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Calculation of load capacity of spur and helical gears —

Part 2: Calculation of surface durability (pitting)

Calcul de la capacité de charge des engrenages cylindriques à dentures droite et hélicoïdale —

Partie 2: Calcul de la résistance à la pression de contact (piqûre)

Reference number
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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 2.

The main task of technical committees is to prepare International Standards. 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 document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 6336-2 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

This second edition cancels and replaces the first edition (ISO 6336-2:1996), Clause 13 of which has been technically revised. It also incorporates the Technical Corrigenda ISO 6336-2:1996/Cor.1:1998 and ISO 6336-2:1996/Cor.2:1999.

ISO 6336 consists of the following parts, under the general title *Calculation of load capacity of spur and helical gears*:

— *Part 1: Basic principles, introduction and general influence factors*

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

— *Part 3: Calculation of tooth bending strength*

— *Part 5: Strength and quality of materials*

— *Part 6: Calculation of service life under variable load*

This corrected version incorporates the following corrections:

— the key to Figure 2 has been inverted, so that the descriptions of the axes now correspond correctly with the figure;

— in Figure 7, the description of the Y axis in the key has been given in English;

— Equation (46) has been corrected;

— the wording of 12.3.1.3.2 has been changed such that it now refers to roughness.

Introduction

Hertzian pressure, which serves as a basis for the calculation of contact stress, is the basic principle used in this part of ISO 6336 for the assessment of the surface durability of cylindrical gears. It is a significant indicator of the stress generated during tooth flank engagement. However, it is not the sole cause of pitting, and nor are the corresponding subsurface shear stresses. There are other contributory influences, for example, coefficient of friction, direction and magnitude of sliding and the influence of lubricant on distribution of pressure. Development has not yet advanced to the stage of directly including these in calculations of load-bearing capacity; however, allowance is made for them to some degree in the derating factors and choice of material property values.

In spite of shortcomings, Hertzian pressure is useful as a working hypothesis. This is attributable to the fact that, for a given material, limiting values of Hertzian pressure are preferably derived from fatigue tests on gear specimens; thus, additional relevant influences are included in the values. Therefore, if the reference datum is located in the application range, Hertzian pressure is acceptable as a design basis for extrapolating from experimental data to values for gears of different dimensions.

Several methods have been approved for the calculation of the permissible contact stress and the determination of a number of factors (see ISO 6336-1).

Calculation of load capacity of spur and helical gears —

Part 2:
Calculation of surface durability (pitting)

IMPORTANT — The user of this part of ISO 6336 is cautioned that when the method specified is used for large helix angles and large pressure angles, the calculated results should be confirmed by experience as by Method A. In addition, it is important to note that best correlation has been obtained for helical gears when high accuracy and optimum modifications are employed.

1 Scope

This part of ISO 6336 specifies the fundamental formulæ for use in the determination of the surface load capacity of cylindrical gears with involute external or internal teeth. It includes formulæ for all influences on surface durability for which quantitative assessments can be made. It applies primarily to oil-lubricated transmissions, but can also be used to obtain approximate values for (slow-running) grease-lubricated transmissions, as long as sufficient lubricant is present in the mesh at all times.

The given formulæ are valid for cylindrical gears with tooth profiles in accordance with the basic rack standardized in ISO 53. They may also be used for teeth conjugate to other basic racks where the actual transverse contact ratio is less than $\varepsilon_{\alpha n} = 2{,}5$. The results are in good agreement with other methods for the range, as indicated in the scope of ISO 6336-1.

These formulæ cannot be directly applied for the assessment of types of gear tooth surface damage such as plastic yielding, scratching, scuffing or any other than that described in Clause 4.

The load capacity determined by way of the permissible contact stress is called the “surface load capacity” or “surface durability”.

2 Normative references

The following referenced documents are indispensable for the application 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 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 6336-1:2006, *Calculation of load capacity of spur and helical gears — Part 1: Basic principles, introduction and general influence factors*

ISO 6336-5:2003, *Calculation of load capacity of spur and helical gears — Part 5: Strength and quality of material*

3 Terms, definitions, symbols and abbreviated terms

For the purposes of this document, the terms, definitions, symbols and abbreviated terms given in ISO 1122-1 and ISO 6336-1 apply.

4 Pitting damage and safety factors

If limits of the surface durability of the meshing flanks are exceeded, particles will break out of the flanks, leaving pits.

The extent to which such pits can be tolerated (in size and number) varies within wide limits, depending largely on the field of application. In some fields, extensive pitting can be accepted; in other fields any appreciable pitting is to be avoided.

The following assessments, relevant to average working conditions, will help in distinguishing between initial pitting and destructive pitting.

Linear or progressive increase of the total area of pits is unacceptable; however, the effective tooth bearing area can be enlarged by initial pitting, and the rate of generation of pits could subsequently reduce (degressive pitting), or cease (arrested pitting). Such pitting is considered tolerable. In the event of dispute, the following rule is determinant.

Pitting involving the formation of pits that increase linearly or progressively with time under unchanged service conditions (linear or progressive pitting) is not acceptable. 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. It is a frequent occurrence that pits are 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 can be based on considerations of the entire quantity of wear debris. In critical cases, the condition of the flanks should be examined at least three times. The first examination should, however, only take place after at least 10^6 cycles of load. Further examination should take place after a period of service depending on the results of the previous examination.

If the deterioration by pitting is such that it puts human life in danger, or there is a risk that it could lead to some grave consequences, then pitting is not tolerable. Due to stress concentration effects, a pit of a diameter of 1 mm near the fillet of a through-hardened or case-hardened tooth of a gear can become the origin of a crack which could lead to tooth breakage; for this reason, such a pit shall be considered as intolerable (e.g. in aerospace transmissions).

Similar considerations are true for turbine gears. In general, during the long life (10^{10} to 10^{11} cycles) which is demanded of these gears, neither pitting nor unduly severe wear is tolerable. Such damage could lead to unacceptable vibrations and excessive dynamic loads. Appropriately generous safety factors should be included in the calculation, i.e. only a low probability of failure can be tolerated.

In contrast, pitting over 100 % of the working flanks can be tolerated for some slow-speed industrial gears with large teeth (e.g. module 25) made from low hardness steel where they will safely transmit the rated power for 10 to 20 years. Individual pits may be up to 20 mm in diameter and 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. A high factor of safety against tooth breakage is necessary.

Comments on the choice of safety factor S_H can be found in ISO 6336-1:2006, 4.1.7. It is recommended that the manufacturer and customer agree on the values of the minimum safety factor.

5 Basic formulæ

5.1 General

The calculation of surface durability is based on the contact stress, σ_H, at the pitch point or at the inner point of single pair tooth contact. The higher of the two values obtained is used to determine the load capacity (determinant). σ_H and the permissible contact stress, σ_{HP}, shall be calculated separately for wheel and pinion. σ_H shall be less than σ_{HP}. This comparison will be expressed in safety factors S_{H1} and S_{H2} which shall be higher than the agreed minimum safety factor S_{Hmin}. Four categories are recognized in the calculation of σ_H, as follows.

a) Spur gears with contact ratio $\varepsilon_\alpha \geqslant 1$:

— for a pinion, σ_H is usually calculated at the inner point of single pair tooth contact. In special cases, σ_H at the pitch point is greater and thus determinant;

— for a spur wheel, in the case of external teeth, σ_H is usually calculated at the pitch point, however, in special cases — particularly in the case of small transmission ratios (see 6.2), — σ_H is greater at the inner point of single pair tooth contact of the wheel and is thus determinant; whereas, for internal teeth, σ_H is always calculated at the pitch point.

b) Helical gears with contact ratio $\varepsilon_\alpha \geqslant 1$ and overlap ratio $\varepsilon_\beta \geqslant 1$: σ_H is always calculated at the pitch point for pinion and wheel.

c) Helical gears with contact ratio $\varepsilon_\alpha \geqslant 1$ and overlap ratio $\varepsilon_\beta < 1$: σ_H is determined by linear interpolation between the two limit values, i.e. σ_H for spur gears and σ_H for helical gears with $\varepsilon_\beta = 1$ in which the determination of σ_H for each is to be based on the numbers of teeth on the actual gears.

d) Helical gears with $\varepsilon_\alpha \leqslant 1$ and with $\varepsilon_\gamma > 1$: not covered by ISO 6336 — a careful analysis of the contact stress along the path of contact is necessary.

5.2 Safety factor for surface durability (against pitting), S_H

Calculate S_H separately for pinion and wheel:

$$S_{H1} = \frac{\sigma_{HG1}}{\sigma_{H1}} > S_{H\min} \tag{1}$$

$$S_{H2} = \frac{\sigma_{HG2}}{\sigma_{H2}} > S_{H\min} \tag{2}$$

Take $\sigma_{H1,2}$ in accordance with Equation (4) for the pinion and in accordance with Equation (5) for the wheel (see 5.1). Calculate σ_{HG} for long life and static stress limits in accordance with Equation (6) and 5.4.2 a) and b). For limited life, calculate σ_{HG} in accordance with Equation (6) and 5.4.3.

NOTE This is the calculated safety factor with regard to contact stress (Hertzian pressure). The corresponding factor relative to torque capacity is equal to the square of S_H.

For notes on minimum safety factor and probability of failure, see Clause 4 and ISO 6336-1:2006, 4.1.7.

5.3 Contact stress, σ_H

The total tangential load in the case of gear trains with multiple transmission paths, planetary gear systems or split-path gear trains is not quite evenly distributed over the individual meshes (depending on design, tangential speed and manufacturing accuracy). This is to be taken into consideration by inserting the mesh load factor K_γ to follow K_A in Equations (4) and (5), and to adjust the average tangential load per mesh as necessary.

$$\sigma_{H0} = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b} \frac{u+1}{u}} \tag{3}$$

$$\sigma_{H1} = Z_B \sigma_{H0} \sqrt{K_A K_v K_{H\beta} K_{H\alpha}} \tag{4}$$

$$\sigma_{H2} = Z_D \sigma_{H0} \sqrt{K_A K_v K_{H\beta} K_{H\alpha}} \tag{5}$$

where

σ_{H0} is the nominal contact stress at the pitch point, which is the stress induced in flawless (error-free) gearing by application of static nominal torque;

Z_B is the pinion single pair tooth contact factor of the pinion (see 6.2 and 6.3), which converts contact stress at the pitch point to the contact stress at the inner point of single pair tooth contact on the pinion;

Z_D is the single pair tooth contact factor of the wheel (see 6.2), which converts contact stress at the pitch point to contact stress at the inner point of single pair tooth contact of the wheel;

K_A is the application factor (see ISO 6336-6), which takes into account the load increment due to externally influenced variations of input or output torque;

K_v is the dynamic factor (see ISO 6336-1), which takes into account load increments due to internal dynamic effects;

$K_{H\beta}$ is the face load factor for contact stress (see ISO 6336-1), which takes into account uneven distribution of load over the facewidth, due to mesh misalignment caused by inaccuracies in manufacture, elastic deformations, etc.;

$K_{H\alpha}$ is the transverse load factor for contact stress (see ISO 6336-1), which takes into account uneven load distribution in the transverse direction resulting, for example, from pitch deviation;[1)]

σ_{HP} is the permissible contact stress (see 5.3);

Z_H is the zone factor (see Clause 6), which takes into account the flank curvatures at the pitch point and transforms tangential load at the reference cylinder to tangential load at the pitch cylinder;

Z_E is the elasticity factor (see Clause 7), which takes into account specific properties of the material, moduli of elasticity E_1, E_2 and Poisson's ratios ν_1, ν_2;

Z_ε is the contact ratio factor (see Clause 8), which takes into account the influence of the effective length of the lines of contact;

Z_β is the helix angle factor (see Clause 9), which takes into account influences of the helix angle, such as the variation of the load along the lines of contact;

F_t is the nominal tangential load, the transverse load tangential to the reference cylinder (see related requirement, below);

b is the facewidth (for a double helix gear $b = 2\, b_B$) (see related requirement, below);

---

1) See ISO 6336-1:2006, 4.1.14, for the sequence in which factors K_A, K_v, $K_{H\beta}$, $K_{H\alpha}$ are calculated.

d_1 is the reference diameter of pinion;

u is the gear ratio $= z_2/z_1$. For external gears u is positive, and for internal gears u is negative.

The total tangential load per mesh shall be introduced for F_t in every case (even with $\varepsilon_{\alpha n} > 2$). See ISO 6336-1:2006, 4.2, for the definition of F_t and comments on particular characteristics of double-helical gearing. The value b of mating gears is the smaller of the facewidths at the root circles of pinion and wheel ignoring any intentional transverse chamfers or tooth-end rounding. Neither unhardened portions of surface-hardened gear tooth flanks nor the transition zones shall be included.

5.4 Permissible contact stress, σ_{HP}

The limit values of contact stresses (see Clause 10) should preferably be derived from material tests using meshing gears as test pieces (see Introduction). The more closely test gears and test conditions resemble the service gears and service conditions, the more relevant to the calculations the derived values will be.

5.4.1 Determination of permissible contact stress σ_{HP} — Principles, assumptions and application

Several procedures for the determination of permissible contact stresses are acceptable. The method adopted shall be validated by carrying out careful comparative studies of well-documented service histories of a number of gears.

5.4.1.1 Method A

In Method A the permissible contact stress σ_{HP} (or the pitting stress limit, σ_{HG}) for reference stress, long and limited life and static stresses is calculated using Equation (4) or (5) from the S-N curve or damage curve derived from tests of actual gear pair duplicates under appropriate service conditions.

The cost required for this method is in general only justifiable for the development of new products, failure of which would have serious consequences (e.g. for manned space flight).

Similarly, the permissible stress values may be derived from consideration of dimensions, service conditions and performance of carefully monitored reference gears. The more closely the dimensions and service conditions of the actual gears resemble those of the reference gears, the more effective will be the application of such values for purposes of design ratings or calculation checks.

5.4.1.2 Method B

Damage curves, characterized by the allowable stress number values, $\sigma_{H\,lim}$, and the limited life factors, Z_{NT}, have been determined for a number of common gear materials and heat treatments from the results of gear loading tests with standard reference test gears.

These test gear values are converted to suit the dimensions and service conditions of the actual gear pair using the (relative) influence factors for lubricant Z_L, pitch line velocity Z_v, flank surface roughness Z_R, work hardening Z_W and size Z_X.

Method B is recommended for reasonably accurate calculation whenever pitting resistance values are available from gear tests, from special tests or, if the material is similar, from ISO 6336-5 (see Introduction).

5.4.1.3 Method B_R

Material characteristic values are determined by rolling pairs of disks in loaded contact. The magnitude and direction of the sliding speed in these tests should be adjusted to represent the in-service slide and roll conditions of the tooth flanks in the areas at risk from pitting.

Method B_R may be used when stress values derived from gear tests are not available. The method is particularly suitable for the determination of the surface durability of various materials relative to one another.

5.4.2 Permissible contact stress, σ_{HP}: Method B

The permissible contact stress is calculated from

$$\sigma_{HP} = \frac{\sigma_{H\,lim} Z_{NT}}{S_{H\,min}} Z_L\, Z_v\, Z_R\, Z_W\, Z_X = \frac{\sigma_{HG}}{S_{H\min}} \tag{6}$$

where

- $\sigma_{H\,lim}$ is the allowable stress number (contact) (see Clause 10 and ISO 6336-5), which accounts for the influence of material, heat treatment and surface roughness of the standard reference test gears;
- Z_{NT} is the life factor for test gears for contact stress (see Clause 11), which accounts for higher load capacity for a limited number of load cycles;
- σ_{HG} is the pitting stress limit ($= \sigma_{HP}\, S_{H\,min}$);
- $S_{H\,min}$ is the minimum required safety factor for surface durability.
- Z_L, Z_R, Z_v are factors that, together, cover the influence of the oil film on tooth contact stress;
- Z_L is the lubricant factor (see Clause 12), which accounts for the influence of the lubricant viscosity;
- Z_R is the roughness factor (see Clause 12), which accounts for the influence of surface roughness;
- Z_v is the velocity factor (see Clause 12), which accounts for the influence of pitch line velocity;
- Z_W is the work hardening factor (see Clause 13), which accounts for the effect of meshing with a surface hardened or similarly hard mating gear.
- Z_X is the size factor for contact stress (see Clause 14), which accounts for the influence of the tooth dimensions for the permissible contact stress.

a) **Permissible contact stress (reference)**, $\sigma_{HP\,ref}$, is derived from Equation (6), with $Z_{NT} = 1$ and the influence factors $\sigma_{H\,lim}$, Z_L, Z_v, Z_R, Z_W, Z_R, Z_X and $S_{H\,min}$ calculated using Method B.

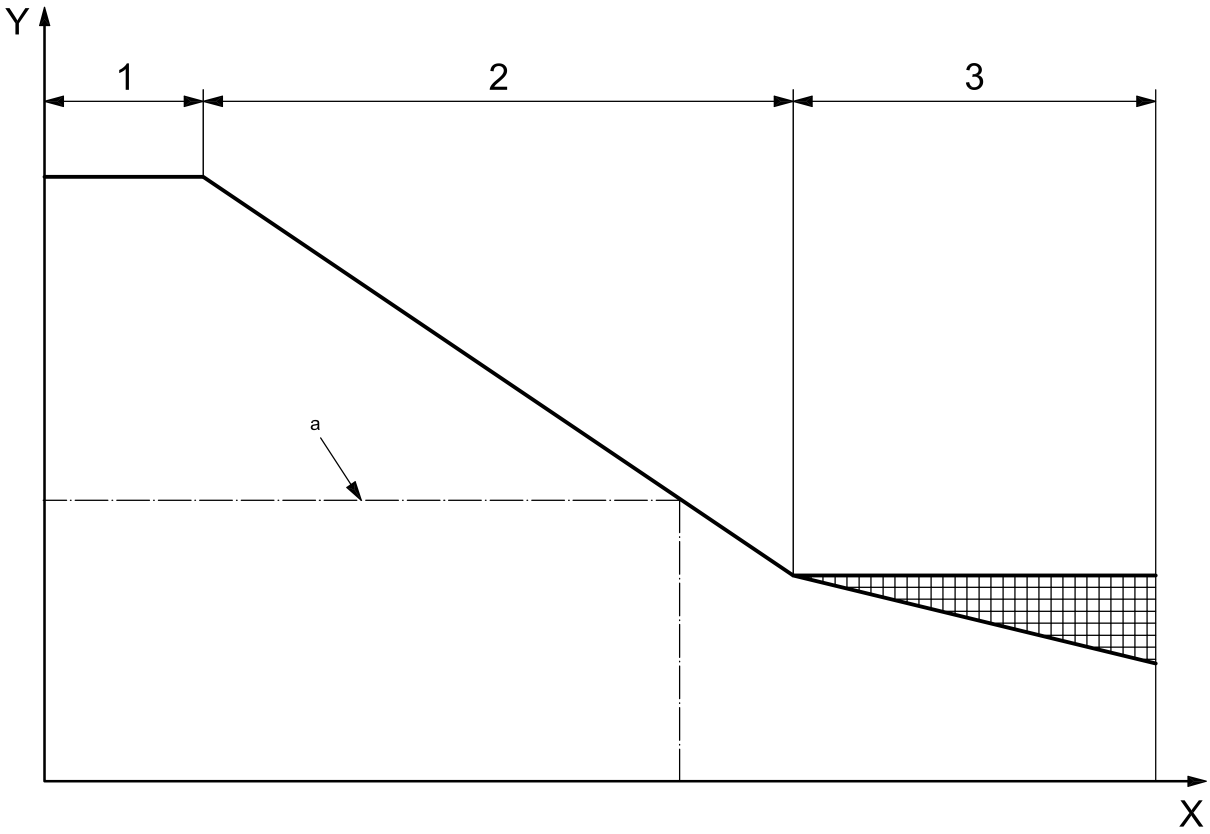
b) **Permissible contact stress (static)**, $\sigma_{HP\,stat}$, is determined in accordance with Equation (6), with all influence factors (for static stress) following Method B.

5.4.3 Permissible contact stress for limited and long life: Method B

In Method B, provision is made for determination of σ_{HP} by graphical or computed linear interpolation on a log-log scale between the value obtained for reference in accordance with 5.4.2 a) and the value obtained for static stress in accordance with 5.4.2 b). Values appropriate to the relevant number of load cycles, N_L, are indicated by the S-N curve. See Clause 11.

5.4.3.1 Graphical values

Calculate σ_{HP} for reference stress and static stress in accordance with 5.4.2 and plot the S-N curve corresponding to the life factor Z_{NT}. See Figure 1 for principle. σ_{HP} for the relevant number of load cycles N_L may be read from this graph.



Key

X number of load cycles, N_L (log)

Y permissible contact stress, σ_{HP} (log)

1 static

2 limited life

3 long life

a Example: permissible contact stress, σ_{HP} for a given number of load cycles.

Figure 1 — Graphic determination of permissible contact stress for limited life — Method B

5.4.3.2 Determination by calculation

Calculate $\sigma_{HP\ ref}$ for reference and $\sigma_{HP\ stat}$ for static strength in accordance with 5.4.2 and, using these results, determine σ_{HP}, in accordance with Method B for limited life and the number of load cycles N_L in the range as follows (see ISO 6336-1:2006, Table 2, for an explanation of the abbreviations used).

a) St, V, GGG(perl., bain.), GTS(perl.), Eh, IF, if a certain number of pits is permissible:

— For the limited life stress range, $6 \times 10^5 < N_L \leqslant 10^7$ in accordance with Figure 6:

$$\sigma_{HP} = \sigma_{HP\ ref}\, Z_N = \sigma_{HP\ ref} \left(\frac{3 \times 10^8}{N_L} \right)^{exp} \tag{7}$$

where

$$\exp = 0{,}370\,5 \log \frac{\sigma_{\text{HP stat}}}{\sigma_{\text{HP ref}}} \tag{8}$$

— For the limited life stress range, $10^7 < N_L \leqslant 10^9$ in accordance with Figure 6:

$$\sigma_{\text{HP}} = \sigma_{\text{HP ref}} Z_N = \sigma_{\text{HP ref}} \left(\frac{10^9}{N_L} \right)^{\exp} \tag{9}$$

where

$$\exp = 0{,}279\,1 \log \frac{\sigma_{\text{HP stat}}}{\sigma_{\text{HP ref}}} \tag{10}$$

b) St, V, GGG(perl., bain.), GTS(perl.), Eh, IF, when no pits are permissible:

— For the limited life stress range, $10^5 < N_L \leqslant 5 \times 10^7$ in accordance with Figure 6:

$$\sigma_{\text{HP}} = \sigma_{\text{HP ref}} Z_N = \sigma_{\text{HP ref}} \left(\frac{5 \times 10^7}{N_L} \right)^{\exp} \tag{11}$$

where

exp is as in Equation (8).

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

— For the limited life stress range, $10^5 < N_L \leqslant 2 \times 10^6$ in accordance with Figure 6:

$$\sigma_{\text{HP}} = \sigma_{\text{HP}}\ \text{ref}\ Z_N = \sigma_{\text{HP ref}} \left(\frac{2 \times 10^6}{N_L} \right)^{\exp} \tag{12}$$

where

$$\exp = 0{,}768\,6 \log \frac{\sigma_{\text{H P stat}}}{\sigma_{\text{HP ref}}} \tag{13}$$

d) NV(nitrocar.)

— For the limited life stress range, $10^5 < N_L \leqslant 2 \times 10^6$ in accordance with Figure 6:

$$\sigma_{\text{HP}} = \sigma_{\text{HP ref}} Z_N = \sigma_{\text{HP ref}} \left(\frac{2 \times 10^6}{N_L} \right)^{\exp} \tag{14}$$

where

$$\exp = 0{,}709\,8 \log \frac{\sigma_{\text{HP stat}}}{\sigma_{\text{HP ref}}} \tag{15}$$

Corresponding calculations may be determined for the range of long life.

6 Zone factor, Z_H, and single pair tooth contact factors, Z_B and Z_D

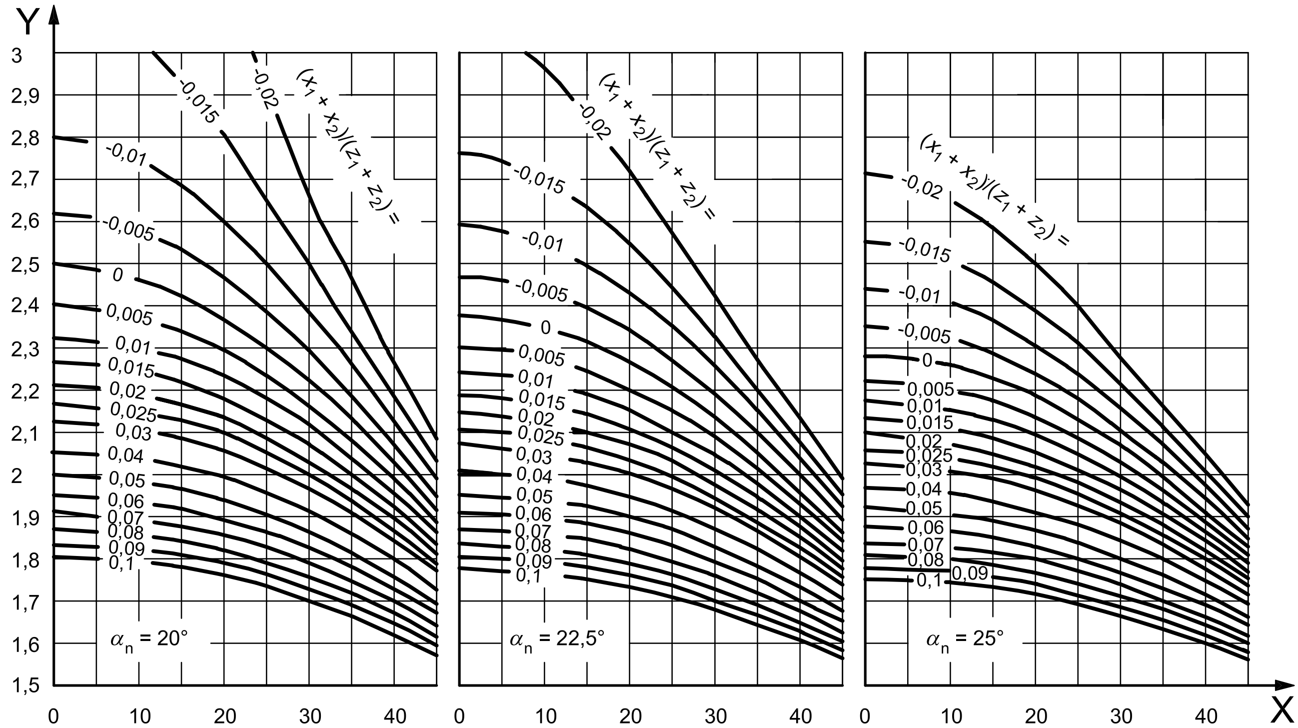
These factors account for the influence of tooth flank curvature on contact stress.

6.1 Zone factor, Z_H

The zone factor, Z_H, accounts for the influence on Hertzian pressure of tooth flank curvature at the pitch point and transforms the tangential load at the reference cylinder to normal load at the pitch cylinder.

6.1.1 Graphical values

Z_H can be taken from Figure 2 as a function of $(x_1 + x_2) / (z_1 + z_2)$ and β for external and internal gears having normal pressure angles $\alpha_n = 20°$, 22,5° or 25°.



Key

Y helix angle at reference circle, β (°)

X zone factor, Z_H

Figure 2 — Zone factor, Z_H

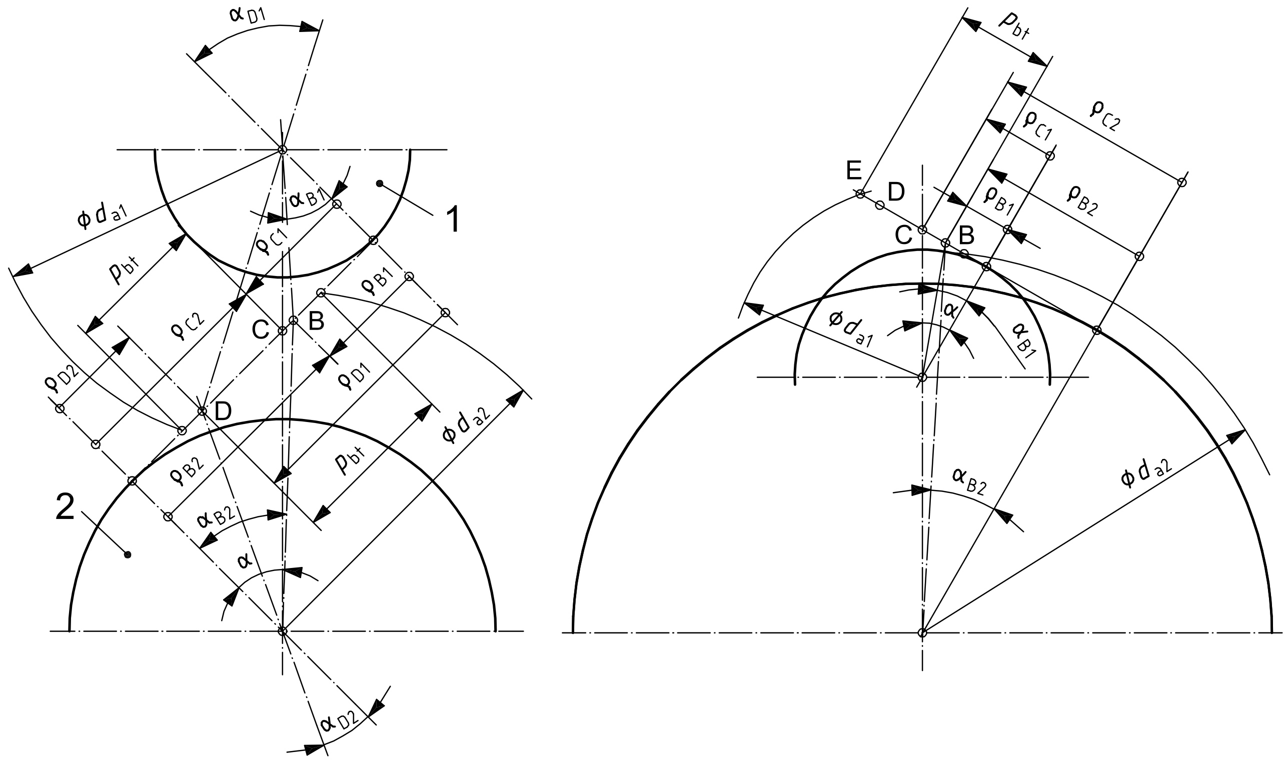
6.1.2 Determination by calculation

The zone factor is calculated by:

$$Z_H = \sqrt{\frac{2 \cos\beta_b \cos\alpha_{wt}}{\cos^2\alpha_t \sin\alpha_{wt}}} \tag{16}$$

6.2 Single pair tooth contact factors, Z_B and Z_D, for $\varepsilon_\alpha \leqslant 2$

The single pair tooth contact factors, Z_B and Z_D, are used to transform the contact stress at the pitch point of spur gears to the contact stress at the inner point B of single pair tooth contact of the pinion or at the inner point D of single pair tooth contact of the wheel if $Z_B > 1$ or $Z_D > 1$. See Figure 3 and 5.1.



External gearing Internal gearing

Key

1 pinion

2 wheel

Figure 3 — Radii of curvature at pitch point C and single pair tooth contact point B of pinion and D of wheel for determination of pinion single pair tooth contact factor Z_B in accordance with Equation (17) and wheel single pair tooth contact factor Z_D in accordance with Equation (18) (only for external spur gears)

In general, Z_D should only be determined for gears when $u < 1{,}5$. When $u > 1{,}5$, M_2 is usually less than 1,0 in which case Z_D is made equal to 1,0 in Equation (17).

For internal gears, Z_D shall be taken as equal to 1,0.

Determination by calculation:

$$M_1 = \sqrt{\frac{\rho_{C1}\,\rho_{C2}}{\rho_{B1}\,\rho_{B2}}} = \frac{\tan \alpha_{wt}}{\sqrt{\left(\sqrt{\frac{d_{a1}^2}{d_{b1}^2} - 1} - \frac{2\pi}{z_1}\right)\left(\sqrt{\frac{d_{a2}^2}{d_{b2}^2} - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_2}\right)}} \tag{17}$$

$$M_2 = \sqrt{\frac{\rho_{C1}\,\rho_{C2}}{\rho_{D1}\,\rho_{D2}}} = \frac{\tan\alpha_{wt}}{\sqrt{\left(\sqrt{\frac{d_{a2}^2}{d_{b2}^2}-1}-\frac{2\pi}{z_2}\right)\left(\sqrt{\frac{d_{a1}^2}{d_{b1}^2}-1}-(\varepsilon_\alpha-1)\frac{2\pi}{z_1}\right)}} \qquad (18)$$

Equation (17) and (18) are not valid, if undercut shortens the path of contact. See 8.2.1 for calculation of the profile contact ratio ε_α.

a) Spur gears with $\varepsilon_\alpha > 1$:

if $M_1 \leqslant 1$ then $Z_B = 1$; if $M_2 \leqslant 1$ then $Z_D = 1$;

if $M_1 > 1$ then $Z_B = M_1$; if $M_2 > 1$ then $Z_D = M_2$.

b) Helical gears with $\varepsilon_\alpha > 1$ and $\varepsilon_\beta \geqslant 1$:

$Z_B = Z_D = 1$

c) Helical gears with $\varepsilon_\alpha > 1$ and $\varepsilon_\beta < 1$:

Z_B and Z_D are determined by linear interpolation between the values for spur and helical gearing with $\varepsilon_\beta \geqslant 1$:

$Z_B = M_1 - \varepsilon_\beta (M_1 - 1)$ and $Z_B \geqslant 1$

$Z_D = M_2 - \varepsilon_\beta (M_2 - 1)$ and $Z_D \geqslant 1$

If Z_B or Z_D are made equal to 1, the contact stresses calculated using Equation (4) or (5) are the values for the contact stress at the pitch cylinder.

d) Helical gears with $\varepsilon_\alpha \leqslant 1$ and with $\varepsilon_\gamma > 1$: not covered by ISO 6336 — a careful analysis of the decisive contact stress along the path of contact is necessary.

Methods a), b) and c) apply to the calculation of contact stress when the pitch point lies in the path of contact. If the pitch point C is determinant and lies outside the path of contact, then Z_B and/or Z_D are determined for contact at the adjacent tip circle. For helical gearing when ε_β is less than 1,0, Z_B and Z_D are determined by linear interpolation between the values (determined at the pitch point or at the adjacent tip circle as appropriate) for spur gears and those helical gears with $\varepsilon_\beta \geqslant 1$.

6.3 Single pair tooth contact factors, Z_B and Z_D, for $\varepsilon_\alpha > 2$

In the case of meshing gear pairs of high precision with $2 < \varepsilon_\alpha \leqslant 2{,}5$, the entire tangential load in any transverse plane is supported by two pairs, or three pairs, of teeth in continued succession. For such gears, the calculation of contact stress is based on the inner point of two pair tooth contact of the pinion.

7 Elasticity factor, Z_E

The elasticity factor, Z_E, takes into account the influences of the material properties E (modulus of elasticity) and ν (Poisson's ratio) on the contact stress.

$$Z_E = \sqrt{\frac{1}{\pi\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}} \qquad (19)$$

When $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = \nu$:

$$Z_E = \sqrt{\frac{E}{2\pi(1-\nu^2)}} \tag{20}$$

For steel and aluminium $\nu = 0{,}3$ and therefore:

$$Z_E = \sqrt{0{,}175\,E} \tag{21}$$

For mating gears in material having different moduli of elasticity E_1 and E_2, the equivalent modulus

$$E = \frac{2E_1\,E_2}{E_1 + E_2} \tag{22}$$

may be used.

For some material combinations Z_E can be taken from Table 1.

Table 1 — Elasticity factor, Z_E, for some material combinations

Wheel 1			**Wheel 2**			
Material [a]	**Modulus of elasticity, E** N/mm²	**Poisson's ratio, ν**	**Material**	**Modulus of elasticity, E** N/mm²	**Poisson's ratio, ν**	Z_E $\sqrt{\text{N/mm}^2}$
St, V, Eh, IF, NT, NV	206 000	0,3	St, V, Eh, IF, NT, NV	206 000	0,3	189,8
			St(cast)	202 000		188,9
			GGG, GTS	173 000		181,4
			GG	126 000 to 118 000		165,4 to 162,0
St(cast)	202 000		St(cast)	202 000		188,0
			GGG, GTS	173 000		180,5
			GG	118 000		161,4
GGG, GTS	173 000		GGG, GTS	173 000		173,9
			GG	118 000		156,6
GG	126 000 to 118 000		GG	118 000		146,0 to 143,7

[a] See ISO 6336-1:2006, Table 2, for explanation of abbreviations used.

8 Contact ratio factor, Z_ε

The contact ratio factor, Z_ε, accounts for the influence of the transverse contact and overlap ratios on the surface load capacity of cylindrical gears. Calculation of the contact stress is based on a virtual facewidth b_{vir} instead of the actual facewidth b:

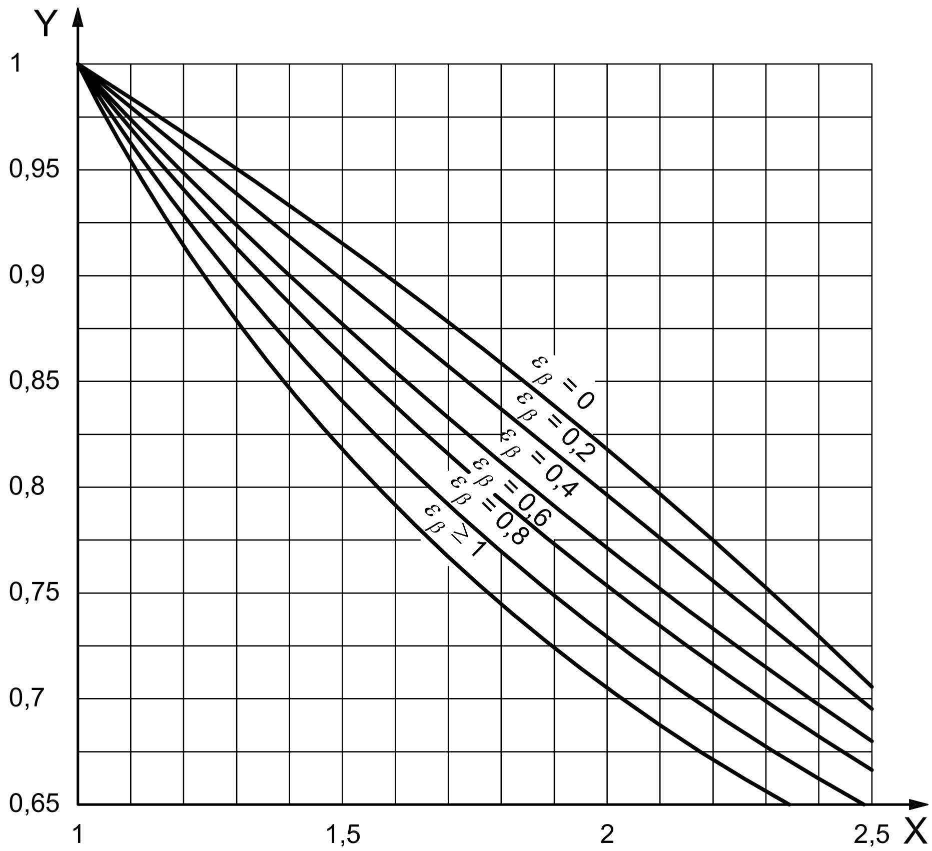
$$\frac{b_{vir}}{b} = \frac{1}{z_\varepsilon^2} \tag{23}$$

The average length of the line of contact calculated on a simplified basis is used as the appropriate value for helical gearing with $\varepsilon_\beta > 1$.

8.1 Determination of contact ratio factor, Z_ε

8.1.1 Graphical values

Z_ε for known contact and overlap ratio factors may be read from Figure 4.



Key

X transverse contact ratio, ε_α

Y contact ratio factor, Z_ε

Figure 4 — Contact ratio factor, Z_ε

8.1.2 Determination by calculation

a) Spur gears:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}} \tag{24}$$

The conservative value of $Z_\varepsilon = 1,0$ may be chosen for spur gears having a contact ratio less than 2,0.

b) Helical gears:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}(1-\varepsilon_\beta)+\frac{\varepsilon_\beta}{\varepsilon_\alpha}} \quad \text{for } \varepsilon_\beta < 1 \tag{25}$$

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}} \quad \text{for } \varepsilon_\beta \geqslant 1 \tag{26}$$

8.2 Calculation of transverse contact ratio, ε_α, and overlap ratio, ε_β

8.2.1 Transverse contact ratio, ε_α

The calculation is based on the roll angle ξ and the angular pitch τ, both expressed in radians in the following equations.

$$\varepsilon_\alpha = \frac{\xi_{\text{fw}1} + \xi_{\text{aw}1}}{\tau_1} = \frac{\xi_{\text{fw}2} + \xi_{\text{aw}2}}{\tau_2} \tag{27}$$

where

$\xi_{\text{fw}1,2}$ are the roll angles from the root form diameters to the working pitch point, taken as the least value of

— limited by the base diameters:

$$\xi_{\text{fw}1,2} = \tan \alpha_{\text{wt}} \tag{28}$$

— limited by the root form diameters:

$$\xi_{\text{fw}1} = \tan \alpha_{\text{wt}} - \tan \arccos \frac{d_{\text{b}1}}{d_{\text{soi}1}} \tag{29}$$

$$\xi_{\text{fw}2} = \tan \alpha_{\text{wt}} - \tan \arccos \frac{d_{\text{b}2}}{d_{\text{soi}2}} \tag{30}$$

— limited by the tip diameters of the wheel/pinion (start of active profile):

$$\xi_{\text{fw}1} = \left(\tan \arccos \frac{d_{\text{b}2}}{d_{\text{a}2}} - \tan \alpha_{\text{wt}} \right) \frac{z_2}{z_1} \tag{31}$$

$$\xi_{\text{fw}2} = \left(\tan \arccos \frac{d_{\text{b}1}}{d_{\text{a}1}} - \tan \alpha_{\text{wt}} \right) \frac{z_1}{z_2} \tag{32}$$

$\xi_{\text{aw}1,2}$ are the roll angles from the working pitch point to the tip diameter

$$\xi_{\text{aw}1} = \xi_{\text{fw}2} \frac{z_2}{z_1}, \xi_{\text{aw}2} = \xi_{\text{fw}1} \frac{z_1}{z_2} \tag{33}$$

$\tau_{1,2}$ is the pinion/wheel angular pitch:

$$\tau_1 = \frac{2\pi}{z_1}, \tau_2 = \frac{2\pi}{z_2} \tag{34}$$

Equations (28) to (34) do not take into account undercut (see Annex A).

8.2.2 Overlap ratio, ε_β

This is calculated by

$$\varepsilon_\beta = \frac{b \sin \beta}{\pi\, m_\text{n}} \tag{35}$$

See Equation (3) for the definition of facewidth.

9 Helix angle factor, Z_β

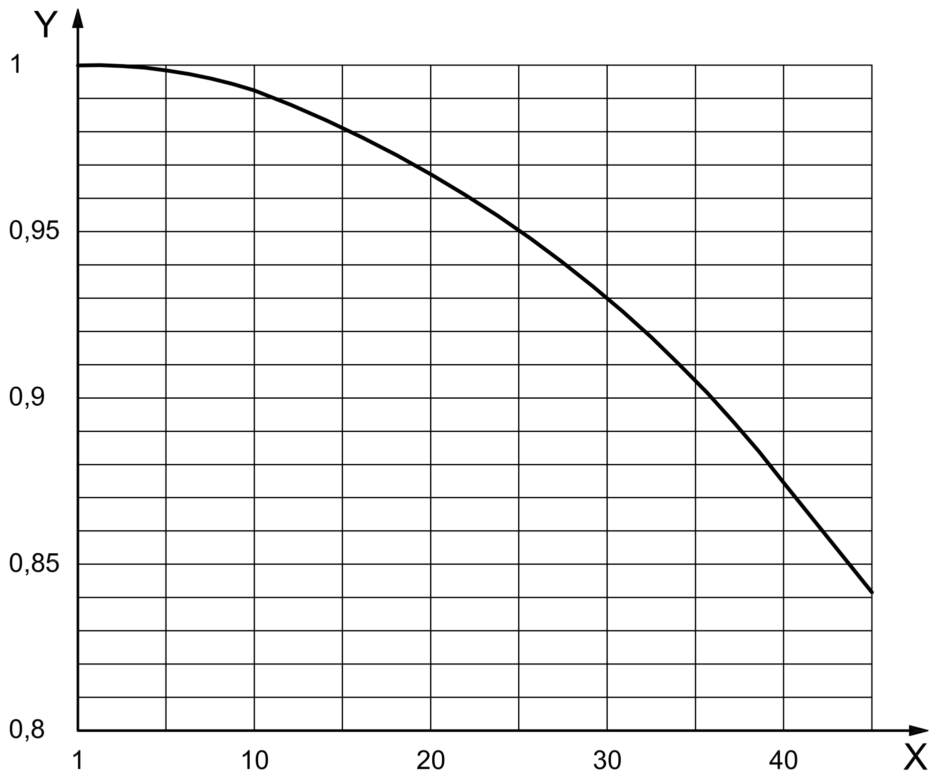
Independent of the influence of the helix angle on the length of path of contact, the helix angle factor, Z_β, accounts for the influence of the helix angle on surface load capacity, allowing for such variables as the distribution of load along the lines of contact.

Z_β is dependent only on the helix angle, β. For most purposes, the following empirical relationship is in sufficiently good agreement with experimental and service experience, but that agreement is only achieved when high accuracy and optimum modifications are employed:

$$Z_\beta = \sqrt{\cos\beta} \tag{36}$$

where β is the reference helix angle.

Z_β can also be read from Figure 5.



Key

X helix angle at reference circle, β (°)

Y helix angle factor, Z_β

Figure 5 — Helix angle factor, Z_β

10 Strength for contact stress

See 5.4 for general notes on the determination of limit values for contact stress; see 5.4.1 for the determination of pitting stress limit values.

10.1 Allowable stress numbers (contact), $\sigma_{\mathrm{H\,lim}}$, for Method B

Refer to 5.4.1.2 for details relevant to the following. For a demonstration of the use of $\sigma_{\mathrm{H\,lim}}$, see Equation (6). The value $\sigma_{\mathrm{H\,lim}}$ for a given material is considered as the highest value of contact stress, calculated in accordance with this part of ISO 6336, which the material will endure for at least 2×10^6 to 5×10^7 load cycles (see Figure 6 for start).

ISO 6336-5 provides information on commonly used gear materials, methods of heat treatment, and the influence of gear quality on values for allowable stress numbers, $\sigma_{\mathrm{H\,lim}}$, derived from test results of standard reference test gears.

Also see ISO 6336-5 for requirements concerning material and heat treatment for qualities ML, MQ and ME. Material quality MQ is generally selected unless otherwise agreed.

10.2 Allowable stress number values for Method B$_\mathrm{R}$

See 5.4.1.3 for detailed information. The allowable stress number values may be determined by means of roller tests or can be taken from the literature.

11 Life factor, Z_{NT} (for flanks)

The life factor, Z_{NT}, accounts for the higher contact stress, including static stress, which may be tolerable for a limited life (number of load cycles), as compared with the allowable stress at the point or “knee” on the curves of Figure 6, where $Z_{\mathrm{NT}} = 1{,}0$. Z_{NT} applies for standard reference use.

The principal influences are

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

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

c) lubrication regime,

d) failure criteria,

e) smoothness of operation required,

f) pitchline velocity,

g) cleanness of gear material,

h) material ductility and fracture toughness, and

i) residual stress.

For the purposes of this part of ISO 6336, the number of load cycles, N_L, is defined as the number of mesh contacts, under load, of the gear tooth being analysed.

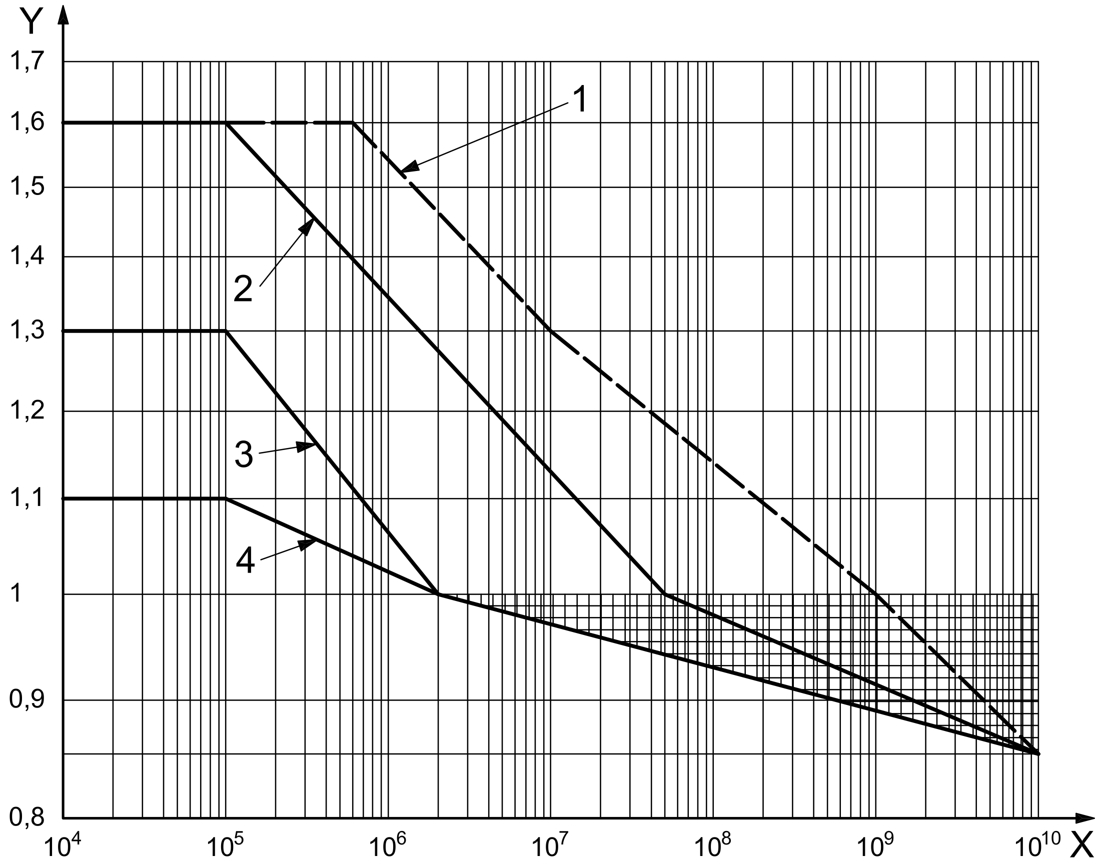
11.1 Life factor Z_{NT}: Method A

The S-N curve or damage curve derived from examples of the actual gear pair is determinant for load capacity at limited service life and is thus also determinant for the materials of both mating gears, the heat treatment, the relevant diameter, module, surface roughness of tooth flanks, pitch line velocity and the lubricant used. Since the S-N curve or 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 formulæ.

11.2 Life factor Z_{NT}: Method B

The permissible stress at limited service life or the safety factor in the limited life stress range is determined using life factor Z_{NT} for the standard reference test gear (see 5.4).

Z_{NT} for static and reference stresses may be taken from Figure 6 or Table 2.



Key

X number of load cycles, N_L

Y life factor, Z_{NT}

1 St, V, GGG (perl., bai.), GTS (perl.), Eh, IF [a]

2 St, V, GGG (perl., bai.), GTS (perl.), Eh, IF

3 GG, GGG (ferr.), NT (nitr.), NV (nitr.)

4 NV (nitrocar.)

[a] When limited pitting is permitted.

Figure 6 — Life factor, Z_{NT}, for standard reference test gears
(see ISO 6336-1:2006, Table 2, for explanation of abbreviations used)

Table 2 — Life factor, Z_{NT}

Material [a]	Number of load cycles	Life factor, Z_{NT}
St, V, GGG (perl., bai.), GTS (perl.), Eh, IF; only when a certain degree of pitting is permissible	$N_L \leqslant 6 \times 10^5$, static	1,6
	$N_L = 10^7$	1,3
	$N_L = 10^9$	1,0
	$N_L = 10^{10}$	0,85 up to 1,0 [b]
St, V, GGG (perl., bai.), GTS (perl.), Eh, IF	$N_L \leqslant 10^5$, static	1,6
	$N_L = 5 \times 10^7$	1,0
	$N_L = 10^9$	1,0
	$N_L = 10^{10}$	0,85 up to 1,0
GG, GGG (ferr.), NT (nitr.), NV (nitr.)	$N_L \leqslant 10^5$, static	1,3
	$N_L = 2 \times 10^6$	1,0
	$N_L = 10^{10}$	0,85 up to 1,0
NV (nitrocar.)	$N_L \leqslant 10^5$, static	1,1
	$N_L = 2 \times 10^6$	1,0
	$N_L = 10^{10}$	0,85 up to 1,0

a See ISO 6336-1:2006, Table 2 for explanation of abbreviations used.

b The lower value of Z_{NT} may be used for critical service, where pitting must be minimal. Values between 0,85 and 1,0 may be used for general purpose gearing. With optimum lubrication, material, manufacturing and experience 1,0 may be used.

12 Influence of lubricant film, factors Z_L, Z_v and Z_R

12.1 General

The lubricant film between the tooth flanks influences surface durability. The following have a significant influence:

a) viscosity of the lubricant in the mesh;

b) sum of the instantaneous velocities of the two tooth surfaces;

c) loading;

d) radius of relative curvature;

e) relationship between the combined values of the surface roughnesses of the tooth flanks, and the minimum thickness of the lubricant film.

According to EHD (elasto-hydrodynamic theory concerning the characteristics of lubricant films in zones of elastic sliding/rolling contact), a) to d) above influence the film dimensions and pressures.

Furthermore, the nature of the lubricant (mineral oil, synthetic oil), its origin, its age, etc. will also have an effect on surface durability.

NOTE Information and recommendations concerning the choice of lubricant type and viscosity can be found in other publications.

12.2 Influence of lubricant film: Method A

By Method A the influence of the lubricant film on surface durability is determined on the basis of reliable service experience or tests on geared transmissions having comparable dimensions, materials, lubricants and operating conditions. The provisions of ISO 6336-1:2006, 4.1.12, are relevant.

12.3 Influence of lubricant film, factors Z_L, Z_v and Z_R: Method B

The information provided is based on tests using standard reference test gears. The shaded fields in Figures 7 to 9 show the tendency of the three factors which are included in the calculation procedure according to Method B:

— Z_L for the influence of the nominal lubricant viscosity (as a characteristic value of the influence of the lubricant) on the effect of the lubricant film;

— Z_v for the influence of the pitch line velocity on the effect of the lubricant film;

— Z_R for the influence of surface roughness of the flanks after running-in (as a manufacturing process) on the effect of the lubricant film.

The considerable scatter (width of the hatched field) indicates that there are influences other than those mentioned above, also involved in the lubricant film, which are not included in the calculation procedure.

These omissions were taken into consideration when plotting the curves in Figures 7 to 9. Clearly, they cannot be considered as representing physical laws. They are, of course, empirical.

The influence factors are presented as independent of one another, but in reality cannot be completely separated. For this reason, test results which were obtained by varying a single variable, while others were held constant, were adjusted to take into account field experiences with gears of different sizes and operating conditions. Thus, some of the recorded values do not correlate directly with test results. In general, through-hardened gears are more sensitive than case-hardened gears to the influences of viscosity, pitch line velocity and surface roughness. This is reflected in the empirical curves drawn in the scatter bands in Figures 7 to 9 inclusive. When a gear pair consists of one which is of hard and one which is of soft material, the factors Z_L, Z_v and Z_R shall be determined for the softer of the materials. See ISO 6336-5 for $\sigma_{H\,lim}$ values of common gear materials.

The influence of the lubricant film is only fully effective at the long life stress level. The influence is low at higher limited-life stress levels (see Clause 11 and 5.4).

The lubricant factor Z_L was derived from tests using mineral oil (with and without EP additives). By comparison, when testing certain synthetic lubricants in combination with case hardened test gears, values of Z_L up to 1,1 times higher and with through-hardened test gears up to 1,4 times higher were observed.

These values should be verified in each individual case (where possible, curves similar to those provided for mineral oils should be prepared for synthetic oils).

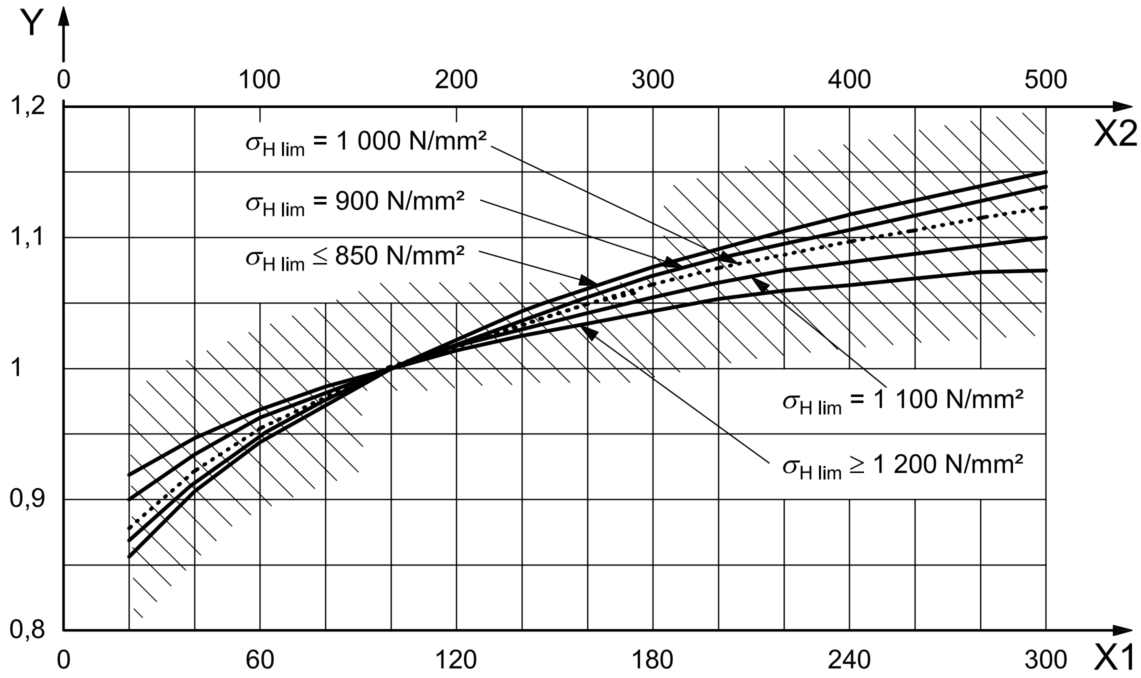
12.3.1 Factors Z_L, Z_v, Z_R for reference stress

12.3.1.1 Lubricant factor, Z_L

The factor Z_L for mineral oils (with or without extreme pressure, EP, additives) can be determined as a function of nominal viscosity at 40 °C (or 50 °C) and the value $\sigma_{H\,lim}$ of the softer of the materials of the mating gear pair, by following the directions in 12.3.1.1.2 a) and b). The values for ν_{40} apply for the viscosity index VI = 95 and viscosities up to 500 cSt at 40 °C; for higher viscosities, use the value obtained at 500 cSt at 40 °C or 300 cSt at 50 °C to determine the value of Z_L.

12.3.1.1.1 Graphical values

Z_L can be read from Figure 7 as a function of the nominal viscosity of the lubricant at 40 °C (or 50 °C) and the $\sigma_{H\,lim}$ value.



Key

X1 nominal viscosity at 50 °C, ν_{50}, mm^2/s

X2 nominal viscosity at 40 °C, ν_{40}, mm^2/s

Y lubricant factor, Z_L

Figure 7 — Lubricant factor, Z_L

12.3.1.1.2 Determination by calculation

a) Z_L can be calculated using Equations (37) to (41) which are consistent with the curves in Figure 7:

$$Z_L = C_{ZL} + \frac{4(1{,}0 - C_{ZL})}{\left(1{,}2 + \frac{80}{\nu_{50}}\right)^2} = C_{ZL} + \frac{4(1{,}0 - C_{ZL})}{\left(1{,}2 + \frac{134}{\nu_{40}}\right)^2} \tag{37}$$

In the range 850 N/mm^2 $\leqslant \sigma_{H\,lim} \leqslant$ 1 200 N/mm^2

$$C_{ZL} = \frac{\sigma_{H\,lim}}{4375} + 0{,}6357 \tag{38}$$

In the range $\sigma_{H\,lim} < 850$ N/mm^2

$$C_{ZL} = 0{,}83 \tag{39}$$

In the range $\sigma_{H\,lim} > 1\,200$ N/mm^2

$$C_{ZL} = 0{,}91 \tag{40}$$

b) Alternatively, Z_L can be calculated from Equation (41):

$$Z_L = C_{ZL} + 4\,(1{,}0 - C_{ZL})\ \nu_f \tag{41}$$

where $\nu_f = 1 / (1{,}2 + 80/\nu_{50})^2$ using viscosity parameters from Table 3.

Table 3 — Viscosity parameters

ISO viscosity class (grade)		**VG 32** [a]	**VG 46** [a]	**VG 68** [a]	**VG 100**	**VG 150**	**VG 220**	**VG 320**
Nominal viscosity, mm^2/s	ν_{40}	32	46	68	100	150	220	320
	ν_{50}	21	30	43	61	89	125	180
Viscosity parameter	ν_f	0,040	0,067	0,107	0,158	0,227	0,295	0,370

[a] Only for high speed transmission.

12.3.1.2 Velocity factor, Z_v

The velocity factor, Z_v, can, as a function of pitch line velocity and the allowable stress number $\sigma_{H\,lim}$ of the softer of the materials of the mating gear pair, be determined in accordance with 12.3.1.2.1 or 12.3.1.2.2.

12.3.1.2.1 Graphical values

Z_v can be taken from Figure 8 as a function of the pitch line velocity and the $\sigma_{H\,lim}$ value.

12.3.1.2.2 Determination by calculation

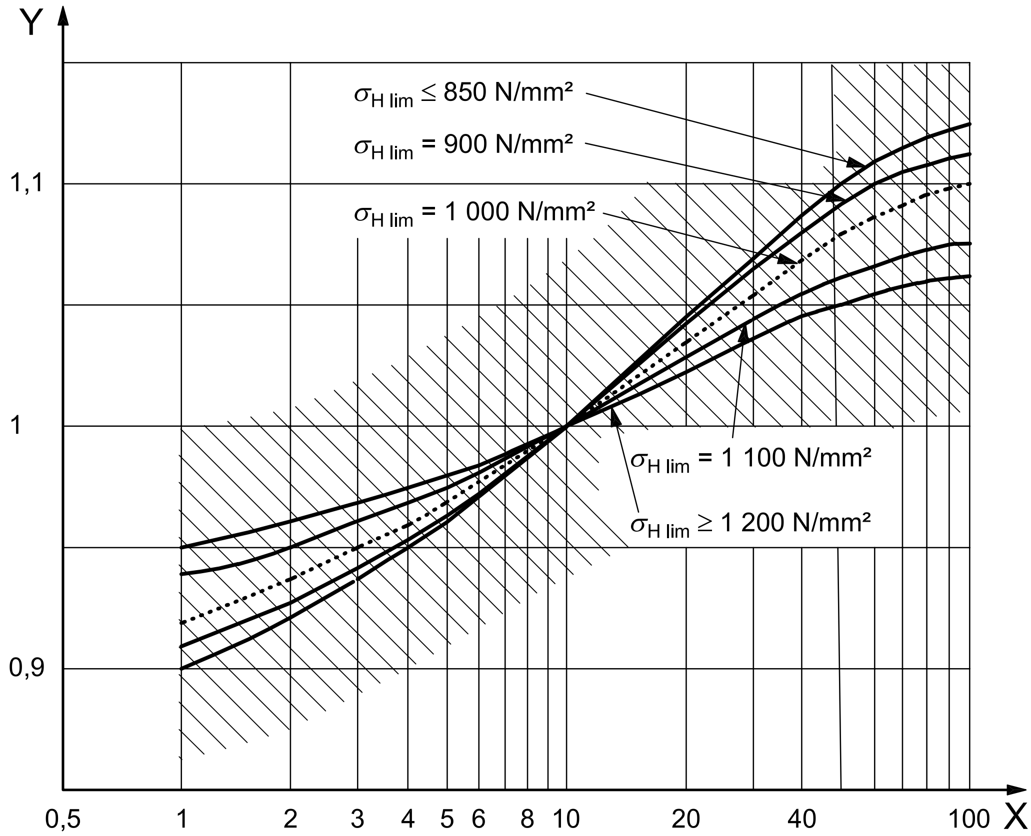
Z_v can be calculated using Equations (42) and (43). They reproduce the curves in Figure 8.

$$Z_v = C_{Zv} + \frac{2(1{,}0 - C_{Zv})}{\sqrt{0{,}8 + \frac{32}{v}}} \tag{42}$$

where

$$C_{Zv} = C_{ZL} + 0{,}02 \tag{43}$$

[see Equations (38) to (40) for values of C_{ZL}].



Key

X pitch line velocity, v, m/s

Y velocity factor, Z_v

Figure 8 — Velocity factor, Z_v

12.3.1.3 Roughness factor, Z_R

12.3.1.3.1 General

The roughness factor, Z_R, can be determined in accordance with the following, as a function of the surface condition (roughness) of the tooth flanks, the dimensions (radius of relative curvature, ρ_{red}) [2)], and the $\sigma_{H\,lim}$ value for the softer material of the mating gear pair.

Z_R can be read from curves or calculated as a function of the "mean relative roughness" (relative to radius of relative curvature at the pitch point $\rho_{red} = 10$ mm).

Mean peak-to-valley roughness of the gear pair:

$$Rz = \frac{Rz_1 + Rz_2}{2} \tag{44}$$

2) ρ_{red} is defined here as the radius of relative curvature at the pitch point. This also applies for internal gear pairs. For pinion – rack contact, $\rho_{red} = \rho_1$.

The peak-to-valley roughness determined for the pinion, Rz_1, and for the wheel, Rz_2, are mean values for the peak-to-valley roughness Rz measured on several tooth flanks [3)].

The mean roughness Rz_1 (pinion flank) and Rz_2 (wheel flank) shall be determined for their surface condition after manufacture, including any running-in treatment, planned as a manufacturing, commissioning or in-service process, when it is safe to assume that it will take place.

Mean relative peak-to-valley roughness for the gear pair:

$$Rz_{10} = Rz \sqrt[3]{\frac{10}{\rho_{red}}} \tag{45}$$

Radius of relative curvature:

$$\rho_{red} = \frac{\rho_1 \, \rho_2}{\rho_1 + \rho_2} \tag{46}$$

where

$$\rho_{1,2} = 0{,}5 \; d_{b1,2} \tan \alpha_{Wt} \tag{47}$$

For external gearing, d_b has a positive sign; for internal gearing, d_b has a negative sign.

12.3.1.3.2 Graphical values

Z_R can be read from Figure 9 as a function of the mean relative peak-to-valley roughness and the $\sigma_{H\,lim}$ value.

12.3.1.3.3 Determination by calculation

Z_R can be calculated using the following equations which are consistent with the curves in Figure 9.

$$Z_R = \left(\frac{3}{Rz_{10}} \right)^{C_{ZR}} \tag{48}$$

In the range 850 N/mm^2 $\leqslant \sigma_{H\,lim} \leqslant$ 1 200 N/mm^2:

$$C_{ZR} = 0{,}32 - 0{,}000\,2\sigma_{H\,lim} \tag{49}$$

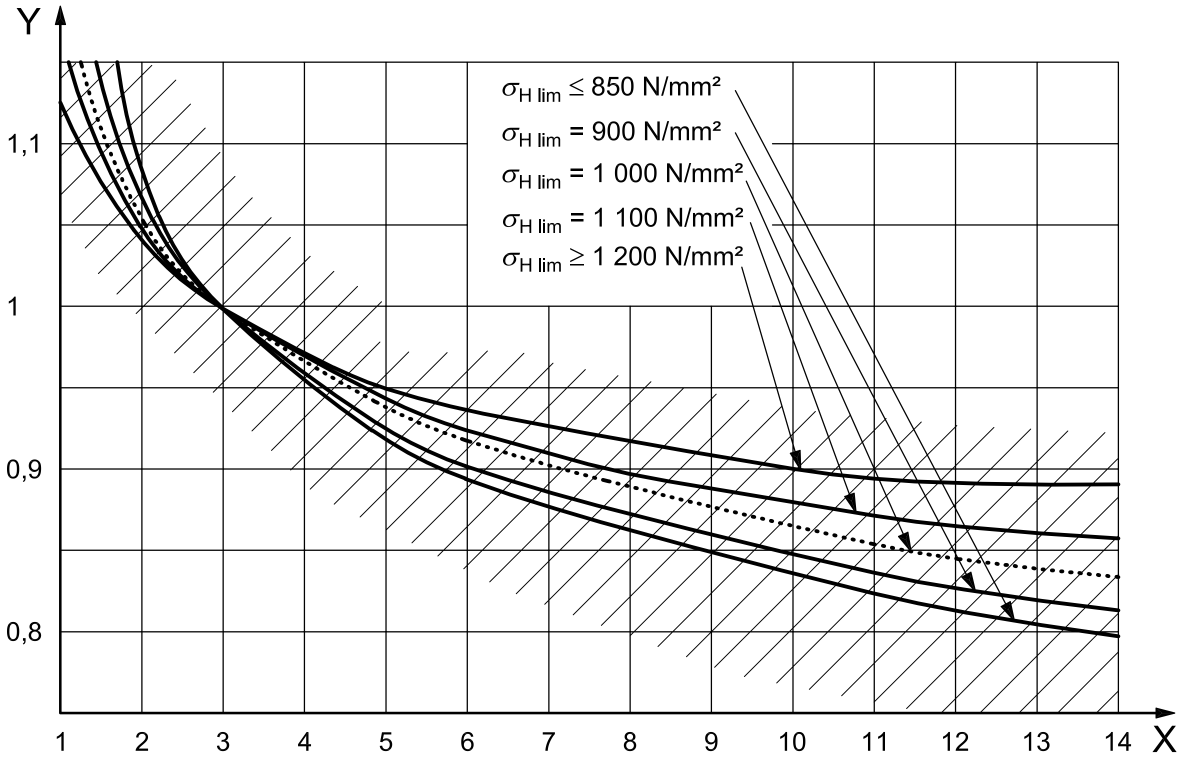
In the range $\sigma_{H\,lim} <$ 850 N/mm^2:

$$C_{ZR} = 0{,}15 \tag{50}$$

In the range $\sigma_{H\,lim} >$ 1 200 N/mm^2:

$$C_{ZR} = 0{,}08 \tag{51}$$

3) If roughness stated is an Ra value (= CLA value) (= AA value), the following approximation may be used for conversion: R_a = CLA = AA = Rz/6.



Key

X mean relative peak-to-valley roughness, Rz_{10}, µm

Y roughness factor, Z_R

Figure 9 — Roughness factor, Z_R

12.3.2 Factors Z_L, Z_v and Z_R for static stress

The relationships in Equation (52) are valid for the static and upper limited life stress ranges (characterized by the upper horizontal branches of S-N curves)

$$Z_L = Z_v = Z_R = 1{,}0 \qquad (52)$$

13 Work hardening factor, Z_W

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.

13.1 Work hardening factor, Z_W: Method A

The increase in load-bearing capacity as a result of the influences listed above is to be determined in accordance with reliable operating experience or tests on geared transmissions of comparable dimensions, materials, lubricants and operating conditions. The provisions given in ISO 6336-1:2006, 4.1.12, are relevant.

13.2 Work hardening factor, Z_W: Method B

13.2.1 Surface-hardened pinion with through-hardened gear

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

Although the curves in Figure 10 were carefully chosen, they cannot be interpreted as a physical law for the reasons mentioned above. They are, like Equation (53), empirical.

The equivalent roughness, Rz_H, is determined as

$$Rz_H = \frac{Rz_1 \,(10 / \rho_{red})^{0,33} \,(Rz_1 / Rz_2)^{0,66}}{(\nu_{40}\, v / 1\,500)^{0,33}} \tag{53}$$

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_{red} is radius of relative curvature at pitch point, in millimetres (mm), see Equation (43);

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

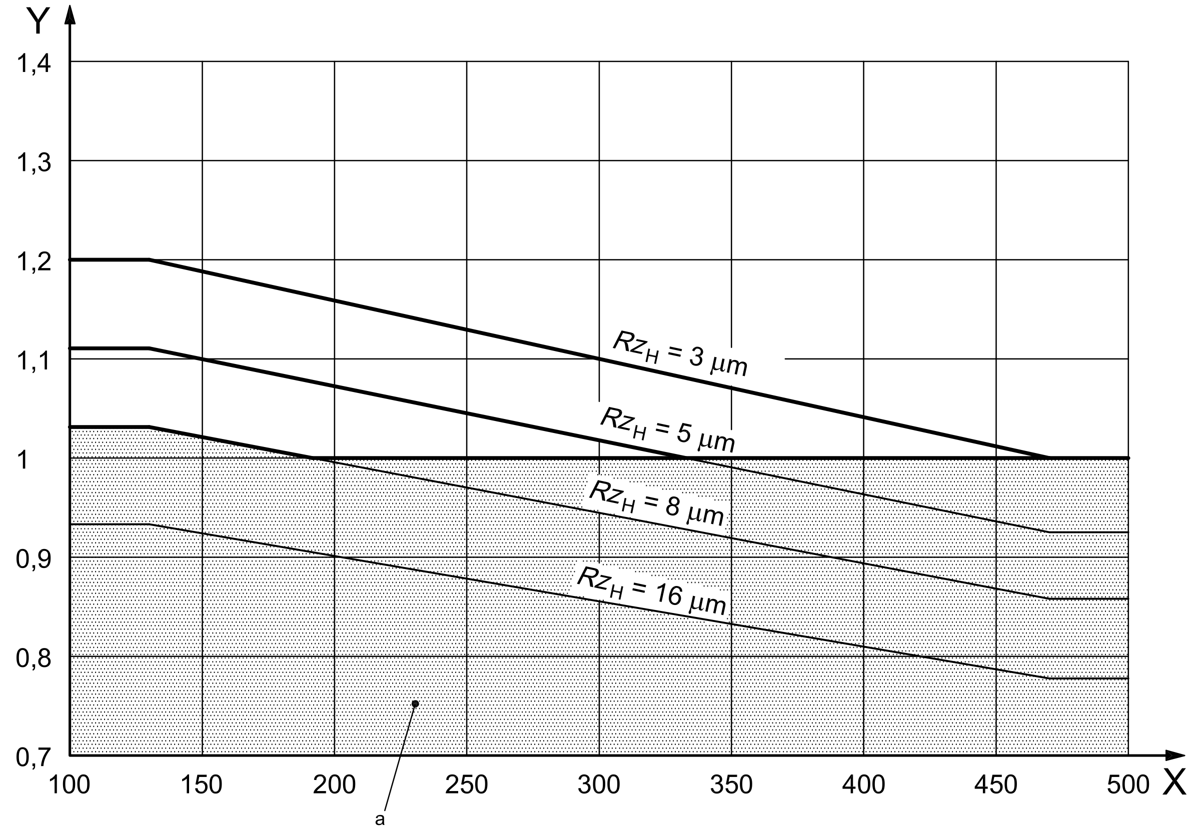
v is the pitch line velocity, in metres per second (m/s).

The value of Z_W is different for static, limited life and reference stress (stress ranges, see Figure 6).

NOTE Especially for rough pinion surfaces, values of $Z_W < 1$ may 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 is recommended to be carried out in this case. Wear of the surface is not covered by ISO 6336.

13.2.1.1 Z_W for reference and long life stress, graphed values

Z_W for reference and long life stress can be taken from Figure 10 for the conditions listed in 12.2 as a function of the flank hardness of the softer wheel.



Key

X tooth flank hardness of softer wheel, HB

Y work hardening factor, Z_W

[a] Shaded area: $Z_W = 1$. Effects of wear (not covered by ISO 6336).

Figure 10 — Work hardening factor Z_W for through-hardened gear/case-hardened pinion, reference stress

13.2.1.2 Z_W for reference and long life stress, determination by calculation

For 130 ⩽ HB ⩽ 470, Z_W for reference and long life stress is calculated as

$$Z_W = \left(1{,}2 - \frac{\mathrm{HB} - 130}{1700}\right)\left(\frac{3}{Rz_H}\right)^{0{,}15} \tag{54}$$

where

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

Rz_H is the equivalent roughness according to Equation (53).

For HB < 130:

$$Z_W = 1{,}2\left(\frac{3}{Rz_H}\right)^{0{,}15} \tag{55}$$

For HB > 470:

$$Z_W = \left(\frac{3}{Rz_H}\right)^{0,15} \tag{56}$$

The calculated values for Z_W are consistent with the curves in Figure 10.

13.2.1.3 Z_W for static stress

For 130 ⩽ HB ⩽ 470, Z_W for the static stress range is calculated as

$$Z_W = 1,05 - \frac{HB - 130}{680} \tag{57}$$

For HB < 130:

$$Z_W = 1,05 \tag{58}$$

For HB > 470:

$$Z_W = 1 \tag{59}$$

13.2.2 Through-hardened pinion and gear

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 to the pinion.

The value of Z_W is different for static, limited life and reference stress (stress ranges, see Figure 6).

13.2.2.1 Z_W for reference and long life stress, graphed values

Values of Z_W for long life stress may be taken from Figure 11.

13.2.2.2 Z_W for reference and long life stress, determination by calculation

For 1,2 ⩽ HB_1/HB_2 ⩽ 1,7, Z_W for long life stress is determined as

$$Z_W = 1,0 + A\ (u - 1,0) \tag{60}$$

where

$$A = 0,008\,98\ HB_1 / HB_2 - 0,008\,29 \tag{61}$$

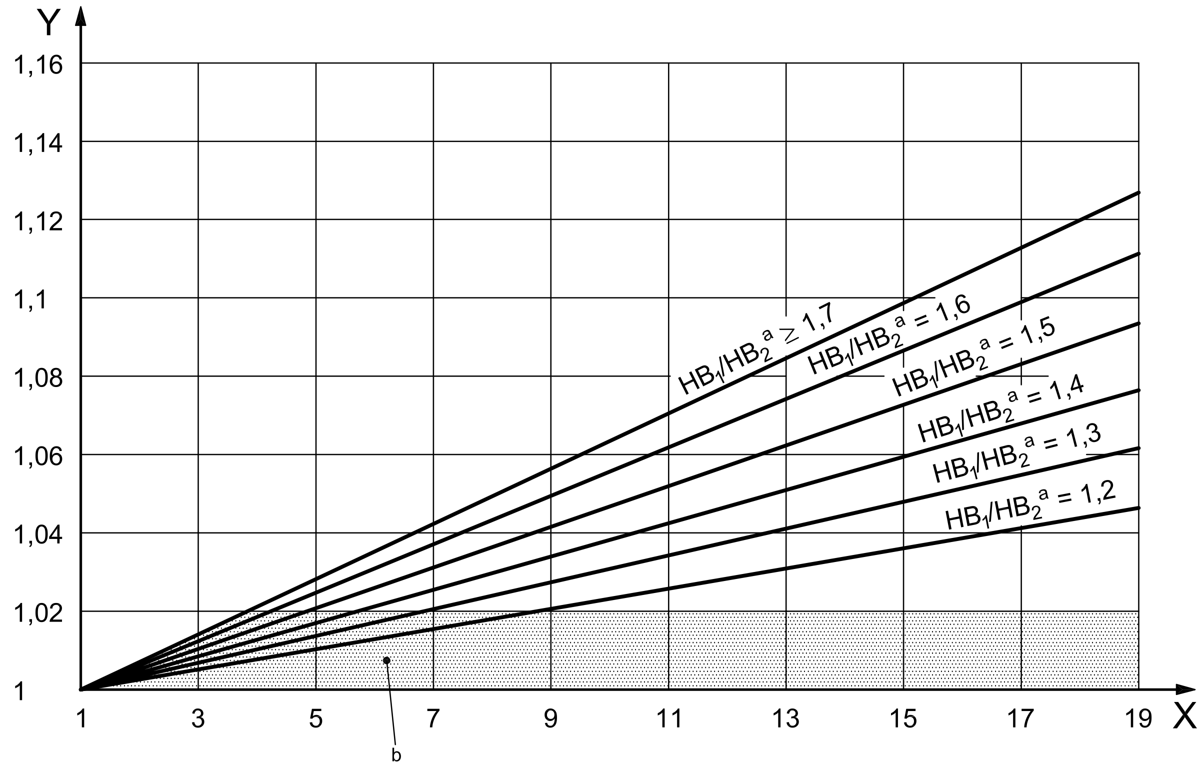
HB_1 is the Brinell hardness number of the pinion;

HB_2 is the Brinell hardness number of the gear;

u is the gear set ratio; if $u > 20$ use $u = 20$.

For $HB_1/HB_2 < 1,2$:

$$Z_W = 1,0 \tag{62}$$



Key

X single reduction gear ratio, u

Y work hardening factor, Z_W

a Calculated hardness ratio.

b For $HB_1/HB_2 < 1,2$, use $Z_W = 1$.

Figure 11 — Work hardening factor Z_W for through-hardened gear and pinion

For $HB_1/HB_2 > 1,7$:

$$Z_W = 1,0 + 0,006\,98\,(u - 1,0) \tag{63}$$

The calculated values for Z_W are is consistent with the curves in Figure 11.

13.2.2.3 Z_W for static stress

For the static stress range, $Z_W = 1,0$.

14 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 of component size (larger number of weak points in structure), as a consequence of the influence on subsurface defects of the smaller stress gradients which occur (theoretical stress analysis) and the influence of size on material quality (effect on forging process, variations in structure, etc.). Important influence parameters are

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

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

c) radius of flank curvature, and

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

In this part of ISO 6336, Z_X is taken to be 1,0.

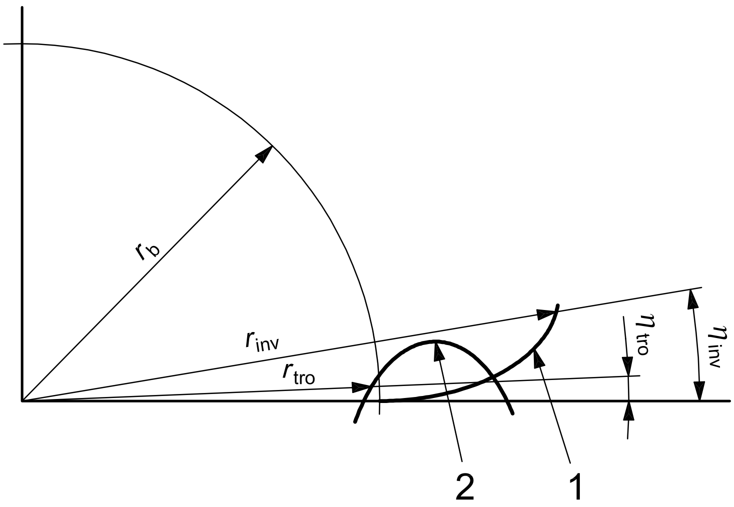
Annex A
(informative)

Start of involute

A.1 Equations of involute and trochoid

For the determination of the start of involute, a polar coordinate system is used (see Figure A.1).

NOTE This method does not account for backlash (although using xE-factor will work) or for shaper cut gears.



Key

1 involute

2 trochoid

Figure A.1 — Polar coordinate system for involute and trochoid

For a rack-type generation, the equations of the involute and the trochoid, taking into account the elliptical tip of the rack and the tool protuberance, are

$$r_{inv} = \frac{d_b}{2}\sqrt{1+\xi^2} \quad \text{(A.1)}$$

$$\eta_{inv} = \xi - \text{arc tan}\,\xi \quad \text{(A.2)}$$

$$r_{tro} = \sqrt{\left(\frac{d}{2} - B\right)^2 + \left(\frac{B\cos\beta}{\tan\varphi}\right)^2} \quad \text{(A.3)}$$

$$\eta_{tro} = \theta + \varepsilon - \alpha_t \quad \text{(A.4)}$$

where

$$B = h_{fP} - x\,m_n - \rho_{fP} + \rho_{fP}\sin\phi \quad \text{(A.5)}$$

$$\theta = \tan \alpha_t + \frac{2}{d}\left(\rho_{fP}\frac{\cos \phi}{\cos \beta} - A - B\frac{\cos \beta}{\tan \phi}\right) \tag{A.6}$$

$$A = \frac{\rho_{fP} - pr}{\cos \alpha_n \cos \beta} + (h_{fP} - x\,m_n - \rho_{fP})\tan \alpha_t \tag{A.7}$$

$$\varepsilon = \text{arc tan}\frac{B\cos \beta}{(d/2 - B)\tan \varphi} \tag{A.8}$$

The trochoid/tip parameter φ is shown in Figure A.2.

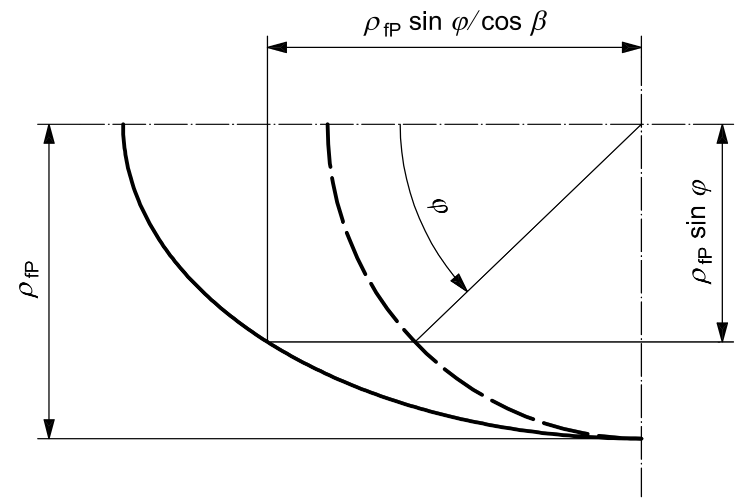


Figure A.2 — Trochoid/tip parameter φ

A.2 Undercut condition

Undercut exists if

$$\frac{d}{2}\sin^2 \alpha_t - [h_{fP} - x\,m_n - \rho_{fP}(1 - \sin \alpha_n)] < 0 \tag{A.9}$$

A.3 Determination of start of involute

If undercut doesn't exist, the radius to the point of start of involute, r_{soi} can be calculated with Equation (A.10):

$$r_{soi} = r_{tro,(\phi = \alpha_n)} = \sqrt{\left[\frac{d}{2} - (h_{fP} - x\,m_n - \rho_{fP} + \rho_{fP}\sin \alpha_n)\right]^2 + \left[\frac{(h_{fP} - x\,m_n - \rho_{fP} + \rho_{fP}\sin \alpha_n)}{\tan \alpha_t}\right]^2} \tag{A.10}$$

If undercut does exist the point of start of involute is located at the intersection of the involute and the trochoid, so that two conditions should be verified:

$$r_{tro} = r_{inv} \tag{A.11}$$

$$\eta_{tro} = \eta_{inv} \tag{A.12}$$

Equation (A.11) allows ξ to be expressed as a function of ν:

$$\xi = \xi(\phi) = \sqrt{\left(\frac{d/2 - B(\phi)}{d_{\mathrm{b}}/2}\right)^2 + \left(\frac{\mathrm{B}(\phi)\cos\beta}{d_{\mathrm{b}}/2\tan\phi}\right)^2 - 1} \qquad \text{(A.13)}$$

Equation (A.12) may be written as:

$$\theta(\phi) + \varepsilon(\phi) - \xi(\phi) + \text{arc}\tan\xi(\phi) - \alpha_{\mathrm{t}} = f(\phi) = 0 \qquad \text{(A.14)}$$

Consequently, the root of $f(\nu)$ is the value of ν at the intersection point, which allows to calculate r_{soi} and η_{soi} according to Equations (A.1) to (A.4).

$f(\nu)$ may have two solutions in between the interval $0 < \nu < \pi/2$. However, the point of start of involute is defined by the root giving the higher value of r_{tro}, which is always the smaller root, as derived from Figure A.2. The correct value of ν_{soi} may be obtained by solving Equation (A.14) with an iteration method, such as by Newton-Raphson, starting from a small value for ν. A few interation steps are required to find a solution.

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