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ISO 10300-2

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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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Foreword

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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.

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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.

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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.*

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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 10300-1:2000, and the following abbreviated terms, apply.

Table 1 — Abbreviated terms

Abbreviation	Description
St	steel (σ_B < 800 N/mm²)
V	through-hardened steel ($\sigma_B \geqslant$ 800 N/mm²)
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

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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:

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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.

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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.

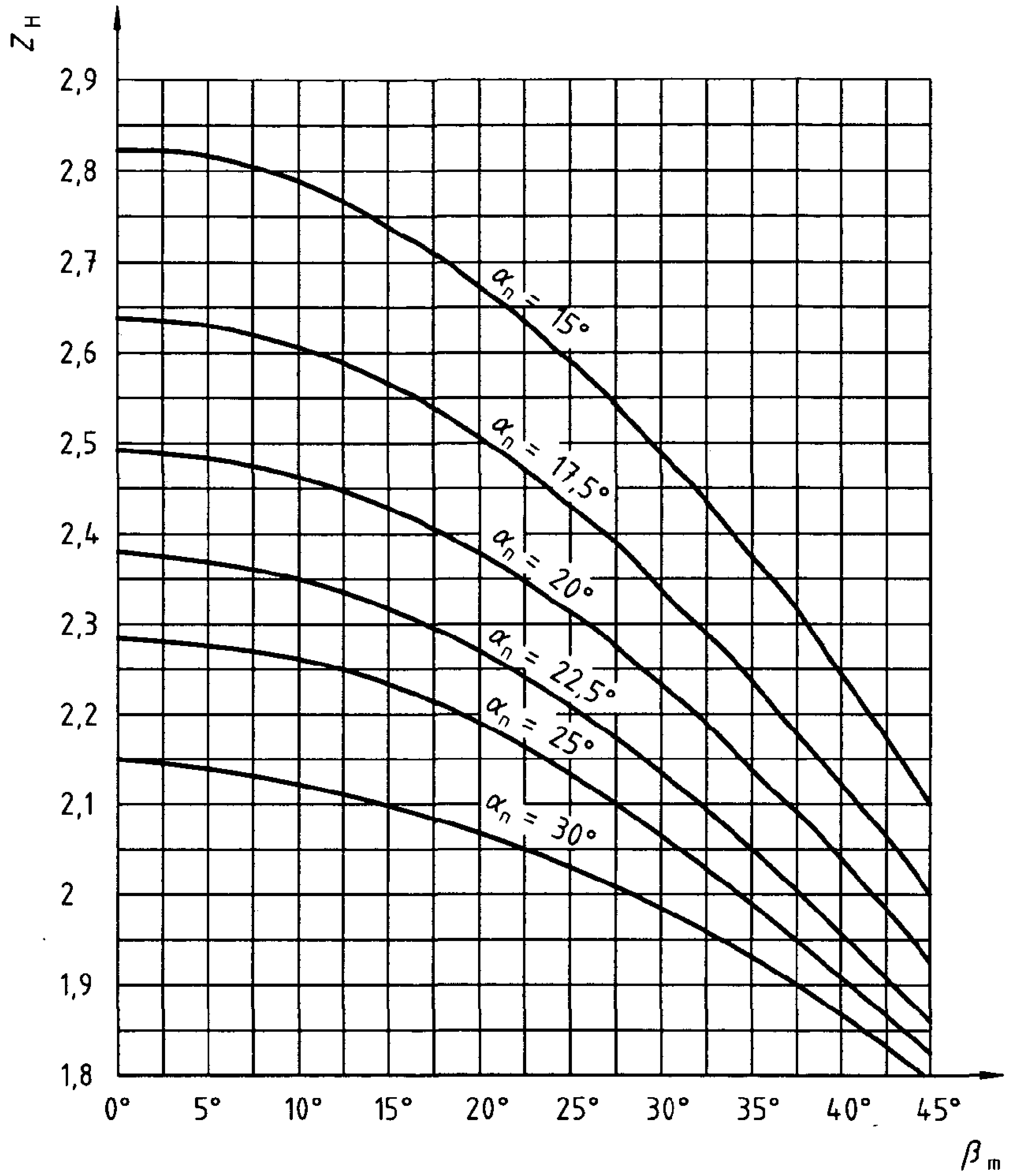

Figure 1 — Zone factor, Z_H, for x-zero bevel gears

8 Mid-zone factor, Z_{M-B}

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

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

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

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Table 2 — Factors for calculation of mid-zone factor, Z_{M-B}

	F_1	F_2
$\varepsilon_{v\beta} = 0$	2	$2\,(\varepsilon_{v\alpha} - 1)$
$0 < \varepsilon_{v\beta} < 1$	$2 + (\varepsilon_{v\alpha} - 2)\,\varepsilon_{v\beta}$	$2\,\varepsilon_{v\alpha} - 2 + (2 - \varepsilon_{v\alpha})\,\varepsilon_{v\beta}$
$\varepsilon_{v\beta} > 1$	$\varepsilon_{v\alpha}$	$\varepsilon_{v\alpha}$

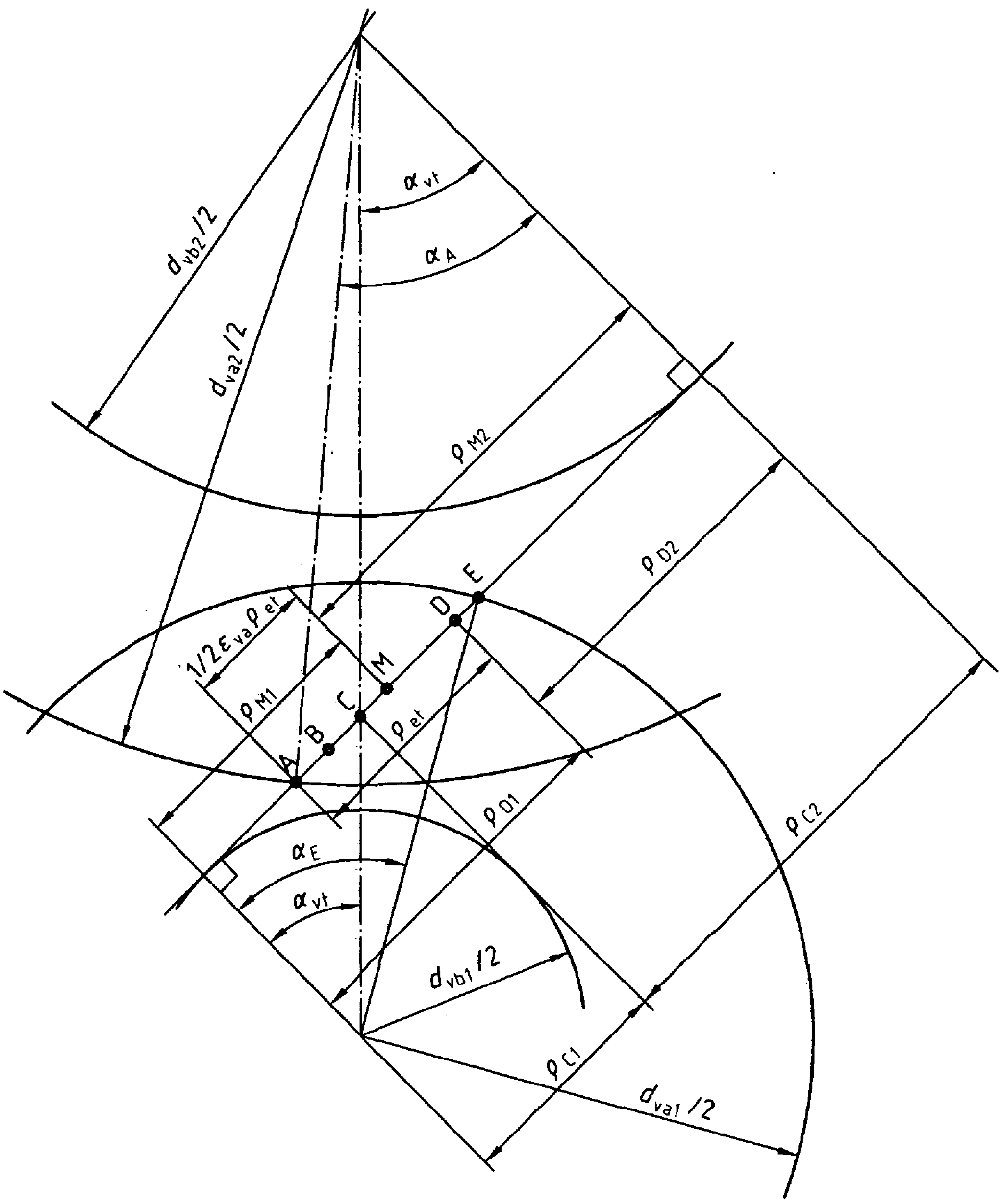

Figure 2 — Radii of curvature at mid-point M and single-pair mesh point B of the pinion, for determination of the mid-zone factor, Z_{M-B} [see Equation (7)]

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9 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.

$$Z_E = \sqrt{\frac{1}{\pi\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}} \tag{8}$$

For $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = \nu$, the following applies:

$$Z_E = \sqrt{\frac{E}{2\pi\left(1-\nu^2\right)}} \tag{9}$$

For steel and light metal $\nu = 0{,}3$, and thus:

$$Z_E = \sqrt{0{,}175\,E} \tag{10}$$

When a pair of gears is made from materials having the moduli of elasticity, E_1 and E_2, E may be determined:

$$E = \frac{2\,E_1\,E_2}{E_1 + E_2} \tag{11}$$

For a steel on steel gear pair: $Z_E = 189{,}8$

For Z_E of some other gear pair materials, see 5, ISO 6336-2:1996.

10 Load-sharing factor, Z_{LS}

The load sharing factor, Z_{LS}, accounts for load sharing between two or more pairs of teeth:

$$Z_{LS} = 1 \qquad \text{for } \varepsilon_{v\gamma} \leqslant 2 \tag{12}$$

$$Z_{LS} = \left\{1 + 2\left[1 - \left(\frac{2}{\varepsilon_{v\gamma}}\right)^{1.5}\right]\sqrt{1 - \frac{4}{\varepsilon_{v\gamma}^2}}\right\}^{-0.5} \qquad \text{for } \varepsilon_{v\gamma} > 2 \text{ and } \varepsilon_{v\beta} > 1 \tag{13}$$

For other cases, such as $\varepsilon_{v\gamma} > 2$ and $\varepsilon_{v\beta} < 1$, and explanations, see annex A.

11 Spiral-angle factor, Z_β

Independent of the influence of the spiral angle on the contact line length, the spiral-angle factor, Z_β, accounts for the influence of the spiral angle on the surface durability in respect of pitting, whereby influences such as load distribution along the contact lines are taken into consideration.

Z_β is only a function of the spiral angle β_m. The following empirical relation corresponds sufficiently well to tests and practical experiences for all practical applications:

$$Z_\beta = \sqrt{\cos\beta_m} \tag{14}$$

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12 Bevel gear factor, Z_K

The factor Z_K is an empirical factor which accounts for the difference between bevel- and cylindrical-gear loading in such a way as to agree with practical experience. It is a stress adjustment constant which permits the rating of bevel, spur and helical gears, with the same allowable contact stress numbers for any material. The following may be used in the absence of more specific knowledge:

$$Z_K = 0{,}8 \tag{15}$$

13 Size factor, Z_X

By means of Z_X, account is taken of statistical evidence indicating that the stress levels at which fatigue damage occurs decrease with an increase in component size (larger number of weak points in structure), as a consequence of the influence on subsurface defects of the resultant smaller stress gradients (theoretical stress analysis) and of size on material quality (effect on forging process, variations in structure, etc.). The main influence parameters related to the size factor are:

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

b) heat treatment, depth of hardening, distribution of hardening;

c) radius of flank curvature;

d) 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.

For the purpose of this part of ISO 10300, the size factor is equal to one ($Z_X = 1$).

14 Lubricant-film influence factors, Z_L, Z_v, Z_R

14.1 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). Figures 3 to 5 show the range of these three influence factors. In addition, the scattering (spread of values) indicates that there are other factors besides the three, not accounted for in the assumptions.

NOTE For further general remarks about these three factors, see clause 10, ISO 6336-2:1996.

14.2 Restrictions

When there is no comprehensive experience or test results (method A), Z_L, Z_v and Z_R shall be determined separately according to method B (14.3). However, in many cases, in fact for most industrial gears, the shorter method, C (14.4), may be used. When a gear pair consists of one member of hard and another of soft material, Z_L, Z_v and Z_R shall be determined for the softer of the materials.

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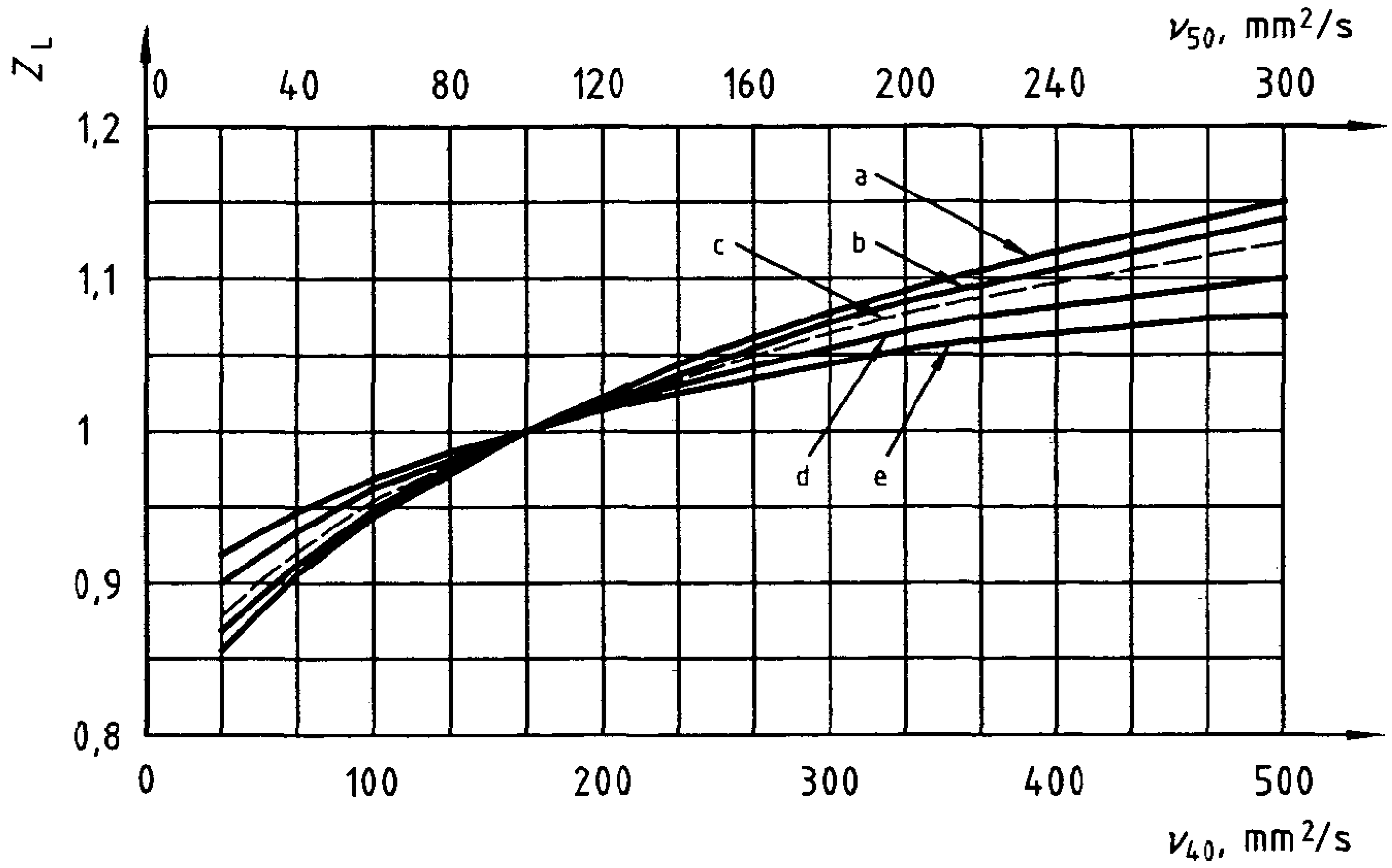


a $\sigma_{Hlim} \leqslant 850$ N/mm²

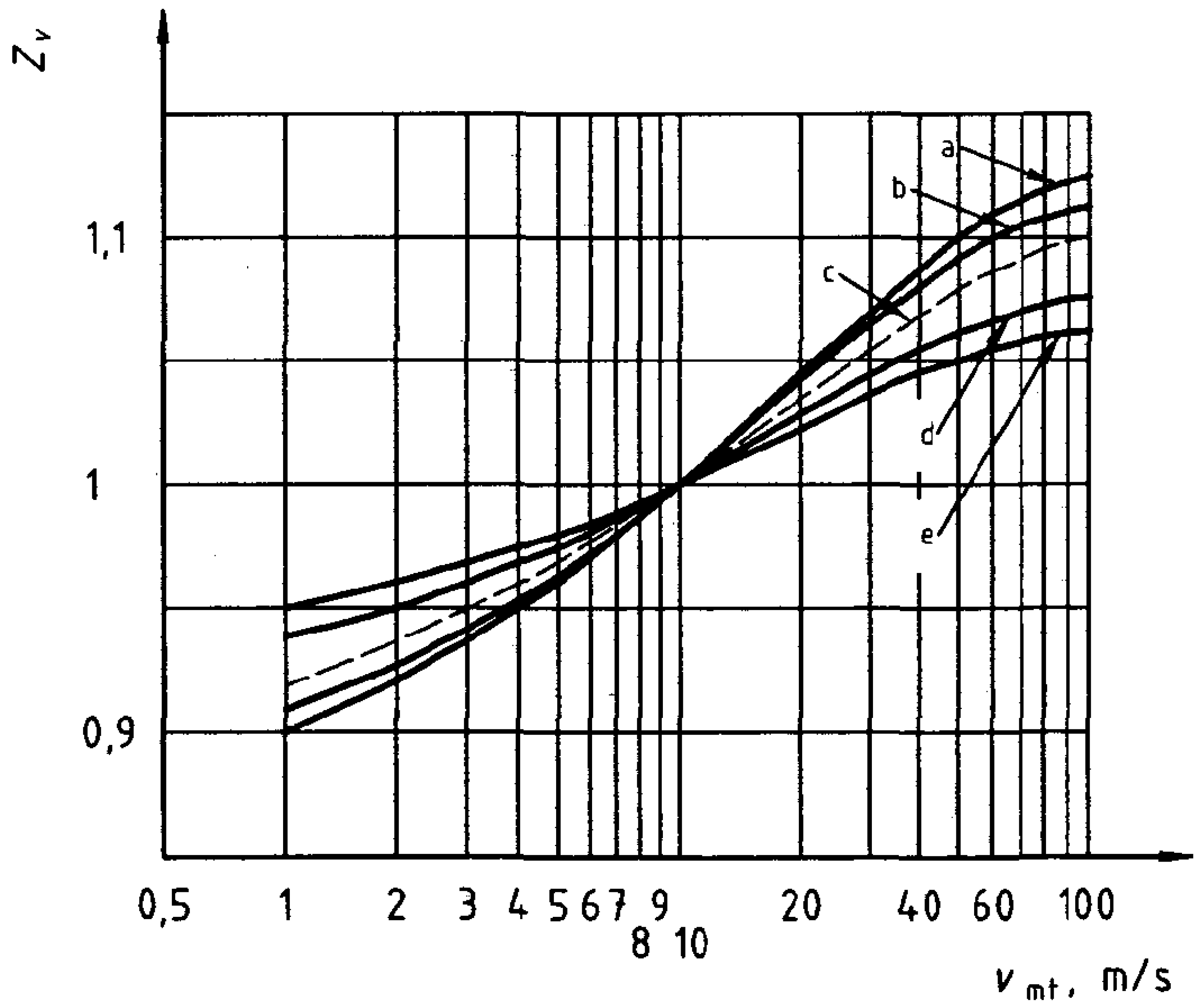
b $\sigma_{Hlim} = 900$ N/mm²

c $\sigma_{Hlim} = 1\,000$ N/mm²

d $\sigma_{Hlim} = 1\,100$ N/mm²

e $\sigma_{Hlim} \geqslant 1\,200$ N/mm²

Figure 3 — Lubricant factor, Z_L



a $\sigma_{Hlim} \leqslant 850$ N/mm²

b $\sigma_{Hlim} = 900$ N/mm²

c $\sigma_{Hlim} = 1\,000$ N/mm²

d $\sigma_{Hlim} = 1\,100$ N/mm²

e $\sigma_{Hlim} \geqslant 1\,200$ N/mm²

Figure 4 — Speed factor, Z_v

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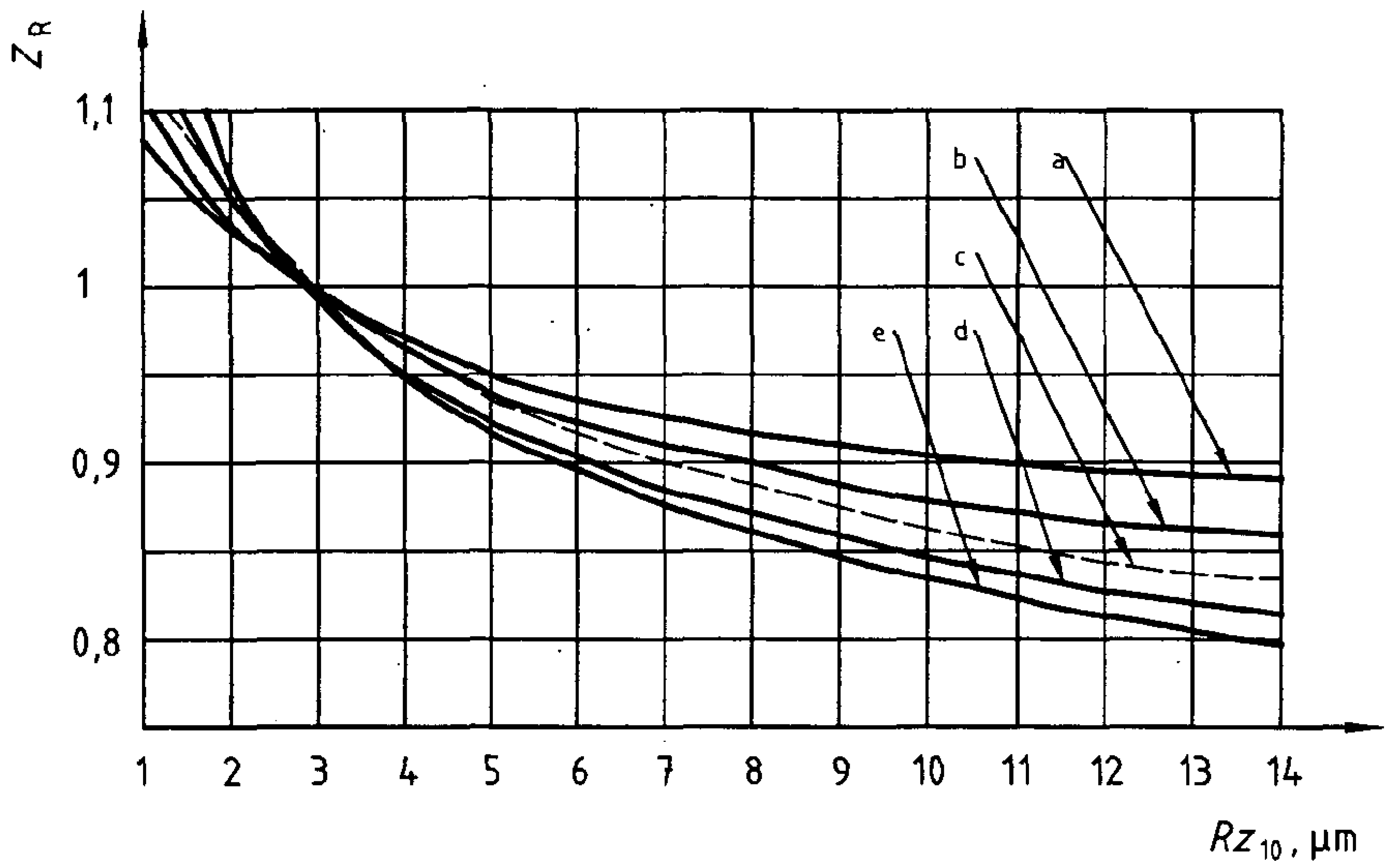

a $\sigma_{Hlim} \geqslant 1\,200$ N/mm²

b $\sigma_{Hlim} = 1\,100$ N/mm²

c $\sigma_{Hlim} = 1\,000$ N/mm²

d $\sigma_{Hlim} = 900$ N/mm²

e $\sigma_{Hlim} \leqslant 850$ N/mm²

Figure 5 — Roughness factor, Z_R

14.3 Method B

14.3.1 Lubricant factor, Z_L

Taking into account the restrictions given in 14.2, the indicated lubricant factor, Z_L, accounts for the influence of the type of lubricant, and its viscosity, on the surface durability (pitting). In Figure 3, the curves of the lubricant factor, Z_L, are plotted for mineral oils (with or without EP-additives) as a function of the nominal viscosity and the value σ_{Hlim} of the softer gear of the mating pair. In case of certain synthetic oils, with lower coefficient of friction, larger values of Z_L than those calculated for mineral oils may be used.

NOTE This part of ISO 10300 does not include a recommendation as to the choice of oil viscosity, which will need to be made with reference to testing, experience or gear-lubrication publications.

Z_L may be calculated using Equations (16) and (17), which represent the course of the curves in Figure 3:

$$Z_L = C_{ZL} + \frac{4\left(1{,}0 - C_{ZL}\right)}{\left(1{,}2 + \dfrac{134}{\nu_{40}}\right)^2} \qquad (16)$$

For the range of $\sigma_{Hlim} = 850$ N/mm² to $\sigma_{Hlim} = 1200$ N/mm², the following applies:

$$C_{ZL} = 0{,}08\,\frac{\sigma_{Hlim} - 850}{350} + 0{,}83 \qquad (17)$$

For σ_{Hlim} values below 850 N/mm², the Z_L value for $\sigma_{Hlim} = 850$ N/mm² is used, while for σ_{Hlim} values above 1200 N/mm², the Z_L value for $\sigma_{Hlim} = 1200$ N/mm² is used.

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14.3.2 Speed factor, Z_v

Taking into account the restrictions given in 14.2, the indicated speed factor, Z_v, accounts for the influence of the tangential speed on the surface durability (pitting). In Figure 4, the curves of the speed factor are plotted as a function of the tangential speed and the value σ_{Hlim} of the softer gear of the mating pair. Z_v may be calculated using Equations (18) and (19), which represent the course of the curves in Figure 4.

$$Z_v = C_{ZV} + \frac{2\left(1{,}0 - C_{ZV}\right)}{\sqrt{0{,}8 + \dfrac{32}{v_{\mathrm{mt}}}}} \tag{18}$$

For the range of $\sigma_{\mathrm{Hlim}} = 850\ \mathrm{N/mm^2}$ to $\sigma_{\mathrm{Hlim}} = 1200\ \mathrm{N/mm^2}$ the following applies:

$$C_{ZV} = 0{,}08\,\frac{\sigma_{\mathrm{Hlim}} - 850}{350} + 0{,}85 \tag{19}$$

For σ_{Hlim} values below 850 N/mm², the Z_v value for $\sigma_{\mathrm{Hlim}} = 850\ \mathrm{N/mm^2}$ is used, while for σ_{Hlim} values above 1200 N/mm², the Z_v value for $\sigma_{\mathrm{Hlim}} = 1200\ \mathrm{N/mm^2}$ is used.

14.3.3 Roughness factor, Z_R

Taking into account the restrictions given in 14.2, the indicated roughness factor Z_R accounts for the influence of the surface condition of the tooth flanks on the surface durability (pitting). In Figure 5, the curves of the roughness factor are plotted as a function of Rz_{10} and the value σ_{Hlim} of the softer gear of the mating pair. The figure is valid for a gear pair with a virtual radius of curvature at the pitch point of $\rho_{\mathrm{red}} = 10$ mm.

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-rolling movement shall be decisive.

The mean relative roughness is[1]:

$$Rz_{10} = \frac{Rz_1 + Rz_2}{2} \cdot \sqrt[3]{\frac{10}{\rho_{\mathrm{red}}}} \tag{20}$$

with the radius of relative curvature:

$$\rho_{\mathrm{red}} = \frac{a_{\mathrm{v}} \sin\alpha_{\mathrm{vt}}}{\cos\beta_{\mathrm{vb}}} \cdot \frac{u_{\mathrm{v}}}{\left(1 + u_{\mathrm{v}}\right)^2} \tag{21}$$

The factor Z_R may be calculated using Equations (22) and (23), which represent the course of the curves in Figure 5.

$$Z_R = \left(\frac{3}{Rz_{10}}\right)^{C_{ZR}} \tag{22}$$

1) When the roughness is given as an Ra value (= CLA value) (= AA value), the following approximation can be used:

$$Ra = \mathrm{CLA} = \mathrm{AA} = \frac{Rz}{6}$$

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In the range of 850 N/mm² $\leqslant \sigma_{\mathrm{Hlim}} \leqslant$ 1200 N/mm² the following applies:

$$C_{\mathrm{ZR}} = 0{,}12 + \frac{1\,000 - \sigma_{\mathrm{Hlim}}}{5\,000} \tag{23}$$

For σ_{Hlim} values below 850 N/mm², use σ_{Hlim} = 850 N/mm², while for σ_{Hlim} values above 1200 N/mm² use σ_{Hlim} = 1200 N/mm².

14.4 Method C (product of Z_{L}, Z_v and Z_R)

It is hereby assumed that a lubricant viscosity has been chosen and which is suited to 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, milled gear pairs: 0,85.

For gear pairs lapped after milling: 0,92.

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

— $Rz_{10} \leqslant 4$ µm: $Z_{\mathrm{L}}\, Z_v\, Z_R = 1{,}0$;

— $Rz_{10} > 4$ µm: $Z_{\mathrm{L}}\, Z_v\, Z_R = 0{,}92$.

If the above conditions do not apply, Z_{L}, Z_v and Z_R shall be determined separately according to Method B.

15 Work-hardening factor, Z_{W}

15.1 General

The work-hardening factor, Z_{W}, accounts for the increase in the surface durability caused by the meshing of a structural- or through-hardened-steel wheel with a surface-hardened pinion having smooth tooth flanks ($Rz \leqslant 6$ µm).

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

15.2 Method B

The data provided here are based on tests on different materials using standard reference test gears as well on field experience with production gears. The extent of scatter (spread of values) indicates the existence of other influences not included in the calculation process. Although the curve in Figure 6 was carefully chosen, it shall not be interpreted as absolute. It is, like Equation (24), empirical. The value of Z_{W} is taken as the same for endurance, limited life, and static stress.

Z_{W} can be taken from Figure 6, for the conditions listed in this clause, as a function of the flank hardness of the softer bevel gear.

For method B, Z_{W} shall be calculated using Equation (25), which is consistent with the curve in Figure 6:

$$Z_{\mathrm{W}} = 1{,}2 - \frac{\mathrm{HB} - 130}{1700} \tag{24}$$

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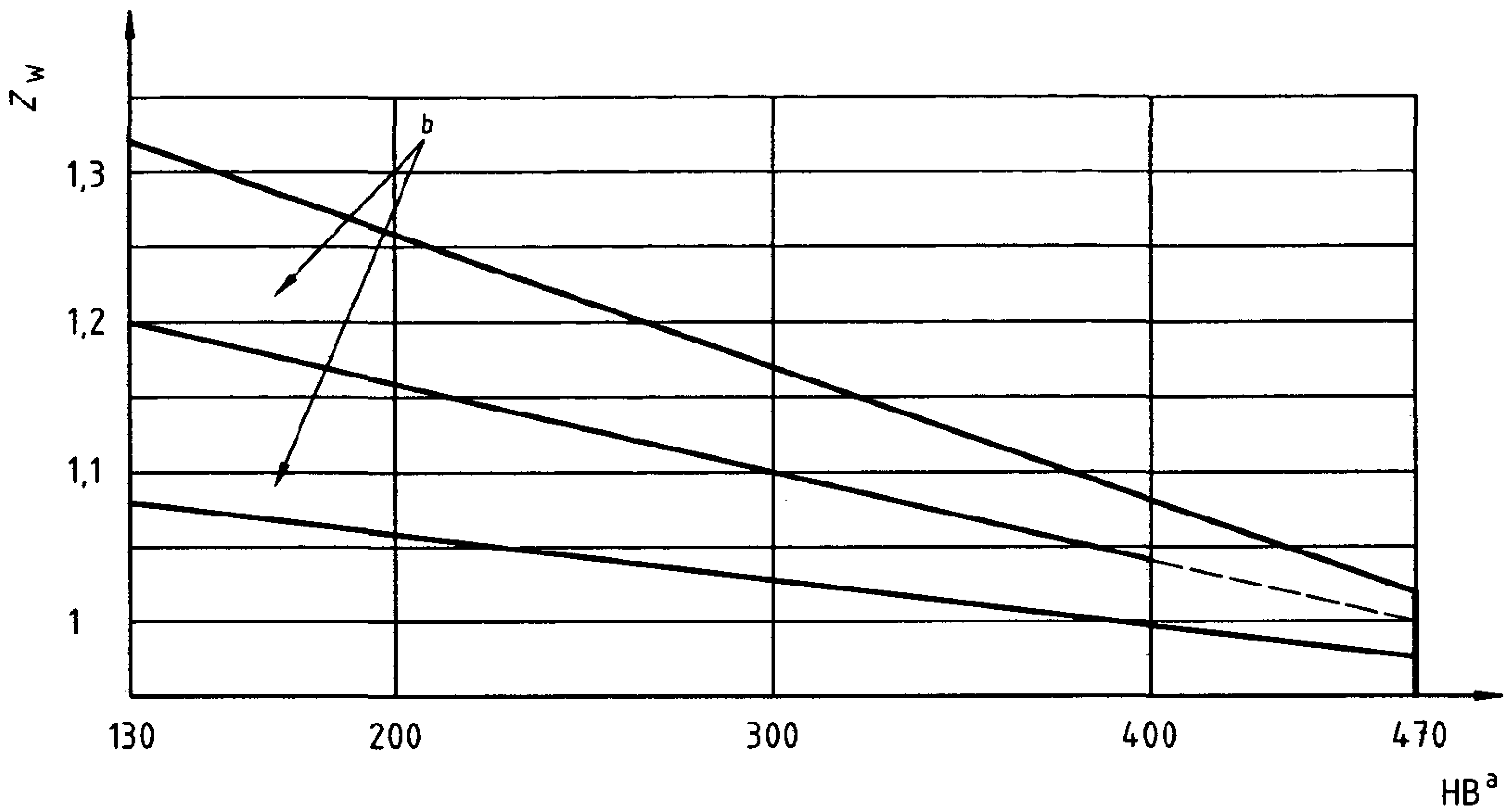
where

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

Z_W is 1,2 for HB < 130, and 1,0 for HB > 470;

Z_W is 1,0 if pinion and gear have the same hardness.



a Tooth flank hardness of the softer wheel

b Range of scatter

Figure 6 — Work-hardening factor, Z_W

16 Life factor, Z_{NT}

16.1 General

The life factor, Z_{NT}, accounts for the higher contact stress, including static stress, which may be acceptable for a limited life (number of load cycles), as compared with the allowable stress at 5×10^7 cycles (the point, or "knee", on the curves of Figure 7 where $Z_{NT} = 1{,}0$). Z_{NT} has been determined for standard test-gear conditions.

The main influences related to Z_{NT} are:

a) material and heat treatment (see 5.2, ISO 6336-5:1996);

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

c) lubrication regime;

d) failure criteria;

e) required smoothness of operation;

f) pitchline velocity;

g) gear material cleanliness;

h) material ductility and fracture toughness;

i) residual stress.

For the purposes of ISO 10300, the number of load cycles, N_L, is identified as the number of mesh contacts, under load, of the gear tooth being analysed.

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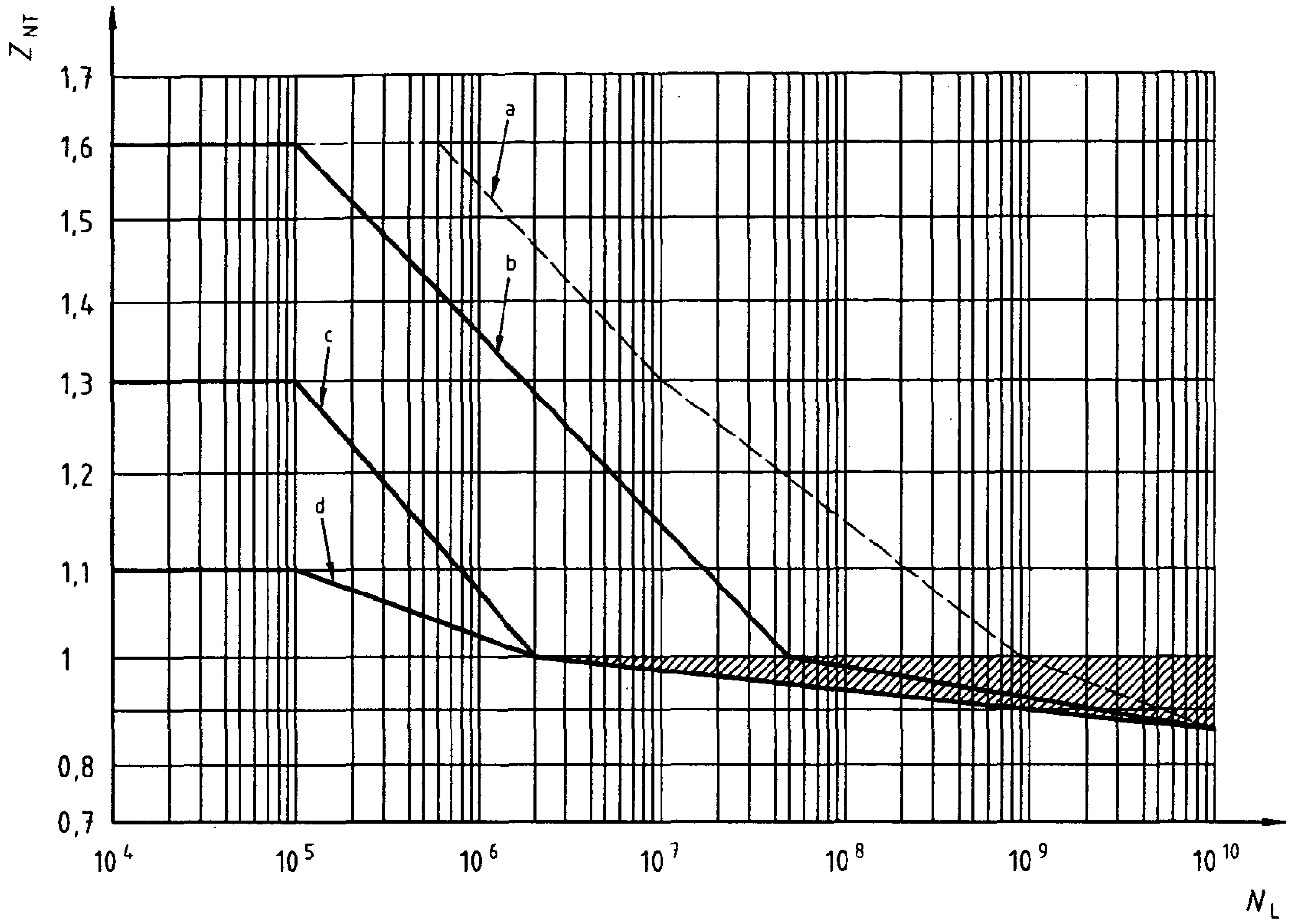

[a] St, V, GGG (perl. bain.), GTS (perl.), Eh, IF, when limited pitting permitted.

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

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

[d] NV (nitrocar.)

Figure 7 — Life factor for pitting resistance, Z_{NT} (for standard-reference test gears)

16.2 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, pitchline 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.

16.3 Method B

The permissible stress at limited service life, or the safety factor in the limited life stress range, shall be determined using the life factor, Z_{NT}, for standard reference test gear (see 5.2, ISO 6336-5:1996). The factors Z_L, Z_R, Z_v and Z_W are not included. However, the modified effect of these factors on limited life shall be considered. Z_{NT} for static and endurance stresses may be taken from Figure 7 or Table 3, and for limited life stress by interpolation between the values for the endurance, and static, stresses (see 5.2, ISO 6336-5:1996).

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Table 3 — Life factor, Z_{NT}, for static- and endurance-stress limits

Material [a]	Number of load cycles	Life factor, Z_{NT}
St, V [b], GGG (perl. bain.) [b]. GTS (perl.), Eh, IF [b]	$N_L = 6 \times 10^5$, static	1,6
	$N_L = 10^7$, endurance	1,3
	$N_L = 10^9$, endurance	1,0
	$N_L = 10^{10}$, endurance	0,85
St, V, GGG (perl. bain.), GTS (perl.), EH, IF	$N_L = 10^5$, static	1,6
	$N_L = 5 \times 10^7$, endurance	1,0
	$N_L = 10^{10}$, endurance	0,85
	Optimum lubrication, material, manufacturing, and experience	1,0
GG, GGG (ferr.), NT (nitr.), NV (nitr.)	$N_L = 10^5$, static	1,3
	$N_L = 2 \times 10^6$, endurance	1,0
	$N_L = 10^{10}$, endurance	0,85
	Optimum lubrication, material, manufacturing, and experience	1,0
NV (nitrocar.)	$N_L = 10^5$, static	1,1
	$N_L = 2 \times 10^6$, endurance	1,0
	$N_L = 10^{10}$, endurance	0,85
	Optimum lubrication, material, manufacturing, and experience	1,0

a For descriptions of the material abbreviations, see Table 1.

b Only if a certain degree of pitting is acceptable.

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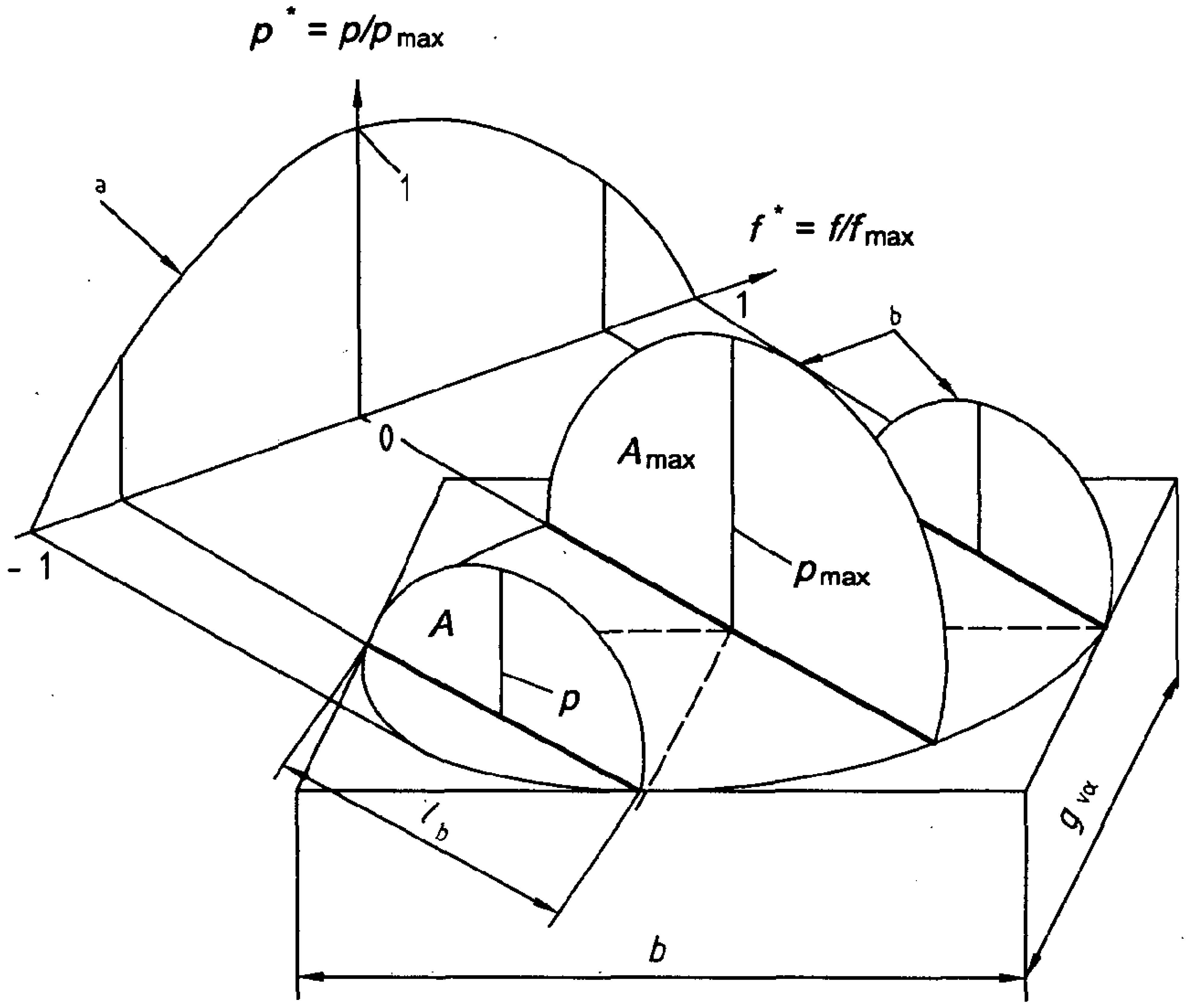
Annex A
(informative)

Load sharing factor, Z_{LS}

The load sharing factor, Z_{LS}, accounts for load sharing between two or more pairs of teeth for $\varepsilon_{v\gamma} > 2$. The load distribution along a contact line is assumed to be elliptical. The distribution of the peak loads (of contact lines) is assumed to be a parabola (exponent 1,5), as shown in Figure A.1.



a Parabolic distribution of peak loads

b Elliptical load distribution

Figure A.1 — Load distribution in the contact area

$$p^* = \frac{p}{p_{max}} = 1 - \left(\frac{|f|}{|f_{max}|} \right)^{1,5} \geqslant 0 \tag{A.1}$$

$$f_{max} = \frac{1}{2}\, \varepsilon_{v\gamma}\, p_{et} \cos\beta_{vb} \tag{A.2}$$

$$A^* = \frac{1}{2} \times \frac{1}{2} p^*\, l_b\, \pi \tag{A.3}$$

For f, $\varepsilon_{v\gamma}$, l_b, see clause A.6, ISO 10300-1:2001.

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The load sharing factor, Z_{LS}, is the area $A_m{}^*$ related to the sum of all areas.

$$Z_{LS} = \sqrt{\frac{A_m{}^*}{A_t{}^* + A_m{}^* + A_r{}^*}} \tag{A.4}$$

where

$A_t{}^*$ is the area over the tip contact line (p^*, l_b calculated with f_t according to ISO 10300-1:2001, Table A.3);

$A_m{}^*$ is the area over the middle contact line (p^*, l_b calculated with f_m according to ISO 10300-1:2001, Table A.3);

$A_r{}^*$ is the area over the root contact line (p^*, l_b calculated with f_r according to ISO 10300-1:2001, Table A.3).

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