

ANSI/AGMA 1003-G93 ERRATA

November, 1995

The following editorial corrections should be made to ANSI/AGMA 1003-G93, *Tooth Proportions for Fine-Pitch Spur and Helical Gearing* (originally printed January, 1993). These changes, discovered after publication, have not been made in the printing of this document. The changes are shown below. Users of ANSI/AGMA 1003-G93 are encouraged to cut out these stickers and insert them in the standard. The equations can be placed over the existing equations.

Page 14

$$N_e = \frac{2.10 \cos \psi}{[\sin \phi (\sin \phi - \cos \phi \tan 5^\circ)]} \quad \dots(20)$$

$$z_g = \frac{2.10 \cos \beta_m}{[\sin \alpha_0 (\sin \alpha_0 - \cos \alpha_0 \tan 5^\circ)]} \quad \dots(20M)$$

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Table C.1 – Enlargement Criteria*

14.5°			20°			25°		
Helix angle, degrees $\psi (\beta_m)$	Transverse profile angle, degrees $\phi_t (\alpha_{pt})$	Number of teeth** $N_e (z_g)$	Helix angle, degrees $\psi (\beta_m)$	Transverse profile angle, degrees $\phi_t (\alpha_{pt})$	Number of teeth** $N_e (z_g)$	Helix angle, degrees $\psi (\beta_m)$	Transverse profile angle, degrees $\phi_t (\alpha_{pt})$	Number of teeth** $N_e (z_g)$
0	14.500000	50.62	0	20.000000	23.63	0	25.000000	14.47
5	14.553040	49.97	5	20.070308	23.36	5	25.083771	14.32
10	14.714048	48.07	10	20.283559	22.55	10	25.337611	13.85
15	14.988849	45.04	15	20.646896	21.25	15	25.769262	13.11
18	15.212411	42.77	18	20.941896	20.27	18	26.118938	12.54
20	15.387707	41.09	20	21.172832	19.54	20	26.392181	12.12
23	15.692808	38.37	23	21.573983	18.36	23	26.865777	11.44
25	15.926252	36.45	25	21.880232	17.52	25	27.226435	10.96
30	16.626985	31.42	30	22.795877	15.30	30	28.300052	9.66
35	17.521624	26.25	35	23.956803	12.99	35	29.650978	8.31
40	18.654748	21.22	40	25.413766	10.71	40	31.329769	6.95
45	20.089512	16.54	45	27.236313	8.54	45	33.403198	5.65

* Based on 5° minimum roll angle at form diameter.

** N_G is a calculated value (see equation 18). Pinions with numbers of teeth that exceed this value do not require enlargement.

ANSI/AGMA 1003-G93

(Revision of AGMA 207.06-1974)

AMERICAN NATIONAL STANDARD

***Tooth Proportions for Fine-Pitch
Spur and Helical Gearing***

ANSI/AGMA 1003-G93



AGMA STANDARD

Tooth Proportions for Fine-Pitch Spur and Helical Gearing
AGMA 1003-G93
Revision of AGMA 207.06-1974

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Approved January 22, 1993
American National Standards Institute, Inc.

ABSTRACT:

Tooth proportions for fine pitch gearing are similar to those of coarse pitch gearing except in the matter of clearance. For 20 degree profile angle fine pitch gearing, this standard provides a system of enlarged pinions which use the involute form above 5 degrees of roll. Data on 14-1/2 and 25 degree profile angle systems are included in the annexes.

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FOREWORD

[The foreword, footnotes, and annexes are provided for informational purposes only, and should not be construed as a part of ANSI/AGMA 1003-G93, *Tooth Proportions for Fine-Pitch Spur and Helical Gearing*.]

As originally developed by the American Gear Manufacturers Association, this standard was in two parts: the first part, *Clearance for 20-Degree Pressure Angle Fine-Pitch Gears* (AGMA 470.01); and the second, *20-Degree Involute Fine-Pitch System for Spur Gears* (AGMA 207.02).

In May, 1949, the two standards were combined and completely re-edited.

The next revision of this standard was begun in 1955.

As a result of the increasing use of gears by sintering and injection molding process, and for greater tooth strength, tooth forms for 25 degree pressure angle were included. Control gearing containing large numbers of teeth was recognized by data on the 14-1/2 degree pressure angle system in the information sheets.

AGMA 207.05, was approved by Sectional Committee B6 and by the sponsors, and designated USA Standard B6.7-1967 as of September 18, 1967.

Due to difficulties encountered in fabricating gears with involute profiles to the base circle, the Fine-Pitch committee developed a new set of tooth proportions for enlarged pinions that would not require active tooth profiles below the five degrees of roll.

AGMA 207.06 was approved by the Fine-Pitch Gearing Committee in June, 1971 and approved by the membership as of May, 1974.

AGMA 1003-G93 is a revision of AGMA 207.06.

The term "profile angle" was introduced in place of the basic rack "pressure angle."

Metric data were added, including ISO symbols. Tables for 20 degree profile angle were revised, and supported with simpler equations and procedures. The lower range of tooth numbers was redone with less enlargement and improved contact ratio and less specific sliding. Data for 7 and 8 tooth pinions were omitted, as they require special design consideration beyond the scope of this standard.

A revised procedure was employed to verify undercut limits, superseding the approximate and more conservative prior method.

Formulas were supplied for all tabulated data.

The data on helical gearing was revised using a simple procedure to allow helical configuration.

Information was added to clarify the distinction between form diameter as generated and the limit diameter established by operational contact which determines the contact ratio.

Clarification was made regarding categories of center distance which often was a source of confusion in the prior standard.

Cautionary notes were added to indicate meshes employing very small numbers of teeth, while geometrically correct, still require analyses for strength, durability, and clearances.

The 5 degree form diameter enlargement method was extended to include the 14-1/2 degree system, and revisions were made to the 25 degree system.

ANSI/AGMA 1003-G93 was approved by the Fine-Pitch Gearing Committee in February, 1992 and approved by the AGMA Board of Directors as of May, 1992.

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.

ANSI/AGMA 1003-G92

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American National Standard –**Tooth Proportions for
Fine-Pitch Spur and
Helical Gearing****1 Scope**

This standard is applicable to external spur and helical gears with diametral pitch of 20 through 120 (1.25 through 0.2 module) and a profile angle of 20 degrees.

It only applies to standard gears with 24 teeth or more; enlarged pinions with 9 through 23 teeth; and reduced gears.

Much of this same information is applicable to internal gears.

1.1 Tooth proportions

The tooth proportions shown herein may be used for many gear designs of finer than 120 diametral pitch (0.2 module); however, such designs should be checked for suitability, particularly in the areas of contact ratio, undercutting, and clearance.

The spur gear portions of this standard closely follow AGMA 201.02, *Tooth Proportions for Coarse-Pitch Involute Spur Gears*.

The main difference between the proportions of fine-pitch gears and those of the coarse-pitch is in the clearance. In fine-pitch gearing, wear on the points of the cutting tools is proportionally greater than in coarse-pitch tools. The fillet radius produced by such tooling will therefore be proportionally greater. The increased clearance in gearing of 20 diametral pitch (1.25 module) and finer provides both for the relatively larger fillet and also for foreign material that tends to accumulate at the bottoms of the teeth.

1.2 Number of teeth

Gear designs with low numbers of teeth should be checked for suitability, particularly in the areas of contact ratio, undercutting, and clearance, as well as for strength and durability for load and life considerations.

1.3 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 201.02, *Tooth Proportions for Coarse-Pitch Involute Spur Gears*

AGMA 370.01, *Design Manual for Fine-Pitch Gearing*

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

ANSI/AGMA 2002-B88, *Tooth Thickness Specification and Measurement*

ANSI B94.7 *Hobs*

ANSI B94.21 *Shaper Cutters*

2 Terms and symbols**2.1 Terms**

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

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

AGMA 904-B89, *Metric Usage*

2.2 Symbols

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

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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 (metric) units of measure are shown in parentheses in table 1 and in the text. ISO symbols are also shown in parentheses in the text and most tables. A metric version of each equation in this standard,

with ISO symbols, is shown after the conventional version, indented, in smaller type, and with "M" included in the equation number.

Example:

$$h_k = \frac{2.000}{P_{nd}} \quad \dots(1)$$

$$h_w = 2.000 m_n \quad \dots(1M)$$

The second expression uses SI units.

Table 1 – Terms and symbols

Symbol		Description	Units	First used
AGMA	ISO			
a	h_a	Addendum	in (mm)	3.4
a_G	h_{a2}	Addendum, gear	in (mm)	table 5
a_P	h_{a1}	Addendum, pinion	in (mm)	table 4
B	j	Backlash	in (mm)	3.6
b	h_f	Dedendum	in (mm)	table 2
C	a	Center distance	in (mm)	table 2
C_e	a_d	Calculated tight mesh center distance	in (mm)	equation 16
c	c	Clearance	in (mm)	3.5
c_e	c_p	Clearance, gear tip to pinion root	in (mm)	table 4
D	d	Pitch diameter	in (mm)	table 2
D_G	d_2	Standard pitch diameter of gear	in (mm)	equation 15
D_o	d_e	Outside diameter	in (mm)	table 2
D_{oG}	d_{e2}	Outside diameter of gear	in (mm)	equation 15
D_{oP}	d_{e1}	Outside diameter of pinion	in (mm)	equation 15
D_P	d_1	Standard pitch diameter of pinion	in (mm)	equation 15
D_r	d_f	Root diameter	in (mm)	table 2
h_k	h_w	Working depth	in (mm)	3.3
h_t	h_t	Whole depth	in (mm)	3.7
--	m_{et}	Module, transverse	(mm)	table 2M
m_c	ϵ_β	Contact ratio	----	equation 15
m_{ce}	$\epsilon_{\beta x}$	Contact ratio, enlarged	----	equation 16
--	m_n	Module, normal	(mm)	equation 1M
N	z	Number of teeth	----	table 2
N_e	z_g	A calculated value	----	equation 20
N_G	z_2	Number of teeth, gear	----	table 2
N_P	z_1	Number of teeth, pinion	----	table 2
P_d	--	Diametral pitch, transverse	in ⁻¹	table 2
P_{nd}	--	Diametral pitch, normal	in ⁻¹	equation 1
p	p_t	Circular pitch, transverse	in (mm)	table 2

(continued)

Table 1 (concluded)

Symbol		Description	Units	First used
AGMA	ISO			
p_h	p_h	Circular pitch, normal	in (mm)	table 2
R_{bG}	r_{b2}	Base radius of gear	in (mm)	figure 4
R_{bP}	r_{b1}	Base radius of pinion	in (mm)	figure 4
R_{oG}	r_{e2}	Outside radius of gear	in (mm)	figure 4
R_{oP}	r_{e1}	Outside radius of pinion	in (mm)	figure 4
$r_{\max}$	$r_{f\max}$	Fillet radius, maximum	in (mm)	table 2
t	s_t	Tooth thickness, transverse	in (mm)	3.6
t_G	s_2	Tooth thickness, gear	in (mm)	equation 19
t_n	s_n	Tooth thickness, normal	in (mm)	table 2
t_o	s_e	Tooth thickness at outside diameter	in (mm)	equation 12
t_P	s_1	Tooth thickness, pinion	in (mm)	equation 19
t_{oG}	s_{ei2}	Top land, gear	in (mm)	table 5
t_{oP}	s_{ei1}	Top land, pinion	in (mm)	table 4
Δa	Δha	Addendum enlargement	in (mm)	4.5
ΔC	Δa_0	Center distance enlargement with rack	in (mm)	4.9
Δe	Δx	Enlargement	in (mm)	equation 3
Δt	Δs	Tooth thickness enlargement	in (mm)	equation 4
ϕ	α_0	Profile angle	----	table 2
ϕ_e	α_u	Transverse pressure angle at calculated tight mesh center distance	----	equation 16
ϕ_n	α_n	Profile angle, normal	----	table 2
ϕ_o	α_e	Pressure angle at outside diameter	----	equation 13
ψ	β_m	Helix angle	----	table 2

3 General features

3.1 Basic rack

The basic rack shown in figure 1 is used to illustrate the tooth proportions covered by this standard. This standard permits freedom of choice in making changes in the gear tooth proportions to meet special design conditions as long as the resulting gears are fully conjugate to the basic rack. Such changes may be indicated when a special contact ratio or modification for tooth strength is desired.

3.1.1 Spur gears

The basic rack shown in figure 1 and the tooth proportions shown in tables 2 and 2M provide the basic design data for spur gear teeth.

3.1.2 Helical gears

The helical teeth covered by this standard are conjugate in the normal plane to the basic rack shown in figure 1 and tables 2 and 2M.

3.2 Pressure angle and profile angle

3.2.1 Pressure angle, ϕ (α_0)

While **profile angle** is the slope of the cutting tool, a **pressure angle** may be defined at any point on the flank of a gear tooth. See ANSI/AGMA 1012-F90, *Gear Nomenclature, Definitions of Terms with Symbols*, for further discussion.

3.2.2 Profile angle

The standard profile angle is 20 degrees, and is recommended for most applications. In the an-

nexes, data may be found on 14-1/2 and 25 degree profile angle systems. Profile angle of helical teeth is taken in the normal plane.

In certain cases, notably some sintered or molded gears, or in gearing where greatest strength and wear resistance are desired, a 25 degree profile angle may be required. Profile angles greater than 20 degrees tend to require the use of generating tools having very narrow point widths. In addition, the larger profile angles require closer control on center distance tolerance for those gear trains in which backlash is critical.

In cases where considerations of angular position or backlash are critical and where both pinions and gears contain relatively large numbers of teeth, a 14-1/2 degree profile angle may be desirable. In general, profile angles of less than 20 degrees require a greater amount of modification to avoid undercut problems and are limited to larger total numbers of teeth in gear and pinion when operating on a standard center distance.

3.3 Working depth, h_k (h_w)

The basic working depth is:

$$h_k = \frac{2.000}{P_{nd}} \quad \dots(1)$$

$$h_w = 2.000 m_n \quad \dots(1M)$$

Teeth with this depth are commonly referred to as full depth teeth.

3.4 Addendum, a (h_a)

Standard addendum teeth are used for applications where the number of teeth are equal to or exceed the minimum numbers shown in table C.1.

Enlarged and reduced addendum proportions are used to avoid objectionable undercut or for considerations of tooth strength, contact ratio or center distance. Table 4 gives recommended tooth proportions to avoid undercut problems in a mesh with a pinion of a small number of teeth. Generally, as the total number of teeth in gear and pinion gets smaller, the contact ratio diminishes. Special attention must be given to avoid contact ratios below 1.2.

3.5 Clearance, c (c)

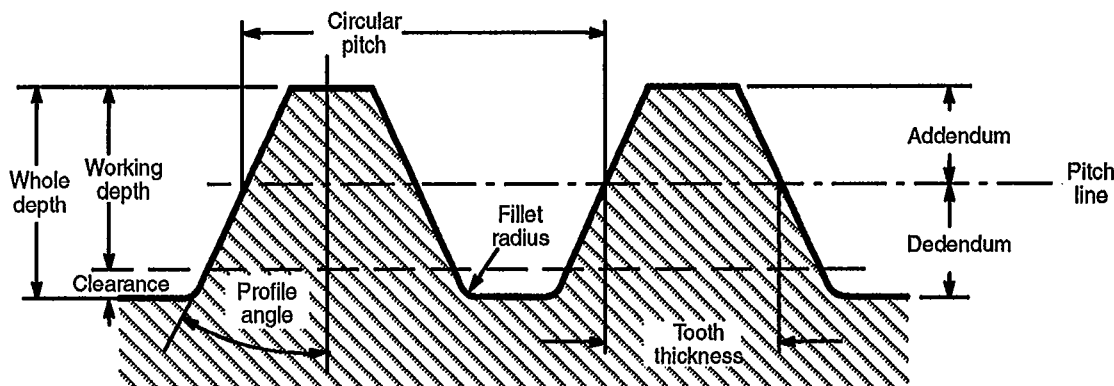
Standard clearance for the diametral pitch system is:

$$c = \frac{0.200}{P_{nd}} + 0.002 \quad \dots(2)$$

For the module system:

$$c = 0.200 m_n + 0.05 \quad \dots(2M)$$

Greater clearance than given in tables 3 and 3M may be required if teeth are to be finished by a secondary operation. While the required clearance may vary with specific gear applications, a value of $0.350/P_{nd}$ ($0.350 m_n$) should provide the necessary amount in most cases. See ANSI B94.7-1966, *Hobs*; ANSI B94.21-1968, *Shaper Cutters*; and AGMA 370.01, *Design Manual for Fine-Pitch Gearing*.



NOTE - The value of the fillet radius is determined by the type and design of the cutting tool.

Figure 1 - Basic rack (normal plane)

3.6 Tooth thickness, $t(s_t)$, and backlash, $B(j)$

The tooth thickness shown in the tables does not include an allowance for backlash when the gears are meshed at standard center distances.

In general, the teeth of both members are reduced in thickness to provide backlash. In cases of pinions having small numbers of teeth, consideration may be given to applying more of the tooth thickness reduction to the gear member to provide the required backlash. See ANSI/AGMA 2002-B88 for a more detailed discussion of tooth thickness specification.

Allowance (thinning) for backlash must be considered to allow for lubricant, temperature effects, and operational meshing conditions including deflections, bearing runouts, and gear element variations. For a detailed discussion see AGMA Paper 239.14, *Assured Backlash Control, the ABC System*, by Leonard J. Smith. [1]*

NOTE – The design tooth thickness is established from engineering considerations. It is determined by gear geometry, gear tooth strength, and backlash. The methods for establishing design tooth thickness, for a given application, are beyond the scope of this standard.

3.7 Whole depth, $h_t(h_t)$

The whole depth values shown in the tables will increase in proportion to the amount of tooth thinning in cutting the teeth, unless the outside diameter is also modified.

The whole depth of enlarged and reduced addendum gearing generated with pinion type shaper cutters may be different from that shown in the tables. In order to control the whole depth of external gears, the root diameter should be specified as a maximum dimension only.

3.8 Generating tools

Standard generating tools (hobs or shaper cutters) are used for either spur or helical gears. See ANSI B94.7-1966, *Hobs*, and ANSI B94.21-1986, *Shaper Cutters*.

Tables 2 and 2M give the formulas for standard tooth proportions without allowance for backlash. In

order to minimize the vast number of tools (cutters and master gears) required for all possible pitches (modules), the following are recommended:

Diametral Pitches:

20
24
32
40
48
64
72
80
96
120

Modules:

1.25
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2

Tables 3 and 3M show the tooth dimensions for each diametral pitch or module. Gear ratios at non-standard center distances which are sometimes fixed by component design requirements can usually be obtained using standard pitch cutters and enlarging one or both of the mating gears.

3.9 Tool tip radius

A zero fillet radius implies a sharp corner on the tip of the generating tool. In actual practice, the corner is made with a small radius. This radius is established by the cutting tool manufacturer, and has not been standardized. However, a maximum value is controlled by the tangency of a radius and the tool flank at the working depth, and tangency with the root circle at the whole depth.

4 Basis for enlarged (long addendum) pinions

4.1 Enlargement, $\Delta e(\Delta x)$

Pinions with small numbers of teeth are enlarged so that a standard tooth thickness rack with an enlarged addendum of $(1.0 + 0.05)/P_d [(1.0 + 0.05) m_{ef}]$

* [] Numbers in brackets refer to the references in annex E.

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will start contact 5 degrees above the base radius. The use of (0.05) extra addendum provides an allowance for center distance variation and eccentricity of mating gear O.D. The 5 degrees avoids the use of the involute in the area near the base circle.

A corresponding increase in tooth thickness is made along with the addendum enlargement.

NOTE – CAUTION should be exercised in using enlarged pinions in speed increasing drives to avoid excessive friction, deflection, and possible lockup.

Table 2 – Diametral pitch, standard tooth proportions and formulas (inch system)

Tooth proportions		
Item	Spur	Helical
Addendum, a	$\frac{1.000}{P_d}$	$\frac{1.000}{P_{nd}}$
Dedendum, b	$\frac{1.200}{P_d} + 0.002$	$\frac{1.200}{P_{nd}} + 0.002$
Working depth, h_k	$\frac{2.000}{P_d}$	$\frac{2.000}{P_{nd}}$
Whole depth, h_t	$\frac{2.200}{P_d} + 0.002$	$\frac{2.200}{P_{nd}} + 0.002$
Clearance, c (standard)	$\frac{0.200}{P_d} + 0.002$	$\frac{0.200}{P_{nd}} + 0.002$
Fillet radius, $r_{\max}$	$\frac{c}{1 - \sin \phi}$	$\frac{c}{1 - \sin \phi_n}$
Tooth thickness, t, t_n at standard pitch diameter	$t = \frac{\pi}{2 P_d}$	$t_n = \frac{\pi}{2 P_{nd}}$
Formulas		
Circular pitch, p, p_n	$p = \frac{\pi D}{N} = \frac{\pi}{P_d}$	$p_n = \frac{\pi}{P_{nd}}$
Pitch diameter, D (standard)	$\frac{N}{P_d}$	$\frac{N}{P_{nd} \cos \psi}$
Outside diameter, D_o	$\frac{N + 2}{P_d}$	$\frac{N}{P_{nd} \cos \psi} + \frac{2}{P_{nd}}$
Root diameter, D_r	$\frac{N - 2.4}{P_d} - 0.004$	$\frac{N}{P_{nd} \cos \psi} - \frac{2.4}{P_{nd}} - 0.004$
Center distance, C (standard)	$\frac{N_P + N_G}{2 P_d}$	$\frac{N_P + N_G}{2 P_{nd} \cos \psi}$
where		
P_d is transverse diametral pitch; P_{nd} is normal diametral pitch; t is transverse tooth thickness at standard pitch diameter; t_n is normal tooth thickness at standard pitch diameter; p is transverse circular pitch; p_n is normal circular pitch; ψ is helix angle; N is number of teeth; N_P is number of pinion teeth; N_G is number of gear teeth; ϕ is profile angle; ϕ_n is normal profile angle.		

Table 2M – Module, standard tooth proportions and formulas (metric system)

Tooth proportions		
Item	Spur	Helical
Addendum, h_a	$1.000 m_{et}$	$1.000 m_n$
Dedendum, h_f	$1.200 m_{et} + 0.05$	$1.200 m_n + 0.05$
Working depth, h_w	$2.000 m_{et}$	$2.000 m_n$
Whole depth, h_t	$2.200 m_{et} + 0.05$	$2.200 m_n + 0.05$
Clearance, c (standard)	$0.200 m_{et} + 0.05$	$0.200 m_n + 0.05$
Fillet radius, r_{fmax}	$\frac{c}{1 - \sin \alpha_0}$	$\frac{c}{1 - \sin \alpha_n}$
Tooth thickness, s_t s_n at standard pitch diameter	$s_t = \frac{\pi m_{et}}{2}$	$s_n = \frac{\pi m_n}{2}$
Formulas		
Circular pitch, p_t p_n	$p_t = \pi m_{et}$	$p_n = \pi m_n$
Pitch diameter, d (standard)	$z m_{et}$	$\frac{z m_n}{\cos \beta_m}$
Outside diameter, d_e	$(z + 2) m_{et}$	$\frac{z m_n}{\cos \beta_m} + 2 m_n$
Root diameter, d_f	$(z - 2.4) m_{et} - 0.100$	$\frac{(z m_n)}{\cos \beta_m} - (2.4 m_n) - 0.100$
Center distance, a (standard)	$\frac{z_1 + z_2}{2} m_{et}$	$\frac{z_1 + z_2}{2 \cos \beta_m} m_n$
where s_t is transverse tooth thickness at standard pitch diameter; s_n is normal tooth thickness at standard pitch diameter; m_{et} is transverse module; m_n is normal module; p_t is transverse circular pitch; p_n is normal circular pitch; β is helix angle; z^m is number of teeth; z_1 is number of pinion teeth; z_2 is number of gear teeth; α_0 is profile angle; α_n is normal profile angle.		

4.2 Form diameter

Teeth designed in accordance with this standard will have an involute profile between the 5 degree form diameter (see figures 2 and 3) and that point where tip chamfer or edge round begins.

The form diameter provides more than enough length of involute profile for meshing with any mating gear, including a rack.

Any special tip relief or modification of involute profile to suit design or operational requirements is beyond the scope of this standard.

Table 3 – Standard diametral pitch tooth dimensions (inch system, in)

1	2	3	4	5	6	7	8
Diametral pitch	Circular pitch	Circular thickness	Working depth	Whole depth	Clearance	Addendum	Dedendum
20	0.15708	0.07854	0.1000	0.1120	0.0120	0.0500	0.0620
24	0.13090	0.06545	0.0833	0.0937	0.0104	0.0417	0.0520
32	0.09818	0.04909	0.0625	0.0708	0.0083	0.0313	0.0395
40	0.07854	0.03927	0.0500	0.0570	0.0070	0.0250	0.0320
48	0.06545	0.03272	0.0417	0.0478	0.0062	0.0208	0.0270
64	0.04909	0.02454	0.0312	0.0364	0.0051	0.0156	0.0208
72	0.04363	0.02182	0.0278	0.0326	0.0048	0.0139	0.0187
80	0.03927	0.01964	0.0250	0.0295	0.0045	0.0125	0.0170
96	0.03272	0.01636	0.0208	0.0249	0.0041	0.0104	0.0145
120	0.02618	0.01309	0.0167	0.0203	0.0037	0.0083	0.0120

NOTE – All dimensions are given in inches.

Table 3M – Standard module tooth dimensions (metric system, mm)

1	2	3	4	5	6	7	8
Module	Circular pitch	Circular thickness	Working depth	Whole depth	Clearance	Addendum	Dedendum
1.25	3.9270	1.9635	2.500	2.800	0.300	1.250	1.550
1.0	3.1416	1.5708	2.000	2.250	0.250	1.000	1.250
0.9	2.8274	1.4137	1.800	2.030	0.230	0.900	1.130
0.8	2.5133	1.2566	1.600	1.810	0.210	0.800	1.010
0.7	2.1991	1.0996	1.400	1.590	0.190	0.700	0.890
0.6	1.8850	0.9425	1.200	1.370	0.170	0.600	0.770
0.5	1.5708	0.7854	1.000	1.150	0.150	0.500	0.650
0.4	1.2566	0.6283	0.800	0.930	0.130	0.400	0.530
0.3	0.9425	0.4712	0.600	0.710	0.110	0.300	0.410
0.2	0.6283	0.3142	0.400	0.490	0.090	0.200	0.290

NOTE – All dimensions are given in millimeters.

4.3 Limit diameter

The limit diameter is based on the actual contact with a mating gear at the operating or working center distance. It may be shown on the drawing as an optional specification thereby confining inspection to functional requirements.

4.4 Top land, $t_o (s_e)$

In order to avoid sharp tips and maintain a minimum top land for strength and durability purposes, the enlarged addendum (enlarged outside diameter) is reduced from the computed enlargement in the case of pinions with very small numbers of teeth. In this standard, the recommended minimum top land is

$0.275/P_d$ ($0.275 m_{et}$) for spur gears and $0.275/P_{nd}$ ($0.275 m_n$) for helical gears. For power gearing, good design practice limits the ratio of the top lands in a mesh.

4.5 Undercut

Conditions of undercut were computed by means of the Davis method [2], and cross checked by the Khiralla equations [3]. Undercut is avoided by addendum enlargement.

The addendum enlargement, Δa (Δh_a), satisfies the requirement that any radial height undercut above the base circle must not exceed the 5 degree form diameter. See figures 2 and 3.

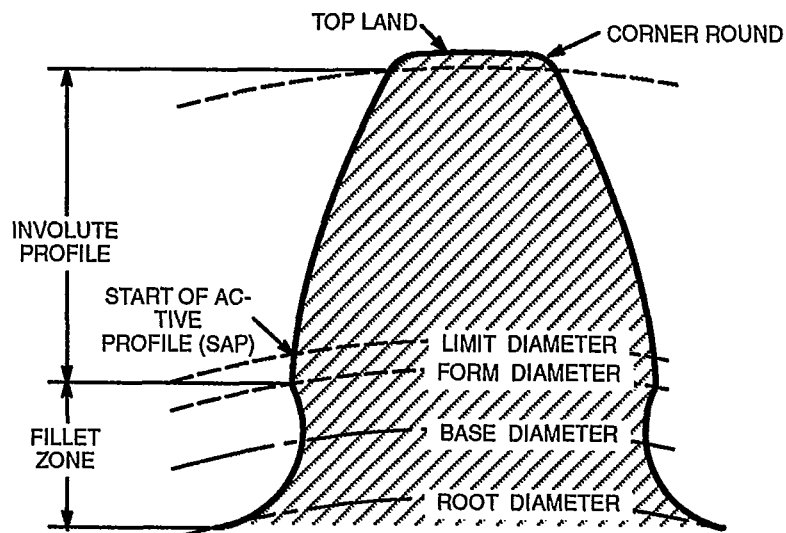


Figure 2 – Form diameter on undercut teeth

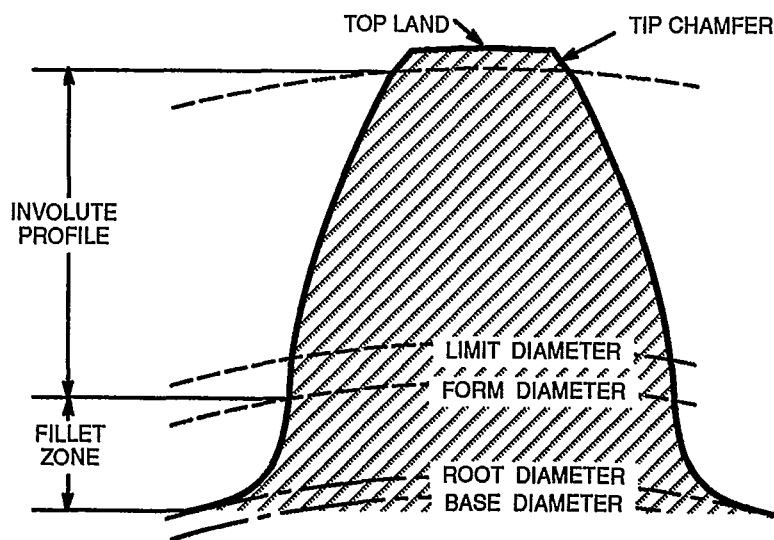


Figure 3 – Form diameter on fillet blend teeth

4.6 Root diameter, D_r (d_f)

Since this system is based on the use of a standard rack, the root diameter derives from the computed (not truncated) outside diameter of the pinion and the outside diameter (reduced) of the gear. The root diameter is specified as a maximum dimension and generally is not toleranced. Highly stressed gears may require some limit, but would be used in conjunction with a controlled root fillet radius as additional specification.

4.7 Mating gear (standard)

The mating gear to an enlarged pinion may be a standard gear, in which case the center distance must be enlarged for operation.

CAUTION – The center distance for tight mesh (zero backlash) does not provide the standard clearance. It is therefore necessary to increase the enlarged center distance if the standard clearance is desired. When doing so, the mesh will incur some backlash increase at the new working center distance. See annex D.

4.8 Mating gear (reduced – short addendum)

The mating gear can be meshed at standard center distance by reducing the mate in the same manner and amount used to enlarge the pinion (except truncation of top land).

4.9 Formula for enlargement of spur pinions

$$\Delta e = \frac{1.05 - 0.5 N_P \sin \phi (\sin \phi - \cos \phi \tan 5^\circ)}{P_d} \quad \dots(3)$$

$$\Delta x = [1.05 - 0.5 z_1 \sin \alpha_0 (\sin \alpha_0 - \cos \alpha_0 \tan 5^\circ)] m_{et} \quad \dots(3M)$$

$$t = \frac{\pi}{2} + \Delta t \quad \dots(4)$$

$$s_t = \frac{\pi}{2} + \Delta s \quad \dots(4M)$$

$$\Delta t = 2 \Delta e (\tan \phi) \quad \dots(5)$$

$$\Delta s = 2 \Delta x (\tan \alpha_0) \quad \dots(5M)$$

$$\Delta a = \Delta e \quad \dots(6)^*$$

$$\Delta h_a = \Delta x \quad \dots(6M)^*$$

where

- N_P (z_1) is number of pinion teeth;
- ϕ (α_0) is profile angle, transverse;
- Δt (Δs) is tooth thickness enlargement;
- Δa (Δh_a) is addendum enlargement;
- Δe (Δx) is enlargement = ΔC ;
- ΔC (Δa_0) is center distance enlargement with rack.

NOTE – Equation 3 is taken from reference [4] and contains a mathematical error in the use of “ $\tan 5^\circ$ ”. This should have been “ $\tan 4.98726^\circ$ ”, which is the equivalent pressure angle for 5 degrees of roll angle. The use of “ $\tan 5^\circ$ ” provides a roll angle of 5.01273° . Since the purpose is to avoid contact in this region, it provides a slight extra allowance.

To avoid wholesale tabular corrections to long standing data, the original equation has been retained.

The 5 degree form diameter is based on the use of a 1.05 addendum rack, and is equivalent to the limit diameter with this rack.

4.9.1 Equations for tables (unit pitch) **

The following equations are used to determine the values in tables 4 and 5:

$$\Delta a = 1.05 - 0.5 N \sin \phi [\sin \phi - \cos \phi \tan 5^\circ] \quad \dots(7)$$

$$\Delta h_a = 1.05 - 0.5 z \sin \alpha_0 [\sin \alpha_0 - \cos \alpha_0 \tan 5^\circ] \quad \dots(7M)$$

$$D_o = N + 2 \pm (2 \Delta a) \quad \dots(8)^*$$

$$d_e = z + 2 \pm (2 \Delta h_a) \quad \dots(8M)^*$$

$$a = 1 \pm \Delta a \quad \dots(9)^*$$

$$h_a = 1 \pm \Delta h_a \quad \dots(9M)^*$$

$$t = \left(\frac{\pi}{2}\right) \pm 2 \Delta t \quad \dots(10)$$

$$s_t = \left(\frac{\pi}{2}\right) \pm 2 \Delta s \quad \dots(10M)$$

$$D_r = N - 2.4 \pm 2 \Delta e \quad \dots(11)^{***}$$

$$d_f = z - 2.4 \pm 2 \Delta x \quad \dots(11M)^{***}$$

* Nominal equation, modified when pinion tooth is truncated for minimum top land.

** Sign determined by enlargement or reduction from standard.

*** Actual root diameter is decreased by 0.004 in (0.10 mm).

$$t_o = 0.275 \text{ min (found by iteration of } D_o) \quad \dots(12)$$

$$s_e = 0.275 \text{ min (found by iteration of } d_e) \quad \dots(12M)$$

$$t_o = D_o \left(\frac{t}{N} + \text{inv } \phi - \text{inv } \phi_o \right) \quad \dots(13)$$

$$s_e = d_e \left(\frac{s_t}{z} + \text{inv } \alpha_o - \text{inv } \alpha_e \right) \quad \dots(13M)$$

$$\phi_o = \cos^{-1} \left(\frac{N \cos \phi}{D_o} \right) \quad \dots(14)$$

$$\alpha_e = \cos^{-1} \left(\frac{z \cos \alpha_o}{d_e} \right) \quad \dots(14M)$$

$$m_c = \frac{\sqrt{D_{oP}^2 - (D_P \cos \phi)^2} + \sqrt{D_{oG}^2 - (D_G \cos \phi)^2}}{2 p \cos \phi} - \frac{2 C \sin \phi}{2 p \cos \phi} \quad \dots(15)$$

$$\epsilon_\beta = \frac{\sqrt{d_{e1}^2 - (d_1 \cos \alpha_o)^2} + \sqrt{d_{e2}^2 - (d_2 \cos \alpha_o)^2}}{2 p_t \cos \alpha_o} - \frac{2 a \sin \alpha_o}{2 p_t \cos \alpha_o} \quad \dots(15M)$$

$$m_{ce} = \frac{\sqrt{D_{oP}^2 - (D_P \cos \phi)^2} + \sqrt{D_{oG}^2 - (D_G \cos \phi)^2}}{2 p \cos \phi} - \frac{2 C_e \sin \phi_e}{2 p \cos \phi} \quad \dots(16)$$

$$\epsilon_{\beta x} = \frac{\sqrt{d_{e1}^2 - (d_1 \cos \alpha_o)^2} + \sqrt{d_{e2}^2 - (d_2 \cos \alpha_o)^2}}{2 p_t \cos \alpha_o} - \frac{2 a_x \sin \alpha_d}{2 p_t \cos \alpha_o} \quad \dots(16M)$$

where

t_o (s_e) is tooth thickness at outside diameter;

ϕ_o (α_e) is pressure angle at outside diameter;

m_c (ϵ_β) is contact ratio;

D_{oP}, D_{oG} (d_{e1}, d_{e2}) is outside diameter of pinion, gear;

D_P, D_G (d_1, d_2) is standard pitch diameter of pinion, gear;

p (p_t) is circular pitch;

C (a) is center distance (standard). See equation 17;

m_{ce} ($\epsilon_{\beta x}$) is calculated tight mesh contact ratio;

C_e (a_d) is calculated tight mesh center distance;

ϕ_e (α_d) is transverse pressure angle at calculated tight mesh center distance (enlarged).

4.10 Standard center distance (for standard spur gears)

Standard gears, made to standard tooth proportions without modification of addendum, dedendum, or tooth thickness (other than for backlash), are run at standard center distance. Data is shown without an allowance for backlash.

$$C = \frac{N_P + N_G}{2 P_d} \quad \dots(17)$$

$$a = \frac{(z_1 + z_2) m_{et}}{2} \quad \dots(17M)$$

4.11 Standard center distance (for enlarged spur pinions and reduced gears)

The data in this standard provide the proper dimensional adjustment of each mating member to allow them to run at the same (standard) center distance as unmodified (standard) gears. Data is shown without an allowance for backlash.

Table 4 provides data for enlarged pinions, and table 5 provides data for reduced gears.

The advantages of this system are: no change in center distance is required; operating pressure angle remains standard; and the contact ratio is slightly greater than if the center distance were increased.

In most cases where gear trains include idler gears, the standard center distance cannot be used with enlarged gears.

Table 4 – 20° Profile angle – enlarged spur pinions

Enlarged pinion dimensions (unit pitch)						Enlarged center distance, pinion with 24 tooth gear		
1	2	3	4	5	6	7	8	9
Number of teeth	Outside diameter (enlarged)	Addendum (enlarged)	Tooth thickness (enlarged)	Root diameter* (enlarged)	Top land	Contact ratio	Clearance gear tip to pinion root**	Center distance
N_P (z_1)	D_{OP} (d_{e1})	a_P (h_{a1})	t_P (s_1)	D_r (d_f)	t_{OP} (s_{e1l})	m_{ce} ($\varepsilon_{\beta x}$)	c_e (c_p)	C_e (a_d)
9	12.0144	1.5072	2.04405	7.9003	0.2750	1.209	0.1308	17.08092
10	13.0256	1.5128	2.01171	8.8114	0.2750	1.261	0.1402	17.54587
11	14.0304	1.5152	1.97937	9.7225	0.2750	1.310	0.1489	18.01010
12	15.0296	1.5148	1.94703	10.6337	0.2750	1.358	0.1568	18.47360
13	15.9448	1.4724	1.91469	11.5448	0.3401	1.383	0.1640	18.93641
14	16.8560	1.4280	1.88234	12.4560	0.3994	1.407	0.1705	19.39851
15	17.7671	1.3836	1.85000	13.3671	0.4513	1.429	0.1764	19.85995
16	18.6783	1.3391	1.81766	14.2783	0.4968	1.450	0.1816	20.32072
17	19.5894	1.2947	1.78532	15.1894	0.5370	1.471	0.1861	20.78083
18	20.5005	1.2503	1.75297	16.1005	0.5728	1.492	0.1900	21.24027
19	21.4117	1.2058	1.72063	17.0117	0.6046	1.511	0.1932	21.69909
20	22.3228	1.1614	1.68829	17.9228	0.6331	1.531	0.1959	22.15726
21	23.2340	1.1170	1.65595	18.8340	0.6585	1.550	0.1978	22.61481
22	24.1451	1.0726	1.62361	19.7451	0.6814	1.569	0.1992	23.07172
23	25.0562	1.0281	1.59126	20.6562	0.7020	1.588	0.1999	23.52799
24	26.0000	1.0000	1.57080	21.6000	0.7156	1.602	0.2000	24.00000

NOTE – Table is dimensionless. Divide values in columns 2, 3, 4, 5, and 6 by diametral pitch for inch units, or multiply by module for millimeter units.

NOTE – Table values for contact ratio are for tight mesh conditions, with no allowance for tooth thinning. Contact ratio is computed for tight mesh and limit diameters. Columns 6 through 9 are for reference only and not to be specified on drawings.

* Actual root diameter is decreased by 0.004 in (0.10 mm).

** Actual clearance is increased by 0.002 in (0.05 mm).

4.12 Enlarged center distance (for enlarged spur pinion mating with a standard gear)

When an enlarged pinion and a standard gear are meshed together, the center distance must be increased. Data for the individual pinions is shown without an allowance for backlash (see table 4). The computation for the tight mesh center distance is dependent upon the summation of the effects of the tooth thickness of the pinion and the tooth thickness of the gear. With an increase of center distance, there is a slight increase in the operating pressure angle.

$$C_e = C \frac{\cos \phi}{\cos \phi_e} \quad \dots(18)$$

$$a_d = a \frac{\cos \alpha_0}{\cos \alpha_d} \quad \dots(18M)$$

$$\text{inv } \phi_e = \text{inv } \phi + \frac{(t_P + t_G - p) P_d}{N_P + N_G} \quad \dots(19)$$

$$\text{inv } \alpha_d = \text{inv } \alpha_0 + \frac{s_1 + s_2 - p_t}{(z_1 + z_2) m_{et}} \quad \dots(19M)$$

where

t_P, t_G (s_1, s_2) is tooth thickness of pinion, gear.

Table 5 – 20° Profile angle – reduced spur gears

Reduced gear dimensions (unit pitch)						Standard center distance $C = 24.000$	
1	2	3	4	5	6	7	8
Minimum no. of teeth in gear N_G (z_2)	Outside diameter (reduced) D_{oG} (d_{e2})	Addendum (reduced) a_G (h_{a2})	Tooth thickness (reduced) t_G (s_2)	Root diameter* (reduced) D_r (d_f)	Top land t_{oG} (s_{ei2})	Contact ratio, N_P mating with N_G m_c (ϵ_β)	Number of teeth in pinion N_P (z_1)
39	39.6997	0.3499	1.09754	35.2997	0.8414	1.250	9
38	38.7886	0.3943	1.12988	34.3886	0.8389	1.313	10
37	37.8775	0.4387	1.16222	33.4775	0.8357	1.372	11
36	36.9663	0.4832	1.19457	32.5663	0.8319	1.427	12
35	36.0552	0.5276	1.22691	31.6552	0.8273	1.457	13
34	35.1440	0.5720	1.25925	30.7440	0.8220	1.484	14
33	34.2329	0.6164	1.29159	29.8329	0.8158	1.507	15
32	33.3218	0.6609	1.32393	28.9218	0.8088	1.528	16
31	32.4106	0.7053	1.35628	28.0106	0.8008	1.546	17
30	31.4995	0.7497	1.38862	27.0995	0.7918	1.562	18
29	30.5883	0.7942	1.42096	26.1883	0.7817	1.574	19
28	29.6772	0.8386	1.45330	25.2772	0.7703	1.585	20
27	28.7660	0.8830	1.48565	24.3660	0.7577	1.593	21
26	27.8549	0.9275	1.51799	23.4549	0.7436	1.598	22
25	26.9438	0.9719	1.55033	22.5438	0.7279	1.601	23
24	26.0000	1.0000	1.57080	21.6000	0.7156	1.602	24

NOTE – Table is dimensionless. Divide values in columns 2, 3, 4, 5, and 6 by diametral pitch for inch units, or multiply by module for millimeter units.

NOTE – Table values for contact ratio are for tight mesh conditions, with no allowance for tooth thinning. Contact ratio is computed for tight mesh and limit diameters. Column 6 through 8 are for reference only, not to be specified on drawing.

* Actual root diameter is decreased by 0.004 in (0.10 mm).

The advantage of this system is that only the pinions need be changed from standard dimensions.

The disadvantages of this system are: center distance must be enlarged over standard; the operating pressure angle increases slightly with different combinations of pinions and gears; and the contact ratio is slightly smaller than that obtained with the standard center distance system.

Special attention must be paid to providing adequate clearance with these meshes since the computed tight mesh center distance does not provide the standard clearance. An additional increase in

the center distance may be required which results in a slight backlash in the mesh. See annex D.

4.13 Center distance caution (enlarged pinion meshing with enlarged pinion)

The design method shown in this standard is not intended for use in meshing identical enlarged pinions together, nor any combination of enlarged pinions meshing together. While some combinations may be successful, they are not recommended and have been removed from this standard. Such gears require special design consideration, not only for clearance and contact ratios, but for analysis for

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strength and endurance and various other considerations beyond the scope of this standard.

4.14 Contact ratio

The contact ratio is the number of angular pitches through which a tooth surface rotates from the beginning to the end of contact. It is obtained as the ratio of the active length of action to the base pitch. See figure 4 and equation 15. Contact ratio is related to the center distance employed; i.e., standard, enlarged, or working.

4.15 Enlargement criteria

Table C.1 in annex C lists the number of teeth below which enlargement should be made to satisfy the 5° angle minimum condition.

$$N_e = \frac{2.10}{[\sin \phi (\sin \phi - \cos \phi \tan 5^\circ)]} \quad \dots(20)$$

$$z_g = \frac{2.10}{[\sin \alpha_0 (\sin \alpha_0 - \cos \alpha_0 \tan 5^\circ)]} \quad \dots(20M)$$

where

$N_e(z_g)$ is a calculated value. Pinions with numbers of teeth that exceed this value do not require enlargement.

Pinions made with tooth numbers larger than $N_e(z_g)$ allow use of standard addendum tooth proportions.

4.16 Example (spur pinion center distance)

Find the enlarged center distance, C_e , of a 9 tooth enlarged pinion running with a standard 24 tooth gear. This method finds the tight mesh center distance by summation of the tooth thicknesses. See equation 19.

$$N_P = 9$$

$$N_G = 24$$

$$\phi = 20^\circ$$

$$P_d = 1$$

$$p = \pi$$

$$t_P = 2.04405$$

$$t_G = \frac{p}{2}$$

The following equations (from subclauses 4.9–4.12) are used:

$$C = \frac{(9 + 24)}{2} = 16.5$$

$$\text{inv } \phi = \tan \phi - \text{arc } \phi = 0.0149044$$

$$\text{inv } \phi_e = 0.0149044 + \frac{(2.04405 + 1.570796 - \pi)(1)}{9 + 24}$$

$$\text{inv } \phi_e = 0.0292454$$

An accurate inverse involute is usually obtained by a computer iteration process.

$$\phi_e = 24.80595^\circ$$

$$C_e = 16.5000 \frac{\cos 20^\circ}{\cos 24.80595^\circ}$$

$$C_e = 17.08092$$

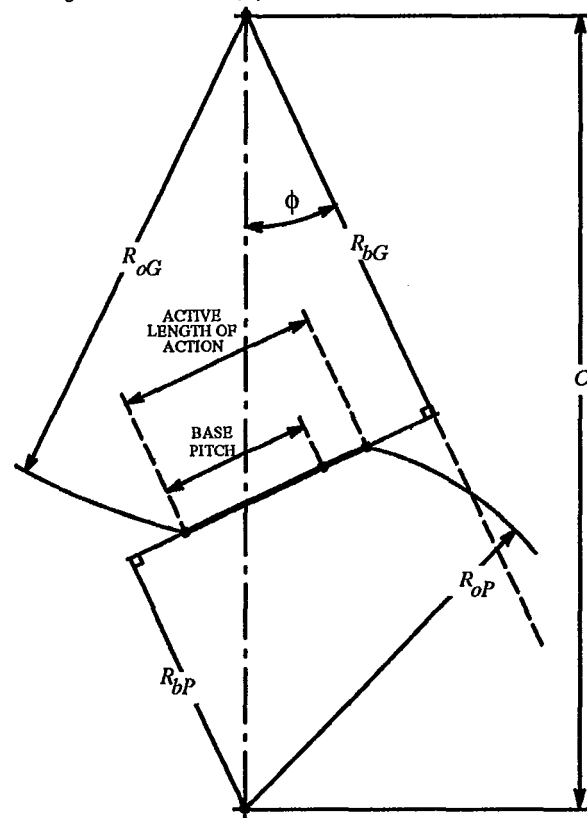
The tight mesh center distance from the above equation is dimensionless. To find the center distance in inches, divide by the diametral pitch, P_d . To find the center distance in millimeters, multiply by the module, m_{et} .

4.16.1 Example (For 20 diametral pitch).

$$C_e = \frac{17.08092}{20} = 0.8540 \text{ in}$$

4.16.2 Example (For 1 module).

$$C_e = 17.08092 (1) = 17.08092 \text{ mm}$$



where

R_{oP} , R_{oG} (r_{e1} , r_{e2}) is outside radius of pinion, gear,

R_{bP} , R_{bG} (r_{b1} , r_{b2}) is base radius of pinion, gear.

Figure 4 – Center distance and line of action

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Annex A (informative)

Tooth proportions for 14-1/2 degree fine-pitch gearing

[The foreword, footnotes, and annexes are provided for informational purposes only, and should not be construed as a part of ANSI/AGMA 1003-G93, *Tooth Proportions for Fine-Pitch Spur and Helical Gearing*.]

Table A.1 – 14-1/2° Profile angle — enlarged spur pinions

Enlarged pinion dimensions (unit pitch)						Enlarged center distance, pinion with 51 tooth gear		
1	2	3	4	5	6	7	8	9
Number of teeth $N_P (z_1)$	Outside diameter (enlarged) $D_{oP} (d_{e1})$	Addendum (enlarged) $a_P (h_{a1})$	Tooth thickness (enlarged) $t_P (s_1)$	Root diameter* (enlarged) $D_r (d_f)$	Top land $t_{oP} (s_{e1})$	Contact ratio $m_{ce} (ε_{βx})$	Clearance gear tip to pinion root** $c_e (c_p)$	Center distance $C_e (a_d)$
11	14.3375	1.6688	1.99588	10.2436	0.2750	1.376	0.0911	31.71286
12	15.3830	1.6915	1.98516	11.2022	0.2750	1.429	0.0968	32.19793
13	16.4241	1.7120	1.97443	12.1608	0.2750	1.479	0.1024	32.68279
14	17.4614	1.7307	1.96370	13.1192	0.2750	1.527	0.1079	33.16748
15	18.4778	1.7389	1.95297	14.0778	0.2884	1.570	0.1131	33.65199
16	19.4363	1.7181	1.94224	15.0363	0.3418	1.595	0.1182	34.13632
17	20.3948	1.6974	1.93152	15.9948	0.3898	1.620	0.1231	34.62049
18	21.3533	1.6767	1.92079	16.9533	0.4333	1.644	0.1278	35.10448
19	22.3118	1.6559	1.91006	17.9118	0.4729	1.666	0.1324	35.58829
20	23.2704	1.6352	1.89933	18.8704	0.5090	1.688	0.1368	36.07195
21	24.2289	1.6144	1.88860	19.8289	0.5423	1.708	0.1410	36.55543
22	25.1874	1.5937	1.87788	20.7874	0.5729	1.728	0.1451	37.03878
23	26.1459	1.5730	1.86715	21.7459	0.6013	1.748	0.1490	37.52195
24	27.1044	1.5522	1.85642	22.7044	0.6275	1.766	0.1528	38.00496
25	28.0629	1.5315	1.84569	23.6629	0.6520	1.784	0.1564	38.48781
26	29.0215	1.5107	1.83496	24.6215	0.6747	1.801	0.1598	38.97051
27	29.9800	1.4900	1.82423	25.5800	0.6960	1.818	0.1631	39.45306
28	30.9385	1.4692	1.81351	26.5385	0.7160	1.835	0.1662	39.93547
29	31.8970	1.4485	1.80278	27.4970	0.7347	1.851	0.1692	40.41772
30	32.8555	1.4278	1.79205	28.4555	0.7523	1.866	0.1721	40.89983
31	33.8140	1.4070	1.78132	29.4140	0.7689	1.881	0.1748	41.38178
32	34.7726	1.3863	1.77059	30.3726	0.7844	1.896	0.1773	41.86360
33	35.7311	1.3655	1.75987	31.3311	0.7991	1.911	0.1797	42.34528
34	36.6896	1.3448	1.74914	32.2896	0.8130	1.925	0.1820	42.82681
35	37.6481	1.3241	1.73841	33.2481	0.8262	1.939	0.1842	43.30820
36	38.6066	1.3033	1.72768	34.2066	0.8386	1.952	0.1862	43.78945
37	39.5651	1.2826	1.71695	35.1651	0.8504	1.966	0.1880	44.27056
38	40.5237	1.2618	1.70623	36.1237	0.8616	1.979	0.1897	44.75155
39	41.4822	1.2411	1.69550	37.0822	0.8722	1.992	0.1913	45.23239
40	42.4407	1.2204	1.68477	38.0407	0.8823	2.005	0.1927	45.71309
41	43.3992	1.1996	1.67404	38.9992	0.8919	2.018	0.1941	46.19366
42	44.3577	1.1789	1.66331	39.9577	0.9010	2.030	0.1952	46.67409
43	45.3163	1.1581	1.65259	40.9163	0.9096	2.042	0.1963	47.15442
44	46.2748	1.1374	1.64186	41.8748	0.9178	2.055	0.1972	47.63459
45	47.2333	1.1166	1.63113	42.8333	0.9256	2.067	0.1980	48.11463
46	48.1918	1.0959	1.62040	43.7918	0.9331	2.079	0.1986	48.59454
47	49.1503	1.0752	1.60967	44.7503	0.9402	2.091	0.1992	49.07432
48	50.1088	1.0544	1.59894	45.7088	0.9470	2.103	0.1996	49.55397
49	51.0674	1.0337	1.58822	46.6674	0.9534	2.115	0.1998	50.03352
50	52.0259	1.0129	1.57749	47.6259	0.9595	2.126	0.2000	50.51292
51	53.0000	1.0000	1.57080	48.6000	0.9637	2.135	0.2000	51.00000

NOTE – Table is dimensionless. Divide values in columns 2, 3, 4, 5 and 6 by diametral pitch for inch units, or multiply by module for millimeter units.

NOTE – Table values for contact ratio are for tight mesh conditions, with no allowance for tooth thinning. Contact ratio is computed for actual mesh and limit diameters. Columns 6 through 9 are for reference only and not to be specified on drawings.

NOTE – Enlarged pinions are designed to use the involute form above 5° of roll.

* Actual root diameter is decreased by 0.004 in (0.10 mm).

** Actual clearance is increased by 0.002 in (0.05 mm).

Table A.2 – 14-1/2 degree profile angle – reduced spur gears

Reduced gear dimensions (unit pitch)						Standard center distance $C = 51.0000$	
1	2	3	4	5	6	7	8
Minimum no. of teeth in gear $N_G (z_2)$	Outside diameter (reduced) $D_{oG} (d_{e2})$	Addendum (reduced) $a_G (h_{a2})$	Tooth thickness (reduced) $t_G (s_2)$	Root diameter* (reduced) $D_r (d_f)$	Top land $t_{oG} (s_{ei2})$	Contact ratio N_P mating with N_G $m_c (\epsilon_\beta)$	Number of teeth in pinion $N_P (z_1)$
91	91.3563	0.1782	1.14571	86.9563	1.0550	1.353	11
90	90.3978	0.1989	1.15644	85.9978	1.0549	1.417	12
89	89.4393	0.2196	1.16716	85.0393	1.0547	1.478	13
88	88.4808	0.2404	1.17789	84.0808	1.0544	1.537	14
87	87.5222	0.2611	1.18862	83.1222	1.0540	1.590	15
86	86.5637	0.2819	1.19935	82.1637	1.0535	1.625	16
85	85.6052	0.3026	1.21008	81.2052	1.0530	1.659	17
84	84.6467	0.3233	1.22081	80.2467	1.0524	1.690	18
83	83.6882	0.3441	1.23153	79.2882	1.0517	1.721	19
82	82.7296	0.3648	1.24226	78.3296	1.0509	1.750	20
81	81.7711	0.3856	1.25299	77.3711	1.0500	1.777	21
80	80.8126	0.4063	1.26372	76.4126	1.0490	1.803	22
79	79.8541	0.4270	1.27445	75.4541	1.0480	1.828	23
78	78.8956	0.4478	1.28517	74.4956	1.0468	1.851	24
77	77.9371	0.4685	1.29590	73.5371	1.0455	1.874	25
76	76.9785	0.4893	1.30663	72.5785	1.0441	1.895	26
75	76.0200	0.5100	1.31736	71.6200	1.0426	1.915	27
74	75.0615	0.5308	1.32809	70.6615	1.0410	1.934	28
73	74.1030	0.5515	1.33881	69.7030	1.0393	1.952	29
72	73.1445	0.5722	1.34954	68.7445	1.0375	1.970	30
71	72.1860	0.5930	1.36027	67.7860	1.0355	1.986	31
70	71.2274	0.6137	1.37100	66.8274	1.0334	2.001	32
69	70.2689	0.6345	1.38173	65.8689	1.0312	2.015	33
68	69.3104	0.6552	1.39245	64.9104	1.0289	2.029	34
67	68.3519	0.6759	1.40318	63.9519	1.0264	2.041	35
66	67.3934	0.6967	1.41391	62.9934	1.0237	2.053	36
65	66.4349	0.7174	1.42464	62.0349	1.0209	2.064	37
64	65.4763	0.7382	1.43537	61.0763	1.0180	2.074	38
63	64.5178	0.7589	1.44610	60.1178	1.0149	2.083	39
62	63.5593	0.7796	1.45682	59.1593	1.0116	2.092	40
61	62.6008	0.8004	1.46755	58.2008	1.0081	2.099	41
60	61.6423	0.8211	1.47828	57.2423	1.0045	2.106	42
59	60.6837	0.8419	1.48901	56.2837	1.0007	2.113	43
58	59.7252	0.8626	1.49974	55.3252	0.9966	2.118	44
57	58.7667	0.8834	1.51046	54.3667	0.9924	2.123	45
56	57.8082	0.9041	1.52119	53.4082	0.9879	2.126	46
55	56.8497	0.9248	1.53192	52.4497	0.9833	2.130	47
54	55.8912	0.9456	1.54265	51.4912	0.9784	2.132	48
53	54.9326	0.9663	1.55338	50.5326	0.9732	2.134	49
52	53.9741	0.9871	1.56410	49.5741	0.9678	2.135	50
51	53.0000	1.0000	1.57080	48.6000	0.9637	2.135	51

NOTE – Table is dimensionless. Divide values in columns 2, 3, 4, 5 and 6 by diametral pitch for inch units, or multiply by module for millimeter units.

NOTE – Table values for contact ratio are for tight mesh conditions, with no allowance for tooth thinning. Contact ratio is computed for tight mesh and limit diameters. Columns 6 through 8 are for reference only, not to be specified on drawing.

* Actual root diameter is decreased by 0.004 in (0.10 mm).

Annex B

(informative)

Tooth proportions for 25 degree fine-pitch gearing

[The foreword, footnotes, and annexes are provided for informational purposes only, and should not be construed as a part of ANSI/AGMA 1003-G93, *Tooth Proportions for Fine-Pitch Spur and Helical Gearing*.]

Table B.1 – 25 degree profile angle – enlarged spur pinions

Enlarged pinion dimensions (unit pitch)						Enlarged center distance, pinion with 15 tooth gear		
1	2	3	4	5	6	7	8	9
Number of teeth	Outside diameter (enlarged)	Addendum (enlarged)	Tooth thickness (enlarged)	Root diameter* (enlarged)	Top land	Contact ratio	Clearance gear tip to pinion root**	Center distance
$N_P (z_1)$	$D_{OP} (d_{el})$	$a_P (h_{a1})$	$t_P (s_1)$	$D_r (d_f)$	$t_{OP} (s_{el})$	$m_{ce} (\epsilon_{\beta x})$	$c_e (c_p)$	$C_e (a_d)$
8	10.6631	1.3316	2.00877	6.5392	0.2750	1.123	0.1654	11.93497
9	11.6203	1.3102	1.94111	7.3942	0.2750	1.174	0.1753	12.37237
10	12.5691	1.2846	1.87345	8.2490	0.2750	1.223	0.1836	12.80806
11	13.5040	1.2520	1.80579	9.1040	0.2807	1.269	0.1901	13.24207
12	14.3588	1.1794	1.73813	9.9588	0.3478	1.294	0.1950	13.67440
13	15.2138	1.1069	1.67047	10.8138	0.4034	1.319	0.1982	14.10509
14	16.0686	1.0343	1.60281	11.6686	0.4500	1.343	0.1999	14.53414
15	17.0000	1.0000	1.57080	12.6000	0.4743	1.358	0.2000	15.00000

NOTE – Table is dimensionless. Divide values in columns 2, 3, 4, 5, and 6 by diametral pitch for inch units, or multiply by module for millimeter units.

NOTE – Table values for contact ratio are for tight mesh conditions, with no allowance for tooth thinning. Contact ratio is computed for actual mesh and limit diameters. Columns 6 through 9 are for reference only and not to be specified on drawings.

NOTE – Enlarged pinions are designed to use the involute form above 5° of roll.

* Actual root diameter is decreased by 0.004 in (0.10 mm).

** Actual clearance is increased by 0.002 in (0.05 mm).

Table B.2 – 25 degree profile angle – reduced spur gears

Reduced gear dimensions (unit pitch)						Standard center distance $C = 15.000$	
1	2	3	4	5	6	7	8
Minimum no. of teeth in gear N_G (z_2)	Outside diameter (reduced) D_{oG} (d_{e2})	Addendum (reduced) a_G (h_{a2})	Tooth thickness (reduced) t_G (s_2)	Root diameter* (reduced) D_r (d_f)	Top land t_{oG} (s_{ei2})	Contact ratio, N_P mating with N_G m_c (ϵ_β)	Number of teeth in pinion N_P (z_1)
22	23.0608	0.5304	1.13283	18.6568	0.6174	1.181	8
21	22.2059	0.6029	1.20049	17.8059	0.6058	1.236	9
20	21.3510	0.6755	1.26814	16.9510	0.5912	1.284	10
19	20.4961	0.7480	1.33580	16.0961	0.5734	1.325	11
18	19.6412	0.8206	1.40346	15.2412	0.5519	1.341	12
17	18.7862	0.8931	1.47112	14.3862	0.5261	1.351	13
16	17.9313	0.9657	1.53878	13.5313	0.4955	1.357	14
15	17.0000	1.0000	1.57080	12.6000	0.4743	1.358	15

NOTE – Table is dimensionless. Divide values in columns 2, 3, 4, 5, and 6 by diametral pitch for inch units, or multiply by module for millimeter units.

NOTE – Table values for contact ratio are for tight mesh conditions, with no allowance for tooth thinning. Contact ratio is computed for tight mesh and limit diameters. Columns 6 through 8 are for reference only, not to be specified on drawing.

* Actual root diameter is decreased by 0.004 in (0.10 mm).

Annex C

(informative)

Helical pinion enlargement

[The foreword, footnotes, and annexes are provided for informational purposes only, and should not be construed as a part of ANSI/AGMA 1003-G93, *Tooth Proportions for Fine-Pitch Spur and Helical Gearing*.]

C.1 Helical gears

Helical gears may be enlarged and reduced in the same manner as spur gears, but caution must be observed in the translation for tooth thickness and top land. Since enlargement is based upon the use of standard tools, made to standard proportions in the normal plane, conversions must be made for transverse plane data and vice versa.

Since tabular data is not practical for the numerous combinations of helical gears, it is necessary to compute center distance, clearance, and contact ratio for each combination. All data would be in accordance with equations in tables 2 or 2M.

C.1.1 Example (standard helical pinion)

For a 23 degree helix angle and a 20 degree normal pressure angle, the minimum number of teeth not requiring enlargement is 20. See table C.1 and equation 20.

$$\begin{aligned}\phi &= \tan^{-1} \frac{\tan \phi_n}{\cos \psi} \\ \phi &= \tan^{-1} \frac{\tan 20^\circ}{\cos 23^\circ} \\ &= 0.39540 \text{ (tangent)} \\ &= 21.57398^\circ\end{aligned}$$

For 20 normal diametral pitch:

$$\begin{aligned}D &= \frac{N}{P_{nd} \cos \psi} = \frac{20}{20 \cos 23^\circ} \\ &= 1.08636 \\ t_n &= \frac{\pi}{2 P_{nd}} = 0.07854 \\ t &= \frac{t_n}{\cos \psi} = 0.08532\end{aligned}$$

where

- ϕ is profile angle, transverse;
- ϕ_n is profile angle, normal;
- ψ is helix angle;
- D is pitch diameter;
- t_n is tooth thickness, normal;
- t is tooth thickness, transverse.

Computations are made in the transverse plane for a summation of tooth thicknesses with the mating gear to obtain the tight mesh center distance, clearance, and contact ratio.

C.1.2 Enlarged helical pinion

When dealing with a pinion having fewer teeth than shown in table C.1, computation should be made for the 5 degree requirement and also an investigation of the undercut must be performed. Detailed procedures for this are best left to a design manual, and are considered beyond the scope of this standard.

Table C.1 – Enlargement criteria*

14.50°			20°			25°		
Helix angle, degrees $\psi (\beta_m)$	Trans. profile angle, degrees $\phi_t (\alpha_{pt})$	Number of teeth** $N_e (z_g)$	Helix angle, degrees $\psi (\beta_m)$	Trans. profile angle, degrees $\phi_t (\alpha_{pt})$	Number of teeth** $N_e (z_g)$	Helix angle, degrees $\psi (\beta_m)$	Trans. profile angle, degrees $\phi_t (\alpha_{pt})$	Number of teeth** $N_e (z_g)$
0	14.50000	50.62	0	20.00000	23.63	0	25.00000	14.47
5	14.55304	50.17	5	20.07031	23.45	5	25.08377	14.37
10	14.71405	48.81	10	20.28356	22.89	10	25.33761	14.07
15	14.98885	46.63	15	20.64690	22.00	15	25.76926	13.57
18	15.21241	44.97	18	20.94190	21.31	18	26.11894	13.19
20	15.38771	43.73	20	21.17283	20.79	20	26.39218	12.90
23	15.69281	41.69	23	21.57398	19.95	23	26.86578	12.43
25	15.92625	40.22	25	21.88023	19.33	25	27.22644	12.09
30	16.62699	36.28	30	22.79588	17.67	30	28.30005	11.16
35	17.52162	32.05	35	23.95680	15.86	35	29.65098	10.14
40	18.65475	27.70	40	25.41377	13.98	40	31.32977	9.07
45	20.08951	23.40	45	27.23631	12.08	45	33.40320	7.99

* Based on 5° minimum roll angle at form diameter.

** N_G is a calculated value (see equation 18). Pinions with numbers of teeth that exceed this value do not require enlargement.

Annex D (informative)

Calculations to obtain standard clearance for enlarged pinions and standard gears

[The foreword, footnotes, and annexes are provided for informational purposes only, and should not be construed as a part of ANSI/AGMA 1003-G93, *Tooth Proportions for Fine-Pitch Spur and Helical Gearing*.]

D.1 Introduction

When using enlarged pinions and standard gears, special attention must be paid to providing adequate clearance since the computed tight mesh distance does not provide the standard clearance. An additional increase in the center distance may be required which results in a slight backlash in the mesh. This approach, however, reduces the contact ratio.

D.2 Clearance

Note that column 8 in table 4 indicates the clearance for the tight mesh condition. Since the clearance is less than 0.200 (for unit pitch), the working center distance should be increased to obtain the desired clearance, or the outside diameter of the reduced gear can be reduced further without tooth thinning adjustment.

D.3 Center distance (working)

The minimum center distance is determined by the requirement for clearance, and is obtained by use of the following equation. This approach also reduces the contact ratio. (See table D.1)

$$C_w = \frac{D_{oG} + D_{rP}}{2} + 0.20 \quad \dots(D.1)$$

$$a_w = \frac{d_{e2} + d_{f1}}{2} + 0.20 \quad \dots(D.1M)$$

where

- C_w is center distance, working;
- D_{oG} is outside diameter of gear;
- D_{rP} is root diameter of pinion.

Using the example from 4.16, where $N_P = 9$ and $N_G = 24$:

$$C_w = \frac{26 + 7.9003}{2} + 0.20 = 17.15015$$

This answer is dimensionless. To obtain the actual number, divide by the diametral pitch, or multiply by the module.

D.4 Backlash (minimum)

A minimum backlash accrues even without any consideration for requirements by the necessity to provide clearance. The resulting minimum backlash is determined by the following equation.

$$B = \frac{(\pi C_w)}{N_P + N_G} - \frac{\pi}{2} \quad \dots(D.2)$$

$$j = \frac{(\pi a_w)}{z_1 + z_2} - \frac{\pi}{2} \quad \dots(D.2M)$$

$$B = \frac{(\pi 17.15015)}{9 + 24} - \frac{\pi}{2} = 0.06189$$

Table D.1 gives dimensionless (unit pitch) values of minimum backlash for meshes of enlarged pinions with a 24 tooth standard gear.

D.4.1 Backlash (20 diametral pitch)

$$B = \frac{0.06189}{20} = 0.00309, \text{ in}$$

D.4.2 Backlash (1 module)

$$B = (0.06189) (1) = 0.062, \text{ mm}$$

D.5 Alternate (reduced gear)

In meshes with insufficient clearance, the outside diameter of the gear can be further reduced without a corresponding reduction in tooth thickness. This approach also reduces the contact ratio.

$$D_{oG} = (2(C_e - 0.200)) - D_{rP} \quad \dots(D.3)$$

$$d_{e2} = (2(a_d - 0.200)) - d_{f1} \quad \dots(D.3M)$$

$$= (2(17.08092 - 0.200)) - 7.9003$$

$$= 25.8615$$

Table D.1 –
Working center distance,* enlarged pinion
with 24 tooth standard gear
(20 degree profile angle, unit pitch)

$NP(z_1)$	$m_{cw} (\epsilon\beta_w)$	$C_{wmin} (a_{wmin})$	$B (j)$
9	1.154	17.15015	0.06189
10	1.212	17.60570	0.05597
11	1.268	18.06125	0.05038
12	1.288	18.55785	0.04868
13	1.353	18.97240	0.04011
14	1.381	19.42800	0.03538
15	1.408	19.88355	0.03090
16	1.434	20.33915	0.02664
17	1.459	20.79470	0.02258
18	1.482	21.25025	0.01872
19	1.505	21.70585	0.01504
20	1.527	22.16140	0.01152
21	1.548	22.61700	0.00817
22	1.568	23.07255	0.00495
23	1.588	23.52810	0.00188
24	1.602	24.00000	0.00000

*Based on standard clearance $(0.200/P_d)$ and data from tables 4 and 5.

NOTE – Table is dimensionless. Divide third and fourth columns by diametral pitch for inch units, or multiply by module for millimeter units.

D.6 Contact ratio (working)

$$m_{cw} = \frac{\sqrt{D_{oP}^2 - (D_P \cos \phi)^2} + \sqrt{D_{oG}^2 - (D_G \cos \phi)^2}}{2 p \cos \phi} - \frac{2 C_w \sin \phi_w}{2 p \cos \phi} \quad \dots(D.4)$$

$$\epsilon\beta_w = \frac{\sqrt{d_{e1}^2 - (d_1 \cos \alpha)^2} + \sqrt{d_{e2}^2 - (d_2 \cos \alpha)^2}}{2 p_t \cos \alpha} - \frac{2 a_w \sin \alpha_w}{2 p_t \cos \alpha} \quad \dots(D.4M)$$

where

$m_{cw} (\epsilon\beta_w)$ is contact ratio at working center distance;

$\phi_w (\alpha_w)$ is pressure angle, working.

Annex E
(informative)
References and bibliography

[The foreword, footnotes, and annexes are provided for informational purposes only, and should not be construed as a part of ANSI/AGMA 1003-G93, *Tooth Proportions for Fine-Pitch Spur and Helical Gearing*.]

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