# INTERNATIONAL STANDARD

# ISO 6336-3

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## Calculation of load capacity of spur and helical gears —

## Part 3:
Calculation of tooth bending strength

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

*Partie 3: Calcul de la résistance à la flexion des dents*

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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 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 6336-3 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 of this part of ISO 6336 is for information only.

## Introduction

The maximum tensile stress at the tooth-root (in the direction of the tooth height) which may not exceed the permissible bending stress for the material, is the basis for rating the bending strength of gear teeth. The stress occurs in the "tension fillets" of the working tooth flanks. If load-induced cracks are formed, the first of these often appears in the fillets where the compressive stress is generated; i.e. in the "compression fillets", which are those of the non-working flanks. When the tooth loading is unidirectional and the teeth are of conventional shape, these cracks seldom propagate to failure. Crack propagation ending in failure is most likely to stem from cracks initiated in tension fillets.

The endurable tooth loading of teeth which are subjected to a reversal of loading during each revolution, such as "idler gears", is less than the endurable unidirectional loading. The full range of stress, in such circumstances, is more than twice the tensile stress which occurs in the root fillets of the loaded flanks. This is taken into consideration when determining permissible stresses (see ISO 6336-5).

When gear rims are thin and tooth spaces adjacent to the root surface are narrow (conditions which can apply in particular to some internal gears), initial cracks commonly occur in the compression fillet. Since, in such circumstances, gear rims themselves can suffer fatigue breakage, special studies are necessary. See clause 1, 5.2.2 and 5.3.2.

Several methods for calculation of the critical tooth-root stress and for evaluating some of the relevant factors have been approved (see ISO 6336-1).

# Calculation of load capacity of spur and helical gears —

## Part 3: Calculation of tooth bending strength

## 1 Scope

This part of ISO 6336 specifies the fundamental formulae for use in tooth bending stress calculations for involute internal and external spur and helical gears with a minimum rim thickness under the root of $s_R \leq 3{,}5\ m_n$. All load influences on tooth stress are included insofar as they are the result of loads transmitted by the gearing and insofar as they can be evaluated quantitatively (see 4.1.1).

The given formulae are valid for spur and helical gears with tooth profiles in accordance with the basic rack standardized in ISO 53 (see introduction). They may also be used for teeth conjugate to other basic racks if the virtual contact ratio is less than $\epsilon_{\alpha n} = 2{,}5$.

NOTE 1 – See 4.1.1 c) and 5.3 for restrictions in the case of method C.

The load capacity determined on the basis of permissible bending stress is termed "tooth bending strength". The results are in good agreement with other methods for the range as indicated in ISO 6336-1.

The user of this part of ISO 6336 standard 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.

## 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 Tooth breakage and safety factors

Tooth breakage usually ends the service life of a transmission. Sometimes the destruction of all gears in a transmission can be a consequence of the breakage of one tooth. In some instances the transmission path between input and output shafts is broken. As a consequence, the chosen value of the safety factor $S_F$ against tooth breakage should be larger than the safety factor against pitting.

General comments on the choice of the minimum safety factor can be found in ISO 6336-1, subclause 4.1.3. It is recommended that manufacturer and customer agree on the value of the minimum safety factor.

This part of ISO 6336 does not apply at stress levels above those permissible for $10^3$ cycles, since stresses in this range may exceed the elastic limit of the gear tooth.

## 4 Basic formulae

NOTE 2 – All symbols, terms and units are defined in ISO 6336-1.

The actual tooth-root stress $\sigma_F$ and the permissible bending stress $\sigma_{FP}$ shall be calculated separately for pinion and wheel; $\sigma_F$ shall be less than $\sigma_{FP}$.

### 4.1 Tooth-root stress, $\sigma_F$

#### 4.1.1 Methods for the determination of tooth-root stress, $\sigma_F$: Principles, assumptions, and application

According to this part of ISO 6336, the local tooth-root stress is determined as the product of nominal bending stress and a stress correction factor (methods B and C[1)]).

**a) Method A**

In principle, the maximum tensile stress can be determined by any appropriate method (e.g. finite element analysis, integral equations, conformal mapping procedures, or experimentally by photo-elastic stress analysis, strain measurement, etc.). In order to determine the maximum tooth-root stress, the effects of load distribution over two or more engaging teeth and changes of stress with changes of meshing phase shall be taken into consideration.

It should be noted that the tooth-root tensile stress has relevance to the plane-strain condition. This is important when making comparisons with the results of photo-elastic stress evaluations (methods B and C) and the permissible stresses.

Method A is only used in special cases and, because of the great effort involved, is only justifiable in such cases.

**b) Method B**

This method involves the assumption that the determinant tooth-root stress occurs with application of load at the outer point of single pair tooth contact of spur gears or of the virtual spur gears of helical gears. However, in the latter case, the "transverse load" shall be replaced by the "normal load", applied over the facewidth of the actual gear of interest.

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1) Stresses such as those caused by the shrink-fitting of gear rims, which are superimposed on stresses due to tooth loading, should be taken into consideration in the calculation of the tooth root stress $\sigma_F$ or the permissible tooth root stress $\sigma_{FP}$.

For gears having virtual contact ratios in the range $2 \le \epsilon_{\alpha n} < 3$, it is assumed that the determinant stress occurs with application of load at the inner point of double pair tooth contact. Formulae are provided for the calculation of the appropriate form factors $Y_\beta$ for the nominal stress and $Y_S$ for the stress correction factors. In the case of helical gears, the factor $Y_\beta$ accounts for deviations from these assumptions.

Method B is suitable for more detailed calculations and is also appropriate for computer programming and for the analysis of pulsator tests (with a given point of application of loading).

**c) Method C**

This simplified method of calculation is derived from method B. The local stress for application of load at the tooth tip is calculated first (with factors $Y_{Fa}$ and $Y_{Sa}$) and then converted to approximate the corresponding value, appropriate to contact at the outer point of single pair tooth contact, using the factor $Y_\epsilon$.

The form factor $Y_{Fa}$ for the nominal stress and the stress correction factor $Y_{Sa}$ have been plotted on a series of graphs for a number of basic rack profiles.

Method C is only acceptable for gears when $\epsilon_{\alpha n} < 2$; it is also useful when no computer program is available. The method is sufficiently accurate for most cases and generally gives slightly higher values of stress than method B.

### 4.1.2 Tooth-root stress, $\sigma_F$: Methods B and C

The total tangential load in the case of gear trains with multiple transmission paths (planetary gear trains, 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 $K_\gamma$ to follow $K_A$ in equation (1), to adjust as necessary the average load per mesh.

$$\sigma_F = \sigma_{F0} K_A K_v K_{F\beta} K_{F\alpha} \le \sigma_{FP} \qquad \text{... (1)}$$

where

$\sigma_{F0}$ is the nominal tooth-root stress, which is the maximum local tensile stress produced at the tooth-root when an error-free gear pair is loaded by the static nominal torque.

$\sigma_{FP}$ is the permissible bending stress (see 4.2).

$K_A$ is the application factor (see ISO 6336-1). It takes into account load increments 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.

$K_{F\beta}$ is the face load factor for tooth-root 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.

$K_{F\alpha}$ is the transverse load factor for tooth-root stress (see ISO 6336-1). It takes into account uneven load distribution in the transverse direction, resulting for example, from pitch deviations.

NOTE 3 – See ISO 6336-1, subclause 4.1.8 for the sequence in which factors $K_A$, $K_v$, $K_{F\beta}$ and $K_{F\alpha}$ are calculated.