7}\left(\frac{\Gamma_y - \Gamma_A}{\Gamma_B - \Gamma_A}\right) \quad \text{for } \Gamma_A \le \Gamma_y < \Gamma_B$$
$$X_\Gamma = 1 \quad \text{for } \Gamma_B \le \Gamma_y \le \Gamma_D$$
$$X_\Gamma = \frac{6}{7}\left(\frac{\Gamma_E - \Gamma_y}{\Gamma_E - \Gamma_D}\right) \quad \text{for } \Gamma_D < \Gamma_y \le \Gamma_E \qquad \text{...(A.56)}$$

If adequate tip and root relief is designed for smooth meshing (see figure A.6):

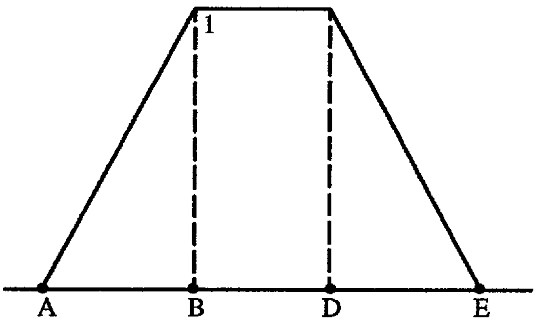


Figure A.6 Smooth meshing

$$X_\Gamma = \frac{\Gamma_y - \Gamma_A}{\Gamma_B - \Gamma_A} \quad \text{for } \Gamma_A \le \Gamma_y < \Gamma_B$$
$$X_\Gamma = 1 \quad \text{for } \Gamma_B \le \Gamma_y \le \Gamma_D \qquad \text{...(A.57)}$$
$$X_\Gamma = \frac{\Gamma_E - \Gamma_y}{\Gamma_E - \Gamma_D} \quad \text{for } \Gamma_D < \Gamma_y \le \Gamma_E$$

A.6 Hertzian contact band

The semi–width of the rectangular contact band is given by:

$$b_H = \left(\frac{8 X_\Gamma w_n \rho_n}{\pi\, E_r}\right)^{0.5} \qquad \text{...(A.58)}$$

where

- X_Γ is load sharing factor (see A.5);
- w_n is normal unit load (see Eq A.53);
- ρ_n is normal relative radius of curvature (see Eq A.40);
- E_r is reduced modulus of elasticity given by:

$$E_r = 2\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)^{-1} \qquad \text{...(A.59)}$$

where

- ν_1 is Poisson's ratio of the pinion;
- ν_2 is Poisson's ratio of the gear;
- E_1, E_2 is modulus of elasticity (pinion, gear).

A.7 Flash Temperature

A.7.1 Fundamental formula

The fundamental formula is based on Blok's [1] equation. In this equation, the coefficient of friction may be approximated by different expressions, for instance as proposed by Kelley [7,8] and AGMA 217.01 [9] . The influence of surface roughness is incorporated in the approximation of the coefficient of friction.

$$\theta_{fl} = K \mu_m \frac{X_\Gamma w_n}{B_M (b_H)^{0.5}} \left| (v_{r1})^{0.5} - (v_{r2})^{0.5} \right| \qquad \text{...(A.60)}$$

where

- K is 0.80,numerical factor valid for a semi–elliptic (Hertzian) distribution of frictional heat over the instantaneous width, 2 b_H, of the rectangular contact band;
- μ_m is mean coefficient of friction (see A.7.3.1);
- X_Γ is load sharing factor (see A.5);
- w_n is normal unit load (see Eq A.53);
- v_{r1} is rolling velocity of the pinion (see Eq A.44);
- v_{r2} is rolling velocity of the gear (see Eq A.45);
- B_M is thermal contact coefficient (see A.7.4);
- b_H is semi–width of Hertzian contact band (see Eq A.58).

A.7.2 Flash temperature equation

The fundamental formula may be used directly, or the formula may be rewritten, by applying the equations in A.3.1, A.4 and A.6, followed by concentrating certain parts of the formula into convenient factors. Commonly, the increasing influence of the generating pressure angle and the decreasing influence of the helix angle are negligibly small, as a result of which the geometry factor is only a function of the gear ratio and the parameter on the line of action. The flash temperature at any point on the line of action is:

$$\theta_{fl} = \mu_m X_M X_G \frac{(X_\Gamma w_t)^{0.75} v_t^{0.5}}{a^{0.25}} \qquad \text{...(A.61)}$$

where

- μ_m is mean coefficient of friction (see A.7.3);

X_M is thermal-elastic factor, (see A.7.4);

X_G is geometry factor (see A.7.5.);

X_Γ is load sharing factor (see A.5);

w_t is transverse unit load (see Eq A.52);

v_t is operating pitch line velocity (see Eq A.43);

a is center distance, mm.

A.7.3 Mean coefficient of friction

The mean coefficient of friction is an approximation of the actual coefficient of friction on the tooth flank, which is an instantaneous and local value depending on several properties of the oil, surface roughness, lay of the surface irregularities like grinding marks, material properties, tangential velocities, forces and dimensions.

A.7.3.1 Approximation by a constant

Apart from the influence of roughness a constant coefficient of friction has been assumed by AGMA 217.01 [9] and Kelley [7]:

$$\mu_m = 0.06 \left(\frac{1.13}{1.13 - R_a}\right) \qquad \text{...(A.62)}$$

Equation A.62 gives a typical value for gears operating in the partial EHD regime. It may be too low for boundary lubricated gears where μ_m may be higher than 0.2 or too high for gears operating in the full-film regime where μ_m may be less than 0.01.

The surface roughness is taken as an average of the rms values:

$$R_a = \frac{\sigma_1 + \sigma_2}{2} \qquad \text{...(A.63)}$$

where

σ_1 is pinion surface roughness (μm);

σ_2 is gear surface roughness (μm).

The surface roughness expression is limited to:

$$\frac{1.13}{1.13 - R_a} \leq 3.0 \qquad \text{...(A.64)}$$

A.7.3.2 Empirical equation

An empirical equation for a variable coefficient of friction is the Benedict and Kelley [10] equation, supplemented with the influence of roughness:

$$\mu = 0.0127 \left(\frac{1.13}{1.13 - R_a}\right) \log_{10}\left(\frac{3.17 \times 10^8\, X_\Gamma w_n}{\eta_{oil}\ v_s v_e^2}\right) \qquad \text{...(A.65)}$$

where the surface roughness expression is taken in accordance with Eq A.63 and Eq A.64. This equation is not valid at or near the operating pitch point, as v_s goes to zero.

where

η_{oil} is dynamic viscosity of the oil at oil temperature, (cp);

v_s is sliding velocity (see Eq A.46);

v_e is entraining velocity (see Eq A.47).

A.7.4 Thermal elastic factor

The thermal elastic factor accounts for the influence of the material properties of pinion and gear:

$$X_M = E_r^{0.25} \left[\frac{\left(1 + \Gamma_y\right)^{0.5} + \left(1 - \Gamma_y/u\right)^{0.5}}{B_{M1}\left(1 + \Gamma_y\right)^{0.5} + B_{M2}\left(1 - \Gamma_y/u\right)^{0.5}}\right] \qquad \text{...(A.66)}$$

where

E_r is reduced modulus of elasticity (see Eq A.59);

Γ_y is parameter on the line of action (see A.3.3);

B_{M1} is $(\lambda_{M1}\ \rho_{M1}\ c_{M1})^{0.5}$ thermal contact coefficient of the pinion material;

B_{M2} is $(\lambda_{M2}\ \rho_{M2}\ c_{M2})^{0.5}$ thermal contact coefficient of the gear material;

In most cases where the thermal contact coefficients are the same for the pinion and the gear, the thermal elastic factor depends solely on the material characteristics:

$$X_M = \frac{E_r^{0.25}}{B_M} \qquad \text{...(A.67)}$$

For martensitic steels the range of heat conductivity, λ_M, is 41 to 52 N/[s K] and the product of density times the specific heat per unit mass, $\rho_M \times c_M$ is about 3.8 N/[mm^{2}K] so that the use of the average value B_M = 13.6 N/[mm s$^{0.5}$ K] for such steels will not introduce a large error when the thermal contact coefficient is unknown. For gears made of representative steels, with E = 207 000 N/mm^2, ν = 0.3, and B_M = 13.6 N/[mm s$^{0.5}$ K], the following can be used:

$$X_M = 50.0\ KN^{-0.75}\, s^{0.5}\, m^{-0.5}\ \text{mm} \qquad \text{...(A.68)}$$

A.7.5 Geometry factor

The geometry factor, X_G, is determined as follows (see figure A.7 and A.8):

$$X_G = 0.51(u \pm 1)^{0.5} \frac{\left| (1+\Gamma_y)^{0.5} - (1 \mp \Gamma_y/u)^{0.5} \right|}{(1+\Gamma_y)^{0.25} (u \mp \Gamma_y)^{0.25}} \quad \text{...(A.69)}$$

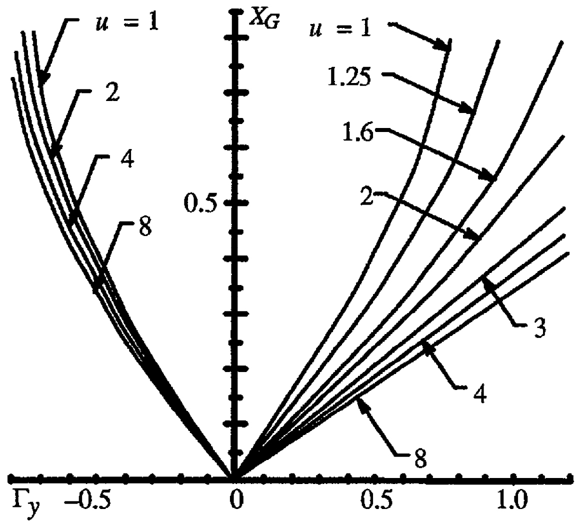


Figure A.7 – Geometry factor external gear

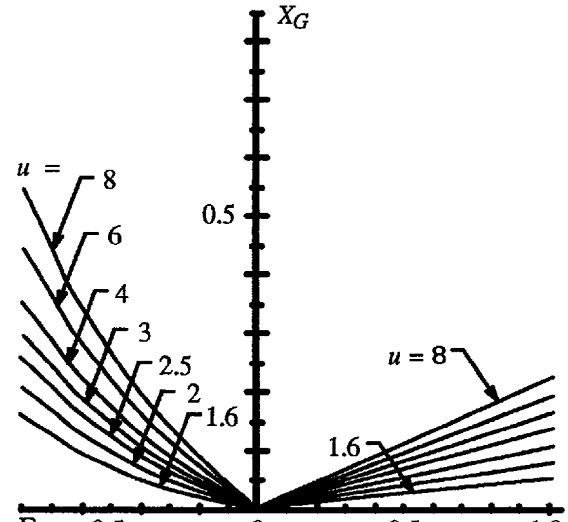


Figure A.8 Geometry factor internal gear

where

Γ_y is parameter on the line of action (see A.3.3);

u is gear ratio.

A.7.6 Maximum flash temperature

The maximum flash temperature $\theta_{fl\,max}$ is the highest of four peak values along the line of action.

The flash temperature should be calculated at a sufficient number of points on the line of action between the specific points, to determine all possible locations of the maximum flash temperature (between SAP and LPSTC during double tooth contact, at LPSTC and HPSTC for single tooth contact, and between HPSTC and EAP during double tooth contact).

The only variable in the computing process for the highest value is the parameter on the line of action. This parameter appears solely in the product of the geometry factor and the load sharing factor, $X_G(X_\Gamma)^{0.75}$. This product may be replaced by the product $X_G X_\Gamma$ if the approximated coefficient of friction depends on the local tooth load, which introduces a factor $(X_\Gamma)^{0.25}$.

A.8 Bulk temperature

The bulk temperature, θ_M, is the equilibrium temperature of the surface of the gear teeth before they enter the contact zone. In some cases, the bulk temperature may be significantly higher than the temperature of the oil supplied to the gear mesh.

A.8.1 Rough approximation

For a very rough approximation, the bulk temperature may be estimated by the sum of the oil temperature, taking into account some impediment in heat transfer for spray lubrication, and a portion that depends mainly on the flash temperature, for which the maximum value is taken:

$$\theta_M = 1.2\,\theta_{oil} + 0.56\,\theta_{fl\max} \quad \text{...(A.70)}$$

where

θ_{oil} is oil temperature, °C;

$\theta_{fl\max}$ is maximum flash temperature, °C, see A.7.

However, for a reliable evaluation of the scuffing risk, it is important that instead of the rough approximation, an accurate value of the gear bulk temperature be used for the analysis.

A.8.2 Measurement and experience

The bulk temperature can be measured by testing, or determined according to the experience of the gear manufacturer.

A.8.3 Thermal network

The bulk temperature can be calculated from a thermal network analysis (see figure A.9).

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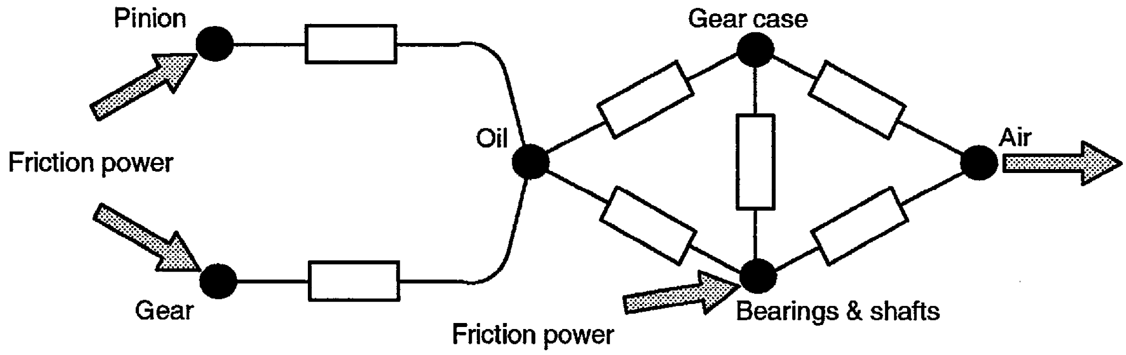

Figure A.9 – Example of thermal network

The bulk temperature is determined by the heat flow balance in the gear box. There are several sources of frictional heat, of which the most important ones are the tooth friction and the bearing friction. Other heat sources, like seals and the oil flow, may contribute to some extent. For gear pitchline velocities above 80 m/s, the churning loss, the expulsion of the lubrication oil between the meshing teeth, and the windage loss become important heat sources which should be considered. The heat is conducted and transferred to the environment by conduction, convection and radiation.

A.9 Contact temperature

A.9.1 Contact temperature at any point

At any point on the line of action (see figure A.10) the contact temperature is:

$$\theta_B = \theta_M + \theta_{fl} \qquad \text{...(A.71)}$$

where

θ_M is the bulk temperature (see A.8);

θ_{fl} is the flash temperature (see A.7).

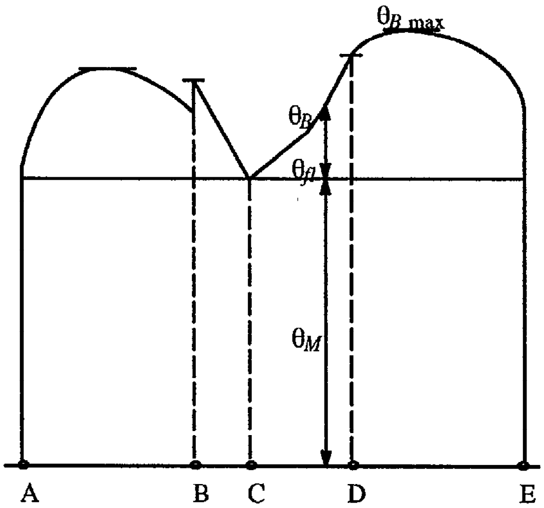

Figure A.10 – Contact temperature along the line of action

A.9.2 Maximum contact temperature

The maximum contact temperature is:

$$\theta_{B\max} = \theta_M + \theta_{fl\max} \qquad \text{...(A.72)}$$

where

θ_M is the bulk temperature (see A.8);

$\theta_{fl\max}$ is the maximum flash temperature (see A.7.6).

A.10 Scuffing temperature

The scuffing temperature is the contact temperature at which scuffing is likely to occur with the chosen combination of lubricant and gear materials. The scuffing temperature is assumed to be a characteristic value for the material–lubricant–material system of a gear pair, to be determined by gear tests with the same material–lubricant–material system.

A.10.1 Scuffing temperature for low–additive mineral oils

When using a low–additive mineral oil, the scuffing temperature is assumed to be independent of operating conditions in a fairly wide range. For such an oil and steel combination, the scuffing temperature may be correlated with the composition of the oil. The viscosity grade has been chosen as a convenient representative index of that composition, and thus of the scuffing temperature. Viscosity grade is suitably expressed in terms of kinematic viscosity. It is emphasized that viscosity grade is introduced as a readily available index of the different chemical composition of the various low–additive mineral oils, and is not to be conceived as an elastohydrodynamically influential characteristic.

$$\theta_S = 118 + 33 \ln \nu_{40} \text{ °C} \qquad \text{...(A.73)}$$

where

ν_{40} is kinematic viscosity at 40°C, mm²/s.

A.10.2 Scuffing risk evaluation

AGMA 217.01 [9] correlated the total contact temperature with the probability of scuffing for

MIL-L-7808 and MIL-L-6081, grade 1005 oils. The data were gathered from a survey of the aerospace industry and includes data for carburized and ground gears (predominantly made of SAE 9310 steel) from field applications and test rigs. The data were reduced using a Gaussian probability function. Table A.2 gives the mean scuffing temperature (50 percent chance of scuffing) and standard deviation of temperatures for MIL-L-7808 and MIL-L-6081 oils from AGMA 217.01. [9]

Table A.2 – MIL Lubricant mean scuffing temperatures

Lubricant	Mean scuffing temperature °C	Standard temperature deviation °C
MIL-L-7808	186	31.4
MIL-L-6081 (grade 1005)	129	41.8

Table A.3 can be used as an approximate guide for nonadditive mineral oils and steels typical of the IAE and FZG test machines. The mean scuffing temperature (50 percent chance of scuffing) was derived from data published by Blok [11]. The standard deviation temperature was assumed to be 15 percent of the mean value.

Table A.3 – Mineral oil mean scuffing temperatures

ISO VG	AGMA lube no.	Mean scuffing temperature °C	Standard temperature deviation °C
32	--	177	29
46	1	189	31
68	2	202	33
100	3	214	35
150	4	227	37
220	5	240	39
320	6	252	41
460	7	264	42
680	8	277	44
1000	8A	289	46
1500	--	303	48

A.10.3 Combination of mineral oil with gear steels

The scuffing temperature of low-additive mineral oils that is determined from test gears may be extended to different gear steels, heat treatments or surface treatments by introducing an empirical welding factor.

$$\theta_S = X_W\, \theta_{fl\max,\,\mathrm{test}} + \theta_{M,\,\mathrm{test}} \quad \text{...(A.74)}$$

where

X_W is welding factor (see table A.4);

$\theta_{fl\max,\,\mathrm{test}}$ is maximum flash temperature of test gears

$\theta_{M,\,\mathrm{test}}$ is bulk temperature of test gears

Table A.4 – Welding factors, X_W

Material	X_W
Through hardened steel	1.00
Phosphated steel	1.25
Copper-plated steel	1.50
Bath or gas nitrided steel	1.50
Hardened carburized steel	
– Less than 20% retained austenite	1.15
– 20 to 30% retained austenite	1.00
– Greater than 30% retained austenite	0.85
Austenite steel (stainless steel)	0.45

Table A.5 gives the evaluation of scuffing risk based on the probability of scuffing.[9]

Table A.5 – Scuffing risk

Probability of scuffing	Scuffing risk
<10%	Low
10 to 30%	Moderate
>30%	High

A.10.4 Scuffing temperature for high-additive mineral oil

When using a high-additive, extreme pressure mineral oil or a synthetic oil, extended research is still needed to determine the nature of a possible non-constancy of the scuffing temperature for the materials and operating conditions concerned. Special attention has to be paid to the correlation between test conditions and actual or design conditions. For instance, the operating conditions of a gear transmission are quite different from the operating conditions of disk tests.

A.10.5 Safety factor

A safety factor has to be introduced to account for inaccuracies in the calculation and to avoid unnecessary risks. In contrast to pitting and fatigue breakage, which show a distinct incubation period, a single short overload can lead to scuffing and failure of gears. The safety factor is defined as a quotient of oil temperature differences to establish a dimensionless factor independent of the temperature scale. The calculated safety factor shall not be less than the minimum demanded safety factor for contact temperature.

where

$$S_B = \frac{\theta_S - \theta_{oil}}{\theta_{B\max} - \theta_{oil}} \geq S_{B\min} \quad \text{...(A.75)}$$

where

θ_{oil} is oil temperature;

θ_S is scuffing temperature;

$\theta_{B\max}$ is maximum contact temperature (see A.7.6);

$S_{B\min}$ is minimum demanded safety factor.

A.11 Alternative scuffing risk evaluation

The Integral Temperature Method [14] has been proposed as an alternative to Blok's method for assessing the risk of scuffing. While Blok's method is based on a critical maximum temperature, the Integral Temperature Method proposes a critical energy level, and is based on integrating the temperature distribution along the path of contact.

For purposes of comparison, the integral temperature may be obtained by numerically integrating (e.g. using Simpson's Rule) the total conjunction temperature given by Eq A.71. Such comparisons have shown the following:

– Blok's method and the Integral Temperature Method give essentially the same assessment of scuffing risk for most gearsets.

– Blok's method and the Integral Temperature Method give different assessments of scuffing risk for those cases where there are local temperature peaks. These cases usually occur in gearsets that have low contact ratio, contact near the base circle or other sensitive geometries.

– Blok's method is sensitive to local temperature peaks because it is concerned with the maximum instantaneous temperature, while the Integral Temperature Method is insensitive to these peaks because it averages the temperature distribution.

A.12 Film thickness equation

The central EHD film thickness is based on the Dowson and Toyoda [5] equation:

Dimensionless central film thickness:

$$H_c = 3.06\, \frac{G^{0.56} U^{0.69}}{W^{0.10}} \quad \text{...(A.76)}$$

where the following are dimensionless parameters:

materials parameter, G

$$G = \alpha E_r \quad \text{...(A.77)}$$

speed parameter, U

$$U = \frac{\mu_o v_e}{2 E_r \rho_n} \quad \text{...(A.78)}$$

load parameter, W

$$W = \frac{X_\Gamma w_n}{E_r \rho_n} \quad \text{...(A.79)}$$

where

α is pressure – viscosity coefficient, (mm^2/N) ranges from 0.725×10^{-2} mm^2/N to 2.9×10^{-2} mm^2/N for typical gear lubricants. Data for pressure – viscosity coefficients versus temperature for typical gear lubricants is given in figure A.11. It has been based on data given in reference [13].

μ_o is absolute viscosity, cP. Figure A.12 gives average values of viscosity versus temperature for typical mineral gear oils with a viscosity index of 95. It has been adapted from reference [12];

It is important that the film thickness calculation be made using values of viscosity and pressure – viscosity coefficients that correspond to the gear bulk temperature.

The central film thickness is given by:

$$h_c = H_c \rho_n \quad \text{...(A.80)}$$

The specific film thickness [17] is given by:

$$\lambda = \frac{h_c}{\sigma} \left(\frac{L}{2 b_H} \right)^{0.5} \quad \text{...(A.81)}$$

where

σ *is* composite surface roughness given by:

$$\sigma = (\sigma_1^2 + \sigma_2^2)^{0.5} \quad \text{...(A.82)}$$

L is the filter cutoff wavelength used in measuring surface roughness.

Ideally, the cutoff wavelengths of the surface measuring instrument should be comparable to the width of the Hertzian contact band, $2b_H$. However, this may not be practical because many surface measuring instruments have fixed cutoff wavelengths (usually 0.8 mm).

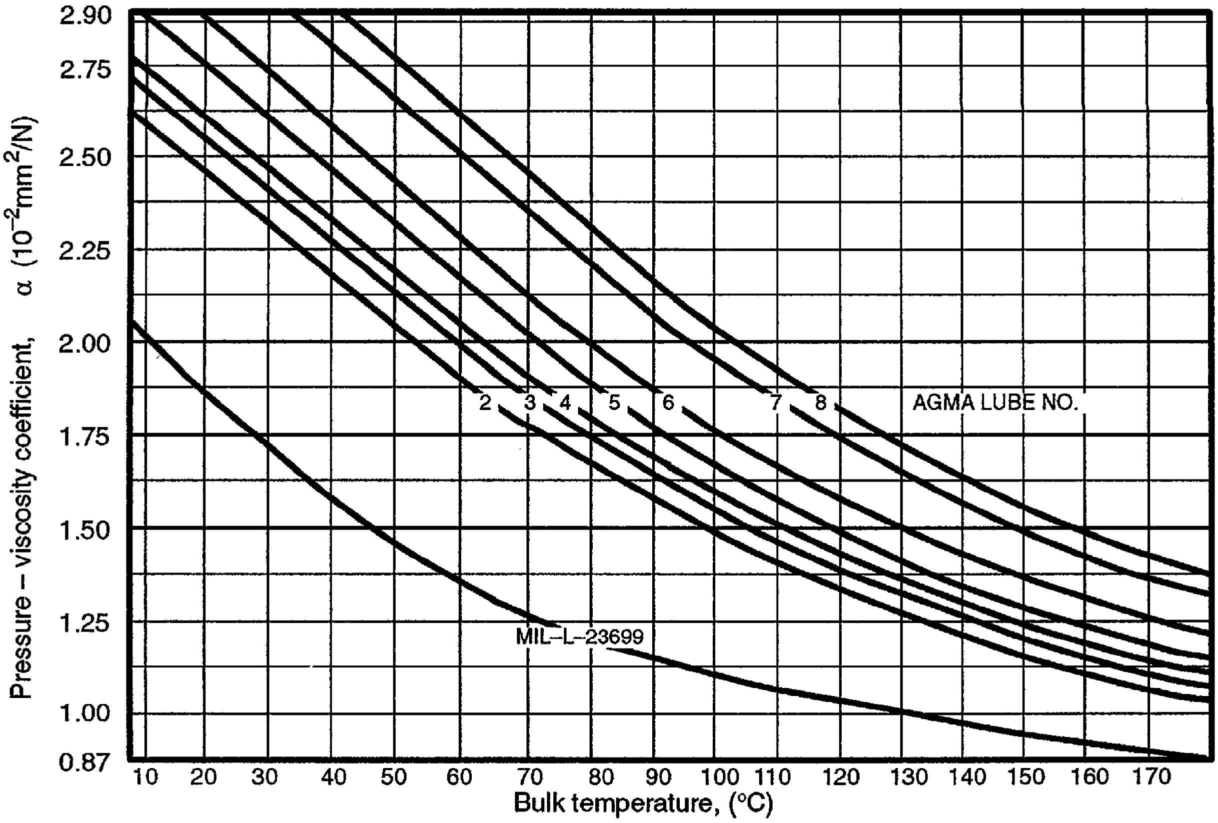

Figure A.11 – Pressure–viscosity coefficient versus temperature

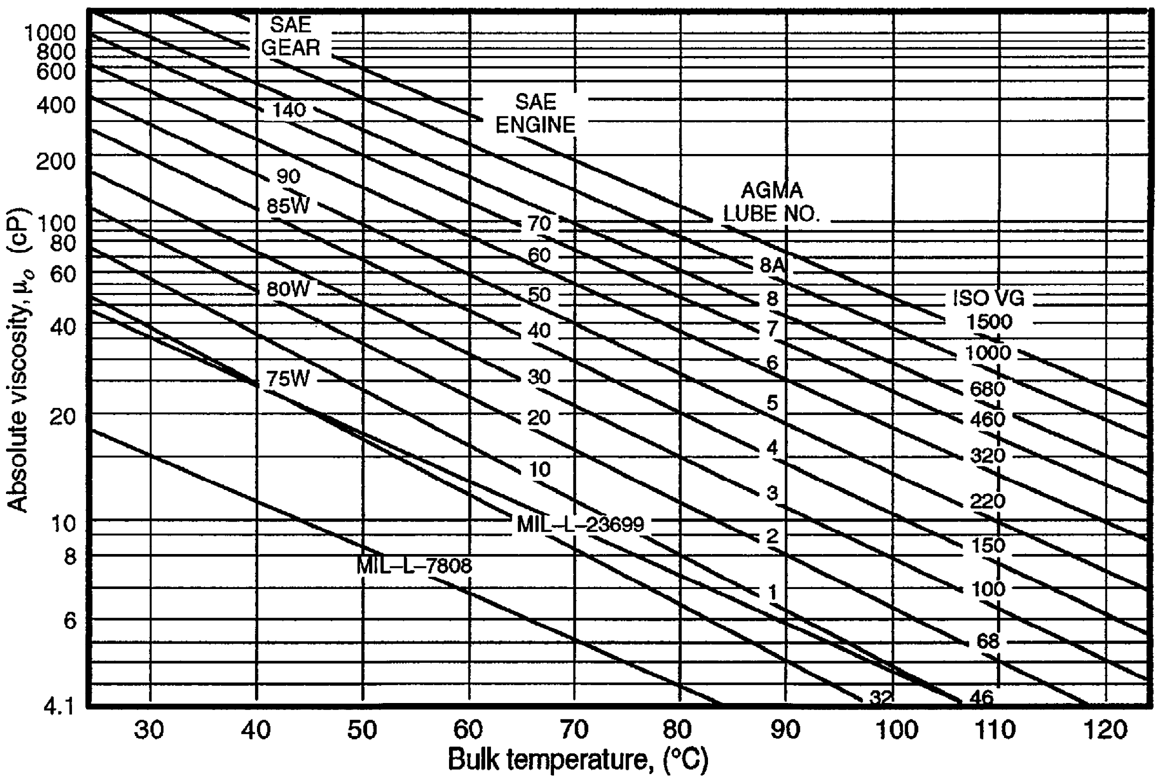

Figure A.12 – Absolute viscosity versus temperature

A.13 Wear risk evaluation

In the boundary lubrication regime, some wear is inevitable. Many gears, because of practical limits on lubricant viscosity, speed and temperature, must operate under boundary lubricated conditions.

Mild adhesive wear occurs during running–in and usually subsides with time, resulting in a tolerable wear rate and a satisfactory lifetime for the gearset. The wear that occurs during running–in may be beneficial if it smoothes the tooth surfaces (increasing the specific film thickness) and increases the area of contact by removing minor imperfections through local wear. The amount of wear that is tolerable depends on the expected lifetime for the gearset, and on requirements for the control of noise and vibration. The wear rate may become excessive if the tooth profiles are worn to the extent that high dynamic loads are encountered. Excessive wear may also be caused by contamination of the lubricant by abrasive particles. When wear becomes aggressive and is not preempted by scuffing (or bending fatigue), wear and pitting will likely compete for the predominate failure mode.

The boundary lubrication regime consists of exceedingly complex interactions between the additives in the lubricant, metal, and atmosphere making it impossible to assess accurately the chance of wear or scuffing from a single parameter such as the specific film thickness. However, the empirical data of figure A.13 has been used as an approximate guide to the probability of wear–related distress. Figure A.13 is based on data published by Wellauer and Holloway [6] which were obtained from several hundred laboratory tests and field applications. The curves of figure A.13 apply to through hardened steel gears ranging in size from 25 mm to 4600 mm in diameter that were lubricated with mineral gear lubricants. The authors [6] defined tooth surface distress as surface pitting or wear which might be destructive or could shorten the gear drive life. Most of the data of figure A.13 pertain to gears that experienced lives in excess of 10 million cycles. The curves of figure A.13 were adjusted to reflect the root–mean–square surface roughness. Reference [6] used the arithmetic average to determine composite surface roughness. They were also adjusted by assuming that the minimum film thicknesses calculated by reference [6] were 76% of the values given by Eq A.80 [5].

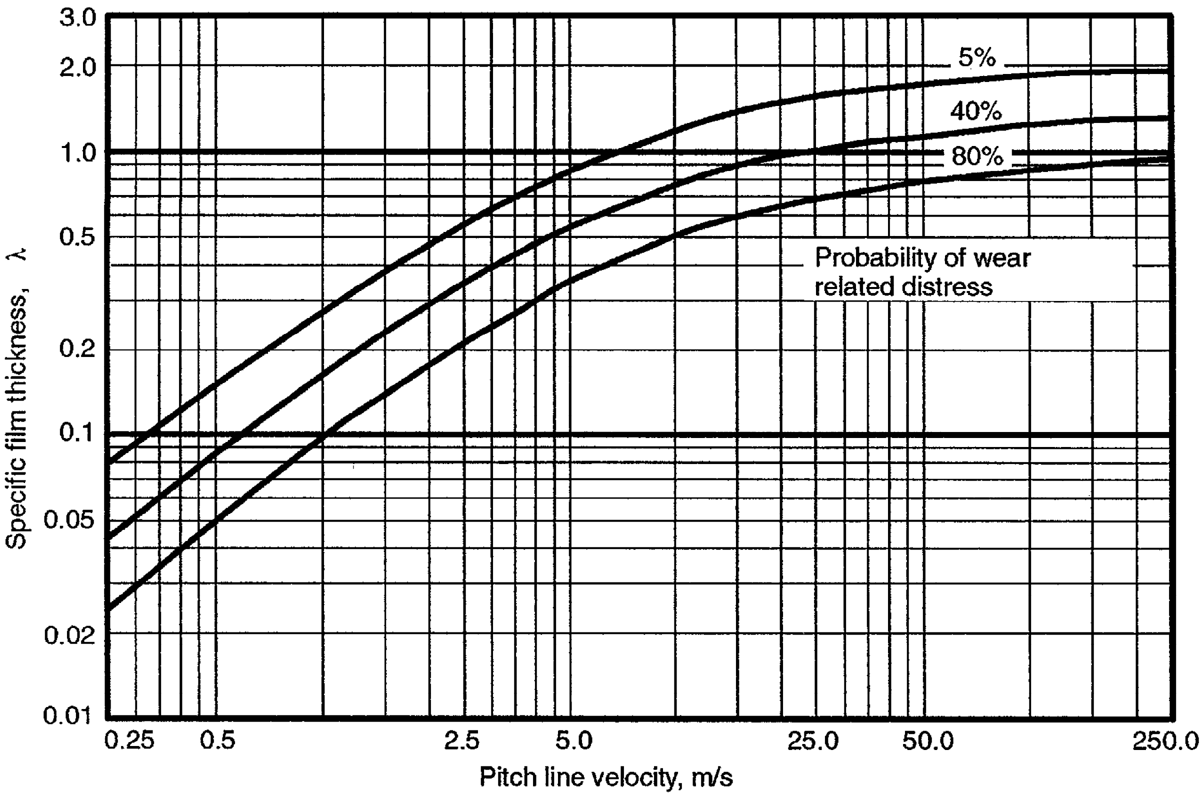

Figure A.13 – Probability of wear distress, percent

Annex A References

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3. Dowson, D., and Higginson, G.R., *Elastohydrodynamic Lubrication – The Fundamentals of Roller and Gear Lubrication,* Pergamon Press (London), 1966.

4. Dowson, D. *Elastohydrodynamics,* Paper No. 10., Proc., Inst. Mech. Engrs., Vol. 182, Pt. 3A, 1967, pp. 151–167.

5. Dowson, D., and Toyoda, S., *A Central Film Thickness Formula for Elastohydrodynamic Line Contacts,* 5th Leeds–Lyon Symposium Proceedings, Paper 11. (VII), 1978, pp. 60–65.

6. Wellauer, E. J. and Holloway, G.A., *Application of EHD Oil Film Theory to Industrial Gear Drives.* Trans ASME, J. Eng., Ind., Vol 98., series B., No 2. May 1976, pp 626–634.

7. Kelley, B.W., *A New Look at the Scoring Phenomena of Gears,* SAE trans, Vol. 61, 1953, pp 175–188.

8. Kelley, B.W., *The Importance of Surface Temperature to Surface Damage,* Chapter in *Engineering Approach to Surface Damage,* Univ. of Michigan Press, Ann Arbor, 1958.

9. AGMA 217.01, AGMA *Information Sheet – Gear Scoring Design for Aerospace Spur and Helical Power Gears,* October, 1965.

10. Benedict G. H. and Kelley B. W., *Instantaneous Coefficients of Gear Tooth Friction,* ASLE trans., Vol 4., 1961, pp 59–70.

11. Blok, H., *The Postulate About the Constancy of Scoring Temperature,* Interdisciplinary Approach to the Lubrication of Concentrated Contacts, NASA SP–237, 1970, pp. 153–248.

12. Neale, M. J., *Tribology Handbook,* Butterworths, (London), 1973.

13. Mobil Oil Corporation, *Mobil EHL Guidebook,* 1979.

14. Winter, H., and Michaelis, K., *Scoring Load Capacity of Gears Lubricated with EP–Oils,* AGMA Paper No. P219.17, October, 1983.

15. Blok, H., *The Thermal–Network Method for Predicting Bulk Temperatures in Gear Transmissions,* Proc. 7th Round Table Discussion on Marine Reduction Gears held in Finspong Sweden, 9–10 September 1969.

16. Blok H., *Thermo–Tribology – Fifty Years On,* keynote address to the Int. Conf. Tribology; Friction, Lubrication and Wear – 50 Years On, Inst. Mech. Engrs., London, 1–3 July 1987, Paper No. C 248/87.

17. Moyer, C. A., and Bahney, L.L., *Modifying the Lambda Ratio to Functional Line Contacts,* STLE Preprint No. 89–TC–5A–1, pp. 1–7.

Annex B
(informative)
Rim thickness factor, K_B

This annex is for information only and should not be construed to be a part of ANSI/AGMA 2101–C95 *Fundamental rating factors and calculation methods for involute spur and helical gear teeth.*

B.1 Purpose

This annex provides a method for considering the effects of gear blank rim thickness on the load carrying capacity of the gear tooth. It is based on Drago's[1] analysis of gear tooth bending fatigue strength.

This analysis shows that bending stresses in gear teeth are adversely affected when the rim thickness below the tooth root, t_R, is relatively thin as compared to the tooth height, h_t. Drago's analysis consisted of photo elastic models where the resulting curves were extrapolated from a limited number of test samples at backup ratios of 0.5 and 2.0 or greater. The study also included finite element analysis and data points from other research. A backup ratio of 2.0 indicated no effect on bending stress, but, an effect began to occur somewhere between 1.0 and 2.0. The curve in this annex is based on this analysis. But in view of the limited data, it is presented as two straight lines with a knee at 1.2. The knee was established based on experience of manufacturers who have successfully operated gears at rated loads with this backup ratio.

The rim thickness factor, K_B, is not sufficiently conservative for components with notches, hoop stresses or keyways. This is based on data for external gears with smooth bores and no notches or keyways.

The concern with notches (such as splines) or keyways in the bore of a gear is an increase in stress concentration which may lead to a fracture through the gear rim. Using large radii in the corners of the keyway (or spline) will help reduce the stress concentration and using a ductile (not brittle) material with good fracture toughness will also help. Another concern is press fitting the gear onto a shaft as this will induce stresses in the gear rim. The amount of effect of all of these items is beyond the scope of this standard.

B.2 Rim thickness factor, K_B

Where the rim thickness is not sufficient to provide full support for the tooth root, the location of bending fatigue failure may be through the gear rim, rather than at the tooth fillet. In such cases, the use of a stress modifying factor, K_B, is recommended.

This factor, entitled rim thickness factor, K_B, adjusts the calculated bending stress number for thin rimmed gears. It is a function of the backup ratio, m_B, or the ratio of the rim thickness below the tooth root, t_R, as compared to the tooth whole depth.

$$m_B = \frac{t_R}{h_t} \qquad \text{...(B.1)}$$

where

t_R is rim thickness below the tooth root, mm;

h_t is whole depth, mm.

Figure B.1 provides recommended values of K_B for backup ratios above 0.5. The effects of webs or stiffeners can be an improvement but are not accounted for. The effect of tapered rims has not been investigated. Ratios less than 0.5 require special analysis and is beyond the scope of this standard. When previous experience justifies, lower values of K_B may be used.

The rim thickness factor, K_B, is applied in addition to the 0.70 reverse loading factor where it is applicable (see 16.2).

1) Drago, R.J., *An Improvement in the Conventional Analysis of Gear Tooth Bending Fatigue Strength.* AGMA P229.24, October 1982.

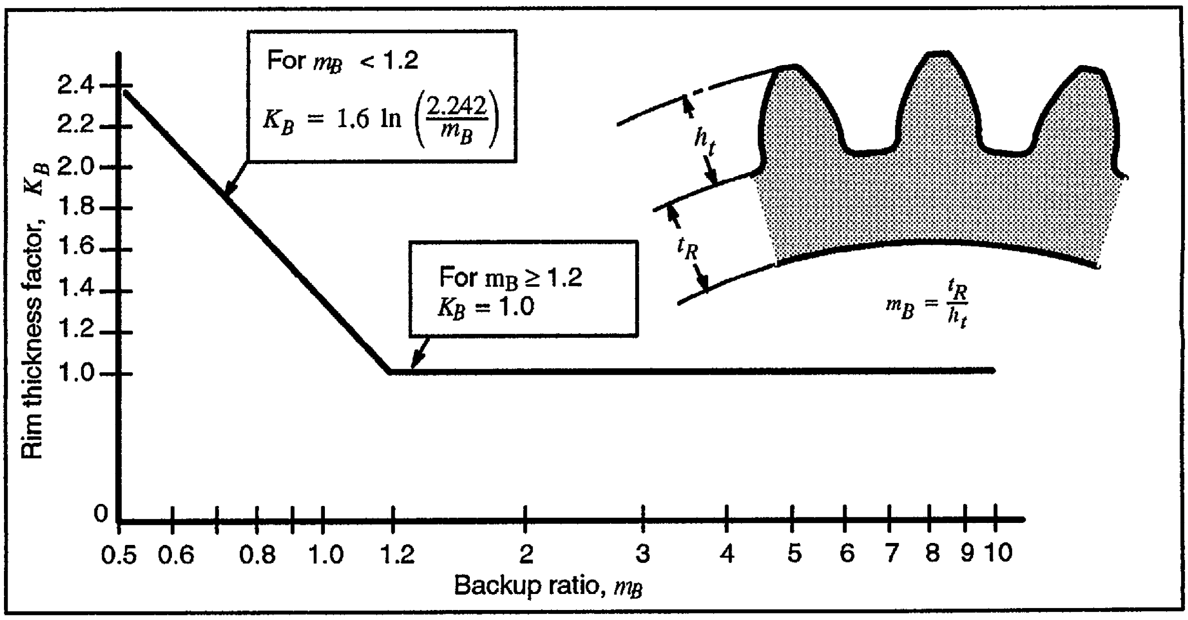

Figure B.1 – Rim thickness factor, K_B

Annex C
(informative)
Application analysis

This annex is for information only and should not be construed to be a part of ANSI/AGMA 2101–C95 *Fundamental rating factors and calculation methods for involute spur and helical gear teeth.*

C.1 Purpose

This annex discusses the use of factors of safety, overload factors, service factors and other considerations for geared systems.

C.2 Background

Many influence factors are used to determine the calculated load capacity of gears to account for various designs, manufacturing methods and uses of gears. Many of these factors have been empirically developed from accumulated experience. Therefore, it is critical that they be used in the manner originally intended. The influence factors are normally used as modifiers to either a calculated stress from part configuration and applied load or to an allowable stress number based on material properties. The gear designer can then compare the modified calculated stress to the modified allowable stress number for a specific design. In any design it is critical to make allowances for unknown variables in materials, machining tolerances, loading. etc. Various terms (factor of safety, service factor, and overload factor) are used in the gear industry to describe this important concept.

The designer, manufacturer, buyer, and user must all have a clear understanding of the meaning and implications of these terms when comparing gear capacity using different standards. The following definitions are given to explain the differences between these terms as applied to gearing:

C.3 Factor of safety

The term "factor of safety" has historically been used in mechanical design to describe a general derating factor to limit the design stress in proportion to the material strength. A factor of safety accounts for uncertainties in:

- Design analysis
- Material characteristics
- Manufacturing quality

Factor of safety also must consider human safety risk and the economic consequences of failure. The greater the uncertainties or consequences of these considerations, the higher the factor of safety should be. As the extent of these factors become known with more certainty, the value of the factor of safety can be more accurately determined. For example, a product such as an automobile transmission which is subjected to full size, full load prototype testing and rigorous quality control of dimensions, materials and processes during manufacture, could have a more precise safety factor than a hoist made in small quantities to normal commercial practices.

As design practices become more comprehensive, some influence factors have been removed from the unknown area of "safety factor" and introduced as predictable portions of the design method. The reliability factor, Y_Z, is an example.

NOTE: Factor of safety has also been used historically to account for uncertainties in "applied loading" or unknown overloads. In gear design, however, service factors or overload factors have been used for this uncertainty.

C.4 Overload factor

An overload factor makes allowance for any externally applied loads in excess of the nominal transmitted load. Overload factors are established only after considerable field experience is gained. In determining the overload factor, consideration should be given to the fact that systems develop momentary peak torques appreciably greater than those determined by the nominal ratings of the prime mover or driven equipment. Also, there are many possible sources of other overloads that should be considered, such as system vibrations, acceleration torques, overspeeds, variations in system operation, split–path load sharing among multiple prime movers, and changes in process applied load conditions.

C.5 Service factor

A service factor is traditionally applied as a multiplier of the nominal application load to determine catalog selections of pre–designed gear units. In AGMA gear rating the service factor has been used to include the combined effects of required life cycles, material reliability, and overload factors in an empirically determined single influence factor. The specific mathematical contribution of each of these items has not been satisfactorily established. In addition, the term "service factor" has been used when including human safety or economic risk, which has developed confusion between the terms factor of safety, overload factor, and service factor.

To avoid confusion, it is recommended that the overload factor be used as defined – for external variability in applied loading. A factor of safety should be applied where there is human risk, economic risk, or remaining uncertainties due to design, material, or manufacturing quality variation.

When an overload factor is used, consideration must be given to the effect of long service life on allowable stress levels.

A service factor should be applied only to a gear assembly and then only in the absence of more specific application load data. In addition, a service factor is only valid with the calculation method used at the time it was developed. It should not be used with other gear calculation methods, unless there is sufficient knowledge and experience to make a satisfactory conversion between methods.

C.6 Other considerations

Other important considerations in the design analysis of gear drive systems which are related to factor of safety, overload factor, and service factor selection are:

C.6.1 Test and experience

The proper selection of overload factors and factors of safety for any power transmission system often are not given enough attention. Without complete testing and field experience on each specific design, the application of gears has many unknowns. Therefore, conservative selection of all gear capacity calculation influence factors is recommended unless operating experience of an identical design is known.

C.6.2 Thermal rating

The thermal power rating of a gear system is defined as the power that the unit will transmit continuously without exceeding established temperature limits. This important consideration is necessary to maintain proper lubrication. Excessive temperatures are detrimental to the lubrication of gear teeth, such that the system may not be able to transmit the rated power without excessive wear and failure.

C.6.3 Non-gear components

Every component of a gear unit must allow for the proper transmission of power, considering both internal and external loading. These components, such as housing supports, shafting, bearings, and fasteners (bolts, nuts, etc.) must be designed and manufactured to maintain the gears in proper position as well as transmit the required power.

C.6.4 Gear quality

The term "quality" can have a number of meanings. In reference to gear manufacture, it is generally used to classify the tolerances applied to the gear tooth geometry. Unless the appropriate gear quality level is used to calculate the power rating of a gear system and that quality level is, in fact, duplicated or exceeded in manufacturing, the unit produced may not have the desired life.

C.6.5 Variation in manufacture

In addition to gear geometry, the metallurgical quality of all stressed parts and the geometrical accuracy of all other components of the drive must exceed the values assumed in the design calculations and test units.

These items in particular, and others in general, are addressed in some standards. Other standards do not mention these topics or, if mentioned, do not cover them thoroughly. It is important to know that factors contained within some AGMA standards, such as a service factors, should not be abstracted and applied to other standard methods of calculating gear capacity. Mixing factors from different standards can result in an inadequate design.

C.7 Summary

In gear design and rating there is a need for the use of factor of safety, service factor, and overload factor. These terms must be clearly defined when they are used. As the uncertainties in design, materials, manufacturing, and loading become known:

- the factor of safety can be reduced toward unity;
- overload factors will represent actual loading or be replaced by a load spectrum analysis, such as Miner's Rule;
- service factors may be replaced with factor of safety, overload factor, stress cycle and reliability factor properly used.

It must be clearly stated that the gear design or analysis must properly account for these uncertainties, based on experience. This is the primary responsibility of the gear engineer.

Annex D
(informative)
Discussion of the analytical face or longitudinal load distribution factor

This annex is for information only and should not be construed to be a part of ANSI/AGMA 2101–C95, *Fundamental rating factors and calculation methods for involute spur and helical gear teeth.*

D.1 Purpose

This annex provides the background information for the development of an accurate assessment of the load distribution across the face width of spur or helical gears.

D.2 Definition

The face load distribution factor is defined as the ratio of the peak load to the average load applied across the face width of a gear. The calculation of an accurate analytical load distribution across the face width of a spur or helical gear is a very complex and tedious process with many influencing factors. The calculation lends itself to computer programs that are dedicated to the task. A correct analytical determination of load distribution across the face width would yield variations in stress across the width of the gear that could be measured with properly applied strain gauges. This stress distribution although analytically and physically correct would not be identical to that predicted by the empirical rating techniques currently utilized in the ANSI/AGMA 2101–C95.

D.3 Empirical versus analytical method

The current rating practice of ANSI/AGMA 2101–C95 is as much empirical as it is analytical. Rating parameters were developed based on extensive testing of gears in service and in test applications. The techniques utilized in the empirical approach for load distribution of ANSI/AGMA 2101–C95 are consistent with this empirical approach, to rating gears. A rigorous analytical approach as will be described later in this annex, can in extreme cases yield results that dramatically derate the capacity of gears as currently rated by ANSI/AGMA 2101–C95. As the basic rating standard evolves to a more correct analytical and physical assessment, the analytical technique to be described will be compatible with this type of overall analysis.

D.4 Influencing parameters

There are many parameters that influence the actual load distribution across the face width of a gear. The influencing parameters can be categorized into four groups, all of which are normal to the manufacturing process but still cause face misalignments of the mating gear teeth. The groups are listed in clause 15 of ANSI/AGMA 2101–C95.

D.5 Guide to the analysis

In a complete analysis the expected values for all basic manufacturing variations of the gearing, housings, and bearings can be estimated and used as an initial gap across the face width of the gears, see figure D.1. In many instances an adjustment feature is provided in the gear assembly such that these variations causing misalignment can be negated. Sometimes the gearing is reground after initial pattern checks to correct for the assembly variations. Sometimes the bores of bearings or housing are scraped and sometimes an eccentric cartridge can provide an effective means of obtaining initial alignment of the mating gears. At this stage the gears are assumed to be initially parallel with no gap if adjustments are planned to be made or the expected gap is combined with the other factors to be determined.

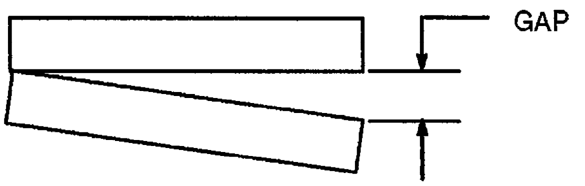

Figure D.1 – Gap due to manufacturing or installation errors

Now the elastic deflections, due to loading, must be dealt with. Two important notes on deflections are that they will be different for each load condition applied to the gears and that the gears can have leads that are intentionally modified to correct for elastic deflections. Ideally this modification would create a lead that is the mirror image of the deflected gear, see figure D.2. Normally the worst loading case is analyzed which will give the greatest mismatch between mating gear elements. The torsional and bending deflections can be calculated by normal strength of material techniques for each rotor. These deflections or gaps can be combined by superposition techniques with the initial misalignment gaps if they haven't been corrected by final adjustment. Centrifugal and thermal distortion should be determined similarly and also combined to give the final total distortion or gap between the mating gear flanks. If the gear teeth were infinitely stiff contact would occur at the intersection of the high point of the mating members causing an exceedingly high localized load. However, as load is applied, cantilever bending deflection of the gear tooth as well as Hertzian deflection occurs and this localized contact is spread across some percentage of the active face width. The amount that the contact spreads and the load variation across the face depends on the applied load, the tooth stiffness and the initial mismatch. Severely misaligned gears would show an extremely localized contact in a no load soft blue type of contact check.

A major problem occurs in the calculation of the deflections. The load distribution curve is needed to calculate the actual deflections but this curve cannot be calculated accurately until the deflection is known. The best solution to this problem is to make an estimate of the load distribution and use this to calculate the actual deflection and iterate on this technique until the assumed load distribution curve and the actual agree within some reasonable tolerance. The final values are plotted and K_H is calculated. This technique is presented in references [1] and [2]. Tooth stiffness values in the range of 1.0×10^4 to 2.1×10^4 N/mm^2 are typically used for determining the actual load distribution by this technique. This iterative type of solution is well suited to computer analysis.

[1] Dudley Darle W. – Practical Gear Design

[2] MAAG Gear Handbook, January 1990.

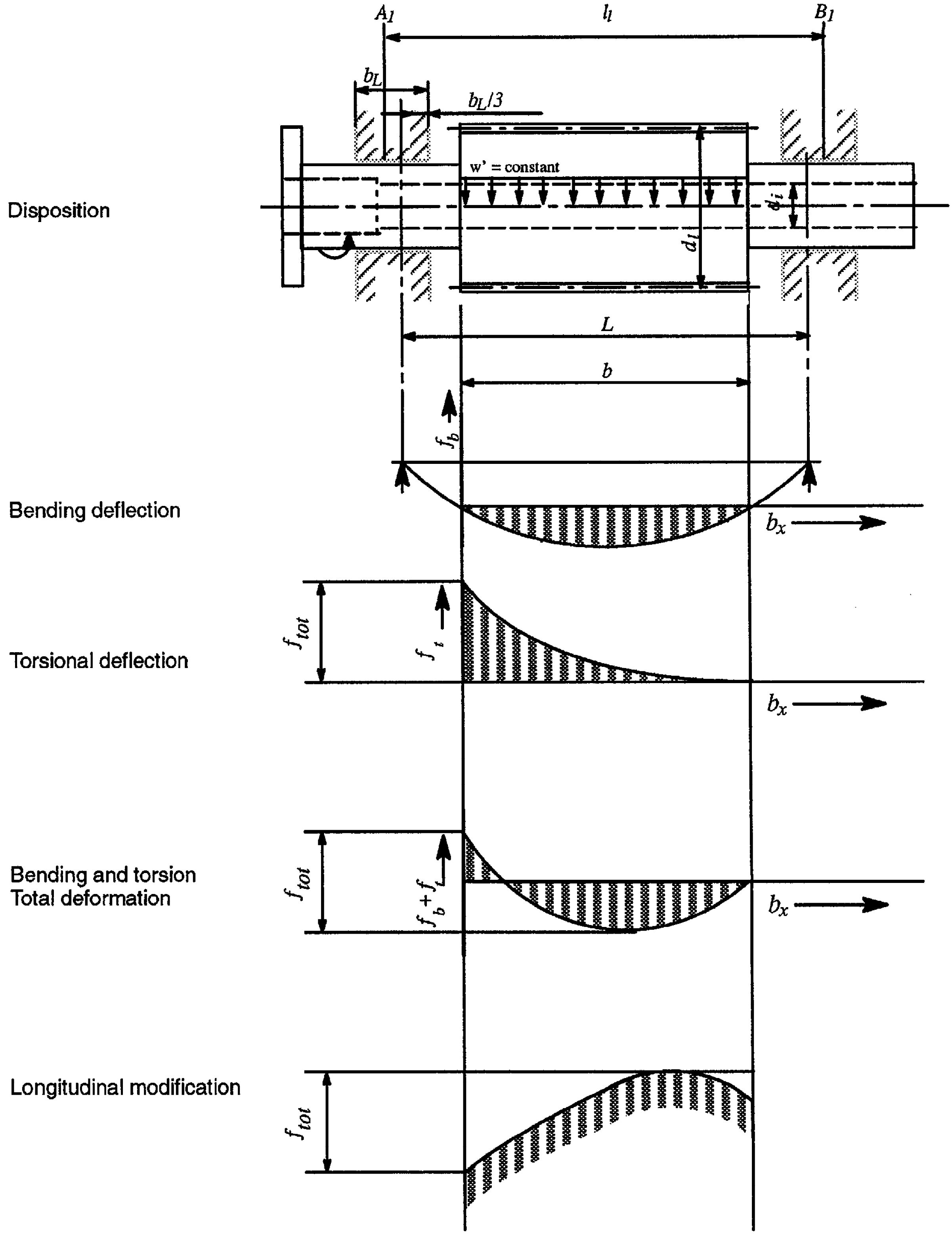

Figure D.2 – Elastic deformation of the pinion [3]

3 MAAG Gear Handbook, January 1990, reproduction of figure 3.07

Annex E
(informative)
Gear material fatigue life

This annex is for information only and should not be construed to be a part of ANSI/AGMA 2001–C95, Fundamental rating factors and calculation methods for involute spur and helical gear teeth.

E.1 Purpose

This annex provides additional and abstracted information concerning the assessment of fatigue life for spur or helical gears using various materials, material quality, heat treatments and criteria for defining life.

CAUTION: This information is for reference only – do not extract any data from the figures without first consulting the appropriate reference literature from which it was taken.

E.2 Variation of fatigue life

In addition to empirical data, gear literature contains test data on the effects on the fatigue life of gears and gear steels due to a various number of items. A collected sample of this data indicates the variability in values that can be used for capacity calculation of life factors.

E.3 Variation with materials

This clause gives three references and selected figures from each, which illustrate variations in fatigue life due to different steel alloys.

E.3.1 Vukovich, D., Pierman, R., and Matovina, M. *Laboratory Evaluation of New Low Alloy Gear Steels.* Reprinted with permission from SAE Paper No. 770416 ©1977, SAE, Inc. (figure 9).

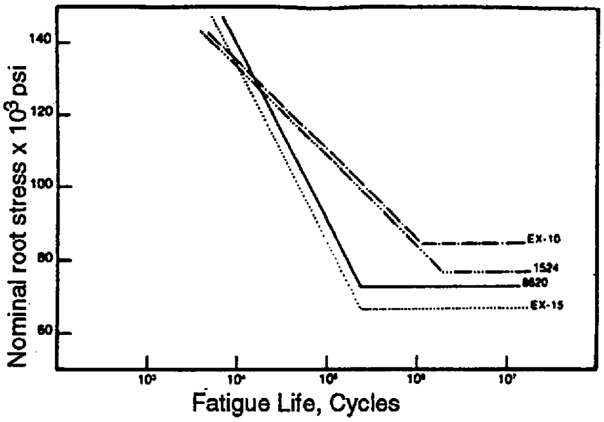

Figure 9 – Median S–N curves for carburized low alloy steel gears

E.3.2 Townsend, D. P., *Endurance and Failure Characteristics of Modified Vasco X–2, CBS 600 and AISI 9310 Spur Gears,* ASME, J. Mech Design, Paper 80–C2/DET–58, San Francisco, August 1980 (figure 11).

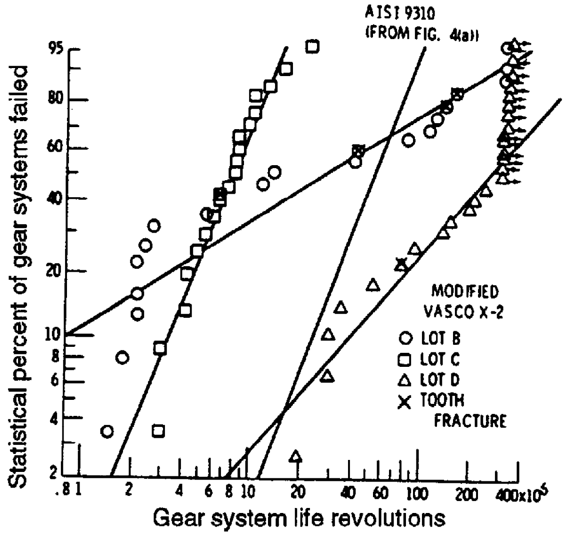

Figure 11 – Surface pitting fatigue lives of CVM modified VASCO X–2 spur gears heat treated to different specifications. (note AISI 9310)

E.3.3 Faure, L., Vasseur, J. L., and LeFleche, C. *Comparison of the Pitting Resistance of Several Steels Used in Case Carburized Gears,* Trans. MPT'91 JSME Inter. Conf., pp 849–854, Hiroshima, November 1991 (figures 5, 6 and 7).

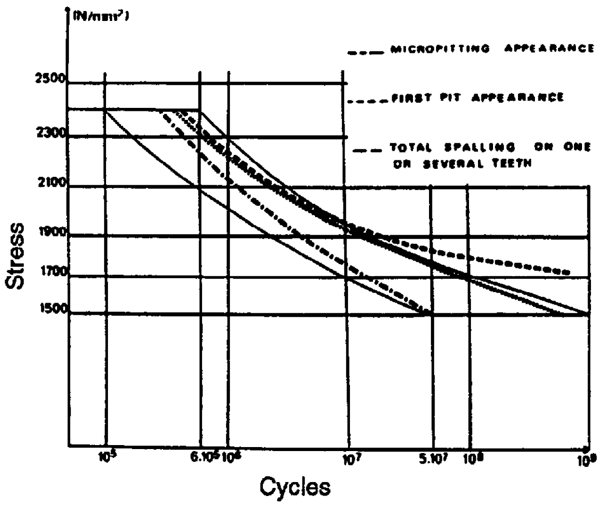

Figure 5 – Endurance curve to superficial pressure obtained with 20MC5 steel

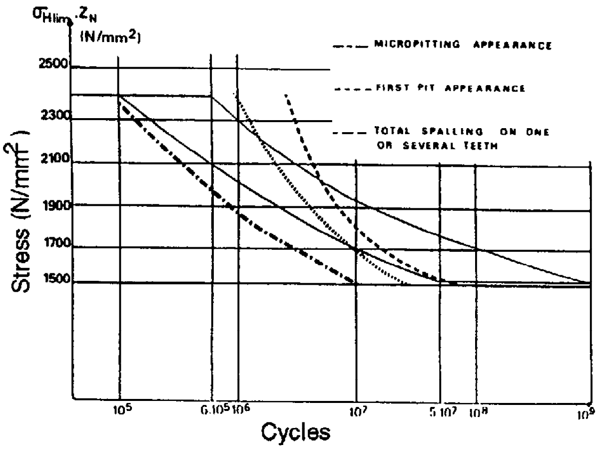

Figure 6 – Endurance curve to superficial pressure obtained with XC18 steel

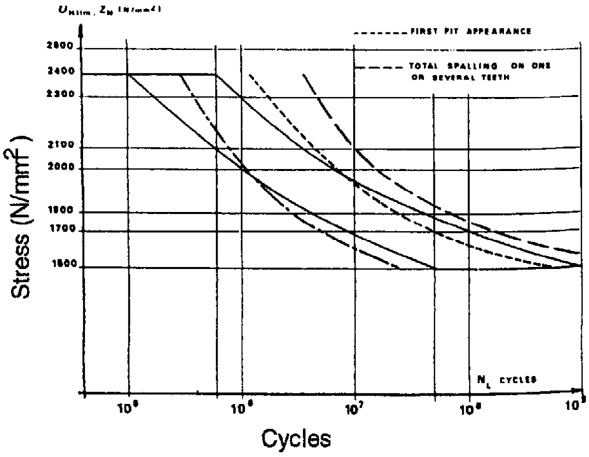

Figure 7 – Endurance curve to superficial pressure obtained with 16NC6 steel

E.3.4 Table 1 gives a list of alternate international gear steels from: Duszak, D. *Alternate Overseas Gear Steels*, ASME Gear Research Institute Transmissions Vol. VI. No 1 May 1989.

E.4 Material quality variations

This clause gives two references and a figure from each, which illustrates microstructure variations in fatigue life.

E.4.1 Parrish G. *The Influence of Microstructure on the Properties of Case–Carburized Components*, Heat Treatment of Metals 1976.3 pp 73–79 (figure 7).

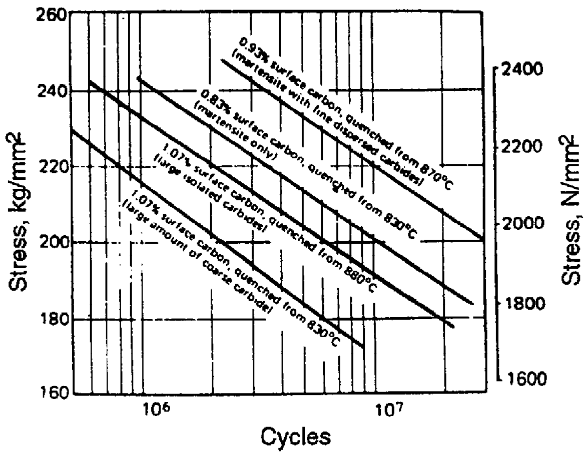

Figure 7 – The contact fatigue strength of carburized 25Kh2GHTA steel[12]

E.4.2 Kern, R. F. *Controlling Carburizing for Top Quality Gears*, Gear Technology, pp 16–21 March/ April 1993 (figure 6).

Table 1 – Alternate Overseas Gear Steels

NORTH AMERICA	FRANCE	GERMANY	JAPAN	UNITED KINGDOM
9310H	–	–	–	832H13[3)]
4118H, PS54H, PS64	–	–	5Cr415H[3)] 5CM415H[2)]	527H17[3)] 805H17[4)]
4620H	–	–	–	665H20[1)]
4820H	18CD4[5)], 8CD4[5)] 20MC5[5)], 20MC6[5)],	5CrNi6 [5)] 20MnCr5 [5)]	–	708H20[5)] 815H17[5)]
8620H, PS15H, PS64	16MC5[3)]	16MnCr5[3)]	SCM415H[4)] SCM418H[4)] 20MoCr4 [4)]	637H17[3)] 805H20[1)] SNCM220HJ[1)]
4140H	40NCD3[7)]	41CrMo4[6)] 41CrMo4[6)]	SCM440H[6)]	708H37[6)]

(refer to original paper for notes on materials in this table)

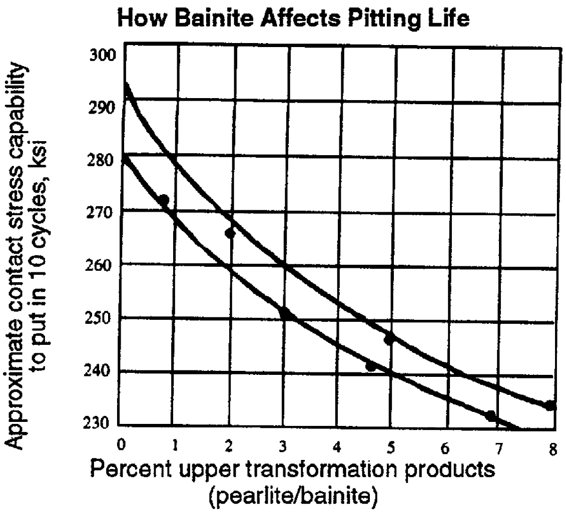

Figure 6 – Bainite, also called quenching pearlite, is soft, and deleterious to pitting life.

E.5 Variation with material heat treatment

This clause gives four references and selected figures from each, which illustrate variations in fatigue life due to heat treatments.

E.5.1 Sheehan, J. P., and Howes, M. A. H., *The Effect of Case Carbon Content and Heat Treatment on the Pitting Fatigue of 8620 Steel.* Reprinted with permission from SAE Paper No. 720268 ©1972, SAE, Inc. (figure 6).

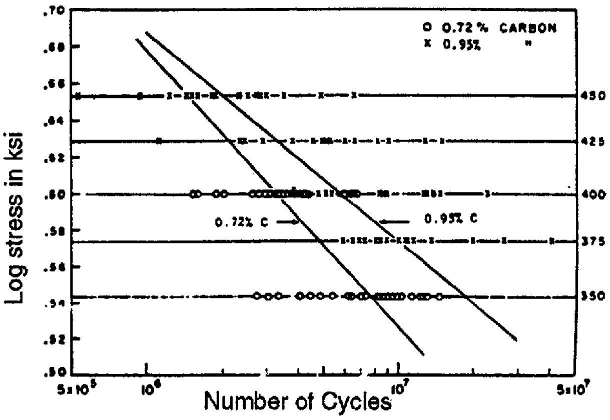

Figure 6 – Log S–log N plot of fatigue data for SAE 8620H steel carburized to 0.72 and 0.95% carbon

E.5.2 Rice, S. L., *Pitting Resistance of Some High Temperature Carburized Cases.* Reprinted with permission from SAE Paper No. 780773 ©1978, SAE, Inc. (figure 2).

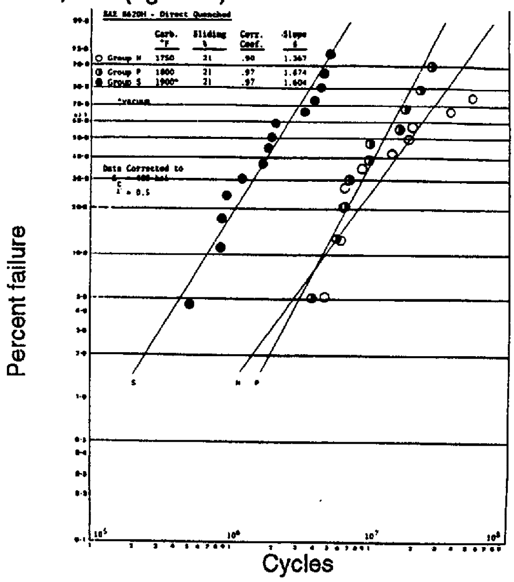

Figure 2 – Weibull probability paper

E.5.3 Kern, R. F. and Suess, M. E., *Steel Selection, a guide for improving performance and profits,* chapter 10, Selection of Steel for Carburized Gears, pp 181–205, John Wiley & Sons, New York 1979 (figure 10.13).

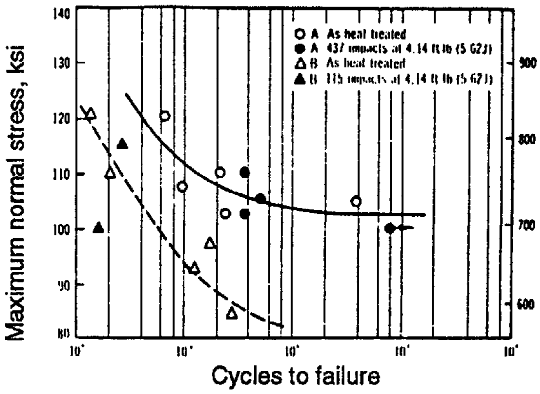

Figure 10.13 – Bending fatigue properties of 4820 steel. Sample A received standard heat treatment, while B was refrigerated at –100°F

E.5.4 Cohen, R. E., Haagensen, J. P., Matlock, D. K., and Krauss, G., *Assessment of Bending Fatigue Limits for Carburized Steel.* Reprinted with permission from SAE Paper No. 910140 ©1991, SAE, Inc. (figure 6).

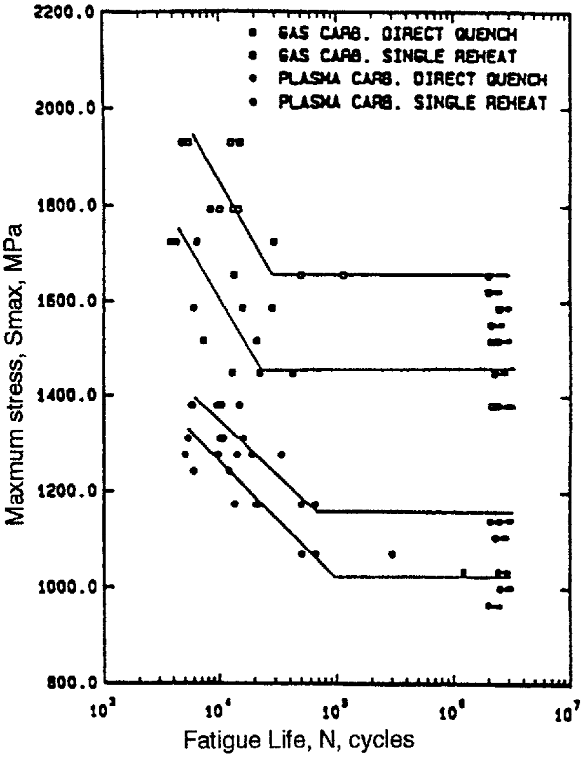

Figure 6 – Bending fatigue curves for SAE 8719 steel. The higher pair of curves corresponds to transgranular crack initiation and the lower pair corresponds to intergranular crack initiation

E.6 Failure definition variation

This clause gives three references, where selected figures illustrate variations in fatigue life due to different definitions or analysis of failure.

E.6.1 Townsend, D. P., Coy, J. J., and Zaretsky, E. V., *Experimental and Analytical Load–Life Relation for AISI 9310 Steel Spur Gears,* Transactions of the ASME, Vol. 100, pp 54–59, January 1978 (figure 5).

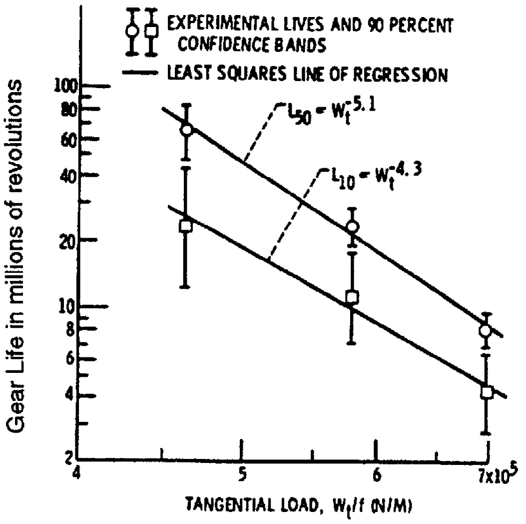

Figure 5 – Load life relationship for (VAR) AISI 9310 steel spur gears speed 10,000 rpm, lubricant naphtenic mineral oil

E.6.2 Nagamura, K., Terauchi, Y., and Martowibowo, S. Y., *Reliability Estimation of Bending Fatigue Strength of Super Carburizing Steel Spur Gears,* Trans. MPT'91 JSME Inter. Conf., pp 795–799, Hiroshima, November1991 (figure 6).

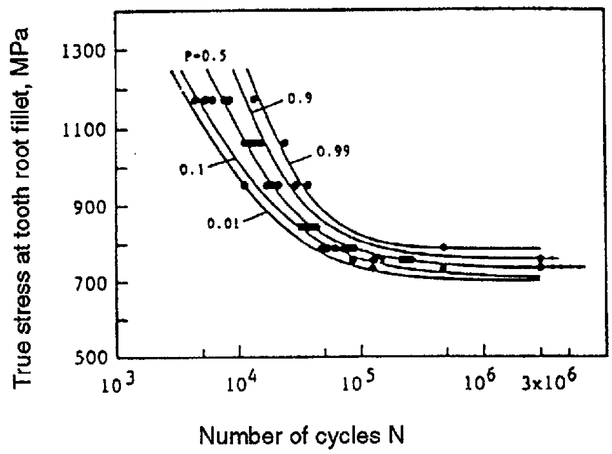

Figure 6 – S–N curve of MAC14

E.6.3. Faure, L., Vasseur, J. L., and LeFleche, C. *Comparison of the Pitting Resistance of Several Steels used in Case Carburized Gears,* American Gear Manufacturers Association, AGMA, Technical Paper 92 FTM6, October 1992 (figures 6, 7, 8 and 9).

The following four figures presents up–dated analysis of data presented in clause E3.3. This, also, illustrates variations in calculations which could result from the different presentation of data.

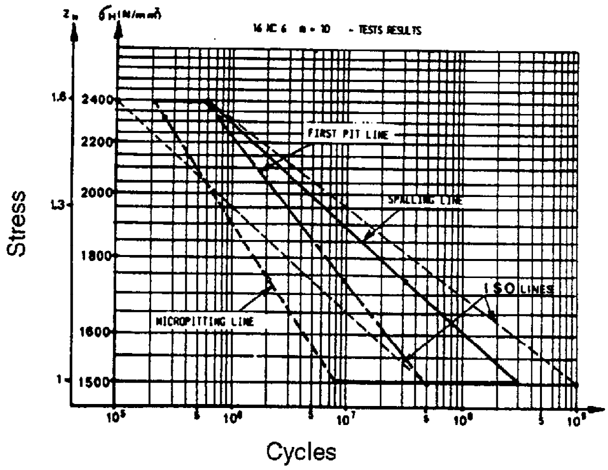

Figure 6 – 16NC6 m=10 Test results

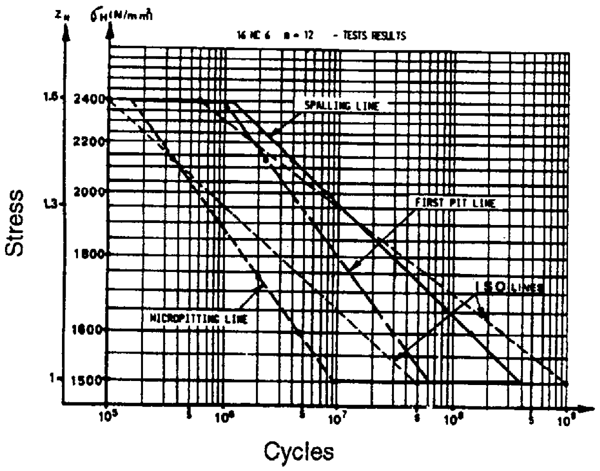

Figure 7 – 16NC6 m=12 Test results

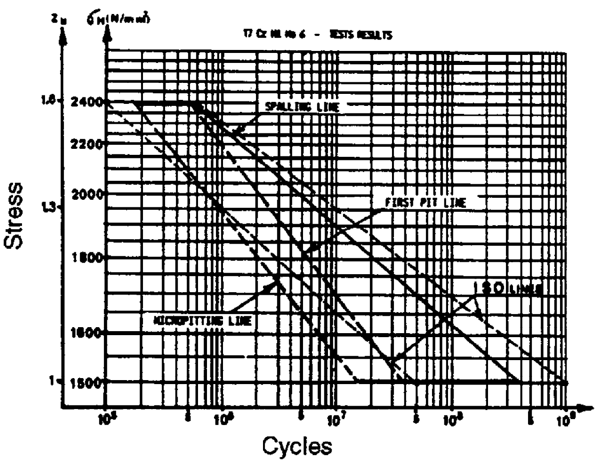

Figure 8 – 17CrNiMo6 Test results

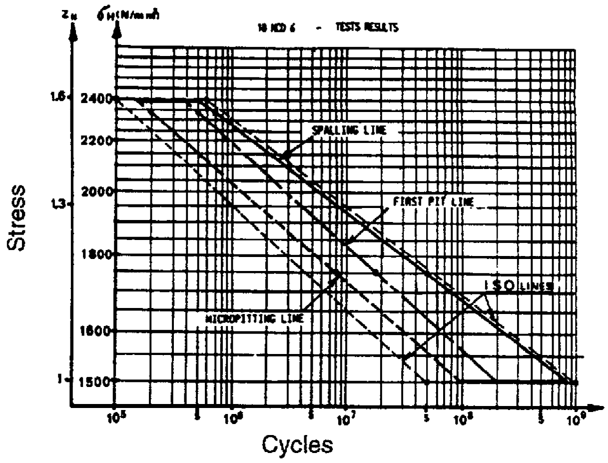

Figure 9 – 18NCD6 Test results

E.7 Other considerations and summary

The illustrated variations in fatigue life are only a sample of the data available. Those presented are used to show many of the considerations which may vary on each application. Other items such shot peening can also affect fatigue life.

E.7.1 Hatano, A,, and Namiki, K., *Application of Hard Shot Peening to Automotive Transmission Gears.* Reprinted with permission from SAE Paper No. 920760 ©1992, SAE, Inc. (figure 14).

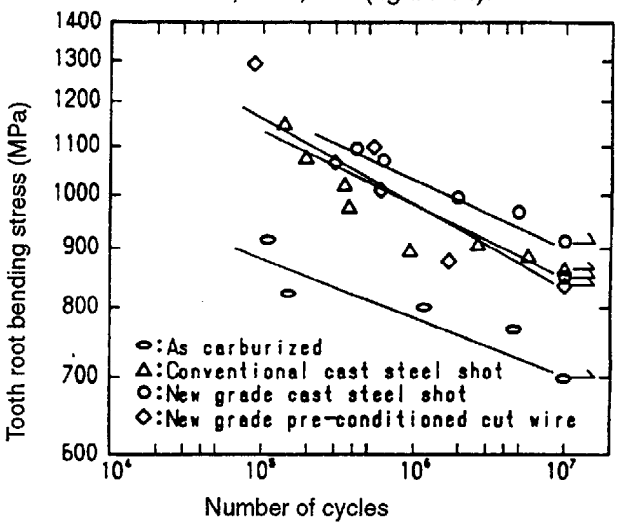

Figure 14 – Comparison of fatigue properties for gears shot peened by conventional and newly developed media

E.7.2 Summary

This annex illustrates that variations in fatigue life are influenced by a great many factors. Therefore, only an experienced engineer should apply knowledge of S–N curves to gear calculations.

Annex F
(informative)
Controlling section size considerations for through hardened gearing

This annex is for information only and should not be construed to be a part of ANSI/AGMA 2101–C95, *Fundamental rating factors and calculation methods for involute spur and helical gear teeth.*

F.1 Purpose

This annex presents approximate maximum controlling section size considerations for through hardened (quench and tempered) gearing. Also presented are factors which affect maximum controlling size, illustrations as to how maximum controlling section size is determined for gearing, and recommended maximum controlling section sizes for some low alloy steels.

F.2 Definition

The controlling section of a part is defined as that section which has the greatest effect on the rate of cooling during quenching at the location (section) where the specified mechanical properties (hardness) are required. The alloy for the part is chosen from the quenching property of the equivalent round bar having a diameter equal to the controlling section size. The maximum controlling section size for a steel is based principally on hardenability, specified hardness, depth of desired hardness, quench rate and tempering temperature.

F.3 Illustrations

Figure F.1 illustrates controlling sections for quenched gear configurations whose teeth are machined after heat treatment.

NOTE: Evaluation of the controlling section size for the selection of an appropriate type of steel and specified hardness need not include consideration of standard rough stock machining allowances. Other special stock allowances such as those used to minimize distortion during heat treatment must be considered.

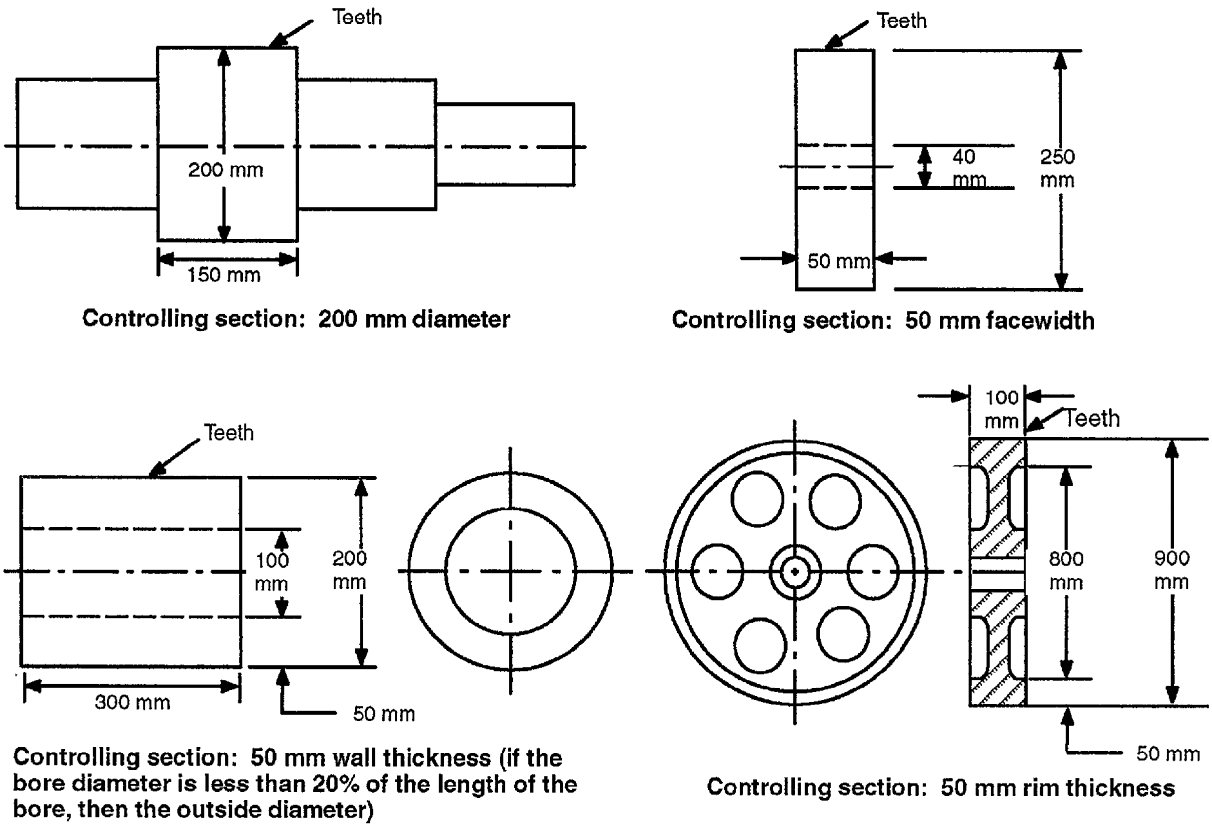

Figure F.1 – Illustrations of controlling section size

F.4 Recommendations

Figure F.2 provides approximate recommended maximum controlling section sizes for oil quenched and tempered gearing (Grossman quench severity value H = 0.5) of low alloy steels based on specified hardness range, normal stock allowance before hardening, minimum tempering temperature of 480°C, and obtaining minimum hardness at the roots of teeth.

F.5 General comments

Maximum controlling section sizes versus specified hardness for section sizes to 200 mm diameter rounds can also be approximated by use of the "Chart Predicting Approximate Cross Section Hardness of Quenched Round Bars from Jominy Test Results" published in *Practical Data for Metallurgists* by Timken Steel Co., and published tempering response/hardenability data.

Maximum controlling section sizes for rounds greater than 200 mm O.D. generally require in–house heat treat experiments of larger sections followed by sectioning and transverse hardness testing.

Normalized and tempered gearing may require a higher hardenability if the design does not permit liquid quenching. Hardnesses obtainable by normalize and temper are lower than those obtained by quench and temper. Normalized and tempered/hardness testing experiments are required.

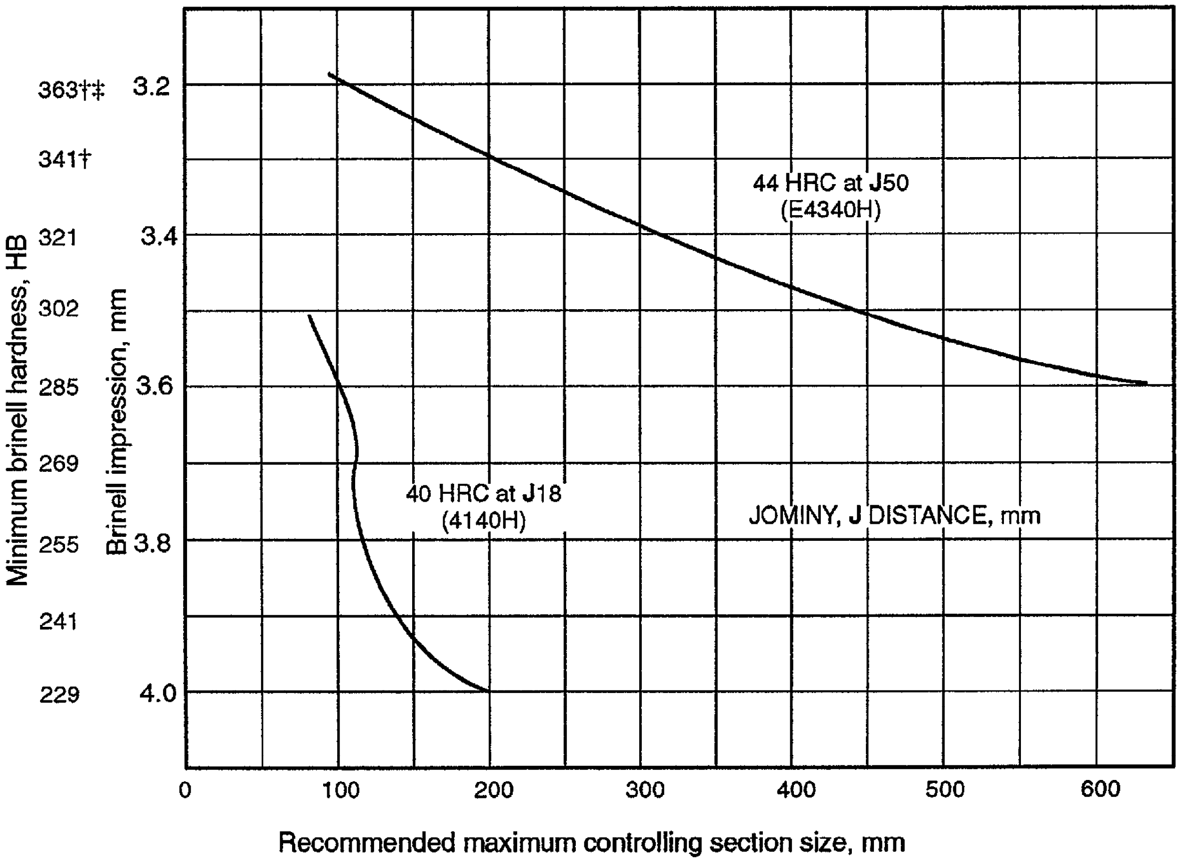

Figure F.2 – Controlling section size for two 0.40% carbon alloy steels*

NOTES:

*Maximum controlling section sizes higher than those above can be recommended when substantiated by test data (heat treat practice).

†480°C minimum temper may be required to meet these hardness specifications.

‡Higher specified hardnesses (e.g., 375–415 HB, 388–421 HB and 401–444 HB) are used for special gearing, but costs should be evaluated due to reduced machinability.

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Annex G
(informative)
Bibliography

This annex is for information only and should not be construed to be a part of ANSI/AGMA 2101–C95, *Fundamental rating factors and calculation methods for involute spur and helical gear teeth.*

G.1 Purpose

The following documents are either referenced in the text of ANSI/AGMA 2001–C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth* or indicated for additional information.

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