

Calculation of load capacity of spur and helical gears —

Part 1: Basic principles, introduction and general influence factors

ICS 21.200

National foreword

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Calculation of load capacity of spur and helical gears —

Part 1:

Basic principles, introduction and general influence factors

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

*Partie 1: Principes de base, introduction et facteurs généraux
d'influence*



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Documents

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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 6336-1 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 6336-1:1996), Clauses 6, 7 and 9 of which have been technically revised. It also incorporates the Amendments ISO 6336-1:1996/Cor.1:1998 and ISO 6336-1:1996/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*

Introduction

This and the other parts of ISO 6336 provide a coherent system of procedures for the calculation of the load capacity of cylindrical involute gears with external or internal teeth. ISO 6336 is designed to facilitate the application of future knowledge and developments, also the exchange of information gained from experience.

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 will need to be analyzed by general machine design methods.

Several methods for the calculation of load capacity, as well as for the calculation of various factors, are permitted (see 4.1.12). The directions in ISO 6336 are thus complex, but also flexible.

Included in the formulae are the major factors which are presently known to affect gear tooth pitting and fractures at the root fillet. The formulae are in a form that will permit the addition of new factors to reflect knowledge gained in the future.

Calculation of load capacity of spur and helical gears —

Part 1:

Basic principles, introduction and general influence factors

1 Scope

This part of ISO 6336 presents the basic principles of, an introduction to, and the general influence factors for, the calculation of the load capacity of spur and helical gears. Together with ISO 6336-2, ISO 6336-3, ISO 6336-5 and ISO 6336-6, it provides a method by which different gear designs can be compared. It is not intended to assure the performance of assembled drive gear systems. It is not 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.

The formulae in ISO 6336 are intended to establish a uniformly acceptable method for calculating the pitting resistance and bending strength capacity of cylindrical gears with straight or helical involute teeth.

ISO 6336 includes procedures based on testing and theoretical studies such as those of Hirt [1], Strasser [2] and Brossmann [3]. The results of rating calculations made by following this method are in good agreement with previously accepted gear calculations methods (see References [4] to [8]) for normal working pressure angles up to 25° and reference helix angles up to 25° .

For larger pressure angles and larger helix angles, the trends of products $Y_F Y_S Y_\beta$ and, respectively, $Z_H Z_\epsilon Z_\beta$ are not the same as those of some earlier methods. The user of ISO 6336 is cautioned that when the methods in ISO 6336 are used for other helix angles and pressure angles, the calculated results will need to be confirmed by experience.

The formulae in ISO 6336 are not applicable when any of the following conditions exist:

- spur or helical gears with transverse contact ratios less than 1,0;
- spur or helical gears with transverse contact ratios greater than 2,5;
- interference between tooth tips and root fillets;
- teeth are pointed;
- backlash is zero.

The rating formulae in ISO 6336 are not applicable to other types of gear tooth deterioration such as plastic yielding, 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 surfaces, failure of the gear rim, or failures of the gear blank through web and hub. ISO 6336 does not apply to teeth finished by forging or sintering. It is not applicable to gears which have a poor contact pattern.

The procedures in ISO 6336 provide rating formulae for the calculation of load capacity, based on pitting and tooth root breakage. At pitch line velocities below 1 m/s the gear load capacity is often limited by abrasive wear (see other literature for information on the calculation for this).

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

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

ISO 4288:1996, *Geometrical Product Specifications (GPS) — Surface texture: Profile method — Rules and procedures for the assessment of surface texture*

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

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

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

ISO 6336-6, *Calculation of load capacity of spur and helical gears — Part 6: Calculation of service life under variable load*

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 the following symbols apply.

NOTE Symbols are based on, and are extensions of, the symbols given in ISO 701 and ISO 1328-1. Only symbols for quantities used for the calculation of the particular factors treated in ISO 6336 are given, together with the preferred units.

Table 1 — Symbols used in ISO 6336-1, ISO 6336-2, ISO 6336-3 and ISO 6336-5

Symbol	Description	Unit
Principal symbols and abbreviations		
A, B, C, D, E	points on path of contact (pinion root to pinion tip, regardless of whether pinion or wheel drives, only for geometrical considerations)	—
a	centre distance ^a	mm
α	pressure angle (without subscript, at reference cylinder)	°
B	total face width of double helical gear including gap width	mm
b	face width	mm
β	helix angle (without subscript, at reference cylinder)	°
C	constant, coefficient	—
	relief of tooth flank	µm
c	constant	—
γ	auxiliary angle	°
D	diameter (design)	mm
d	diameter (without subscript, reference diameter)	mm
δ	deflection	µm
E	modulus of elasticity	N/mm ²
Eh	material designation for case-hardened wrought steel	—
Eht	case depth, see ISO 6336-5	mm
e	auxiliary quantity	—
ε	contact ratio, overlap ratio, relative eccentricity (see Clause 7)	—
ζ	roll angle	°
F	composite and cumulative deviations	µm
	force or load	N
f	deviation, tooth deformation	µm
G	shear modulus	N/mm ²
GG	material designation for grey cast iron	—
GGG	material designation for nodular cast iron (perlitic, bainitic, ferritic structure)	—
GTS	material designation for black malleable cast iron (perlitic structure)	—
g	path of contact	mm
ϑ	temperature	°C
HB	Brinell hardness	—
HRC	Rockwell hardness (C scale)	—
HR 30N	Rockwell hardness (30 N scale) (see ISO 6336-5)	—
HV	Vickers hardness	—
HV 1	Vickers hardness at load $F = 9,81$ N (see ISO 6336-5)	—
HV 10	Vickers hardness at load $F = 98,10$ N (see ISO 6336-5)	—
h	tooth depth (without subscript, root circle to tip circle)	mm
η	effective dynamic viscosity of the oil wedge at the mean temperature of wedge	mPa s
IF	material designation for flame or induction hardened wrought special steel	—
i	transmission ratio	—
	bin	—
J	Jominy hardenability (see ISO 6336-5)	—
K	constant, factors concerning tooth load	—

Table 1 (continued)

Symbol	Description	Unit
L	lengths (design)	mm
l	bearing span	mm
Γ	parameter on the line of action	—
M	moment of a force	Nm
	mean stress ratio	—
ME	symbols identifying material and heat-treatment requirements (see ISO 6336-5)	—
MQ		—
ML		—
m	module	mm
	mass	kg
μ	coefficient of friction	—
N	number, exponent, resonance ratio	—
NT	material designation for nitrided wrought steel, nitriding steel	—
NV	material designation for through-hardened wrought steel, nitrided, nitrocarburized	—
n	rotational speed	s^{-1} or min^{-1}
	number of load cycles	
ν	Poisson's ratio	—
	kinematic viscosity of the oil	mm^2/s
P	transmitted power	kW
p	pitch	mm
	number of planet gears	—
	slope of the Woehler-damage line	—
q	auxiliary factor	—
	flexibility of pair of meshing teeth (see Clause 9)	$(mm \cdot \mu m)/N$
	material allowance for finish machining (see ISO 6336-3)	mm
r	radius (without subscript, reference radius)	mm
ρ	radius of curvature	mm
	density (for steel, $\rho = 7,83 \times 10^{-6}$)	kg/mm^3
S	safety factor	—
St	material designation for normalized base steel ($\sigma_B < 800 N/mm^2$)	—
s	tooth thickness, distance between mid-plane of pinion and the middle of the bearing span	mm
σ	normal stress	N/mm^2
T	torque	Nm
	tolerance	μm
τ	shear stress	N/mm^2
	angular pitch	mm
u	gear ratio ($z_2 / z_1 \geq 1^a$)	—
U	Miner sum	—
V	material designation for through-hardened wrought special steel, alloy or carbon ($\sigma_B \geq 800 N/mm^2$)	—
v	tangential velocity (without subscript, at the reference circle = tangential velocity at pitch circle)	m/s
w	specific load (per unit face width, F_t / b)	N/mm
ψ	auxiliary angle	°
x	profile shift coefficient	—

Table 1 (continued)

Symbol	Description	Unit
χ	running-in factor	—
Y	factor related to tooth root stress	—
y	running-in allowance (only with subscript α or β)	μm
Z	factor related to contact stress	—
z	number of teeth ^a	—
ω	angular velocity	rad/s
Subscripts to symbols		
—	reference values (without subscript)	
A	application	
	external shock loads	
a	addendum	
	tooth tip	
ann	annulus gear	
α	transverse contact	
	profile	
b	base circle	
	face width	
be	bearing	
β	helix	
	face width	
	crowning	
C	pitch point	
	profile and helix modification	
ca	case	
cal	calculated	
co	contact pattern	
γ	total (total value)	
D	speed transformation	
	reducing or increasing	
dyn	dynamic	
Δ	rough specimen	
E	elasticity of material	
	resonance	
e	outer limit of single pair tooth contact	
eff	effective value, real stress	
ε	contact ratio	
F	tooth root stress	
f	tooth root, dedendum	
G	geometry	
H	Hertzian stress (contact stress)	
i	internal	
	bin number	
k	values related to the notched test piece	

Table 1 (continued)

Symbol	Description
L	lubrication
lim	value of reference strength
M	mean stress influence
m	mean or average value (mean section)
ma	manufacturing
max	maximum value
min	minimum value
N	number (a specific number may be inserted after the subscript N in the life factor)
n	normal plane
	virtual spur gear of a helical gear
	number of revolutions
oil	oil
P	permissible value
	rack profile
p	pitch
	values related to the smooth polished test piece
par	parallel
pla	planet gear
R	roughness
r	radial
red	reduced
rel	relative
s	tooth thickness
	notch effect
sh	shaft
stat	static (load)
sun	sun pinion, sun gear
T	test gear
	values related to the standard reference test gear
t	transverse plane
th	theoretical
v	velocity
	losses
W	pairing of materials
w	working (this subscript may replace the prime)
X	dimension (absolute)
y	running-in
	any point on the tooth flank
z	sun
0	basic value
	tool
1	pinion
2	wheel

Table 1 (continued)

Symbol	Description	
1..9	general numbering	
I(II)	end relief	
	reference (nonreference) face	
'	single-flank (subscript w possible) single-pair tooth contact	
"	double-flank contact (simultaneous contact between working and non-working flanks)	
Combined symbols		Unit
α_{en}	form-factor pressure angle, 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'_t or α_{wt}	pressure angle at the pitch cylinder	°
α_{Fen}	load direction angle, relevant to direction of application of load at the outer point of single pair tooth contact of virtual spur gears	°
α_{Pn}	normal pressure angle of the basic rack for cylindrical gears	°
B^*	constant (see equations in Clause 7)	—
b_{cal}	calculated face width	mm
b_{c0}	length of tooth bearing pattern at no load (contact marking)	mm
b_{red}	reduced face width (face width minus end reliefs)	mm
b_s	web thickness	mm
b_B	face width of one helix on a double helical gear	mm
$b_{I(II)}$	length of end relief	mm
β_b	base helix angle	°
β_e	form-factor helix angle, helix angle at the outer point of single tooth contact	°
C_a	tip relief	µm
C_{ay}	tip relief by running-in	µm
C_B	basic rack factor (same rack for pinion and wheel)	—
C_{B1}	basic rack factor (pinion)	—
C_{B2}	basic rack factor (wheel)	—
C_M	correction factor (see Clause 9)	—
C_R	gear blank factor (see Clause 9)	—
$C_{ZL, ZR, Zv}$	factors for determining lubricant film factors (see ISO 6336-2)	—
C_β	crowning height	µm
$C_{I(II)}$	end relief	µm
c_γ	mean value of mesh stiffness per unit face width	N/(mm·µm)
$c_{\gamma\alpha}$	mean value of mesh stiffness per unit face width (used for $K_v, K_{H\alpha}, K_{F\alpha}$)	N/(mm·µm)
$c_{\gamma\beta}$	mean value of mesh stiffness per unit face width (used for $K_{H\beta}, K_{F\beta}$)	N/(mm·µm)
c'	maximum tooth stiffness per unit face width (single stiffness) of a tooth pair	N/(mm·µm)
c'_{th}	theoretical single stiffness	N/(mm·µm)
D_{be}	bearing bore diameter (plain bearings)	mm
D_{sh}	journal diameter (plain bearings)	mm
d_a	tip diameter	mm
d_b	base diameter	mm
d_e	diameter of circle through outer point of single pair tooth contact	mm

Table 1 (continued)

Symbol	Description	Unit
d_f	root diameter	mm
d_{f2}	root diameter of internal gear	mm
d_{Nf}	root form diameter	mm
d_{sh}	external diameter of shaft, nominal for bending deflection	mm
d_{shi}	internal diameter of a hollow shaft	mm
d_{soi}	diameter at start of involute	mm
d_w	pitch diameter	mm
$d_{1,2}$	reference diameter of pinion (or wheel)	mm
$\delta_{1,2}$	deformation of bearing (1, 2) in direction of load	μm , mm
$\delta_{b\ th}$	combined deflection of mating teeth assuming even load distribution over the face width	μm
δ_g	difference in feeler gauge thickness measurement of mesh misalignment f_{ma}	μm
δ_S	elongation on fracture	%
ε_α	transverse contact ratio	—
$\varepsilon_{\alpha n}$	virtual contact ratio, transverse contact ratio of a virtual spur gear	—
ε_β	overlap ratio	—
ε_γ	total contact ratio, $\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$	—
ε_1	addendum contact ratio of the pinion, $\varepsilon_1 = CE/p_{bt}$	—
ε_2	addendum contact ratio of the wheel, $\varepsilon_2 = AC/p_{bt}$	—
ζ_{aw}	roll angle from working pitch point to tip diameter	°
ζ_{fw}	roll angle from root form diameter to working pitch point	°
$F_{be\ r}$	radial force on bearing	N
F_{bn}	(nominal) load, normal to the line of contact	N
F_{bt}	nominal transverse load in plane of action (base tangent plane)	N
F_m	mean transverse tangential load at the reference circle relevant to mesh calculations, $F_m = (F_t K_A K_V)$	N
$F_{m\ T}$	mean transverse tangential part load at reference circle	N
F_{max}	maximum tangential tooth load for the mesh calculated	N
F_t	(nominal) transverse tangential load at reference cylinder per mesh	N
F_{tH}	determinant tangential load in a transverse plane for $K_{H\alpha}$ and $K_{F\alpha}$ $F_{tH} = F_t K_A K_V K_{H\beta}$	N
F_α	total profile deviation	μm
F_β	total helix deviation	μm
$F_{\beta\delta}$	tolerance on total helix deviation for ISO accuracy grade 6	μm
$F_{\beta x}$	initial equivalent misalignment (before running-in)	μm
$F_{\beta x\ cv}$	initial equivalent misalignment for the determination of the crowning height (estimate)	μm
$F_{\beta x\ T}$	equivalent misalignment measured under a partial load	μm
$F_{\beta y}$	effective equivalent misalignment (after running-in)	μm
f_{be}	component of equivalent misalignment ^b due to bearing deformation	μm
f_{ca}	component of equivalent misalignment ^b due to case deformation	μm
$f_{f\alpha}$	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}	mesh misalignment ^b due to manufacturing deviations	μm
f_{pt}	transverse single pitch deviation	μm

Table 1 (continued)

Symbol	Description	Unit
f_{par}	non-parallelism of pinion and wheel axes (manufacturing deviation) ^b	μm
f_{pb}	transverse base pitch deviation (the values of f_{pt} may be used for calculations in accordance with ISO 6336, using tolerances complying with ISO 1328-1)	μm
f_{sh}	component of equivalent misalignment ^b due to deformations of pinion and wheel shafts	μm
f_{shT}	component of misalignment due to shaft and pinion deformation measured at a partial load	μm
f_{sh0}	shaft deformation under specific load ^b	$\mu\text{m}\cdot\text{mm}/\text{N}$
$f_{\text{H}\beta}$	helix slope deviation (the value for the total helix deviation F_{β} may be used alternatively for this, if tolerances complying with ISO 1328-1 are used)	μm
$[AC_1] f_{\text{H}\beta 5}$	tolerance on helix slope deviation for ISO accuracy grade 5 $[AC_1]$	μm
g_{α}	length of path of contact	mm
h_{aP}	addendum of basic rack of cylindrical gears	mm
h_{fP}	dedendum of basic rack of cylindrical gears	mm
h_{f2}	dedendum of tooth of an internal gear	mm
h_{min}	minimum lubricant film thickness	mm
h_{Fe}	bending moment arm for tooth root stress relevant to load application at the outer point of single pair tooth contact	mm
h_{t}	tooth height	mm
J^*	moment of inertia per unit face width	$\text{kg}\cdot\text{mm}^2/\text{mm}$
K'	constant of the pinion offset	—
K_{v}	dynamic factor	—
K_{A}	application factor	—
$K_{\text{F}\alpha}$	transverse load factor (root stress)	—
$K_{\text{F}\beta}$	face load factor (root stress)	—
$K_{\text{H}\alpha}$	transverse load factor (contact stress)	—
$K_{\text{H}\beta}$	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)	—
l_{a}	effective length of roller (roller bearings)	mm
m^*	relative individual gear mass per unit face width referenced to line of action	kg/mm
m_{n}	normal module	mm
m_{red}	reduced gear pair mass per unit face width referenced to the line of action	kg/mm
m_{t}	transverse module	mm
N_{F}	exponent	—
N_{i}	number of cycles to failure for bin i	—
N_{L}	number of load cycles	—
N_{S}	resonance ratio in the main resonance range	—
$n_{1,2}$	rotation speed of pinion (or wheel)	min^{-1} or s^{-1}
n_{i}	number of cycles for bin i	—
n_{E}	resonance speed	min^{-1}
p_{bn}	normal base pitch	mm
p_{bt}	transverse base pitch	mm
q'	minimum value for the flexibility of a pair of meshing teeth	$(\text{mm}\cdot\mu\text{m})/\text{N}$
q_{pr}	protuberance of the tool (see ISO 6336-3)	mm
q_{s}	notch parameter, $q_{\text{s}} = s_{\text{Fn}} / 2\rho_{\text{F}}$	—

Table 1 (continued)

Symbol	Description	Unit
q_{sk}	notch parameter of the notched test piece	—
q_{sT}	notch parameter of the standard reference test gear, $q_{sT} = 2,5$	—
q_{α}	auxiliary factor	—
Ra	arithmetic mean roughness value, $Ra = 1/6 Rz$	μm
Rz	mean peak-to-valley roughness (as specified in ISO 4287 and ISO 4288)	μm
Rz_k	mean peak-to-valley roughness of the notched, rough test piece	μm
Rz_T	mean peak-to-valley roughness of the standard reference test gear, $Rz_T = 10$	μm
r_b	base radius	mm
ρ_{FP}	root fillet radius of the basic rack for cylindrical gears	mm
ρ_g	radius of grinding notch	mm
ρ_{red}	radius of relative curvature	mm
ρ_C	radius of relative curvature at the pitch surface	mm
ρ_F	tooth root radius at the critical section	mm
ρ'	slip-layer thickness	mm
S_F	safety factor for tooth breakage	—
S_H	safety factor for pitting	—
s_c	film thickness of marking compound used in contact pattern determination	μm
s_{pr}	residual fillet undercut, $s_{pr} = q_{pr} - q$	mm
s_{Fn}	tooth root chord at the critical section	mm
s_R	rim thickness	mm
$\sigma_{k \text{ lim}}$	nominal notched-bar stress number (bending)	N/mm ²
$\sigma_{p \text{ lim}}$	nominal plain-bar stress number (bending)	N/mm ²
σ_B	tensile strength	N/mm ²
σ_F	tooth root stress	N/mm ²
σ_{Fi}	tooth root stress for bin i	N/mm ²
$\sigma_{F \text{ lim}}$	nominal stress number (bending)	N/mm ²
σ_{FE}	allowable stress number (bending), $\sigma_{FE} = \sigma_{F \text{ lim}} Y_{ST}$	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	contact stress	N/mm ²
σ_{Hi}	contact stress for bin i	N/mm ²
$\sigma_{H \text{ lim}}$	allowable stress number (contact)	N/mm ²
σ_{HG}	pitting stress limit	N/mm ²
σ_{HP}	permissible contact stress	N/mm ²
σ_{H0}	nominal contact stress	N/mm ²
σ_S	yield stress	N/mm ²
σ_i	stress for bin i	N/mm ²
$\sigma_{0,2}$	proof stress (0,2 % permanent set)	N/mm ²
$T_{1,2}$	nominal torque at the pinion (or wheel)	N m
T_{eq}	equivalent torque	N m
T_i	torque for bin i	N m
T_n	nominal torque	N m

Table 1 (continued)

Symbol	Description	Unit
t_g	maximum depth of grinding notch	mm
U	sum of individual damage parts	—
w_m	mean specific load (per unit face width)	N/mm
w_t	tangential load per unit tooth width, including overload factors	N/mm
$x_{1,2}$	profile shift coefficient of pinion (or wheel)	—
χ^*	relative stress gradient in the root of a notch	mm^{-1}
χ_β	factor characterizing the equivalent misalignment after running-in	—
χ_p^*	relative stress gradient in a smooth polished test piece	mm^{-1}
Y_{DT}	deep tooth factor	—
Y_F	tooth form factor, for the influence on nominal tooth root stress with load applied at the outer point of single pair tooth contact	—
Y_M	mean stress influence factor	—
Y_{Nk}	life factor for tooth root stress, relevant to the notched test piece	—
Y_{Np}	life factor for tooth root stress, relevant to the plain polished test piece	—
Y_{NT}	life factor for tooth root stress for reference test conditions	—
Y_R	tooth root surface factor (relevant to the plain polished test piece)	—
$Y_{R \text{ rel } k}$	relative roughness factor, the quotient of the gear tooth root surface factor of interest divided by the notch test piece factor, $Y_{R \text{ rel } k} = Y_R/Y_{Rk}$	—
$Y_{R \text{ rel } T}$	relative surface factor, the quotient of the gear tooth root surface factor of interest divided by the tooth root surface factor of the reference test gear, $Y_{R \text{ rel } T} = Y_R/Y_{RT}$	—
Y_S	stress correction factor, for the conversion of the nominal tooth root stress, determined for application of load at the outer point of single pair tooth contact, to the local tooth root stress	—
Y_{Sg}	stress correction factors for teeth with grinding notches	—
Y_{Sk}	stress correction factor, relevant to the notched test piece	—
Y_{ST}	stress correction factor, relevant to the dimensions of the reference test gears	—
Y_X	size factor (tooth root)	—
Y_β	helix angle factor (tooth root)	—
Y_δ	notch sensitivity factor of the actual gear (relative to a polished test piece)	—
$Y_{\delta k}$	sensitivity factor of a notched test piece, relative to a smooth polished test piece	—
$Y_{\delta T}$	sensitivity factor of the standard reference test gear, relative to the smooth polished test piece	—
$Y_{\delta \text{ rel } k}$	test relative notch sensitivity factor, the quotient of the gear notch sensitivity factor of interest divided by the notched test piece factor, $Y_{\delta \text{ rel } k} = Y_{\delta}/Y_{\delta k}$	—
$Y_{\delta \text{ rel } T}$	relative notch sensitivity factor, the quotient of the gear notch sensitivity factor of interest divided by the standard test gear factor, $Y_{\delta \text{ rel } T} = Y_{\delta}/Y_{\delta T}$	—
y_α	running-in allowance for a gear pair	μm
y_β	running-in allowance (equivalent misalignment)	μm
Z_v	velocity factor	—
Z_B, Z_D	single pair tooth contact factors for the pinion, for the wheel	—
Z_E	elasticity factor	$\sqrt{\text{N/mm}^2}$
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	—

Table 1 (continued)

Symbol	Description	Unit
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)	—
z_n	virtual number of teeth of a helical gear	—
$z_{1,2}$	number of teeth of pinion (or wheel) ^a	—
$\omega_{1,2}$	angular velocity of pinion (or wheel)	rad/s
^a For external gears a , z_1 and z_2 are positive; for internal gearing, a and z_2 have a negative sign, z_1 has a positive sign.		
^b The components in the plane of action are determinant.		

4 Basic principles

4.1 Application

4.1.1 Scuffing

Formulae for scuffing resistance on cylindrical gear teeth are not included in ISO 6336. At the present time there is insufficient agreement concerning the method for designing cylindrical gears to resist scuffing failure.

4.1.2 Wear

Very little attention and concern have been devoted to the study of gear tooth wear. This subject primarily concerns gear teeth with low surface hardness or gears with improper lubrication. No attempt has been made to cover the subject in ISO 6336.

4.1.3 Micropitting

ISO 6336 does not cover micropitting, which is an additional type of surface distress that may occur on gear teeth.

4.1.4 Plastic yielding

ISO 6336 does not extend to stress levels greater than those permissible at 10^3 cycles or less, since stresses in this range may exceed the elastic limit of the gear tooth in bending or in surface compressive stress. Depending on the material and the load imposed, a single stress cycle greater than the limit level at $< 10^3$ cycles could result in plastic yielding of the gear tooth.

4.1.5 Particular categories

Pitting resistance and bending strength rating systems for a particular category of cylindrical gearing may be established by selecting proper values for the factors used in these general formulae.

4.1.6 Specific applications

For the design of gears it is very important to recognize that requirements for different fields of application vary considerably. Use of ISO 6336 procedures for specific applications demands a realistic and knowledgeable appraisal of all applicable considerations, particularly:

- the allowable stress of the material and the number of load repetitions,
- the consequences of any percentage of failure (failure rate), and
- the appropriate safety factor.

The following three application fields exemplify the requirements of the above-mentioned characteristics.

4.1.6.1 Vehicle final drive gears

For vehicle final drive gears, which are relatively low speed, coarse pitch teeth, are chosen for adequate strength. As a consequence, pinions have small numbers of teeth (z_1 of about 14), whereas a value z_1 of about 28 would be chosen for a comparatively high speed gear of similar size. Thus, the tooth bending strength of the former would be about twice that of the latter.

The computed reliability of vehicle gears can be as low as 80 % to 90 % whereas that of high speed industrial gears should be at least 99 %.

In general, the material used in high volume vehicle gear production may be of more uniform quality than that used for gears produced in small numbers.

Comparison of applied gear designs has indicated that for about 10 000 cycles, the load transmitted by truck final drive gears is about four times greater than that transmitted by aircraft or space vehicle gears, where the material, quality, size and design are the same.

For low speed vehicle gears which are intended to have short lives (less than 100 000 cycles), small amounts of plastic deformation, pitting and abrasive wear can usually be tolerated. Consequently, the levels of surface stress which are permissible are substantially higher than would be permissible for long life, high speed gears.

4.1.6.2 Main drive for aircraft and space vehicles

For main drives of aircraft and space vehicles, which are found in helicopter rotor drives and the main pump drives of space vehicle boosters, gears of the highest quality material and accuracy are used. Such gears are extensively tested. For example, 10 to 20 transmissions of the same production series may be tested under operational conditions for the full design life. The tolerable wear rate is established on the basis of test results. Lubricant spray rate, position of injection points and direction of spray is optimized.

For these reasons, higher loading is permissible for a design life up to 100 times longer (in cycles of tooth loading), and speeds about 10 times greater than those of a typical vehicle transmission. The probability of damage in such cases shall not exceed 0,1 % to 1 %. Overall loading cannot be as high as for vehicle gears, since neither surface wear nor minor damage can be tolerated.

4.1.6.3 Industrial high speed gears

For industrial high speed gears, where the pitch line velocities exceed 50 m/s, the pinions are often designed with 30 or more teeth with the objective of minimizing the risk of scuffing and wear. A typical gear pair would consist of a pinion with 45 teeth and a wheel with 248.

Industrial high speed gears should be better than 99 % reliable for a normal life of more than 10^{10} cycles. Extensive prototype testing is normally excluded because of the cost. As a consequence, the load capacity ratings of high speed gears tend to be conservative with relatively high safety factors.

4.1.7 Safety factors

It is necessary to distinguish between the safety factor relative to pitting, S_H , and that relative to tooth breakage, S_F .

For a given application, adequate gear load capacity is demonstrated by the computed values of S_H and S_F being greater than or equal to the values $S_{H \min}$ and $S_{F \min}$, respectively.

Certain minimum values for safety factors shall be determined. Recommendations concerning these minimum values are made in ISO 6336, but values are not proposed.

An appropriate probability of failure and the safety factor shall be carefully chosen to meet the required reliability at a justifiable cost. If the performance of the gears can be accurately appraised through testing of the actual unit under actual load conditions, a lower safety factor and more economical manufacturing procedures may be permissible:

$$\text{Safety factor} = \frac{\text{Modified allowable stress number}}{\text{Calculated stress}}$$

Safety factors based on load are permitted. When they are based on load the safety factor equals the specific calculated load capacity divided by the specific operating load transmitted. When the factor is based on load, this shall be stated clearly.

NOTE Safety factors based on load (power) relative to tooth bending are proportional to S_F . Safety factors based on load (power) relative to pitting are proportional to S_H^2 .

In addition to the general requirements mentioned and the special requirements for surface durability, pitting, (see ISO 6336-2) and tooth bending strength (see ISO 6336-3), the safety factors shall be chosen after careful consideration of the following influences.

- Reliability of material data: ISO 6336-applicable materials, for which data are given in ISO 6336-5, and their abbreviations, are listed in Table 2. The allowable stress numbers used in the calculation are valid for a given probability of failure; the material values in ISO 6336-5 are valid for 1 % probability of damage. This risk of damage reduces with the increase of the safety factor and vice versa.
- Reliability of load values used for calculation: if loads or the response of the system to vibration, are estimated rather than measured, a larger safety factor should be used.
- Variations in gear geometry due to manufacturing tolerances.
- Variations in alignment.
- Variations in material due to process variations in chemistry, cleanliness and microstructure (material quality and heat treatment).
- Variations in lubrication and its maintenance over the service life of the gears.

Depending on the reliability of the assumptions on which the calculations are based (e.g. load assumptions) and according to the reliability requirements (consequences of damage occurrence), a corresponding safety factor is to be chosen.

Where gears are produced under a specification or a request for proposal (quotation), in which the gear supplier is to provide gears or assembled gear drives having specified calculated capacities (ratings) in accordance with ISO 6336, the value of the safety factor for each mode of failure (pitting, tooth breakage) is to be agreed upon between the parties.

Table 2 — Materials

Material	Type	Abbreviation
Normalized low carbon steels / cast steels	Wrought normalized low carbon steels	St
	Cast steels	St (cast)
Cast iron materials	Black malleable cast iron (perlitic structure)	GTS (perl.)
	Nodular cast iron (perlitic, bainitic, ferritic structure)	GGG (perl., bai., ferr.)
	Grey cast iron	GG
Through-hardened wrought steels	Carbon steels, alloy steels	V
Through-hardened cast steels	Carbon steels, alloy steels	V(cast)
Case-hardened wrought steels		Eh
Flame or induction hardened wrought or cast steels		IF
Nitrided wrought steels / nitriding steels / through-hardening steels, nitrided	Nitriding steels	NT(nitr.)
	Through hardening steels	NV (nitr.)
Wrought steels, nitrocarburized	Through hardening steels	NV (nitrocar.)

4.1.8 Testing

The most reliable known approach to the appraisal of overall system performance is that of testing a proposed new design. Where sufficient field or test experience is available, satisfactory results can be obtained by extrapolation of previous tests or field data.

When suitable test results or field data are not available, values for the rating factors should be chosen conservatively.

4.1.9 Manufacturing tolerances

Evaluation of rating factors should be based on the minimum accuracy grade limits specified for the component parts in the manufacturing process.

4.1.10 Implied accuracy

Where empirical values for rating factors are given by curves, curve fitting equations are provided to facilitate computer programming. The constants and coefficients used in curve fitting often have significant digits in excess of those appropriate to the reliability of the empirical data.

4.1.11 Other considerations

In addition to the factors considered in ISO 6336 influencing pitting resistance and bending strength, other interrelated system factors can have a significant influence on overall transmission performance. The following factors are particularly significant.

4.1.11.1 Lubrication

The ratings determined by these formulae are valid only if the gear teeth are operated with a lubricant of proper viscosity and additives for the load, speed and surface finish, and if there is a sufficient quantity of lubricant supplied to the gear teeth and bearings to lubricate and maintain an acceptable operating temperature.

4.1.11.2 Misalignment and deflection of foundations

Many gear systems depend on external supports such as machinery foundations to maintain alignment of the gear mesh. If these supports are poorly designed, initially misaligned, or become misaligned during operation through elastic or thermal deflection or other influences, overall gear system performance will be adversely affected.

4.1.11.3 Deflections

Deflections of gear teeth, gear blanks, gear shafts, bearings and housings affect performance and distribution of total tooth load over meshing flanks. Since these deflections vary with load, it is impossible to obtain optimum tooth contact at different loads in those transmissions that encounter variable load. When gear tooth flanks are not modified, the face load factor increases with increasing deflection, thereby lowering rated capacity.

4.1.11.4 System dynamics

The method of analysis used in ISO 6336 provides a dynamic factor in the formulae by derating the gears for increased loads caused by gear tooth inaccuracies and for harmonic effects. In general, simplified values are given for easy application. The dynamic response of the system results in additional gear tooth loads due to the relative motions of the connected masses of the driver and the driven equipment. The application factor, K_A , is intended to account for the operating characteristics of the driving and driven equipment. It must be recognized, however, that if the operating roughness of the driver, gearbox or driven equipment causes an excitation with a frequency that is near to one of the system's major natural frequencies, resonant vibrations can cause severe overloads which could be several times higher than the nominal load.

For critical service applications, it is recommended that a vibration analysis be performed. This analysis shall include the total system of driver, gearbox, driven equipment, couplings, mounting conditions and sources of excitation. Natural frequencies, mode shapes and the dynamic response amplitudes should be calculated. The resulting load spectrum cumulative fatigue effect calculation, if necessary or required, is given in ISO 6336-6.

4.1.11.5 Contact pattern

The teeth of most cylindrical gears are modified in both profile and helix directions during the manufacturing operation to accommodate deflection of the mountings and thermal distortions. This results in a localized contact pattern during roll testing under light loads. Under design load, the contact should spread over the tooth flank without any concentration of the pattern at the edges. This influence shall be taken into account by the corresponding load distribution factor.

4.1.11.6 Corrosion

Corrosion of gear tooth surfaces can significantly reduce the bending strength and pitting resistance of the teeth. Quantifying the extent of these reductions is beyond the scope of ISO 6336.

4.1.12 Influence factors

The influence factors presented in ISO 6336 are derived from results of research and field service. It is convenient to distinguish between the following.

- a) Factors which are determined by gear geometry or which have been established by convention. They shall be calculated in accordance with the equations given in ISO 6336.
- b) Factors which account for several influences and which are dealt with as independent of each other, but which may nevertheless influence each other to a degree for which no numerical value can be assigned. These include the factors K_A , K_V , $K_{H\alpha}$, $K_{H\beta}$ or $K_{F\alpha}$ and the factors influencing allowable stress.

The factors K_v , $K_{H\beta}$ and $K_{H\alpha}$ also depend on the magnitudes of the profile and helix modifications. Profile and helix modifications are only effective if they are significantly larger than the manufacturing deviations. For this reason, the influence of the profile and helix modifications may only be taken into consideration if the gear manufacturing deviations do not exceed specific limit values. The minimum required gear manufacturing accuracy is stated, together with reference to ISO 1328-1, for each factor.

The influence factors can be determined by various methods. These are qualified, as necessary, by adding subscripts A through C to the symbols. Unless otherwise specified, e.g. in an application standard, the more accurate of the methods is to be preferred for important transmissions. In cases of dispute, when proof of accuracy and reliability is supplied, Method A is superior to Method B, and Method B to Method C.

It is recommended that supplementary subscripts be used whenever the method used for evaluation of a factor would not be readily identifiable.

In some applications it could be necessary to choose between factors which have been determined using alternative methods (e.g. the alternatives for the determination of the equivalent misalignment). When necessary, the relevant method can be indicated by extending the subscript, e.g. $K_{H\beta-B1}$.

ISO 6336 is primarily intended for verifying the load capacity of gears for which essential calculation data are available by way of detail drawings, or in similar form.

The data available at the primary design stage is usually restricted. It is therefore necessary, at this stage, to make use of approximations or empirical values for some factors.

For given fields of application or for rough calculations, it is often permissible to substitute unity or some other constant for some factors. In doing so, it is necessary to verify that a good margin of safety is assured. Otherwise, the safety factor shall be adequately increased.

More precise evaluation is possible when manufacture and inspection is completed, for then data obtained by direct measurement are available.

Contractual provisions relating to the nature of the calculation proof shall be agreed in advance between manufacturer and purchaser.

4.1.12.1 Method A

Method A factors are derived from the results of full scale load tests, precise measurements or comprehensive mathematical analysis of the transmission system on the basis of proven operating experience, or any combination of these. All gear and loading data shall be available. In such cases the accuracy and reliability of the method used shall be demonstrated and the assumptions clearly stated.

In general, and for the following reasons, Method A is seldom used:

- the relevant relationships have not been more extensively researched than those in Methods B and C;
- details of the operating conditions are incomplete;
- suitable measuring equipment is not available;
- the costs of analysis and measurements exceed their value.

4.1.12.2 Method B

Method B factors are derived with sufficient accuracy for most applications. Assumptions involved in their determination are listed. In each case, it is necessary to assess whether or not these assumptions apply to the conditions of interest. Additional subscripts should be inserted when necessary, e.g. K_{v-B} .

4.1.12.3 Method C

Method C is where simplified approximations are specified for some factors. The assumptions under which they have been determined are listed. On each occasion an assessment should be made as to whether or not these assumptions apply to the existing conditions. The additional subscript C shall be used when necessary, e.g. K_{V-C} .

4.1.13 Numerical equations

It is necessary to apply the numerical equations specified in ISO 6336 with the stated units. Any exceptions are specially noted.

4.1.14 Succession of factors in course of calculation

The factors K_V , $K_{H\beta}$ or $K_{F\beta}$ and $K_{H\alpha}$ or $K_{F\alpha}$ depend on a nominal tangential load. They are also to some extent interdependent and shall therefore be calculated successively as follows:

- a) K_V with the load $F_t K_A$;
- b) $K_{H\beta}$ or $K_{F\beta}$ with the load $F_t K_A K_V$;
- c) $K_{H\alpha}$ or $K_{F\alpha}$ with the load $F_t K_A K_V K_{H\beta} (F\beta)^{-1}$.

When a gear drives two or more mating gears, it is necessary to multiply by $(K_A K_\gamma)$ instead of K_A ; also see 4.2.

4.1.15 Determination of allowable values of gear deviations

The allowable values of gear deviation shall be determined in accordance with ISO 1328-1.

4.2 Tangential load, torque and power

When assessing the load acting on gear teeth, all loads affecting the gearing shall be considered.

In the case of double helical gears, it is assumed that the total tangential load is divided equally between the two helices. If this is not the case, for examples as a consequence of externally applied axial loads, this shall be taken into consideration. The two halves of the helix should be treated as two helical gears arranged in parallel.

Concerning multiple-path transmissions, the total tangential load is not quite evenly distributed to the various load paths (irrespective of design, tangential velocity or accuracy of manufacture). Allowance is made for this by means of the mesh load factor K_γ (see also 4.1.14). If possible, K_γ should preferably be determined by measurement; alternatively, its value may be estimated from the literature.

If the operating speed is near to a resonance speed, a careful study is necessary. See Clauses 5 and 6.

4.2.1 Nominal tangential load, nominal torque and nominal power

The nominal tangential load F_t is determined in the transverse plane at the reference cylinder. It is derived from the nominal torque or power transmitted by the gear pair.

The load capacity rating of gears is effectively based on the input torque to the driven machine. This is the torque corresponding to the heaviest regular working condition. Alternatively, the nominal torque of the prime

1) $K_{H\beta}$ is also to be used in the evaluation of $K_{F\alpha}$, since for tooth bending it is $K_{H\beta}$ which represents the determinant load due to uneven distribution of F_t over the face width (see 7.2.1).

mover can be used as a basis if it corresponds to the torque requirement of the driven machine, or some other suitable basis can be chosen.

F_t is defined as the nominal tangential load per mesh, i.e. for the mesh under consideration. T and P are defined accordingly. In the following equations, $n_{1,2}$ is given in revolutions per minute.

$$F_t = \frac{2\,000\,T_{1,2}}{d_{1,2}} = \frac{19\,098 \times 1\,000\,P}{d_{1,2}\,n_{1,2}} = \frac{1\,000\,P}{v} \quad (1)$$

$$T_{1,2} = \frac{F_t\,d_{1,2}}{2\,000} = \frac{1\,000\,P}{\omega_{1,2}} = \frac{9\,549\,P}{n_{1,2}} \quad (2)$$

$$P = \frac{F_t\,v}{1\,000} = \frac{T_{1,2}\,\omega_{1,2}}{1\,000} = \frac{T_{1,2}\,n_{1,2}}{9\,549} \quad (3)$$

$$v = \frac{d_{1,2}\,\omega_{1,2}}{2\,000} = \frac{d_{1,2}\,n_{1,2}}{19\,098} \quad (4)$$

$$\omega_{1,2} = \frac{2\,000\,v}{d_{1,2}} = \frac{n_{1,2}}{9,549} \quad (5)$$

4.2.2 Equivalent tangential load, equivalent torque and equivalent power

When the transmitted load is not uniform, consideration should be given not only to the peak load and its anticipated number of cycles, but also to intermediate loads and their numbers of cycles. This type of load is classed as a *duty cycle* and may be represented by a load spectrum. In such cases, the cumulative fatigue effect of the duty cycle is considered in rating the gear set. A method of calculating the effect of the loads under this condition is given in ISO 6336-6.

4.2.3 Maximum tangential load, maximum torque and maximum power

This is the maximum tangential load $F_{t\max}$ (or corresponding torque $T_{\max}$, corresponding power $P_{\max}$) in the variable duty range. Its magnitude can be limited by a suitably responsive safety clutch. $F_{t\max}$, $T_{\max}$ and $P_{\max}$ are required to determine the safety from pitting damage and from sudden tooth breakage due to loading corresponding to the static stress limit.

5 Application factor K_A

The factor K_A adjusts the nominal load F_t in order to compensate for incremental gear loads from external sources. These additional loads are largely dependent on the characteristics of the driving and driven machines, as well as the masses and stiffness of the system, including shafts and couplings used in service.

For applications such as marine gears and others subjected to cyclic peak torque (torsional vibrations) and designed for infinite life, the application factor can be defined as the ratio between the peak cyclic torques and the nominal rated torque. The nominal rated torque is defined by the rated power and speed. It is the torque used in the load capacity calculations.

If the gear is subjected to a limited number of known loads in excess of the amount of the peak cyclic torques, this influence may be covered directly by means of cumulative fatigue or by means of an increased application factor, representing the influence of the load spectrum.

It is recommended that the purchaser and manufacturer/designer agree on the value of the application factor.

5.1 Method A — Factor K_{A-A}

K_A is determined in this method by means of careful measurements and a comprehensive analysis of the system, or on the basis of reliable operational experience in the field of application concerned. See 4.2.2.

5.2 Method B — Factor K_{A-B}

If no reliable data, obtained as described in 5.1, are available, even as early as the first design phase, it is possible to use the guideline values for K_A as described in ISO 6336-6.

6 Internal dynamic factor K_V

The internal dynamic factor makes allowance for the effects of gear tooth accuracy grade as related to speed and load. High accuracy gearing requires less derating than low accuracy gearing.

It is generally accepted that the internal dynamic load on the gear teeth is influenced by both design and manufacturing.

The internal dynamic factor makes allowance for the effects of gear tooth accuracy and modifications as related to speed and load:

$$K_V = \frac{\text{Total mesh torque at operating speed}}{\text{Mesh torque with "perfect" gears}}$$

$$K_A \cdot K_V = \frac{\text{Total mesh torque at operating speed}}{\text{Nominal transmitted (design) mesh torque}}$$

"Perfect" gears are defined as having zero quasi-static transmission error at the nominal transmitted (design) mesh torque. They can only exist for a single load and, with proper modifications, have zero dynamic effects [e.g. zero transmission error (perfect conjugate action), zero excitation, no fluctuation at tooth mesh frequency and no fluctuation at rotational frequencies]. With zero excitation from the gears, there is zero response at any speed.

6.1 Parameters affecting internal dynamic load and calculations

6.1.1 Design

The design parameters include the following:

- pitchline velocity;
- tooth load;
- inertia and stiffness of the rotating elements;
- tooth stiffness variation;
- lubricant properties;
- stiffness of bearings and case structure;
- critical speeds and internal vibration within the gear itself.

6.1.2 Manufacturing

The manufacturing considerations include the following:

- pitch deviations;
- runout of reference surfaces with respect to the axis of rotation;
- tooth flank deviations;
- compatibility of mating gear tooth elements;
- balance of parts;
- bearing fit and preload.

6.1.3 Transmission perturbation

Even when the input torque and speed are constant, significant vibration of the gear masses, and resultant dynamic tooth loads, can exist. These loads result from the relative displacements between the mating gears as they vibrate in response to an excitation known as transmission error. The ideal kinematics of a gear pair require a constant ratio between the input and output rotations. Transmission error is defined as the departure from uniform relative angular motion of a pair of meshing gears. It is influenced by all deviations from the ideal gear tooth form and spacing due to the design and manufacture of the gears, and to the operational conditions under which the gears shall perform. The latter include the following.

- a) Pitch line velocity: the frequencies of excitation depend on the pitch line velocity and module.
- b) Gear mesh stiffness variations as the gear teeth pass through the meshing cycle: this source of excitation is especially pronounced in spur gears. Spur and helical gears with total contact ratios greater than 2,0 have less stiffness variation.
- c) Transmitted tooth load: since deflections are load-dependent, gear tooth profile modifications can be designed to give uniform velocity ratio only for one magnitude of load. Loads different from the design load will give increased transmission error.
- d) Dynamic unbalance of the gears and shafts.
- e) Application environment: excessive wear and plastic deformation of the gear tooth profiles increase the transmission error. Gears shall have a properly designed lubrication system, enclosure and seals to maintain a safe operating temperature and a contamination-free environment.
- f) Shaft alignment: gear tooth alignment is influenced by load and thermal deformations of the gears, shafts, bearings and housing.
- g) Excitation induced by tooth friction.

6.1.4 Dynamic response

The effects of dynamic tooth loads are influenced by the following:

- mass of the gears, shafts, and other major internal components;
- stiffness of the gear teeth, gear blanks, shafts, bearings and housings;
- damping, the principal sources of which are the shaft bearings and seals, while other sources include hysteresis of the gear shafts and viscous damping at sliding interfaces and shaft couplings.

6.1.5 Resonance

When excitation frequencies (such as tooth meshing frequency and its harmonics) coincide or nearly coincide with a natural frequency of vibration of the gearing system, the resonant forced vibration can cause high dynamic tooth loading. When the magnitude of internal dynamic load at a speed involving resonance becomes large, operation near this speed should be avoided.

a) Gear blank resonance

The gear blanks of high speed, lightweight gearing can have natural frequencies within the operating speed range. If the gear blank is excited by a frequency which is close to one of its natural frequencies, the resonant deflections could cause high dynamic tooth loads. Also, there is the possibility of plate or shell mode vibrations which can cause the gear blank to fail. The dynamic factors K_v (from the following Methods B and C) do not take account of gear blank resonance.

b) System resonance

The gearbox is only one component of a system comprised of a power source, gearbox, driven equipment and interconnecting shafts and couplings. The dynamic response of this system depends on the configuration of the system. In certain cases a system could possess a natural frequency close to the excitation frequency associated with an operating speed. Under such resonant conditions, the operation shall be worked out carefully as mentioned above. For critical drives, a detailed analysis of the entire system is recommended. This should also be taken into account when determining the effects on the application factor.

6.2 Principles and assumptions

In accordance with the specifications of 4.1.12, methods for determining K_v are given in 6.3, from Method A (K_{v-A}) to Method C (K_{v-C}).

Given optimum profile modification appropriate to the loading, a large overlap ratio, even distribution of load over the face width, highly accurate teeth and high specific tooth loading, the value of the dynamic factor approaches 1,0.

In gear trains which include multiple mesh gears such as idler gears and epicyclic gearing, planet and sun gears, there are several natural frequencies. These can be higher or lower than the natural frequency of a single gear pair which has only one mesh. When such gears run in the supercritical range, resonance of higher natural frequencies occur and may lead to a higher dynamic factor.

Transverse vibrations of the shaft-gear systems will also influence the dynamic load. Transverse compliance of a shaft-gear system can result in coupled vibrations where the pinion and/or wheel combine torsional and lateral vibrations. This results in more natural frequencies than the one formed by just a pinion and wheel and may lead to a higher dynamic factor K_v .

In the case of high specific loading, high values of $(K_A F_t) / b$, high values of $(v z_1 / 100) \sqrt{u^2 / (1 + u^2)}$, and having the corresponding accuracy of gear cutting, tooth tips and/or roots should be suitably relieved.

6.3 Methods for determination of dynamic factor

6.3.1 Method A — Factor K_{v-A}

The maximum tooth loads, including the internally generated dynamic additional loads and the uneven distribution of loads as described in Clause 8, are determined in Method A by measurement or by a comprehensive dynamic analysis of the general system. Under these circumstances K_v (just as $K_{H\alpha}$ and $K_{F\alpha}$) is assumed to have a value of 1,0.

K_v can also be assessed by measuring the tooth root stresses of the gears when transmitting load at working speed and at a lower speed, then comparing the results.

The factor K_v may be determined by a comprehensive analytical procedure which is supported by experience of similar designs. Guidance on procedures can be found in the literature.

Reliable values of the dynamic factor, K_v , can best be predicted by a mathematical model which has been satisfactorily verified by measurement.

6.3.2 Method B — Factor K_{v-B}

For this method the simplifying assumption is made that the gear pair consists of an elementary single mass and spring system comprising the combined masses of pinion and wheel, the stiffness being the mesh stiffness of the contacting teeth. It is also assumed that each gear pair functions as a single stage gear pair, i.e. the influence of other stages in a multiple-stage gear system is ignored. This assumption is permissible if the torsional stiffness of the shafts connecting the wheel of one stage with the pinion of the next is relatively low. See 6.4.2 for the procedure dealing with very stiff shafts.

In accordance with this assumption, loads due to torsional vibration of the shafts and coupled masses are not covered by K_v . These latter loads should be included with other externally applied loads (e.g. with the application factor).

It is further assumed, in the evaluation of dynamic factors by Method B, that damping at the gear mesh has an average value. (Other sources of damping such as friction at component interfaces, hysteresis, bearings, couplings, etc., are not taken into consideration.) Because of these additional sources of damping, the actual dynamic tooth loads are normally somewhat smaller than those calculated by this method. This does not apply in the range of main resonance (see 6.4.4).

Calculation of K_v by this method serves no useful purpose when the value $(v z_1 / 100) \sqrt{u^2 / (1 + u^2)}$ is less than 3 m/s. Method C is sufficiently accurate for all cases in this range.

6.3.3 Method C — Factor K_{v-C}

Method C is derived from Method B by introducing the following additional simplifying assumptions.

- a) The running speed range is subcritical.
- b) Steel solid disc wheels.
- c) The pressure angle $\alpha_t = 20^\circ$; $f_{pb} = f_{pt} \cos 20^\circ$ according to ISO/TR 10064-1.
- d) Helix angle $\beta = 20^\circ$ for helical gearing (refers to c' , $c_{\gamma\alpha}$).
- e) Total contact ratio $\varepsilon_\gamma = 2,5$ for helical gearing.
- f) Tooth stiffness (see 9.3):
 - For spur gears, $c' = 14 \text{ N}/(\text{mm} \cdot \mu\text{m})$, $c_{\gamma\alpha} = 20 \text{ N}/(\text{mm} \cdot \mu\text{m})$;
 - For helical gears, $c' = 13,1 \text{ N}/(\text{mm} \cdot \mu\text{m})$, $c_{\gamma\alpha} = 18,7 \text{ N}/(\text{mm} \cdot \mu\text{m})$.
- g) Tip relief $C_a = 0 \mu\text{m}$ and tip relief after running-in $C_{ay} = 0 \mu\text{m}$.
- h) Effective deviation $f_{pb \text{ eff}} = f_{t\alpha \text{ eff}}$.
- i) For assumed values for f_{pb} , y_p and $f_{pb \text{ eff}}$, see Equations (18) and (19) and Table 3.

The features described in 6.4.1 a) have not been taken into consideration in the application of Method C. The influence of specific loading is taken into account.

Table 3 — Assumed mean values of effective base pitch deviation, $f_{pb \text{ eff}}$, for factor K_{V-C}

	Accuracy grades as specified in ISO 1328-1									
	3	4	5	6	7	8	9	10	11	12
$f_{pb \text{ eff}}$	2,8	5,1	9,8	19,5	35	51	69	100	134	191

6.4 Determination of dynamic factor using Method B: K_{V-B}

According to the preconditions and assumptions given in 6.3.2, Method B is suited for all types of transmission (spur and helical gearing with any basic rack profile and any gear accuracy grade) and, in principle, for all operating conditions. However, there are restrictions for certain fields of application and operation which will be noted in each case and should be taken into account.

The resonance ratio N (ratio of the running speed to the resonance speed) is determined as described in 6.4.2²⁾. The entire running speed range can be divided into three sectors — subcritical, main resonance and supercritical. Formulae are provided for calculating K_V in each sector.

NOTE The dynamic factors calculated from the equations in 6.4.3 to 6.4.6 correspond to the experimentally determined mean dynamic tooth load values. In the subcritical and main resonance ranges, values of K_V derived from measurement data usually deviate from the calculated values by up to +10 %. Even greater deviations can occur when there are other natural frequencies in the gear and shaft system. See 6.4.1 a), 6.4.3, and 6.4.4.

6.4.1 Running speed ranges

a) Subcritical range ($N < 1$)³⁾

In this sector, resonances can exist if the tooth mesh frequency coincides with $N = 1/2$ and $N = 1/3$. Under such circumstances the dynamic loads can exceed the values calculated using Equation (13). The risk of this is slight for precision helical or spur gears, if the latter have suitable profile modification (gears to accuracy grade 5 of ISO 1328-1 or better).

When the contact ratio of spur gears is small or if the accuracy is low, K_V can be just as great as in the main resonance speed range. If this occurs, the design or operating parameters should be altered.

Resonances at $N = 1/4, 1/5, \dots$ are seldom troublesome because the associated vibration amplitudes are usually small.

When the specific loading $(F_t K_A)/b < 50 \text{ N/mm}$, a particular risk of vibration exists (under some circumstances, with separation of working tooth flanks) — above all for spur or helical gears of coarse accuracy grade running at higher speed.

b) Main resonance range ($N = 1$)

Operation in this range should generally be avoided, especially for spur gears with unmodified tooth profiles, or helical gears of accuracy grade 6 or coarser as specified in ISO 1328-1. Helical gears of high accuracy with a high total contact ratio can function satisfactorily in this sector. Spur gears of grade 5 or better as specified in ISO 1328-1 shall have suitable profile modification.

c) Supercritical range ($N > 1$)

The same limitations on gear accuracy grade as in b) apply to gears operating in this speed range. Resonance peaks can occur at $N = 2, 3 \dots$ in this range. However, in the majority of cases vibration amplitudes are small, since excitation loads with lower frequencies than meshing frequency are usually small.

2) When it is known in advance that gears will operate in the supercritical sector, there is no need to evaluate the resonance speed. As a consequence, the dynamic factor can be directly determined in accordance with 6.4.5.

3) For a definition of N , see Equation (9). In practice, the calculated resonance sector is broadened to ensure a safe margin. See Equations (10) and (11) and the preamble thereto.

For some gears in this speed range, it is also necessary to consider dynamic loads due to transverse vibration of the gear and shaft assemblies (see 6.3.2). If the critical frequency is near the frequency of rotation, the associated effective value of K_v can exceed the value calculated using Equation (21) by up to 100 %. This condition should be avoided.

6.4.2 Determination of resonance running speed (main resonance) of a gear pair ³⁾

This is as follows:

$$n_{E1} = \frac{30\,000}{\pi z_1} \sqrt{\frac{c_{\gamma\alpha}}{m_{\text{red}}}} \quad (6)$$

where

m_{red} is the relative mass of a gear pair, i.e. of the mass per unit face width of each gear, referred to its base radius or to the line of action

$$m_{\text{red}} = \frac{m_1^* m_2^*}{m_1^* + m_2^*} = \frac{J_1^* J_2^*}{J_1^* r_{b2}^2 + J_2^* r_{b1}^2} \quad (7)$$

where

$$m_{1,2}^* = \frac{J_{1,2}^*}{r_{b1,2}^2} \quad (8)$$

and r_b is the base radius

See 6.4.8 for the method of calculating an approximate value of m_{red} . See Clause 9 for the stiffness $c_{\gamma\alpha}$

Method A is to be preferred for less common transmission designs. A method for deriving approximate values is specified for the following cases:

- a) pinion on large diameter shaft;
- b) two neighbouring gears rigidly joined together;
- c) one big wheel driven by two pinions;
- d) simple planetary gears;
- e) idler gears.

The ratio of pinion speed to resonance speed is termed the “resonance ratio”, N (n_1 is in revolutions per minute):

$$N = \frac{n_1}{n_{E1}} = \frac{n_1 \pi z_1}{30\,000} \sqrt{\frac{m_{\text{red}}}{c_{\gamma\alpha}}} \quad (9)$$

The resonance running speed n_{E1} may be above or below the running speed calculated from Equation (6) because of stiffnesses which have not been included (the stiffness of shafts, bearings, housings, etc.) and as a result of damping. For reasons of safety, the resonance ratio in the main resonance range, N_S , is defined by the following upper limit.

$$N_S < N \leq 1,15 \quad (10)$$

The lower limit of resonance ratio N_S is determined:

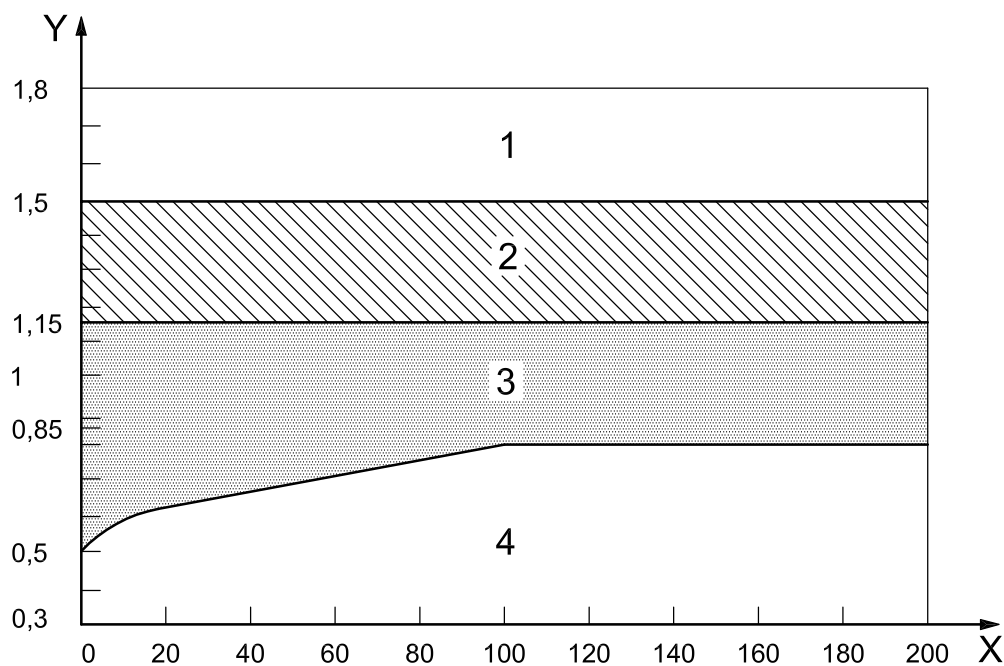
a) at loads such that $(F_t K_A) / b$ is less than 100 N/mm, as

$$N_S = 0,5 + 0,35 \sqrt{\frac{F_t K_A}{100 b}} \quad (11)$$

b) for loads where $(F_t K_A) / b \geq 100$ N/mm, as

$$N_S = 0,85 \quad (12)$$

For resonance ratio, N , in the main resonance range, see Figure 1.



Key

X specific loading, $\frac{F_t K_A}{b}$, N/mm

Y resonance ratio, N

- 1 supercritical range
- 2 intermediate range
- 3 main resonance range
- 4 subcritical range

Figure 1 — Resonance range