SLOVENSKI STANDARD
SIST ISO 6336-2:2002

01-julij-2002

Izraļ un nosilnosti ravnozobih in poġevnozobih zobnikov - 2. del: Izraļ un obratovalne vzdrǵljivosti zobnih bokov (jamiļ enje)

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 (piqûres)

Ta slovenski standard je istoveten z: **ISO 6336-2:1996**

ICS:

21.200	Gonila	Gears

SIST ISO 6336-2:2002 **en**

INTERNATIONAL STANDARD

ISO 6336-2

First edition
1996-06-15

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 superficielle (piquage)

Reference number
ISO 6336-2:1996(E)

Contents

International Organization for Standardization
Case Postale 56 • CH-1211 Genève 20 • Switzerland

Printed in Switzerland

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 liason with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

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

International Standard 6336-2 was prepared by Technical Committee ISO/TC60, Gears, Subcommittee SC2, Gear capacity calculation.

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*

Annex A is for information only.

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 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, 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)

1 Scope

This part of ISO 6336 specifies the fundamental formulae for use in the determination of the surface load capacity of cylindrical gears with involute internal or external teeth. It includes formulae for all influences on surface durability for which quantitative assessments can be made. It applies primarily to oil-lubricated transmissions, but may 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 formulae are valid for cylindrical gears with tooth profiles in accordance with the basic rack standardized in ISO 53. They may be used for teeth where the actual transverse contact ratio is less than $\epsilon_{\alpha n}$ = 2,5. The results are in good agreement with other methods for the range as indicated in the scope of ISO 6336-1.

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.

These formulae 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 3.

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 standards contain provisions which, through reference in this text, constitute provisions of this part of ISO 6336. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this part of ISO 6336 are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 53: 1974, *Cylindrical gears for general and heavy engineering - Basic rack.*

ISO 6336-1: 1996, *Calculation of load capacity of spur and helical cylindrical gears - Part 1: Basic principles, introduction and general influence factors.*

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

3 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 definitions, relevant to average working conditions 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 may be enlarged by initial pitting, and the rate of generation of pits may 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 which 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 of leading 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 may 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) can 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, subclause 4.1.3. It is recommended that the manufacturer and customer agree on the values of the minimum safety factor.

4 Basic formulae

NOTE 1 – All symbols, terms and units are defined in ISO 6336-1.

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 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}. Three categories are recognized in the calculation of σ_H as follows.

a) Spur gears

1) Spur pinion: 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.

2) Spur wheel: in the case of external teeth, σ_H is usually calculated at the pitch point. In special cases, particularly in the case of small transmission ratios (see 5.2), σ_H is greater at the inner point of single pair tooth contact of the wheel and is thus determinant. For internal teeth, σ_H is always calculated at the pitch point.

b) Helical gearing with overlap ratio $\epsilon_\beta \geq 1$

σ_H is always calculated at the pitch point for pinion and wheel.

c) Helical gearing with overlap ratio $\epsilon_\beta < 1$

In this case 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 $\epsilon_\beta = 1$ in which the determination of σ_H for each is to be based on the numbers of teeth on the actual gears.

4.1 Contact stress σ_H

As stated, the contact stress is to be calculated on the basis of Hertzian pressure (see introduction).

4.1.1 Contact stress for the pinion

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 a distribution factor K_γ to follow K_A in equation (1), to adjust the average tangential load per mesh as necessary.

$$\sigma_H = Z_B \, \sigma_{H0} \sqrt{K_A \, K_v \, K_{H\beta} \, K_{H\alpha}} \leq \sigma_{HP} \quad \text{... (1)}$$

$$\sigma_{H0} = Z_H \, Z_E \, Z_\epsilon \, Z_\beta \sqrt{\frac{F_t}{d_1 \, b} \, \frac{u+1}{u}} \quad \text{... (2)}$$

where

σ_{H0} is the nominal contact stress at the pitch point; this 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 (see 5.2). This converts contact stress at the pitch point to the contact stress at the inner point of single pair tooth contact on the pinion.

K_A is the application factor (see ISO 6336-1). It 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). It 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). It 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). It takes into account uneven load distribution in the transverse direction resulting, for example, from pitch deviation.

NOTE 2 – See ISO 6336-1, subclause 4.1.10 for the sequence in which factors K_A, K_v, $K_{H\beta}$, $K_{H\alpha}$ are calculated.

σ_{HP} is the permissible contact stress (see 4.2).

Z_H is the zone factor (see clause 5). It 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 6). It 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 7). It takes into account the influence of the effective length of the lines of contact.

Z_β is the helix angle factor (see clause 8). It 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. The total tangential load per mesh shall be introduced for F_t in every case (even with $\epsilon_{\alpha n} > 2$). See ISO 6336-1, subclause 4.2, for the definition of F_t and comments on particular characteristics of double-helical gearing.

b is the facewidth (for a double helix gear $b = 2\, b_B$). 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.

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.

4.1.2 Contact stress for the wheel

$$\sigma_H = Z_D\, \sigma_{H0} \sqrt{K_A\, K_v\, K_{H\beta}\, K_{H\alpha}} \le \sigma_{HP} \qquad \dots (3)$$

where

Z_D is the single pair tooth contact factor of the wheel (see 5.2). This transforms contact stress at the pitch point to contact stress at the inner point of single pair tooth contact of the wheel.

See 4.1.1 for explanations of other symbols.

4.2 Permissible contact stress, σ_{HP}

The limit values of contact stresses (see clause 9) 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.

4.2.1 Determination of the permissible contact stress, σ_{HP}, principles, assumptions and application

a) 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 are calculated using equation (2) or (3) 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.

b) 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 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).

c) Methods C and D

In these methods which are derived from method B, the influence factors Z_L, Z_v, Z_R, Z_W and Z_X are determined using simplified procedures.

d) 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.

4.2.2 Permissible contact stress, σ_{HP}, Method B

$$\sigma_{HP} = \frac{\sigma_{H\,lim} Z_{NT}}{S_{H\,min}} Z_L Z_v Z_R Z_W Z_X = \frac{\sigma_{HG}}{S_{H\,min}} \quad \text{... (4)}$$

where

$\sigma_{H\,lim}$ is the allowable stress number (contact) (see clause 9 and ISO 6336-5). It accounts for the influence of material, heat treatment and surface roughness of the standard reference test gears.

Z_{NT} is the life factor for contact stress (see clause 10). It 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.

Factors Z_L, Z_R and Z_v together cover the influence of the oil film on tooth contact stress.

Z_L is the lubricant factor (see clause 11). It accounts for the influence of the lubricant viscosity.

Z_R is the roughness factor (see clause 11). It accounts for the influence of surface roughness.

Z_v is the velocity factor (see clause 11). It accounts for the influence of pitch line velocity.

Z_W is the work hardening factor (see clause 12). It 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 13). It accounts for the influence of the tooth dimensions for the permissible contact stress.

a) Permissible contact stress (reference)

The permissible contact stress (reference), $\sigma_{HP\,ref}$, is derived from equation (4), 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 following method B.

b) Permissible contact stress (static)

The permissible contact stress (static), $\sigma_{HP\,stat}$, is determined in accordance with equation (4) with all method B influence factors (for static stress).

4.2.3 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 interpolation between the value obtained for reference in accordance with 4.2.2 a) and the value obtained for static stress in accordance with 4.2.2 b). Values appropriate to the relevant number of load cycles N_L are indicated by the S-N curve. See clause 10.

4.2.3.1 Graphical values

Calculate σ_{HP} for reference stress and static stress in accordance with 4.2.2 and plot the S-N curve corresponding to the life factor Z_{NT}. See figure 1 for the principle. σ_{HP} for the relevant number of load cycles N_L may be read from this graph.