



American
Gear Manufacturers
Association

Technical Resources

ANSI/AGMA 6032-A94

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American National Standard

Standard for Marine Gear Units: Rating

ANSI/AGMA 6032-A94

ANSI/AGMA 6032-A94



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Standard for Marine Gear Units: Rating ANSI/AGMA 6032-A94

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American National Standards Institute, Inc.

Abstract

This document considers rating practices for marine main propulsion, power take-off and auxiliary propulsion service. Practical suggestions are included for various factors. Allowable contact stress numbers and allowable bending stress numbers for materials of this standard covered in ANSI/AGMA 6033-A88 are included.

This standard, ANSI/AGMA 6032-A94, provides the rating practice and recommended allowable according to type of the subject marine gearing.

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Foreword

[The foreword, footnotes, and annexes, if any, are provided for informational purposes only, and should not be construed as a part of American Gear Manufacturers Association 6032-A94, *Standard For Marine Gear Units: Rating*.]

The purpose of ANSI/AGMA 6032-A94 is to provide for rating marine reduction and reversing gears: driven by internal combustion engines or electric motors from 1500 to 20 000 horsepower, driven by steam or gas turbine from 1500 to 30 000 horsepower, power take-off (PTO) gearing that is integral to the propulsion unit, auxiliary propulsion gears and combinations of these drives. It does not cover separate power generation drives, pump set drives, conveyor drives or deck machinery. Nor does it cover the design and application of epicyclic drives.

This standard interprets ANSI/AGMA 2001-A88 for marine practice considering the successful practice of marine gear manufacturers and the existing American Bureau of Shipping Rules for Building and Classing Steel Vessels. These latter requirements are based on AGMA rating Standards 211 and 221 as they were published in the early 1970's.

There are six annexes in this standard. Annex A through Annex F are all informative and are not considered part of this standard.

This standard was approved by the AGMA membership in June, 1994 and as an American National Standard on August 23, 1994.

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

ERRATA June, 1999

The following editorial correction has been made to ANSI/AGMA 6032-A94, *Standard for Marine Gear Units: Rating*, (originally printed August, 1994). This change, discovered after publication, has been made in the second standard printing, as shown below:

<u>PAGE</u>	<u>ITEM</u>	<u>CHANGE</u>
17	Equation 10.5	The change of sign in the third term would bring the equation into agreement with the plot of Figure 2 and with Equation 40 of ANSI/AGMA 2001-C95. The equation should read as follows: $C_{pf} = \frac{F}{10 d} - 0.1109 + 0.0207F - 2.28 \times 10^{-4}F^2$

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American National Standard – Standard for Marine Gear Units: Rating

1 Scope

1.1 Application

This standard is applicable to marine reduction and reversing gears as follows:

- marine propulsion reduction gears driven by internal combustion engines or electric motors from 1500 to 20 000 horsepower (1120 to 14 900 kW) per prime mover, when the rotor speeds do not exceed 3600 rpm.
- marine propulsion reduction gears driven by steam or gas turbine from 1500 to 30 000 horsepower (1120 to 22 370 kW) per prime mover.
- power take-off (PTO) gearing that is integral to the propulsion unit.
- auxiliary propulsion gears.
- combination of drives listed above.

The fundamental rating formulas are applicable for rating the pitting resistance and bending strength of external spur, helical, or double helical involute gear teeth. The formulas evaluate gear tooth capacity as influenced by the major factors which affect gear tooth pitting and gear tooth fracture.

This standard also addresses bearings, clutches, lubricating oil systems, shafts and certain aspects of vibrations.

1.2 Intended use

This standard provides a method by which different gear designs can be compared. It is intended for use by the experienced gear designer capable of selecting reasonable values for the factors, based on his knowledge of performance of similar designs and the effects of such items as lubrication, deflection, manufacturing tolerances, metallurgy, residual stress and system dynamics. It is not intended to assure performance of assembled gear drive systems. It is not intended for use by the engineering public at large.

The majority of marine gears are of helical or double helical tooth design. Spur gear tooth designs may also be used, but are generally not, except for power take off (PTO) drives and reversing sections of marine gear units.

This standard assumes adequate support and alignment for the gearing.

1.3 Exceptions

This standard does not cover separate power generation drives, pump set drives, conveyor drives or deck machinery. Nor does it cover the design and application of epicyclic drives or gear blank design. It is beyond the scope of this standard to present a detailed analysis of efficiency.

This standard does not cover the rating of gear drives due to the wear or scoring (scuffing) of gear teeth or components. See ANSI/AGMA 2001-B88, appendix A, for a consideration of scuffing.

1.4 Marine gear performance parameter sheet

It is recognized that marine units may be rated in different manners for the same service. To assist in determining the actual applicability of a gear to its intended service, annex C is included as a marine gear performance parameter sheet.

1.5 Annexes

The annexes are for reference only and are not a part of this standard. The annexes can be used to make a more detailed analysis of certain rating factors.

1.6 References

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

AGMA 908-B89, *Information Sheet – Geometry Factors for Determining the Strength of Spur, Helical, Herringbone and Bevel Gear Teeth*.

ANSI/AGMA 2001-B88, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*.

ANSI/AGMA 6001-C88, *Design and Selection of Components for Enclosed Gear Drives*.

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

2 Definitions and symbols

2.1 Definitions

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

ANSI Y10.3-1968, *Letter Symbols for Quantities Used in Mechanics of Solids*.

AGMA 112.05, *Gear Nomenclature, Terms, Definitions, Symbols, and Abbreviations*.

AGMA 904-B89, *Standard for Metric Usage*.

2.2 Symbols

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

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

SI units of measure are shown in parentheses in table 1 and in the text. Where equations require a different format or constant for use with SI units, a second expression is shown after the first, indented, in smaller type, and with *M* included in the equation number.

Example

$$U_L = \frac{W_t P_{nd}}{F} \quad \dots(\text{Eq 5.12})$$

$$U_L = \frac{F_t}{b m_n} \quad \dots(\text{Eq 5.12M})$$

Table 1 - Symbols used in gear rating equations

Symbol		Description	Units	First used	Ref. para
AGMA	ISO				
A_h	C_{Ah}	Variable used in the hardness ratio factor	--	Eq 12.1	12.1
A_v	A	Variable used in the dynamic factor calculation	--	Eq 8.2	8.3.2
B_H	C_{Bh}	Variable used in the hardness ratio factor	--	Eq 12.6	12.2
B_v	B	Variable used in the dynamic factor calculation	--	Eq 8.2	8.3.2
b	a_b	Shaft type factor	--	Eq 10.11	10.3
C	a	Center distance	in (mm)	Eq 5.2	5.1.1
C_a	K_A	Application factor for pitting resistance	--	Eq 5.1	9.1
C_c	C_c	Curvature factor at pitch line	--	Eq 6.1	6.1
C_e	K_e	Mesh alignment correction factor	--	Eq 10.2	10.2.1
C_f	Z_R	Surface condition factor for pitting resistance	--	Eq 5.1	14.2
C_G	C_u	Gear ratio factor	--	Eq 11.1	11.2
C_H	Z_W	Hardness ratio factor for pitting resistance	--	Eq 5.5	12.1
C_L	Z_N	Life factor for pitting resistance	--	Eq 5.5	13
C_m	K_H	Load distribution factor for pitting resistance	--	Eq 5.1	10.1
C_{ma}	K_{ma}	Mesh alignment factor	--	Eq 10.2	10.2.1
C_{mc}	K_{mc}	Lead correction factor	--	Eq 10.2	10.2.1
C_{mf}	K_{mf}	Total load distribution factor	--	Eq 10.2	10.2.1
C_p	Z_E	Elastic coefficient	$[\text{lb/in}^2]^{0.5} ([\text{N/mm}^2]^{0.5})$	Eq 5.1	5.1.1
C_{pf}	K_{pf}	Pinion proportion factor	--	Eq 10.2	10.2.1
C_{pm}	K_{pm}	Pinion proportion modifier	--	Eq 10.2	10.2.1
C_R	Z_z	Reliability factor for pitting resistance	--	Eq 5.5	5.1.2
C_s	C_s	Size factor for pitting resistance	--	Eq 5.1	5.1.1
C_T	Z_θ	Temperature factor for pitting resistance	--	Eq 5.5	5.1.2
C_v	K_v	Dynamic factor for pitting resistance	--	Eq 5.1	8.1
C_x	Z_h	Contact height factor	--	Eq 6.1	6.1

(continued)

Table 1 (continued)

Symbol		Description	Units	First used	Ref. para
AGMA	ISO				
C_ψ	Z_β	Helical overlap factor	--	Eq 6.1	6.1
d	d_1	Pinion operating pitch diameter	in (mm)	Eq 5.1	5.1.1
d_i	d_{B1}	Pinion bore diameter	in (mm)	Eq 10.11	10.3
d_{is}	d_{shi}	Shaft inside diameter	in (mm)	Eq 18.6	18.1.2
d_o	d_{K1}	Outside diameter of the pinion	in (mm)	Eq 10.11	10.3
d_{os}	d_e	Shaft outside diameter	in (mm)	Eq 18.1	18.1
F	b	Net face width of narrowest member	in (mm)	Eq 5.1	5.1.1
F_s	S_s	Shaft factor of safety	--	Eq 18.1	18.1.1
f_p	R_{a1}	Pinion surface finish	$\mu\text{in } (\mu\text{m})$	Eq 12.7	12.2
H_B	H_B	Brinell hardness	--	Fig 5	11.1
H_{BG}	H_{B2}	Brinell hardness of gear	--	Eq 12.2	12.1
H_{BP}	H_{B1}	Brinell hardness of pinion	--	Eq 12.2	12.1
h_c	Eht_{nt}	Total case depth for nitrided gears	in (mm)	Eq 11.4	11.2
h_e	$Eht_{\min}$	Minimum effective case depth	in (mm)	Eq 11.1	11.2
$h_{e \max}$	$Eht_{\max}$	Maximum effective case depth	in (mm)	Eq 11.3	11.2
h_t	h_t	Gear tooth whole depth	in (mm)	Eq 5.14	5.2.4
I	Z_I	Geometry factor for pitting resistance	--	Eq 5.1	6.1
J	Y_J	Geometry factor for bending strength	--	Eq 5.8	6.2
K	Z_c	Contact load factor for pitting resistance	lb/in ² (N/mm ²)	Eq 5.4	5.1.1
K_a	K_A	Application factor for bending strength	--	Eq 5.8	9.1
K_{ac}	Z_{ac}	Allowable contact load factor	lb/in ² (N/mm ²)	Eq 5.6	5.1.2
K_B	Y_{SR}	Rim thickness factor	--	Eq 5.8	5.2.4
K_b	K_b	Theoretical stress concentration factor (bending)	--	Eq 18.2	18.1.1
K_f	Y_s	Stress correction factor	--	Eq 6.2	6.2
K_{fb}	K_{fb}	Fatigue stress concentration factor in bending	--	18.1	18.1.1
K_{ft}	K_{ft}	Fatigue stress concentration factor in torsion	--	18.1	18.1.1
K_L	Y_N	Life factor for bending strength	--	Eq 5.11	13
K_m	K_H	Load distribution factor for bending strength	--	Eq 5.8	10.1
K_R	Y_Z	Reliability factor for bending strength	--	Eq 5.11	5.2.2
K_S	Y_x	Size factor for bending strength	--	Eq 5.8	5.2.1
K_T	Y_θ	Temperature factor for bending strength	--	Eq 5.11	5.2.2
K_{ts}	K_{ts}	Theoretical stress concentration factor (torsion)	--	Eq 18.3	18.1.1
K_v	K_v	Dynamic factor for bending strength	--	Eq 5.8	8.1
k	k	Shaft fatigue limit modification factor	--	Eq 18.9	18.2.3
k_a	k_a	Shaft surface finish factor	--	Eq 18.9	18.2.4
k_b	k_b	Size factor	--	Eq 18.9	18.2.5
k_c	k_c	Reliability factor	--	Eq 18.9	18.2.6
k_d	k_d	Temperature factor	--	Eq 18.9	18.2.7
k_e	k_e	Life factor	--	Eq 18.9	18.2.8
k_g	k_g	Miscellaneous factor	--	Eq 18.9	18.2.9
L	B_L	Overall face width including gap for double helical gears	in (mm)	Eq 10.7	10.3
M_a	M_a	Alternating reverse bending moment	lb in (Nm)	Eq 18.1	18.1.1
M_m	M_m	Steady state shaft bending moment	lb in (Nm)	Eq 18.1	18.1.1
m	m	Metric module, nominal in plane of rotation	(mm)	Eq 5.8M	5.2.1

(continued)

Table 1 (continued)

Symbol		Description	Units	First used	Ref. para
AGMA	ISO				
m_B	u_B	Back-up ratio	--	Eq 5.14	5.2.4
m_G	u	Gear ratio (always > 1.0)	--	Eq 5.2	5.1.1
m_N	ε_N	Load sharing ratio	--	Eq 6.1	6.1
--	m_n	Normal metric module, nominal	(mm)	Eq 5.9M	5.2.1
N	N	Number of cycles	--	Fig 12	13
N_m	N_m	Number of meshes	--	Eq 10.9	10.3
n	n	RPM at rated speed	rpm	Eq 14.1	14.3
n_p	n_1	Pinion speed	rpm	Eq 7.1	7.1
P	P	Transmitted power per mesh	hp (kW)	Eq 7.1	7.1
P_d	--	Diametral pitch, nominal, in plane of rotation	in ⁻¹	Eq 5.8	5.2.1
P_{nd}	--	Normal diametral pitch, nominal	in ⁻¹	Eq 5.9	5.2.1
p_x	--	Axial pitch	in (mm)	Eq 5.9	5.2.1
Q_v	Q	Quality number	--	Eq 8.4	8.3.2
q_b	q_b	Bending notch sensitivity factor	--	Eq 18.2	18.1
q_t	q_t	Torsion notch sensitivity factor	--	Eq 18.3	18.1
R_a	R_a	Average surface roughness	μ in (μ m)	Table 7	14.2
S	S	Bearing span	in (mm)	Fig 3	10.2
S_F	S_F	Safety factor for tooth bending	--	Eq 5.11	9.3
S_f	σ_f	Shaft bending fatigue (endurance) limit	lb/in ² (N/mm ²)	Eq 18.1	18.1.1
S_{fe}	σ_{fe}	Shaft fatigue (endurance) limit of polished, unnotched test specimen in reverse bending	lb/in ² (N/mm ²)	Eq 18.7	18.2.2
S_H	S_H	Safety factor for tooth contact	--	Eq 5.5	9.3
S_u	σ_u	Shaft ultimate tensile strength	lb/in ² (N/mm ²)	Eq 18.8	18.2.1
S_y	σ_y	Shaft tensile yield strength	lb/in ² (N/mm ²)	Eq 18.1	18.2.1
S_1	S_1	Offset of pinion	in (mm)	Fig 3	10.2
s_{ac}	σ_{HP}	Allowable contact stress number	lb/in ² (N/mm ²)	Eq 5.5	11.1
s_{at}	σ_{FP}	Allowable bending stress number	lb/in ² (N/mm ²)	Eq 5.11	11.1
s_{ay}	σ_{ay}	Allowable yield stress number	lb/in ² (N/mm ²)	Eq 11.5	11.5
s_c	σ_H	Contact stress number	lb/in ² (N/mm ²)	Eq 5.1	5.1.1
s_t	σ_F	Bending stress number	lb/in ² (N/mm ²)	Eq 5.8	5.2.1
T	T_1	Transmitted pinion torque	lb in (Nm)	Eq 7.1	7.1
T_a	T_a	Alternating shaft torque	lb in (Nm)	Eq 18.1	18.1.1
T_m	T_m	Mean pinion torque at operating speed	lb in (Nm)	Eq 9.1	9.2
T_{ms}	T_{ms}	Mean shaft torque	lb in (Nm)	Eq 18.1	18.1.1
T_R	T_R	Rated pinion torque at operating speed	lb in (Nm)	Eq 9.1	9.2
T_v	T_v	Vibrating component of torque	lb in (Nm)	Eq 9.1	9.2
t_o	s_a	Normal tooth thickness at the top land of gear	in (mm)	Eq 11.3	11.2
t_R	s_R	Gear rim thickness	in (mm)	Eq 5.14	5.2.4
U	U_{max}	Maximum allowable residual unbalance	oz in (g mm)	Eq 14.1	14.3
U_{at}	U_{at}	Allowable unit load for bending strength	lb/in ² (N/mm ²)	Eq 5.13	5.2.3
U_c	U_c	Core hardness coefficient	--	Eq 11.4	11.2
U_H	U_H	Hardening process factor	--	Eq 11.1	11.2
U_L	U_L	Unit load for bending strength	lb/in ² (N/mm ²)	Eq 5.12	5.2.3
v_t	ω_t	Pitch line velocity	ft/min (m/s)	Eq 7.1	7.1
$v_{t \max}$	$\omega_{t \max}$	Pitch line velocity maximum	ft/min (m/s)	Eq 8.5	8.3.2

(continued)

Table 1 (concluded)

Symbol		Description	Units	First used	Ref. para
AGMA	ISO				
W	m_r	Weight of rotating part	lb (N)	Eq 14.1	14.3
W_d	F_{dyn}	Incremental dynamic tooth load	lb (N)	Eq 8.1	8.1
$W_{t\ max}$	$F_{t\ max}$	Maximum peak tangential load	lb (N)	Eq 11.5	11.5
W_t	F_t	Tangential load	lb (N)	Eq 5.1	7
Y	Y	Tooth form factor	- -	Eq 6.2	6.2
Φ_t	α_{wt}	Operating transverse pressure angle	- -	Eq 11.1	11.2
δ	δ_1	Pinion elastic deflection at pitch cylinder	in (mm)	Eq 10.7	10.3
ψ_s	β	Helix angle at standard pitch diameter	- -	Eq 5.9	5.2.1
ψ_b	β_b	Base helix angle	- -	Eq 11.1	11.2

3 Application

3.1 System responsibility

The successful performance of a marine reduction gear requires a proper complete design of the system in which it operates. To ensure a successful gear design, it is necessary that the complete propulsion system be properly tuned so that the system is free of harmful vibratory forces, properly manufactured, properly aligned, properly supported, and properly operated. To ensure this, clear designation of responsibility and clear communication between the owner, shipbuilder, designer, and machinery supplier is a necessity.

The party having contractual responsibility for the system analysis of the unit should be clearly identified in the specification, contract or purchase order. Substantial costs are involved in a system analysis. All parties supplying components to the system have the responsibility to furnish correct and accurate data so that the analysis is meaningful.

3.2 Design basis

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

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

3.3 Manufacturing tolerances

Rating factors should be evaluated on the basis of the expected variation of component parts in the production run.

3.4 Implied accuracy

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

3.5 Other considerations

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

3.5.1 Misalignment and deflection of foundations

Many gear systems depend on external supports such as machinery foundations to maintain alignment of the gear mesh. If these supports are not properly aligned or deviate from specified tolerances during operation through elastic or thermal deflection, overall reduction gear unit(s) performance will be adversely affected.

The factors in this standard assume proper support of the gear unit. If proper support is not provided, the factors in this standard are not applicable.

3.5.2 Deflection due to external forces

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

3.5.3 Metallurgy

The allowable stress numbers, s_{ac} (σ_{HP}) and s_{at} (σ_{FP}), included herein are a function of melting, casting, forging and heat treating practice. Hardness, tensile strength and cleanliness are some of the criteria for determining allowable stress numbers.

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

Material recommendations may be found in ANSI/AGMA 6033-A88.

3.5.4 Residual stress

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

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

Shot peening is often performed to assure the presence of residual compressive stresses. Care must be taken to avoid unnecessarily reducing or eliminating such compressive stresses by subsequent machining, grinding, or thermal treatment.

3.5.5 Lubrication

The ratings determined by these formulas are only valid when the gear teeth are operated with a

lubricant of proper viscosity for the load, gear tooth surface finish, temperature, and pitch line velocity. This standard does not refer to scuffing or other forms of wear. These phenomena should be evaluated and reference is made to appendix A of ANSI/AGMA 2001-B88.

For further details on marine reduction gear lubrication see clause 16.

3.5.6 System dynamics

The dynamic response of the system results in additional gear tooth loads due to the relative motions of the connected masses of the driver and the driven equipment. The application factors C_a and K_a (K_A) are intended to account for those additional tooth loads due to dynamic and steady state operating characteristics of the driving and driven equipment.

Torsional, axial and transverse vibrations of the system must be considered to assure satisfactory performance of marine gears. The natural frequencies and amplitudes of vibration depend on the mass and elastic distributions and damping for the total power train. Operating roughness such as the firing of the engine cylinders, propeller blades passing through a varying flow field, damaged or fouled propellers, rotational unbalance and gear tooth meshing variations produce pulsating excitation forces which may sustain vibrations of the system. If the value of the excitation frequency is near the value of a natural frequency, resonance will occur, which may cause overloads several times higher than the nominal load on the system.

An analysis by the party having contractual responsibility must include the total system of driver, gearbox, driven equipment, couplings, mounting conditions, and sources of excitation. Natural frequencies, node shapes, and the dynamic response amplitudes should be calculated. For more information, refer to AGMA 427, *Information Sheet - Systems Considerations for Critical Service Gear Drives*.

The steady state propeller loading may vary due to such things as hull fouling, draft change and shallow bottoms, when the propeller speed is maintained constant. This additional loading should be supplied by the shipyard, naval architect or purchaser.

The party with contractual responsibility for the system analysis (see 3.1) will obtain data from the

component manufacturers to define the mass, elastic, damping and excitation characteristics of the system. Annex D indicates the type of data required of each component manufacturer.

3.5.7 Corrosion

Corrosion of gear unit components and of the gear tooth surface can have a significant detrimental effect on the life and pitting resistance of the teeth and bearings. Operational procedures such as dehumidification or lube oil centrifuging should be considered to minimize corrosion. Quantification of the effect of corrosion on gear teeth is beyond the scope of this standard.

3.5.8 Cold temperature operation

If gears are to be operated below 32°F (0°C), special care must be given to select materials which will have adequate impact properties at the operating temperature. Consideration should be given to:

- low temperature Charpy specification;
- fracture appearance transition or nil ductility temperature specification;
- reduction of carbon content in base material to less than 0.4 percent carbon;
- the use of higher nickel alloy steels.

In these instances, gear unit preheating is often provided for marine units.

4 Criteria for tooth capacity

4.1 Relationship of pitting resistance and bending strength ratings

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

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

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

4.2 Pitting resistance

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

For through hardened gears, corrective and non-progressive initial pitting is not deemed serious. Initial pitting is characterized by small pits which do not extend over the entire face width or profile height of the affected teeth. Initial pitting occurs in localized, overstressed areas. It tends to redistribute the load by progressively removing high contact spots. Generally, when the localized load has been reduced or redistributed, the pitting stops.

For surface hardened gears, destructive pitting is not acceptable. Micropitting or frosting is not considered detrimental to the gear, if it does not progress.

The aim 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. The ratings for pitting resistance are based on the formulas developed by Hertz for contact pressure between two curved surfaces, modified for the effect of load sharing between adjacent teeth.

4.3 Bending strength

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

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

The user should satisfy himself that the gear blank construction is representative of the basic theory

embodied in this standard (see 5.2.4). Gear blank design is beyond the scope of this standard.

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

- the compressive stress at tooth roots caused by the radial component of tooth loading;
- non-uniform moment distribution resulting from the inclined angle of the load lines on the teeth;
- stress concentrations at the tooth root fillets;
- the load sharing between adjacent teeth in contact.

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

Occasionally, 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.

5 Rating formula

5.1 Pitting resistance

5.1.1 Fundamental formula

The fundamental pitting resistance formula for gear teeth is:

$$s_c = C_p \sqrt{\frac{W_t C_a C_s C_m C_f}{C_v d F I}} \quad \dots(\text{Eq 5.1})$$

$$\sigma_H = Z_E \sqrt{\frac{F_t K_A C_s K_H Z_R}{K_v d_1 b Z_I}} \quad \dots(\text{Eq 5.1M})$$

where

NOTE: The symbols for SI equations are given in ().

$s_c (\sigma_H)$ is contact stress number, lb/in² (N/mm²);

$C_p (Z_E)$ is elastic coefficient, (lb/in²)^{0.5} [N/mm²]^{0.5} taken as 2300 (191) for steel on steel gears. For other materials, see ANSI/AGMA 2001-B88;

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

$C_a (K_A)$ is application factor for pitting resistance (see clause 9);

$C_s (C_s)$ is size factor for pitting resistance normally taken as unity for marine gears;

$C_m (K_H)$ is load distribution factor for pitting resistance (see clause 10);

$C_f (Z_R)$ is surface condition factor for pitting resistance normally taken as unity for marine gears (see 14.2);

$C_v (K_v)$ is dynamic factor for pitting resistance (see clause 8);

$F (b)$ is net face width of narrowest member, in (mm);

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

$d (d_1)$ is operating pitch diameter of pinion, in (mm).

$$d = \frac{2C}{m_G + 1.0} \quad \dots(\text{Eq 5.2})$$

$$d_1 = \frac{2a}{u + 1.0} \quad \dots(\text{Eq 5.2M})$$

where

$C (a)$ is operating center distance, in (mm);

$m_G (u)$ is gear ratio (never less than 1.0).

For conventional steel marine gears the pitting resistance formula can be expressed as

$$s_c = 2300 \sqrt{\frac{W_t C_a}{C_v} \frac{1}{d F} \frac{C_m}{I}} \quad \dots(\text{Eq 5.3})$$

$$\sigma_H = 191 \sqrt{\frac{F_t K_A}{K_v} \frac{1}{d_1 b} \frac{K_H}{Z_I}} \quad \dots(\text{Eq 5.3M})$$

Traditional large marine gears have used K factors for pitting:

$$K = \frac{W_t}{d F} \frac{m_G + 1}{m_G} \quad \dots(\text{Eq 5.4})$$

$$Z_c = \frac{F_t}{d_1 b} \frac{u + 1}{u} \quad \dots(\text{Eq 5.4M})$$

where

$K (Z_c)$ is contact load factor, lb/in² (N/mm²).

5.1.2 Allowable contact stress number

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

$$s_c \leq \frac{s_{ac} C_L C_H}{S_H C_T C_R} \quad \dots(\text{Eq 5.5})$$

$$\sigma_H \leq \frac{\sigma_{HP} Z_N Z_W}{S_H Z_0 Z_z} \quad \dots(\text{Eq 5.5M})$$

where

$s_{ac} (\sigma_{HP})$ is allowable contact stress number, lb/in² (N/mm²) (see clause 11);

$C_L (Z_N)$ is life factor for pitting resistance (see clause 13);

$C_H (Z_W)$ is hardness ratio factor (see clause 12);

$C_R (Z_z)$ is reliability factor for pitting resistance, taken as unity for marine gears;

$C_T (Z_\theta)$ is temperature factor for pitting resistance. As blank temperatures are not expected to exceed 200°F (93°C), this is taken as unity for marine gears;

$S_H (S_H)$ is a safety factor for tooth contact (see clause 9.3).

In terms of coefficient K :

$$K \leq \frac{K_{ac}}{S_H} \quad \dots(\text{Eq 5.6})$$

$$Z_c = \frac{Z_{ac}}{S_H} \quad \dots(\text{Eq 5.6M})$$

and

$$K_{ac} = I \left(\frac{m_G + 1}{m_G} \right) \left(\frac{C_v}{C_a C_s C_m C_F} \right) \left(\frac{s_{ac} C_L C_H}{C_p C_T C_R} \right)^2 \quad \dots(\text{Eq 5.7})$$

$$Z_{ac} = Z_I \left(\frac{u + 1}{u} \right) \left(\frac{K_v}{K_A C_s C_m Z_R} \right) \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Z_\theta Z_z} \right)^2 \quad \dots(\text{Eq 5.7M})$$

where

$K_{ac} (Z_{ac})$ is allowable contact load factor, lb/in² (N/mm²).

Further information appears in annex A.

5.2 Bending strength

5.2.1 Fundamental formula

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

$$s_t = \frac{W_t K_a}{K_v} \frac{P_d}{F} \frac{K_s K_m K_B}{J} \quad \dots(\text{Eq 5.8})$$

$$\sigma_F = \frac{F_t K_A}{K_v} \frac{1}{b m} \frac{Y_X K_H Y_{SR}}{Y_J} \quad \dots(\text{Eq 5.8M})$$

where

$s_t (\sigma_F)$ is bending stress number, lb/in² (N/mm²);

$K_a (K_A)$ is application factor for bending strength (see clause 9);

$K_B (Y_{SR})$ is rim thickness factor, normally taken as unity for standard marine gears (see 5.2.4);

$K_s (Y_X)$ is size factor for bending strength, normally taken as unity for standard marine gears;

$K_m (K_H)$ is load distribution factor for bending strength (see clause 10);

$K_v (K_v)$ is dynamic factor for bending strength (see clause 8);

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

m is nominal metric module in plane of rotation, mm;

P_d is nominal diametral pitch in plane of rotation*, inch⁻¹;

$P_d = P_{nd}$ for spur gears;

$m = m_n$ for spur gears.

$P_d = \frac{\pi}{p_x \tan \psi_s} = P_{nd} \cos \psi_s$ for helical gears
... (Eq 5.9)

$m = \frac{p_x \tan \beta}{\pi} = \frac{m_n}{\cos \beta}$ for helical gears
... (Eq 5.9M)

where

P_{nd} is normal diametral pitch (nominal), inch⁻¹;

m_n is normal metric module (nominal), mm;

p_x is axial pitch, in (mm);

$\psi_s (\beta)$ is helix angle at standard pitch diameter.

$$\psi_s = \sin^{-1} \left(\frac{\pi}{p_x P_{nd}} \right) \quad \dots(\text{Eq 5.10})$$

$$\beta = \sin^{-1} \left(\frac{\pi m_n}{p_x} \right) \quad \dots(\text{Eq 5.10M})$$

5.2.2 Allowable bending stress number

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

$$s_t \leq \frac{s_{at} K_L}{S_F K_T K_R} \quad \dots(\text{Eq 5.11})$$

$$\sigma_F \leq \frac{\sigma_{FP} Y_N}{S_F Y_\theta Y_Z} \quad \dots(\text{Eq 5.11M})$$

where

$s_{at} (\sigma_{FP})$ is allowable bending stress number, lb/in² (N/mm²) (see clause 11);

$K_L (Y_N)$ is life factor for bending strength (see clause 13);

$K_T (Y_\theta)$ is temperature factor for bending strength. As blank temperatures are not expected to exceed 200°F (93°C), this is taken as unity for marine gears;

$K_R (Y_Z)$ is reliability factor for bending strength, taken as unity for marine gears;

$S_F (S_F)$ is a safety factor for tooth bending (see 9.3).

* This calculation is based on standard gear hobbing practice, with P_d and p_x given. For a detailed text on geometry, see ANSI/AGMA 115.01, *Reference Information-Basic Gear Geometry*.

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

lowest value of the product of $\frac{s_{at} K_L J}{K_B}$ for each of the mating gears.

5.2.3 Unit load, U_L

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

$$U_L = \frac{W_t P_{nd}}{F} \quad \dots(\text{Eq 5.12})$$

$$U_L = \frac{F_t}{b m_n} \quad \dots(\text{Eq 5.12M})$$

where

U_L is unit load, lb/in² (N/mm²).

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

$$U_{at} = \frac{J}{\cos \psi_s} \frac{K_v}{K_a K_m K_s K_B} \frac{s_{at} K_L}{K_R K_T} \frac{1}{S_F} \quad \dots(\text{Eq 5.13})$$

$$U_{at} = \frac{Y_J}{\cos \beta} \frac{K_v}{K_A K_H Y_x Y_{SR}} \frac{\sigma_{FP} Y_N}{Y_Z Y_\theta} \frac{1}{S_F} \quad \dots(\text{Eq 5.13M})$$

where

U_{at} (U_{at}) is allowable unit load, lb/in² (N/mm²).

The allowable unit load, U_{at} , is the lowest of the ratings calculated using the different values of s_{at} , K_B , K_L , and J for pinion and gear.

$$U_L \leq U_{at} \quad \dots(\text{Eq 5.14})$$

5.2.4 Rim thickness factor, K_B (Y_{SR})

Where the minimum rim thickness is not sufficient to provide full support for the tooth root, the location of bending fatigue failure may be through the gear rim, rather than at the root fillet. Published data [6] suggest the use of a stress modifying factor, K_B , in this case.

The rim thickness factor, K_B , adjusts the calculated bending stress number for thin rimmed gears. It is a function of the backup ratio, m_B (u_B). See appendix C of ANSI/AGMA 2001-B88.

$$m_B = \frac{t_R}{h_t} \quad \dots(\text{Eq 5.15})$$

$$u_B = \frac{s_R}{h_t} \quad \dots(\text{Eq 5.15M})$$

where

h_t is whole depth, in (mm);

t_R (s_R) is minimum rim thickness, in (mm).

The effects of webs and stiffeners are ignored. The effect of tapered rims has not been investigated. When previous experience justifies, lower values of K_B (Y_{SR}) may be used.

For marine gears, backup ratios of 1.5 have been used, but 2.0 is recommended and K_B (Y_{SR}) is taken as unity. Where thinner rims are used, detailed analysis is recommended.

6 Geometry factors, I (Z_I) and J (Y_J)

6.1 Pitting resistance geometry factor

The geometry factor I , evaluates the radii of curvature of the contacting tooth profiles based on tooth geometry. These radii are used to evaluate the Hertzian contact stress in the tooth flank. Effects of modified tooth proportions and load sharing are considered. AGMA 908-B89 includes complete information and tables of geometry factors, I (Z_I), for some common tooth forms.

The geometry factor, I , is defined as:

$$I = \frac{C_c C_x C_\psi^2}{m_N} \quad \dots(\text{Eq 6.1})$$

$$Z_I = \frac{C_c Z_h Z_\beta^2}{\epsilon_N} \quad \dots(\text{Eq 6.1M})$$

where

C_c (C_c) is curvature factor at pitch line;

C_x (Z_h) is contact height factor;

C_ψ (Z_β) is helical overlap factor;

m_N (ϵ_N) is load sharing ratio.

For further discussion on marine gear units, see annex A.

6.2 Bending strength geometry factor, J (Y_J)

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

In AGMA 908-B89, the geometry factor, J (Y_J), is defined as:

$$J = \frac{Y C_\psi}{K_f m_N} \quad \dots(\text{Eq 6.2})$$

$$Y_J = \frac{Y Z_\beta}{Y_s \epsilon_N} \quad \dots(\text{Eq 6.2M})$$

where

Y (Y) is tooth form factor;

K_f (Y_s) is stress correction factor.

6.3 Analysis method

For involute gears with generated root fillets, the analytical method of AGMA 908 is recommended.

7 Transmitted tangential load, W_t (F_t)

7.1 Uniform load

The transmitted tangential load calculation is based on the nominal ratings of the prime mover or the driven equipment, and is assumed to be uniform.

The rating is calculated on the basis of uniform load. The transmitted uniform tangential load is:

$$W_t = \frac{33\,000\,P}{v_t} = \frac{2\,T}{d} = \frac{126\,050\,P}{n_p\,d} \quad \dots(\text{Eq 7.1})$$

$$F_t = \frac{1000\,P}{\omega_t} = \frac{2\,T_1}{d_1} = \frac{1.91 \times 10^7\,P}{n_1\,d_1} \quad \dots(\text{Eq 7.1M})$$

where

W_t (F_t) is transmitted tangential load per mesh, lb (N);

v_t (ω_t) is pitch line velocity at operating pitch diameter, ft/min (m/s).

$$v_t = \frac{\pi\,n_p\,d}{12} \quad \dots(\text{Eq 7.2})$$

$$\omega_{t\max} = \frac{\pi\,n_1\,d_1}{60\,000} \quad \dots(\text{Eq 7.2M})$$

P (P) is transmitted power per mesh, hp (kW);

T (T_1) is transmitted pinion torque per mesh, lb in (Nm);

n_p (n_1) is pinion speed, rpm.

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

Application factors, C_a and K_a (K_A) (see clause 9) and dynamic factors, C_v and K_v (K_v) (see clause 8), are included in the rating formulas (see clause 5) to account for loads in excess of W_t (F_t).

In most gear applications the transmitted tangential load will vary to obtain values of the operating tangential loads the designer normally applies.

7.2 Non-uniform load

When the transmitted load is not uniform, consideration should be given not only to the peak

load and its anticipated number of cycles, but also to intermediate loads and their numbers of cycles. This type of load is often considered a *duty cycle* and may be represented by a load spectrum. In such cases, the cumulative fatigue effect of the duty cycle is considered in rating the gear set.

A method of calculating the effect of loads under these conditions is given in annex B.

7.3 Double helical gears

For double helical gears it is assumed that the total tangential force is evenly distributed between the two helices. If this is not the case unequal load distribution between the two helices exists. Therefore, the two helices should be treated as two separate helical gears in parallel.

8 Dynamic factors, C_v and K_v

8.1 Dynamic Factors

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

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

$$C_v = K_v = \frac{W_t}{W_d + W_t} \quad \dots(\text{Eq 8.1})$$

$$K_v = \frac{F_t}{F_{\text{dyn}} + F_t} \quad \dots(\text{Eq 8.1M})$$

where

W_d (F_{dyn}) is incremental dynamic tooth load due to the dynamic response of the gear pair to the transmission error excitation, not including the transmitted tangential load, lb (N).

8.1.1 Excitation

The transmission error is influenced by:

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

8.1.2 Dynamic response

The dynamic tooth forces are influenced by:

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

8.2 Resonance

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

8.2.1 Gear pair resonance

If a particular frequency of the transmission error excitation is close to the natural frequency of the gear masses, or some multiple of the natural frequency such as $1/2$ or 2 , a resonant vibration may cause high dynamic tooth forces due to large relative displacements of the gear masses. The dynamic factors, C_v and K_v , do not account for gear pair resonance, and operation in this regime is to be avoided.

8.2.2 Gear blank resonance

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

8.2.3 System resonance

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

8.2.4 Shaft critical speeds

Due to the high bending stiffness of gear shafts the natural frequencies of lateral vibration of the gear shafts are usually much higher than the operating speeds. For high speed gears, however, it is recommended that the shaft critical speeds be analyzed to ensure that they are well removed from the operating speed range. The dynamic factors, C_v