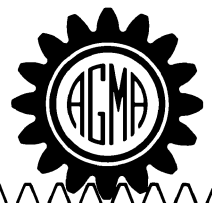


AMERICAN NATIONAL STANDARD

***Specification for High Speed Helical Gear
Units***

ANSI/AGMA 6011-I03



AGMA STANDARD

American National Standard

Specification for High Speed Helical Gear Units

ANSI/AGMA 6011-I03

[Revision of ANSI/AGMA 6011-H98]

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Approved February 12, 2004

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.

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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 Standard 6011-103, *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-C95 which is the metric version of ANSI/AGMA 2001-C95. To provide uniform rating practices, clearly defined rating factors were included in the standard (ANSI/AGMA 6011-H98). While some equations may slightly change 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 has 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 drive 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 service 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 has been withdrawn. The information found in AGMA 427.01 was included in annex D of ANSI/AGMA 6011-H98.

ANSI/AGMA 6011-I03 is a further refinement to ANSI/AGMA 6011-H98. Symbols have been changed where possible to conform with ANSI/AGMA 2101-C95 and ISO standards. The minimum rotational speed has been increased to 4500 rpm. Helix angle limits have changed, and a minimum axial contact ratio limit has been added. The L/D limits have changed, and use of modified leads is now encouraged with the use of predicted rotor deflection and distortion. Bearing load design limits have also changed. For gear tooth accuracy, reference is now 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 10^{10} . A table of dynamic factor as a function of accuracy grade has been added. References to AGMA oil grades have been removed; now only ISO viscosity grades are listed. To facilitate communications between purchaser and vendor, an annex with data sheets has been added.

Realistic evaluation of the various rating factors of ANSI/AGMA 6011-I03 requires specific knowledge and judgment which come from years of accumulated experience in designing, manufacturing and operating high speed gear units. This input has been provided by the AGMA High Speed Gear Committee.

The first draft of 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.

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 – Specification for High Speed Helical Gear Units

1 Scope

This high speed helical gear unit 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 meters per second 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 meters per second and other stages having pitch line velocities equal to or greater than 8 meters per second.

Limits specified are generally accepted design limits. When specific experience exists for gear units of similar requirements above or below these limits, this experience may be applied.

Marine propulsion, aircraft, automotive, and epicyclic gearing are not covered by this standard.

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.

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

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

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

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

ANSI/AGMA 6001-D97, *Design and Selection of Components for Enclosed Gear Drives*

ANSI/AGMA 6025-D98, *Sound for Enclosed Helical, Herringbone, and Spiral Bevel Gear Drives*

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

Table 1 - Symbols used in equations

Symbol	Term	Units	Reference paragraph
A	Allowable double amplitude of unfiltered vibration	μm	7.5
A_{ct}	Amplitude at N_{ct}	μm	7.3.3.3
AF	Amplification factor	--	7.3.3.3
C_{SF}	Service factor for pitting resistance	--	5.2
CRE	Critical response envelope	rpm	7.3.3.3
c_p	Specific heat of lubricant	$\text{kJ}/(\text{kg}^\circ\text{C})$	8.2.5
D_J	Nominal bearing bore diameter	mm	Table 4
d	Pinion operating pitch diameter	mm	4.6
F_d	Incremental dynamic load	N	5.3.3
F_t	Transmitted tangential load	N	5.3.3
K_B	Rim thickness factor	--	5.4
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
K_{Hmc}	Lead correction factor	--	5.3.2
K_{Hpm}	Pinion proportion modifier	--	5.3.2
K_s	Size factor	--	5.3
K_{SF}	Service factor for bending strength	--	5.2
K_v	Dynamic factor	--	5.3.3
L	Face width including gap	mm	4.6
N_{cm}	Initial (lesser) speed at $0.707 \times$ peak amplitude (critical)	rpm	7.3.3.3
N_{cp}	Final (greater) speed at $0.707 \times$ peak amplitude (critical)	rpm	7.3.3.3
N_{ct}	Rotor first critical, center frequency	rpm	7.3.3.3
N_{mc}	Maximum continuous rotor speed	rpm	4.1
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 pitting resistance at unity service factor	kW	5.1
P_L	Power loss	kW	8.2.5
P_S	Service power of enclosed drive	kW	4.1
Q_{LUBE}	Lubricant flow	kg/sec	8.2.5
S_J	Diametral clearance	mm	Table 4
SM	Separation margin	rpm	7.3.3.3
U_{max}	Amount of residual rotor unbalance	g-mm	7.4
W	Journal static loading	kg	7.4
W_{cpl}	Half weight of coupling and spacer	kg	7.3.3.2
W_r	Total weight of rotor	kg	7.3.3.2
Y_N	Stress cycle factor for bending strength	--	5.4.1
Y_θ	Temperature factor	--	5.3
Z_N	Stress cycle factor for pitting resistance	--	5.3.1
Z_R	Surface condition factor for pitting resistance	--	5.3
Z_W	Hardness ratio factor for pitting resistance	--	5.3
ΔT	Change in lubricant temperature	$^\circ\text{C}$	8.2.5
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:

AGMA 904-C96, *Metric Usage*

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

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

4 Design considerations

This standard should be used in conjunction with appropriate current AGMA standards. External loads must be considered as acting in directions and rotations producing the most unfavorable stresses unless more specific information is available. Allowances must be made for peak loads.

4.1 Service power, P_S

Service power of an application is defined as the maximum installed continuous power capacity of the prime mover, unless specifically agreed to by the purchaser and vendor. For example, for electric motors, maximum continuous power will be the motor nameplate power rating multiplied by the motor service factor.

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

- 110 percent of the maximum power required by the equipment driven by the gear;
- maximum power of the driver prorated between the driven equipment, based on normal power demands.

If maximum torque occurs at a speed other than maximum continuous 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 specified (or nominal) pinion speed for variable speed units and is the rated pinion speed for constant speed units.

All components shall be capable of transmitting the service power.

4.2 High transient torque levels

Where unusual torque variations develop peak loads which exceed the application power by a ratio greater than the service factor, C_{SF} or K_{SF} , specified for the application, the magnitude and frequency of such torque variations should be evaluated with regard to the endurance and yield properties of the materials used. See annex D and also ANSI/AGMA 2101-C95, subclause 16.3.

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 in which high speed gears are typically used, two modes of vibration are normally considered.

- Lateral or radial vibration, which considers shaft dynamic motion that is in a direction perpendicular to the shaft centerline; and
- Torsional vibration, which considers 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 may be necessary. In addition, appropriate responsibility between the vendor and purchaser must be clearly delineated.

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, a helix angle of 5 to 45 degrees, and a minimum axial contact ratio of 1.1 per helix.

4.5 Recommended accuracy grade

Table 2 presents recommended ANSI/AGMA 2015-1-A01 accuracy grades as a function of pitch line velocity. Based on experience and application, other accuracy grades may be appropriate.

Table 2 - Recommended accuracy grades

Pitch line velocity, m/s	ANSI/AGMA 2015-1-A01 accuracy grade
35 – 100	A5
100 – 160	A4
Over 160	A3

4.6 Pinion proportions

Table 3 presents maximum length-to-diameter (L/d) ratios for material hardening methods in current use. The L/d values shown in table 3 apply to helical gears when designed to transmit the service power. Generally, 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 per table 3. A detailed analytical method should include, but not be limited to, bending and torsional deflection and thermal distortion.

Table 3 – Recommended maximum length-to-diameter (L/d) ratios

Hardening method	Maximum L/d ratio	
	Double helical	Single helical
Through hardened	2.2	1.6
Case hardened	2.0	1.6
NOTE: L = face width including gap, mm; d = pinion operating pitch diameter, mm		

No matter what the L/d ratio is, if the combination of tooth and rotor deflection and distortion exceeds 25 μm for through hardened gears, or 18 μm for case hardened gears, then an analytically determined lead modification should be applied in order to reduce the total mismatch to a magnitude below these values. Determination of the combined tooth and rotor deflection shall be based on the service power. The modification is intended to provide a uniform load distribution across the entire face width. Working flanks of the pinion or gear wheel should be modified when necessary to compensate for torsional and bending deflections and thermal distortion. 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 lead modifications to gear tooth accuracy.

When a higher L/d ratio than tabulated in table 3 is proposed, the gear vendor shall submit justification

in the proposal for using the higher L/d ratio. Purchasers should be notified when L/d ratios exceed those in table 3. When operating conditions other than gear rated power are specified by the purchaser, such as the normal transmitted power, the gear vendor shall consider in the analysis the length of time and load range at which the gear unit will operate at each condition so that the correct lead modification can be determined. When modified leads are to be furnished, purchaser and vendor shall agree on the tooth contact patterns obtained in the checking stand, housing or test stand.

4.7 Rotor construction

Several configurations may 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 meters per second. 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). When no torque transmitting device is used, the shrink fit must provide ample capacity to transmit torque when considering centrifugal and thermal effects. When a torque transmitting device is used, the shrink fit must provide ample location support 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 meters per second;
- Forged rim shrunk onto a substructure.** The substructure may be forged, cast, or fabricated. The shaft may be a part of the substructure. Shrunk rims shall consider stresses and torque transmitting capacity due to fit, centrifugal, and thermal effects (refer to item b). The normal design limit for this type of construction is 60 meters per second.

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, may require special material specifications and/or testing. Construction features such as holes in the gear body should be analyzed for their influence on the stress. The influence of real or virtual inclusions and/or 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.

4.8 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 in which they will operate. The vendor's design of the housing must 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 must maintain proper and stable alignment of the gearing. Alignment must consider all specified torsional, radial and thrust loadings, and thermal conditions present during operation.

4.8.1 Special housing considerations

Certain applications may 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 which may affect lubricant distribution, create excessive temperature rise, or cause other adverse conditions.

4.8.2 Shaft seals

Where shafts pass through the housing, the housings shall be equipped with seals and deflectors that shall effectively retain lubricant in the housing and prevent entry of foreign material into the housing. Easily replaceable labyrinth-type end seals and deflectors are preferred. The seals and deflectors shall be made of nonsparking materials. Lip-type 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 conservative recommendation.

4.9 Bearings

Proper design of bearings is critical to the operation of high speed enclosed drive units. The bearing design shall consider normal service power.

Radial bearings are normally of the hydrodynamic sleeve or pad type. Thrust bearings are usually flat land, tapered land, or thrust pad type. Rolling element bearings are occasionally used when speeds are at the very low end of the high speed range. Bearing design shall consider start-up and unloaded conditions, as well as normal service power.

4.9.1 Hydrodynamic bearings

Hydrodynamic bearings shall be lined with suitable bearing material. Tin and lead based babbits (white metal) are among the most widely used bearing materials. Tin alloy is usually preferred over lead alloys 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 at running speeds 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 as related to bearing lubricant discharge temperatures should be kept within a range that is compatible with the bearing material and lubricant characteristics. See table 4 for design limits.

Table 4 - Hydrodynamic babbitt bearing design limits¹⁾

Type of bearing	Projected unit load, ³⁾ N/mm ²	Minimum lubricant film thickness, mm	Bearing metal temperature, ^{2) 3)} °C	Maximum velocity, ³⁾ m/s
Radial bearing				
- Fixed geometry	3.8	0.020	115	100
- Tilting pad	4.2	0.020	115	125
Thrust bearings				
- Tapered land	2.5	0.020	115	125
- Flat face	0.5	N/A	115	50
- Tilt pad	3.5	0.015	115	125

NOTE: Table limits will generally not occur all together; one parameter alone may 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.

²⁾ Bearing temperature measurements are taken in the backing material within 3 mm of the backing material/babbitt interface at the hottest operational zone of the bearing circumference.

³⁾ Higher values are acceptable if supported either with special engineering or testing and field experience.

4.9.2 Radial bearing stability

Hydrodynamic radial bearings shall be designed such that damaging self generated instabilities (e.g., half frequency whirl) do not occur at any anticipated operational load or speed. Hydrodynamic instability occurs when a journal does not return to its established equilibrium position after being momentarily displaced. Displacement introduces an instability in which the journal whirls around the bearing axis at less than one-half journal speed. Known as "half frequency whirl", this instability may occur in lightly loaded high speed bearings.

4.9.3 Thrust bearings

Thrust bearings shall be furnished with all gear units unless otherwise specified. Thrust bearings are generally provided on the low speed shaft for all double helical gears and on single helical gears fitted with thrust collars (see 4.9.4). Thrust bearings are generally provided on each shaft for all single helical gears not fitted with thrust type 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 ele-

ments with the gear casing shall be provided unless otherwise specifically agreed to by the purchaser.

The design of a hydrodynamic bearing to sustain thrust is as complicated as the design of a radial hydrodynamic bearing. Complete analysis requires consideration of heat generation, lubricant flow, bearing material, load capacity, speed and stiffness. Thrust bearing load capacity should consider the possibility of torque lock-up loads from couplings. When other external thrust forces are anticipated, the vendor must be notified of their magnitudes.

4.9.4 Thrust collars

Thrust collars (also known as rider rings) may be used to counteract the axial gear thrust developed by single helical gear sets.

Thrust collars arranged near each end of the teeth on a single helical pinion and having bearing surface contact diameters greater than that of the pinion outside diameter may be used to carry the gear mesh thrust forces. Typically the thrust collars have a conical shape where they contact a similarly shaped surface on the mating gear rim located below the root diameter of the gear. Other designs also exist and may be used. Single helical gear sets using thrust collars may be positioned in the housing in a similar fashion to that of double helical gear elements.

4.9.5 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 normal applications, an L_{10} life of 50 000 hours minimum is required.

4.10 Threaded fasteners

Refer to ANSI/AGMA 6001-D97, *Design and Selection of Components for Enclosed Gear Drives*, clause 8.

4.11 Shafting

The pinion and gear shafts may normally be designed for the maximum bending and maximum torsional shear stresses at service power (see 4.1) by the appropriate methods and allowable values from ANSI/AGMA 6001-D97, clause 4, or other equivalent standards. In some instances, this may result in an oversized or undersized shaft. Therefore, an in-depth study using other available analysis methods may be required.

5 Rating of gears

5.1 Rating criteria

The pitting resistance power rating and bending strength power rating for each gear mesh in the unit must be calculated. 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:

$$P_a = \text{the lesser of } \frac{P_{azu}}{C_{SF}} \text{ and } \frac{P_{ayu}}{K_{SF}} \quad (1)$$

where

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

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

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

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

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

$$P_S \leq P_a \quad (2)$$

where:

P_S is service power, kW.

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

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

The service factor includes the combined effects of overload, reliability, life, 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, that 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 Pitting resistance power rating

The pitting resistance of gear teeth is considered to be a Hertzian contact fatigue phenomenon. Initial pitting and destructive pitting are illustrated and discussed in ANSI/AGMA 1010-E95.

The purpose of the pitting resistance formula is to determine a load rating at which destructive 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.

The pitting resistance power rating for gearing within the scope of this standard shall be determined by the rating methods and procedures of ANSI/AGMA 2101-C95, clause 10, when using service factors, with the following values:

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

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

K_S is size factor, $K_S = 1.0$;

Z_R is surface condition factor, $Z_R = 1.0$;

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 , is calculated by the lower curve of figure 17 of ANSI/AGMA 2101-C95, and

should be based on 40 000 hours of service at rated operating speed.

$$Z_N = 2.466 n_L^{-0.056} \quad (3)$$

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), Z_N should be less than or equal to 0.68.

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

5.3.2 Load distribution factor, K_H

K_H is the load distribution factor. Values are to be per ANSI/AGMA 2101-C95. The following values shall be used with the empirical method:

K_{Hma} is mesh alignment factor. Use values from curve 3, precision enclosed gear units, of figure 7 and table 2 of ANSI/AGMA 2101-C95;

K_{Hmc} is lead 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 calculated value of K_H shall not be less than 1.1.

NOTE: The above empirical rating method assumes properly matched leads 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-A01 provides one such approach.

5.3.3 Dynamic factor, K_v

Dynamic factors account for internally generated gear tooth dynamic loads, which 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, which includes the dynamic effects.

$$K_v = \frac{F_d + F_t}{F_t} \quad (4)$$

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, which is defined as the departure from uniform relative angular motion of a pair of meshing gears. The transmission error is caused by:

- inherent variations in gear accuracy as manufactured;
- gear tooth deflections which 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 stiffnesses;
- damping characteristics of the rotor and bearing system.

The AGMA accuracy grades per ANSI/AGMA 2015-1-A01, specifically tooth element tolerances for pitch and profile, and the pitch line velocity may be used as parameters to guide the selection of dynamic factors. Within the 1.09 to 1.15 dynamic factor range, the trend is for K_v to vary in nearly a direct relationship with AGMA accuracy grades from A5 to A2 as shown in table 5.

Table 5 - Dynamic factor as a function of accuracy grade

ANSI/AGMA 2015-1-A01 accuracy grade	Dynamic factor, K_v
A5	1.15
A4	1.13
A3	1.11
A2	1.09

The dynamic factor, K_v , does not account for dynamic tooth loads which may 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 may be very high.

Refer to ANSI/AGMA 2101-C95 for additional considerations influencing dynamic factors.

5.4 Bending strength power rating

The bending strength of gear teeth is a measure of the resistance to fatigue cracking at the tooth root fillet.

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

The gear rim thickness must be sufficient for the calculated rim thickness factor to be 1.0.

Occasionally, manufacturing tool marks, wear, surface fatigue, or plastic flow may 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-C95, clause 10, when using service factors, with the following values:

- Y_θ is temperature factor, $Y_\theta = 1.0$;
- K_s is size factor, $K_s = 1.0$;
- K_B is rim thickness factor, $K_B = 1.0$;
- 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 the lower curve of figure 18 of ANSI/AGMA 2101-C95, and should be based on 40 000 hours of service at rated operating speed.

$$Y_N = 1.6831 n_L^{-0.0323} \quad (5)$$

where

n_L is number of stress cycles.

When the number of stress cycles exceeds 10^{10} , Y_N should be less than or equal to 0.80.

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

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

Allowable stress numbers, which are dependent upon material and processing, are given in ANSI/AGMA 2101-C95, clause 16. That clause also

specifies the treatment of momentary overload conditions.

Three grades of material have been established. Grade 1 is normal commercial quality steel and shall not be used for gears rated by this standard. Grade 2 is high quality steel meeting SAE/AMS 2301 cleanliness requirements. Grade 3 is premium quality steel meeting SAE/AMS 2300 cleanliness requirements. Both Grade 2 and Grade 3 are heat treated under carefully controlled conditions. The choice of material, hardness and grade is left to the gear designer; however, values of σ_{HP} and σ_{FP} shall be for grade 2 materials.

Due consideration should be given to additional testing, such as ultrasonic or magnetic particle inspection of high speed gear rotors which are subject to high fatigue cycles or high stress, or both, during operation.

For details on tooth failure, refer to ANSI/AGMA 1010-E95.

5.6 Reverse loading

For idler gears and other gears where the teeth are completely reverse loaded on every cycle, use 70 percent of the allowable bending stress number, σ_{FP} , in ANSI/AGMA 2101-C95.

5.7 Scuffing resistance

Scuffing failure (sometimes incorrectly referred to as scoring) has been known for many years and is a concern for high speed gear units. When high speed gears are subject to highly loaded conditions and high sliding velocities, the lubricant film may not adequately separate the surfaces. This localized damage to the tooth surface is referred to as "scuffing". Scuffing will exhibit itself as a dull matte or rough finish usually at the extreme end regions of the contact path or near the points of a single pair of teeth contact resulting in severe adhesive wear.

Scuffing is not a fatigue phenomenon and may occur instantaneously. The risk of scuffing damage varies with the material of the gear, lubricant being used, viscosity of the lubricant, surface roughness of the tooth flanks, sliding velocity of the mating gear set under load, and geometry of the gear teeth. Changes in any or all of these factors can reduce scuffing risk.

Further information is provided in annex B. Annex B is not a requirement of this standard. However, it is recommended that either annex B or some other

method be used to check for the probability of scuffing failure. See AGMA 925-A03 for further information.

6 Lubrication

6.1 Design parameters

High speed gear units shall be designed with a pressurized lubrication supply system to provide lubrication and cooling to the gears and bearings. A normal lubricant inlet pressure of 1 to 2 bar is an industry accepted value. Special applications may require other lubricant pressures. If a gear element extends below the lubricant level in the gear casing, it is said to be dipping in the lubricant. Dipping at high speed can result in high power losses, rapid overheating, possible fire hazard, and should be avoided.

The following minimum parameters should be considered to ensure that proper lubrication is provided for the gear unit:

- type of lubricant;
- lubricant viscosity;
- film thickness;
- surface roughness;
- inlet lubricant pressure;
- inlet lubricant temperature;
- filtration;
- drainage;
- retention or settling time;
- lubricant flow rate;
- cooling requirements.

6.2 Choice of lubricant

Certain lubricant additives, such as those in extreme pressure (EP) lubricants, may be removed by fine filtration. Changes to the level of filtration should only be done in consultation with both the gear unit and lubricant manufacturers. Extreme pressure lubricants are not normally used in high speed units.

To avoid dependency on extreme pressure additives, unless otherwise specified, the gear unit shall be designed for use with a lubricant that fails ISO 14635-1 load stage 6. The lubricant used shall pass ISO 14635-1 load stage 5. When an alternate

lubricant is requested, the vendor shall provide calculations and an experience list to support a request for an alternate lubricant selection.

6.2.1 Lubricant viscosity

Selection of an appropriate lubricant viscosity is a compromise of factors. In addition, lubrication systems are oftentimes integrated with other drive train equipment whose viscosity requirements are different from the gear unit. This complicates the selection of the lubricant.

Load carrying capacity of the lubricant film increases with the viscosity of the lubricant. Therefore, a higher viscosity is preferred at the gear mesh. Development of an adequate elastohydrodynamic lubricant film thickness and reduction in tooth roughness are of primary importance to the life of the gearset. However, in high speed gear units, particularly those with high bearing loads and high journal velocities, heat created in the bearings is considerable. Here, the viscosity must be low enough to permit adequate cooling of the bearings.

Lubricant viscosity recommendations are specified as ISO viscosity grades. Recommendations for high speed applications are listed in table 6. For turbine driven speed increasers where the lubrication system supplies both the bearings and the gear mesh, an ISO VG32 is usually provided for the gear drive. A lubricant with a viscosity index (VI) of 90 or better should be employed. Special considerations may require the use of lubricants not listed in table 6. The gear vendor should always be consulted when selecting or changing viscosity grades.

Table 6 - Recommended lubricants

ISO viscosity grade (VG)	Viscosity range mm ² /s (cSt) at 40°C	Minimum viscosity index (VI)
32	28.8 to 35.2	90
46	41.4 to 50.6	90
68	61.2 to 74.8	90
100	90.0 to 100.0	90

NOTE:

When operating at low ambient temperatures, the lubricant selected should have a pour point at least 6°C below the lowest expected ambient temperature.

6.2.2 Synthetic lubricants

Synthetic lubricants may be advantageous in some applications, especially where extremes of temperature are involved. There are many types of synthetic

lubricants, and some have distinct disadvantages. The gear vendor should be consulted before using any synthetic lubricant.

6.3 Lubrication considerations

6.3.1 Ambient temperature

Ambient temperature is defined as the temperature of the air in the immediate vicinity of the gear unit. The normal ambient temperature range for high speed gear unit operation is from -10°C to 55°C . The vendor should be informed what the ambient temperature will be, or if a large radiant heat source is located near the gear unit. Furthermore, if low ambient temperature causes the sump temperature to drop below 20°C at start-up, the vendor should be advised. Special procedures or equipment, such as heaters, may be required to ensure adequate lubrication.

6.3.2 Environment

If a gear unit is to be operated in an extremely humid, salt water, chemical, or dust laden atmosphere, the vendor must be advised. Special care must be taken to prevent lubricant contamination.

6.3.3 Temperature control

The lubricant temperature control system must be designed to maintain a lubricant inlet temperature within design limits at any expected ambient temperature or operating condition. Design inlet temperature may vary, but 50°C is a generally accepted value. Lubricant temperature rise through the gear unit should be limited to 30°C . Special operating conditions, such as high pitch line velocity, high inlet lubricant temperature, and high ambient temperature may result in higher operating temperatures.

6.3.4 Gear element cooling and lubrication

The size and location of the spray nozzles is critical to the cooling and proper lubrication of the gear mesh.

Spray nozzles may be positioned to supply lubricant at either the in-mesh, out-mesh, or both sides of the gear mesh (or at other points) at the discretion of the vendor.

6.3.5 Lubricant sump

The lubricant reservoir may be in the bottom of the gear case (wet sump) or in a separate tank (dry sump). In either case, the reservoir and/or gear case

should be sized, vented, and baffled to adequately deaerate the lubricant and control foaming. In dry sump applications, the external drainage system must be adequately sized, sloped and vented to avoid residual lubricant buildup in the gear case. Drain velocities may vary, but 0.3 meters per second in a half full opening is a generally accepted maximum value.

6.3.6 Filtration

A good filtering system for the lubricant is very important. The design filtration level may vary, but filtration to a 25 micron or finer nominal particle size is a generally accepted value. Filtration finer than 25 microns is recommended when light turbine lubricants are used, particularly for higher operating temperatures. ISO 4406 may be used as a more complete specification of the oil cleanliness required. An ISO 4406:1999 cleanliness level of 17/15/12 is recommended if there is no other recommendation from the gear unit manufacturer. To remove the finer particles, systems may be installed downstream of the filters. It has been found that removing very fine particles can greatly extend lubricant life. It is good practice to locate the filter as near as possible to the gear unit lubricant inlet. Further, it is recommended to provide a duplex filter to facilitate cleaning of the filter when the unit can not be conveniently shut down for filter change. Any kind of bypass of the filter is prohibited. A mechanism to indicate the cleanliness of the filter is recommended. Systems that take a portion of the filtered lubricant and further clean it are acceptable.

6.3.7 Drain lines

Location of the drain line must be in accordance with the vendor's recommendations. Drain lines should be sized so they are no more than half full. The lines should slope down at a minimum of 20 millimeters per meter and have a minimum number of bends and elbows.

6.4 Lubricant maintenance

The lubricant must be filtered and tested, or changed periodically, to assure that adequate lubricant properties are maintained.

Prior to initial start-up of the gear unit, the lubrication system should be thoroughly cleaned and flushed. It is recommended that the initial charge of lubricant be changed or tested after 500 hours of operation.

6.4.1 Change interval

Unless the vendor recommends different intervals, under normal operating conditions subsequent change or test intervals should be 2500 operating hours, or 6 months, whichever occurs first. Extended change periods may be established through periodic testing of lubricants. With periodic lubricant testing and conditioning, it is not uncommon to operate lubrication systems without lubricant changes for the life of the gear drive.

6.4.2 Water contamination

Where operating conditions result in water collecting in the lubrication system, the lubricant should be processed, or changed as required, to keep water content below the lubricant manufacturer's recommendation. Failure to control moisture may result in damage to the gear unit. Some lubricants are hygroscopic (absorb water) and may need special consideration to eliminate or control the water content and total acid number.

7 Vibration and sound

7.1 Vibration analysis

When the frequency of a periodic forcing phenomenon (exciting frequency) applied to a rotor-bearing support system coincides with a natural frequency of that system, the system may be in a state of resonance. A shaft rotational speed at which the rotor-bearing support system is in a state of resonance with any exciting frequency associated with that speed, is called a "critical speed".

Vibration of any component of the gear unit can result in additional dynamic loads being superimposed on the normal operating loads. Vibration of sufficient amplitude may result in impact loading of the gear teeth, interference in the gear mesh, or damage to close clearance parts of the gear unit. Where torque variations exceed 20 percent of the rated torque at the service power, the magnitude and frequency of such torque variations should be evaluated with regard to the endurance properties of the materials used.

The types of vibration which are generally of concern for gear units are the torsional, lateral and axial modes of the rotating elements, since these can have a direct influence on the tooth load. Of these,

the two that are normally reviewed analytically during design are the lateral critical speeds of the gear unit rotating shafts and the torsional critical frequencies of all connected rotating elements.

7.2 Torsional vibration analysis

Any torsional vibration analysis must consider the complete system including prime mover, gear unit, driven equipment and couplings. Dynamic loads imposed on a gear unit from torsional vibrations are the result of the dynamic behavior of the entire system and not the gear unit alone. Thus, a coupled system has to be analyzed in its entirety. A common method used is to separate the system into a series of discrete spring connected masses. When applied to a multi-mass system, this method is known as using lumped parameters. These parameters are developed into a model in order to analyze the system as a whole and solve its torsional mechanical vibrations.

It is important to note that this result is only as good as its model. In fact, the process of lumping parameters could be the largest source of errors. The result of the torsional system analysis is not within the control of the vendor, since the gear unit itself is only one of several elements in a coupled train.

The gear unit vendor is seldom the system designer and in normal cases the gear unit vendor is responsible only for providing mass elastic data. The system designer, not the gear vendor, is responsible for the torsional vibration analysis.

7.3 Lateral vibration analysis

The rating equations used in this standard assume smooth operation of the rotors. To insure smooth operation, these rotors should be analyzed for lateral critical speeds. It is imperative that slow roll, start-up, and shutdown of rotating equipment not cause any damage as the rotating elements pass through their critical speeds. See annex C.

7.3.1 Undamped lateral critical speed map

An undamped lateral critical speed analysis is sufficient in some cases to determine rotor suitability. If this method is chosen as the sole criterion for determining the suitability of a rotor, it should be based upon significant experience in designing high speed gear drives utilizing this method. It includes a lateral critical speed map, showing the undamped critical speeds versus support stiffness or percentage of torque load. This graphic display shows all