



American
Gear Manufacturers
Association®

DIVISION OF THE
Motion + Power
Manufacturers Alliance

ANSI/AGMA 6011-K25
(Revision of ANSI/AGMA 6011-J14)

American National Standard

Specification for High Speed Helical Gear Units

ANSI/AGMA 6011-K25

**American
National
Standard*****Specification for High Speed Helical Gear Units***

ANSI/AGMA 6011-K25

[Revision of ANSI/AGMA 6011-J14]

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Approved September 2, 2025

ABSTRACT

This standard includes design, lubrication, bearings, testing and rating for single and double helical external tooth, parallel shaft speed reducers or increasers. Units covered include those operating with at least one stage having a pitch line velocity equal to or greater than 35 meters per second or rotational speeds greater than 4500 rpm and other stages having pitch line velocities equal to or greater than 8 meters per second.

Published by

Motion + Power Manufacturers Alliance (MPMA)
1001 N. Fairfax Street, Suite 500, Alexandria, Virginia 22314

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Printed in the United States of America

ISBN: 978-1-64353-201-1

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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 ANSI/AGMA 6011-K25, *Specification for High Speed Helical Gear Units*.]

The first high speed gear unit standard, AGMA 421.01, was adopted as a tentative standard in October 1943. It contained formulas for computing the durability horsepower rating of gearing, allowable shaft stresses, and included a short table of application factors. AGMA 421.01 was revised and adopted as a full status standard in September 1947 and issued as AGMA 421.02.

The High Speed Gear Committee began work on the revision of AGMA 421.02 in 1951, which included: classification of applications not previously listed; changing the application factors from “K” values to equivalent *Service Factors*; revision of the rating formula to allow for the use of heat treated gearing; and develop a uniform selection method for high speed gear units. This Uniform Selection Method Data Sheet became AGMA 421.03A.

AGMA 421.03 was approved as a revision by the AGMA membership in October 1954.

The standard was reprinted as AGMA 421.04 in June 1957. It included the correction of typographical errors and the addition of a paragraph on pinion proportions and bearing span, which had been approved by the committee for addition to the standard at the October 1955 meeting.

In October 1959 the Committee undertook revisions to cover developments in the design, manufacture, and operation of high speed units with specific references to high hardness materials and sound level limits. The revisions were incorporated in AGMA 421.05 which was approved by the AGMA membership as of October 22, 1963.

The significant changes of 421.06 from 421.05 were: minimum pitch line speed was increased to 5000 feet per minute (25 meters per second); strength and durability ratings were changed; and some service factors were added. AGMA 421.06 was approved by the High Speed Gear Committee as of June 27, 1968, and by the AGMA membership as of November 26, 1968.

ANSI/AGMA 6011-G92 was a revision of 421.06 approved by the AGMA membership in October 1991. The most significant changes were the adaptation of ratings per ANSI/AGMA 2001-B88 and the addition of normal design limits for babbbitted bearings. ANSI/AGMA 6011-G92 used “application factor” and not “service factor”.

ANSI/AGMA 6011-H98 was a further refinement of ANSI/AGMA 6011-G92. One of the most significant changes was the conversion to an all metric standard. The rating methods were changed to be per ANSI/AGMA 2101-D04 which is the metric version of ANSI/AGMA 2001-D04. To provide uniform rating practices, clearly defined rating factors were included in the standard (ANSI/AGMA 6011-H98). While some equations slightly changed to conform to metric practices, no substantial changes were made to the rating practice for durability and strength rating. In addition, minimum pitch line velocity was raised from 25 m/s to 35 m/s and minimum rotational speed increased to 4000 rpm.

AGMA had reverted to the term “service factor” in their standards, which was reflected in ANSI/AGMA 6011-H98. The service factor approach is more descriptive of enclosed gear unit applications and can be defined as the combined effects of overload, reliability, desired life, and other application related factors. The service factor is applied only to the gear tooth rating, rather than to the ratings of all components. Components are designed based on the rated power and the guidelines given in this standard.

In continued recognition of the effects of scuffing in the rating of the gear sets, additional information on scuffing resistance was added to Annex B of ANSI/AGMA 6011-H98.

AGMA 427.01 was withdrawn. The information found in AGMA 427.01 was included in Annex D of ANSI/AGMA 6011-H98.

ANSI/AGMA 6011-I03 was a further refinement to ANSI/AGMA 6011-H98. Symbols were changed where possible to conform with ANSI/AGMA 2101-D04 and ISO standards. The minimum rotational speed was increased to 4500 rpm. Helix angle limits were changed, and a minimum axial contact ratio was added. The L/d limits were changed, and use of modified helices is encouraged based on the use of predicted rotor deflection and distortion. Bearing load design limits also changed. For gear tooth accuracy, reference was made to ANSI/AGMA 2015-1-A01 rather than to ANSI/AGMA 2000-A88. The Z_N and Y_N life factors now have a maximum rather than a minimum limit when the number of load cycles exceeds

10¹⁰. A table of dynamic factor as a function of accuracy grade was added. References to AGMA oil grades were removed; now only ISO viscosity grades are listed. To facilitate communications between purchaser and vendor, an annex with data sheets was added.

The first draft of ANSI/AGMA 6011-I03 was made in May 2001. It was approved by the AGMA membership in October 2003. It was approved as an American National Standard on February 12, 2004.

ANSI/AGMA 6011-J14 is a further refinement of ANSI/AGMA 6011-I03. For flank tolerance classification, previously referred to as accuracy grade or quality number, the referenced quality standard has been changed to ANSI/AGMA ISO 1328-1-B14.

This revision expanded the definition section and provides detailed attention to improving quality and reliability of high speed gearing. The most notable changes are:

- normative referral to material requirements in accordance with AGMA 923-B05 replacing ANSI/AGMA 2101-D04;
- tightened controls on allowable and recommended filtration of lubricants;
- clarified test requirements and test options;
- eliminated the rotor dynamics annex and transferred it for adoption to the Sound and Vibration Committee;
- expanded the annex on systems considerations for high speed gears;
- introduced new material in gear efficiency annex eliminating a specific method for determining mesh and windage losses;
- added an annex on metallurgical consideration for high speed gearing;
- added an annex on lubrication considerations;
- added an annex on procedures for assembly and functional testing of gear units.

The first draft of ANSI/AGMA 6011-J14 was made in May 2011. It was approved by the AGMA membership on July 28, 2014. It was approved as an American National Standard on August 8, 2014.

ANSI/AGMA 6011-K25 is a further refinement of ANSI/AGMA 6011-J14. For allowable stress numbers, Clause 5.5 was expanded to apply values for case hardened gears at 55 HRC minimum, reintroducing similar values from AGMA 421.06. The lubrication section, Clause 6, tightened the maximum allowable filtration values as defined in ISO 4406. The minimum recommended helix angle was increased.

In Annex B, the derating factor, C_w , was assigned to plv limits in calculating the load function. The example problem in Annex D was expanded to provide examples for both through hardened and case carburized gears. Formula calculations have been added to the summary tables for guidance on how the calculations should be applied.

In August 2022, the AGMA Technical Division Executive Committee (TDEC) approved the start of the ANSI/AGMA 6011-K25 project, which was written by the AGMA High Speed Units Committee. It was approved by the MPMA Board of Directors in August 2025. It was approved as an American National Standard on September 2, 2025.

Suggestions for improvement of this standard will be welcome. They may be submitted to tech@motionpower.org.

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American National Standard –

Specification for High Speed Helical Gear Units

1 Scope

This high speed helical gear units standard is provided as a basis for improved communication regarding:

- establishment of uniform criteria for rating;
- guidance for design considerations; and,
- identification of the unique features of high speed gear units.

1.1 Application

Operational characteristics such as lubrication, maintenance, vibration limits and testing are discussed. This standard is applicable to enclosed high speed, external toothed, single and double helical gear units of the parallel axis type. Units in this classification are:

- single stage units with pitch line velocities equal to or greater than 35 m/s or rotational speeds greater than 4500 rpm;
- multi-stage units with at least one stage having a pitch line velocity equal to or greater than 35 m/s and other stages having pitch line velocities equal to or greater than 8 m/s.

When specific experience exists, this standard may be applied to gear units that operate below the limits specified above.

Marine propulsion, aerospace and vehicle gearing are not covered by this standard.

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. 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.

2 Normative references

The following standards contain provisions that, 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 915-1, *Inspection Practices – Part 1: Cylindrical Gears – Tangential Measurements*

AGMA 923, *Metallurgical Specifications for Steel Gearing*

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

AGMA 947, *Gears – Thermal capacity*

AGMA 955, *Guidance for Industrial Gear Lubrication*

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

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

ANSI/AGMA 6000, *Specification for Measurement of Linear Vibration on Gear Units*

ANSI/AGMA 6001, *Design and Selection of Components for Enclosed Gear Drives*

ANSI/AGMA 6101, *Design and Selection of Components for Enclosed Gear Drives (Metric Edition)*

ANSI/AGMA 6025, *Sound for Enclosed Helical, Herringbone, and Spiral Bevel Gear Drives*

ANSI/AGMA 9005, *Industrial Gear Lubrication*

ANSI/AGMA ISO 1328-1, *Cylindrical Gears – ISO System of Flank Tolerance Classification – Part 1: Definitions and Allowable Values of Deviations Relevant to Flanks of Gear Teeth*

ISO 4406, *Hydraulic fluid power – Fluids – Method for coding the level of contamination by solid particles*

ISO 14635-1, *Gears – FZG test procedures – Part 1: FZG test method A/8,3/90 for relative scuffing load carrying capacity of oils*

3 Symbols, terminology and definitions

3.1 Symbols

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

NOTE: The symbols and terms contained in this document can 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.

Table 1 – Symbols used in equations

Symbol	Term	Units	Where first used
A	Allowable amplitude of unfiltered vibration	μm	8.4
A_{ct}	Amplitude at N_{ct}	mm	Figure 3
A_F	Amplification factor	- -	3.3.2
A_α	Pressure angle factor	- -	B.2
C_{SF}	Service factor for macropitting resistance	- -	5.2
C_w	Derating factor	N/mm ²	B.2
c_h	Lubricant specific heat at $(T_{\text{out}} + T_{\text{in}})/2$	kJ/kg °C	6.6.6
c_p	Specific heat of lubricant	kJ/kg °C	8.3
D_J	Nominal bearing bore diameter	mm	Table 3
d_{w1}	Pinion pitch diameter	mm	4.7
F_d	Incremental dynamic load	N	5.3.3
F_t	Transmitted tangential load	N	5.3.3
H_g	Heat generated at design point	kJ/min	6.6.6
H_{RC}	Surface hardness	HRC	5.5
K_H	Load distribution factor	- -	5.3.2
K_{He}	Mesh alignment correction factor	- -	5.3.2
K_{Hma}	Mesh alignment factor	- -	5.3.2

Symbol	Term	Units	Where first used
K_{Hmc}	Helix correction factor	--	5.3.2
K_{Hpm}	Pinion proportion modifier	--	5.3.2
K_{SF}	Service factor for bending strength	--	5.2
K_v	Dynamic factor	--	5.3.3
K_γ	Mesh load factor	--	5.3
L	Face width including gap	mm	4.7
L/d	Length to diameter ratio	--	4.7
N_{cm}	Initial (lesser) speed at 0.707 peak amplitude (critical)	rpm	7.2.4
N_{cp}	Final (greater) speed at 0.707 peak amplitude (critical)	rpm	7.2.4
N_{ct}	Rotor first critical, center frequency	rpm	7.2.4
N_{mc}	Maximum continuous rotor speed	rpm	4.12.1
n_L	Number of stress cycles	--	5.3.1
P_a	Allowable transmitted power for the gear set	kW	5.1
P_{ayu}	Allowable transmitted power for bending strength at unity service factor	kW	5.1
P_{azu}	Allowable transmitted power for macropitting resistance at unity service factor	kW	5.1
P_L	Power loss	kW	8.3
P_r	Gear rated power on the nameplate	kW	4.1
Q	Lubricant flow rate	l/min	6.6.6
SM	Separation margin	rpm	Figure 3
S_J	Diametral clearance	mm	Table 3
T_{in}	Average oil in temperature	°C	6.6.6
T_{out}	Average oil out temperature	°C	6.6.6
U_{max}	Amount of maximum residual rotor unbalance per plane	g-mm	4.12.1
W	Individual journal static loading	kg	4.12.1
W_{cpl}	Half weight of coupling and spacer	kg	7.2.1
W_r	Total weight of rotor	kg	7.2.1
Y_B	Rim thickness factor	--	5.4
Y_N	Stress cycle factor for bending strength	--	5.4.1
Y_θ	Temperature factor	--	5.3
Z_N	Stress cycle factor for macropitting resistance	--	5.3.1
Z_W	Hardness ratio factor for macropitting resistance	--	5.3
ΔT	Change in lubricant temperature	°C	8.3
ε_α	Standard transverse contact ratio	--	Annex K
ρ_{oil}	Lubricant density	kg/m ³	6.6.6
σ_{FP}	Allowable bending stress number	N/mm ²	5.5
σ_{HP}	Allowable contact stress number	N/mm ²	5.5

3.2 Nomenclature

The terms used, wherever applicable, conform to the following standards:

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

ISO 701, *International gear notation – Symbols for geometrical data*

3.3 Definitions

3.3.1 Ambient temperature

The temperature of the air in the immediate vicinity of the gear unit.

3.3.2 Amplification factor, A_F

The critical speed divided by the band width of the response frequencies at the half power point.

3.3.3 Babbitt

Babbitt, also known as White Metal, is a thin surface layer material that is often utilized in sliding bearings. These babbitt materials typically possess several key characteristics, per Wilcock [1]. They have excellent compatibility with steels and are non-scuffing under boundary and mixed lubrication regimes. They allow embedment of foreign particles and debris. They conform to errors in geometric construction under loaded operation. A variety of alloying elements are utilized in the manufacturing of babbitt. The majority of Babbitt alloys used with high speed gearing are tin based. Many babbitt formulations are standardized and well documented, such as in ASTM B23 [2] and AMS 4800F [3]. ASTM B23 Grade 2 is more typically used, while others are proprietary trade secrets. The first known babbitt alloy was developed by Isaac Babbitt in 1839.

3.3.4 Checking stand

A testing device that holds the pinion and gear very accurately in the correct position, with the shaft axes parallel at the proper center distance, so they can be rotated and the contact pattern of the teeth checked.

3.3.5 Critical speed

A shaft rotational speed at which the rotor bearing support system is in a state of resonance.

3.3.6 Damping coefficient

A characteristic of the bearings that is generally obtained from the bearing manufacturer or calculated with specialized bearing software.

3.3.7 Dynamic tooth contact pattern check

A check of the tooth contact pattern after operation.

3.3.8 Functional test

A mechanical running test performed on the assembled gear unit.

3.3.9 Gear rated power

The maximum power the gear is designed to transmit at a specified speed.

3.3.10 Half frequency whirl

An instability where the shaft whirls around the bearing axis at a frequency less than one-half shaft speed.

3.3.11 Helix modification

An intentional design change to the shape of the lead, generally defined relative to a helix on a specified pitch cylinder.

NOTE: Helix modification usually does not mean that the helix angle is modified. There are many possible helix modifications, including end relief, full crowning, or far more complex asymmetrical modifications such as those designed to compensate for thermal deformation and distortions expected under load.

3.3.12 Lateral vibration

Dynamic motion that is primarily in a direction perpendicular to the shaft centerline, i.e., in a radial direction.

3.3.13 Layout lacquer

A hard lacquer commonly used to make scribed lines more visible on smooth metal surfaces.

3.3.14 Normal transmitted power

An optional power level specified by the purchaser.

3.3.15 Overload

A load that is in excess of the torque equivalent to the gear rated power at rated speed.

3.3.16 Purchaser

Entity that purchases and is responsible for operating and maintaining the gear unit.

NOTE: In practice, the purchaser may contract different legal entities to operate, service, and maintain the gear unit. This distinction is not further reflected in this standard.

3.3.17 Profile modification

An intentional design change to the shape of the profile, generally defined relative to an involute in the transverse plane.

NOTE: Profile modification usually does not mean that the profile angle is modified. One common profile modification is tip relief, used so that highly loaded gears transfer loads between teeth smoothly.

3.3.18 Service power

The maximum installed continuous power capacity of the prime mover

3.3.19 Thrust collars

A pair of flanges or collars on single helical pinions that overlap the sides of the gear wheel such that the axial position of the pinion is controlled by the gear wheel.

NOTE: Thrust collars are used so that a high speed thrust bearing is not needed; the pinion thrust forces are transmitted to the gear wheel where a low speed thrust bearing can be used.

3.3.20 User

The company operating the gear unit according to the specification requirements.

3.3.21 Vendor

The entity responsible for the design, manufacture, and warranty of the gear unit.

NOTE: In practice, the gear unit design may be provided to the vendor by the purchaser. This distinction is not further reflected in this standard.

4 Design considerations

This standard shall be used in conjunction with appropriate AGMA standards. External loads shall be considered as acting in directions and rotations producing the most unfavorable stresses unless more specific information is available.

4.1 Gear rated power on the nameplate, P_r

Unless specifically agreed to by the purchaser and vendor, the rated power of the gear unit shall be equal to or greater than the service power.

For gear units between two items of driven equipment such as two compressor units, rated power of such gear units should normally not be less than item (a) or (b) below, whichever is greater.

- a. 110% of the maximum power required by the equipment driven by the gear unit;
- b. maximum power of the driver prorated between the driven equipment, based on normal transmitted power demands.

If maximum torque occurs at a speed other than rated speed, this torque and its corresponding speed shall be specified by the purchaser. Maximum continuous speed, N_{mc} , is normally the speed at least equal to 105% of the rated speed for variable speed units and is the rated speed for constant speed units.

All components shall be capable of handling both the maximum torque and the rated power.

4.2 Peak loads

The peak loads should be provided by the purchaser. Where torque variations develop peak loads that exceed 20% of the torque of the rated power, the magnitude and frequency of such torque variations shall be evaluated with regard to the endurance and yield properties of the materials used. See Annex C of this standard, and also ANSI/AGMA 2101, for further information.

4.3 Torsional and lateral vibrations

When an elastic system is subjected to externally applied, cyclic or harmonic forces, the periodic motion that results is called forced vibration. For the systems that high speed gears are typically used, two modes of vibration are normally considered.

- a. Lateral vibration, consisting of shaft dynamic motion that is in a direction perpendicular to the shaft centerline; and
- b. Torsional vibration, consisting of the amplitude modulation of torque measured peak to peak referenced to the axis of rotation.

In certain cases, axial or longitudinal vibration might also be considered.

Because of the wide variation of gear driven systems, Clause 7 of this standard outlines areas where proper assessment of the system can be necessary.

4.4 Tooth proportions and geometry

Any practical combination of tooth height, pressure angle and helix angle may be used. However, it is recommended that the gears have a minimum working depth of 1.80 times the normal module, a maximum normal pressure angle of 25 degrees, typical helix angles of 10 to 35 degrees, and a minimum axial contact ratio of 1.1 per helix.

NOTE: Lower helix angles may be used with a detailed analysis.

4.5 Spur gears

High speed spur gearing is not a typical configuration. For high-speed spur and combinations of helical and spur gear units see Annex K for more information.

4.6 Recommended tooth flank tolerance classification

Previously flank tolerance classification was referred to as accuracy grade or quality number. Table 2 presents recommended ANSI/AGMA ISO 1328-1 flank tolerance class as a function of pitch line velocity. Based on the experience and application, other flank tolerance classes may be appropriate.

Table 2 – Recommended flank tolerance class

Pitch line velocity, m/s	ANSI/AGMA ISO 1328-1 flank tolerance class
below 50	5
50 – 150	4
Over 150	3

NOTE: It is allowed to evaluate the gear accuracy by taking into consideration the measured data of the two mating gears (pinion and bull gear).

NOTE: If each pair of mating gears is a matched set, the profile and helix modifications should be evaluated when replacing either member of the set.

4.7 Pinion proportions

The maximum length-to-diameter (L/d) ratios are 1.6 for single helical and 2.2 for double helical designs. L/d shall be calculated by:

$$L/d = \frac{L}{d_{w1}} \quad (1)$$

where

L/d is length to diameter ratio;

L is face width including gap, mm;

d_{w1} is pinion pitch diameter, mm.

Higher L/d ratios are permitted when analytical load distribution methods are employed that yield load distribution values, K_H , that are less than the value calculated per 5.3.2 at the maximum L/d ratio and the proposal to the purchaser includes notice and justification of the L/d ratio to be used. The analytical method should include, but not be limited to, bending and torsional deflection and thermal distortion.

If the total helix mismatch, including bending deflection, torsional deflection, and thermal distortion exceeds 15 μm based on the gear rated power, an appropriate helix modification should be applied to reduce the total mismatch to $\leq 15 \mu\text{m}$, regardless of the L/d ratio. The modification is intended to provide a uniform load distribution across the entire face width. Gears with pitch line velocities in excess of 100 m/s are particularly susceptible to thermal distortion. Consideration should be given to the relationship of helix modifications to gear tooth accuracy; tighter helix tolerances can be appropriate. In addition, the profile of the pinion or gear wheel should be modified when necessary to compensate for bending deflections and thermal distortion.

When operating conditions other than gear rated power are specified by the purchaser, such as the normal transmitted power, the vendor should consider in the analysis the length of time and load range that the gear unit will operate at each condition, such as through a Miner's Rule analysis, so that the correct helix modification can be determined. When contact checks are to be furnished on gears with helix or profile modifications, the purchaser and vendor shall agree on the acceptable range of tooth contact patterns obtained in the checking stand, housing, or test stand.

4.8 Rotor construction

Several configurations can be applied in the construction of rotors. The most commonly used are listed below:

- Integral shaft and gear element. This configuration is commonly used for pinions, smaller gears, or rotating elements operating above a pitch line velocity of 150 m/s. The pinion or gear, integral with its shaft, is machined from a single blank;
- Solid blank shrunk on a shaft. The shrink fit may be used either with or without a mechanical torque transmitting device (such as key or spline). The connection of the gear to the shaft shall provide ample capacity to transmit torque, axial forces, and to ensure accurate positioning when considering centrifugal and thermal effects.

- Fabricated gear. A forged rim is welded directly to the fabricated substructure producing a one-piece welded gear. The shaft may be a part of the weldment. Fabricated gears should be analyzed to consider centrifugal and thermal stresses and fatigue life. Maximum pitch line velocity for welded gear construction is 130 m/s;
- Forged rim shrunk onto a substructure. The substructure may be forged, cast, or fabricated. The shaft may be a part of the substructure. During design, shrunk rims shall be analyzed for stresses and torque transmitting capacity. The analysis shall include the range of shaft and bore tolerances, centrifugal, and thermal effects (refer to solid blank shrunk on shaft above). The normal design limit for this type of construction is 60 m/s.

Combinations of the above are often used on multistage units.

Stresses and deflections at high speeds often dictate limits for a specific type of construction. High pitchline velocity, especially when combined with high loads, can require special material specifications and testing. Construction features such as holes in the gear body should be analyzed for their influence on the stresses. The influence of assumed maximum size inclusions and cracks may need to be considered using the methods of fracture mechanics, with testing of the material to ensure that there are no inclusions greater than the assumed maximum. Overall, a careful analysis of actual operating stresses and deflection should be made to ensure reliable operation.

4.9 Gear housing

The gear housing should be designed to provide a sufficiently rigid enclosed structure for the rotating elements that enables them to transmit the loads imposed by the system and protects them from the environment that they will operate. The vendor's design of the housing shall provide for proper alignment of the gearing when operating under the user's specified thermal conditions, and the torsional, radial and thrust loadings applied to its shaft extensions. In addition, it should be designed to facilitate proper lubricant drainage from the gear mesh and bearings.

The user's design of the supporting structure shall maintain the alignment of the gear unit within the specified tolerance. The tolerance for alignment shall be given in the gear unit's manual. Alignment should consider all specified torsional, radial and thrust loadings, and thermal conditions present during operation.

4.9.1 Breathers

Breathers shall be located and of sized to avoid pressure buildup and allow the exchange of air between the gear unit and the environment without the loss of lubricant. Breathers shall be capable of preventing the ingestion of contaminants from the environment into the gear unit.

Powered oil mist eliminators may be used on breathers to assist in preventing lubricant mist from escaping the gear unit. These usually keep the interior of the gear unit and lubricant reservoir slightly below atmospheric pressure and can also help minimize oil leakage.

Breathers may also incorporate a desiccant material to assist in reducing the water content in the gear unit. See 6.7.1 and 6.7.5 for information on desiccant breathers.

4.9.2 Special housing considerations

Certain applications can be subjected to operating conditions requiring special consideration. Some of these operating conditions are:

- temperature variations in the vicinity of the gear unit;
- relative thermal growth between mating system components;
- environmental elements that will attack the unit housing, rotating components, bearings or lubricant;
- inadequate support for the housing;
- high pitch line velocities that can affect lubricant distribution, create excessive temperature rise, or cause other adverse conditions.

4.9.3 Shaft seals

Where shafts pass through the housing, the housings shall be equipped with seals and deflectors that retain lubricant in the housing and prevent entry of foreign material into the housing. Easily replaceable labyrinth end seals and deflectors are preferred. The seals and deflectors shall be made of non-sparking materials. Lip seals have a very finite life and can generate enough heat at higher speeds to be a fire hazard. Surface velocity should be kept within the seal manufacturer's recommendation.

4.10 Bearings

Proper design of bearings is critical to the operation of high speed enclosed drive units. The bearing design is based on the gear rated power. Bearing design includes start-up and unloaded conditions.

Radial bearings are normally of the hydrodynamic sleeve or pad type. Thrust bearings are usually flat land, tapered land, or tilting pad type. Rolling element bearings are occasionally used.

4.10.1 Hydrodynamic bearings

Hydrodynamic bearings shall be lined with suitable bearing material, such as tin based babbitts. Tin alloy is usually preferred because of its higher corrosion resistance, easier bonding, and better high temperature characteristics. Hydrodynamic bearings shall have a rigid steel or other suitable metallic backing and be properly installed and secured in the housing against axial and rotational movement. Bearings are generally supplied split for ease of assembly. Selection of the particular design (sleeve, pad type or land bearing) shall be based on evaluation of surface velocity, surface loading, hydrodynamic film thickness, calculated bearing temperature, lubricant viscosity, lubricant flow rate, and bearing stability.

Heat is generated as a result of lubricant shear. Temperature is regulated by controlling the lubricant flow through the bearing and external cooling of the lubricant. The anticipated peak babbitt temperature and peak pressure should be kept within a range that is compatible with the babbitt and lubricant characteristics. See Table 3 for recommended design criteria.

Table 3 – Hydrodynamic babbitt bearing recommended design criteria ¹⁾

Type of bearing	Projected unit load ²⁾ (N/mm ²)	Minimum average lubricant film thickness ³⁾ (mm)	Measured bearing temperature ^{2) 4) 5)} (°C)	Maximum velocity ²⁾ (m/s)
Radial bearing				
- Fixed geometry	3.8	0.020	115	100
- Tilting pad	3.8	0.020	115	125
Thrust bearings				
- Tapered land	2.5	0.020	115	125
- Flat face	0.5	N/A	115	50
- Tilting pad	3.5	0.015	115	125

NOTES:

Table limits will generally not occur all together; one parameter alone can dictate the design.

¹⁾ Limits are for babbitt on steel backing. When other materials are used, established limits for these materials are permissible. Bearing clearances should be chosen to yield proper temperature, high stiffness and stability, as well as to ensure adequate clearance to cope with thermal gradients, whether steady, static, or transient. The average ratio of diametral clearance (S_J) to the nominal bore size (D_J), S_J / D_J , for radial bearings is approximately 0.002 mm/mm. Chrome-copper backing with greater thermal conductivity can be used to reduce babbitt temperature.

²⁾ Higher values are acceptable if supported by special engineering and testing. Higher values are also acceptable if supported by field experience. Projected load refers to the load applied to the bearing divided by the projected area of the bearing. The projected area for a fixed geometry (plane) or tilting pad bearing is calculated by multiplying the length of the bearing by its inside diameter.

³⁾ Average film thickness is calculated across the bearing width.

⁴⁾ Bearing temperature measurements are taken in the metallic backing material within 3 mm of the backing material/babbitt interface at the hottest operational zone of the bearing circumference.

⁵⁾ See AGMA 955 for more information on various lubricant temperature limits.

4.10.2 Radial bearings

Unless otherwise agreed by purchaser and gear vendor, at rated speed the predicted bearing temperature should be a maximum of 100°C, with an inlet oil temperature of 49°C and with the specified nominal lubricant oil inlet pressure. The predicted bearing temperature shall be a maximum of 115°C per Table 3. The calculation shall be at rated load for radial bearings and at the maximum operating thrust load for thrust bearings. If the above criteria cannot be met, the purchaser and the gear vendor shall agree on acceptable bearing temperatures.

The bearing temperature limit is applicable to babbbitted (white metal) bearings and to mineral oil lubricants. Use of bearing materials and synthetic lubricants with higher temperature capability can allow a higher bearing temperature limit. Bearings with high velocity or high loading or both can require special bearing material or special lubrication, and, in some cases, the use of higher temperature limits can result in an improved design.

4.10.3 Radial bearing stability

Hydrodynamic radial bearings shall be designed such that damaging self-generated instabilities (e.g., half frequency whirl or pad flutter) do not occur at any anticipated operational load or speed. Half frequency whirl can occur in lightly loaded high speed bearings.

Tilting pad bearings can be prone to pad oscillations, also known as pad flutter, that can occur in regions of high speed with low load as well as low oil flow rate. Pad flutter is a tilting pad bearing phenomenon that can occur on the pads that are located opposite of the load vector. If these pads become unloaded, they might not be able to establish an equilibrium position, as reported by Nicholas [4]. Pad flutter can cause babbbitt damage but rarely leads to high synchronous vibration.

4.10.4 Thrust bearings

Thrust bearings shall be furnished with all gear units unless otherwise specified. A thrust bearing is usually provided only on the low speed shaft for all double helical gears and on single helical gears fitted with thrust collars, see 4.10.5. Thrust bearings are generally provided on each shaft for all single helical gears not fitted with thrust collars. If the axial position of any of the shafts depends on items outside the gear unit, the purchaser and vendor shall agree to the arrangement relative to the thrust bearings.

When gear units are supplied without thrust bearings, some type of end float limitation shall be provided at shaft couplings to maintain positive axial positioning of the gear rotors and connected rotors. Provisions to prevent contact of the rotating elements with the gear housing shall be provided unless otherwise specifically agreed to by the purchaser.

The design of a hydrodynamic bearing to sustain thrust shall include consideration of heat generation, lubricant flow, bearing material, load capacity, speed and stiffness. Thrust bearing load capacity shall consider the maximum expected loads created by coupling displacement due to thermal expansion and the possibility of torque lock-up loads from gear couplings. When other external thrust forces are anticipated, the vendor shall be notified of their magnitudes. See C.5.2 for further information on coupling axial alignment.

4.10.5 Thrust collars

Thrust collars may be used on single helical gears to maintain alignment and transfer axial forces from the pinion to the gear wheel. The thrust collars extend beyond the pinion outside diameter and typically have a conical shape that contacts a similarly shaped surface on the mating gear rim at a diameter below the root diameter of the gear. Other designs also exist and may be used.

4.10.6 Rolling element bearings

Selection of rolling element bearings shall be based upon the application requirements and the bearing manufacturer's recommendations and rating methods. For these applications, an L_{10} life shall be 50 000 hours minimum at gear rated power.

4.11 Threaded fasteners

Refer to ANSI/AGMA 6101, ANSI/AGMA 6001, or equivalent fastener standards.

4.12 Shafting

The pinion and gear shafts should be designed for the maximum bending and maximum torsional shear stresses at the gear rated power by the appropriate methods and allowable values from ANSI/AGMA 6101. To ensure that the shaft is not under, or oversized, other available analysis methods may be used for a more in-depth study.

4.12.1 Balance

All gear rotating elements within the speed limits of this standard should be dynamically balanced in two planes after assembly of the rotor. Rotors with single keys for couplings should be balanced with their keyway fitted with a fully crowned half-key so that the shaft keyway is filled for its entire length. The rotating elements should be balanced to the level of Equation (2) or better:

$$U_{\max} = \frac{6350W}{N_{\text{mc}}} \quad (2)$$

where

$U_{\max}$ is maximum amount of residual rotor unbalance per plane, g-mm;

W is individual journal static loading per plane, kg;

N_{mc} is maximum continuous rotor speed, rpm.

The balancing machine shall be calibrated, with documentation of the calibration available upon purchaser request.

4.13 Sound pressure level

If ANSI/AGMA 6025 is specified by the purchaser, the gear vendor shall provide the expected sound pressure level and the expected sound intensity level with or without load.

NOTE: The sound intensity level is the value to be used for calculating the sound emission of the complete shaft train that is in the scope of the entity responsible for the sound emission of the complete shaft train.

5 Rating of gears

5.1 Rating criteria

The macropitting resistance power rating and bending strength power rating for each gear mesh in the unit shall be calculated per ANSI/AGMA 2101 using the factors specified in this clause. The lowest value obtained shall be used as the allowable transmitted power of the gear set.

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

$$P_a = \text{the lesser of } \frac{P_{\text{azu}}}{C_{\text{SF}}} \text{ and } \frac{P_{\text{ayu}}}{K_{\text{SF}}} \quad (3)$$

where

P_{azu} is allowable transmitted power for macropitting resistance at unity service factor, kW ($C_{\text{SF}} = 1.0$);

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

C_{SF} is service factor for macropitting resistance; recommended values are shown in Annex A;

K_{SF} is service factor for bending strength; recommended values are shown in Annex A.

The gear rated power shall be less than, or equal to, the allowable transmitted gearset power rating:

$$P_r \leq P_a \quad (4)$$

where

P_r is gear rated power on the nameplate, kW.

It is recognized that all prime movers have overload capacity that should be specified.

5.2 Service factor, C_{SF} and K_{SF}

The service factor includes the combined effects of overload, reliability, and other application related influences. The AGMA service factor used in this standard depends on experience acquired in each specific application.

In determining the service factor, consideration should be given to the fact that systems develop a peak torque, whether from the prime mover, driven machinery, or transitional system vibrations, which is greater than the nominal torque.

When an acceptable service factor is not known from experience, the values shown in Annex A should be used as minimum allowable values.

5.3 Macropitting resistance power rating

The macropitting resistance of gear teeth is considered to be a Hertzian fatigue phenomenon. Non-progressive macropitting and progressive macropitting are illustrated and discussed in ANSI/AGMA 1010.

The purpose of the macropitting resistance formula is to determine a load rating where destructive macropitting of the teeth does not occur during their design life. Ratings for macropitting 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.

The macropitting resistance power rating for gearing within the scope of this standard shall be determined by the rating methods and procedures of ANSI/AGMA 2101, when using service factors, with the values for factors as specified in this clause and 5.3.1 through 5.3.3:

Z_W is hardness ratio factor, $Z_W = 1.0$;

Y_θ is temperature factor, $Y_\theta = 1.0$;

K_γ is mesh load factor, $K_\gamma = 1.0$;

Z_N is stress cycle factor (see 5.3.1);

K_H is load distribution factor (see 5.3.2);

K_v is dynamic factor (see 5.3.3).

5.3.1 Stress cycle factor, Z_N

Stress cycle factor, Z_N , shall be calculated using the lower curve for Z_N and should be based on a minimum of 40 000 hours of service at rated operating speed.

If less than 40 000 hours is used for rating, it shall be with the specific approval of the purchaser and shall be so stated along with the rating.

$$Z_N = 2.466 n_L^{-0.056} \quad (5)$$

where

n_L is the number of stress cycles.

When the number of stress cycles exceeds 10^{10} (i.e., speed above 4167 rpm for 40 000 hours), Z_N should be less than or equal to 0.68.

5.3.2 Load distribution factor, K_H

K_H is the load distribution factor. Values are to be per ANSI/AGMA 2101. The following values shall be used with the empirical method:

K_{Hma} is mesh alignment factor. Use values for precision gear units as defined in ANSI/AGMA 2101;

K_{Hmc} is helix correction factor,

$$K_{Hmc} = 0.8;$$

K_{Hpm} is pinion proportion factor,
 $K_{Hpm} = 1.0$;
 K_{He} is mesh alignment correction factor,
 $K_{He} = 0.8$.

The value used for K_H shall not be less than 1.1.

NOTE: The above empirical rating method assumes properly matched helix whether unmodified or modified, teeth central to the bearing span and tooth contact checked at assembly with contact adjustments as required. If these conditions are not met, or for wide face gears, it may be desirable to use an analytical approach to determine load distribution factor. AGMA 927 [5] provides one such approach.

5.3.3 Dynamic factor, K_v

Dynamic factors account for internally generated gear tooth dynamic loads that are caused by gear tooth meshing action at a non-uniform relative angular velocity.

The dynamic factor is the ratio of transmitted tangential tooth load to the total tooth load, including the dynamic effects.

$$K_v = \frac{F_d + F_t}{F_t} \quad (6)$$

where

F_d is incremental dynamic tooth load due to the dynamic response of the gear pair to transmission error excitation, N;
 F_t is transmitted tangential load, N.

Dynamic forces on gear teeth result from gear transmission error. The transmission error is caused by:

- inherent variations in gear accuracy as manufactured;
- gear tooth deflections that are dependent on the variable mesh stiffness and the transmitted load.

The dynamic response to transmission error excitation is influenced by:

- masses of the gears and connected rotors;
- shaft and coupling stiffness;
- damping characteristics of the rotor and bearing system.

ANSI/AGMA ISO 1328-1 flank tolerances for pitch and profile shall be used for the selection of dynamic factor, K_v as shown in Table 4. Within the 1.09 to 1.15 dynamic factor range, the trend is for K_v to vary in nearly a direct relationship with flank tolerance classes from 2 to 5 as shown in Table 4.

Table 4 – Dynamic factor as a function of flank tolerance classification

ANSI/AGMA ISO 1328-1 flank tolerance class	Dynamic factor, K_v
5	1.15
4	1.13
3	1.11
2	1.09

The dynamic factor, K_v , does not account for dynamic tooth loads that can occur due to torsional or lateral natural frequencies. System designs should avoid having such natural frequencies close to an excitation frequency associated with an operating speed, since the resulting gear tooth dynamic loads can be very high.

Refer to ANSI/AGMA 2101 for additional considerations influencing dynamic factors.

5.4 Bending strength power rating

The intent of the AGMA strength rating formula is to determine the load that can be transmitted for the design life of the gear unit without causing root fillet cracking or fatigue failure.

The gear rim thickness shall be sufficient for the calculated rim thickness factor to be 1.0 per ANSI/AGMA 2101.

Occasionally, manufacturing tool marks, wear, surface fatigue, or plastic flow can limit bending strength due to stress concentration around large, sharp cornered pits or wear steps on the tooth surface.

The bending strength power rating for gearing within the scope of this standard shall be determined by the rating methods and procedures of ANSI/AGMA 2101, when using service factors, with the values for factors as specified in this clause and 5.4.1, 5.3.3, and 5.3.2:

- Y_θ is temperature factor, $Y_\theta = 1.0$;
- Y_B is rim thickness factor, $Y_B = 1.0$;
- Y_N is stress cycle factor (see 5.4.1);
- K_v is dynamic factor (see 5.3.3);
- K_H is load distribution factor (see 5.3.2).

5.4.1 Stress cycle factor, Y_N

Stress cycle factor, Y_N , is calculated by Equation (7) that is based on the lower curve for Y_N from ANSI/AGMA 2101 and should be based on a minimum of 40 000 hours of service at rated operating speed.

If less than 40 000 hours is used for rating, it shall be with the specific approval of the purchaser and shall be so stated along with the rating.

$$Y_N = 1.6831 n_L^{-0.0323} \quad (7)$$

where

n_L is number of stress cycles.

When the number of stress cycles exceeds 10^{10} (i.e., speed above 4167 rpm for 40 000 hours), Y_N should be less than or equal to 0.80.

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

Allowable stress numbers that are dependent upon material and processing, are given in ANSI/AGMA 2101. ANSI/AGMA 2101 also specifies the treatment of momentary overload conditions. Carburized gears with a hardness of 55 HRC and higher may be used. For gears with hardness in the range of 55–58 HRC, the allowable contact and bending stress numbers in Figure 1 and Figure 2 shall be used in Equation (8) and Equation (9).

$$\sigma_{HP} = 43 H_{RC} - 944 \text{ (maximum } 1550 \text{ N/mm}^2\text{)} \quad (8)$$

$$\sigma_{FP} = 8 H_{RC} - 14 \text{ (maximum } 450 \text{ N/mm}^2\text{)} \quad (9)$$

where

- σ_{HP} is the allowable contact stress number, N/mm²;
- σ_{FP} is the allowable bending stress number, N/mm²;
- H_{RC} is the surface hardness, HRC.

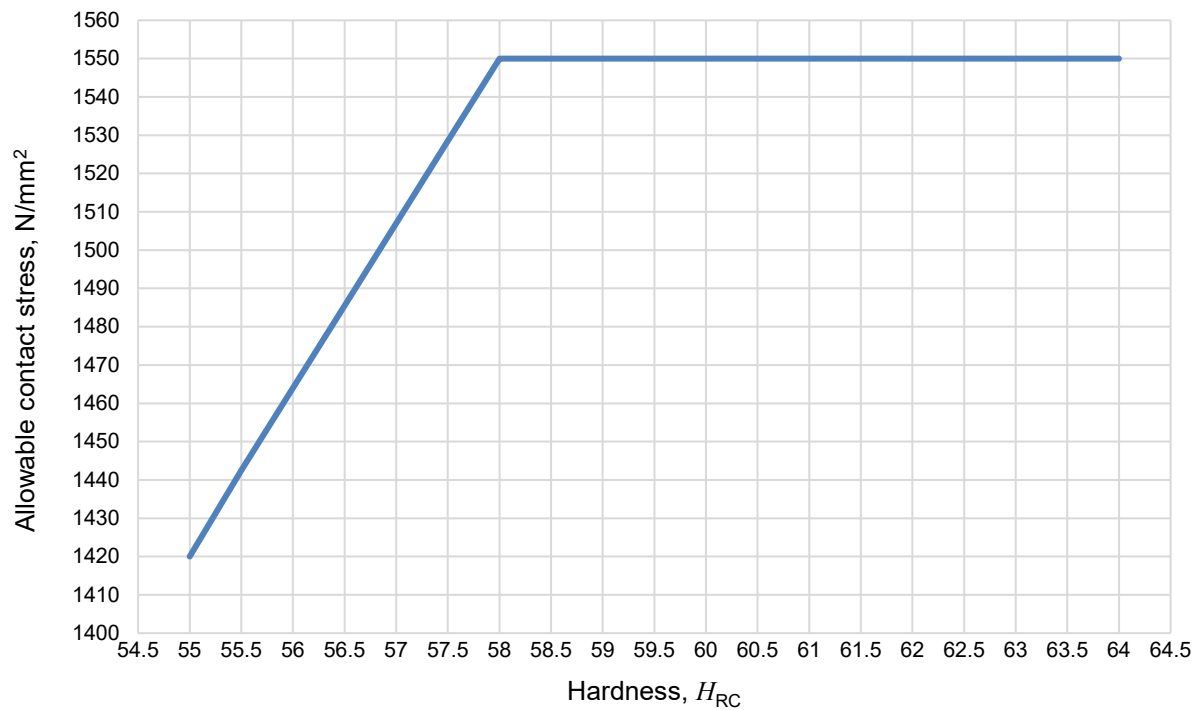


Figure 1 – Allowable contact stress, σ_{HP} , carburized steel

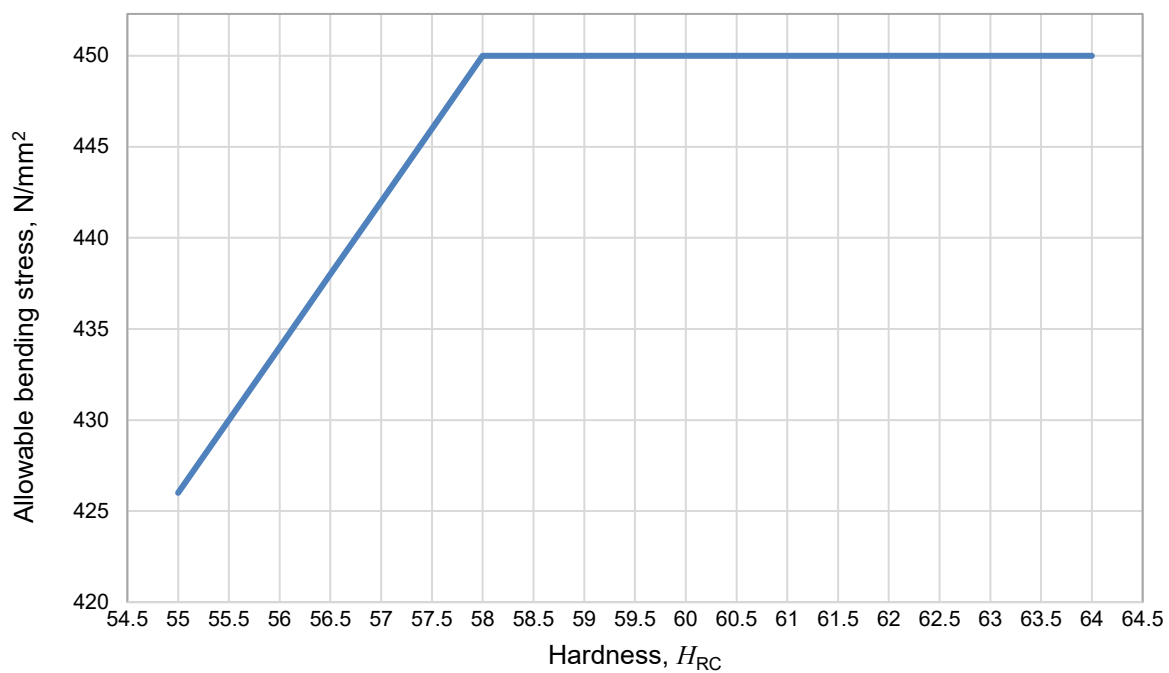


Figure 2 – Allowable bending stress, σ_{FP} , carburized steel

AGMA 923 shall be used in conjunction with this standard. Tables 1, 2, and 4 from AGMA 923-C22 supersede Tables 7, 9, and 10 from ANSI/AGMA 2101-D04.

Three grades of material have been established. Grade 1 material shall not be used for gears rated by this standard. Both Grade 2 and Grade 3 shall be heat treated under carefully controlled conditions. The choice of material, hardness and selection between grade 2 or 3 is left to the gear designer; however, values of σ_{HP} and σ_{FP} shall be for grade 2 materials. Castings and surface hardened gearing, such as flame or induction hardened gearing, are not approved for compliance with this standard.

The cleanliness of steel used for gears controlled by this standard, specifically nonmetallic inclusions in the raw material, shall be controlled by AGMA 923-C22, Table 2, Item 4, Alternative A, B, or C; but not Alternative D.

The minimum total raw material area reduction ratio shall be achieved by hot working processes that ensure through-section working.

Additional testing, such as ultrasonic, surface temper, and magnetic particle inspection of high speed gear rotors that are subject to high fatigue cycles or high stress, or both, during operation, shall be conducted. The concern is not only contact and bending stresses but also includes residual stresses and operating centrifugal radial and hoop stresses.

Refer to Annex F for additional metallurgical considerations. For details on tooth failure, refer to ANSI/AGMA 1010.

5.6 Reverse loading

For idler gears and other gears where the teeth are completely reverse loaded on every cycle, use 70% of the allowable bending stress number, σ_{FP} , in ANSI/AGMA 2101.

5.7 Scuffing resistance

High speed gear units can be subject to high load and high sliding velocity operating conditions. Under these conditions, the lubricant film might not adequately separate the flank surfaces and full elastohydrodynamic lubrication (EHL) conditions might not exist. The consequence can be an instantaneous metal-to-metal contact resulting in a localized solid-phase welding and tearing of metal away from the mating teeth. This failure mode is known as scuffing and is a serious concern for high speed helical gear units. The risk for scuffing increases if the high load and or high sliding velocity operating conditions are varying rather than in a steady state.

Scuffing appears as a dull matte or rough finish on gear flanks. It commonly occurs at the extreme end regions of the contact path or near the points of a single pair of teeth contact. Scuffing is a form of severe adhesive wear.

Scuffing is not a fatigue phenomenon. The risk of scuffing damage varies with the material and heat treatment, surface roughness, sliding velocity, geometry of the gear teeth, and viscosity and composition of the lubricant. Changes in any or all of these factors can change the risk of scuffing.

Further information on scuffing is provided in Annex B of this document. It is highly recommended that Annex B be used as a minimum to assess for the risk of scuffing. More detailed methods for assessing the risk of scuffing can be found in AGMA 925, Blok's flash temperature method. Additional information related to scuffing in aerospace gearing can be found in AGMA 911-B21 [6], Clause 7.4.4.

Anything that can reduce the tooth flank temperature or the flash temperatures will lower the risk for scuffing. See ANSI/AGMA 1010 for additional information on methods to mitigate the risk of scuffing.

NOTE: For rolling element bearings the risk of scuffing exists. Consult the bearing manufacturer for more information.

5.8 Hertzian Fatigue

Hertzian fatigue can result in three forms of tooth surface distress; macropitting, micropitting and subcase fatigue. ANSI/AGMA 2101 provides the general calculation for predicting macropitting and subcase fatigue. See 5.3 for additional information on macropitting.