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Calculation of load capacity of bevel gears —

Part 2:
Calculation of surface durability (macropitting)

Calcul de la capacité de charge des engrenages coniques —

Partie 2: Calcul de la résistance à la pression superficielle (macro-écaillage)

Reference number
ISO 10300-2:2023(E)

Contents

Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO document should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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This document was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

This third edition cancels and replaces the second edition (ISO 10300-2:2014), which has been technically revised.

The main changes are as follows:

— Table 1 has been inserted;

— Table 2 has been inserted;

— the term “pitting” has been replaced by “macropitting”;

— bevel gear factor, Z_K, for the calculation of the nominal value of the contact stress has been removed; instead, a new bevel gear factor, Z_{KP}, has been introduced for the calculation of the permissible contact stress;

— Formula (37) for the calculation of the length of action considering adjacent teeth has been modified;

— subclause 8.3 — work hardening factor, Z_W, has been updated and method A added;

— Figure 2 — load distribution in the contact area has been updated as the symbol for exponent e has been changed to e_{LS};

— Figure 6 — facewidth factor, Z_{FW} has been removed;

— Figure 7 — lubricant factor, Z_L, for mineral oils has been removed;

— Figure 8 — speed factor, Z_V has been removed;

— Figure 9 — roughness factor, Z_R has been removed;

— Figure 10 — work hardening factor, Z_W has been removed;

— former Annex A has been replaced by new Annex A describing a local calculation method for surface durability (macropitting) – Method B1-localised.

A list of all parts in the ISO 10300 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

When ISO 10300:2001 (all parts) became due for its first revision, the opportunity was taken to include hypoid gears, since previously the series only allowed for calculating the load capacity of bevel gears without offset axes. The former structure is retained, i.e. three parts of the ISO 10300 series, together with ISO 6336-5, and it is intended to establish general principles and procedures for rating of bevel gears. Moreover, ISO 10300 (all parts) is designed to facilitate the application of future knowledge and developments, as well as the exchange of information gained from experience.

In view of the decision for ISO 10300 (all parts) to cover hypoid gears also, a separate clause: "Gear flank rating formulae — Method B2" has been included in this document, while the former method B was renamed method B1. It became necessary to present a new, clearer structure of the three parts, which is illustrated in ISO 10300-1:2023, Figure 1.

NOTE ISO 10300 (all parts) gives no preferences in terms of when to use method B1 and when to use method B2.

This document deals with the failure of gear teeth by macropitting, a fatigue phenomenon. Two varieties of macropitting are recognized, initial and destructive macropitting.

In applications employing low hardness steel or through hardened steel, initial macropitting frequently occurs during early use and is not deemed serious. Initial macropitting is characterized by small pits which do not extend over the entire facewidth or profile depth of the affected tooth. The degree of acceptability of initial macropitting varies widely, depending on the gear application. Initial macropitting occurs in localized overstressed areas and tends to redistribute the load by progressively removing high contact spots. Generally, when the load has been redistributed, the macropitting stops.

In applications employing high hardness steel and case carburized steel, the variety of macropitting that occurs is usually destructive. The formulae for macropitting resistance given in this document are intended to assist in the design of bevel gears which stay free from destructive macropitting during their design lives (for additional information, see ISO/TR 22849[5]).

The basic formulae, first developed by Hertz for the contact pressure between two curved surfaces, have been modified to consider the following four items: the load sharing between adjacent teeth, the position of the centre of pressure on the tooth, the shape of the instantaneous area of contact and the load concentration resulting from manufacturing uncertainties. The Hertzian contact pressure serves as the theory for the assessment of surface durability with respect to macropitting. Although all premises for a gear mesh are not satisfied by Hertzian relations, their use can be justified by the fact that, for a gear material, the limits of the Hertzian pressure are determined on the basis of running tests with gears, which include the additional influences in the analysis of the limit values. Therefore, if the reference is within the application range, Hertzian pressure can be used to convert test gear data to gears of various types and sizes.

NOTE Contrary to cylindrical gears, where the contact is usually linear, bevel gears are generally manufactured with profile and lengthwise crowning, i.e. the tooth flanks are curved on all sides and the contact develops an elliptical pressure surface. This is taken into consideration when determining the load factors by the fact that the rectangular zone of action (in the case of spur and helical gears) is replaced by an inscribed parallelogram for method B1 and an inscribed ellipse for method B2 (see ISO 10300-1:2023, Annex A for method B1 and Annex B for method B2). The conditions for bevel gears, different from cylindrical gears in their contact, are thus taken into consideration by the face and transverse load distribution factors.

Calculation of load capacity of bevel gears —

Part 2:
Calculation of surface durability (macropitting)

1 Scope

This document specifies the basic formulae for use in the determination of the surface load capacity of straight and helical (skew), Zerol and spiral bevel gears including hypoid gears, and comprises all the influences on surface durability for which quantitative assessments can be made. This document is applicable to oil lubricated bevel gears, as long as sufficient lubricant is present in the mesh at all times.

The formulae in this document are based on virtual cylindrical gears and restricted to bevel gears whose virtual cylindrical gears have transverse contact ratios of $\varepsilon_{v\alpha} < 2$. The results are valid within the range of the applied factors as specified in ISO 10300-1.

The formulae in this document are not directly applicable to the assessment of other types of gear tooth surface damage, such as plastic yielding, scratching, scuffing or any other type not specified.

NOTE This document is not applicable to bevel gears which have an inadequate contact pattern under load.

The user is cautioned that when the formulae are used for large average mean spiral angles $(\beta_{m1} + \beta_{m2})/2 > 45°$, for effective pressure angles $\alpha_e > 30°$ and/or for large facewidths $b > 13\ m_{mn}$, the calculated results of this document should be confirmed by experience.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

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

ISO 1122-1, *Vocabulary of gear terms — Part 1: Definitions related to geometry*

ISO 6336-5, *Calculation of load capacity of spur and helical gears — Part 5: Strength and quality of materials*

ISO 10300-1, *Calculation of load capacity of bevel gears — Part 1: Introduction and general influence factors*

ISO 17485, *Bevel gears — ISO system of accuracy*

ISO 23509, *Bevel and hypoid gear geometry*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 1122-1 and ISO 23509 and the following apply.

ISO and IEC maintain terminology databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at https://www.electropedia.org/

3.1
macropitting
material fatigue phenomenon of two meshing surfaces under load

3.2
nominal contact stress
σ_{H0}
contact stress calculated on the basis of the Hertzian theory at the critical point of load application for error-free gears loaded by a constant nominal torque

3.3
contact stress
σ_{H}
determinant contact stress at the critical point of load application including the load factors which consider static and dynamic loads and load distribution

3.4
allowable stress number (contact)
$\sigma_{H,lim}$
maximum contact stress of standardized test gears and determined at standardized operating conditions, as specified in ISO 6336-5

3.5
permissible contact stress
σ_{HP}
maximum contact stress of the evaluated gear set including all influence factors

4 Symbols, general subscripts and abbreviated terms

For the purposes of this document, the symbols given in ISO 701, ISO 17485, ISO 23509 and the following shall apply.

Table 1 — Symbols

Symbol	Description or term	Unit
A	Auxiliary factor for calculating the dynamic factor $K_{v\text{-}C}$	—
A^*	Related area for calculating the load sharing factor Z_{LS}	mm
A_m^*	Area above the middle contact line	mm
A_r^*	Area above the root contact line	mm
A_t^*	Area above the tip contact line	mm
b	Facewidth	mm
b_b	Relative base facewidth	—
C_{ZL}, C_{ZR}, C_{ZV}	Constants for determining lubricant film factors	—
d_e	Outer pitch diameter	mm
d_m	Mean pitch diameter	mm
d_v	Reference diameter of virtual cylindrical gear	mm
d_{va}	Tip diameter of virtual cylindrical gear	mm
d_{vb}	Base diameter of virtual cylindrical gear	mm
E	Modulus of elasticity, Young's modulus	N/mm^2
e_{LS}	Exponent for the load distribution along the lines of contact	—
F	Auxiliary variable for mid-zone factor	—
F_{mt}	Nominal tangential force at mid-facewidth of the reference cone	N

Table 1 *(continued)*

Symbol	Description or term	Unit
F_n	Nominal normal force	N
f	Distance from the centre of the zone of action to a contact line	mm
f_{max}	Maximum distance to middle contact line	mm
g_c	Length of contact line (method B2)	mm
g_{va}	Length of path of contact of virtual cylindrical gear in transverse section	mm
g_{van}	Relative length of action in normal section	—
g_η	Relative length of action within the contact ellipse	—
$g_{\eta I}$	Relative length of action at critical point in contact ellipse	—
$g_{\eta I \Sigma}$	Relative length of action considering adjacent teeth	—
HBW	Brinell hardness	—
K_v	Dynamic factor	—
K_A	Application factor	—
$K_{H\alpha}$	Transverse load factor for contact stress	—
$K_{H\beta}$	Face load factor for contact stress	—
k	Positive integer	—
k'	Contact shift factor	—
l_b	Length of contact line (method B1)	mm
l_{bm}	Theoretical length of middle contact line	mm
m_{et}	Outer transverse module	mm
m_{mn}	Mean normal module	mm
N_L	Number of load cycles	—
p	Peak load	N/mm
p_{max}	Maximum peak load	N/mm
p^*	Relative peak load for calculating the load sharing factor (method B1)	—
p_{nb}	Relative mean normal base pitch	—
Ra	Centre line average (CLA) = AA arithmetic average roughness	µm
R_{mpt}	Relative mean back cone distance	—
Rz	Mean peak-to-valley roughness	µm
Rz_H	Equivalent roughness	µm
Rz_{10}	Mean roughness for gear pairs with relative curvature radius ρ_{rel} = 10 mm	µm
r_{va}	Relative mean virtual tip radius	—
r_{vn}	Relative mean virtual pitch radius	—
S_H	Safety factor for contact stress (against macropitting)	—
$S_{H,min}$	Minimum safety factor for contact stress	—
u	Gear ratio of bevel gear	—
V	Ratio of maximum load over the middle contact line and total load	—
v_g	Sliding velocity in the mean point P	m/s
$v_{g,par}$	Sliding velocity parallel to the contact line	m/s
$v_{g,vert}$	Sliding velocity vertical to the contact line	m/s
v_{mt}	Tangential speed at mid-facewidth of the reference cone	m/s
v_w	Circumferential velocity at the pitch line	m/s
v_Σ	Sum of velocities in the mean point P	m/s
$v_{\Sigma h}$	Sum of velocities in profile direction	m/s
$v_{\Sigma l}$	Sum of velocities in lengthwise direction	m/s

Table 1 *(continued)*

Symbol	Description or term	Unit
$v_{\Sigma s}$	Sum of velocities in lengthwise direction	m/s
$v_{\Sigma,vert}$	Sum of velocities vertical to the contact line	m/s
w	Angle of contact line relative to the root cone	°
w_t	Surface velocity	m/s
$w_{t,h}$	Surface velocity in profile direction	m/s
$w_{t,s}$	Surface velocity in lengthwise direction	m/s
$w_{t,vert}$	Surface velocity vertical to the contact line	m/s
X	Intermediate value	—
Z_i	Inertia factor (macropitting)	—
Z_v	Speed factor	—
Z_A	Contact stress adjustment factor (method B2)	—
Z_E	Elasticity factor	$(N/mm^2)^{1/2}$
Z_{FW}	Facewidth factor	—
Z_{Hyp}	Hypoid factor	—
Z_I	Macropitting resistance geometry factor (method B2)	—
Z_{KP}	Bevel gear factor (method B1)	—
Z_L	Lubricant factor	—
Z_{LS}	Load sharing factor (method B1)	—
$Z_{M\text{-}B}$	Mid-zone factor	—
Z_{NT}	Life factor (macropitting)	—
Z_R	Roughness factor for contact stress	—
Z_S	Bevel slip factor	—
Z_W	Work hardening factor	—
Z_X	Size factor	—
z	Number of teeth	—
z_v	Number of teeth of virtual cylindrical gear	—
α_L	Normal pressure angle at point of load application (method B2)	°
α_a	Adjusted pressure angle (method B2)	°
α_{an}	Normal pressure angle at tooth tip	°
$\alpha_{eD,C}$	Effective pressure angle for drive side/coast side	°
α_{lim}	Limit pressure angle	°
$\alpha_{nD,C}$	Generated pressure angle for drive side/coast side	°
α_{vet}	Transverse pressure angle of virtual cylindrical gears	°
β_B	Inclination angle of contact line	°
β_{bm}	Mean base spiral angle	°
β_m	Mean spiral angle	°
ε_{NI}	Load sharing ratio for macropitting (method B2)	—
ζ_R	Pinion offset angle in root plane	°
ζ_{vert}	Slip vertical to the contact line	°
λ	Adjustment angle for contact angle of hypoid gears (method B2)	°
λ_r	Adjustment angle for virtual spiral angle of hypoid gears (method B2)	°
ρ	Density of gear material	kg/mm^3
ρ_{rel}	Local equivalent radius of curvature vertical to contact line	mm
ρ_t	Relative radius of profile curvature between pinion and wheel (method B2)	—

Table 1 *(continued)*

Symbol	Description or term	Unit
$\rho_{\Delta red}$	Relative radius of curvature change	—
$\rho_{\Delta 1,2}$	Relative radius of curvature difference between point of load application and mean point	—
Σ	Shaft angle	°
σ_H	Contact stress	N/mm^2
$\sigma_{H,lim}$	Allowable stress number for contact stress	N/mm^2
σ_{HP}	Permissible contact stress	N/mm^2
ν	Poisson's ratio	—
ν_{40}, ν_{50}	Nominal kinematic viscosity of the oil at 40 °C and 50 °C respectively	mm^2/s
ω_{wt}	Angle between surface velocity in lengthwise and profile direction	°
ω_{Σ}	Inclination angle of the sum of velocities vector results	°

Table 2 — General subscripts

Subscripts	Description
0	Tool
1	Pinion
2	Wheel
A, B, B1, B2, C	Value according to method A, B, B1, B2 or C
D	Drive flank
C	Coast flank
T	Relative to standardized test gear dimensions
(1), (2)	Trials of interpolation

Table 3 — Abbreviated terms

Abbreviated term	Material	Type
St	Normalized low carbon steels/cast steels	Wrought normalized low carbon steels
St (cast.)		Cast steels
GTS (perl.)	Cast iron materials	Black malleable cast iron (perlitic structure)
GGG (perl., bai., ferr.)		Nodular cast iron (perlitic, bainitic, ferritic structure)
GG		Grey cast iron
V	Through hardened wrought steels	Carbon steels, alloy steels
V (cast)	Through hardened cast steels	Carbon steels, alloy steels
Eh	Case-hardened wrought steels	
IF	Flame or induction hardened wrought or cast steels	
NT (nitr.)	Nitrided wrought steels/nitriding steels/ through hardening steels, nitrided	Nitriding steels
NV (nitr.)		Through hardening steels
NV (nitrocar.)	Wrought steels, nitrocarburized	Through hardening steels

5 Macropitting damage — General aspects

5.1 Acceptable versus unacceptable macropitting

When limits of the surface durability of the meshing flanks are exceeded, particles break out of the flank, thus leaving pits. This damage is called pitting, also known as macropitting. The extent, to which such pits may be tolerated, in terms of their size and number, varies within wide limits which depend largely on the field of application. In some fields, extensive macropitting is acceptable; in others, no macropitting is acceptable. The descriptions in 5.2 and 5.3 are relevant to average working conditions and give guidelines to distinguish between initial and destructive, and acceptable and unacceptable macropitting varieties.

A linear or progressive increase in the total area of pits (linear or progressive macropitting) is generally considered to be unacceptable. However, it is possible that the effective tooth bearing area is enlarged by initial macropitting, and the rate of pit generation subsequently decreases (degressive macropitting), or even ceases (arrested macropitting), and then may be considered tolerable. Nevertheless, where there is dispute over the acceptability of macropitting the next subclause shall be determinant.

5.2 Assessment requirements

Macropitting involving the formation of pits which increase linearly or progressively with time under unchanged service conditions shall be unacceptable. Damage assessment shall include the entire active area of all the tooth flanks. The number and size of newly developed pits in unhardened tooth flanks shall be taken into consideration. Pits are frequently formed on just one, or only a few, of the surface hardened gear tooth flanks. In such circumstances, assessment shall be centred on the flanks actually pitted.

Teeth suspected of being especially at risk should be marked for critical examination if a quantitative evaluation is required.

In special cases, it is possible that a first, rough assessment can be based on considerations of the entire quantity of wear debris. But in critical cases, the condition of the flanks should be examined at least three times. The first time, however, the examination should take place only after at least 10^6 cycles of load. Depending on the results of previous examinations, further ones should be carried out after a period of service.

When deterioration caused by macropitting is such that it puts human life in danger, or poses a risk of other grave consequences, the macropitting shall not be tolerated. Due to stress concentration effects, a pit of 1 mm in diameter near the fillet of a through hardened or case-hardened gear tooth can become the origin of a crack which can lead to tooth breakage; for this reason, such a pit shall be considered unacceptable (for example, in aerospace transmissions).

Similar considerations should be taken into account in respect of turbine gears. In general, during the long life (10^{10} to 10^{11} cycles) demanded of these gears, neither macropitting nor unduly severe wear should be considered acceptable as such damage can lead to unacceptable vibrations and excessive dynamic loads. Appropriately generous safety factors should be included in the calculation: only a low probability of failure shall be tolerated.

In contrast, macropitting on the operating flanks may be tolerated for some slow speed industrial gears with large teeth (e.g. module 25) made from low hardness steel, which can safely transmit the rated power for 10 years to 20 years. Individual pits can be up to 20 mm in diameter and 0,8 mm deep. The apparently “destructive macropitting”, which occurs during the first two or three years of service, normally slows down. In such cases, the tooth flanks become smoothed and work hardened to the extent of increasing the surface Brinell hardness number by 50 % or more. For such conditions, relatively low safety factors (in some, less than 1) may be chosen, with a correspondingly higher probability of tooth surface damage. However, a high safety factor against tooth breakage shall be chosen.

5.3 General rating procedure

There are two main methods for rating the surface durability of bevel and hypoid gears: method B1 and method B2. They are provided in Clause 6 and Clause 7, while Clause 8 contains those influence factors which are equal for both. Although methods B1 and B2 use the same basis of calculation, the calculation procedure is unique to each method.

With both methods, the capability of a gear tooth to resist macropitting shall be determined by the comparison of the following stress values:

— contact stress, σ_H, based on the geometry of the tooth, the accuracy of its manufacture, the rigidity of the gear blanks, bearings and housing, and the operating torque, expressed by the contact stress Formulae (1) and (20) (see 6.1 and 7.1);

— permissible contact stress, σ_{HP}, based on the endurance limit for contact stress, $\sigma_{H,lim}$, and the effect of the operating conditions under which the gears operate, expressed by the permissible contact stress Formulae (4) and (22) (see 6.2 and 7.2).

The ratio of the permissible contact stress and the calculated contact stress is the safety factor S_H. The value of the minimum safety factor for contact stress, $S_{H,min}$, should be 1,0. For further recommendations on the choice of this safety factor and other minimum values, see ISO 10300-1.

The gear designer and customer should agree on the value of the minimum safety factor.

Information on a local calculation method based on method B1 can be found in Annex A.

6 Gear flank rating formulae — Method B1

6.1 Contact stress formula

The calculation of macropitting resistance is based on the contact (Hertzian) stress, in which the load is distributed along the lines of contact (see ISO 10300-1:2023, Annex A). Calculations shall be carried out for pinion and wheel together; however, in case of different pressure angles of drive and coast side (hypoid gears, asymmetric bevel gears) separately for drive and coast side flank.

$$\sigma_{H-B1} = \sigma_{H0-B1} \cdot \sqrt{K_A \cdot K_v \cdot K_{H\beta} \cdot K_{H\alpha}} \leq \sigma_{HP-B1} \tag{1}$$

with load factors K_A, K_v, $K_{H\beta}$, $K_{H\alpha}$ as specified in ISO 10300-1.

The nominal value of the contact stress is given by Formula (2):

$$\sigma_{H0-B1} = \sqrt{\frac{F_n}{l_{bm} \cdot \rho_{rel}}} Z_{M-B} \cdot Z_{LS} \cdot Z_E \tag{2}$$

where F_n is the nominal normal force of the virtual cylindrical gear at mean point P according to Formula (3):

$$F_n = \frac{F_{mt1}}{\cos\alpha_n \cdot \cos\alpha_{m1}} \tag{3}$$

where

$\alpha_n = \alpha_{nD}$ is the generated pressure angle for drive side in accordance with ISO 23509;

$\alpha_n = \alpha_{nC}$ is the generated pressure angle for coast side in accordance with ISO 23509;

l_{bm} is the length of contact line in the middle of the zone of action as specified in ISO 10300-1:2023, A.2.7;

ρ_{rel} is the local equivalent radius of curvature vertical to the contact line as specified in ISO 10300-1:2023, A.2.8;

Z_{M-B} is the mid-zone factor which accounts for the conversion of the contact stress determined at the mean point to the determinant position (see 6.4.1);

Z_{LS} is the load sharing factor that considers the load sharing between two or more pairs of teeth (see 6.4.2);

Z_E is the elasticity factor which accounts for the influence of the material's E-Module and Poisson's ratio (see 8.1).

The determinant position of load application is:

a) the inner point of single tooth contact, if $\varepsilon_{v\beta} = 0$;

b) the midpoint of the zone of action, if $\varepsilon_{v\beta} \geq 1$;

c) interpolation between a) and b), if $0 < \varepsilon_{v\beta} < 1$.

6.2 Permissible contact stress

The permissible contact stress shall be calculated separately for pinion (suffix 1) and wheel (suffix 2):

$$\sigma_{HP-B1} = \sigma_{H,lim} \cdot Z_{NT} \cdot Z_X \cdot Z_L \cdot Z_v \cdot Z_R \cdot Z_W \cdot Z_{KP} \cdot Z_{Hyp} \tag{4}$$

where

$\sigma_{H,lim}$ is the allowable stress number (contact), which accounts for material, heat treatment and surface influence at test gear dimensions as specified in ISO 6336-5;

Z_{NT} is the life factor (see 8.4), which accounts for the influence of required numbers of cycles of operation;

Z_X is the size factor (see 6.5.2), which accounts for the influence of the tooth size, given by the module, on the permissible contact stress;

Z_L, Z_v, Z_R are the lubricant film factors (see 8.2) for the influence of the lubrication conditions;

Z_W is the work hardening factor (see 8.3), which considers the hardening of a softer wheel running with a surface-hardened pinion;

Z_{KP} is the bevel gear factor which accounts for stress adjustment (see 6.5.1);

Z_{Hyp} is the hypoid factor (see 6.5.3), which accounts for the influence of lengthwise sliding onto the surface durability.

6.3 Calculated safety factor for contact stress

The calculated safety factor for contact stress according to Formula (5) shall be checked separately for pinion and wheel, if the values of permissible contact stress are different:

$$S_{H-B1} = \frac{\sigma_{HP-B1}}{\sigma_{H-B1}} > S_{H,min} \tag{5}$$

where $S_{H,min}$ is the minimum safety factor; see ISO 10300-1:2023, 5.2 for recommended numerical values for the minimum safety factor or the risk of failure (damage probability).

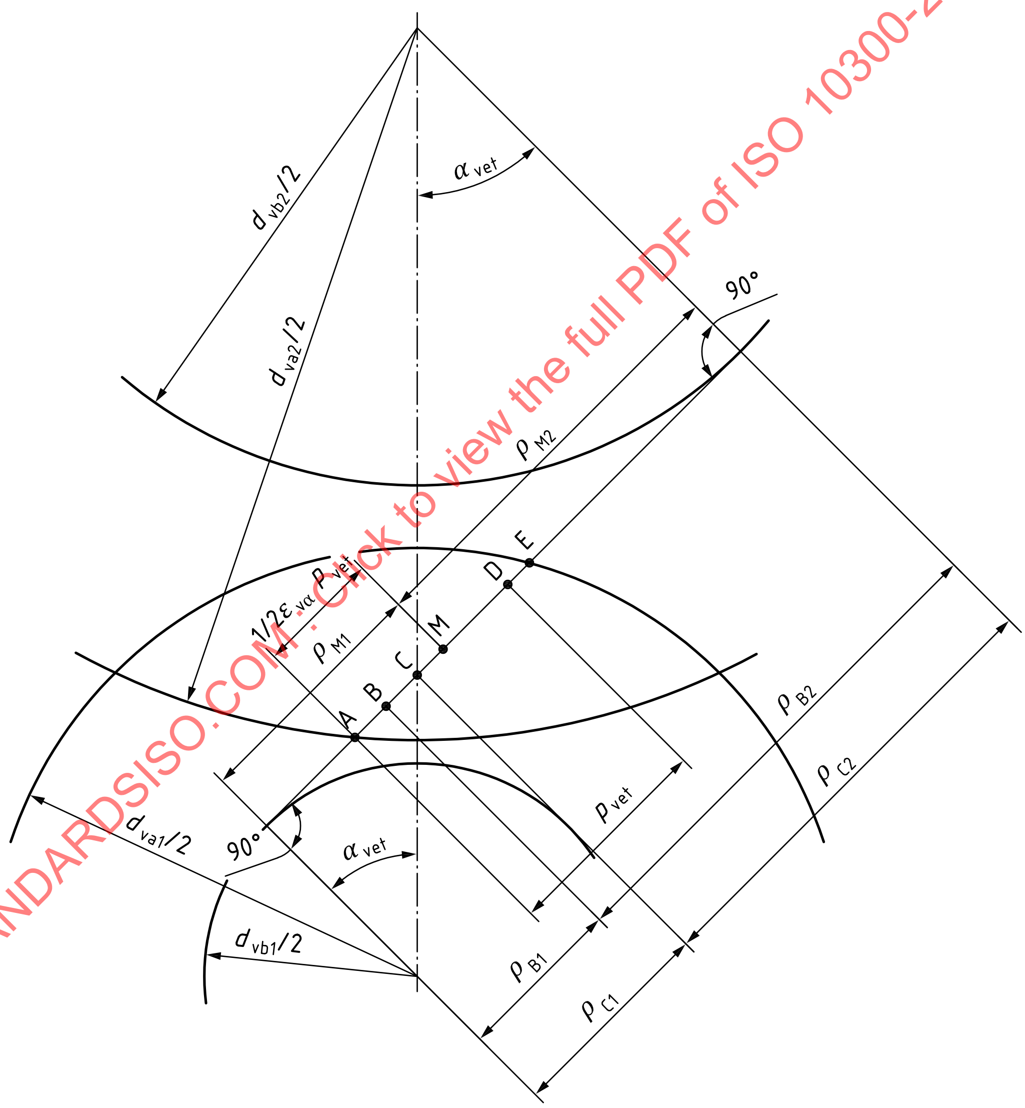
NOTE Formula (5) defines the relationship of the calculated safety factor, S_H, with respect to contact stress. A safety factor related to the transferable torque is equal to the square of S_H.

6.4 Contact stress factors

6.4.1 Mid-zone factor, $Z_{M\text{-}B}$

The mid-zone factor, $Z_{M\text{-}B}$, considers the difference between the local equivalent radius of curvature ρ_{rel} at the mean point and at the critical point of load application of the pinion. The radius ρ_{rel} at the mean point P can directly be calculated from the data of the bevel gears in mesh (see ISO 10300-1:2023, Annex A). For the conversion to the critical point of mesh, the corresponding virtual cylindrical gears are used. Depending on the face contact ratio it can be the inner point of single contact B of the pinion ($\varepsilon_{v\beta} = 0$) or point M in the middle of the path of contact ($\varepsilon_{v\beta} \geq 1$) or a point interpolated between B and M for $0 < \varepsilon_{v\beta} < 1$ (see Figure 1). The comparison with the results of tooth contact analyses shows a good approximation for bevel gear as well as for hypoid gear sets.

For hypoid gears, the mid-zone factor shall be determined for both, drive and coast flank, separately.



NOTE The schematic view of a cylindrical gear set in transverse section shows the line of action being tangent to both base circles d_{vb1} and d_{vb2} of pinion and wheel. The tip circles d_{va2} and d_{va1} intersect the line of action in points A and E, which define the path of contact. In between there are pitch point C, midpoint M and inner point of single contact B, for which different radii of profile curvature are specified: $\rho_{C1,2}$, $\rho_{M1,2}$, $\rho_{B1,2}$, the basis for Formula (6).

Figure 1 — Radii of curvature at midpoint M and inner point of single contact B of the pinion for determination of the mid-zone factor, $Z_{M\text{-}B}$

The mid-zone factor, $Z_{M\text{-}B}$, is calculated by Formula (6):

$$Z_{M-B} = \frac{\tan \alpha_{vet}}{\sqrt{\left[\sqrt{\left(\frac{d_{va1}}{d_{vb1}}\right)^2 - 1} - F_1 \cdot \frac{\pi}{z_{v1}}\right] \cdot \left[\sqrt{\left(\frac{d_{va2}}{d_{vb2}}\right)^2 - 1} - F_2 \cdot \frac{\pi}{z_{v2}}\right]}} \quad (6)$$

The auxiliary factors F_1 and F_2 for the mid-zone factor are given in Table 4.

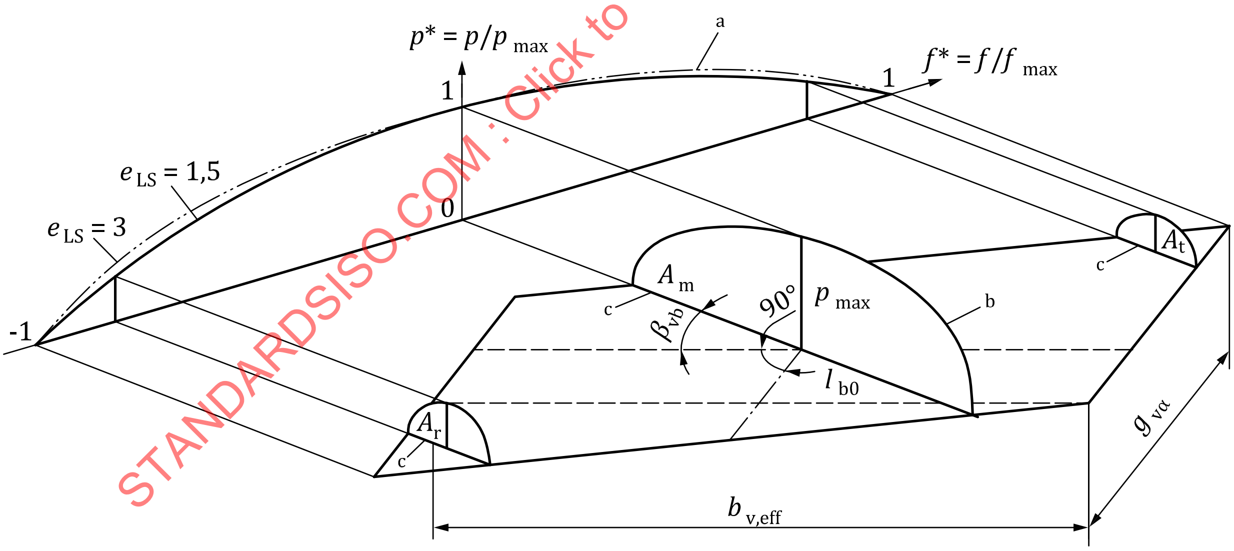
Table 4 — Factors for calculation of mid-zone factor, $Z_{M\text{-}B}$

Parameters	F_1	F_2
$\varepsilon_{v\beta} = 0$	2	$2 \cdot (\varepsilon_{v\alpha} - 1)$
$0 < \varepsilon_{v\beta} < 1$	$2 + (\varepsilon_{v\alpha} - 2) \cdot \varepsilon_{v\beta}$	$2 \cdot \varepsilon_{v\alpha} - 2 + (2 - \varepsilon_{v\alpha}) \cdot \varepsilon_{v\beta}$
$\varepsilon_{v\beta} \geq 1$	$\varepsilon_{v\alpha}$	$\varepsilon_{v\alpha}$

6.4.2 Load sharing factor, Z_{LS}

The load sharing factor, Z_{LS}, accounts for load sharing between two or more pairs of teeth. That means this factor determines the maximum portion of the total load which affects one tooth. The load distribution along each contact line in the zone of action is assumed to be elliptical. The area, A, of each semi-ellipse (see Figure 2) represents the load on the respective contact line, and the sum of all areas over all contact lines being simultaneously in mesh, represents the total load on the gear set. Additionally, the distribution of the peak loads, p, over the line of action is assumed to follow a parabola (exponent e_{LS}). On this basis, the maximum load over the middle contact line divided by the total load is a measure for load sharing.

NOTE In this context, contact line means the major axis of the Hertzian contact ellipse under load.



a Parabolic distribution of peak loads.

b Elliptical load distribution.

c Contact lines simultaneously in mesh.

Figure 2 — Load distribution in the contact area

For easier calculation, dimensionless parameters related to their maximum values are used (marked by *) for the peak load, p, and the distance, f, of the relevant contact line from the centre of the zone of action.

Related peak load p^* is given by Formula (7):

$$p^* = \frac{p}{p_{\max}} = 1 - \left(\frac{|f|}{|f_{\max}|} \right)^{e_{LS}} = 1 - \left| f^* \right|^{e_{LS}} \quad (7)$$

with f given in Table A.2 of ISO 10300-1:2023, and exponent, e_{LS}, given in Table 5.

The related area, A^*, is calculated by the formula of an ellipse whose major axis is half the length of the contact line l_b and whose minor axis is given by the related peak load p^*.

For the related area A^*, Formula (8) applies:

$$A^* = \frac{1}{4} \cdot p^* \cdot l_b \cdot \pi \quad (8)$$

with l_b in accordance with ISO 10300-1:2023, A.2.7.

Table 5 — Exponent, e_{LS}, for calculation of parabolic distribution of peak loads, p^*

Profile crowning	Exponent e_{LS}
Low (e.g. automotive gears)	3
High (e.g. industrial gears)	1,5

The ratio, V, of maximum load over the middle contact line and total load can be expressed by Formula (9):

$$V = \frac{A_m^*}{A_t^* + A_m^* + A_r^*} \quad (9)$$

As the contact stress is a function of the square root of load, this is necessarily also applied to the ratio of the maximum load and the total load, when determining the load sharing factor Z_{LS} according to Formula (10):

$$Z_{LS} = \sqrt{\frac{A_m^*}{A_t^* + A_m^* + A_r^*}} \quad (10)$$

where

A_t^* is the area above the tip contact line, where p^*, l_b shall be calculated with f_t according to ISO 10300-1:2023, Table A.2;

A_m^* is the area above the middle contact line where p^*, l_b shall be calculated with f_m in accordance with ISO 10300-1:2023, Table A.2;

A_r^* is the area above the root contact line, where p^*, l_b shall be calculated with f_r in accordance with ISO 10300-1:2023, Table A.2.

6.5 Permissible contact stress factors

6.5.1 Bevel gear factor, Z_{KP}

The factor Z_{KP} is an empirical factor which accounts for the differences between cylindrical and bevel gears in such a way as to agree with practical experience. It is a stress adjustment constant which permits the rating of bevel gears, using the same allowable contact stress numbers as for cylindrical gears.

The following value for Z_{KP} should be used in the absence of more specific knowledge:

$$Z_{KP} = 1,2 \tag{11}$$

6.5.2 Size factor, Z_X

Factor Z_X accounts for statistical evidence indicating that the stress levels, at which fatigue damage occurs, decreases with an increase in component size. This results from the influence of lower stress gradients on subsurface defects (theoretical stress analysis) and of gear size on material quality (e.g. effect on forging process, variations in structure). The main influence parameters related to the size factor are:

a) material quality (furnace charge, cleanliness, forging);

b) heat treatment, distribution of hardening;

c) module in the case of surface hardening; depth of hardened layer relative to the size of teeth (core-supporting effect).

The size factor, Z_X, shall be determined separately for pinion and wheel. However, reasonable size factors for gear teeth have not yet been established. So, in Clause 6, the size factor is set equal to unity ($Z_{X1,2}$ = 1) for most gears, provided a proper choice of material is made.

6.5.3 Hypoid factor, Z_{Hyp}

6.5.3.1 General

Tests were carried out on series of bevel gears with increasing relative hypoid offset values a_{rel}. They show that the permanent transmissible torque increases from zero offset to typical offset values but decreases again at very high offset values. The Hertzian pressure caused by the respective permanent transmissible torque has its maximum on bevel gears without offset; however, this pressure decreases immediately with increasing offset. The only interpretation seems to be that higher sliding velocities lower the allowable contact stresses on the flank. Rising contact temperatures and debasing lubricant film developments are regarded as main reasons for this lowering effect.

In order to realize this effect, different components of velocities in the mean point shall be considered: the sliding velocity parallel to the contact line, $v_{g,par}$, which is unfavourable for the temperature and the oil film thickness, and the sum of velocities vertical to the contact line, $v_{\Sigma,vert}$, which is advantageous for the oil film. For bevel gears without offset, $v_{g,par}$ is negligibly small compared to $v_{\Sigma,vert}$, while for hypoid gears, both components increase with the offset value, but $v_{g,par}$ more than $v_{\Sigma,vert}$. So, the ratio of both components is an indicator for the different temperature and oil film behaviour of hypoid gears in comparison to cylindrical gears and bevel gears without offset. To consider this effect in the rating formula for surface durability the hypoid factor, Z_{Hyp}, is included in method B1.

NOTE In this context, contact line means the major axis of the Hertzian contact ellipse under load.

Empirical Formula (12) for the hypoid factor, Z_{Hyp}, was derived from test results:

$$Z_{Hyp} = 1 - 0,3 \cdot \left(\frac{v_{g,par}}{v_{\Sigma,vert}} - 0,15 \right) \tag{12}$$

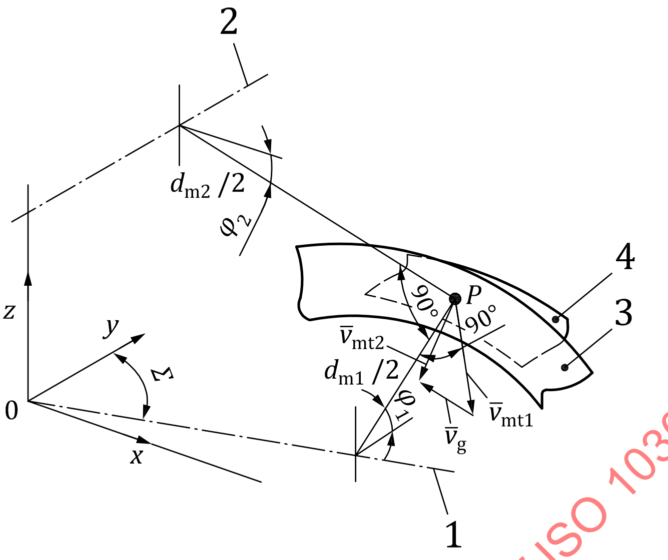
where

the range of validity is given by: $0,6 \leq Z_{Hyp} \leq 1,0$; for bevel gears without offset, Z_{Hyp} = 1,0;

$v_{g,par}$ is the sliding velocity parallel to the contact line;

$v_{\Sigma\ vert}$ is the sum of velocities vertical to the contact line.

In order to evaluate the different velocities, Figure 3 shows both tangential velocities v_{mt1} and v_{mt2} as well as the sliding velocity, v_g, for a hypoid gear pair with $\Sigma < 90°$.



Key

1 pinion axis
2 wheel axis
3 pinion flank
4 wheel flank

Figure 3 — Tangential velocities in the mean point P

6.5.3.2 Sliding velocity

The sliding velocity parallel to the contact line, as shown in Figure 4, is derived from Formulae (13) and (14):

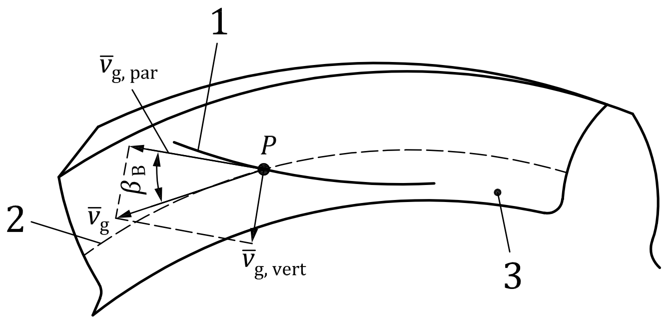
$$v_g = v_{mt1} \cdot \cos\beta_{m1} \cdot (\tan\beta_{m1} - \tan\beta_{m2}) \tag{13}$$

$$v_{g,par} = v_g \cdot \cos|\beta_B| \tag{14}$$

where

v_g is the sliding velocity in the mean point P;

β_B is the inclination angle of contact line as specified in ISO 10300-1:2023, Formula (A.38).



Key

1 contact line
2 trace of pitch cone
3 pinion flank

Figure 4 — Sliding velocity vertical and parallel to the contact line in the mean point P

6.5.3.3 Sum of velocities

The sum of velocities in profile direction is calculated with Formula (15):

$$v_{\Sigma h} = \left|2 \cdot v_{mt1} \cdot \cos\beta_{m1} \cdot \sin\alpha_n\right| \tag{15}$$

where

$\alpha_n = \alpha_{nD}$ is the generated pressure angle for drive side in accordance with ISO 23509;

$\alpha_n = \alpha_{nC}$ is the generated pressure angle for coast side in accordance with ISO 23509.

The sum of velocities in the lengthwise direction is calculated with Formula (16):

$$v_{\Sigma l} = \left|v_{mt1} \cdot \left(\sin\beta_{m1} + \frac{\sin\beta_{m2} \cdot \cos\beta_{m1}}{\cos\beta_{m2}}\right)\right| \tag{16}$$

According to Figure 5, the sum of velocities results from Formula (17):

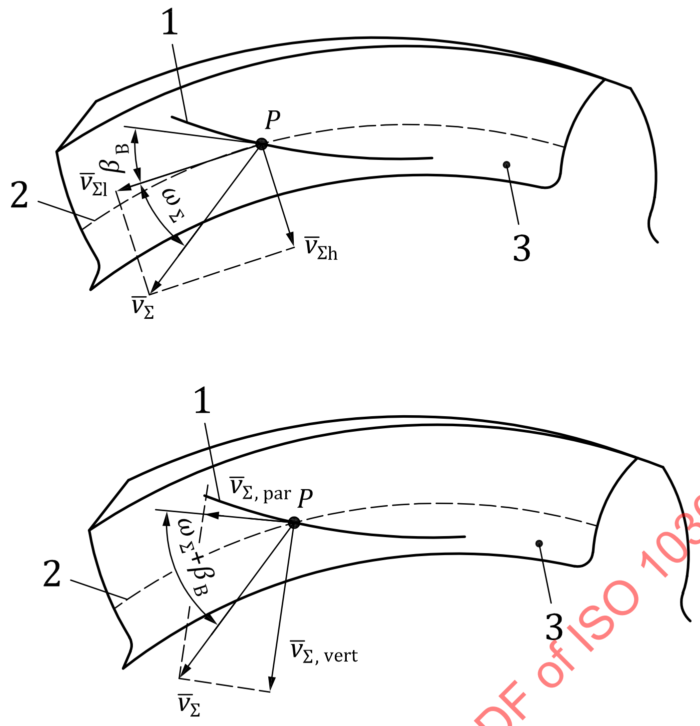
$$v_{\Sigma} = \sqrt{v_{\Sigma h}^2 + v_{\Sigma l}^2} \tag{17}$$

The inclination angle of the sum of velocities vector results from Formula (18):

$$\omega_{\Sigma} = \left|\arctan\left(\frac{v_{\Sigma h}}{v_{\Sigma l}}\right)\right| \tag{18}$$

The sum of velocities vertical to the contact line derives from Formula (19):

$$v_{\Sigma,vert} = v_{\Sigma} \cdot \sin\left(\omega_{\Sigma} + \left|\beta_B\right|\right) \tag{19}$$



Key

1 contact line
2 trace of pitch cone
3 pinion flank

Figure 5 — Sum of velocities vertical and parallel to the contact line in the mean point *P*

7 Gear flank rating formulae — Method B2

7.1 Contact stress formula

Calculations shall be carried out for pinion and wheel together; in case of hypoid gears, generally only the drive side is considered:

$$\sigma_{H-B2} = \sigma_{H0-B2} \cdot \sqrt{K_A \cdot K_v \cdot K_{H\beta}} \cdot Z_A \leq \sigma_{HP-B2} \tag{20}$$

where

the load factors K_A, K_v and $K_{H\beta}$ are specified in ISO 10300-1;

Z_A is the contact stress adjustment factor (see 7.4.4).

The nominal value of the contact stress is calculated with Formula (21):

$$\sigma_{H0-B2} = \sqrt{\frac{F_{mt1} \cdot d_{m1} \cdot Z_{FW}}{b_2 \cdot Z_I} \cdot \left(\frac{z_2}{d_{e2} \cdot z_1}\right)^2} \cdot Z_E \tag{21}$$

where

F_{mt1} is the nominal tangential force at mid-facewidth of the pinion (see ISO 10300-1:2023, Clause 6);

d_{m1} is the mean pitch diameter of the pinion;

Z_{FW} is the facewidth factor (see 7.4.3);

d_{e2} is the outer pitch diameter of the wheel;

b_2 is the facewidth of the wheel;

$z_{1,2}$ is the number of teeth of pinion and wheel;

Z_I is the macropitting resistance geometry factor (see 7.4.2);

Z_E is the elasticity factor (see 8.1).

7.2 Permissible contact stress

The permissible contact stress is calculated separately for pinion (suffix 1) and wheel (suffix 2) according to Formula (22):

$$\sigma_{HP-B2} = \sigma_{H,lim} \cdot Z_{NT} \cdot Z_L \cdot Z_v \cdot Z_R \cdot Z_W \tag{22}$$

where

$\sigma_{H,lim}$ is the allowable stress number (contact), which accounts for material, heat treatment, and surface influence at test gear dimensions as specified in ISO 6336-5;

Z_{NT} is the life factor, which accounts for the influence of required numbers of cycles of operation (see 8.4);

Z_L, Z_v, Z_R are the lubricant film factors (see 8.2) for the influence of the lubrication conditions;

Z_W is the work hardening factor (see 8.3), which considers less hardening of the softer wheel running with a surface hardened pinion.

7.3 Calculated safety factor for contact stress

The calculated safety factor for contact stress according to Formula (23) shall be checked separately for pinion and wheel, if the values of permissible contact stress are different:

$$S_{H-B2} = \frac{\sigma_{HP-B2}}{\sigma_{H-B2}} > S_{H,min} \tag{23}$$

where $S_{H,lim}$ is the minimum safety factor; for recommended values, see ISO 10300-1:2023, 5.2.

NOTE Formula (23) defines the calculated safety factor S_H with respect to contact stress. A safety factor related to the transferable torque is equal to the square of S_H.

7.4 Contact stress factors

7.4.1 General

The base unit of one diametral pitch, 1,0/m_{et2}, is used in the formulae in 7.4.

7.4.2 Macropitting resistance geometry factor, Z_I

7.4.2.1 General

The geometry factor evaluates the relative radius of curvature of the mating tooth flanks and the load sharing between adjacent pairs of teeth at that point on the tooth surfaces where the calculated contact pressure reaches its maximum value. Formulae (24) to (46) should be used; see ISO 10300-1:2023, Clause B.2. Because of the complexity of the calculation, computerization is recommended.

ANSI/AGMA 2003[6] provides graphs for bevel gears that may be used to determine the geometry factor, Z_I, whenever the gear parameters correspond to those in the graphs. Corresponding graphs for hypoid gears may be taken from AGMA 932[7].

7.4.2.2 Initial formulae

The angle between contact direction and tooth tangent in pitch plane:

$$\cot(\beta_{m1} - \lambda_1) = \frac{\cos\zeta_{mp}}{\cos\beta_{m1} \cdot \cos\beta_{m2} \cdot \tan(\beta_{m1} - \lambda_r)} - \tan\beta_{m2} \quad (24)$$

with $(\beta_{m1} - \lambda_r)$ as specified in ISO 10300-1:2023, Formula (B.15).

The angle between projection of pinion axis and direction of contact in pitch plane is given by:

$$\lambda_1 = \beta_{m1} - (\beta_{m1} - \lambda_1) \quad (25)$$

with $(\beta_{m1} - \lambda_1)$ as defined in Formula (24).

The angle of contact line relative to the root cone is given by:

$$\tan w = \frac{\sin\alpha_a \cdot \tan(\beta_{m1} - \lambda_r)}{\cos\alpha_{lim}} \quad (26)$$

with α_a = adjusted pressure angle, see ISO 10300-1:2023, Formula (B.16).

Mean base spiral angle:

$$\cos\beta_{bm} = \frac{1{,}0}{\sqrt{\tan^2(\beta_{m1} - \lambda_r)\cdot\cos^2\alpha_a + 1{,}0}} \tag{27}$$

Relative mean normal base pitch:

$$p_{nb} = \frac{\pi\cdot m_{mn}\cdot\cos\alpha_a\cdot\cos\beta_{bm}}{m_{et2}\cdot\cos(\beta_{m1} - \lambda_r)} \tag{28}$$

The angle between projection of wheel axis and direction of contact in pitch plane is given by:

$$\lambda_2 = (\beta_{m1} - \lambda_r) - \beta_{m2} \tag{29}$$

Relative base facewidth:

$$b_b = \frac{b_2}{m_{et2}\cdot\cos\lambda_2} \tag{30}$$

Pressure angle at point of load application:

$$\cos\alpha_{L1,2} = \cos\alpha_a\cdot\left[1{,}0 - \frac{(r_{va1,2} - r_{vn1,2})\cdot\cos^2\beta_{m1,2}}{r_{va1,2} - r_{vn1,2} + R_{mpt1,2}}\right] \tag{31}$$

Relative radius of curvature difference between point of load application and mean point:

$$\rho_{\Delta1,2} = \frac{r_{va1,2} - r_{vn1,2} + R_{mpt1,2}}{(\cos\beta_{m1,2})^2}\cdot\cos\alpha_{L1,2}\cdot(\tan\alpha_{L1,2} - \tan\alpha_a) \tag{32}$$

Relative radius of curvature change:

$$\rho_{\Delta red} = \cos\beta_{bm}\cdot(\rho_{\Delta1} + \rho_{\Delta2}) \tag{33}$$

Relative length of action within the contact ellipse is given by:

$$g_\eta = \sqrt{\rho_{\Delta red}^2\cdot(\cos\beta_{bm})^2 + b_b^2\cdot(\sin\beta_{bm})^2} \tag{34}$$

7.4.2.3 Radius of relative profile curvature, ρ_0, and load sharing ratio at critical point, ε_{NI}

The critical point on the tooth surface occurs when the contact line passes through a point at a distance, y_I, from the midpoint of the length of action. The value of y_I is chosen to produce the minimum value of Z_I which corresponds to the point of maximum contact stress.

a) For straight bevel and Zerol bevel gears, the contact line passes close to the lowest point of single tooth contact on the pinion, in which case:

$$y_I = 0{,}5\cdot g_{v\alpha n} - p_{nb} \tag{35}$$

In this case, the geometry factor, Z_I, is calculated using Formulae (36) to (46) without iteration.

b) For spiral bevel and hypoid gears, it is necessary to start an iteration procedure. As initial value y_I = 0 is assumed and Z_I is calculated using Formulae (36) to (46).

Relative length of action at critical point in contact ellipse:

$$g_{\eta\mathrm{I}} = \sqrt{g_{\eta}^{2} - 4{,}0 \cdot y_{\mathrm{I}}^{2}} \tag{36}$$

Relative length of action considering adjacent teeth:

$$g_{\eta\mathrm{I}\Sigma}^{3} = g_{\eta\mathrm{I}}^{3} + \sum_{k=1}^{k=x} \sqrt{\left[g_{\eta\mathrm{I}}^{2} - 4 \cdot k \cdot p_{\mathrm{nb}} \cdot \left(k \cdot p_{\mathrm{nb}} + 2 \cdot y_{\mathrm{I}}\right)\right]^{3}} + \sum_{k=1}^{k=y} \sqrt{\left[g_{\eta\mathrm{I}}^{2} - 4 \cdot k \cdot p_{\mathrm{nb}} \cdot \left(k \cdot p_{\mathrm{nb}} - 2 \cdot y_{\mathrm{I}}\right)\right]^{3}} \tag{37}$$

In Formula (37), k is a positive integer which takes on successive values from 1 to x or y, generating all real terms (positive values under the radical) in each series. Imaginary terms (negative values under the radical) shall be set to zero. For most designs, x and y will not be greater than 2 until summation ends.

Load sharing ratio:

$$\varepsilon_{\mathrm{NI}} = g_{\eta\mathrm{I}}^{3} / g_{\eta\mathrm{I}\Sigma}^{3} \tag{38}$$

Relative length of contact line:

$$g_{\mathrm{c}} = g_{\eta\mathrm{I}} \cdot \rho_{\Delta\mathrm{red}} \cdot b_{\mathrm{b}} / g_{\eta}^{2} \tag{39}$$

Relative position change along path of contact:

$$g_{\eta\Delta} = \frac{\rho_{\Delta\mathrm{red}}^{2} \cdot y_{\mathrm{I}}}{g_{\eta\mathrm{I}}^{2}} + k' \cdot g_{\mathrm{c}} \cdot \tan\beta_{\mathrm{bm}} + \frac{0{,}5 \cdot \rho_{\Delta\mathrm{red}}}{\cos\beta_{\mathrm{bm}}} - \rho_{\Delta 2} \tag{40}$$

where k' is the contact shift factor as specified in ISO 10300-1:2023, Clause B.5.

Intermediate value:

$$X = \frac{\sin^{2} w \cdot \cos\alpha_{\lim} \cdot \cos\left(\zeta_{\mathrm{mp}} - \lambda_{1}\right) \cdot \cos\lambda_{1}}{\sin^{2}\left(\beta_{\mathrm{m1}} - \lambda_{1}\right) \cdot \sin\alpha_{\mathrm{a}} \cdot \cos\zeta_{\mathrm{mp}}} \tag{41}$$

Relative profile radius of curvature:

$$\rho_{1,2} = R_{\mathrm{mpt}1,2} \cdot X \pm g_{\eta\Delta} \tag{42}$$

Relative radius of profile curvature between pinion and wheel:

$$\rho_{\mathrm{t}} = \frac{\rho_{1} \cdot \rho_{2}}{\rho_{1} + \rho_{2}} \tag{43}$$

The inertia factor, Z_{i}, is determined depending on the modified contact ratio, $\varepsilon_{\mathrm{v}\gamma}$:

a) For $\varepsilon_{\mathrm{v}\gamma} \leq 2{,}0$:

$$Z_{\mathrm{i}} = 2{,}0 / \varepsilon_{\mathrm{v}\gamma} \tag{44}$$

b) For $\varepsilon_{\mathrm{v}\gamma} > 2{,}0$:

$$Z_{\mathrm{i}} = 1{,}0 \tag{45}$$

Macropitting resistance geometry factor:

$$Z_{\mathrm{I}} = \frac{g_{\mathrm{c}} \cdot \rho_{\mathrm{t}} \cdot \cos\alpha_{\mathrm{a}} \cdot m_{\mathrm{mn}}}{b_{\mathrm{b}} \cdot z_1 \cdot Z_{\mathrm{i}} \cdot \varepsilon_{\mathrm{NI}} \cdot m_{\mathrm{et2}}} \tag{46}$$

Formulae (36) to (46) shall be recalculated by stepping y_{I} in both directions until a minimum of Z_{I} is found. End of iteration

7.4.3 Facewidth factor, Z_{FW}

The facewidth factor, Z_{FW}, reflecting non-uniformity of material properties, depends primarily on:

— tooth size (diameter of part);

— ratio of tooth size to diameter of part;

— facewidth;

— area of stress pattern;

— material characteristics.

Although the facewidth factor is a function of the strength of the material and therefore should appear in the formulae for permissible stress, it is more practicable to include it in the formulae for calculated stress as is done in method B2. Then, it is possible to prepare S-N-diagrams from experimental data using a wide range of gears with varying tooth sizes. Otherwise, one would be limited to using gears of only one tooth size. The facewidth factor for macropitting resistance of bevel gears, without sufficient experience, is determined as a size factor being dependent on the facewidth according to Formulae (47) to (49):

a) for $b_2 < 12{,}7$ mm:

$$Z_{\mathrm{FW}} = 0{,}5 \tag{47}$$

b) for $12{,}7 \le b_2 \le 79{,}8$ mm:

$$Z_{\mathrm{FW}} = 0{,}004\ 92 \cdot b_2 + 0{,}437\ 5 \tag{48}$$

In Formula (48), b_2 shall be used without unit.

c) for $b_2 > 79{,}8$ mm:

$$Z_{\mathrm{FW}} = 0{,}83 \tag{49}$$

7.4.4 Contact stress adjustment factor, Z_{A}

The contact stress adjustment factor Z_{A}, adjusts the calculation results of method B2 so that the contact stress numbers in ISO 6336-5 can be used. The determination of Z_{A} is based on the comparison of ISO 6336-5 MQ grade carburized case-hardened steel which has an allowable stress number of 1 500 N/mm^2, to the equivalent ANSI/AGMA 2003[6] grade 2 case hardened steel, which has an allowable stress number of 1 550 N/mm^2.

Hence it follows for carburized case-hardened steel that according to Formula (50):

$$Z_{\mathrm{A}} = 0{,}967 \tag{50}$$

For other specific materials and qualities, Z_{A} shall be calculated by taking the ratio of the allowable contact stress number in ISO 6336-5 to equivalent ANSI/AGMA 2003[6] steel.

8 Factors for contact stress and permissible contact stress common for method B1 and method B2

8.1 Elasticity factor, Z_E

The elasticity factor, Z_E, accounts for the influence of the material specific quantities E (modulus of elasticity) and ν (Poisson's ratio) on the contact stress, see Formula (51).

$$Z_E = \sqrt{\frac{1}{\pi \cdot \left[\frac{\left(1-\nu_1^2\right)}{E_1} + \frac{\left(1-\nu_2^2\right)}{E_2} \right]}} \quad (51)$$

For $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = \nu$, Formula (52) applies:

$$Z_E = \sqrt{\frac{E}{2 \cdot \pi \cdot \left(1-\nu^2\right)}} \quad (52)$$

If a pair of gears is made from materials having modules of elasticity, E_1 and E_2, E should be determined by Formula (53):

$$E = \frac{2 \cdot E_1 \cdot E_2}{E_1 + E_2} \quad (53)$$

For Z_E of some other gear pair materials, see ISO 6336-2[1].

8.2 Lubricant film influence factors, Z_L, Z_v, Z_R

8.2.1 General

The influences on the lubricant film between the tooth flanks are approximated by the factors Z_L (oil viscosity), Z_v (tangential speed) and Z_R (flank roughness). When a gear pair consists of one member with a lower surface hardness than the other one, the lubricant film influence factors shall be determined for the member with the lower surface hardness. In Formulae (54) to (60), the allowable stress number for contact stress $\sigma_{H,lim}$ of the material with the lower surface hardness shall be used. These factors will be applied to both members.

NOTE For additional general remarks about these three factors, see ISO 6336-2[1].

8.2.2 Restrictions

When there are no comprehensive experiences or test results (method A), Z_L, Z_v and Z_R shall be determined separately according to method B (see 8.2.3). However, in many cases, in fact for most industrial gears, the shorter method, i.e. method C (see 8.2.4), may be used instead.

8.2.3 Method B

8.2.3.1 Lubricant factor, Z_L

Taking into account the restrictions given in 8.2.2, the lubricant factor, Z_L, accounts for the influence of the type of lubricant, and its viscosity, on the surface durability (macropitting).

This document does not include a recommendation as to the choice of oil viscosity, which shall only be made with reference to testing, experience or gear lubrication publications.

Z_L should be calculated using Formulae (54) and (55).

$$Z_L = C_{ZL} + \frac{4 \cdot (1{,}0 - C_{ZL})}{\left(1{,}2 + \frac{134}{v_{40}}\right)^2} \tag{54}$$

For the range of $\sigma_{H,lim}$ = 850 N/mm² to $\sigma_{H,lim}$ = 1 200 N/mm², the following applies:

$$C_{ZL} = 0{,}08 \cdot \frac{\sigma_{H,lim} - 850}{350} + 0{,}83 \tag{55}$$

For $\sigma_{H,lim}$ values below 850 N/mm², the Z_L value for $\sigma_{H,lim}$ = 850 N/mm² is used, while for $\sigma_{H,lim}$ values above 1 200 N/mm², the Z_L value for $\sigma_{H,lim}$ = 1 200 N/mm² is used.

8.2.3.2 Speed factor, Z_v

Taking into account the restrictions given in 8.2.2, the indicated speed factor, Z_v, accounts for the influence of the tangential speed on the surface durability (macropitting).

Z_v should be calculated using Formulae (56) and (57).

$$Z_v = C_{ZV} + \frac{2 \cdot (1{,}0 - C_{ZV})}{\sqrt{0{,}8 + \frac{32}{v_{mt2}}}} \tag{56}$$

For the range of $\sigma_{H,lim}$ = 850 N/mm² to $\sigma_{H,lim}$ = 1 200 N/mm², Formula (57) applies:

$$C_{ZV} = 0{,}08 \cdot \frac{\sigma_{H,lim} - 850}{350} + 0{,}85 \tag{57}$$

For $\sigma_{H,lim}$ values below 850 N/mm², the Z_v value for $\sigma_{H,lim}$ = 850 N/mm² is used, while for $\sigma_{H,lim}$ values above 1 200 N/mm², the Z_v value for $\sigma_{H,lim}$ = 1 200 N/mm² is used.

8.2.3.3 Roughness factor, Z_R

Taking into account the restrictions given in 8.2.2, the indicated roughness factor, Z_R, accounts for the influence of the surface condition of the tooth flanks on the surface durability (macropitting).

The mean roughness shall be determined for the values Rz_1 and Rz_2 of the pinion and the wheel after manufacturing. Allowance shall be made for any special surface treatment or running-in process. The roughness measured in the direction of the sliding and rolling movement shall be decisive.

The mean relative roughness is given by Formula (58):

$$Rz_{10} = \frac{Rz_1 + Rz_2}{2} \cdot \sqrt[3]{\frac{10}{\rho}} \tag{58}$$

with the radius of relative curvature $\rho = \rho_{rel}$ for method B1 (see ISO 10300-1:2023, Annex A), $\rho = \rho_t \cdot m_{et2}$ for method B2 [see Formula (43)].

When the roughness is given as an Ra value (= CLA value, AA value), the following approximation may be used: Ra = CLA = AA = Rz/6.

The factor Z_R should be calculated using Formulae (59) and (60).

Roughness factor:

$$Z_R = \left(\frac{3}{Rz_{10}} \right)^{C_{ZR}} \tag{59}$$

In the range of 850 N/mm² ≤ $\sigma_{H,lim}$ ≤ 1 200 N/mm², the following applies:

$$C_{ZR} = 0{,}12 + \frac{1\ 000 - \sigma_{H,lim}}{5\ 000} \tag{60}$$

For $\sigma_{H,lim}$ values below 850 N/mm², use $\sigma_{H,lim}$ = 850 N/mm², while for $\sigma_{H,lim}$ values above 1 200 N/mm², use $\sigma_{H,lim}$ = 1 200 N/mm².

8.2.4 Method C (product of Z_L, Z_v and Z_R)

As a simplification of method B, it is assumed that a proper lubricant viscosity has been chosen for the operating conditions (tangential speed, load, structural size).

The following values apply for the product of Z_L, Z_v and Z_R:

— for through hardened gear pairs without finishing process: 0,85;

— for gear pairs lapped after hardening: 0,92;

— for gear pairs ground after hardening, or for hard cut gear pairs, with:

— Rz_{10} ≤ 4 µm: $Z_L \cdot Z_v \cdot Z_R$ = 1,0;

— Rz_{10} > 4 µm: $Z_L \cdot Z_v \cdot Z_R$ = 0,92.

If these conditions do not apply, Z_L, Z_v and Z_R shall be determined separately according to method B.

8.3 Work hardening factor, Z_W

8.3.1 General

The work hardening factor, Z_W, takes account of the increase in the surface durability due to meshing a steel wheel (structural steel, through-hardened steel) with a hardened or substantially harder pinion with smooth tooth flanks.

The increase in the surface durability of the soft wheel depends not only on any work hardening of this wheel, but also on other influences such as polishing (lubricant), alloying element and internal stresses in the soft material, surface roughness of the hard pinion, contact stress and hardening processes.

8.3.2 Work hardening factor, Z_W: Method A

The increase in load-bearing capacity as a result of the influences listed above shall be determined in accordance with the reliable operating experience or tests on geared transmissions of comparable dimensions, materials, lubricants and operating conditions. The provisions given in ISO 10300-1:2023, 5.1.1 are relevant.

8.3.3 Work hardening factor, Z_W: Method B

8.3.3.1 Surface-hardened steel pinion with through-hardened steel gear

8.3.3.1.1 General

The data provided are based on tests of different materials, using standard reference test gears, as well as production gearing field experience.

Although the Formulae (61) to (67) were carefully determined, they cannot be interpreted as a physical law for the reasons mentioned above. They are empirical, like Formula (61).

The equivalent roughness Rz_H, is determined as

$$Rz_H = \frac{Rz_1 \cdot (10/\rho_{rel})^{0,33} \cdot (Rz_1/Rz_2)^{0,66}}{(\nu_{40} \cdot v_{mt2}/1\ 500)^{0,33}} \tag{61}$$

if $Rz_H > 16$ then $Rz_H = 16$ µm

if $Rz_H < 3$ then $Rz_H = 3$ µm

where

Rz_1 is the surface roughness of the harder pinion, in micrometres (µm) before running-in;

Rz_2 is the surface roughness of the softer wheel, in micrometres (µm) before running-in;

ρ_{rel} is the local equivalent radius of curvature, in millimetres (mm);

ν_{40} is the nominal kinematic viscosity at 40 °C, in square millimetres per second (mm^2/s);

v_{mt2} is the tangential speed at mid-force of reference cone wheel, in metres per second (m/s).

The value of Z_W is different for static, limited life and reference stress.

Especially for rough pinion surfaces, values of $Z_W < 1$ can be evaluated. As in this range, effects of wear can limit the surface durability, Z_W is fixed at $Z_W = 1$. An additional analysis concerning wear should be carried out in this case. Wear of the surface is not covered by the ISO 10300 series.

8.3.3.1.2 Z_W for reference and long-life stress

For 130 ≤HBW ≤ 470, Z_W for reference and long-life stress is calculated using Formula (62):

$$Z_W = \left(1,2 - \frac{\mathrm{HBW} - 130}{1\ 700}\right) \cdot \left(\frac{3}{Rz_H}\right)^{0,15} \tag{62}$$

where

HBW is the Brinell hardness of the tooth flanks of the softer gear of the pair;

Rz_H is the equivalent roughness according to Formula (61).

For HBW < 130, Formula (63) applies:

$$Z_W = 1{,}2 \cdot \left(\frac{3}{Rz_H} \right)^{0{,}15} \tag{63}$$

For HBW > 470, Formula (64) applies:

$$Z_W = \left(\frac{3}{Rz_H} \right)^{0{,}15} \tag{64}$$

8.3.3.1.3 Z_W for static stress

For 130 ≤ HBW ≤ 470, Z_W for the static stress range is calculated using Formula (65):

$$Z_W = 1{,}05 - \frac{\text{HBW} - 130}{6\ 800} \tag{65}$$

For HBW < 130, Formula (66) applies:

$$Z_W = 1{,}05 \tag{66}$$

For HBW > 470, Formula (67) applies:

$$Z_W = 1 \tag{67}$$

8.3.3.2 Through-hardened steel pinion with through-hardened steel gear

8.3.3.2.1 General

When the pinion is substantially harder than the gear the work hardening effect increases the load capacity of the gear flanks. Z_W applies to the gear only, not on the pinion.

The value of Z_W is different for static, limited life and reference stress.

8.3.3.2.2 Z_W for reference and long-life stress

For 1,2 ≤ $\text{HBW}_1/\text{HBW}_2$ ≤ 1,7, Z_W for long-life stress is determined by using Formula (68):

$$Z_W = 1{,}0 + A \cdot (u - 1{,}0) \tag{68}$$

where

$$A = 0{,}008\ 98 \cdot \text{HBW}_1 / \text{HBW}_2 - 0{,}008\ 29 \tag{69}$$

HBW_1 is the Brinell hardness of the pinion;

HBW_2 is the Brinell hardness of the gears;

u is the gear ratio; if u > 20 then u = 20.

For $\text{HBW}_1/\text{HBW}_2$ < 1,2, Formula (70) applies:

$$Z_W = 1{,}0 \tag{70}$$

For $\text{HBW}_1/\text{HBW}_2$ > 1,7, Formula (71) applies:

$$Z_W = 1{,}0 + 0{,}006\ 98 \cdot (u - 1{,}0) \tag{71}$$

8.3.3.2.3 Z_W for static stress

For the static stress range, Formula (72) applies:

$$Z_W = 1{,}0 \tag{72}$$

8.3.3.3 Surface-hardened steel pinion with ductile iron gear

8.3.3.3.1 General

When the pinion is surface hardened significantly harder than the ductile iron gear the work hardening effect increases the load capacity of the gear flanks. Z_w applies to the gear only, not to the pinion.

The value of Z_W is different for static, limited life and reference stress.

8.3.3.3.2 Z_W for reference and long-life stress

For 162 ≤ HBW ≤ 344, Z_W for long life stress is determined by using Formula (73):

$$Z_W = 1{,}2 - \frac{1{,}87 \cdot \mathrm{HBW} - 303{,}6}{1\ 700} \tag{73}$$

For HBW < 162, Formula (74) applies:

$$Z_W = 1{,}2 \tag{74}$$

For HBW > 344, Formula (75) applies:

$$Z_W = 1{,}0 \tag{75}$$

where HBW is the Brinell hardness of the gear.

8.3.3.3.3 Z_W for static stress

For the static stress range, Formula (76) applies:

$$Z_W = 1{,}0 \tag{76}$$

8.4 Life factor, Z_{NT}

8.4.1 General

The life factor, Z_{NT}, accounts for the higher permissible contact stress, which may be acceptable for a limited life (number of load cycles), as compared with the allowable stress at the lower point or “knee” on the curves of Figure 6 where Z_{NT} = 1,0. For extended life, Z_{NT} may be less than 1,0. Z_{NT} has been determined for standard test gear conditions (see ISO 6336-6[2] for further information). Z_{NT} shall be determined for pinion and wheel separately.

The main influences related to Z_{NT} are:

a) material and heat treatment (see 5.2 and ISO 6336-5);

b) number of load cycles (service life), N_L;

c) lubrication regime;

d) failure criteria;

e) required smoothness of operation;

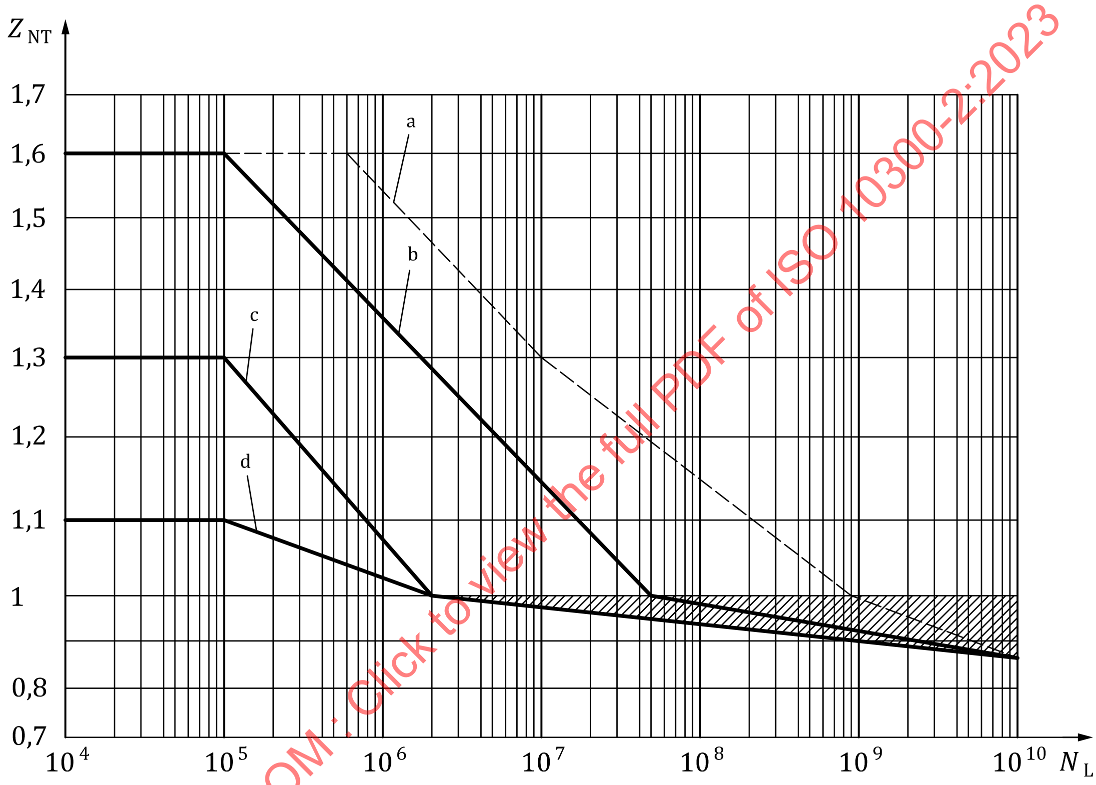
f) pitch line velocity;

g) gear material cleanliness;

h) material ductility and fracture toughness;

i) residual stress.

For the purposes of this document, the number of load cycles, N_L, is identified as the number of mesh contacts, under load, of the gear being analysed.



Key

N_L number of load cycles

Z_{NT} life factor

[a] St, V, GGG (perl. bain.), GTS (perl.), Eh, IF, when limited macropitting permitted.

[b] St, V, Eh, IF, GGG (perl. bain.), GTS (perl.).

[c] GG, NT (nitr.), GGG (ferr.), NV (nitr.).

[d] NV (nitrocar.).

Figure 6 — Life factor for macropitting resistance, Z_{NT}

8.4.2 Method A

The S-N, or damage curve, derived from examples of the actual gear pair, is determinant for load capacity at limited service life. Thus, it is also determinant for the materials of both mating gears, heat treatment, relevant diameter, module, surface roughness of tooth flanks, pitch line velocity and lubricant. Since the S-N/damage curve is directly valid for the conditions mentioned, the influences represented by the factors Z_R, Z_v, Z_L, Z_W and Z_X are included in the curve, and should therefore be assigned the value 1,0 in the calculation formulae.