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Calculation of load capacity of spur and helical gears — Application for industrial gears

Calcul de la capacité de charge des engrenages à denture droite et hélicoïdale — Application aux engrenages industriels

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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 3.

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

Attention is drawn to the possibility that some of the elements of this International Standard may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 9085 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

Annexes A and B form a normative part of this International Standard. Annexes C and D are for information only.

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Introduction

Procedures for the calculation of the load capacity of general spur and helical gears with respect to pitting and bending strength appear in ISO 6336-1, ISO 6336-2, ISO 6336-3 and ISO 6336-5. This International Standard is derived from ISO 6336-1, ISO 6336-2 and ISO 6336-3 by the use of specific methods and assumptions which are considered to be applicable to industrial gears. Its application requires the use of allowable stresses and material requirements which are to be found in ISO 6336-5.

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Calculation of load capacity of spur and helical gears — Application for industrial gears

1 Scope

The formulae specified in this International Standard are intended to establish a uniformly acceptable method for calculating the pitting resistance and bending strength capacity of industrial gears with spur or helical teeth.

The rating formulae in this International Standard are not applicable to other types of gear tooth deterioration such as plastic yielding, micropitting, scuffing, case crushing, welding and wear, and are not applicable under vibratory conditions where there may be an unpredictable profile breakdown. The bending strength formulae are applicable to fractures at the tooth fillet, but are not applicable to fractures on the tooth working profile surfaces, failure of the gear rim, or failures of the gear blank through web and hub. This International Standard does not apply to teeth finished by forging or sintering. It is not applicable to gears which have a poor contact pattern.

This International Standard provides a method by which different gear designs can be compared. It is not intended to assure the performance of assembled drive gear systems. Neither is it intended for use by the general engineering public. Instead, it is intended for use by the experienced gear designer who is capable of selecting reasonable values for the factors in these formulae based on knowledge of similar designs and awareness of the effects of the items discussed.

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CAUTION — The user is cautioned that the calculated results of this International Standard should be confirmed by experience.

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2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this International Standard. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 53:1998, *Cylindrical gears for general and heavy engineering — Standard basic rack tooth profile*

ISO 54:1996, *Cylindrical gears for general and heavy engineering — Modules*

ISO 1122-1:1998, *Vocabulary of gear terms — Part 1: Definitions related to geometry*

ISO 1328-1:1995, *Cylindrical gears — ISO system of accuracy — Part 1: Definitions and allowable values of deviations relevant to corresponding flanks of gear teeth*[1)]

ISO 4287:1997, *Geometrical Product Specifications (GPS) — Surface texture: Profile method — Terms, definitions and surface texture parameters*

1) This was corrected and reprinted in 1997.

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

ISO 6336-2:1996, *Calculation of load capacity of spur and helical gears — Part 2: Calculation of surface durability (pitting)*

ISO 6336-3:1996, *Calculation of load capacity of spur and helical gears — Part 3: Calculation of tooth bending strength*

ISO 6336-5:1996, *Calculation of load capacity of spur and helical gears — Part 5: Strength and quality of materials*

ISO 9084:2000, *Calculation of load capacity of spur and helical gears — Application to high speed gears and gears of similar requirements*

ISO/TR 10495:1997, *Cylindrical gears — Calculation of service life under variable loads — Conditions for cylindrical gears accordance with ISO 6336*

ISO/TR 13593:1999, *Enclosed gear drives for industrial applications*

3 Terms and definitions

For the purposes of this International Standard, the terms and definitions given in ISO 1122-1 apply. For the symbols, see Table 1.

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Table 1 — Symbols and abbreviations used in this International Standard

Symbol	Description or term	Unit
a	centre distance[a]	mm
b	facewidth	mm
b_{B}	facewidth of an individual helix of a double helical gear	mm
b_{H}	facewidth (pitting)	mm
b_{F}	facewidth (tooth root)	mm
b_{red}	reduced facewidth (facewidth minus end reliefs)	mm
b_{s}	web thickness	mm
$b_{\mathrm{I(II)}}$	length of end relief	mm
c_{γ}	mean value of mesh stiffness per unit facewidth	N/(mm·µm)
c'	maximum tooth stiffness of one pair of teeth per unit facewidth (single stiffness)	N/(mm·µm)
$d_{\mathrm{a}1,2}$	tip diameter of pinion (or wheel)	mm
$d_{\mathrm{an}1,2}$	tip diameter of pinion (or wheel) of virtual spur gear	mm
$d_{\mathrm{b}1,2}$	base diameter of pinion (or wheel)	mm
$d_{\mathrm{bn}1,2}$	base diameter of pinion (or wheel) of virtual spur gear	mm
$d_{\mathrm{en}1,2}$	diameter of circle through outer point of single pair tooth contact of pinion, wheel of virtual spur gear	mm
$d_{\mathrm{f}1,2}$	root diameter of pinion, wheel	mm
$d_{\mathrm{m}1,2}$	diameter at mid-tooth depth of pinion, wheel	mm
$d_{\mathrm{n}1,2}$	reference diameter of pinion, wheel of virtual spur gear	mm
d_{sh}	nominal shaft diameter for bending	mm
d_{shi}	internal diameter of hollow shaft	mm
$d_{\mathrm{w}1,2}$	pitch diameter of pinion, wheel	mm
$d_{\mathrm{Nf}2}$	diameter of a circle near the tooth-roots, containing the limits of the usable flanks of an internal gear or the larger external gear of a mating gear	mm
$d_{1,2}$	reference diameter of pinion, wheel	mm
$f_{\mathrm{f\ eff}}$	effective profile form deviation	µm
$f_{\mathrm{f}\alpha}$	profile form deviation (the value for the total profile deviation F_{α} may be used alternatively for this, if tolerances complying with ISO 1328-1 are used)	µm
f_{ma}	helix deviation due to manufacturing inaccuracies	µm
f_{pb}	transverse base pitch deviation (the values of f_{pt} may be used for calculations in accordance with ISO 6336:1996, using tolerances complying with ISO 1328-1)	µm
$f_{\mathrm{pb\ eff}}$	effective transverse base pitch deviation	µm
f_{sh}	helix deviation due to elastic deflections	µm
$f_{\mathrm{H}\beta}$	tooth alignment deviation (not including helix form deviation)	µm
g_{α}	path length of contact	mm
h	tooth depth	mm

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Table 1 (*continued*)

Symbol	Description or term	Unit
h_a	addendum	mm
h_{a0}	tool addendum	mm
h_{f2}	dedendum of tooth of an internal gear	mm
h_{fP}	dedendum of basic rack of cylindrical gears	mm
h_{Fe}	bending moment arm for load application at the outer point of single pair tooth contact	mm
h_{Nf2}	dedendum of tooth of an internal gear, containing the limits of the usable flanks of an internal gear or the larger external gear of a mating gear	mm
l	bearing span	mm
m_n	normal module	mm
m_{red}	reduced gear pair mass per unit facewidth referenced to the line of action	kg/mm
n_E	resonance speed	min^{-1}
$n_{1,2}$	rotation speed of pinion, wheel	min^{-1}
p_{bn}	normal base pitch	mm
p_{bt}	transverse base pitch	mm
pr	protuberance of the tool	mm
q	finishing stock allowance	mm
q_s	notch parameter s_{Fn} / $2\rho_F$	—
q_{sT}	notch parameter of standard reference test gear	—
r_b	base radius	mm
s	pinion offset from shaft centre line	mm
s_{Fn}	tooth-root chord at the critical section	mm
s_R	rim thickness	mm
s_{pr}	residual fillet undercut	mm
u	gear ratio $\mid u \mid = \mid z_2/z_1 \mid \geqslant 1$ [a]	—
v	circumferential speed (without subscript: at reference circle ≈ circumferential speed at working pitch circle)	m/s
$x_{1,2}$	profile shift coefficient of pinion, wheel	—
y_f	running-in allowance (pitch deviation)	µm
y_p	running-in allowance (profile deviation)	µm
y_α	running-in allowance for a gear pair	µm
y_β	running-in allowance (equivalent misalignment)	µm
z_n	virtual number of teeth of a helical gear	—
$z_{1,2}$	number of teeth of pinion, wheel [a]	—
B	total facewidth of a double helical gear including the gap	mm
B_f	running-in parameter for determination of constant K	—
B_k	running-in parameter for determination of constant K	—

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Table 1 (*continued*)

Symbol	Description or term	Unit
B_p	running-in parameter for determination of constant K	—
$B_{1,2}$	constants for determination of $F_{\beta x}$	—
B^*	constant for determination of the pinion offset	—
C_a	tip relief	µm
C_{ay}	tip relief resulting from running-in	µm
$C_{v1,2,3}$	constants for determination of constant K	—
C_B	basic rack factor	—
C_R	gear blank factor	—
C_β	crowning height	µm
$C_{1...9}$	constants for determination of q_s	—
E	modulus of elasticity, Young's modulus	N/mm^2
E	auxiliary value for calculation of Y_F	—
F_m	mean transverse force at the reference cylinder (= $F_t\, K_A\, K_v$)	N
F_t	(nominal) transverse tangential force at reference cylinder	N
$F_{t\,max}$	maximum transverse tangential force at reference cylinder	N
F_{tH}	determinant transverse force at the reference cylinder (= $F_t\, K_A\, K_v\, K_{H\beta}$)	N
F_β	total helix deviation	µm
$F_{\beta x}$	initial equivalent misalignment (before running-in)	µm
G	auxiliary value for calculation of Y_F	—
H	auxiliary value for calculation of Y_F	—
$J^*_{1,2}$	polar moment of inertia per unit face width	Kg/mm
K	constant for determination of K_v	—
K_v	dynamic factor	—
K_A	application factor	—
$K_{F\alpha}$	transverse load factor (root stress)	—
$K_{F\beta}$	face load factor (root stress)	—
$K_{H\alpha}$	transverse load factor (contact stress)	—
$K_{H\beta}$	face load factor (contact stress)	—
K_γ	mesh load factor (takes into account the uneven distribution of the load between meshes for multiple transmission paths)	—
$K_{1,2}$	constant	—
K'	constant for the pinion offset in relation to the torqued end	—
L	tooth root chord at the critical section, related to the bending moment arm relevant to load application at the outer point of single pair tooth contact	—
N	resonance ratio	—
N_F	exponent	—

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Table 1 (*continued*)

Symbol	Description or term	Unit
N_L	number of load cycles	—
N_S	resonance ratio in the main resonance range	—
$M_{1,2}$	auxiliary values for the determination of $Z_{B,D}$	—
P	transmitted power	kW
P_{max}	maximum transmitted power	kW
Ra	arithmetic mean roughness value (as specified in ISO 4287:1997)	µm
Rz	mean peak-to-valley roughness (as specified in ISO 4287:1997)	µm
Rz_{10}	mean peak-to-valley roughness for the gear pair	µm
S_F	safety factor from tooth breakage	—
$S_{F\ min}$	minimum safety factor (tooth breakage)	—
S_H	safety factor from pitting	—
$S_{H\ min}$	minimum safety factor (pitting)	—
$T_{1,2}$	pinion torque (nominal); wheel torque	Nm
T_{max}	maximum torque	Nm
Y_F	tooth form factor	—
Y_N	life factor for tooth-root stress	—
Y_{NT}	life factor for tooth-root stress for reference test conditions	—
$Y_{R\ rel\ T}$	surface factor	—
Y_S	stress correction factor	—
Y_X	size factor (tooth root)	—
Y_β	helix angle factor (tooth root)	—
$Y_{\delta\ rel\ T}$	relative notch sensitivity factor	—
Y_ε	contact ratio factor (tooth root)	—
Z_v	speed factor	—
$Z_{B,D}$	single pair tooth contact factors for the pinion, wheel	—
Z_E	elasticity factor	$\sqrt{N/mm^2}$
Z_H	zone factor	—
Z_L	lubricant factor	—
Z_N	life factor for contact stress	—
Z_{NT}	life factor for contact stress for reference test conditions	—
Z_R	roughness factor affecting surface durability	—
Z_W	work-hardening factor	—
Z_X	size factor (pitting)	—
Z_β	helix angle factor (pitting)	—
Z_ε	contact ratio factor (pitting)	—

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Table 1 (*continued*)

Symbol	Description or term	Unit
α_{en}	pressure angle at the outer point of single pair tooth contact of virtual spur gears	°
α_n	normal pressure angle	°
α_t	transverse pressure angle	°
α_{wt}	transverse pressure angle at the pitch cylinder	°
α_{Fen}	load direction angle, relevant to direction of application of load at the outer single pair tooth contact of virtual spur gears	°
α_{Pn}	normal pressure angle of the basic rack for cylindrical gears	°
β	helix angle at the reference cylinder	°
β_b	base helix angle	°
γ_e	auxiliary angle for determination of α_{Fen}	°
δ_{bth}	combined deflection of mating teeth assuming even load distribution over the facewidth	µm
ε_α	transverse contact ratio	—
$\varepsilon_{\alpha n}$	transverse contact ratio of a virtual spur gear	—
ε_β	axial overlap ratio	—
ε_γ	total contact ratio ($\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$)	—
ν	Poisson's contact ratio	—
θ	auxiliary value for calculation of Y_F	—
ρ_{a0}	tip radius of the tool	mm
ρ_{fP}	root fillet radius of the basic rack for cylindrical gears	mm
ρ_{rel}	radius of relative curvature	mm
ρ_F	tooth-root fillet radius at the critical section	mm
ρ'	slip-layer thickness	mm
σ_B	tensile strength	N/mm^2
σ_F	tooth-root stress	N/mm^2
$\sigma_{F\,lim}$	nominal stress number (bending)	N/mm^2
σ_{FE}	allowable stress number (bending) = $\sigma_{F\,lim}\, Y_{ST}$	N/mm^2
σ_{FG}	tooth-root stress limit	N/mm^2
σ_{FP}	permissible tooth-root stress	N/mm^2
σ_{F0}	nominal tooth-root stress	N/mm^2
σ_H	calculated contact stress	N/mm^2
$\sigma_{H\,lim}$	allowable stress number (contact)	N/mm^2
σ_{HG}	modified allowable stress number = $\sigma_{HP}\, S_{H\,min}$	N/mm^2
σ_{HP}	permissible contact stress	N/mm^2
σ_{H0}	nominal contact stress	N/mm^2

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Table 1 (*continued*)

Symbol	Description or term	Unit
σ_S	yield point	N/mm^2
$\sigma_{0,2}$	0,2 % proof stress	N/mm^2
χ^*	relative stress gradient in the root of a notch	mm^{-1}
χ^*_p	relative stress gradient in a smooth polished test piece	mm^{-1}
χ^*_T	relative stress gradient in the root of the standard reference test gear	mm^{-1}
$\omega_{1,2}$	angular velocity of pinion, wheel	rad/s

[a] For external gear pairs, a, u, z_1 and z_2 are positive; for internal gear pairs, a, u and z_2 are negative, and z_1 positive.

4 Application

4.1 Design, specific applications

4.1.1 General

Gear designers must recognize that requirements for different applications vary considerably. Use of the procedures of this International Standard for specific applications demands a careful appraisal of all applicable considerations, in particular:

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— the allowable stress of the material and the number of load repetitions;

— the consequences of any percentage of failure (failure rate);

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— the appropriate factor of safety.

Design considerations to prevent fractures emanating from stress raisers in the tooth flank, tip chipping and failures of the gear blank through the web or hub should be analysed by general machine design methods.

Any variances according to the following shall be reported in the calculation statement.

a) If a more refined method of calculation is desired or if compliance with the restrictions given in 4.1 is for any reason impractical, relevant factors may be evaluated according to the basic standard or another application standard.

b) Factors derived from reliable experience or test data may be used instead of individual factors according to this International Standard. Concerning this, the criteria for Method A in 4.1.8.1 of ISO 6336-1:1996 are applicable.

In other respects, rating calculations shall be strictly in accordance with this International Standard wherever stresses, safety factors etc. are to be classified as being in accordance with this International Standard.

This International Standard recognizes the following types of industrial drive design.

— **Catalogue enclosed drives** are designed to nominal load ratings for sale from catalogues or from stock. The actual loads and operation conditions are not exactly known at the time of design.

NOTE The actual loads for each application are evaluated to select an appropriately sized unit from the catalogue. A selection factor, based on experience with similar applications, is often used to reduce the catalogue rating to match the application conditions (see ISO TR 13593).

— **Custom designed drives** are aimed at a specific application where the operating conditions are known or specified at the time of design.

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This International Standard is applicable when the wheel blank, shaft/hub connections, shafts, bearings, housings, threaded connections, foundations and couplings conform to the requirements regarding accuracy, load capacity and stiffness which form the basis for the calculation of the load capacity of gears.

Although the method described in this International Standard is mainly intended for recalculation purposes, by means of iteration it can also be used to determine the load capacities of gears. The iteration is accomplished by selecting a load and calculating the corresponding safety factor against pitting, S_{H1}, for the pinion. If S_{H1} is greater than $S_{H\,min}$, the load is increased; if it is smaller than $S_{H\,min}$, the load is reduced. This is done until the chosen load corresponds to $S_{H1} = S_{H\,min}$. The same method is used for the wheel ($S_{H2} = S_{H\,min}$), and also for the safety factors against tooth breakage, $S_{F1} = S_{F2} = S_{F\,min}$.

4.1.2 Gear data

This International Standard is applicable within the following constraints.

a) Types of gear:

— external and internal, involute spur, helical and double helical gears;

— for double helical gears, it is assumed that the total tangential load is evenly distributed between the two helices; if this is not the case (e.g. due to externally applied axial forces), this shall be taken into account; the two helices are treated as two single helical gears in parallel.

b) Range of speeds:

— n_1 less than or equal to 3 600 min^{-1} (synchronous speed of two-pole motor at 60 Hz current frequency)[2];

— subcritical range of speed (see K_v in 5.6);

— at speeds of $v < 1$ m/s, gear load capacity is often limited by wear.

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c) Gear accuracy:

— accuracy grade 10 or better according to ISO 1328-1 (affects K_v, $K_{H\alpha}$ and $K_{H\beta}$).

d) Range of the transverse contact ratios of virtual spur gear pairs:

— $1{,}2 < \varepsilon_\alpha < 1{,}9$ (affects c', c_γ, K_v, $K_{H\beta}$, $K_{F\alpha}$, $K_{H\alpha}$ and $K_{F\beta}$).

e) Range of helix angles:

— β less than or equal to 30° (affects c', c_γ, K_v and $K_{H\beta}$).

4.1.3 Pinion and pinion shaft

This International Standard is applicable to pinions integral with shafts or bored pinions with $s_R/d_1 \geqslant 0{,}2$ (this affects c', c_γ, K_v, $K_{H\beta}$). It is assumed that the bored pinions will be mounted on solid shafts or on hollow shafts with $d_{shi}/d_{sh} < 0{,}5$ (this affects $K_{H\beta}$).

4.1.4 Wheel blank, wheel rim

The given formulae are valid for spur and helical gears with a minimum rim thickness under the root of $s_R \geqslant 3{,}5\, m_n$. The calculation of $K_{H\beta}$ assumes that wheel and wheel shaft are sufficiently stiff such that their deflections can be ignored.

2) For higher speeds, the requirements of ISO 6336 or ISO 9084 apply.

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