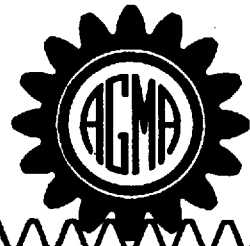


AMERICAN GEAR MANUFACTURERS ASSOCIATION

***A Rational Procedure for the Preliminary
Design of Minimum Volume Gears***



AGMA INFORMATION SHEET

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AGMA 901-A92, A Rational Procedure for the Preliminary Design of Minimum Volume Gears

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ABSTRACT:

A simple, closed-form procedure is presented as a first step in the design of minimum volume spur and helical gearsets. The procedure includes methods for selecting geometry and dimensions, considering maximum pitting resistance, bending strength, and scuffing resistance. It also includes methods for selecting profile shift.

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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 AGMA 901-A92, *A Rational Procedure for the Preliminary Design of Minimum Volume Gears*.]

Gear design is a process of synthesis where gear geometry, materials, heat treatment, manufacturing methods, and lubrication are selected to meet the performance requirements of a given application. The designer must design the gearset with adequate pitting resistance, bending strength, and scuffing resistance to transmit the required power for the design life. With the algorithm presented here, one can select materials and heat treatment within the economic constraints and limitations of manufacturing facilities, and select the gear geometry to satisfy constraints on weight, size and configuration. The gear designer can minimize noise level and operating temperature by minimizing the pitchline velocity and sliding velocity. This is done by specifying high gear accuracy and selecting material strengths consistent with maximum material hardness, to obtain minimum volume gearsets with teeth no larger than necessary to balance the pitting resistance and bending strength.

Gear design is not the same as gear analysis. Existing gearsets can only be analyzed, not designed. While design is more challenging than analysis, current textbooks do not provide procedures for designing minimum volume gears. They usually recommend that the number of teeth in the pinion be chosen based solely on avoiding undercut. This information sheet, based on the work of R. Errichello [1]*, will show why this practice, or any procedure which arbitrarily selects the number of pinion teeth, will not necessarily result in minimum volume gearsets. Although there have been many technical papers on gear design (for example [2] and [3]), most advocate using computer-based search algorithms which are unnecessary. Tucker [4] came the closest to an efficient algorithm, but he did not show how to find the preferred number of teeth for the pinion.

This information sheet includes the design of spur and helical gears. Other gear types could be designed by a similar algorithm with some slight modifications to the one presented here.

AGMA 901-A92 was approved by the Helical Gear Rating Committee in March, 1992 and approved by the AGMA Board of Directors as of May, 1992.

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

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

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A Rational Procedure for the Preliminary Design of Minimum Volume Gears

1 Scope

This information sheet is intended for the student or beginning gear designer, to provide an outline of a preliminary design procedure which will lead to a rational design for spur and helical gear pairs within constraints such as:

- required gear ratio;
- required torque capacity;
- specified center distance;
- material selection.

This method could be extended to other gear types given the appropriate constants and factors.

1.1 Procedure

The simple closed form of the procedure allows the designer to explore options with a minimum of computation so that the important design decisions regarding loads, overloads, material, and tooling selections are not obscured by the need to spend a long time calculating each possibility.

This information sheet will demonstrate to the user that the traditional beginning point for gear design, selecting the minimum number of pinion teeth to avoid undercut, will rarely lead to the best design.

As this procedure is approximate, it is necessary to audit the design (see clause 6).

1.2 Exceptions

The procedure described herein incorporates major design considerations and leads toward minimum volume gear designs based upon the criteria chosen. For the final gear design, additional influencing factors beyond those in this information sheet include shaft deflection limits, sound level, cost, etc. Any of these could influence the design of the gear envelope and final volume.

It is not the intent of this information sheet to include the calculation of the profile shift coefficient (addendum modification coefficient). It is, however, necessary to inform the reader that profile shift exists, how it can affect gear design, and where it comes into play in designing a gearset. Some of the important factors relating to profile shift are discussed in 7.4.

Overhung pinions or gears are not covered by this information sheet because of the difficulty in determining an accurate value for the load distribution factor.

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*

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

AGMA 904–B89, *Metric Usage*

2.2 Symbols

The symbols used in this information sheet are shown in table 1.

NOTE—The symbols, definitions and terminology used in this information sheet may differ from other AGMA publications. The user should not assume that familiar symbols can be used without a careful study of these definitions.

3 Input variables

This clause discusses the significant parameters relating to a preferred gear design. It is not intended to be all inclusive, but to be limited by the scope of this information sheet.

3.1 Materials and heat treatment

Many materials have been used in gearing, but the most common today is steel. This information sheet only applies to steel gearing. There are two commonly used types of heat treatment for steel gear materials, surface hardening and through hardening. The choice of steel alloy must be compatible with the chosen heat treatment process.

Table 1 – Symbols used in equations

Symbols		Terms	Units	Equation where first used
AGMA	ISO			
b	N_b	number of power paths	—	8
C_a	K_A	application factor – pitting	—	9
C_d	Z_d	combined derating factor – pitting	—	30
C_L	Z_{NT}	pitting resistance life factor	—	26
C_m	K_H	load distribution factor – pitting	—	9
C_p	Z_E	elastic coefficient	$[\text{lb/in}^2]^{0.5} ([\text{N/mm}^2]^{0.5})$	32
C_r	a_w	operating center distance	in (mm)	21
C_v	K_v	dynamic factor – pitting	—	30
d	d_1	operating pitch diameter of pinion	in (mm)	10
F	b	net face width (without gap for double helical)	in (mm)	25
H_B	H_B	Brinell hardness	—	1
I	Z_I	pitting resistance geometry factor	—	11
J	Y_J	bending strength geometry factor	—	12
K_a	K_A	application factor – bending	—	31
K_B	Y_B	rim thickness factor	—	31
K_c	Z_c	pitting resistance constant	$\text{in}^3 (\text{mm}^3)$	32
K_d	Y_d	combined derating factor – bending	—	31
K_L	Y_{NT}	bending strength life factor	—	27
K_m	K_H	load distribution factor – bending	—	9
K_t	Y_t	bending strength constant	$\text{in}^3 (\text{mm}^3)$	33
K_v	*	dynamic factor – bending	—	31
L	L	gear life	hours	3
m_a	u_b	aspect (F/d) ratio	—	4
m_F	ϵ_β	face contact ratio	—	25
m_G	u	gear ratio ($m_G \geq 1$)	—	4
m_{G1}	u_1	gear ratio of high speed gearset ($m_{G1} \geq 1$)	—	15
—	m_n	normal module	mm	25M
M_o	u_o	overall gear ratio of double stage gear drive ($M_o \geq 1$)	—	15
N	N_L	total number of load cycles in gear life	—	3
N_G	z_2	number of teeth in gear	—	6
N_P	z_1	number of teeth in pinion	—	6
$N_{P \text{ pre}}$	$z_{1 \text{ pre}}$	preferred number of pinion teeth	—	34
n	ω_1	speed	rpm	3
n_c	S_H	pitting resistance factor of safety	—	32
n_t	S_F	bending strength factor of safety	—	33
P	P	input power	hp (kW)	7
P_{nd}	—	normal diametral pitch	in^{-1}	25

* The symbol K_v is also used for the dynamic factor in ISO standards. However, its value is the inverse (equal to $1/K_v$) of the value used in ANSI/AGMA 2001-B88.

continued

Table 1 (concluded)

Symbols		Terms	Units	Equation where first used
AGMA	ISO			
q	N_q	number of contacts per revolution	—	3
s_{ac}	σ_{HP}	allowable contact stress number	lb/in ² (N/mm ²)	1
s_{at}	σ_{FP}	allowable bending stress number	lb/in ² (N/mm ²)	2
s_{nc}	σ_{HN}	contact strength	lb/in ² (N/mm ²)	28
s_{nt}	σ_{FN}	bending strength	lb/in ² (N/mm ²)	29
T_1	T_1	torque on high speed shaft	in lbs (Nm)	7
T_P	T_P	transmitted pinion torque, per mesh	in lbs (Nm)	8
ϕ_n	α_{no}	normal profile angle of cutter	degrees	11
ψ	β_m	standard helix angle	degrees	25
Subscripts/Sign convention				
1	high speed (pinion)			
2	low speed (wheel)			
n	normal (no subscript indicates transverse)			
G	gear (wheel)			
P	pinion			
r	operating or running			
(±)	upper sign external gearsets, lower sign internal gearsets			

3.1.1 Surface hardening

Surface hardening takes place after tooth cutting, usually on gears made from hot rolled bar or forged steel.

3.1.1.1 Carburized

Carburized steel is most commonly used for highly loaded, compact designs such as aircraft gears, vehicle transmissions of all types, machine tools, industrial gear drives, mining machines and similar uses. This material has the highest strength and greatest overload capacity, but carburized gears must be carefully manufactured. Carburized gears often result in the least expensive overall transmission design, if their advantage in small size for a given capacity can be utilized. Few manufacturers can produce carburized gears larger than 40 inches in diameter, though some can make them over 100 inches in diameter. Secondary finishing operations after carburizing, such as tooth grinding, are often required to achieve the desired tooth form. This is often required to eliminate the distortions caused by heating and cooling employed in the carburizing process.

3.1.1.2 Nitrided

Nitrided steel is most commonly used for small gears, finer than 10 diametral pitch (2.5 module) because the maximum case depth is limited. Some large gears are nitrided to avoid the distortion inherent in the carburizing process. Typical applications are industrial gear drives and small machine tools. Nitrided gears have limited shock resistance. This information sheet does not address nitrided gears, as reference [5] does not provide life factor curves for nitrided gears.

3.1.1.3 Induction and flame hardened

Induction and flame hardened steels are used to achieve intermediate capacities between carburized and through hardened gears. These processes are difficult to control, but give good results when carefully controlled. This information sheet does not address induction or flame hardened gears, as they are not recommended for inexperienced designers.

3.1.2 Through hardening

Through hardened gears typically have teeth cut in pre-heat-treated gear blanks, with no further heat treatment after cutting. The raw material can be hot

rolled, cast, or forged. Hardness is chosen on the basis of machinability, using the lowest hardness which will carry the load on the required center distance.

The allowable stress numbers shall be based on the lowest hardness in the heat treatment specifications. Typical heat treatment specifications have a 40 BHN tolerance between the minimum and maximum hardnesses. The hardest heat treatment range that can be machined without special techniques is 320–360 BHN. The normal lowest range of hardness is 180–220 BHN, because lower values are difficult to machine.

Through hardened gear sizes commonly range from less than one inch to over 20 feet in diameter. Typical applications vary from instrument gears to girth gears on large mills and kilns. When gears cannot be of minimum size because of required center distance, rigidity requirements or thermal limits, or when loads are low, through hardened

gears are commonly used. Internal gears are often through hardened.

The selection of a proper alloy for hardenability and reliability as well as the quality control of the steel manufacturing and heat treatment process are beyond the scope of this information sheet. Guidance can be found in section 14 of [5], as well as [6] and [7].

3.1.3 Elastic coefficient, C_p

The rating of gears also depends on the elastic coefficient, C_p . Further information can be found in section 10 of [5]. The elastic coefficient for a steel pinion and gear is $2300[\text{lbs/in}^2]^{0.5}$ ($191[\text{N/mm}^2]^{0.5}$).

3.1.4 Allowable stress numbers

The allowable stress numbers for some heat treatments, surface hardness and steel quality grades are shown in tables 2 and 3. There are two grades of allowables shown in tables 2 and 3. The allowable stress numbers are valid only when the requirements of ANSI/AGMA 2001-B88, section 14 are met.

Table 2 – Allowable contact stress numbers for steel gears¹

Material designation	Heat treatment	Minimum hardness at surface	Allowable contact stress number, s_{ac} lb/in ² (N/mm ²)	
			Grade 1	Grade 2
Steel	Through hardened	180 BHN and less	85 000 (590)	95 000 (660)
		240 BHN	105 000 (720)	115 000 (790)
		300 BHN	120 000 (830)	135 000 (930)
		360 BHN	145 000 (1000)	160 000 (1100)
		400 BHN	155 000 (1050)	170 000 (1150)
	Carburized and case hardened		180 000 (1250)	225 000 (1550)

1) The data in this table has been extracted from ANSI/AGMA 2001-B88. The metric values have been revised per AGMA 904-B89. The allowable stress numbers are valid only when the requirements of ANSI/AGMA 2001-B88, section 14 are met.

Table 3 – Allowable bending stress numbers for steel gears¹

Material designation	Heat treatment	Minimum hardness at surface	Allowable bending stress number, s_{at} lb/in ² (N/mm ²)	
			Grade 1	Grade 2
Steel	Through hardened	180 BHN and less	25 000 (170)	33 000 (230)
		240 BHN	31 000 (215)	41 000 (285)
		300 BHN	36 000 (250)	47 000 (325)
		360 BHN	40 000 (275)	52 000 (360)
		400 BHN	42 000 (290)	56 000 (385)
	Carburized and case hardened		55 000 (380)	65 000 (450)
1) The data in this table has been extracted from ANSI/AGMA 2001–B88. The metric values have been revised per AGMA 904–B89. The allowable stress numbers are valid only when the requirements of ANSI/AGMA 2001–B88, section 14 are met.				

Allowable stress numbers for grade 1 through hardened steel for unity life factor are:

$$s_{ac} = 26\,000 + 327 H_B \quad \dots(1)$$

$$\sigma_{HP} = 179 + 2.25 H_B \quad \dots(1M)$$

$$s_{at} = -274 + 167 H_B - 0.152 H_B^2 \quad \dots(2)$$

$$\sigma_{FP} = -1.89 + 1.15 H_B - 0.00105 H_B^2 \quad \dots(2M)$$

where

s_{ac} (σ_{HP}) is allowable contact stress number in pounds per square inch (newtons per square millimeter);

s_{at} (σ_{FP}) is allowable bending stress number in pounds per square inch (newtons per square millimeter);

H_B is Brinell hardness.

3.2 Design life

When evaluating gearing, it is important to know how many stress cycles the individual gears will experience during the intended life of the equipment. Some machines will run twenty-four hours per day and operate for twenty or more years. Other machines have gears that have total life requirements of a few hours. The gear designer

should obtain the design life that is appropriate for the application. The required life in load cycles, N , will be used to determine the life factors.

$$N = 60 L n q \quad \dots(3)$$

where

N is the total number of load cycles in the gear life;

L is the life, in hours;

n is the speed, in rpm;

q is the number of contacts per revolution.

3.3 Aspect ratio

The aspect ratio, m_a , also known as the pinion face width to diameter ratio, F/d , is an indicator of how sensitive a gear set is to misalignment. In this algorithm the aspect ratio is input, rather than the face width. The ratio used will affect the value for C_m and K_m . This is because changes in the aspect ratio change the face width of a gearset. This will, in turn, increase or decrease the effect of any mounting errors or deflections under load. For this reason, as one changes the aspect ratio, one must appropriately alter C_m and K_m .

Opinions vary regarding what is good design practice for an aspect ratio. Some factors that influence one's selection are ratio, materials, shaft deflection,

housing deflection, housing accuracy, bearing clearances, centrifugal and thermal deflections, and tooth crowning. In the absence of experience, equations 4 and 5 may be used. These will result in aspect ratios which will be conservative for most applications.

Aspect ratio, m_a (F/d):

$$m_a = \frac{m_G}{m_G + 1} \quad (\text{for spur and single helical}) \quad \dots(4)$$

$$m_a = \frac{2 m_G}{m_G + 1} \quad (\text{for double helical without including the gap}) \quad \dots(5)$$

where

$$m_G = \frac{N_G}{N_P} \geq 1.0 \quad \dots(6)$$

3.4 Input power, P

Input power, P , is the total power input. If there are multiple power paths (where $b > 1$), each path takes a portion of the input power. This document assumes that each path receives an equal share of the input power, but in practice extra steps must be taken to approach equal load sharing between power paths. See figure 1 for an example where $b = 2$.

3.5 Combined derating factors, C_d and K_d

For the purposes of this information sheet, the additional factors (application factor, load distribution factor, dynamic factor and the rim thickness factor) that affect gearset rating are combined into one derating factor for pitting resistance, C_d , and a second for bending strength, K_d . These are defined in equations 30 and 31.

3.5.1 Application factor, C_a and K_a

The application factors make allowance for any externally applied loads in excess of the nominal tangential load. This factor must be based upon the past experience of gear drive users and manufacturers. Examples may be found in appendix A of [8]. Typical application factors are shown in table 4.

Table 4 – Typical application factors*, C_a and K_a

Application factor C_a, K_a	Driven Equipment
1.25	Uniformly loaded conveyors Pure liquid mixers Centrifugal compressors Rotary or centrifugal pumps
1.50	Non-uniformly fed conveyors Variable density mixers Lobe compressors Reciprocating pumps
1.75	Multi-cylinder reciprocating compressors Rubber extruders
2.0	Reciprocating conveyors Single-cylinder reciprocating compressors Laundry washers

*Taken from appendix A of [8], for gearing driven by electric or hydraulic motors.

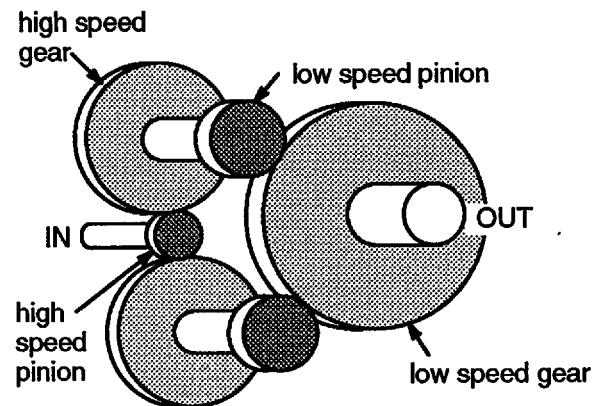


Figure 1 – Two branch double stage gearing

Most electric or hydraulic motors and steam or gas turbines are considered smoothly operating drivers, while multi-cylinder engines produce medium level shock loading and single-cylinder engines produce heavy shock loads. Add a value of 0.25 to the application factor given in table 4 if the driver is a

multi-cylinder engine, or 0.50 if the driver is a single-cylinder engine.

3.5.2 Load distribution factor, C_m and K_m

The load distribution factor modifies the rating equations to reflect the non-uniform distribution of load along the lines of contact. Reference [5] provides two methods for determining this factor, analytical and empirical. The analytical method requires knowledge of the design, manufacturing, and mounting to evaluate the load distribution factor. For this reason, the analytical method is typically used only by experienced engineers and should not be used for preliminary design. The empirical method requires a minimum amount of information. This method is recommended for relatively stiff gear designs which meet the following requirements:

- Aspect ratio, $F/d \leq 2.0$. (For double helical gears the gap is not included in the face width.);
- The gear elements are mounted between bearings. Designs having overhung pinions, overhung gears, or both, require extensive analysis to determine the load distribution factor and are not covered by this information sheet;
- Face width up to 40 inches;
- Contact across full face width of narrowest member when loaded.

When using the empirical method, the calculated value of C_m and K_m depends on many items but is basically a function of net face width and alignment. Unfortunately, at this point in the design, the net face width is not known. To get an approximation for C_m and K_m based on pinion torque, application factor, and aspect ratio, use equations 9 or 9M, or table 5.

$$T_1 = \frac{P (63\ 000)}{n_{P1}} \quad \dots(7)$$

$$T_1 = \frac{P (9550)}{\omega_1} \quad \dots(7M)$$

$$T_{P1} = \frac{T_1}{b} \quad \dots(8)$$

where

b is the number of power paths;

n_{P1} is the speed of high speed pinion, in rpm;

T_1 is the torque on high speed shaft, in in-lbs (Nm);

T_{P1} is the transmitted pinion torque, per high speed mesh, in in-lbs (Nm);

P is the input power, in hp (kW).

$$C_m = K_m = 1 + m_a \left[0.2 + 0.0054 \left(\frac{T_P C_a}{m_a} \right)^{0.33} \right] \quad \dots(9)$$

$$K_H = 1 + u_b \left[0.2 + 0.0112 \left(\frac{T_P K_A}{u_b} \right)^{0.33} \right] \quad \dots(9M)$$

Once the pinion operating pitch diameter, d , is known, a more accurate approximation of load distribution factor can be found with equations 10 or 10M.

$$C_m = K_m = 1 + m_a (0.2 + 0.03 d) \quad \dots(10)$$

$$K_H = 1 + u_b (0.2 + 0.0012 d_1) \quad \dots(10M)$$

Do not use equations 9, 9M, 10, 10M, or table 5 for a final design audit, but rather follow the procedure given in clause 6. See [5] for a complete discussion.

Table 5 – Typical load distribution factors

$T_P C_a$ (Pinion torque) x (Application factor)		C_m and K_m			
in lbs	(Nm)	$m_a = 0.25$	$m_a = 0.50$	$m_a = 0.75$	$m_a = 1.00$
5000	(550)	1.10	1.15	1.25	1.30
50 000	(5600)	1.15	1.25	1.30	1.40
500 000	(56 500)	1.20	1.35	1.50	1.65
5 000 000	(565 000)	1.40	1.70	1.90	2.10

3.5.3 Dynamic factor, C_v and K_v

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. Externally applied dynamic loading or resonance operation is not applicable to the dynamic factor and is usually accommodated in the application factor. Dynamic factors are essentially dependent on gear pitch line velocity and gear quality. For simplicity, this information sheet uses $C_v = K_v = 0.7$ as a first approximation, which is conservative for most applications. See [5] for a more detailed explanation of dynamic factor.

3.5.4 Rim thickness factor, K_B

Where the rim thickness is not sufficient to provide full support for the tooth root, the location of bending fatigue failure may be through the gear rim, rather than at the root fillet. The rim thickness factor, K_B , adjusts the calculated bending stress number for thin rimmed gears.

In general, this factor can be taken as unity if the rim section below the tooth is at least 20% greater than the tooth height, unless keyways, splines, or notches are present in the bore. Otherwise, see appendix C of [5] for a more detailed analysis.

3.6 Geometry factors, I and J

Geometry factors for pitting resistance, I , and bending strength, J , are approximated by the following equations.

Geometry factors for spur gears:

$$I \approx \frac{\sin \phi_n \cos \phi_n}{2} \frac{m_G}{m_G \pm 1} \quad \dots(11)$$

$$J \approx 0.45 \quad \dots(12)$$

Geometry factors for helical gears:

$$I \approx \frac{1 + 0.00682 \phi_n}{4.0584} \frac{m_G}{m_G \pm 1} \quad \dots(13)$$

$$J \approx 0.50 \quad \dots(14)$$

where

($\pm$) upper sign is for external gearsets, lower sign is for internal gearsets;

ϕ_n is the normal profile angle in degrees. See 3.8.

Equations 13 and 14 above assume $\psi = 15^\circ$ and will change for other values of ψ .

Values of $J = 0.45$ for spur gears and $J = 0.50$ for helical gears are starting points that can normally be achieved through good design procedures when selecting profile shift and tooling. J factors as low as 0.20 and as high as 0.75 are not uncommon, but usually are not found in optimum designs.

3.7 Gear ratio

The gear ratio, m_G , is an input that must be known. In many situations the ratio is so large that there should be more than one stage. When the gear ratio approaches or exceeds 6:1, it may be more economical to add a second stage. To optimize the design, one must first optimize the overall ratio split between the various meshes. This subclause provides two methods for optimizing the ratio split; one based on minimum volume and a second based on preexisting housing designs.

In either case, internal clearances between rotating and stationary parts must be maintained. Also, adequate room must be available for bearings with enough capacity for the application. Both methods balance the rating of each mesh by equalizing the pitting rating of each pinion.

3.7.1 Minimum volume gearsets

One method is to split the overall ratio to minimize the sum of the solid rotor volumes. This method is discussed in [9] and annex B. The "minimum volume" gear ratio of the high speed gearset of a two stage gearbox with b power branches is given by:

$$\left(\frac{M_o^2}{b m_{G1}^2} \right) - 1 = \left(\frac{C_{m1}}{C_{m2}} \right) \left(\frac{I_2}{I_1} \right) \left(\frac{s_{ac2}}{s_{ac1}} \right)^2 \times \left[\left(\frac{0.112}{b^{0.888} m_{G1}^{0.888}} \right) + 2.112 b^{0.112} m_{G1}^{1.112} \right] \quad \dots(15)$$

where

C_{m1} is the load distribution factor for pitting resistance – high speed mesh;

C_{m2} is the load distribution factor for pitting resistance – low speed mesh;

I_1 is the geometry factor for pitting resistance – high speed mesh;