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Calculation of load capacity of bevel gears —

Part 2: Calculation of surface durability (pitting)

Calcul de la capacité de charge des engrenages coniques —

Partie 2: Calcul de la résistance à la pression superficielle (formation des piqûres)

Reference number
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Foreword

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The committee responsible for this document is ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

This second edition cancels and replaces the first edition (ISO 10300-2:2001), which has been technically revised.

ISO 10300 consists of the following parts, under the general title *Calculation of load capacity of bevel gears*:

— *Part 1: Introduction and general influence factors*

— *Part 2: Calculation of surface durability (pitting)*

— *Part 3: Calculation of tooth root strength*

Introduction

When ISO 10300:2001 (all parts, withdrawn) became due for (its first) revision, the opportunity was taken to include hypoid gears, since previously the series only allowed for calculating the load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, it was agreed to include a separate clause: “Gear flank rating formulae — Method B2” in this part of ISO 10300, while the former method B was renamed method B1. So, it became necessary to present a new, clearer structure of the three parts, which is illustrated in ISO 10300-1:2014, Figure 1. Note, ISO 10300 (all parts) gives no preferences in terms of when to use method B1 and when method B2.

This part of ISO 10300 deals with the failure of gear teeth by pitting, a fatigue phenomenon. Two varieties of pitting are recognized, initial and destructive pitting.

In applications employing low hardness steel or through hardened steel, initial pitting frequently occurs during early use and is not deemed serious. Initial pitting is characterized by small pits which do not extend over the entire face width or profile depth of the affected tooth. The degree of acceptability of initial pitting varies widely, depending on the gear application. Initial pitting 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 pitting stops.

In applications employing high hardness steel and case carburized steel, the variety of pitting that occurs is usually destructive. The formulae for pitting resistance given in this part of ISO 10300 are intended to assist in the design of bevel gears which stay free from destructive pitting during their design lives (for additional information, see ISO/TR 22849[4]).

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 pitting. 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 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 longitudinal and transverse load distribution factors.

Calculation of load capacity of bevel gears —

Part 2: Calculation of surface durability (pitting)

1 Scope

This part of ISO 10300 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 part of ISO 10300 is applicable to oil lubricated bevel gears, as long as sufficient lubricant is present in the mesh at all times.

The formulae in this part of ISO 10300 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 (see ISO 6336-2[1]). Additionally, the given relations are valid for bevel gears of which the sum of profile shift coefficients of pinion and wheel is zero (see ISO 23509).

The formulae in this part of ISO 10300 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.

WARNING — 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 face widths $b > 13\ m_{mn}$, the calculated results of ISO 10300 should be confirmed by experience.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable to its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

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 10300-1:2014, *Calculation of load capacity of bevel gears — Part 1: Introduction and general influence factors*

ISO 10300-3, *Calculation of load capacity of bevel gears — Part 3: Calculation of tooth root strength*

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 (geometrical gear terms) and the following apply.

3.1
pitting
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
$\sigma_{H,lim}$
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, units and abbreviated terms

For the purposes of this document, the symbols and units given in ISO 10300-1:2014, Table 1 and Table 2, as well as the following abbreviated terms, apply (see ISO 6336-5).

Table 1 — 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 Pitting damage — General aspects

5.1 Acceptable versus unacceptable pitting

When limits of the surface durability of the meshing flanks are exceeded, particles break out of the flank, thus leaving pits. 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 pitting is acceptable; in others, no pitting 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, pitting varieties.

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

5.2 Assessment requirements

Pitting 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 pitting is such that it puts human life in danger, or poses a risk of other grave consequences, the pitting 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 could 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 pitting nor unduly severe wear should be considered acceptable as such damage could 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, pitting 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 pitting”, 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 pitting 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 formula (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 (14) 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.

It is recommended that the gear designer and customer agree on the value of the minimum safety factor.

6 Gear flank rating formulae — Method B1

6.1 Contact stress formula

The calculation of pitting resistance is based on the contact (Hertzian) stress, in which the load is distributed along the lines of contact (see ISO 10300-1:2014, Annex A). Calculations are to be carried out for pinion and wheel together; however, in case of hypoid gears separately for drive (suffix D) and coast side flank (suffix C):

$$\sigma_{H\text{-}B1} = \sigma_{H0\text{-}B1}\sqrt{K_A K_v K_{H\beta} K_{H\alpha}} \le \sigma_{HP\text{-}B1} \qquad (1)$$

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:

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

where F_n is the nominal normal force of the virtual cylindrical gear at mean point P:

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

with

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

$\alpha_n = \alpha_{nC}$ = 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 10300-1:2014, A.2.7;

ρ_{rel} is the radius of relative curvature vertical to the contact line as specified in ISO 10300-1:2014, A.2.8;

$Z_{M\text{-}B}$ 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);

Z_K is the bevel gear factor which accounts for the influence of the bevel gear geometry (see 6.4.3).

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):

$$\sigma_{\text{HP-B1}} = \sigma_{\text{H,lim}} Z_{\text{NT}} Z_{\text{X}} Z_{\text{L}} Z_{\text{v}} Z_{\text{R}} Z_{\text{W}} Z_{\text{Hyp}} \quad (4)$$

where

$\sigma_{\text{H,lim}}$	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 6.5.1), 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_{Hyp}	is the hypoid factor (see 6.5.2), 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 shall be checked separately for pinion and wheel, if the values of permissible contact stress are different:

$$S_{\text{H-B1}} = \frac{\sigma_{\text{HP-B1}}}{\sigma_{\text{H-B1}}} > S_{\text{H,min}} \quad (5)$$

where $S_{\text{H,min}}$ is the minimum safety factor; see 5.2 of ISO 10300-1:2014 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, $Z_{\text{M-B}}$

The mid-zone factor, $Z_{\text{M-B}}$, considers the difference between the radius of relative 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 10300-1:2014, 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 ($\varepsilon_{\text{v}\beta} = 0$) or point M in the middle of the path of contact ($\varepsilon_{\text{v}\beta} \geq 1$) or a point interpolated between B and M for $0 < \varepsilon_{\text{v}\beta} < 1$ (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.

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