ANSI/AGMA 2001-D04
(Revision of
ANSI/AGMA 2001-C95)

AMERICAN NATIONAL STANDARD

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

ANSI/AGMA 2001-D04

AGMA STANDARD

American National Standard

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

ANSI/AGMA 2001-D04

[Revision of ANSI/AGMA 2001-C95]

Approval of an American National Standard requires verification by ANSI that the requirements for due process, consensus, and other criteria for approval have been met by the standards developer.

Consensus is established when, in the judgment of the ANSI Board of Standards Review, substantial agreement has been reached by directly and materially affected interests. Substantial agreement means much more than a simple majority, but not necessarily unanimity. Consensus requires that all views and objections be considered, and that a concerted effort be made toward their resolution.

The use of American National Standards is completely voluntary; their existence does not in any respect preclude anyone, whether he has approved the standards or not, from manufacturing, marketing, purchasing, or using products, processes, or procedures not conforming to the standards.

The American National Standards Institute does not develop standards and will in no circumstances give an interpretation of any American National Standard. Moreover, no person shall have the right or authority to issue an interpretation of an American National Standard in the name of the American National Standards Institute. Requests for interpretation of this standard should be addressed to the American Gear Manufacturers Association.

CAUTION NOTICE: AGMA technical publications are subject to constant improvement, revision, or withdrawal as dictated by experience. Any person who refers to any AGMA technical publication should be sure that the publication is the latest available from the Association on the subject matter.

[Tables or other self-supporting sections may be referenced. Citations should read: See ANSI/AGMA 2001-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth,* published by the American Gear Manufacturers Association, 500 Montgomery Street, Suite 350, Alexandria, Virginia 22314, http://www.agma.org.]

Approved December 28, 2004

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.

Published by

American Gear Manufacturers Association
500 Montgomery Street, Suite 350, Alexandria, Virginia 22314

Printed in the United States of America

ISBN: 1-55589-839-4

Contents

Annexes

Figures

Tables

Foreword

[The foreword, footnotes and annexes, if any, in this document are provided for informational purposes only and are not to be construed as a part of ANSI/AGMA 2001-D04, *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, and supersedes ANSI/AGMA 2001-C95.

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, C_L and K_L, Dynamic Factor, C_v and K_v, and Load Distribution Factor, C_m and K_m, 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, I and 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 I and 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.

ANSI/AGMA 2001-C95 was a revision of the rating method described in its superseded publications. The changes included: the Miner's rule annex was removed; the analytical method for load distribution factors, C_m and K_m, 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; life factor was replaced by stress cycle factor and its use with service factor redefined; and, the dynamic factor was redefined as the reciprocal of that used in previous AGMA standards and was relocated to the denominator of the power equation.

This standard, ANSI/AGMA 2001-D04, is a revision of its superseded version. Clause 8 was changed to incorporate ANSI/AGMA 2015-1-A01 and the K_v method using AGMA 2000-A88 was moved to Annex A. References to old Annex A, "Method for Evaluating the

Risk of Scuffing and Wear" were changed to AGMA 925-A03. It also reflects a change to clause 10, dealing with the relationship between service factor and stress cycle factor. Editorial corrections were implemented to table 8, figure 14 and table E-1, and style was updated to latest standards.

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 ANSI/AGMA 2001-D04 was completed in February 2002. It was approved by the AGMA membership on October 23, 2004. It was approved as an American National Standard on December 28, 2004.

Suggestions for improvement of this standard will be welcome. They should be sent to the American Gear Manufacturers Association, 500 Montgomery Street, Suite 350, Alexandria, Virginia 22314.

PERSONNEL of the AGMA Helical Gear Rating Committee

Chairman: John V. Lisiecki . . . . Falk Corporation
Vice Chairman: Michael B. Antosiewicz . . . . Falk Corporation

ACTIVE MEMBERS

K.E. Acheson . . . . Gear Works - Seattle, Inc.
J.B. Amendola . . . . MAAG Gear AG
T.A. Beveridge . . . . Caterpillar, Inc.
M. Broglie . . . . Dudley Technical Group
G.A. DeLange . . . . Hansen Transmissions
G. Elliott . . . . Lufkin Industries, Inc.
R.L. Errichello . . . . GEARTECH
R.W. Holzman . . . . Innovative Gearing Solutions LLC
O.A. LaBath . . . . Gear Consulting Services of Cincinnati, LLC
G. Lian . . . . Amarillo Gear Company
L. Lloyd . . . . Lufkin Industries, Inc.
D. McCarthy . . . . Gear Products, Inc.
D.R. McVittie . . . . Gear Engineers, Inc.
A.G. Milburn . . . . Milburn Engineering, Inc.
G.W. Nagorny . . . . Nagorny & Associates
F.C. Uherek . . . . Falk Corporation

ASSOCIATE MEMBERS

M. Bartolomeo . . . . Rolls-Royce Corporation
E.J. Bodensieck . . . . Bodensieck Engineering Company
D.L. Borden . . . . D.L. Borden, Inc.
K.J. Buzdygon . . . . ExxonMobil Research and Engineering
A.B. Cardis . . . . Consultant
M.R. Chaplin . . . . Contour Hardening, Inc.
J. Chen . . . . General Motors Corporation
E. Chermet . . . . CETIM
R.J. Ciszak . . . . GE - Rail
A.S. Cohen . . . . Engranes y Maquinaria Arco, S.A.
S. Copeland . . . . Gear Products, Inc.
R.L. Cragg . . . . Steward Machine Company, Inc.
T.J. Dansdill . . . . General Electric Company AE Marine Engines
F. Eberle . . . . Hi-Lex Controls, Inc.
J.M. Escanaverino . . . . Instituto Superior Politecnico
L. Faure . . . . Compagnie Engrenages Et Reducteurs
T. Funk . . . . Gear Products, Inc.
M.J. Gardner . . . . Boeing Commercial Airplane Group
C. Gay . . . . Charles E. Gay & Company, Ltd.
T.C. Glasener . . . . Xtek, Inc.
G.G. Rey . . . . Instituto Superior Politecnico
H. Hagan . . . . Philadelphia Gear Corporation
H. Hagiwara . . . . Nippon Gear Company, Ltd.
E.C. Hahlbeck . . . . Powertrain Engineers, Inc.
R.W. Hankes . . . . A-C Equipment Services Corporation
M.A. Hartman . . . . ITW

J.M. Hawkins .... Rolls-Royce Corporation
G. Henriot .... Consultant
M. Hirt .... Renk AG
M.R. Hoeprich .... Timken Company
R.S. Hyde .... Timken Company
K.T. Jones .... Boeing Commercial Airplane Group
J.R. Keough .... Applied Process, Inc.
H.J. Kim .... General Motors Corporation
J.G. Kish .... Sikorsky Aircraft Division
R.H. Klundt .... Timken Company
I. Laskin .... Consultant
D.A. Lauer .... Kluber Lubrication North America L.P.
S. Luchetta .... Philadelphia Gear Corporation
W. Luo .... Chicago Gear - D.O. James Corporation
J.J. Luz .... General Electric Company AE Marine Engines
J. Maddock .... Consultant
K. Miller .... Dana Spicer Off Highway Products
S. Miller .... David Brown Engineering, Ltd.
H. Minasian .... Consultant
G.P. Mowers .... Gary P. Mowers, Inc.
R.A. Nay .... Hamilton Sundstrand
A. Noll .... Horsburgh & Scott Company
B. O'Connor .... The Lubrizol Corporation
M. Octrue .... CETIM
J.A. Pennell .... University of Newcastle-Upon-Tyne, Design Unit
A.E. Phillips .... Rockwell Automation/Dodge (Retired)
A. Piazza .... Centro Ricerche Fiat S.p.A.
W.P. Pizzichil .... Rockwell Automation/Dodge
J.W. Polder .... Delft University of Technology
S. Rao .... Philadelphia Gear Corporation
E. Sandberg .... Det Norske Veritas
H. Sanderow .... Management & Engineering Technologies
C.D. Schultz .... Pittsburgh Gear Company
E.S. Scott .... Consultant
Y. Sharma .... Rockwell Automation/Dodge
B.W. Shirley .... Emerson Power Transmission, Gearing Facility
D.F. Smith .... Solar Turbines, Inc. Gear Systems
L.J. Smith .... Consultant
G.L. Snelling .... General Motors Corporation
L. Spiers .... Emerison Power Transmission Corporation
W.T. Sumi .... Cognis Corporation - Lubricant Technologies
A.A. Swiglo .... Alion Science and Technology
K. Taliaferro .... Rockwell Automation/Dodge
F.A. Thoma .... F.A. Thoma, Inc.
D. Townsend .... Townsend Engineering
A. von Graefe .... MAAG Gear AG
H.W. Wallis .... Cognis Corporation - Lubricant Technologies
C.C. Wang .... 3E Software & Engineering Consulting
B. Ward .... Recovery Systems, LLC
R.F. Wasilewski .... Arrow Gear Company

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, m_p, less than 1.0.
- Spur or helical gears with transverse contact ratio, m_p, 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 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 this standard. A method to evaluate scuffing risk can be found in AGMA 925-A03. This information is provided for

[ ] Numbers in brackets refer to the reference number listed in the Bibliography.
* Refer to ANSI/AGMA 1012-F90 for further discussion of standard (reference) diameters.

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

2 Normative references, definitions and symbols

2.1 Normative references

The following documents contain provisions which, through reference in this text, constitute provisions of this standard. At the time of development, the editions were valid. All publications are subject to revision, and the users of this standard are encouraged to investigate the possibility of applying the most recent editions of the publications listed.

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

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

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

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

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

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

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

ANSI/AGMA 2015-1-A01, *Accuracy Classification System - Tangential Measurements for Cylindrical Gears.*

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 Sand Castings.*

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

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

ASTM E709-91, *Guide for Magnetic Particle Examination.*

2.2 Definitions

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

2.3 Symbols

The symbols used in this standard are shown in table 1.

NOTE: The symbols and terms contained in this document may vary from those used in other AGMA standards. Users of this standard should assure themselves that they are using these symbols and terms in the manner indicated herein.

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.

Table 1 - Symbols used in gear rating equations

Symbol	Description	Units	First Used	Ref. Clause
A_v	Transmission accuracy level number	--	Eq 22	8.3
C	Operating center distance	in	Eq 2	5.1.1
C_e	Mesh alignment correction factor	--	Eq 38	15.3
C_f	Surface condition factor for pitting resistance	--	Eq 1	13
C_G	Gear ratio factor	--	Eq 6	5.1.4
C_H	Hardness ratio factor for pitting resistance	--	Eq 4	14
C_{ma}	Mesh alignment factor	--	Eq 38	15.3
C_{mc}	Lead correction factor	--	Eq 38	15.3
C_{mf}	Face load distribution factor	--	Eq 36	15.3
C_{mt}	Transverse load distribution factor	--	Eq 36	15.2
C_p	Elastic coefficient	$[\text{lb/in}^2]^{0.5}$	Eq 1	12
C_{pf}	Pinion proportion factor	--	Eq 38	15.3
C_{pm}	Pinion proportion modifier	--	Eq 38	15.3
C_{SF}	Service factor for pitting resistance	--	Eq 30	10
d	Operating pitch diameter of pinion	in	Eq 1	5.1.1
d_e	Outside diameter of pinion or gear	in	Eq 27	8.3.3
d_T	Tolerance diameter	in	Eq 25	8.3.3
E_G	Modulus of elasticity for gear	lb/in^2	Eq 31	12
E_P	Modulus of elasticity for pinion	lb/in^2	Eq 31	12
F	Net face width of narrowest member	in	Eq 1	5.1.1
f_p	Pinion surface finish	micro-inches, R_a	Eq 35	14.2
H_{BG}	Brinell hardness of gear	HB	Eq 33	14.1
H_{BP}	Brinell hardness of pinion	HB	Eq 33	14.1
h_{cmin}	Minimum total case depth for external nitrided gear teeth	in	Eq 45	16.1
h_{emax}	Maximum effective case depth for external carburized and induction hardened gear teeth	in	Eq 44	16.1
h_{emin}	Minimum effective case depth for external carburized and induction hardened gear teeth	in	Eq 43	16.1
h_t	Gear tooth whole depth	in	Eq 17	5.2.5
I	Geometry factor for pitting resistance	--	Eq 1	6.1
J	Geometry factor for bending strength	--	Eq 10	6.2
K	Contact load factor for pitting resistance	lb/in^2	Eq 6	5.1.4
K_{ac}	Allowable contact load factor	lb/in^2	Eq 9	5.1.4
K_B	Rim thickness factor	--	Eq 10	5.2.5
K_f	Stress correction factor	--	Eq 46	16.4
K_m	Load distribution factor	--	Eq 1	15
K_{my}	Load distribution factor under overload conditions	--	Eq 46	16.4
K_o	Overload factor	--	Eq 1	9
K_R	Reliability factor	--	Eq 4	18
K_s	Size factor	--	Eq 1	20

(continued)

Table 1 *(continued)*

Symbol	Description	Units	First Used	Ref. Clause
K_{SF}	Service factor for bending strength	--	Eq 30	10
K_T	Temperature factor	--	Eq 4	19
K_v	Dynamic factor	--	Eq 1	8
K_y	Yield strength factor	--	Eq 46	16.4
L	Life	hours	Eq 48	17.1
m_B	Back-up ratio	--	Eq 17	5.2.5
m_G	Gear ratio (never less than 1.0)	--	Eq 2	5.1.1
N	Number of stress cycles	--	Eq 48	17.1
N_G	Number of teeth in gear	--	Eq 7	5.1.4
N_P	Number of teeth in pinion	--	Eq 7	5.1.4
n	Speed	rpm	Eq 48	17.1
n_p	Pinion speed	rpm	Eq 5	5.1.3
P	Transmitted power	hp	Eq 18	7.1
P_a	Allowable transmitted power for gear set	hp	Eq 30	10
P_{ac}	Allowable transmitted power for pitting resistance	hp	Eq 5	5.1.3
P_{acu}	Allowable transmitted power for pitting resistance at unity service factor	hp	Eq 28	10
P_{at}	Allowable transmitted power for bending strength	hp	Eq 14	5.2.3
P_{atu}	Allowable transmitted power for bending strength at unity service factor	hp	Eq 29	10
P_d	Transverse diametral pitch	in^{-1}	Eq 10	5.2.1
P_{nd}	Normal diametral pitch	in^{-1}	Eq 11	5.2.1
p_x	Axial pitch	in	Eq 11	5.2.1
q	Number of contacts per revolution	--	Eq 48	17.1
S	Bearing span	in	Fig 6	15.3
S_1	Pinion offset	in	Fig 6	15.3
S_F	Safety factor - bending	--	Eq 13	11
S_H	Safety factor - pitting	--	Eq 4	11
s_{ac}	Allowable contact stress number	lb/in^2	Eq 4	16
s_{at}	Allowable bending stress number	lb/in^2	Eq 13	16
s_{ay}	Allowable yield stress number	lb/in^2	Eq 46	16.4
s_c	Contact stress number	lb/in^2	Eq 1	5.1.1
s_t	Bending stress number	lb/in^2	Eq 10	5.2.1
T	Transmitted pinion torque	lb in	Eq 18	7.1
t_o	Normal tooth thickness at the top land of gear	in	Eq 44	16.1
t_R	Gear rim thickness	in	Eq 17	5.2.5
U_{at}	Allowable unit load for bending strength	lb/in^2	Eq 16	5.2.4
U_c	Core hardness coefficient	--	Eq 45	16.1
U_H	Hardening process factor	--	Eq 43	16.1
U_L	Unit load for bending strength	lb/in^2	Eq 15	5.2.4
v_t	Pitch line velocity at operating pitch diameter	ft/min	Eq 18	7.1
v_{tmax}	Pitch line velocity maximum at operating pitch diameter	ft/min	Eq 24	8.3.2

(continued)

Table 1 *(continued)*

Symbol	Description	Units	First Used	Ref. Clause
W_d	Incremental dynamic tooth load	lb	Eq 20	8.1
W_{max}	Maximum peak tangential load	lb	Eq 46	16.4
W_t	Transmitted tangential load	lb	Eq 1	7.1
Y_N	Stress cycle factor for bending strength	--	Eq 13	17
Z_N	Stress cycle factor for pitting resistance	--	Eq 4	17
μ_G	Poisson's ratio for gear	--	Eq 31	12
μ_P	Poisson's ratio for pinion	--	Eq 31	12
ϕ_t	Operating transverse pressure angle	--	Eq 43	16.1
ψ_b	Base helix angle	--	Eq 43	16.1
ψ_s	Helix angle at standard pitch diameter	--	Eq 11	5.2.1

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, s_{ac} and s_{at}, 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. AGMA 925-A03 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 1000 to 10 000 ft/min, 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 100 ft/min, require special design consideration to avoid premature failure due to inadequate lubrication.

At low surface speeds [below 100 ft/min 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 100 ft/min but less than 1000 ft/min, 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 32°F, 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.

- Reducing 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 300°F.

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 ANSI/AGMA 6011-I03, *Specification for High Speed Helical Gear Units,* Annex D.

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 1010-E95 [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 1010-E95.

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 1010-E95 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 0.0008 inch 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, 800 to 2000 ft/min 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 AGMA 925-A03).

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. AGMA 925-A03 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 1010-E95.

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:

$$s_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d F} \frac{C_f}{I}} \qquad (1)$$

where

s_c is contact stress number, lb/in^2;

C_p is elastic coefficient, [lb/in^2]$^{0.5}$ (see clause 12);

W_t is transmitted tangential load, lb (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_m is load distribution factor (see clause15);

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

F is net face width of narrowest member, in;

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

d is operating pitch diameter of pinion, in.

$$d = \frac{2C}{m_G + 1} \text{ for external gears} \qquad (2)$$

$$d = \frac{2C}{m_G - 1} \text{ for internal gears} \qquad (3)$$

where

C is operating center distance, in;

m_G 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:

$$s_c \le \frac{s_{ac}}{S_H} \frac{Z_N}{K_T} \frac{C_H}{K_R} \qquad (4)$$

where

s_{ac} is allowable contact stress number, lb/in^2 (see clause 16);

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

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

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

K_T is temperature factor (see clause 19);

K_R is reliability factor (see clause 18).

5.1.3 Pitting resistance power rating

The pitting resistance power rating is:

$$P_{ac} = \frac{\pi \; n_p F}{396\,000} \frac{I}{K_o K_v K_s K_m C_f} \left(\frac{d\, s_{ac}}{C_p S_H} \frac{Z_N C_H}{K_T K_R} \right)^2 \tag{5}$$

where

P_{ac} is allowable transmitted power for pitting resistance, hp;

n_p 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 $s_{ac} Z_N C_H$ 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.

$$K = \frac{W_t}{d\,F} \frac{1}{C_G} \tag{6}$$

where

K is contact load factor for pitting resistance, lb/in^2;

C_G is gear ratio factor.

$$C_G = \frac{m_G}{m_G + 1} \text{ or } \frac{N_G}{N_G + N_P} \text{ for external gears} \tag{7}$$

and

$$C_G = \frac{m_G}{m_G - 1} \text{ or } \frac{N_G}{N_G - N_P} \text{ for internal gears} \tag{8}$$

where

N_G is number of teeth in gear;

N_P is number of teeth in pinion.

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

$$K_{ac} = \frac{I}{K_o K_v K_s K_m C_f C_G} \left(\frac{s_{ac}}{C_p S_H} \frac{Z_N C_H}{K_T K_R} \right)^2 \tag{9}$$

where

K_{ac} is allowable contact load factor, lb/in^2.

The allowable contact load factor, K_{ac}, is the lowest of the ratings calculated using the different values of s_{ac}, C_H 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:

$$s_t = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J} \tag{10}$$

where

s_t is bending stress number, lb/in^2;

K_B is rim thickness factor (see 5.2.5);

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

P_d is transverse diametral pitch, in^{-1}*;

P_d is P_{nd} for spur gears.

$$P_d = \frac{\pi}{p_x \tan \psi_s} = P_{nd} \cos \psi_s \text{ for helical gears} \tag{11}$$

where

P_{nd} is normal diametral pitch, in^{-1};

p_x is axial pitch, in;

ψ_s is helix angle at standard pitch diameter.

$$\psi_s = \arcsin \left(\frac{\pi}{p_x P_{nd}} \right) \tag{12}$$

5.2.2 Allowable bending stress number

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

$$s_t \le \frac{s_{at} Y_N}{S_F K_T K_R} \tag{13}$$

where

s_{at} is allowable bending stress number, lb/in^2 (see clause 16);

* This calculation is based on standard gear hobbing practice, with P_{nd} and p_x given. For a detailed text on geometry, see AGMA 933-B03, *Information Sheet - Basic Gear Geometry.*.

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:

$$P_{at} = \frac{\pi\, n_p\, d}{396\,000\, K_o\, K_v} \frac{F}{P_d} \frac{J}{K_s\, K_m\, K_B} \frac{s_{at}\, Y_N}{S_F\, K_T\, K_R} \quad (14)$$

where

P_{at} is allowable transmitted power for bending strength, hp.

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

$\frac{s_{at}\, Y_N\, J}{K_B}$ 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.

$$U_L = \frac{W_t\, P_{nd}}{F} \quad (15)$$

where

U_L is unit load for bending strength, lb/in^2.

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

$$U_{at} = \frac{J}{\cos\psi_s\, K_o\, K_v\, K_s\, K_m\, K_B} \frac{s_{at}\, Y_N}{K_T\, K_R\, S_F} \quad (16)$$

where

U_{at} is allowable unit load for bending strength, lb/in^2.

The allowable unit load, U_{at}, is the lowest of the ratings calculated using the different values of s_{at}, K_B, Y_N and 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).

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

where

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

h_t is gear tooth whole depth, in.

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, I and J

6.1 Pitting resistance geometry factor, I

The geometry factor, 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, J

The geometry factor, 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 geometry factors, I and 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, W_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. W_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 W_t.

7.1 Uniform load

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

$$W_t = \frac{33\,000\,P}{v_t} = \frac{2\,T}{d} = \frac{396\,000\,P}{\pi\,n_p\,d} \qquad (18)$$

where

- P is transmitted power, hp;
- T is transmitted pinion torque, lb in;
- v_t is pitch line velocity at operating pitch diameter, ft/min.

$$v_t = \frac{\pi\,n_p\,d}{12} \qquad (19)$$

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.

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

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, lbs.

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, the dynamic factor curve should be selected based on experience with the manufacturing methods and operating considerations of the design.

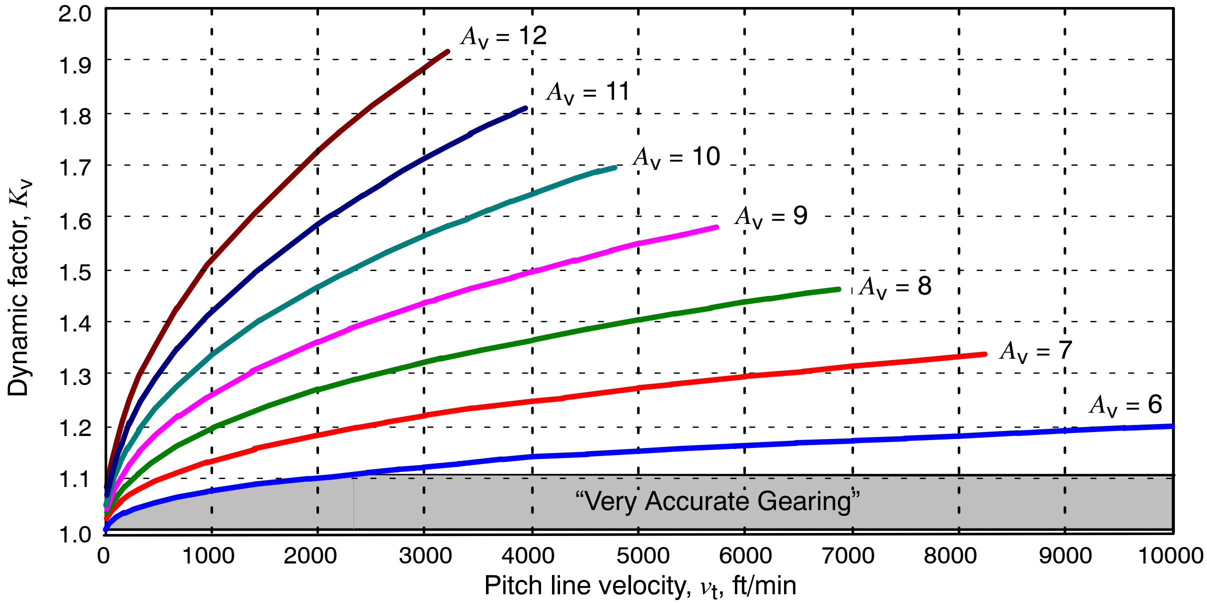


Figure 1 - Dynamic factor, K_v

Choice of curves A_v = 6 through A_v = 12 and "very accurate gearing" should be based on transmission error.

The transmission accuracy level number, A_v, can be estimated as the appropriate accuracy grade, A, for the expected pitch and profile deviations in accordance with ANSI/AGMA 2015-1-A01. See Annex A for use with AGMA 2000-A88.

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 Calculating K_v

Empirical curves labeled A_v = 6 through A_v = 12 in figure 1 are generated by the following equations for integer values of A_v, such that $6 \leq A_v \leq 12$. A_v is related to the transmission accuracy grade number.

$$K_v = \left(\frac{C}{C + \sqrt{v_t}} \right)^{-B} \tag{21}$$

where

$$C = 50 + 56\,(1.0 - B) \quad \text{for } 6 \leq A_v \leq 12 \tag{22}$$

$$B = 0.25\,(A_v - 5.0)^{0.667} \tag{23}$$

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

$$v_{t\,max} = [C + (14 - A_v)]^2 \tag{24}$$

where

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

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, equation 24 defines the end points of the curves in figure 1.

8.3.3 Estimating A_v

When A_v or A are 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. A slight variation from the selected "A_v" value is not considered significant to the gearset rating.

A_v can be approximated using the pitch variation of the pinion and gear with the following formulas, rounded to the next higher integer. Values of A_v should be calculated for both gear and pinion, and the higher value should be used for calculating the dynamic factor, K_v.

For 0.20 < $d_T \leq$ 15.75 in

$$A_v = \frac{\ln\left(0.0254\left|f_{pt}\right|\right) - \ln\left(\frac{7.62}{P_{nd}} + 0.0762\, d_T + 5.2\right)}{0.3466} + 5 \quad (25)$$

(rounded to the next highest integer)

For 15.75 < $d_T \leq$ 39.37 in

$$A_v = \frac{\ln\left(0.0254\left|f_{pt}\right|\right) - \ln\left(\frac{7.62}{P_{nd}} + 0.6048\, d_T^{0.5} + 4\right)}{0.3466} + 5 \quad (26)$$

(rounded to the next highest integer)

where

ln is natural log, $\log_e$;

f_{pt} is single pitch deviation, microinch;

NOTE: 1 microinch = 10^{-6} inches.

P_{nd} is normal diametral pitch (in^{-1}), where $0.5 \leq P_{nd} \leq 20$;

d_T is tolerance diameter, in;

$$d_T = d_e - \frac{2}{P_{nd}} \quad (27)$$

d_e is outside diameter of pinion or gear, in.

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, W_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 28 and 29 are used to establish power ratings for unity service factor to which established service factors may be applied using equation 30. 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. Where specific experience and satisfactory performance has been demonstrated by successful use of established service factors, values of Z_N and Y_N of 1.0 may be appropriate.

From equation 5:

$$P_{acu} = \frac{\pi\, n_P\, F}{396\ 000} \frac{I}{K_v\, K_s\, K_m\, C_f} \left(\frac{d\, s_{ac}}{C_p} \frac{Z_N\, C_H}{K_T} \right)^2 \quad (28)$$

and from equation 14:

$$P_{atu} = \frac{\pi\, n_P\, d}{396\ 000\, K_v} \frac{F}{P_d\, K_s} \frac{J}{K_m\, K_B} \frac{s_{at}\, Y_N}{K_T} \quad (29)$$

where

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

P_{atu} 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.

$s_{ac}\, Z_N\, C_H$ for pitting resistance

$\frac{s_{at}\, Y_N\, J}{K_B}$ for bending strength

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

$$P_a = \text{the lesser of } \frac{P_{acu}}{C_{SF}} \text{ and } \frac{P_{atu}}{K_{SF}} \quad (30)$$

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 K_R 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, C_p

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

$$C_p = \sqrt{\frac{1}{\pi\left[\left(\frac{1-\mu_P^2}{E_P}\right) + \left(\frac{1-\mu_G^2}{E_G}\right)\right]}} \quad (31)$$

where

C_p is elastic coefficient, $[\text{lb/in}^2]^{0.5}$;

μ_P and μ_G is Poisson's ratio for pinion and gear, respectively;

E_P and E_G is modulus of elasticity for pinion and gear, respectively, lb/in^2.

For example, C_p equals 2300 $[\text{lb/in}^2]^{0.5}$, for a steel pinion and gear with μ=0.3 and E=3 $\times$ 10^7 lb/in^2 for both members.

13 Surface condition factor, C_f

The surface condition factor, C_f, 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, C_H

The hardness ratio factor, C_H, depends upon:

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

The value of C_H for the pinion is set at 1.0. The value of C_H for the gear is either 1.0 or as outlined in 14.1 or 14.2.

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 C_H are shown in figure 2.

The values from figure 2 can be calculated as follows:

$$C_H = 1.0 + A\left(m_G - 1.0\right) \tag{32}$$

where

$$A = 0.008\,98\left[\frac{H_{BP}}{H_{BG}}\right] - 0.008\,29 \tag{33}$$

H_{BP} is pinion Brinell hardness number, HB;

H_{BG} is gear Brinell hardness number, HB.

This equation is valid for the range

$1.2 \leq H_{BP} / H_{BG} \leq 1.7$

For $H_{BP} / H_{BG} < 1.2$, $A = 0.0$
$H_{BP} / H_{BG} > 1.7$, $A = 0.006\,98$

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 C_H factor varies with the surface finish of the pinion, f_p, and the mating gear hardness.

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

$$C_H = 1.0 + B\left(450 - H_{BG}\right) \tag{34}$$

where

$$B = 0.000\,75\ (e)^{-0.0112\left(f_p\right)} \tag{35}$$

e is base of natural or Napierian logarithms = 2.718 28

f_p is surface finish of pinion, microinches, Ra.

15 Load distribution factor, K_m

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:

<u>Manufacturing variation of gears</u>

- Lead, profile, spacing and runout of both the pinion and the gear.
- Tooth crowning and end relief.

<u>Assembly variations of installed gears</u>

- 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.

<u>Deflections due to applied loads</u>

- 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.

<u>Distortions due to thermal and centrifugal effects</u>

- 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.

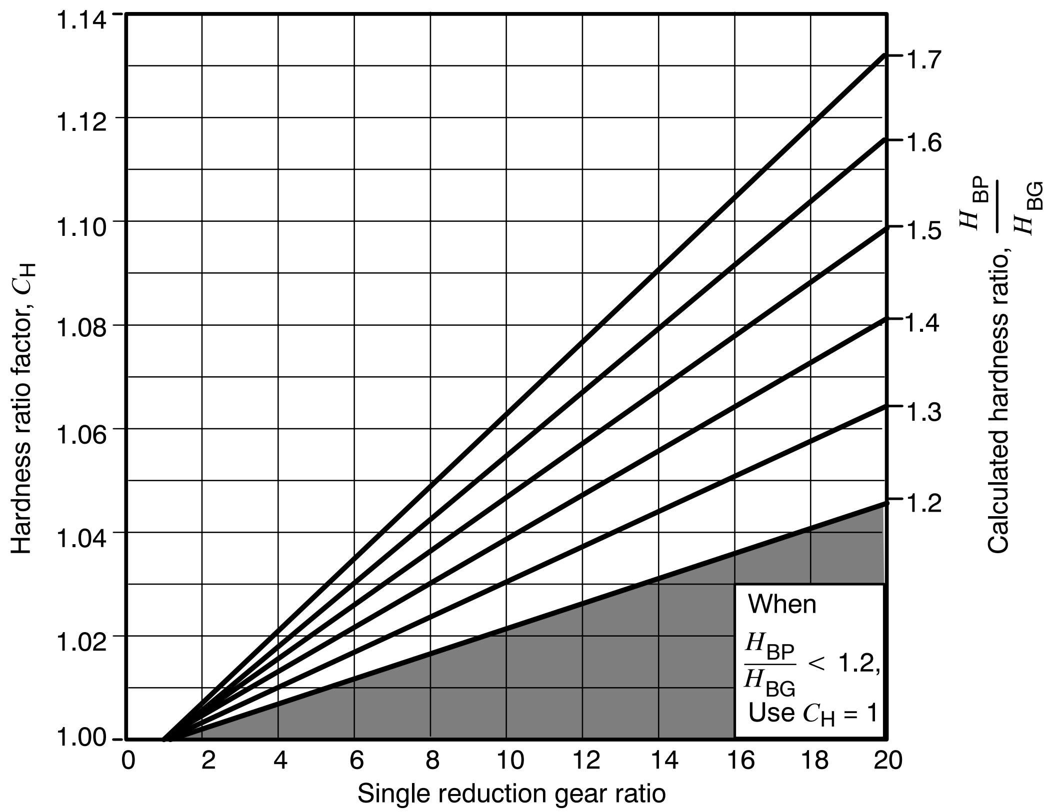


Figure 2 - Hardness ratio factor, C_H (through hardened)

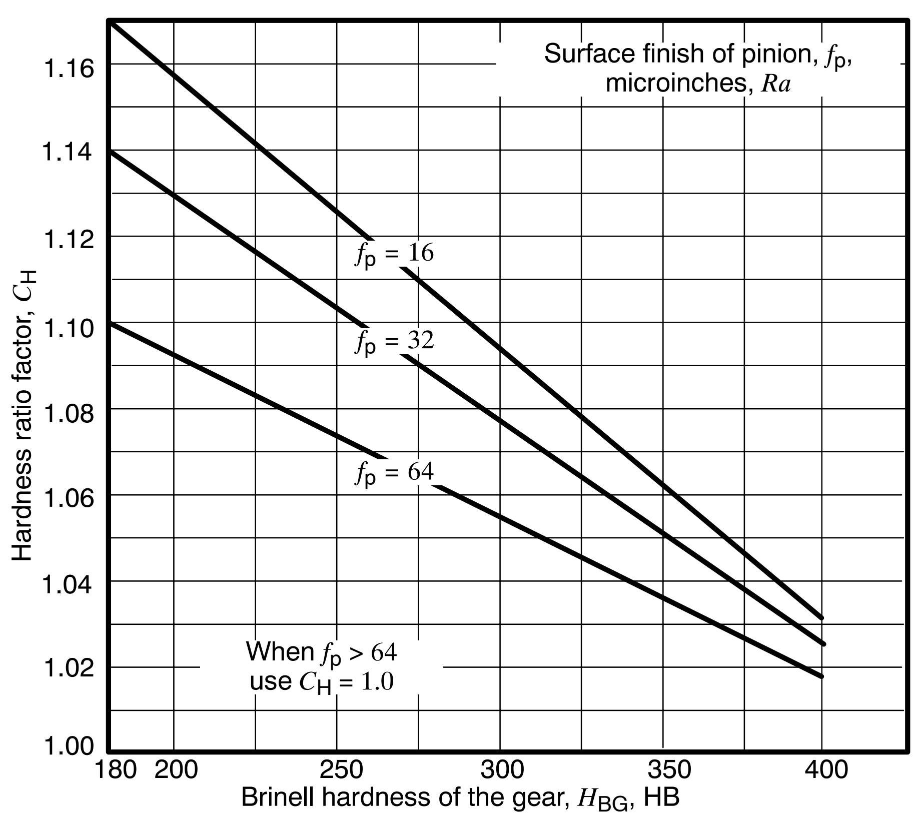


Figure 3 - Hardness ratio factor, C_H (surface hardened pinions)

15.1 Values for load distribution factor, K_m

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:

C_{mf} is face load distribution factor;
C_{mt} is transverse load distribution factor.

C_{mf} and C_{mt} 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,

$$K_m = f\left(C_{mf}\,,\ C_{mt}\right) \tag{36}$$

For helical gears, having three or more axial overlaps, the face load distribution factor, C_{mf}, 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, C_{mf} is affected primarily by lead and parallelism (see figure 4(B)). In this case, C_{mt} 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, C_{mf} and C_{mt} , 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, C_{mt}

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.

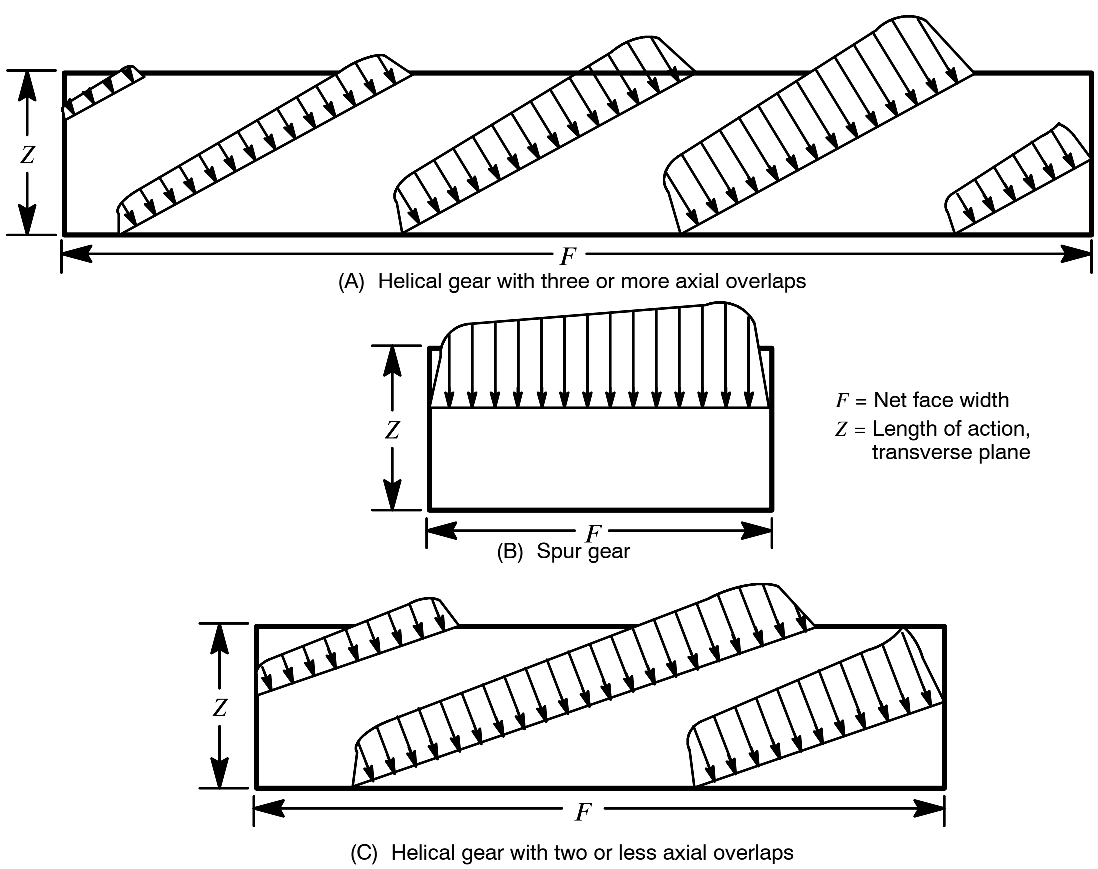


Figure 4 - Instantaneous contact lines in the plane of action

Standard procedures to evaluate the influence of C_{mt} 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 36 therefore, can be modified to:

$$K_m = C_{mf} \tag{37}$$

15.3 Face load distribution factor, C_{mf}

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. Also see AGMA 927-A01.

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, F/d, ≤ 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 40 inches.
- Contact across full face width of narrowest member when loaded.

CAUTION: If $F/d > 2.4 - 0.002K$ where K = the contact load factor (see equation 6), the value of K_m 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, C_{pm}. 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 F/d 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:

$$C_{mf} = 1.0 + C_{mc}\left(C_{pf}\,C_{pm} + C_{ma}\,C_e\right) \tag{38}$$

where

C_{mc} is lead correction factor;

C_{pf} is pinion proportion factor;

C_{pm} is pinion proportion modifier;

C_{ma} is mesh alignment factor;

C_e is mesh alignment correction factor.

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

C_{mc} is 1.0 for gear with unmodified leads;

C_{mc} is 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, C_{pf}, accounts for deflections due to load. These deflections are normally higher for wide face widths or higher F/d ratios. The pinion proportion factor can be obtained from figure 5.

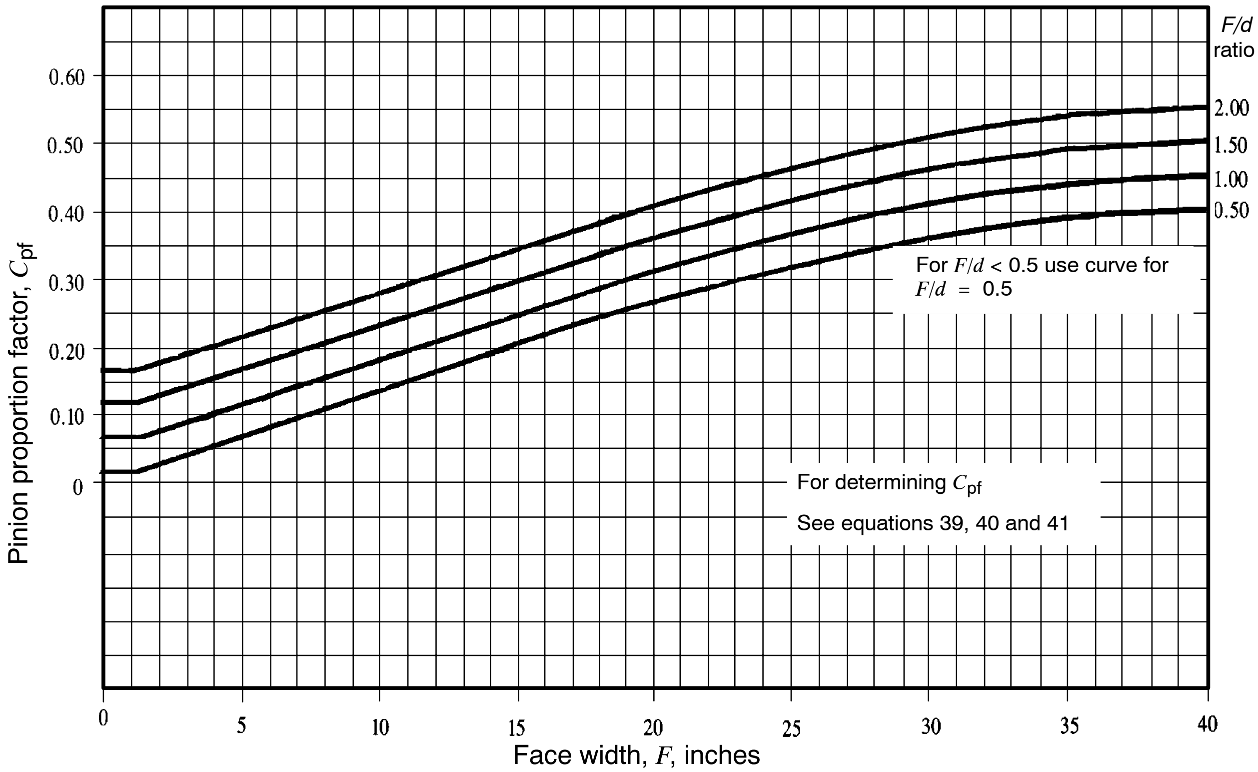


Figure 5 - Pinion proportion factor, C_{pf}

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

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

when $F \leq 1.0$

$$C_{pf} = \frac{F}{10\,d} - 0.025 \tag{39}$$

when $1.0 < F \leq 17$

$$C_{pf} = \frac{F}{10\,d} - 0.0375 + 0.0125F \tag{40}$$

when $17 < F \leq 40$

$$C_{pf} = \frac{F}{10\,d} - 0.1109 + 0.0207F - 0.000\,228\,F^2 \tag{41}$$

NOTE: For values of $\frac{F}{10\,d}$ less than 0.05, use 0.05 for this value in equations 39, 40 or 41.

The pinion proportion modifier, C_{pm}, alters C_{pf}, based on the location of the pinion relative to its bearing centerline.

C_{pm} is 1.0 for straddle mounted pinions with $(S_1/S) < 0.175$;

C_{pm} is 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, in (see figure 6);

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

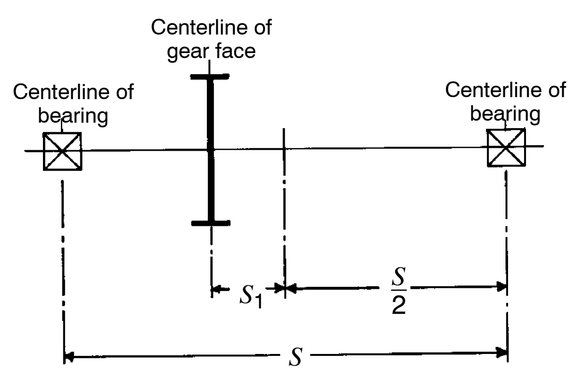


Figure 6 - Evaluation of S and S_1

The mesh alignment factor, C_{ma}, 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 C_{ma} based on the accuracy of gearing and misalignment effects which can be expected for the four classes of gearing shown.

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

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

$$C_{ma} = A + B(F) + C(F)^2 \tag{42}$$

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:

C_e is 0.80 when the gearing is adjusted at assembly;

is 0.80 when the compatibility of the gearing is improved by lapping;

is 1.0 for all other conditions.

When gears are lapped and mountings are adjusted at assembly, the suggested value of C_e is 0.80.

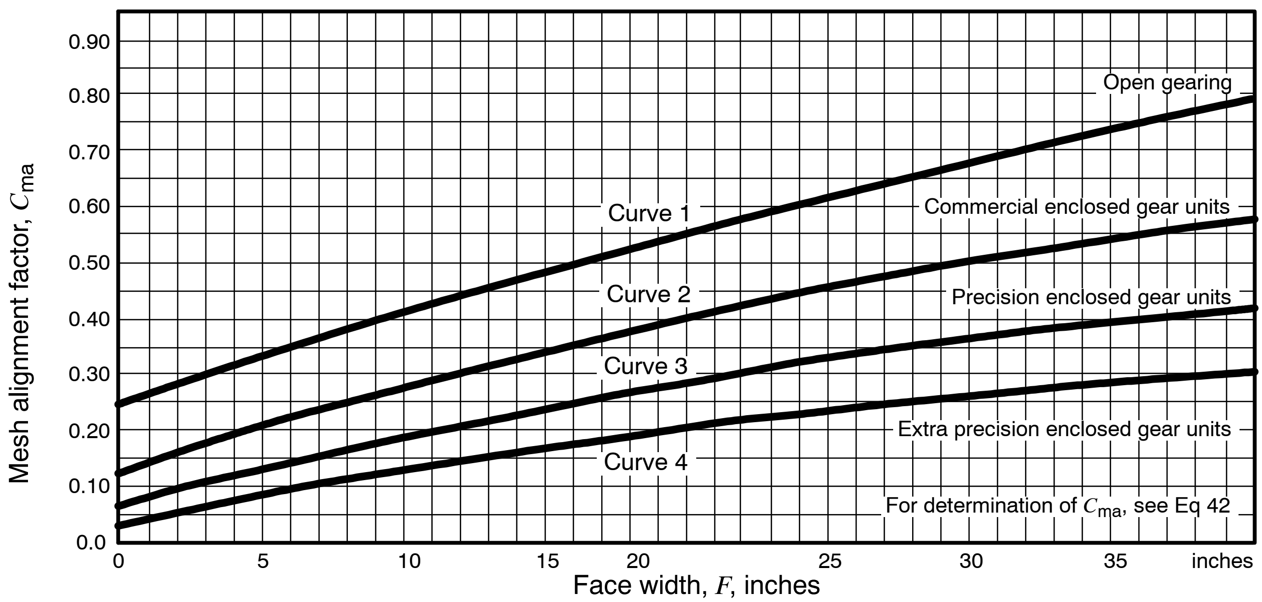


Figure 7 - Mesh alignment factor, C_{ma}

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

Curve	A	B	C
Curve 1 Open gearing	2.47×10^{-1}	0.167×10^{-1}	-0.765×10^{-4}
Curve 2 Commercial enclosed gear units	1.27×10^{-1}	0.158×10^{-1}	-1.093×10^{-4}
Curve 3 Precision enclosed gear units	0.675×10^{-1}	0.128×10^{-1}	-0.926×10^{-4}
Curve 4 Extra precision enclosed gear units	0.380×10^{-1}	0.102×10^{-1}	-0.822×10^{-4}

16 Allowable stress numbers, s_{ac} and s_{at}

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 s_{ac} and s_{at}, 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).

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.

Table 3 - Allowable contact stress number, s_{ac}, for steel gears

Material designation	Heat treatment	Minimum surface hardness[1)]	Allowable contact stress number[2)], s_{ac} lb/in^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	170 000	190 000	--
		54 HRC	175 000	195 000	--
	Carburized and hardened[5)]	see table 9	180 000	225 000	275 000
	Nitrided[5)] (through hardened steels)	83.5 HR15N	150 000	163 000	175 000
		84.5 HR15N	155 000	168 000	180 000
2.5% Chrome (no aluminum)	Nitrided[5)]	87.5 HR15N	155 000	172 000	189 000
Nitralloy 135M	Nitrided[5)]	90.0 HR15N	170 000	183 000	195 000
Nitralloy N	Nitrided[5)]	90.0 HR15N	172 000	188 000	205 000
2.5% Chrome (no aluminum)	Nitrided[5)]	90.0 HR15N	176 000	196 000	216 000

NOTES

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.

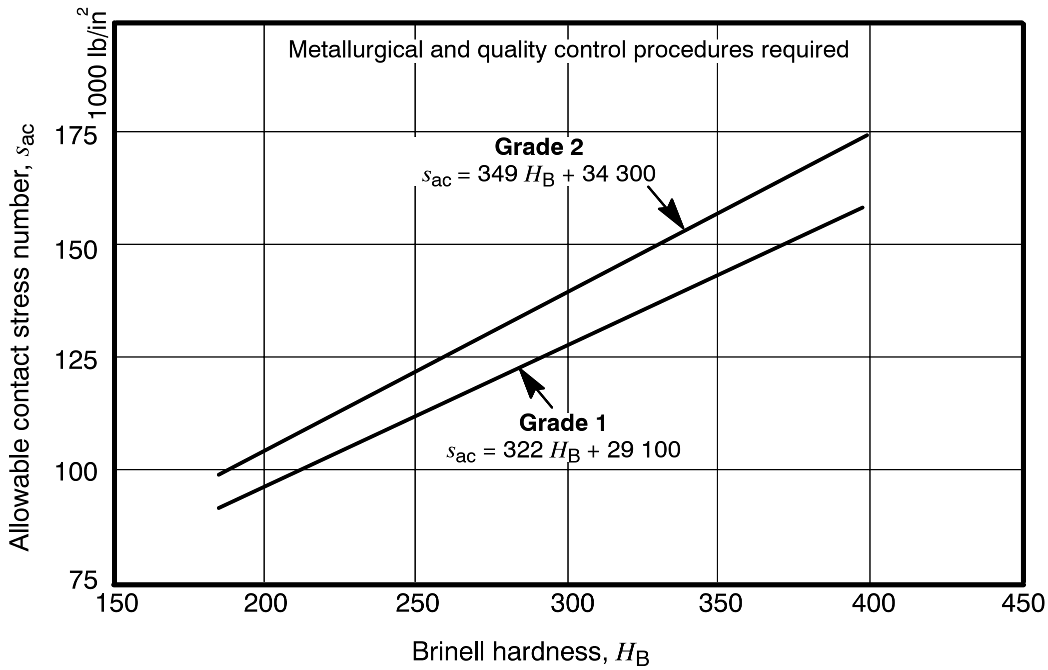


Figure 8 - Allowable contact stress number for through hardened steel gears, s_{ac}

Table 4 - Allowable bending stress number, s_{at}, for steel gears

Material designation	Heat treatment	Minimum surface hardness[1)]	Allowable bending stress number[2)], s_{at} lb/in²		
			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	45 000	55 000	--
	Flame[4)] or induction hardened[4)] with type B pattern[5)]	see table 8	22 000	22 000	--
	Carburized and hardened[4)]	see table 9	55 000	65 000 or 70 000[6)]	75 000
	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, 70,000 psi 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]

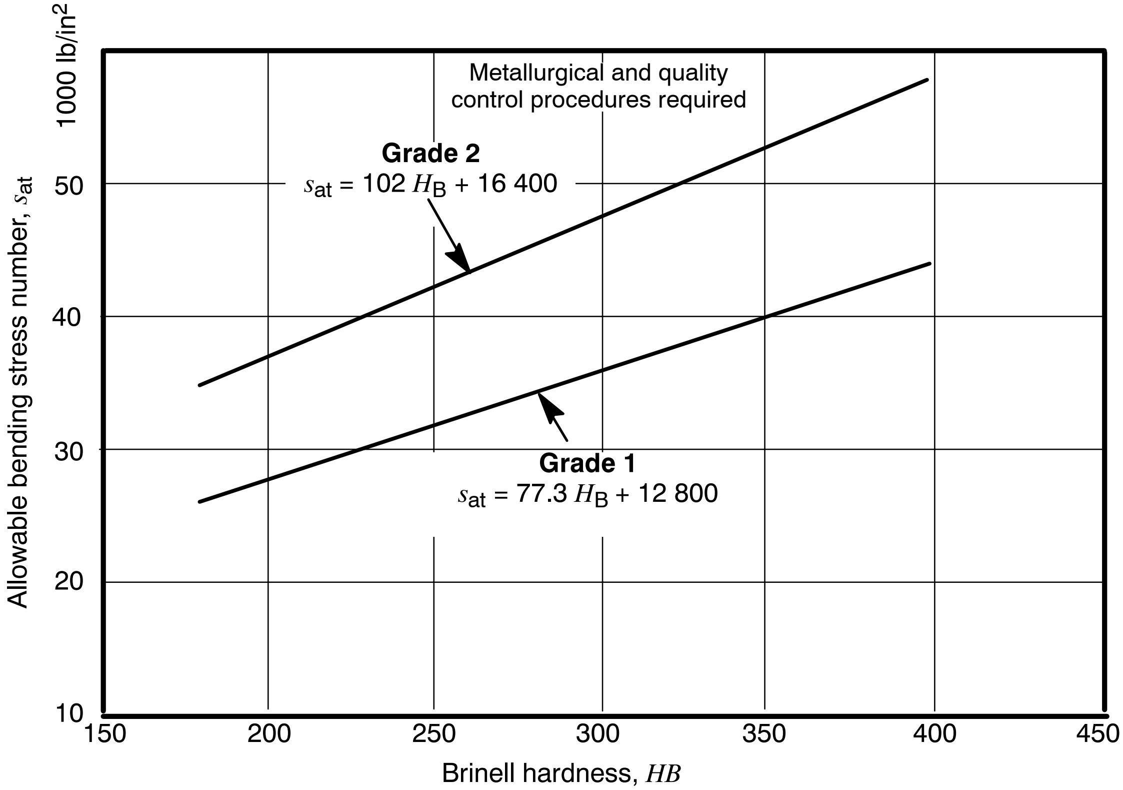


Figure 9 - Allowable bending stress number for through hardened steel gears, s_{at}

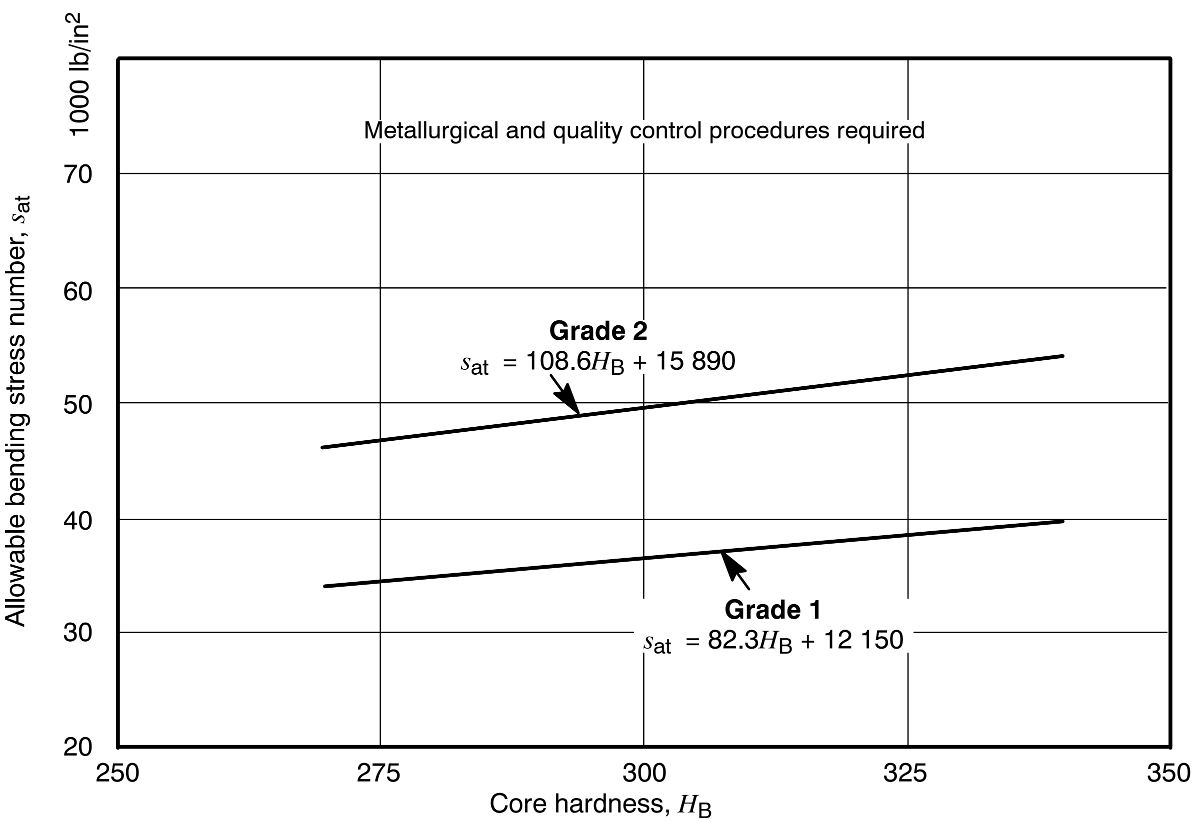


Figure 10 - Allowable bending stress numbers for nitrided through hardened steel gears (i.e., AISI 4140, AISI 4340), s_{at}

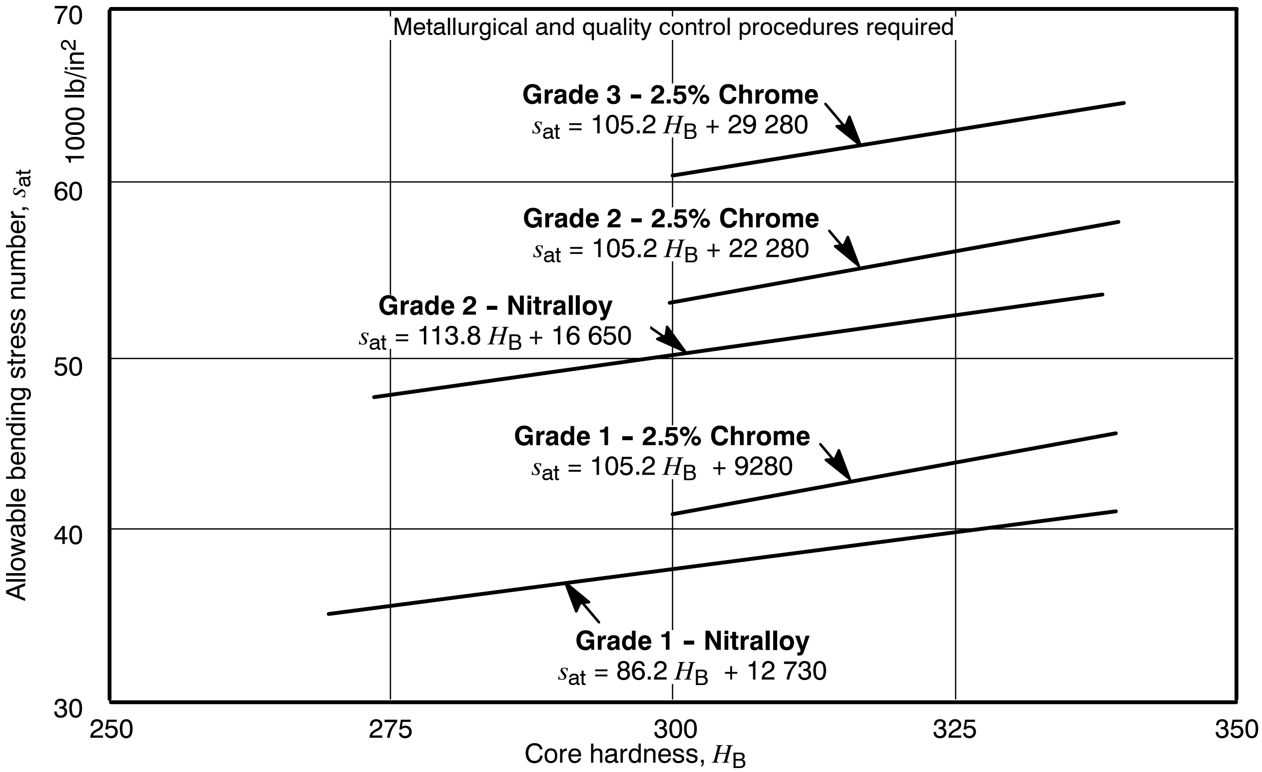


Figure 11 - Allowable bending stress numbers for nitriding steel gears, s_{at}

Table 5 - Allowable contact stress number, s_{ac}, for iron and bronze gears

Material	Material designation[1)]	Heat treatment	Typical minimum surface hardness[2)]	Allowable contact stress number[3)] s_{ac}, lb/in²
ASTM A48 Gray Cast Iron	Class 20	As cast	--	50 000 - 60 000
	Class 30	As cast	174 HB	65 000 - 75 000
	Class 40	As cast	201 HB	75 000 - 85 000
ASTM A536 Ductile (Nodular) Iron	Grade 60-40-18	Annealed	140 HB	77 000 - 92 000
	Grade 80-55-06	Quenched & tempered	179 HB	77 000 - 92 000
	Grade 100-70-03	Quenched & tempered	229 HB	92 000 - 112 000
	Grade 120-90-02	Quenched & tempered	269 HB	103 000 - 126 000
Bronze	--	Sand cast	Minimum tensile strength 40 000 lb/in²	30 000
	ASTM B-148 Alloy 954	Heat treated	Minimum tensile strength 90 000 lb/in²	65 000

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.

Table 6 - Allowable bending stress number, s_{at}, for iron and bronze gears

Material	Material designation[1)]	Heat treatment	Typical minimum surface hardness[2)]	Allowable bending stress number[3)], s_{at} lb/in^2
ASTM A48 Gray Cast Iron	Class 20	As cast	--	5000
	Class 30	As cast	174 HB	8500
	Class 40	As cast	201 HB	13 000
ASTM A536 Ductile (Nodular) Iron	Grade 60-40-18	Annealed	140 HB	22 000 - 33 000
	Grade 80-55-06	Quenched & tempered	179 HB	22 000 - 33 000
	Grade 100-70-03	Quenched & tempered	229 HB	27 000 - 40 000
	Grade 120-90-02	Quenched & tempered	269 HB	31 000 - 44 000
Bronze		Sand cast	Minimum tensile strength 40 000 lb/in^2	5700
	ASTM B-148 Alloy 954	Heat treated	Minimum tensile strength 90 000 lb/in^2	23 600

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.

Table 7 - Major metallurgical factors affecting the allowable contact stress number, s_{ac}, and allowable bending stress number, s_{at}, 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, inches (see annex F) / Max upper transformation products at 400X to 10.0 incl — 10% Over 10.0 — 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, s_{ac} only	See figure 8	See figure 8
Specified hardness at root, s_{at} 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 s_{ac} and table 4 for values of s_{at}. 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 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 32° F, see 3.6.1.

Table 8 - Major metallurgical factors affecting the allowable contact stress number, s_{ac}, and allowable bending stress number, s_{at}, of flame or induction hardened steel gears[1) 2) 3)]

Metallurgical factor	Grade 1	Grade 2	
ASTM E112 grain size	Predominantly 5 or finer	Predominantly 5 or finer	
Material composition	Not specified	Medium carbon alloy steel	
Prior structure	Not specified	Quenched and tempered	
Material form	Not specified	Forgings and wrought steel; castings with magnetic particle inspection of gear tooth area	
Cleanliness[4)]	Not specified	AMS 2301 or ASTM A866 for wrought steel (certification not required); castings are permissible with primarily round (Type 1) sulfide inclusions.	
Sulfur content	Not specified	0.025% maximum for wrought 0.040% maximum for castings	
Core hardness, center of tooth at root diameter, s_{ac} only	Not specified	28 HRC minimum	
Core hardness, center of tooth at root diameter, s_{at} only	Not specified	Type A - 28 HRC minimum Type B - not specified	
Non-martensitic transformation products in hardened zone	Limited by effect on specified hardness	10% maximum, no free ferrite	
Surface hardness, s_{ac} only	See table 3	See table 3	
Surface hardness at root, s_{at} only	Type A - 50 HRC min Type B - not specified	Type A - 54 HRC min Type B - not specified	
Hardness pattern (see figure 12), s_{at} only	As required per table 4	Type A - Contour pattern with a ductile core Type B - not specified	
Magnetic particle (method per ASTM E709 on teeth)[5)]	Not specified	Pitch P_{nd}	Maximum indication, inch
Magnetic particle (method per ASTM E709 on teeth)[5)]	Not specified	≤3 >3 to <10 ≥10	1/8 3/32 1/16

NOTES

1) See table 3 for values of s_{ac} and table 4 for values of s_{at}. 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 inch of face width and maximum of five in one tooth flank. No indications allowed below 1/2 working depth of tooth. Indications smaller than 1/64 inch 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, s_{ac}, and allowable bending stress number, s_{at}, 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	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), s_{ac} 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	Pitch P_{nd} / Maximum indication, inch ≤3 / 1/8 >3 to <10 / 3/32 ≥10 / 1/16	Pitch P_{nd} / Maximum indication, inch ≤3 / 3/32 >3 to <10 / 1/16 ≥10 / 1/32
Decarburization in case (to 0.005 inch depth), s_{ac} 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 0.005 inch depth), s_{at} 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, s_{at} 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 0.0001 in^2 field at 400X
Secondary transformation products, (upper bainite) in case along flank above root, or on representative coupon, to 0.010 inch deep, s_{ac} 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 0.010 inch deep, s_{at} 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, in / IGO, in <0.030 / 0.0007 $0.030 \le h_e < 0.059$ / 0.0010 $0.059 \le h_e < 0.089$ / 0.0015 $0.089 \le h_e < 0.118$ / 0.0020 ≥0.118 / 0.0024 If excessive, salvage is allowed by controlled shotpeening, with the agreement of the customer.	Case depth, in / IGO, in <0.030 / 0.0005 $0.030 \le h_e < 0.059$ / 0.0008 $0.059 \le h_e < 0.089$ / 0.0008 $0.089 \le h_e < 0.118$ / 0.0010 ≥0.118 / 0.0012 (If excessive, salvage is allowed by controlled shotpeening, with the agreement of the customer — spans Grades 2 and 3)

(continued)

Table 9 *(concluded)*

Metallurgical factor[4) 5)]	Grade 1	Grade 2	Grade 3
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), s_{ac} only[11)]	Not specified	21 HRC minimum	21 HRC minimum
Core hardness (at center of tooth at root diameter or on representative coupon), s_{at} 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, s_{at} only	Not specified	Recommended if the root is ground	Required in tooth root area

NOTES

1) See table 3 for values of s_{ac}, and table 4 for values of s_{at}. 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 $6/P_{nd}$ and minimum length $12/P_{nd}$ are used in ISO 6336-5. Microhardness is to be measured on the test coupon at a depth not more than 0.003 inch below the depth corresponding to the finished tooth surface.

5) For low temperature service, 32°F, 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 350°F.

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 8/64 inch 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 1.0 inch below roots) and Level 2 in Zone 2 (remainder of rim) using 8/64 inch 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 inch of face width and maximum of five in one tooth flank. No indications allowed below 1/2 working depth of tooth. Indications smaller than 1/64 inch 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 300° F 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 6 P_{nd}. 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, s_{ac}, and allowable bending stress number, s_{at}, of 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)	0.0010 inch	0.0008 inch	0.0005 inch
Upper transformation products which primarily include bainite and fine pearlite[5)]	Not specified	Max controlling section, in. (see annex F). — Max upper transformation products @ 400X to 10.0 inc. — 10% Over 10.0 — 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	Pitch P_{nd} — Maximum indication, inch ≤ 3 — 3/32 >3 to <10 — 1/16 ≥ 10 — 1/32
Grinding burns	Not specified	See note 8	See note 8

NOTES

1) See table 3 for values of s_{ac}, and table 4 for values of s_{at}.

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).

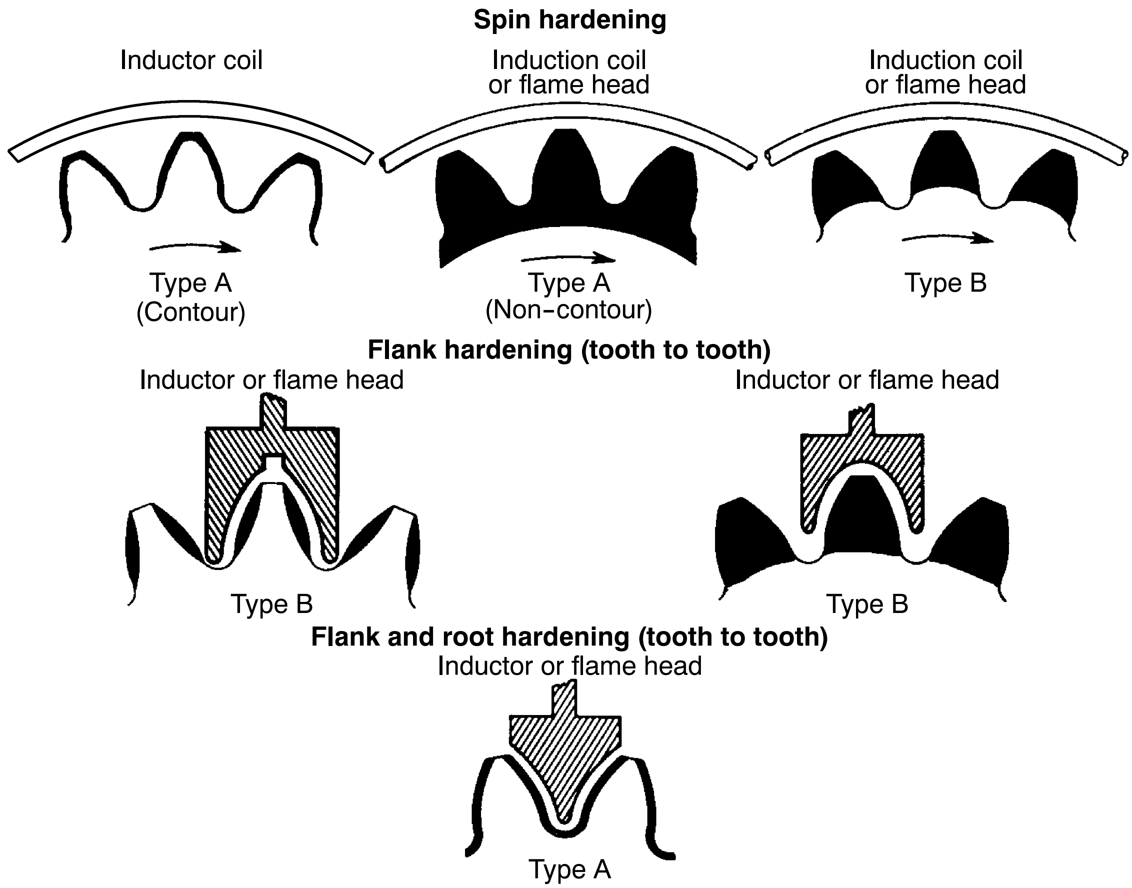
4) The grade cleanliness requirements apply only to those portions of the gear material where the teeth will be located to a distance below the finished tip diameter of at least two times the tooth depth. On external gears, this portion of the gear blank normally will be less than 25 percent of the radius.

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 8/64 inch FBH (8-0400) per ASTM E428 (also described in ANSI/AGMA 6033-A88). A distance amplitude correction curve is not in tended. 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 1.0 inch below roots) and Level 2 in Zone 2 (remainder of rim) using 8/64 inch 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 inch of face width and maximum of five in one tooth flank. No indications allowed below 1/2 working depth of tooth. Indications smaller than 1/64 inch 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

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.

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, $h_{\mathrm{e\ min}}$, 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 [6]:

$$h_{\mathrm{e\ min}} = \frac{s_{\mathrm{c}}\, d\, \sin\phi_{\mathrm{t}}}{U_{\mathrm{H}} \cos\psi_{\mathrm{b}}} C_{\mathrm{G}} \qquad (43)$$

where

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

s_c is contact stress number lb/in^2. The maximum value recommended is 200 000 lb/in^2 for this equation;

ϕ_t is operating transverse pressure angle;

U_H is hardening process factor, lb/in^2;

= 6.4 x 10^6 lb/in^2 for carburized and hardened;

= 4.4 x 10^6 lb/in^2 for tooth-to-tooth induction hardened;

ψ_b 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 43.

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:

$$h_{e\,max} = \text{the lesser of } \frac{0.4}{P_{nd}} \text{ or } 0.56\,t_o \qquad (44)$$

where

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

t_o is normal tooth thickness at the top land of the gear in question, in.

If $h_{e\,min}$ from equation 43 (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 coarser pitch 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:

$$h_{c\,min} = \frac{U_c\, s_c\, d\, \sin\phi_t}{1.66 \times 10^7 \cos\psi_b} C_G \qquad (45)$$

where

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

U_c is core hardness coefficient, from figure 14.

If the value of $h_{c\,min}$ from equation 45 is less than the value for normal case depth from figure 15, then the minimum value from figure 15 should be used.

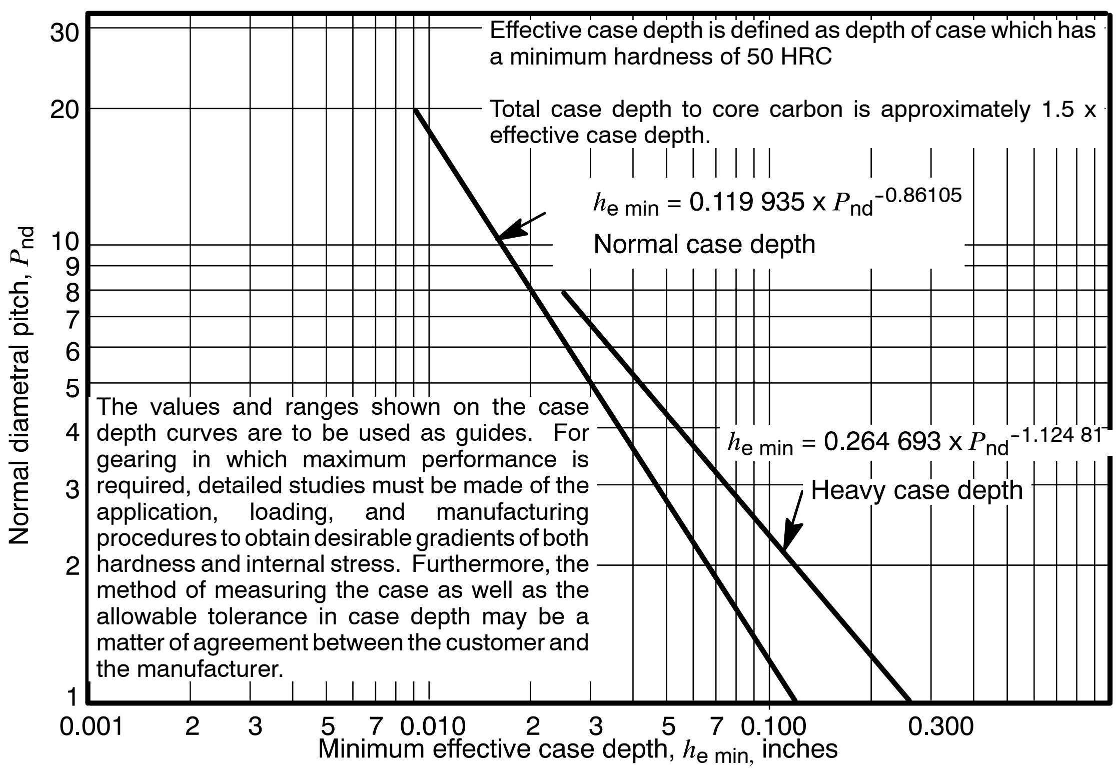
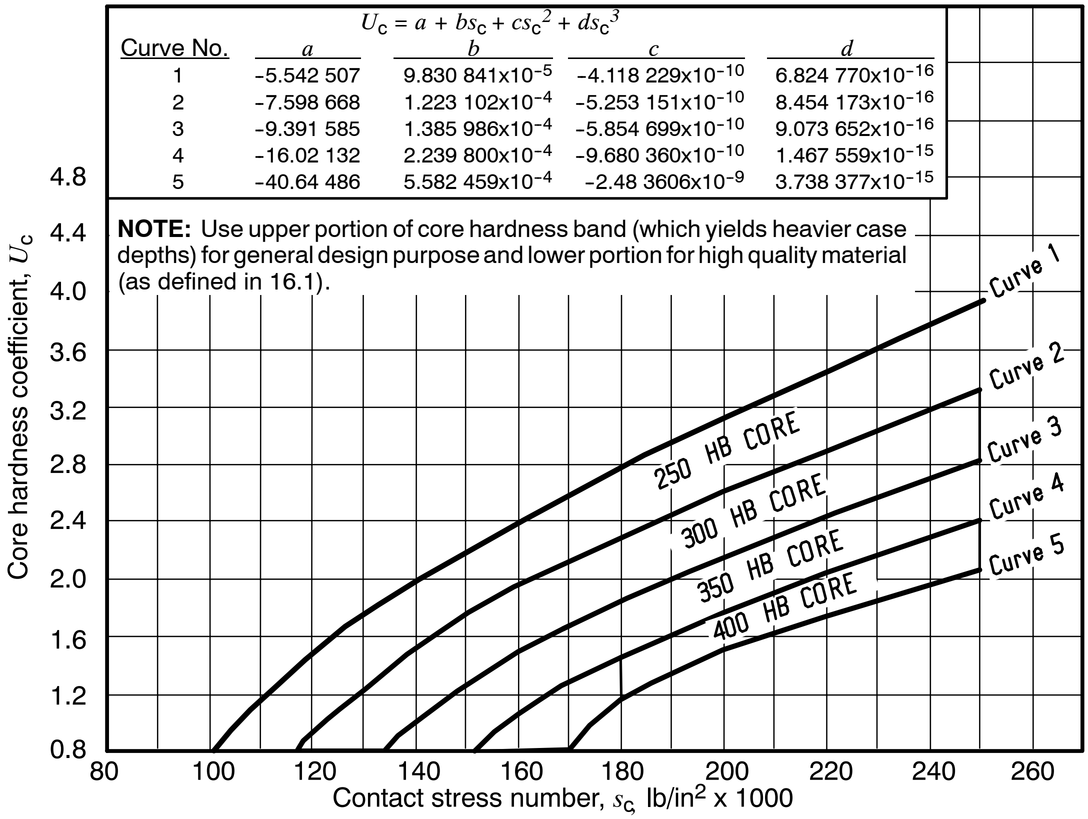


Figure 13 - Minimum effective case depth for carburized gears, $h_{e\,min}$



$$U_c = a + bs_c + cs_c^2 + ds_c^3$$

Curve No.	a	b	c	d
1	−5.542 507	9.830 841x10^{-5}	−4.118 229x10^{-10}	6.824 770x10^{-16}
2	−7.598 668	1.223 102x10^{-4}	−5.253 151x10^{-10}	8.454 173x10^{-16}
3	−9.391 585	1.385 986x10^{-4}	−5.854 699x10^{-10}	9.073 652x10^{-16}
4	−16.02 132	2.239 800x10^{-4}	−9.680 360x10^{-10}	1.467 559x10^{-15}
5	−40.64 486	5.582 459x10^{-4}	−2.48 3606x10^{-9}	3.738 377x10^{-15}

Figure 14 - Core hardness coefficient, U_c

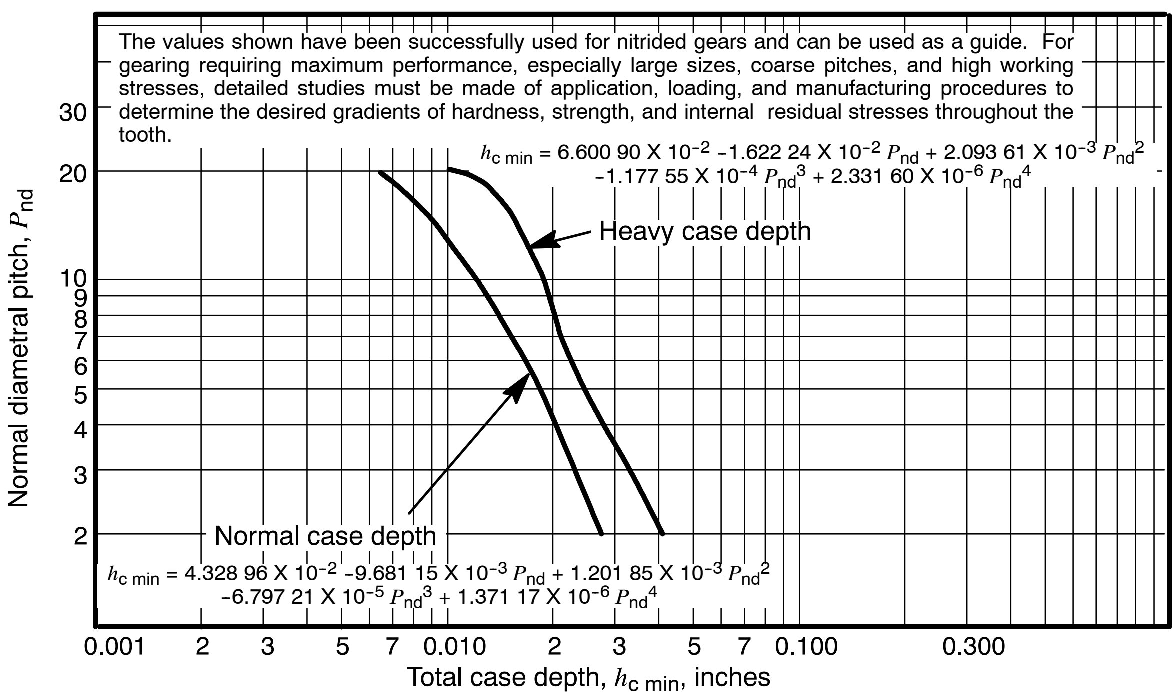


Figure 15 - Minimum total case depth for nitrided gears, $h_{c\,min}$

16.2 Reverse loading

Use 70 percent of the s_{at} 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 s_{ay}. 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 46, 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.

$$s_{ay}\, K_y \geq W_{max} \frac{P_d}{F} \frac{K_m}{J\, K_f} \tag{46}$$

where

s_{ay} is allowable yield strength number from figure 16, lb/in^2;

K_y is yield strength factor from the following tabulation;

Requirements of application	K_y
Conservative practice	0.50
Industrial practice	0.75

W_{max} is maximum peak tangential load, lb;

K_f is stress correction factor (see AGMA 908-B89).

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.

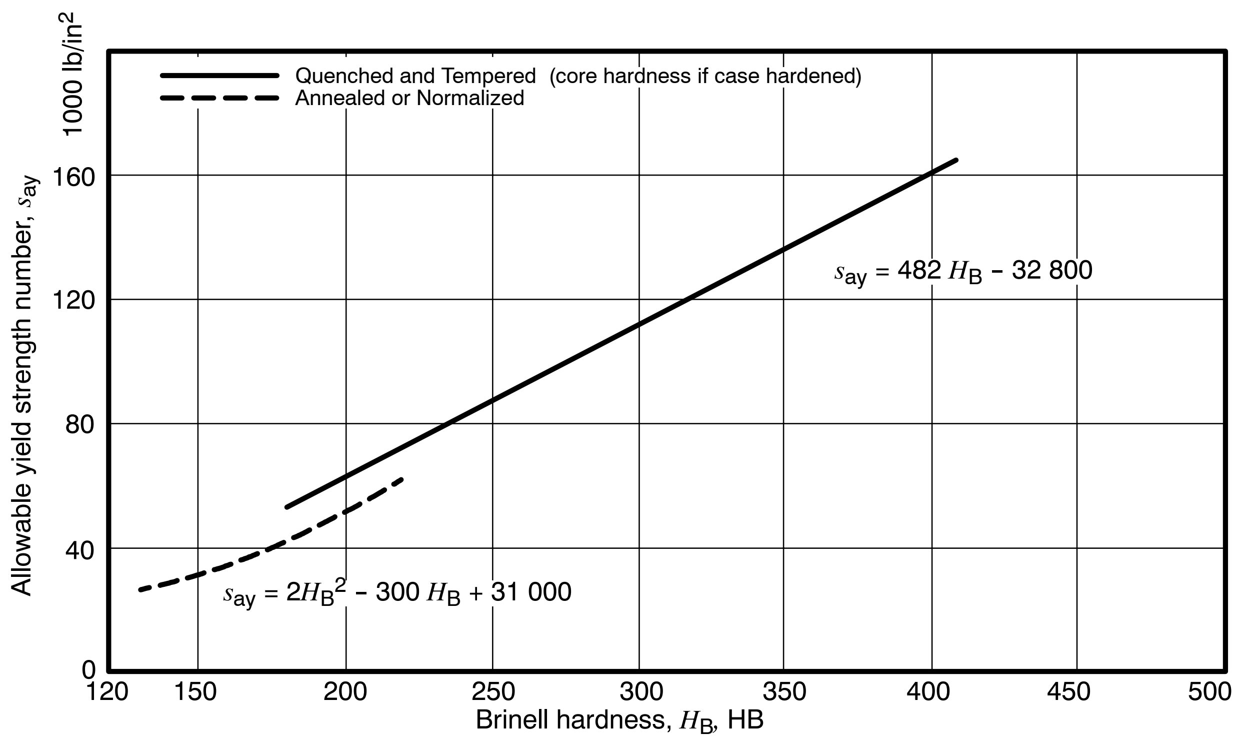


Figure 16 - Allowable yield strength number for steel gears, s_{ay}

K_{my} is load distribution factor under overload condition.

For a straddle mounted gear or pinion in an enclosed drive, K_{my} may be calculated from equation 47.

$$K_{my} = 0.0144\, F + 1.07 \tag{47}$$

Equation 47 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, 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 number of stress cycles that are appropriate for the application. The number of stress cycles, N, is used to determine the stress cycle factor as follows:

$$N = 60\, Lnq \tag{48}$$

where

N is the number of stress cycles;

L is life (hours);

n 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 stop at 10^{10} due to insufficient data at the time the standard was developed. Application beyond this point must be reviewed. 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.

Intermediate values of Y_N for hardnesses of through hardened gearing between 1×10^3 and 3×10^6 may be approximated by first determining the value using logarithmic interpolation at $N = 10^3$ cycles (see figure 18). The second point of a straight line for the desired hardness on a log-log plot is at 3×10^6 cycles where $Y_N = 1.04$. Below 1×10^3 cycles the value is a constant. An equation for the line between 1×10^3 and 3×10^6 would be of the form as shown in the figure. Above 3×10^6 cycles, the values within the existing figure are to be used.

17.3 Localized yielding

If the product of $s_{at}\, Y_N$ exceeds the allowable yield stress, s_{ay}, 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 yielding of the gear tooth.

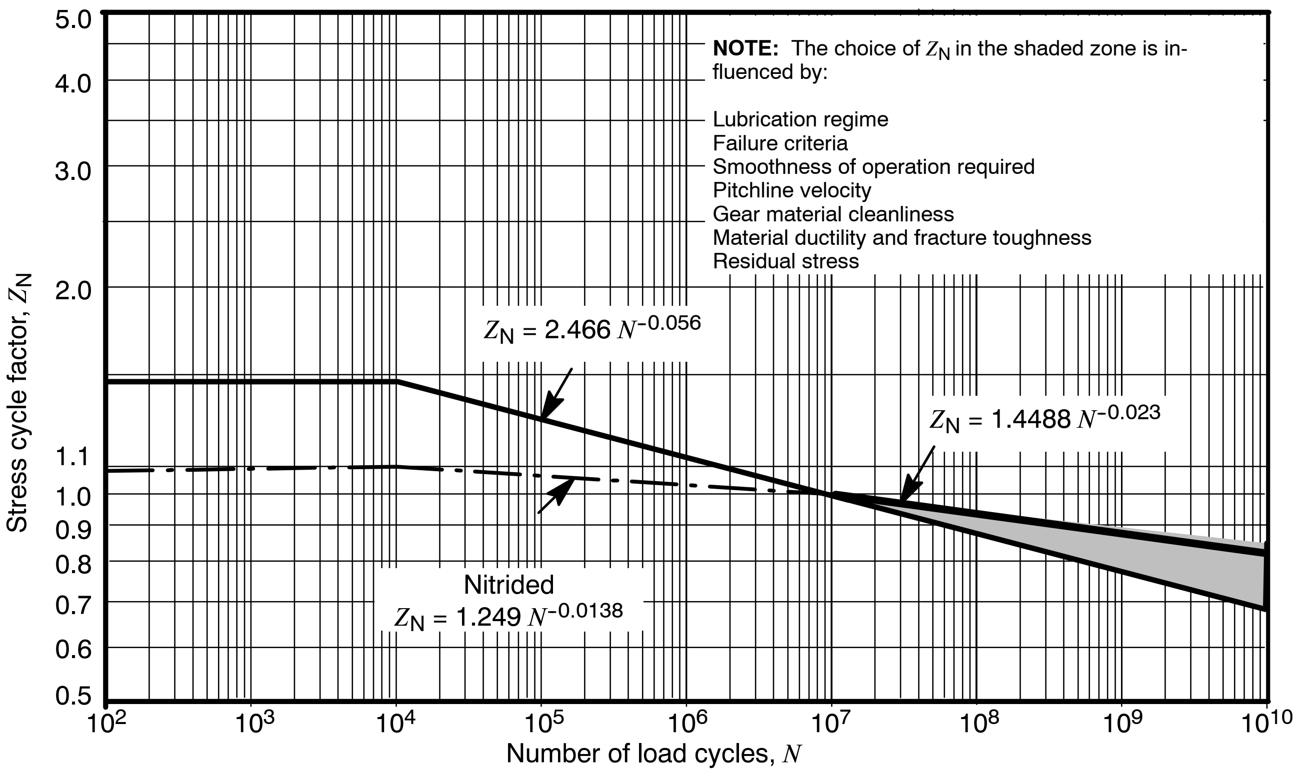


Figure 17 - Pitting resistance stress cycle factor, Z_N

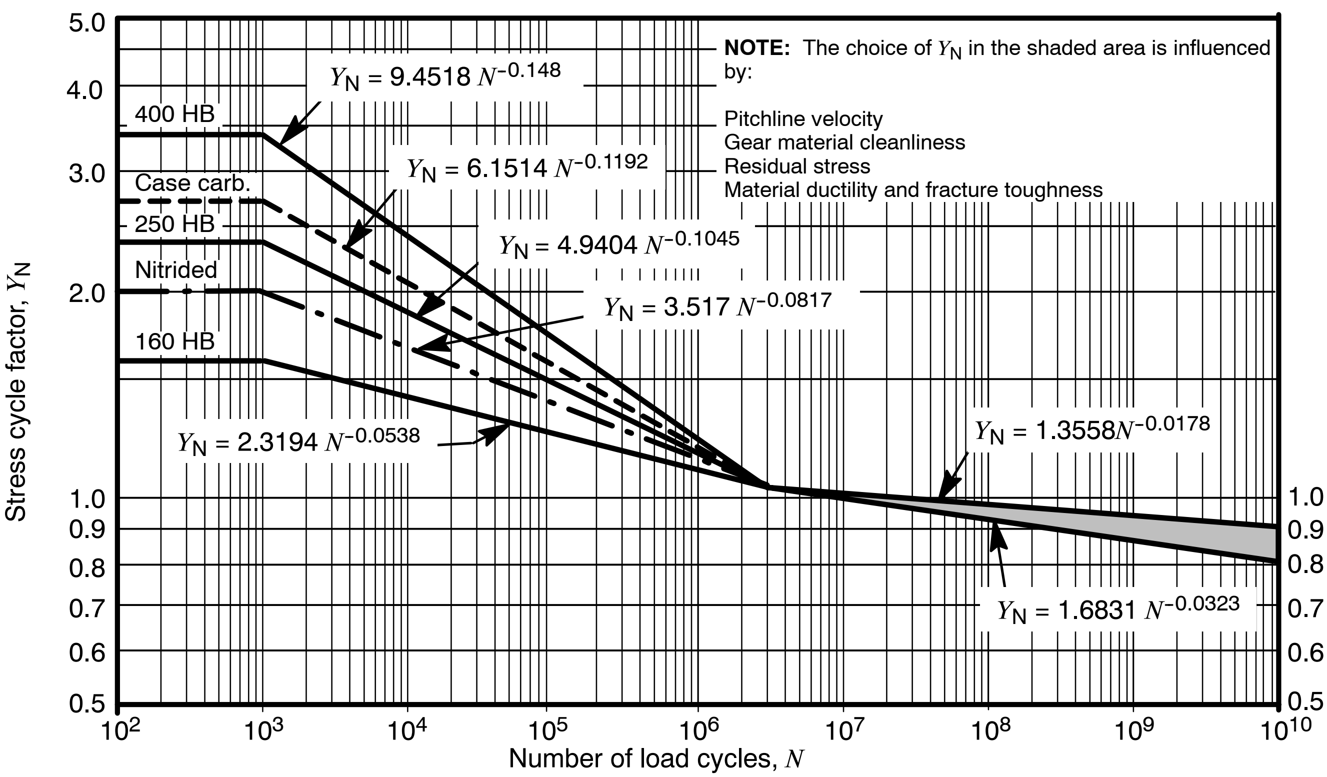


Figure 18 - Bending strength stress cycle factor, Y_N

18 Reliability factor, K_R

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.

When strength rating is based on yield strength, s_{ay}, the values of K_y from 16.4 should be used instead of K_R.

Table 11 - Reliability factors, K_R

Requirements of application	K_R 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 K_R is selected for bending.
2) At this value plastic flow might occur rather than pitting.
3) From test data extrapolation.

19 Temperature factor, K_T

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 250°F. When operating temperatures result in gear blank temperatures below 32° F, special care must be given, see 3.6.1.

19.2 High temperature operation

When operating at oil or gear blank temperature above 250°F, K_T 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 300°F.

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 determination of dynamic factor with AGMA 2000-A88

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ANSI/AGMA 2001-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.*]

A.1 Purpose

The standard method for the determination of dynamic factor is given in clause 8, with the use of ANSI/AGMA 2015-1-A01. This annex provides an alternative method using the older standard AGMA 2000-A88. A specific geometry, procedure and operating conditions should result in a comparable dynamic factor using this annex or clause 8.

A.2 Approximate dynamic factor, K_v

Figure A.1 shows dynamic factors which can be used in the absence of specific knowledge of the dynamic loads. The curves of figure A.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.

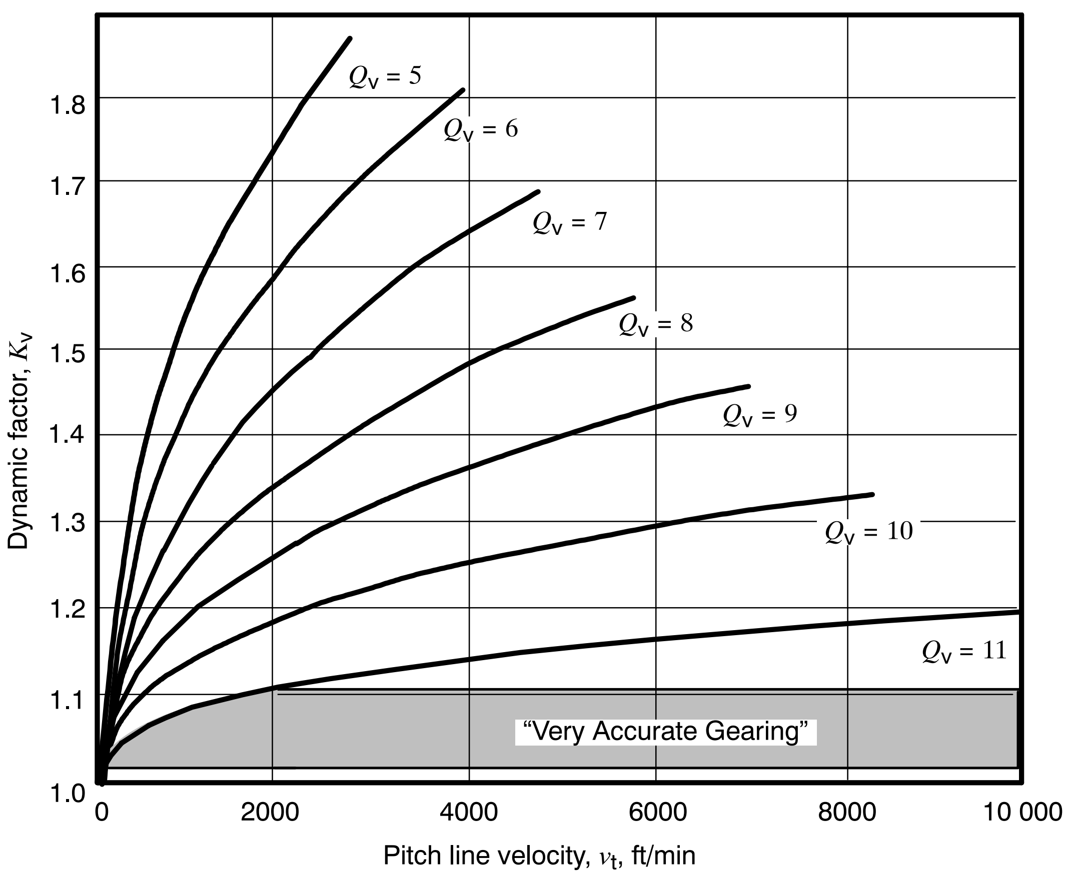


Figure A.1 - Dynamic factor, K_v

A.2.1 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.

A.2.2 Curves labeled $Q_v = 5$ through $Q_v = 11$

The empirical curves of figure A.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.

Q_v can be estimated as the appropriate quality number for the expected pitch and profile variations in accordance with 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 A.1 based on experience and careful consideration of the factors influencing dynamic load. For purposes of calculation, equation A.4 defines the end points of the curves in figure A.1.

$$K_v = \left(\frac{A + \sqrt{v_t}}{A} \right)^B \tag{A.1}$$

where

$$A = 50 + 56(1.0 - B) \text{ for } 5 \leq Q_v \leq 11 \tag{A.2}$$

$$B = 0.25(12 - Q_v)^{0.667} \tag{A.3}$$

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

$$v_{t\,max} = [A + (Q_v - 3)]^2 \tag{A.4}$$

where

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

Annex B
(informative)
Rim thickness factor, K_B

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ANSI/AGMA 2001-D04, *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, in;

h_t is whole depth, in.

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.

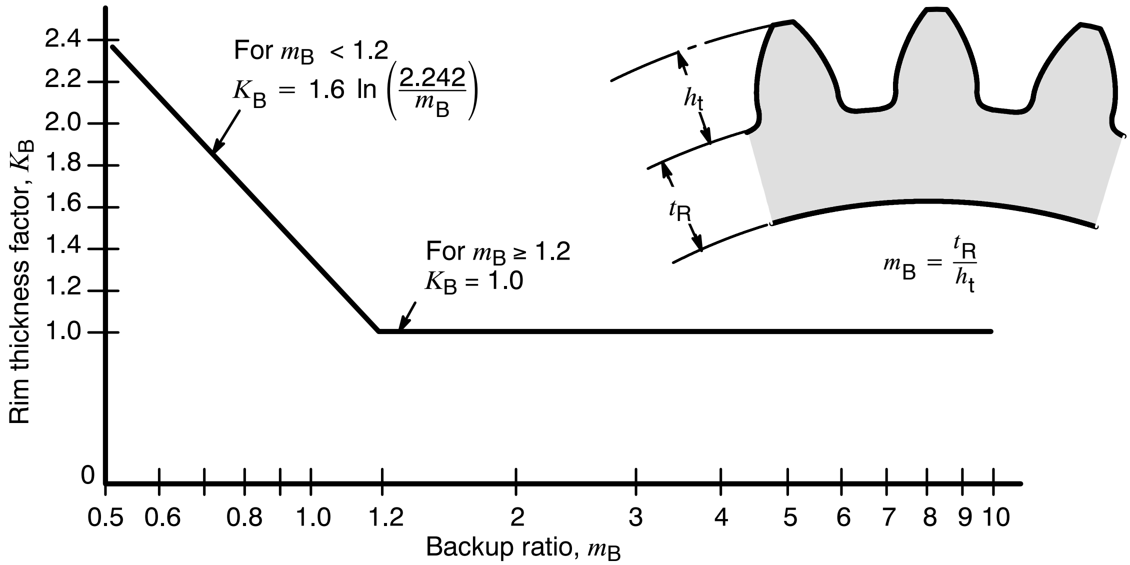


Figure B.1 - Rim thickness factor, K_B

Annex C
(informative)
Application analysis

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ANSI/AGMA 2001-D04, *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, K_R, 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 factor 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

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ANSI/AGMA 2001-D04, *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 ANSI/AGMA 2001-D04.

D.3 Empirical versus analytical method

The current rating practice of ANSI/AGMA 2001-D04 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 2001-D04 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 2001-D04. 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 2001-D04.

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.

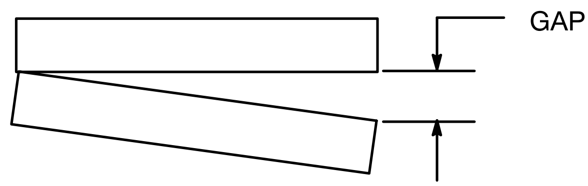


Figure D.1 - Gap due to manufacturing or installation errors

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_m is calculated. This technique is presented in references [1] and [2]. Tooth stiffness values in the range of 1.5×10^6 to 3.0×10^6 lbs/in^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.

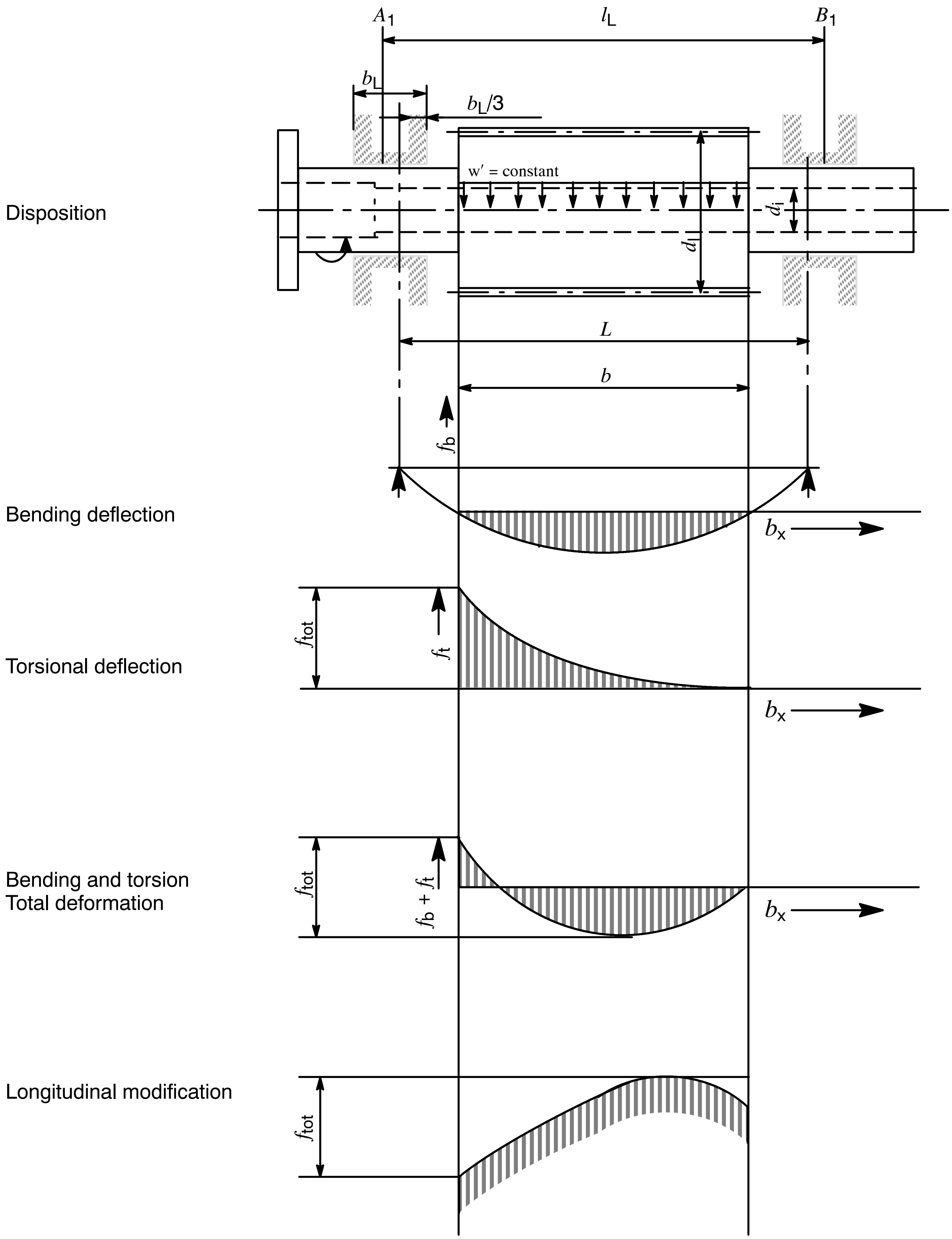


Figure D.2 - Elastic deformation of the pinion [3]

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-D04, *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).

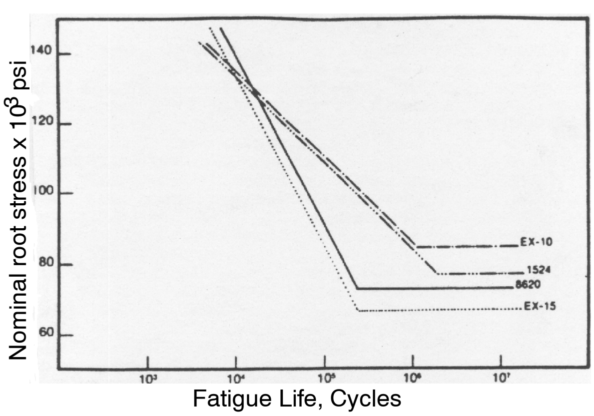


Figure 9 - Median S-N curves for carburized low alloy steel gears

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).

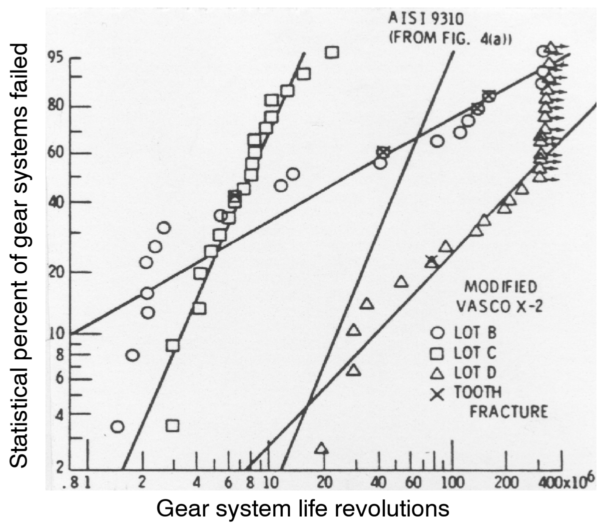


Figure 11 - Surface pitting fatigue lives of CVM modified VASCO X-2 spur gears heat treated to different specifications. (note AISI 9310)

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).

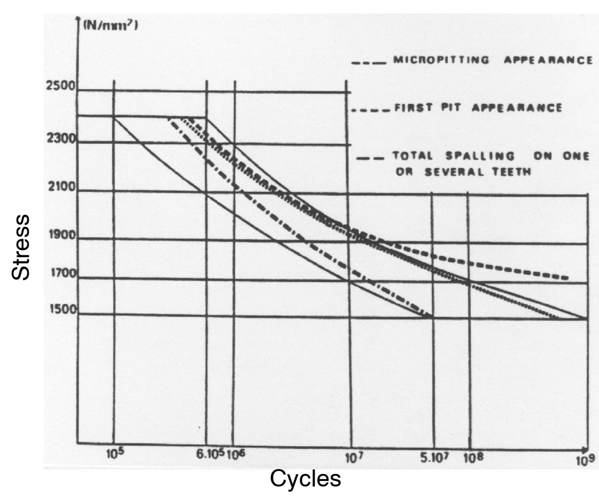


Figure 5 - Endurance curve to superficial pressure obtained with 20MC5 steel

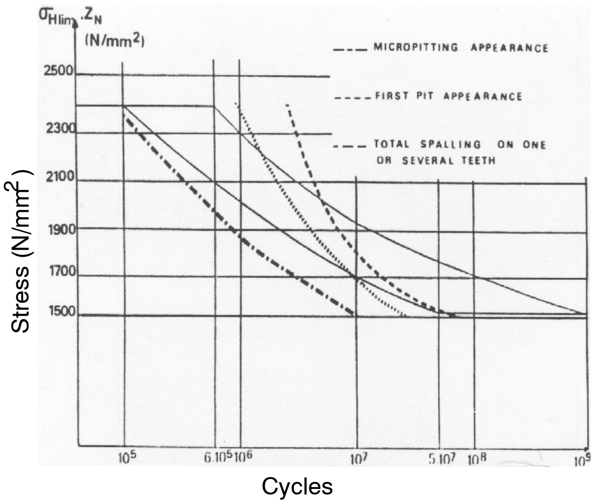


Figure 6 - Endurance curve to superficial pressure obtained with XC18 steel

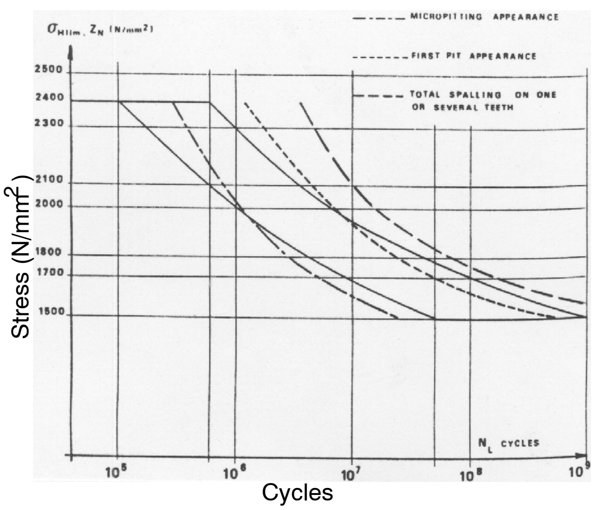


Figure 7 - Endurance curve to superficial pressure obtained with 16NC6 steel

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).

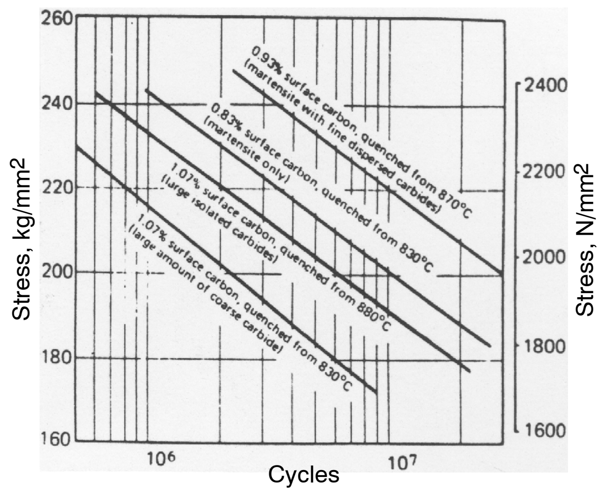


Figure 7 - The contact fatigue strength of carburized 25Kh2GHTA steel[12]

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)

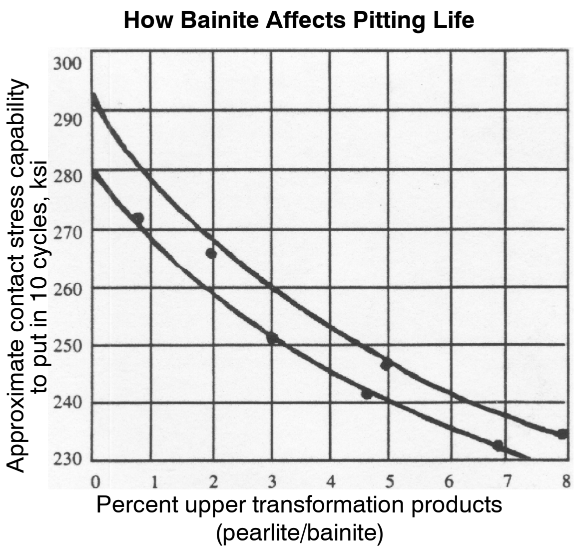


Figure 6 - Bainite, also called quenching pearlite, is soft, and deleterious to pitting life.

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).

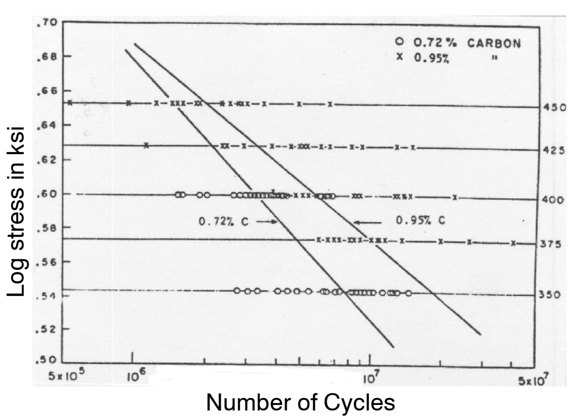


Figure 6 - Log S-log N plot of fatigue data for SAE 8620H steel carburized to 0.72 and 0.95% carbon

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).

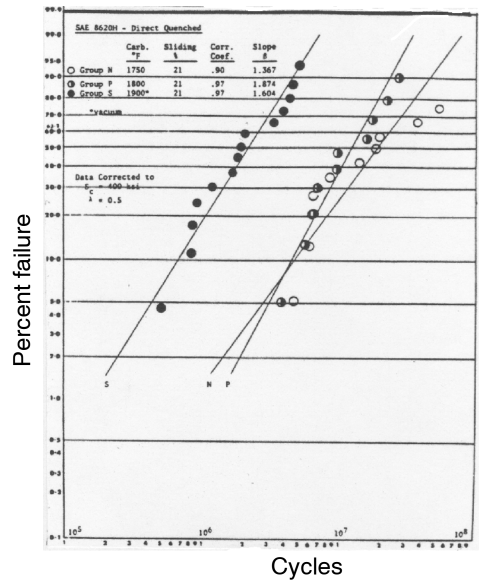


Figure 2 - Weibull probability paper

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).

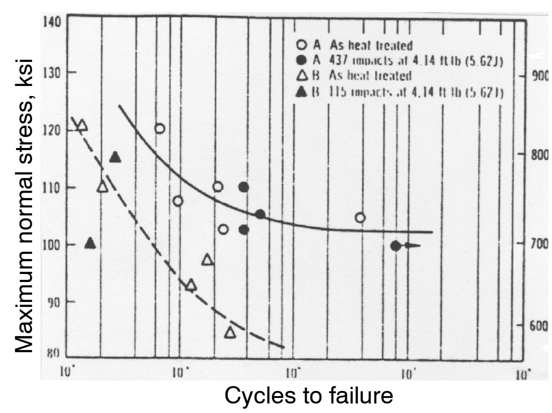


Figure 10.13 - Bending fatigue properties of 4820 steel. Sample A received standard heat treatment, while B was refrigerated at -100^0F

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).

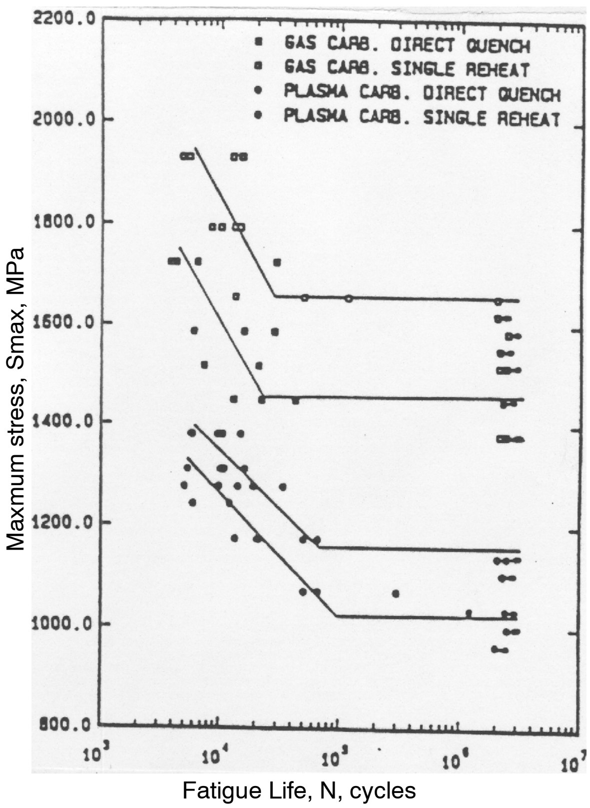


Figure 6 - Bending fatigue curves for SAE 8719 steel. The higher pair of curves corresponds to transgranular crack initiation and the lower pair corresponds to intergranular crack initiation

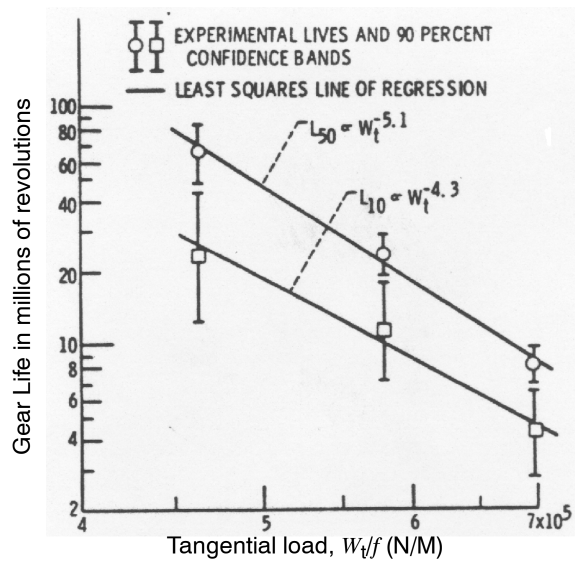


Figure 5 - Load life relationship for (VAR) AISI 9310 steel spur gears speed 10,000 rpm, lubricant naphtenic mineral oil

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).

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).

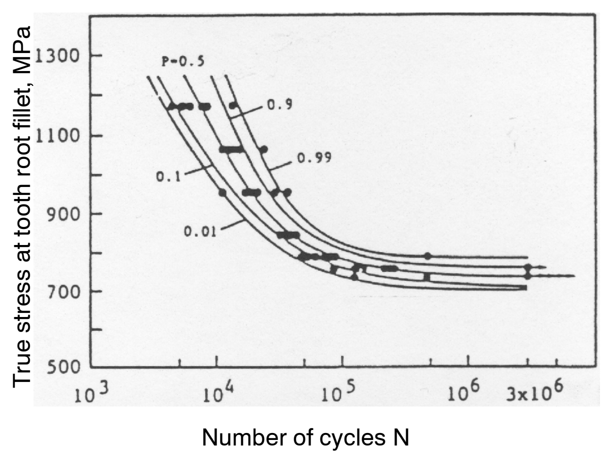


Figure 6 - S-N curve of MAC14

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.

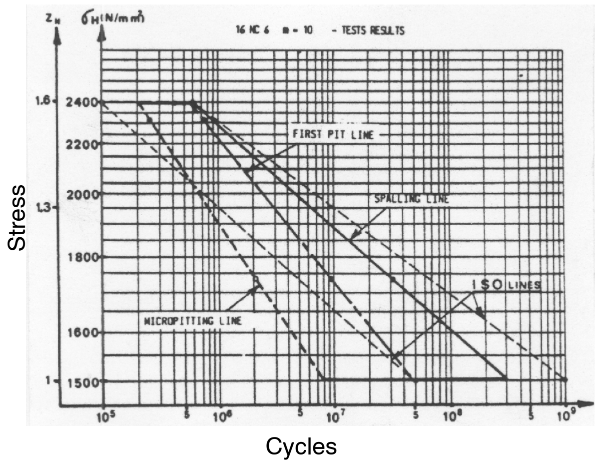


Figure 6 - 16NC6 m=10 Test results

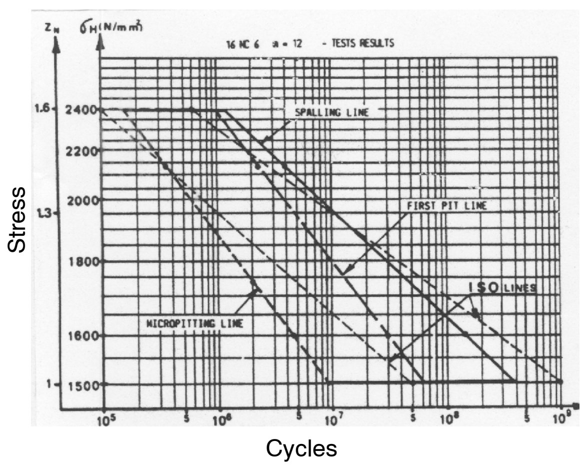


Figure 7 - 16NC6 m=12 Test results

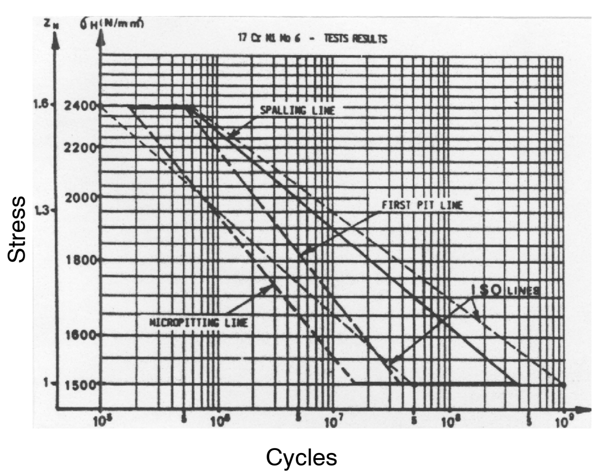


Figure 8 - 17CrNiMo6 Test results

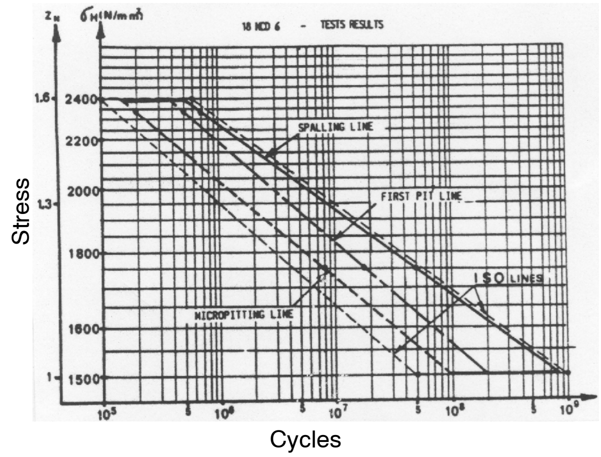


Figure 9 - 18NCD6 Test results

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).

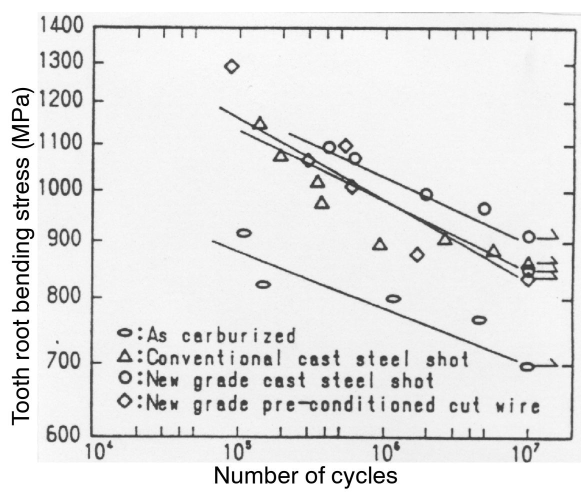


Figure 14 - Comparison of fatigue properties for gears shot peened by conventional and newly developed media

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

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ANSI/AGMA 2001-D04, *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.

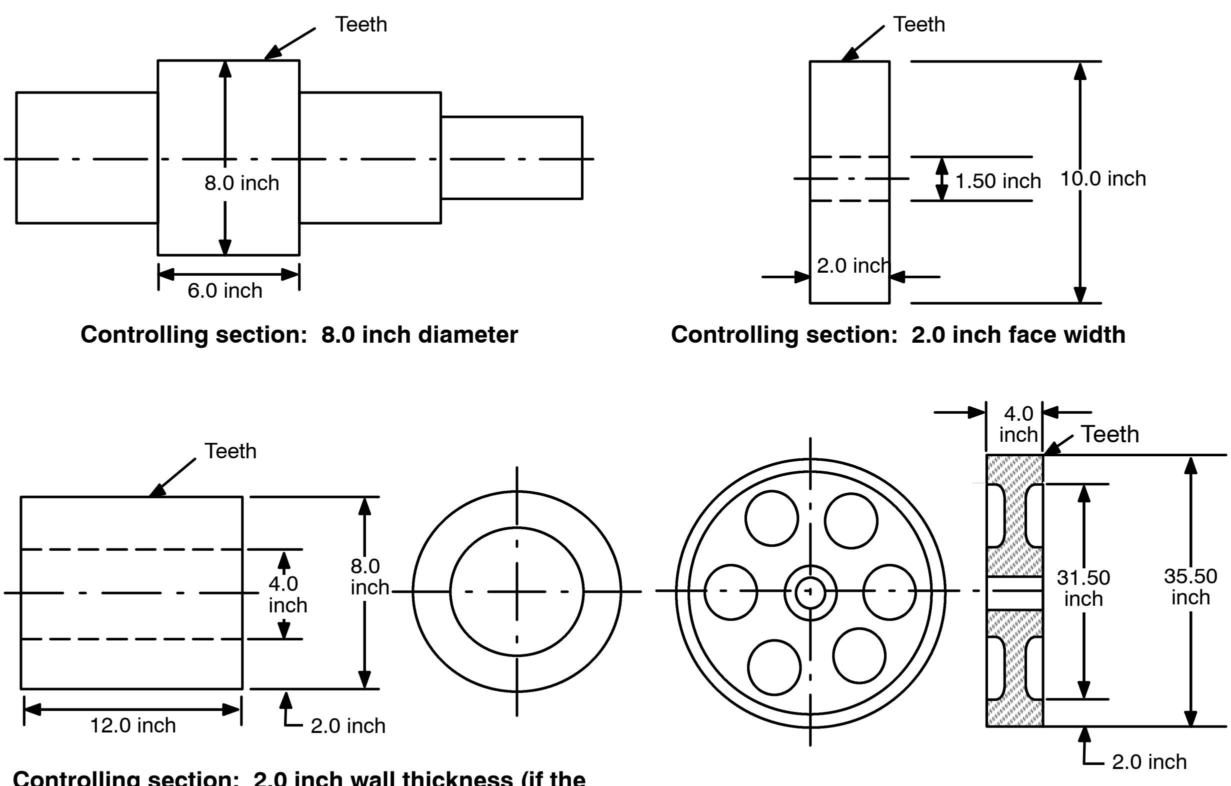
F.2 Definition

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.



Controlling section: 8.0 inch diameter

Controlling section: 2.0 inch face width

Controlling section: 2.0 inch wall thickness (if the bore diameter is less than 20% of the length of the bore, then the outside diameter)

Controlling section: 2.0 inch rim thickness

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 900°F, and obtaining minimum hardness at the roots of teeth.

F.5 General comments

Maximum controlling section sizes versus specified hardness for section sizes to 8.0 inch 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 8.0 inch 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.

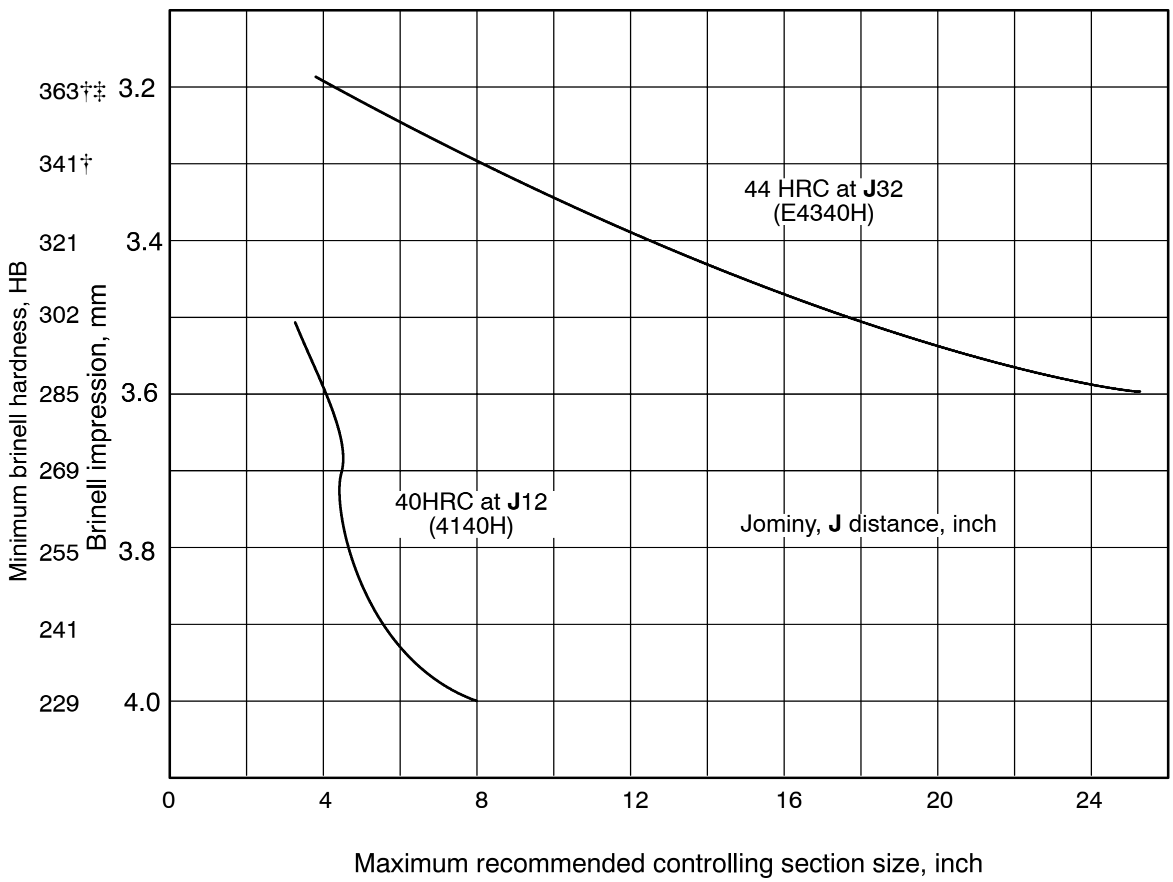


Figure F.2 - Controlling section size for two 0.40% carbon alloy steels*

NOTES:

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

†900°F 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.

Bibliography

The following documents are either referenced in the text of ANSI/AGMA 2001-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*, or indicated for additional information.

1. ISO/TR 10495, *Calculation of Service Life under Variable Load.*

2. American National Standards Institute - ANSI Y12.3-1968, *Letter Symbols for Quantities Used in Mechanics of Solids.*

3. American Gear Manufacturers Association - ANSI/AGMA 1010-E95, *Appearance of Gear Teeth - Terminology of Wear and Failure.*

4. American Gear Manufacturers Association - AGMA 933-B03, *Basic Gear Geometry*

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

6. Kron, H. O., *Gear Teeth Sub-Surface Stress Analysis*, International Symposium on Gearing, Paris, France, June 23, 1977.

7. Winter, H., and Weiss, T., *Some Factors Influencing the Pitting, Micro-Pitting (Frosted Areas) and Slow Speed Wear of Surface Hardened Gears*, ASME Paper No. 80-C2/Det-89.

8. Dudley, Darle. W., *Handbook of Practical Gear Design*, McGraw-Hill, New York, 1984.

9. Dudley, Darle. W., *Characteristics of Regimes of Gear Lubrication*, International Symposium on Gearing and Power Transmissions, Tokyo, 1981.

10. Dudley, Darle. W., *Elastohydrodynamic Behavior Observed in Gear Tooth Action*, Institution of Mechanical Engineers, Leeds, England, September 1965.

11. Bowen, C. W., *The Practical Significance of Designing to Gear Pitting Fatigue Life Criteria*, ASME Paper 77-DET-122, September 1977.

12. Peterson, M. B. and Winer, W. O., *Wear Control Handbook*, ASME, New York, 1980.

13. Ishibashi, A. and Tanaka, S., *Effects of Hunting Gear Ratio Upon Surface Durability of Gear Teeth*, ASME Paper 80-C2/DET-35, August, 1980.

14. Ichimaru, K., Nakajimi, A. and Hirano, F., *Effect of Asperity Interaction on Pitting in Rollers and Gears*, ASME Paper 80-C2/DET-36, August, 1980.

15. ASTM A148-83, *Specifications for Steel Castings for High Strength Stuctural Purposes.*

16. ASTM A291-82, *Specification for Carbon and Alloy Steel Forgings for Pinions and Gears for Reduction Gears.*

17. ASTM A356-83, *Specifications for Steel Castings, Carbon and Low Alloy, Heavy-Walled, for Steam Turbines.*

18. Massey, C., Reeves C. and Shipley, E.E., *The Influence of Lubrication on the Onset of Surface Pitting in Machinable Hardness Gear Teeth*, AGMA Paper 91FTM17.

19. Dolan T.J. and Broghamer E.L., *A Photoelastic Study of the Stresses in Gear Tooth Fillets*, University of Illinois, Engineering Experiment Station, Bulletin No. 335, 1942.

20. Kern, R.F., and Suess, M.E., *Steel Selection a Guide for improving Performance and Profits,* John Wiley and Sons, New York, 1979.

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

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

AGMA 925-A03, *Effect of Lubrication on Gear Surface Distress*

AGMA 927-A01, *Load Distribution Factors - Analytical Methods for Cylindrical Gears*

ANSI/AGMA 6011-I03, *Specification for High Speed Helical Gear Units*

PUBLISHED BY
AMERICAN GEAR MANUFACTURERS ASSOCIATION
500 MONTGOMERY STREET, SUITE 350, ALEXANDRIA, VIRGINIA 22314