INTERNATIONAL STANDARD

ISO 6336-2

Second edition
2006-09-01

Corrected version
2007-04-01

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
ISO 6336-2:2006(E)

PDF disclaimer

This PDF file may contain embedded typefaces. In accordance with Adobe's licensing policy, this file may be printed or viewed but shall not be edited unless the typefaces which are embedded are licensed to and installed on the computer performing the editing. In downloading this file, parties accept therein the responsibility of not infringing Adobe's licensing policy. The ISO Central Secretariat accepts no liability in this area.

Adobe is a trademark of Adobe Systems Incorporated.

Details of the software products used to create this PDF file can be found in the General Info relative to the file; the PDF-creation parameters were optimized for printing. Every care has been taken to ensure that the file is suitable for use by ISO member bodies. In the unlikely event that a problem relating to it is found, please inform the Central Secretariat at the address given below.

ISO copyright office
Case postale 56 • CH-1211 Geneva 20
Tel. + 41 22 749 01 11
Fax + 41 22 749 09 47
E-mail copyright@iso.org
Web www.iso.org

Published in Switzerland

Contents

Page

Foreword

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

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

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

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.

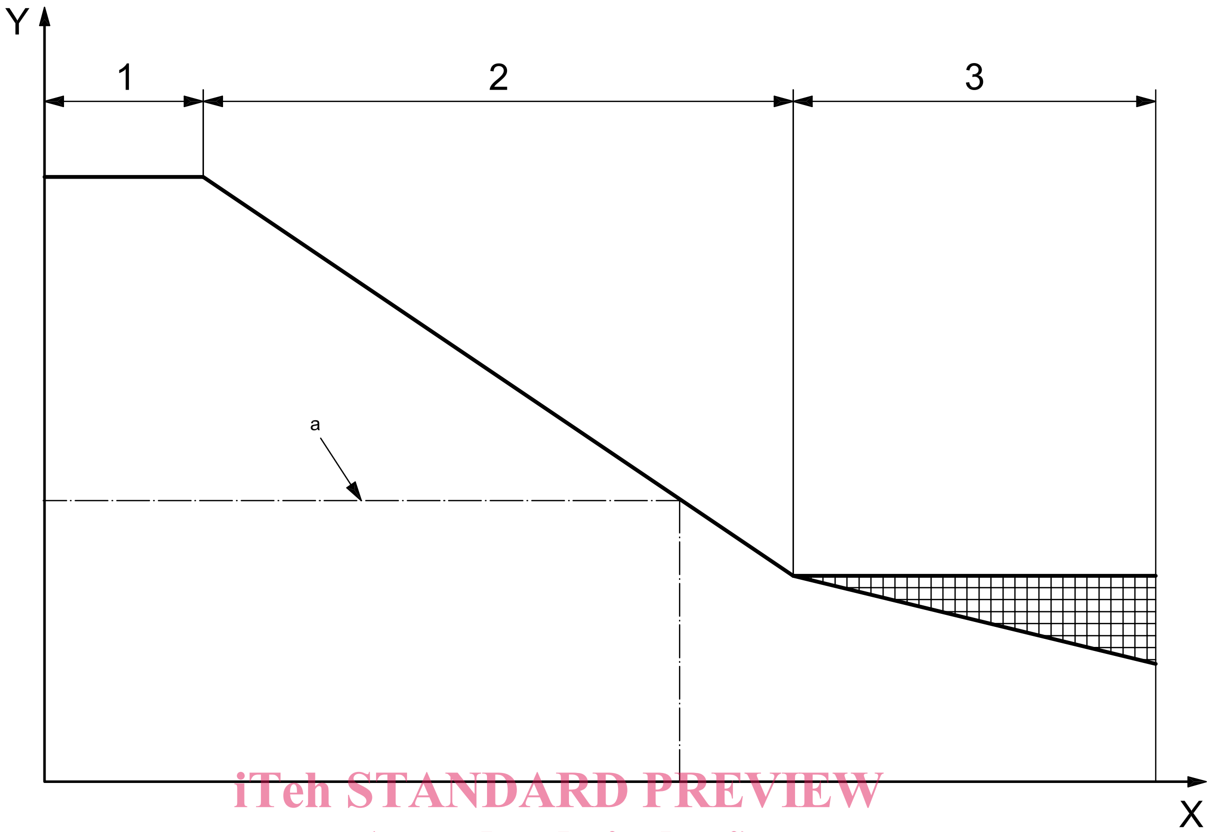
b) **Permissible contact stress (static)**, $\sigma_{HP\,stat}$, is determined in accordance with Equation (6), with all influence factors (for static stress) following Method B.

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:

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

