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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
ISO 10300-2:2001(E)

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Contents

Page

Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this part of ISO 10300 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 10300-2 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

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*

Annex A of this part of ISO 10300 is for information only.

This corrected version of ISO 10300-2:2001 incorporates the following corrections:

equations (16) and (18), and the date of publication of ISO 10300-1 given in Clause 4, have been corrected.

Introduction

Parts 1, 2 and 3 of ISO 10300, taken together with ISO 6336-5, are intended to establish general principles and procedures for the calculation of the load capacity of bevel gears. Moreover, ISO 10300 has been designed to facilitate the application of future knowledge and developments, as well as the exchange of information gained from experience.

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.

On the one hand, in applications employing low-hardness steel or through-hardened steel, corrective (non-progressive) 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 over-stressed areas, and tends to redistribute the load by progressively removing high contact spots. Generally, when the load has been redistributed, the pitting stops.

On the other hand, 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 ISO 10300 are intended to assist in the design of gears that will be free from destructive pitting during their design life.

The basic formulae, first developed by Hertz for the contact pressure between two curved surfaces, have been modified to consider 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 in respect of 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 given 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 points lie within the field of application range, Hertzian pressure can be used as a type of model theory to aid in the conversion of test-gear data to gears of various types and sizes.

NOTE In contrast to cylindrical gears, where the contact is mostly linear, bevel gears are generally manufactured with 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 $K_{H\beta}$ and $K_{H\alpha}$ (see ISO 10300-1) by the fact that the rectangular pressure surface (in the case of linear contact) is replaced by an inscribed pressure ellipse. 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. Therefore, the general equations for the calculation of Hertzian pressure are similar for cylindrical and bevel gears.

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, and includes all the influences on surface durability for which quantitative assessments can be made. This part of ISO 10300 is applicable to oil-lubricated transmissions, as long as sufficient lubricant is present in the mesh at all times.

The formulae in ISO 10300 are valid for bevel gears with teeth where the transverse contact ratio is $\varepsilon_{v\alpha} < 2$. The results are valid within the range of the applied factors as indicated in ISO 10300-1, and in ISO 6336-2. However, the formulae in this part of ISO 10300 are not directly applicable in the assessment of certain types of gear-tooth surface damage, such as plastic yielding, scratching, scuffing or any other type not specified.

CAUTION — The user is cautioned that when the methods are used for large spiral and pressure angles, and for large face width $b > 10\ m_{mn}$, the calculated results of ISO 10300 should be confirmed by experience.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO 10300. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this part of ISO 10300 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 53:1998, *Cylindrical gears for general and heavy engineering — Standard basic rack tooth profile*.

ISO 1122-1:1998, *Vocabulary of gear terms — Part 1: Definitions related to geometry*.

ISO 1328-1, *Cylindrical gears — ISO system of accuracy — Part 1: Definitions and allowable values of deviations relevant to corresponding flanks of gear teeth*.

ISO 6336-2:1996, *Calculation of load capacity of spur and helical gears — Part 2: Calculation of surface durability (pitting)*.

ISO 6336-5:1996, *Calculation of load capacity of spur and helical gears — Part 5: Strength and quality of materials*.

ISO 10300-1:2001, *Calculation of load capacity of bevel gears — Part 1: Introduction and general influence factors*.

3 Terms and definitions

For the purposes of this part of ISO 10300, the geometrical gear terms given in ISO 53 and ISO 1122-1, and the following term and definition, apply.

3.1
surface load capacity
surface durability
load capacity determined by way of the permissible contact stress

4 Symbols and abbreviated terms

For the purposes of this part of ISO 10300, the symbols and abbreviated terms given in Table 1 of ISO 10300-1:2001, and the following abbreviated terms, apply.

Table 1 — Abbreviated terms

Abbreviation	Description
St	steel (σ_B < 800 N/mm^2)
V	through-hardened steel ($\sigma_B \geqslant$ 800 N/mm^2)
GG	grey cast iron
GGG (perl., bai., ferr.)	spheroidal cast iron (perlitic, bainitic, ferritic structure)
GTS (perl.)	black malleable cast iron (perlitic structure)
Eh	case-hardening steel, case hardened
IF	steel and GGG, flame or induction hardened
NT (nitr.)	nitriding steels, nitrided
NV (nitr.)	through-hardened and case-hardening steel, nitrided
NV (nitrocar.)	through-hardened and case-hardening steels, nitro-carburized

5 Pitting damage-assessment requirements and safety factors

5.1 Overview

When limits of the surface durability of the meshing flanks are exceeded, particles break out of the flanks, leaving pits. The extent to which such pits can 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 following descriptions are relevant to average working conditions, and give guidelines for distinguishing between the initial and destructive, acceptable and unacceptable, pitting varieties.

5.2 Acceptable vs. unacceptable pitting

A linear or progressive increase in the total area of the pits is generally considered to be unacceptable. However, the effective tooth bearing area can be enlarged by initial pitting, and the rate of pit generation could subsequently decrease (degressive pitting), or even cease (arrested pitting), and then be considered tolerable. Nevertheless, where there is dispute over the acceptability of pitting, the following shall be determinant.

Pitting involving the formation of pits which increase linearly or progressively with time under unchanged service conditions (linear or progressive pitting) 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, a first, rough assessment may 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 only take place after at least 10^6 cycles of load. Depending on the results of previous examinations, further ones should be made 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).

Considerations similar to those above 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 may be considered as 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 in over 100 % of the working 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 to 20 years. Here, 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. 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 cases less than one) may be chosen, with a correspondingly higher probability of tooth surface damage. However, a high factor of safety against tooth breakage shall be chosen.

The value of the minimum safety factor for contact stress, S_{Hmin}, should be 1,0 (for further recommendations on the choice of the contact-stress safety factor, S_H, and other minimum values, see ISO 10300-1).

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

6 Gear-tooth rating formulae

6.1 General

The capacity of a gear tooth to resist pitting shall be determined by the comparison of the following stress values:

— **contact stress**, 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 6.2.1);

— **allowable stress**, and the effect of the working conditions under which the gears operate, expressed by the permissible contact stress formula (see 6.2.2).

The calculation of pitting resistance is based on the contact (Hertzian) stress, in which the load is distributed over the lines of contact (see annex A of ISO 10300-1:2001). The determinant positions of load application are:

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

b) the mid-point of the zone of contact, ($\varepsilon_{v\beta} > 1$);

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

6.2 Contact stress

6.2.1 Contact stress formula

Calculations are to be made for pinion and wheel together:

$$\sigma_{H} = \sigma_{H0} \sqrt{K_{A} K_{v} K_{H\beta} K_{H\alpha}} \leqslant \sigma_{HP} \tag{1}$$

Hereby, the nominal value of the contact stress is:

$$\sigma_{H0} = \sqrt{\frac{F_{mt}}{d_{v1} l_{bm}} \cdot \frac{u_{v} + 1}{u_{v}}} \, Z_{M\text{-}B} Z_{H} Z_{E} Z_{LS} Z_{\beta} Z_{K} \tag{2}$$

For the shaft angle $\Sigma = \delta_1 + \delta_2 = 90°$ the following applies:

$$\sigma_{H0} = \sqrt{\frac{F_{mt}}{d_{m1} l_{bm}} \cdot \frac{\sqrt{u^{2} + 1}}{u}} \, Z_{M\text{-}B} Z_{H} Z_{E} Z_{LS} Z_{\beta} Z_{K} \tag{3}$$

For K_A, K_v, $K_{H\beta}$, $K_{H\alpha}$, F_{mt}, and d_v and u_v, l_{bm}, see ISO 10300-1:2001, in particular annex A for d_v and u_v, and l_{bm} for equations (A.42) and (A.43).

6.2.2 Permissible contact stress

The permissible contact stress is to be calculated separately for pinion and wheel:

$$\sigma_{HP} = \frac{\sigma_{Hlim} Z_{NT}}{S_{Hlim}} Z_{X} Z_{L} Z_{R} Z_{v} Z_{W} \tag{4}$$

For σ_{Hlim}, the endurance limit for contact stress, see ISO 6336-5.

6.2.3 Calculated safety factors for contact stress (against pitting)

The calculated safety factor for contact stress is to be checked separately for pinion and wheel:

$$S_{H} = \frac{\sigma_{Hlim} Z_{NT}}{\sigma_{H0}} \cdot \frac{Z_{X} Z_{L} Z_{R} Z_{v} Z_{W}}{\sqrt{K_{A} K_{v} K_{H\beta} K_{H\alpha}}} \tag{5}$$

NOTE This is the relationship of the calculated safety factor with respect to contact stress. Safety related to the transferable torque is equal to the square of S_H. See ISO 10300-1 for numerical values for the minimum safety factor, or the risk of failure (damage probability).

7 Zone factor, Z_H

The zone factor, Z_H, accounts for the influence of the flank curvature in the profile direction at the pitch point on the Hertzian pressure.

When an involute tooth profile is assumed, the following applies for x-zero bevel gears, where $x_1 + x_2 = 0$ and $\alpha_t = \alpha_{wt}$:

$$Z_H = 2\sqrt{\frac{\cos\beta_{vb}}{\sin(2\alpha_{vt})}} \tag{6}$$

For some common normal pressure angles, Z_H may be taken from Figure 1.

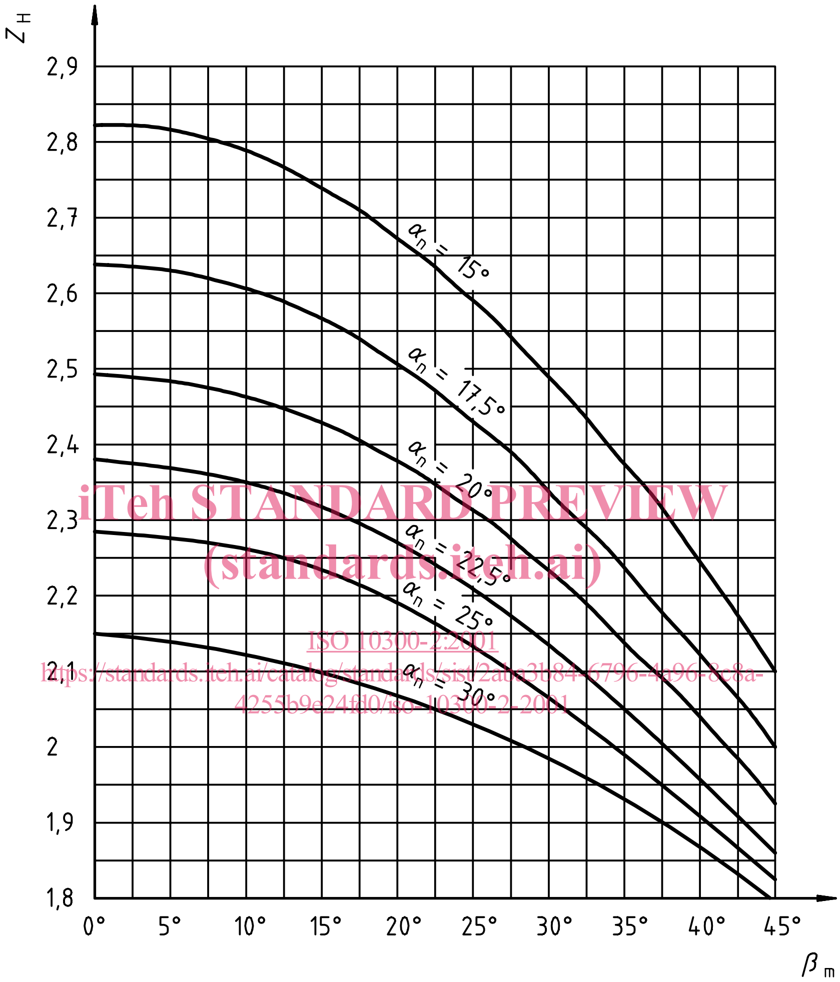

Figure 1 — Zone factor, Z_H, for x-zero bevel gears

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

The mid-zone factor, $Z_{M\text{-}B}$, transforms Z_H, and thereby contact pressure at the pitch point, to that at the determinant point of load application.

$$Z_{M\text{-}B} = \frac{\tan\alpha_{vt}}{\sqrt{\left[\sqrt{\left(\frac{d_{va1}}{d_{vb1}}\right)^2 - 1} - F_1\frac{\pi}{z_{v1}}\right]\cdot\left[\sqrt{\left(\frac{d_{va2}}{d_{vb2}}\right)^2 - 1} - F_2\frac{\pi}{z_{v2}}\right]}} \tag{7}$$

The auxiliary factors F_1 and F_2 for the mid-zone factor are given in Table 2.