

AMERICAN NATIONAL STANDARD

*Gear Power Rating for Cylindrical Shell
and Trunnion Supported Equipment
(Metric Edition)*

ANSI/AGMA 6114-A06



AGMA STANDARD

American National Standard

Gear Power Rating for Cylindrical Shell and Trunnion Supported Equipment (Metric Edition)

ANSI/AGMA 6114-A06

(Metric Edition of ANSI/AGMA 6014-A06)

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ABSTRACT

This standard specifies a method for rating the pitting resistance and bending strength of open or semi-enclosed spur, single helical, double helical, and herringbone gears made from steel and spheroidal graphitic iron for use on cylindrical shell and trunnion supported equipment such as cylindrical grinding mills, kilns, coolers and dryers. Annexes cover installation, alignment, maintenance, lubrication, and a rating method for gears made from ausferritic ductile iron.

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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 AGMA Standard 6114-A06, *Gear Power Rating for Cylindrical Shell and Trunnion Supported Equipment (Metric Edition)*.]

This standard presents formulas and information using ISO symbology and SI units.

AGMA 321.01 was originally developed to cover gears used primarily for ball and rod mills, and for kilns and dryers. It was approved in October 1943, and later modified in June 1946. In June 1951, AGMA 321.03 was approved as a standard. Further changes and additions were approved in June 1959, and AGMA 321.04 was issued in March 1960. AGMA 321.05 was approved in March 1968 and issued in March 1970.

In February 1979, the mill gearing committee was reorganized to review AGMA 321.05 and revise it in accordance with AGMA 218.01, *Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Gear Teeth*. With AGMA 218.01 as a guide, the committee submitted the first draft of ANSI/AGMA 6004-F88 in March 1984.

ANSI/AGMA 6004-F88 superseded AGMA 321.05, *Design Practice for Helical and Herringbone Gears for Cylindrical Grinding Mills, Kilns, Coolers, and Dryers*. It was approved by the AGMA membership in January 1988 and approved as an American National Standard on May 31, 1988.

ANSI/AGMA 6004 was not widely accepted by the industry and many continued to use AGMA 321.05. As such, the AGMA Mill Gearing began work on ANSI/AGMA 6114-A06 in November 2001. Changes to the standard include a new dynamic factor analysis as a function of transmission accuracy number, revised allowable stress numbers, the use of the stress cycle factor in the rating practice, and ratings for gears made from spheroidal graphitic iron. Extensive discussions on new equipment installation and alignment, lubrication, and use of ausferritic ductile iron were added to the annex.

Values for factors assigned in previous standards are not applicable to this Standard, nor are the values assigned in this Standard applicable to previous standards. The ability to design gears, and the knowledge and judgment required to properly evaluate the various rating factors comes primarily from years of accumulated experience in gearing. The detailed treatment of the general rating formulas for specific applications is best accomplished by those experienced in the field.

ANSI/AGMA 6114-A06 supersedes ANSI/AGMA 6004-F88, *Gear Power Rating for Cylindrical Grinding Mills, Kilns, Coolers and Dryers*. The first draft of ANSI/AGMA 6114-A06 was made in November, 2001. It was approved by the AGMA membership in July 2006 and approved as an American National Standard on September 29, 2006.

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.

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American National Standard – Gear Power Rating for Cylindrical Shell and Trunnion Supported Equipment (Metric Edition)

1 Scope

1.1 Applicability

This standard provides a method to determine the power rating of gear sets for cylindrical grinding mills, kilns, coolers, and dryers. The formulas are applicable to steel and spheroidal graphitic iron gears with machined spur, single helical, double helical, or herringbone gear teeth commonly used for this purpose. Calculations determine the allowable rating for pitting resistance and bending strength of external spur and helical involute gear teeth.

1.2 Rating formulas

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

These rating formulas are applicable for rating the pitting resistance and bending strength of external spur and helical involute gear teeth operating on parallel axes with adjustable center distances. 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.

This standard is intended for use by experienced gear designers, capable of selecting reasonable values for the rating factors. It is not intended for use by the engineering public at large.

Values for factors assigned in other standards are not applicable to this standard nor are the values assigned in this standard applicable to other standards. Mixing values from other standards with those from this standard could lead to erroneous ratings.

The gear designer or manufacturer is not responsible for the total system unless such a requirement is clearly identified in the contractual agreement.

It is imperative that the system designer be satisfied that the system of connected rotating parts is compatible, free from critical speeds and from torsional or other vibrations within the specified speed range, no matter how induced.

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.

CAUTION: Compliance with this standard does not constitute a warranty of the rating of the gear set under installed field service conditions.

1.3 Limitations

- 1) Rating procedures are limited to open or semi-enclosed gearing where the gear reaction forces are transmitted through a structure which provides independent bearing support for the gear and pinion. Open gears operate without any enclosure. Semi-enclosed gears operate with a guard that provides some degree of protection against contamination from dust or dirt and retains lubricant.
- 2) Enclosed gear drives or speed reducers are expressly excluded from this standard.
- 3) When multiple pinions are used, the number of contacts per revolution, q , shall be the same as the number of pinions.
- 4) Unless otherwise specified by contractual agreement, the connected motor nameplate power including motor service factor shall be used to determine service factors as defined later within

this standard. When not provided by the purchaser, motor service factor equal to 1.0 shall be used.

- 5) This standard does not include gearing which has been surface hardened by nitriding or flame hardening. This gear rating practice is limited to through hardened steel and spheroidal graphitic iron gears operating with through hardened, carburized, or induction hardened steel pinions.
- 6) Axial contact ratio of helical gear sets, ε_β , shall be equal to or greater than 1.0.
- 7) Information on alignment and drive characteristics is given in annexes A and B.
- 8) Formulas do not apply to external loads such as dropped charges, electrical short circuits and earthquakes.
- 9) Spheroidal graphitic iron data presented in the body of the standard does not apply to austempered spheroidal graphitic iron (ADI). Rating of ADI gearing is not covered by this standard. ADI is discussed further in annex J.
- 10) This gear rating practice is limited to maximum operating speeds of 10.2 meters per second gear pitch line velocity.
- 11) This gear rating practice is limited to gears with module of 8.0 or coarser.

1.4 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-B96.

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

- Damaged gear teeth;
- Spur gears with transverse contact ratio, ε_α , less than 1.0;
- Spur or helical gears with transverse contact ratio, ε_α , greater than 2.0;
- Interference exists between tips of teeth and root fillets;
- Teeth are pointed as defined by this standard, see clause 7;
- Backlash is zero;
- Undercut exists in an area above the theoretical start of active profile;

- The root profiles are stepped or irregular, or deviate from the generated form. The Y_J factor calculation uses the stress concentration factors developed by Dolan and Broghamer [1]. 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;
- The helix angle at the standard (reference) pitch diameter is greater than 20 degrees for single helical and 35 degrees for double helical.

Fractures emanating from stress risers on the tooth profile, tip chipping, and failures of the gear blank through the web or rim should be analyzed by general machine design methods.

2 Normative references

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

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

AGMA 923-A00, *Metallurgical Specifications for Steel Gearing*

ANSI/AGMA 1010-E95, *Appearance of Gear Teeth – Terminology of Wear and Failure*

ANSI/AGMA 1012-G05, *Gear Nomenclature, Definitions of Terms with Symbols*

ANSI/AGMA 2101-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*

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

ANSI/AGMA 2007-C00, *Gears – Surface Temper Etch Inspection After Grinding*

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

ANSI/AGMA 6000-B96, *Specification for Measurement of Lateral Vibration on Gear Units*

ASTM A29-99, *Standard Specification for Steel Bars, Carbon and Alloy, Hot-Wrought and Cold-Finished, General Requirements for*

ASTM A148-03, *Specification for Steel Castings, High Strength, for Structural Purposes*

ASTM A247-67(1998), *Standard Test Method for Evaluating the Microstructure of Graphite in Iron Castings*

ASTM A255-02, *Standard Test Method for Determining Hardenability of Steel*

ASTM A290-02, *Standard Specification for Carbon and Alloy Steel Forgings for Rings for Reduction Gears*

ASTM A291-03, *Standard Specification for Steel Forgings, Carbon and Alloy, for Pinions, Gears and Shafts for Reduction Gears*

ASTM A304-02, *Standard Specification for Carbon and Alloy Steel Bars Subject to End-Quench Hardenability Requirements*

ASTM A370-03a, *Standard Test Methods and Definitions for Mechanical Testing of Steel Products*

ASTM A388-01, *Practice for Ultrasonic Examination of Heavy Steel Forgings*

ASTM A488-04, *Standard Practice for Steel Castings, Welding, Qualifications of Procedures and Personnel*

ASTM A534-01, *Standard Specification for Carburizing Steels for Anti-Friction Bearings*

ASTM A536-84(1999), *Standard Specification for Ductile Iron Castings*

ASTM A578-96, *Specification for Straight-Beam Ultrasonic Examination of Plain and Clad Steel Plates for Special Applications*

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

ASTM A751-01, *Standard Test Methods, Practices, and Terminology for Chemical Analysis of Steel Products*

ASTM A866-01, *Standard Specification for Medium Carbon Anti-Friction Bearing Steel*

ASTM E8-01, *Test Methods for Tension Testing of Metallic Materials*

ASTM E45-97(2002), *Standard Test Methods for Determining the Inclusion Content of Steel*

ASTM E112-96, *Test Methods for Determining Average Grain Size*

ASTM E140-02, *Standard Hardness Conversion Tables for Metals - Relationship Among Brinell Hardness, Vickers Hardness, Rockwell Hardness,*

Superficial Hardness, Knoop Hardness and Scleroscope Hardness

ASTM E351-93(2000), *Standard Test Methods for Chemical Analysis of Cast Iron - All Types*

ASTM E1019-02, *Standard Test Methods for Determination of Carbon, Sulfur, Nitrogen, and Oxygen in Steel and in Iron, Nickel, and Cobalt Alloys*

ASTM E1444-01, *Standard Practice for Magnetic Particle Examination*

AWS D1.1, *Structural Welding Code as Applicable to Cyclically Loaded Non Tubular Connections*

ISO 642:1999, *Steel - Hardenability test by end quenching (Jominy test)*

ISO 643:2003, *Steels - Micrographic determination of the apparent grain size*

ISO 683-1:1987, *Heat treatable steels, alloy steels and free cutting steels - Part 1: Direct hardening and low-alloyed wrought steel in form of different black products*

ISO 683-11-11:1987, *Heat treatable steels, alloy steels and free-cutting steels - Part 11: Wrought case-hardening steels*

ISO 945:1975, *Cast iron - Designation of microstructure of graphite*

ISO 1083:1987, *Spheroidal graphite cast iron - Classification*

ISO 4967:1998, *Steel - Determination of content of nonmetallic inclusions - Micrographic method using standard diagrams*

ISO 6336-5:2003, *Calculation of load capacity of spur and helical gears - Part 5: Strength and quality of materials*

ISO/FDIS 6336-6, *Calculation of load capacity of spur and helical gears - Part 6: Calculation of service life under variable load*

SAE/AMS 2301, *Steel Cleanliness, Aircraft Quality Magnetic Particle Inspection Procedure*

SAE/AMS-S-13165, *Shot Peening of Metal Parts*

SAE J422, *Microscopic Determination of Inclusions in Steels*

3 Definitions and symbols

3.1 Definitions

The symbols, terms and definitions, when applicable, conform to ANSI/AGMA 1012-G05, *Gear No-*

menclature, Definitions of Terms with Symbols, and AGMA 908-B89, Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth.

Throughout the standard the terms gearing or gear teeth can refer to either pinion or gear. The user is cautioned to take the context of the reference under consideration to determine if this refers to a specific

element or the mesh.

3.2 Symbols

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

NOTE: The symbols and terminology used in this standard may differ from other AGMA standards. The user should not assume that familiar symbols can be used without careful study of table 1.

Table 1 - Symbols and definitions

Symbol	Description	Units	Where first used
A_v	Transmission accuracy number per ANSI/AGMA 2015-1-A01	--	Eq 6
a	Operating center distance	mm	Eq 2
B_{mill}	Exponential accuracy adjustment to K_{vm} for open gearing	--	Eq 9
b	Net face width of narrowest member	mm	Eq 1
C_{mill}	Linear adjustment to K_{vm} for open gearing	--	Eq 9
C_{SF}	Service factor for pitting resistance	--	Eq 14
d_e	Outside diameter of pinion/gear	mm	Eq 8
d_T	Tolerance diameter	mm	Eq 8
d_{w1}	Operating pitch diameter of pinion	mm	Eq 1
E_1	Modulus of elasticity for pinion	N/mm ²	Eq 13
E_2	Modulus of elasticity for gear	N/mm ²	Eq 13
F_{max}	Maximum peak tangential load	N	Eq 25
F_t	Transmitted tangential load	N	12.2
f_{pt}	Single pitch deviation	μm	Eq 6
H_{B1}	Brinell hardness of pinion	HB	Eq 16
H_{B2}	Brinell hardness of gear	HB	Eq 16
H_{R1}	Rockwell C hardness of pinion	HRC	Eq 17
$h_{e \text{ max}}$	Maximum effective case depth for external carburized and induction hardened gear teeth	mm	Eq 23
$h_{e \text{ min}}$	Minimum effective case depth for external carburized and induction hardened gear teeth	mm	Figure 11
h_t	Gear tooth whole depth	mm	Eq 5
K_{Bm}	Rim thickness factor	--	Figure 1
K_f	Stress correction factor	--	Eq 25
K_H	Load distribution factor	--	Eq 1
K_{He}	Mesh alignment correction factor	--	Eq 19
K_{Hma}	Mesh alignment factor	--	Eq 19
K_{Hmc}	Lead correction factor	--	Eq 19
K_{Hpf}	Pinion proportion factor	--	Eq 19
K_{Hpm}	Pinion proportion modifier	--	Eq 19
$K_{H\beta}$	Face load distribution factor	--	Eq 18
K_{SF}	Service factor for bending strength	--	Eq 14
K_{vm}	Dynamic factor	--	Eq 1
L	Life	hours	Eq 29
m_B	Back-up ratio	--	Eq 5

(continued)

Table 1 (concluded)

Symbol	Description	Units	Where first used
m_n	Normal module	mm	Eq 4
m_T	Transverse module	mm	Eq 3
n_L	Number of stress cycles	--	Eq 29
P	Maximum peak power	kW	Eq 26
P_a	Allowable transmitted power for gear set	kw	Eq 14
P_{aym}	Allowable transmitted power for bending strength at unity service factor	kw	Eq 3
P_{azm}	Allowable transmitted power for pitting resistance at unity service factor	kw	Eq 1
p_x	Axial pitch	mm	Eq 4
Q_v	Transmission accuracy level number	--	8.2
q	Number of contacts per revolution	--	1.3
s_{an}	Normal tooth thickness at the top land of gear	mm	Eq 23
T	Maximum transmitted pinion torque	Nm	Eq 26
t_R	Gear rim thickness	mm	Eq 5
u	Gear ratio (never less than 1.0)	--	Eq 2
v_t	Pitchline velocity at operating pitch diameter	m/s	Eq 9
ω	Speed	rpm	Eq 29
ω_1	Pinion speed	rpm	Eq 1
Y_J	Geometry factor for bending strength	--	Eq 3
Y_N	Stress cycle factor for bending strength	--	Eq 3
Z_E	Elastic coefficient	$[N/mm^2]^{0.5}$	Eq 1
Z_I	Geometry factor for pitting resistance	--	Eq 1
Z_N	Stress cycle factor for pitting resistance	--	Eq 1
Z_w	Hardness ratio factor for pitting resistance	--	Eq 15
β	Helix angle at standard pitch diameter	degree	Eq 4
ε_β	Axial contact ratio	--	1.3
ν_2	Poisson's ratio for gear	--	Eq 13
ν_1	Poisson's ratio for pinion	--	Eq 13
σ_{HP}	Allowable contact stress number	N/mm^2	Eq 1
σ_{FP}	Allowable bending stress number	N/mm^2	Eq 3
σ_s	Allowable yield stress number	N/mm^2	Eq 25

4 Application

4.1 Manufacturing quality

Rating factors shall be selected on the basis of expected process variations of component parts as manufactured. The formulas of this standard are only valid for appropriate material quality and geometric quality that conforms to manufacturing tolerances.

4.1.1 Geometric quality

The rating formulas of this standard are only valid if gear tooth accuracy and gear element support accuracy assumed in the calculations are actually achieved in manufacture.

Flange mounted gears require that the flange of the gear and the mounting flange of the equipment be parallel with one another. At least one gear rim end face should be parallel with the gear mounting flange surfaces, or pin boss faces for spring mounted

gears. That gear rim end face should be clearly identified for alignment.

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

Gear element support considerations include: assembled gear mesh alignment, mounted gear flange and rim face axial runouts, shaft pinion axial and radial runouts, and either gear flange mount or spring mount pin circle radial runouts, see annex A.

4.1.2 Metallurgy

Allowable stress numbers, σ_{HP} and σ_{FP} , included herein are a function of melting, casting, forging and heat treating practice. Allowable stress numbers in this standard are based on 10^7 cycles, 99 percent reliability, and unidirectional loading.

Allowable stress numbers are only valid for materials and conditions listed in this standard.

4.1.3 Residual stress

Residual stress in gear segments is an important consideration. Presence of residual stress may be checked by inspecting the gear joint assembly. Typically this is done before finish gear cutting while there is stock present to re-machine if the gear has distorted, or as a minimum, after finished machining. For acceptance, the joint should reassemble without excessive force or misalignment. If the gear joint cannot be satisfactorily reassembled, the gear segments should be disassembled and re-machined.

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 bending strength performance of the gear teeth. Shot peening, case carburizing, and induction hardening are common methods of inducing compressive pre-stress in the surface of gear teeth.

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

4.2 Lubrication

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 pitchline velocity, see annex D.

4.3 Temperature extremes

4.3.1 Cold temperature operation

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

- low temperature Charpy specification for either steel or spheroidal graphitic iron;
- nil ductility temperature specification;
- reduce carbon content of steel to less than 0.4 percent;
- use of nickel alloy or vanadium modified steels;
- using heating elements to increase lubricant and gear temperatures.

4.3.2 Temperature gradient

The gear design should consider the effects of operating temperature difference between the mounting flange and the gear rim.

4.4 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.

4.4.1 Service damaged teeth

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

4.4.2 Misalignment and deflection of foundations

Many gear systems depend on external supports such as equipment 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.

4.4.3 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.

4.4.4 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 driving and driven equipment. Overloads are a part of the service factor which is intended to account for the operating characteristics of the driving and driven equipment.

However, it must be recognized that if the operating roughness of the driving 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. It is recommended that a vibration analysis be performed. This analysis must include the total system of driver, driven equipment, couplings, mounting conditions, and sources of excitation. Natural frequencies, mode shapes, and dynamic response amplitudes should be calculated. Responsibility for the vibration analysis of the system rests with the purchaser of the gearing, unless otherwise specified by contractual agreements.

4.4.5 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. Efforts should be made to minimize corrosion and its effects.

5 Criteria for tooth capacity

5.1 Relationship of pitting resistance and bending strength ratings

There are major differences between the pitting resistance and 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.

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

5.2 Pitting resistance

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.

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. 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.

In most industrial applications non-progressive initial pitting is not deemed serious on through hardened elements. Initial pitting is characterized by small pits which do not extend over the entire face width or profile height of the effected 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.

5.3 Surface conditions not covered by pitting resistance formula

Conditions such as micropitting, electric discharge pitting, wear and scuffing are not covered by this standard. See ANSI/AGMA 1010-E95 for more information.

5.3.1 Micropitting

Micropitting is one type of gear tooth surface fatigue. It is characterized by very small pits on the surface of the material, usually less than 20 micrometers 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 influences from gear loading, material

and its heat treatment, type of lubricant, and degree of lubrication.

5.3.2 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 a similar 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.

5.3.3 Wear capacity of gears

Wear resistance of mating gears can be a dictating performance limitation, particularly in these low speed, heavily loaded gears. Gear wear is a difficult phenomenon to predict analytically, however consideration should be given to the following:

- operation in a contaminated environment;
- sealing and design of the gear guard;
- lubrication, see annex D.

5.3.4 Scuffing

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

Scuffing is not a fatigue phenomenon and it may occur instantaneously. This phenomenon is a function of lubricant viscosity and additives, operating bulk temperature of gear blanks, sliding velocity, surface roughness of teeth, gear materials, heat treatments, and surface pressure.

5.4 Bending strength

Bending strength of gear teeth is a fatigue phenomenon related to the resistance to cracking at the tooth root fillet in external gears. Typical cracks and fractures are illustrated in ANSI/AGMA 1010-E95.

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

- 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;
- 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 gearing without causing root fillet cracking.

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. In such cases, the rim thickness factor, K_B , adjusts the calculated bending stress number.

5.5 Conditions not covered by the bending strength formula

The gear designer 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 annex C.

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. However, these considerations are outside the scope of this standard.

5.6 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 6336-6.

6 Rating formulas

6.1 Pitting resistance

6.1.1 Pitting resistance power rating

The allowable transmitted pitting resistance power rating is:

$$P_{azm} = \frac{\pi \omega_1 b}{6 \times 10^7} \frac{Z_l}{K_{vm} K_H} \left(\frac{d_{w1} \sigma_{HP} Z_N Z_W}{Z_E} \right)^2 \quad (1)$$

where

P_{azm} is allowable transmitted power for pitting resistance per clause 1.0, at unity service factor ($C_{SF}=1.0$), kw;

ω_1 is pinion speed, rpm;

b is net face width of narrowest member, mm;

Z_l is geometry factor for pitting resistance, see clause 7;

K_{vm} is dynamic factor, see clause 8;

K_H is load distribution factor, see clause 12;

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

$$d_{w1} = \frac{2a}{u+1} \quad (2)$$

a is operating center distance, mm;

u is gear ratio (never less than 1.0);

σ_{HP} is allowable contact stress number, N/mm², see clause 13;

Z_N is stress cycle factor for pitting resistance, see clause 15;

Z_W is hardness ratio factor for pitting resistance, see clause 11;

Z_E is elastic coefficient, (N/mm²)^{0.5}, see clause 9.

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

6.2 Bending strength

6.2.1 Bending strength power rating

The allowable transmitted bending strength power rating is:

$$P_{aym} = \frac{\pi \omega_1 d_{w1}}{6 \times 10^7} \frac{b m_t Y_J \sigma_{FP} Y_N}{K_H K_{Bm}} \quad (3)$$

where

P_{aym} is allowable transmitted power for bending strength per clause 1.0, at unity service factor ($K_{SF} = 1.0$), kw;

m_t is m_n for spur gears;

$$m_t = \frac{p_x \tan \beta}{\pi} = \frac{m_n}{\cos \beta} \quad \text{for helical gears} \quad (4)$$

p_x is axial pitch, mm;

β is helix angle at the standard pitch diameter, degrees. See ANSI/AGMA 1012-G05 for additional information.

m_n is normal metric module, mm;

Y_J is geometry factor for bending strength, see clause 7;

σ_{FP} is allowable bending stress number, N/mm², see clause 13;

Y_N is stress cycle factor for bending strength, see clause 15;

K_{Bm} is rim thickness factor, see 6.2.2.

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

value of the term $\frac{\sigma_{FP} Y_N Y_J}{K_{Bm}}$ for each of the mating gears.

6.2.2 Rim thickness factor, K_{Bm}

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 [2] suggest the use of a stress modifying factor in this case.

The rim thickness factor, K_{Bm} , 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_{Bm} , adjusts the calculated bending stress number for thin rimmed gears. It is a function of the backup ratio, m_B , see figure 1.

$$m_B = \frac{t_R}{h_t} \quad (5)$$

where

m_B is back-up ratio;

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

h_t is gear tooth whole depth, mm.

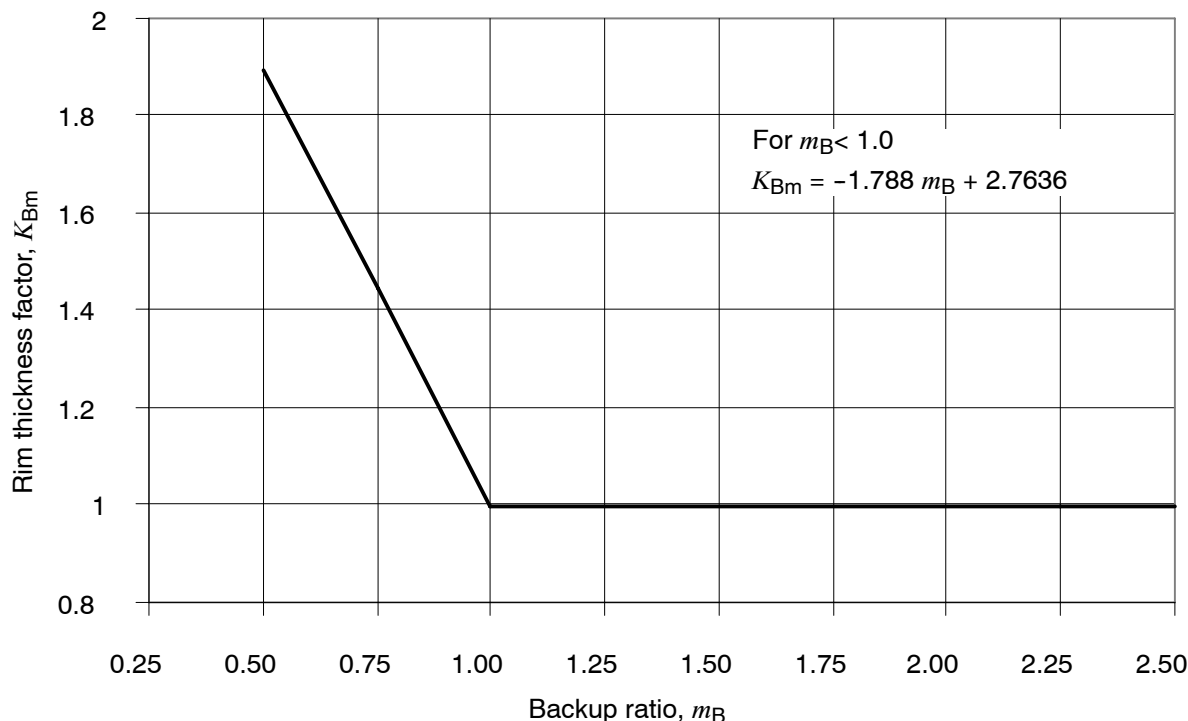


Figure 1 – Rim thickness factor, K_{Bm}

The effects of webs and stiffeners can be an improvement but are not accounted for in figure 1. 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 or detailed analysis justifies, different values of K_{Bm} may be used. The minimum value of K_{Bm} is 1.0.

7 Geometry factors, Z_I and Y_J

7.1 Pitting resistance geometry factor, Z_I

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

7.2 Bending strength geometry factor, Y_J

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

7.3 Calculation method

It is recommended that geometry factors, Z_I and Y_J , be determined by AGMA 908-B89. It includes tables for some common tooth forms and the analytical method for involute gears with generated root fillets.

For most designs covered by this standard, pointed teeth are defined as normal chordal top land thickness, s_{an} , less than $0.25 m_n$. Top lands smaller than this value require additional review.

8 Dynamic factor, K_{vm}

CAUTION: Dynamic factor 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_{vm} , 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.

8.2 Approximate dynamic factor, K_{vm}

Figure 2 shows dynamic factors that shall be used. The curves of figure 2 and the equations given are based on empirical data, and do not account for resonance.

CAUTION: Dynamic factor has been redefined to use A_v as defined in ANSI/AGMA 2015-1-A01. In previous AGMA standards K_{vm} was based on Q_v as defined in AGMA 2000-A88. See annex J for a method to calculate K_{vm} using Q_v values.

Choices of curves $A_v = 7$ through $A_v = 11$ 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. A_v is related to the transmission accuracy grade number.

8.2.1 Curves labeled $A_v = 7$ through $A_v = 11$

The empirical curves of figure 2 are generated by the following equations for integer values of A_v , such that $7 \leq A_v \leq 11$.

The dynamic factor can be expressed as a function of A_v . A_v can be approximated using the pitch variation of the pinion and gear with the following formula, 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 K_v .

For $d_T \leq 400$ mm

$$A_v = \frac{\ln(|f_{pt}|) - \ln(0.3 m_n + 0.003 d_T + 5.2)}{0.3466} + 5 \quad (6)$$

(rounded to the next highest integer)

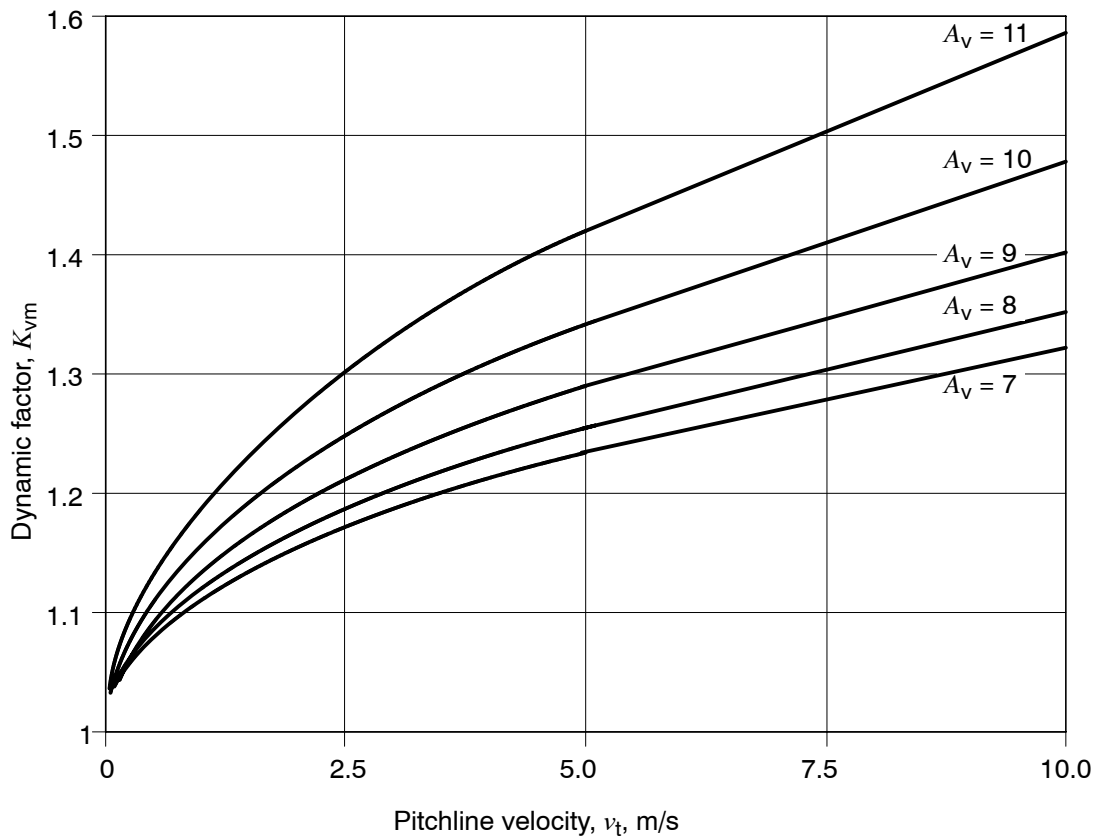


Figure 2 - Dynamic factor, K_{vm}

For 400 mm < d_T

$$A_v = \frac{\ln(|f_{pt}|) - \ln(0.3m_n + 0.12d_T^{0.5} + 4)}{0.3466} + 5 \quad (7)$$

(rounded to the next highest integer)

where

A_v is transmission accuracy number per ANSI/AGMA 2015-1-A01;

$\ln$ is natural log, $\log_e$;

f_{pt} is single pitch deviation, μm ;

m_n is normal module, where
 $1.25 \leq m_n \leq 50$;

d_T is tolerance diameter, mm;

$$d_T = d_e - 2m_n \quad (8)$$

d_e is outside diameter of pinion or gear, mm.

A_v can also be estimated as the appropriate accuracy grade for the expected pitch and profile deviations in accordance with ANSI/AGMA 2015-1-A01.

$$K_{vm} = \left[\frac{C_{mill} + \sqrt{196.85 v_t}}{C_{mill}} \right]^{B_{mill}} \quad (9)$$

where

$$C_{mill} = 50 + 56(1 - B_{mill}) \quad (10)$$

$$B_{mill} = \frac{\left(12 - \left(17 - A_v - \frac{(11 - A_v)^2}{10} \right) \right)^{0.667}}{4} \quad (11)$$

$$v_t = \frac{\pi \omega_1 d_{w1}}{60\,000} \quad (12)$$

where

C_{mill} is linear adjustment to K_{vm} for open gearing;

B_{mill} is exponential accuracy adjustment to K_{vm} for open gearing;

v_t is pitchline velocity at operating pitch diameter, m/s.

Values greater than $A_v = 11$ are not allowed. When $A_v < 7$, use $A_v = 7$.

The minimum value of K_{vm} when using this standard is 1.02.

9 Elastic coefficient, Z_E

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

$$Z_E = \sqrt{\frac{1}{\pi \left[\left(\frac{1 - \nu_1^2}{E_1} \right) + \left(\frac{1 - \nu_2^2}{E_2} \right) \right]}} \quad (13)$$

where

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

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

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

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

When using an spheroidal graphitic (SG) iron gear meshing with a steel pinion, Z_E equals 184 $[\text{N/mm}^2]^{0.5}$ for a steel pinion with $\nu = 0.3$ and $E = 2.05 \times 10^5 \text{ N/mm}^2$ and for the SG iron gear, $\nu = 0.27$ and $E = 1.850 \times 10^5 \text{ N/mm}^2$.

10 Service factor

The AGMA service factor as traditionally used in these gear applications depends on experience acquired in each specific application.

Service factor includes the combined effects of overload, reliability, and other application related factors. See annex I for additional information.

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

$$P_a = \text{the lesser of } \frac{P_{azm}}{C_{SF}} \text{ and } \frac{P_{aym}}{K_{SF}} \quad (14)$$

where

P_{azm} is allowable transmitted power for pitting resistance at unity service factor, kw;

C_{SF} is service factor for pitting resistance;

P_{aym} is allowable transmitted power for bending strength at unity service factor, kw;

K_{SF} is service factor for bending strength.

11 Hardness ratio factor, Z_W

The hardness ratio factor, Z_W , depends upon:

- gear ratio;
- hardness of pinion and gear;
- surface finish of surface hardened pinions.

When the pinion is substantially harder than the gear, the work hardening effect increases the gear capacity. Typical values of Z_W are shown in figure 3. These values are applied to the gear only. For the pinion, $Z_W = 1.0$.

Values of the hardness ratio factor, Z_W , can be calculated as follows:

$$Z_W = 1.0 + A(u - 1.0) \quad (15)$$

where

Z_W is hardness ratio factor for pitting resistance;

u is gear ratio (never less than 1.0).

A is determined from the hardness ratio as:

For through hardened pinions meshing with through hardened gears for the range:

$$1.2 \leq \frac{H_{B1}}{H_{B2}} \leq 1.7$$

and for surface hardened pinions with a surface finish less than 3.2 Ra meshing with through hardened gears for the range:

$$1.2 \leq \frac{H_{B1}}{H_{B2}} \leq 2.0$$

$$A = 0.00898 \left[\frac{H_{B1}}{H_{B2}} \right] - 0.00829 \quad (16)$$

where

H_{B1} is Brinell hardness of pinion, HB;

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

For $\frac{H_{B1}}{H_{B2}} < 1.2$, $A = 0.00$

For through hardened pinions meshing with through hardened gears, and

$\frac{H_{B1}}{H_{B2}} > 1.7$, $A = 0.00698$

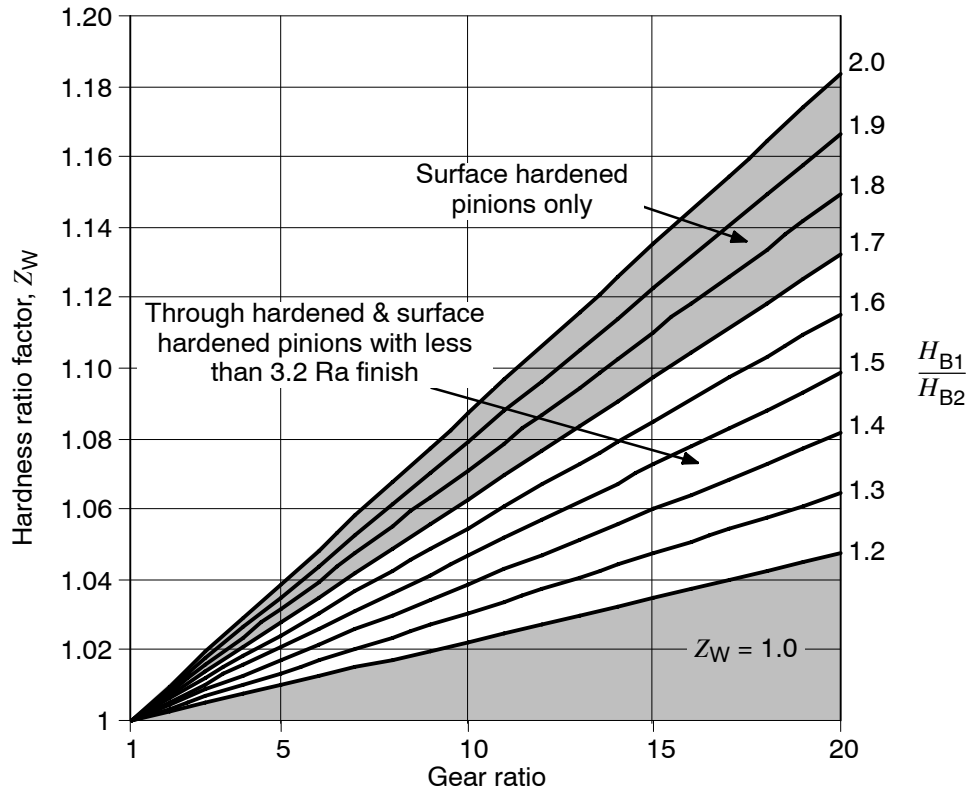


Figure 3 - Typical Hardness ratio factors, Z_W

For surface hardened pinions with a surface finish less than 3.2 Ra, meshing with through hardened gears, and

$$\frac{H_{B1}}{H_{B2}} > 2.0, A = 0.00967$$

For surface hardened pinions with a surface finish greater than or equal to 3.2 Ra, meshing with through hardened gears

$$Z_W = 1.0$$

H_{B1} may be determined for surface hardened pinions measured with Rockwell C scale hardness values between 45 Rc and 62 Rc by equation 17

$$H_{B1} = 6.96608 (10^{-2}) + \left[\frac{1.0}{-9.68063(10^{-5}) - 1.62552(10^{-9})(H_{R1}^3) + \frac{0.117524}{H_{R1}}} \right] \quad (17)$$

where

H_{R1} is Rockwell C hardness of the pinion, HRC.

12 Load distribution factor, K_H

The load distribution factor modifies the rating equations to reflect the non-uniform distribution of load along the lines of contact. The amount of non-uniformity of the load distribution is caused by many influences. See AGMA 927-A01 and ANSI/AGMA 2101-D04 for additional information on this topic.

12.1 Values for load distribution factor, K_H

Load distribution factor is defined as: peak load intensity divided by the average, or uniformly distributed, load intensity; i.e., ratio of peak to mean loading.

$$K_H = K_{H\beta} \quad (18)$$

where

K_H is load distribution factor;

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

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

The face load distribution factor accounts for the non-uniform distribution of load across the gearing face width. The magnitude of the face load

distribution factor is defined as peak load intensity divided by average load intensity across the face width.

This factor can be determined empirically or analytically. This standard provides an empirical method only.

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

- net face width to pinion pitch diameter ratio, $b/d_{w1} \leq 2.0$ (for double helical gears the gap is not included in the face width);
- face width up to 1270 mm;
- contact across full face width of narrowest member when loaded.

CAUTION: If $b/d_{w1} > 2.4 - 0.029 (F_t(u+1)/(d_{w1} b u))$ the value of K_H 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.

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 gear contact. Bearing clearances affect gear contact in the same way as offset straddle mounted pinions. However, gear elements with their overhang to the same support side can compound the effect. This effect is addressed by the pinion proportion modifying factor, K_{Hpm} . When 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.

The following method will be used:

$$K_{H\beta} = 1.0 + K_{Hmc} (K_{Hpf} K_{Hpm} + K_{Hma} K_{He}) \quad (19)$$

where

K_{Hmc} is lead correction factor;

K_{Hpf} is pinion proportion factor;

K_{Hpm} is pinion proportion modifier;

K_{Hma} is mesh alignment factor;

K_{He} is mesh alignment correction factor.

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

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

K_{Hmc} is 0.95 for gearing with leads properly modified by crowning or lead correction.

Pinion proportion factor, K_{Hpf} , accounts for deflections due to load. These deflections are normally higher for wide face widths or higher b/d_{w1} ratios. The pinion proportion factor can be obtained from figure 4.

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

Values for K_{Hpf} as shown in figure 4 can be determined by the following equations:

when $25 < b \leq 432$

$$K_{Hpf} = \frac{b}{10 d_{w1}} - 0.0375 + 0.0004926 b \quad (20)$$

when $432 < b \leq 1250$

$$K_{Hpf} = \frac{b}{10 d_{w1}} - 0.1109 + 0.000815 b - 0.000000353 b^2 \quad (21)$$

NOTE: For values of $\frac{b}{10 d_{w1}}$ less than 0.05, use 0.05 for this value in equations 20 or 21.

Pinion proportion modifier, K_{Hpm} , is 1.0 due to alignment correction at assembly.

Mesh alignment factor, K_{Hma} , accounts for 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 mesh alignment factor can be obtained from figure 5. When the driven gear is mounted to equipment that is supported by rollers and tires, use the roller supported curve. When the driven gear is mounted to equipment that is supported by bearings, use the bearing supported curve.

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

Values for the two curves of figure 5 are defined as follows:

$$K_{Hma} = A + B(b) + C(b)^2 \quad (22)$$

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

Mesh alignment correction factor is used to modify the mesh alignment factor when assembly techniques improve the effective mesh alignment. The following value is suggested for the mesh alignment correction factor:

K_{He} is 0.80 because the gearing is adjusted at assembly.

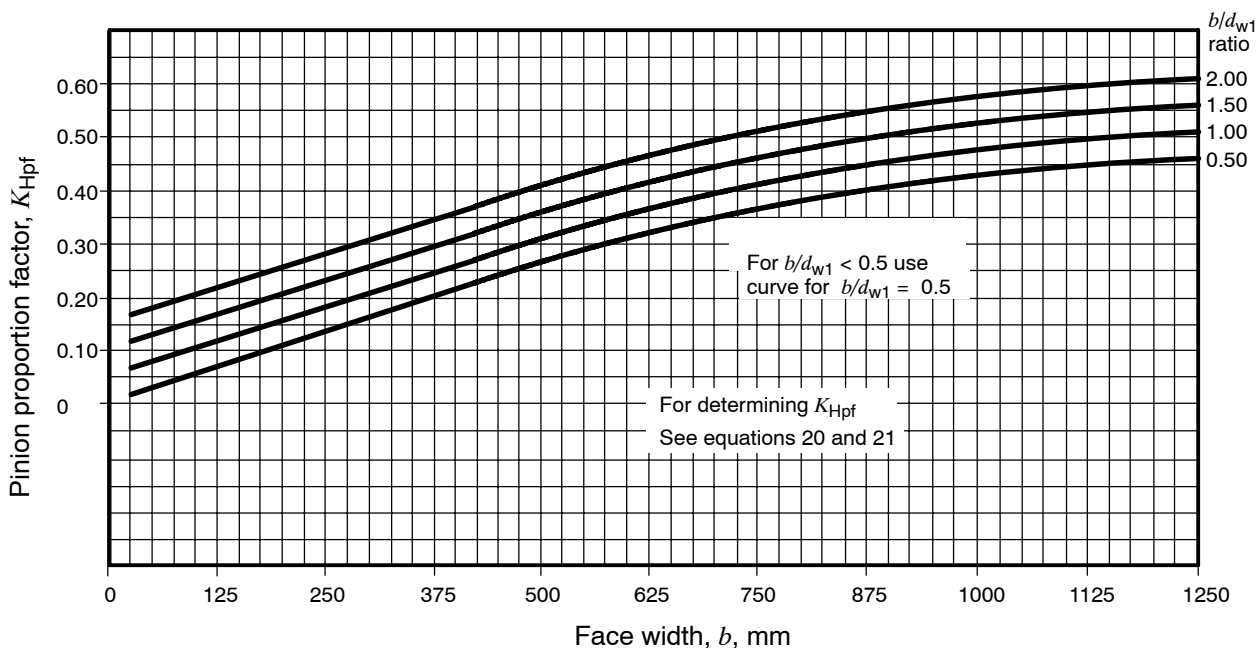
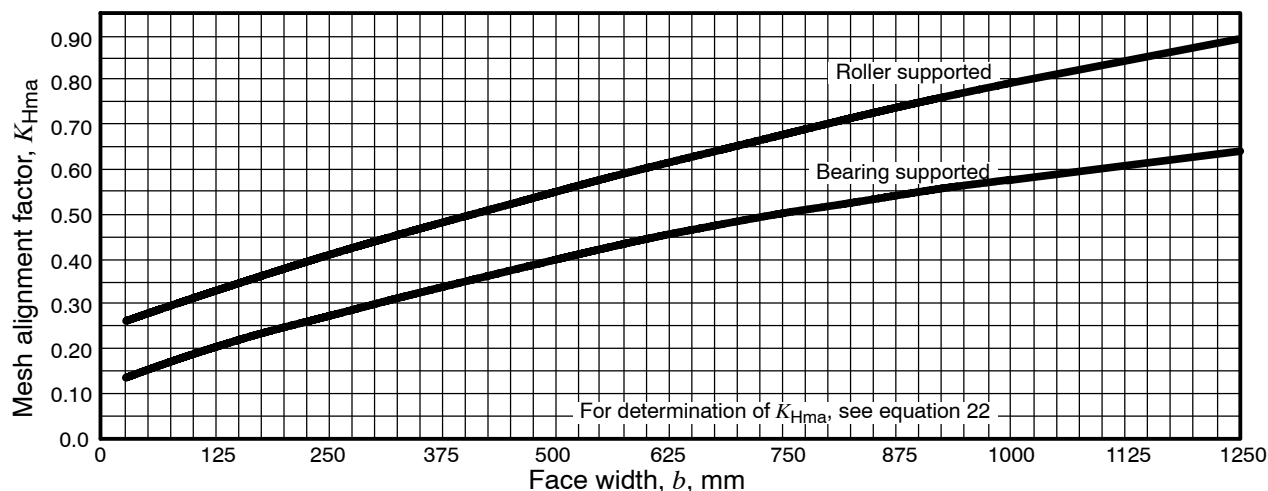


Figure 4 - Pinion proportion factor, K_{Hpf}

Figure 5 - Mesh alignment factor, K_{Hma} Table 2 - Empirical constants: A , B , and C

Curve	A	B	C
Roller supported	2.47×10^{-1}	0.657×10^{-3}	-1.186×10^{-7}
Bearing supported	1.27×10^{-1}	0.622×10^{-3}	-1.69×10^{-7}

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

Allowable stress numbers for gear materials vary with items such as material composition, cleanliness, residual stress, microstructure, quality, heat treatment, and processing practices.

Allowable stress numbers in this standard (tables 3 and 4) are determined or estimated from laboratory

tests and accumulated field experiences. They are based on 10 million stress cycles, unidirectional loading and 99 percent reliability. Allowable stress numbers are designated as σ_{HP} and σ_{FP} , for pitting resistance and bending strength respectively. For service life other than 10 million cycles, allowable stress numbers are adjusted by the use of stress cycle factors, see clause 15.

Table 3 - Allowable contact stress number, σ_{HP} , for steel and spheroidal graphitic iron gears

Material designation	Heat treatment	Minimum surface hardness	Allowable contact stress number ²⁾ , σ_{HP} , N/mm ²	
			Grade M1	Grade M2
Steel ³⁾	Through hardened ^{1), 4)}	see figure 6	See figure 6	See figure 6
	Induction hardened ⁵⁾	see table 8	1170	1345
	Carburized & hardened ⁵⁾	55 HRC	1240	1450
Spheroidal graphitic iron	Spheroidal graphitic iron through hardened ¹⁾	See figure 8		

NOTES:

- 1) Hardness to be equivalent to that at the start of active profile in the center of the face width.
- 2) See tables 5, 7, and 8 for major metallurgical factors for each stress grade of steel gears and table 6 for spheroidal graphitic iron.
- 3) 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) Allowable stress numbers indicated may be used with the minimum case depths prescribed in 13.1.

Table 4 - Allowable bending stress number, σ_{FP} , for steel and spheroidal graphitic iron gears

Material designation	Heat treatment	Minimum surface hardness	Allowable bending stress number ²⁾ , σ_{FP} , N/mm ²	
			Grade M1	Grade M2
Steel ³⁾	Through hardened ¹⁾	See figure 7	See figure 7	See figure 7
	Induction hardened ⁴⁾ with type A pattern ⁵⁾	See table 8	310	380
	Carburized & hardened ⁴⁾	55 HRC	380	425
Spheroidal graphitic iron	Spheroidal graphitic iron through hardened ¹⁾	See figure 9		
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 5, 7, and 8 for major metallurgical factors for each stress grade of steel gears and table 6 for spheroidal graphitic iron. 3) Steel selected must be compatible with the heat treatment process selected and hardness required. 4) Allowable stress numbers indicated may be used with the minimum case depths prescribed in 13.1. 5) See figure 10 for type A flank and root hardness patterns.				

The effective case depth for induction hardened pinions 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.

Allowable stress numbers for 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 when specified. 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 shall not be used 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. Allowable stress numbers are shown in tables 3 and 4, and figures 6 through 9.

13.1 Guide for case depth of surface hardened pinions

Surface hardened pinion 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 tooth tips and high residual tensile stress in the core.

The effective case depth for carburized and hardened pinions is defined as the depth below the

surface at which the Rockwell 'C' hardness, HRC, has dropped to 50 HRC or equivalent.

A minimum effective case depth, $h_{e \min}$, at the pitch line for carburized and induction hardened pinion teeth as a function of pitch is shown in figure 11.

Care should be exercised when choosing case depth, such that adequate case depths prevail at the tooth and root fillet, and that tooth tips are not over hardened and brittle. The actual case depth tolerance is determined by the manufacturer or by contractual agreement. It is recommended that the case depth does not exceed $h_{e \max}$.

$$h_{e \max} = \text{the lesser of } 0.4 m_n \text{ or } 0.56 s_{an} \quad (23)$$

where

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

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

If $h_{e \min}$ from figure 11 (with case depth 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 induction hardened teeth using a single tooth hardening method, it is critical that the hardened areas from each tooth flank do not cross at the tip of the tooth. This prevents tempering back of a previously hardened tooth surface.

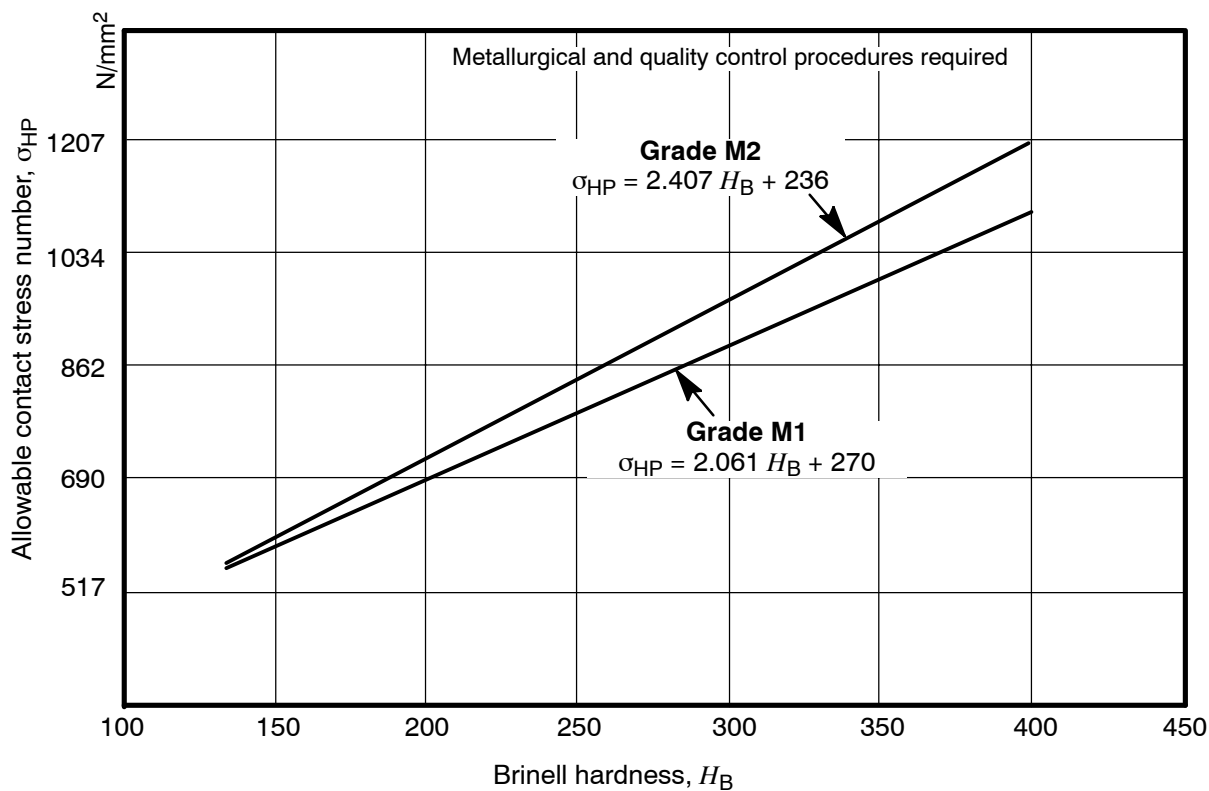


Figure 6 - Allowable contact stress number for through hardened steel gears, σ_{HP}

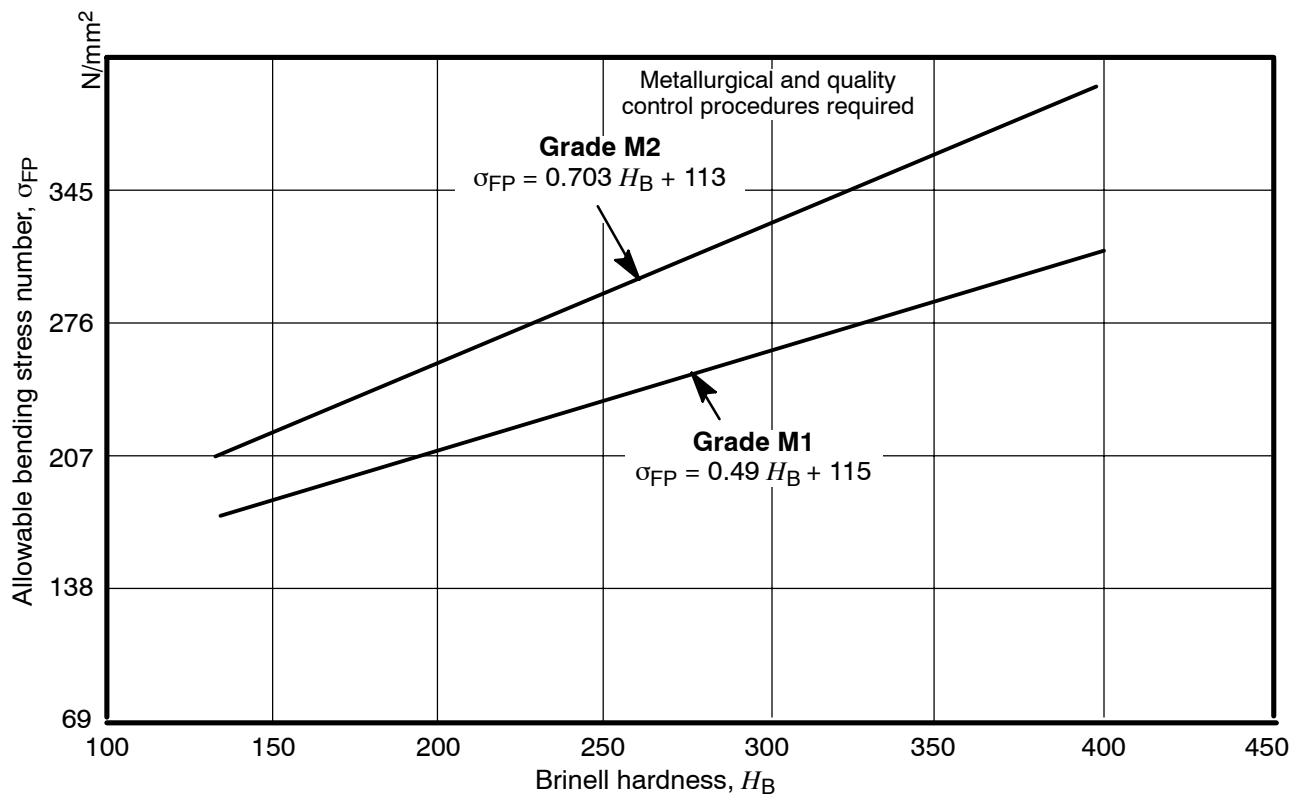


Figure 7 - Allowable bending stress number for through hardened steel gears, σ_{FP}