



BSI Standards Publication

Calculation of load capacity of spur and helical gears

Part 1: Basic principles, introduction and general influence factors

National foreword

This British Standard is the UK implementation of ISO 6336-1:2019. It supersedes BS ISO 6336-1:2006, which is withdrawn.

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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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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.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

This third edition cancels and replaces the second edition (ISO 6336-1:2006), which has been technically revised. It also incorporates the Technical Corrigendum ISO 6336-1:2006/Cor.1:2008.

The main changes compared to the previous edition are as follows:

- incorporation of ISO/TS 6336-4, ISO/TS 6336-20, ISO/TS 6336-21 and ISO/TS 6336-22 into [Clause 4](#) (failure mode);
- update of application factors in [Clause 5](#);
- integration of [Clause 10](#) "Parameters of Hertzian contact";
- integration of [Clause 11](#) "Lubricant parameters at given temperature".

A list of all parts in the ISO 6336 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

ISO 6336 (all parts) consists of International Standards, Technical Specifications (TS) and Technical Reports (TR) under the general title *Calculation of load capacity of spur and helical gears* (see [Table 1](#)).

- International Standards contain calculation methods that are based on widely accepted practices and have been validated.
- Technical Specifications (TS) contain calculation methods that are still subject to further development.
- Technical Reports (TR) contain data that is informative, such as example calculations.

The procedures specified in parts 1 to 19 of the ISO 6336 series cover fatigue analyses for gear rating. The procedures described in parts 20 to 29 of the ISO 6336 series are predominantly related to the tribological behavior of the lubricated flank surface contact. Parts 30 to 39 of the ISO 6336 series include example calculations. The ISO 6336 series allows the addition of new parts under appropriate numbers to reflect knowledge gained in the future.

Requesting standardized calculations according to the ISO 6336 series without referring to specific parts requires the use of only those parts that are currently designated as International Standards (see [Table 1](#) for listing). When requesting further calculations, the relevant part or parts of the ISO 6336 series need to be specified. Use of a Technical Specification as acceptance criteria for a specific design need to be agreed in advance between the manufacturer and the purchaser.

Table 1 — Parts of the ISO 6336 series (status as of DATE OF PUBLICATION)

Calculation of load capacity of spur and helical gears	International Standard	Technical Specification	Technical Report
<i>Part 1: Basic principles, introduction and general influence factors</i>	X		
<i>Part 2: Calculation of surface durability (pitting)</i>	X		
<i>Part 3: Calculation of tooth bending strength</i>	X		
<i>Part 4: Calculation of tooth flank fracture load capacity</i>		X	
<i>Part 5: Strength and quality of materials</i>	X		
<i>Part 6: Calculation of service life under variable load</i>	X		
<i>Part 20: Calculation of scuffing load capacity (also applicable to bevel and hypoid gears) — Flash temperature method</i> (replaces: ISO/TR 13989-1)		X	
<i>Part 21: Calculation of scuffing load capacity (also applicable to bevel and hypoid gears) — Integral temperature method</i> (replaces: ISO/TR 13989-2)		X	
<i>Part 22: Calculation of micropitting load capacity</i> (replaces: ISO/TR 15144-1)		X	
<i>Part 30: Calculation examples for the application of ISO 6336 parts 1,2,3,5</i>			X
<i>Part 31: Calculation examples of micropitting load capacity</i> (replaces: ISO/TR 15144-2)			X

This document and the other parts of the ISO 6336 series provide a coherent system of procedures for the calculation of the load capacity of cylindrical involute gears with external or internal teeth. The ISO 6336 series 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 analysed 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 [4.1.16](#)). 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 damages which are covered by the ISO 6336 series. 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 document 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 the other documents in the ISO 6336 series, 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 the knowledge of similar designs and the awareness of the effects of the items discussed.

The formulae in the ISO 6336 series are intended to establish a uniformly acceptable method for calculating the load capacity of cylindrical gears with straight or helical involute teeth.

The ISO 6336 series includes procedures based on testing and theoretical studies as referenced by each method. The methods are validated for:

- normal working pressure angle from 15° to 25°;
- reference helix angle up to 30°;
- transverse contact ratio from 1,0 to 2,5.

If this scope is exceeded, the calculated results will need to be confirmed by experience.

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

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

The rating formulae in the ISO 6336 series are not applicable to other types of gear tooth deterioration such as plastic deformation, case crushing and wear, and are not applicable under vibratory conditions where there can be an unpredictable profile breakdown. The ISO 6336 series does not apply to teeth finished by forging or sintering. It is not applicable to gears which have a poor contact pattern.

The influence factors presented in these methods form a method to predict the risk of damage that aligns with industry and experimental experience. It is possible that they are not entirely scientifically exact. Therefore, the calculation methods from one part of the ISO 6336 series is not applicable in another part of the ISO 6336 series unless specifically referenced.

The procedures in the ISO 6336 series provide rating formulae for the calculation of load capacity with regard to different failure modes such as pitting, tooth root breakage, tooth flank fracture, scuffing and micropitting. At pitch line velocities below 1 m/s the gear load capacity is often limited by abrasive wear (see other literature such as References [23] and [22] for further information on such calculation).

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements 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 1328-1:2013, *Cylindrical gears — ISO system of flank tolerance classification — Part 1: Definitions and allowable values of deviations relevant to flanks of gear teeth*

ISO 21771:2007, *Gears — Cylindrical involute gears and gear pairs — Concepts and geometry*

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

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 1122-1:1998 and ISO 21771:2007 apply.

ISO and IEC maintain terminological databases for use in standardization at following addresses:

- ISO Online browsing platform: available at <https://www.iso.org/obp>
- IEC Electropedia: available at <http://www.electropedia.org/>

3.2 Symbols and abbreviated terms

For the purpose of this document, the symbols and abbreviated terms given in ISO 1122-1:1998, ISO 21771:2007 and [Table 2](#) apply. Further general symbols and abbreviated terms used for the calculation of load capacity of spur and helical gears can be found in [Annex F](#).

NOTE Symbols are based on, and are extensions of, the symbols given in ISO 701 and ISO 1328-1:2013. Only symbols for quantities used for the calculation of the particular factors treated in the ISO 6336 series are given, together with the preferred units.

Table 2 — Abbreviated terms and symbols used in this document

Abbreviated terms		
Terms	Description	
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)	
CP	contact point	
EAP	end of active profile (for driving pinion: contact point E, for driving wheel: contact point A)	
Eh	material designation for case-hardened wrought steel	
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)	
IF	material designation for flame or induction hardened wrought special steel	
NT	material designation for nitrided wrought steel, nitriding steel	
NV	material designation for through-hardened wrought steel, nitrided, nitrocarburized	
SAP	start of active profile (for driving pinion: contact point A, for driving wheel: contact point E)	
St	material designation for normalized base steel ($\sigma_B < 800 \text{ N/mm}^2$)	
V	material designation for through-hardened wrought steel, alloy or carbon ($\sigma_B \geq 800 \text{ N/mm}^2$)	
Symbols		
Symbol	Description	Unit
B	total face width of double helical gear including gap width	mm
B_f	non-dimensional parameter taking into account the effect of profile form deviations on the dynamic load	—
B_k	non-dimensional parameter taking into account the effect of tip and root reliefs on the dynamic load	—
B_p	non-dimensional parameter taking into account the effect of transverse base pitch deviations on the dynamic load	—
B^*	constant (see formulae in Clause 7)	—
b	face width	mm
b_{cal}	calculated face width	mm
b_{c0}	length of tooth bearing pattern at low load (contact marking)	mm
b_{H}	half of the Hertzian contact width	mm
b_{red}	reduced face width (face width minus end reliefs)	mm
b_{s}	web thickness	mm
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		
^b The components in the plane of action are determinant.		

Table 2 (continued)

Symbols		
Symbol	Description	Unit
b_B	face width of one helix on a double helical gear	mm
$b_{I(II)}$	length of end relief	mm
C	constant, coefficient	—
	relief of tooth flank	μm
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_f	root relief	μm
C_M	correction factor (see Clause 9)	—
C_R	gear blank factor (see Clause 9)	—
C_β	crowning height	μm
$C_{I(II)}$	end relief	μm
c	constant	—
c_γ	mean value of mesh stiffness per unit face width	$\text{N}/(\text{mm} \cdot \mu\text{m})$
$c_{\gamma\alpha}$	mean value of mesh stiffness per unit face width (used for K_v , $K_{H\alpha}$, $K_{F\alpha}$)	$\text{N}/(\text{mm} \cdot \mu\text{m})$
$c_{\gamma\beta}$	mean value of mesh stiffness per unit face width (used for $K_{H\beta}$, $K_{F\beta}$)	$\text{N}/(\text{mm} \cdot \mu\text{m})$
c'	maximum tooth stiffness per unit face width (single stiffness) of a tooth pair	$\text{N}/(\text{mm} \cdot \mu\text{m})$
c'_{th}	theoretical single stiffness	$\text{N}/(\text{mm} \cdot \mu\text{m})$
D	diameter (design)	mm
D_I	deflection increment	μm
d	diameter (without subscript, reference diameter) ^a	mm
	effective twist diameter (Annex E)	mm
d_a	tip diameter ^a	mm
d_b	base diameter	mm
d_f	root diameter	mm
d_{in}	inside shaft diameter (Annex E)	mm
d_m	mean diameter for calculating reduced gear pair mass	mm
d_{Na}	active tip diameter of pinion or wheel	mm
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		
^b The components in the plane of action are determinant.		

Table 2 (continued)

Symbols		
Symbol	Description	Unit
d_{sh}	external diameter of shaft, nominal for bending deflection	mm
d_{shi}	internal diameter of a hollow shaft	mm
d_w	pitch diameter	mm
$d_{1,2}$	reference diameter of pinion (or wheel)	mm
E	modulus of elasticity	N/mm ²
E_r	reduced modulus of elasticity	N/mm ²
F	composite and cumulative deviations	µm
	force or load	N
F_{bt}	nominal transverse load in plane of action (base tangent plane)	N
$F_{bt\ eff}$	total load in the plane of action	N
F_g	total load on the gearset	N
F_m	mean transverse tangential load at the reference circle relevant to mesh calculations, $F_m = F_t K_A K_V K_V$	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 $K_{F\alpha'}$ $F_{tH} = F_t K_A K_V K_V K_{H\beta}$	N
$F_{\beta x}$	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	deviation, tooth deformation	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	load correction factor	—
$f_{f\alpha}$	profile form deviation (the value for the total profile deviation F_α may be used alternatively for this, if tolerances complying with ISO 1328-1:2013 are used)	µm
$f_{f\alpha\ eff}$	effective profile form deviation after running-in	µm
f_{ma}	mesh misalignment ^b due to manufacturing deviations	µm
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		
^b The components in the plane of action are determinant.		

Table 2 (continued)

Symbols		
Symbol	Description	Unit
$f_{pb\ eff}$	transverse effective base pitch deviation after running-in	μm
f_{pt}	transverse single pitch deviation	μm
$f_{par\ act}$	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 the ISO 6336 series, using tolerances complying with ISO 1328-1:2013)	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
$f_{\Sigma\beta}$	shaft parallelism out-of-plane deviation according to ISO/TR 10064-3:1996	—
$f_{H\beta}$	helix slope deviation (the value for the total helix deviation F_{β} may be used alternatively for this, if tolerances complying with ISO 1328-1:2013 are used)	μm
$f_{\alpha\ eff}$	effective single profile deviation	μm
f_{δ}	torsional deflection	μm
$f_{H\beta 5}$	tolerance on helix slope deviation for ISO tolerance class 5	μm
G	shear modulus	N/mm^2
g	path of contact	mm
g_{α}	length of path of contact	mm
h	tooth depth (without subscript, root circle to tip circle)	mm
h_{aP}	addendum of basic rack of cylindrical gears	mm
h_{fP}	dedendum of basic rack of cylindrical gears	mm
h_t	tooth height	mm
I	moment of inertia	mm^4
I_{CS}	integration constant	μm
J^*	moment of inertia per unit face width	$\text{kg}\cdot\text{mm}^2/\text{mm}$
K	constant, factors concerning tooth load	—
K'	constant of the pinion offset	—
K_A	application factor	—
K_{A-A}	application factor (Method A)	—
K_{A-B}	application factor (Method B)	—
K_{FA-A}	application factor for tooth root breakage along ISO 6336-3 (Method A)	—
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		
^b The components in the plane of action are determinant.		

Table 2 (continued)

Symbols		
Symbol	Description	Unit
K_{FA-B}	application factor for tooth root breakage along ISO 6336-3 (Method B)	—
$K_{F\alpha}$	transverse load factor (root stress)	—
$K_{F\alpha-A}$	transverse load factor (root stress) (Method A)	—
$K_{F\alpha-B}$	transverse load factor (root stress) (Method B)	—
K_{FFA-A}	application factor for tooth flank fracture along ISO/TS 6336-4 (Method A)	—
K_{FFA-B}	application factor for tooth flank fracture along ISO/TS 6336-4 (Method B)	—
$K_{F\beta}$	face load factor (root stress)	—
$K_{F\beta-A}$	face load factor (root stress) (Method A)	—
$K_{F\beta-B}$	face load factor (root stress) (Method B)	—
$K_{F\beta-C}$	face load factor (root stress) (Method C)	—
$K_{H\alpha}$	transverse load factor (contact stress)	—
$K_{H\alpha-A}$	transverse load factor (contact stress) (Method A)	—
$K_{H\alpha-B}$	transverse load factor (contact stress) (Method B)	—
$K_{H\beta}$	face load factor (contact stress)	—
$K_{H\beta-A}$	face load factor (contact stress) (Method A)	—
$K_{H\beta-B}$	face load factor (contact stress) (Method B)	—
$K_{H\beta-C}$	face load factor (contact stress) (Method C)	—
K_v	dynamic factor	—
K_{v-A}	dynamic factor (Method A)	—
K_{v-B}	dynamic factor (Method B)	—
K_{v-C}	dynamic factor (Method C)	—
K_γ	mesh load factor (takes into account the uneven distribution of the load between meshes for multiple transmission paths)	—
K_λ	application factor for micropitting along ISO/TS 6336-22	—
$K_{\lambda A-A}$	application factor for micropitting along ISO/TS 6336-22 (Method A)	—
$K_{\lambda A-B}$	application factor for micropitting along ISO/TS 6336-22 (Method B)	—
K_θ	application factor for scuffing along ISO/TS 6336-20/ISO/TS 6336-21	—
$K_{\theta A-A}$	application factor for scuffing along ISO/TS 6336-20/ISO/TS 6336-21 (Method A)	—
$K_{\theta A-B}$	application factor for scuffing along ISO/TS 6336-20/ISO/TS 6336-21 (Method B)	—
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		
^b The components in the plane of action are determinant.		

Table 2 (continued)

Symbols		
Symbol	Description	Unit
L_i	load at a specific point i	N
$L_{i\text{ ave}}$	average load	N
$L_{i\text{ peak}}$	peak load	N
L_S	distance between two supports	mm
$L_{\delta i}$	load intensity	N/mm
l	bearing span	mm
M	moment of a force, bending moment	Nm
m	module	mm
	mass	kg
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
N	number, exponent, resonance ratio	—
N_S	resonance ratio in the main resonance range	—
n	rotational speed	s^{-1} or min^{-1}
$n_{1,2}$	rotation speed of pinion (or wheel)	min^{-1} or s^{-1}
n_E	resonance speed	min^{-1}
P	transmitted power	kW
p	number of planet gears	—
p_{bt}	transverse pitch on the base cylinder	mm
p_{dyn}	Hertzian contact stress	N/mm^2
$p_{\text{dyn,CP}}$	local Hertzian contact stress including the load factors, K	N/mm^2
p_{et}	transverse base pitch on the path of contact	mm
p_H	nominal Hertzian contact stress	N/mm^2
q	auxiliary factor	—
	flexibility of pair of meshing teeth (see Clause 9)	$(\text{mm}\cdot\mu\text{m})/\text{N}$
q'	minimum value for the flexibility of a pair of meshing teeth	$(\text{mm}\cdot\mu\text{m})/\text{N}$
R	R_R reaction at the right side support	N
	R_L reaction at the left side support	N
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		
^b The components in the plane of action are determinant.		

Table 2 (continued)

Symbols		
Symbol	Description	Unit
r	radius (without subscript, reference radius)	mm
r_b	base radius	mm
S	safety factor	—
S_F	safety factor for tooth breakage	—
S_H	safety factor for pitting	—
S_l	slope	μrad
$\bar{S}_l$	average slope	μrad
S_Y	sum of the deflection increment values	μm
s	tooth thickness, distance between mid-plane of pinion and the middle of the bearing span	mm
s_c	film thickness of marking compound used in contact pattern determination	μm
s_R	rim thickness	mm
T	torque	Nm
$T_{1,2}$	nominal torque at the pinion (or wheel)	Nm
u	gear ratio ($z_2 / z_1 \geq 1^a$)	—
V	shear	N
v	circumferential velocity (without subscript at the reference circle)	m/s
v_r	tangential velocity	m/s
v_Σ	sum of tangential velocities	m/s
w	specific load (per unit face width, F_t / b)	N/mm
X_{CP}	load sharing factor	—
X_{fi}	distance from the left support	mm
X_i	length of face where the point load is applied	mm
x	distance between stations	mm
	profile shift coefficient	—
$x_{1,2}$	profile shift coefficient of pinion (or wheel)	—
Y	factor related to tooth root bending	—
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	—
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		
^b The components in the plane of action are determinant.		

Table 2 (continued)

Symbols		
Symbol	Description	Unit
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_β	helix angle factor (tooth root)	—
y	running-in allowance (only with subscript α or β)	μm
y	calculated deflection	μm
y_f	estimated running-in allowance for profile form deviation	μm
y_p	estimated running-in allowance for base pitch deviation	μm
y_α	running-in allowance for a gear pair	μm
y_β	running-in allowance (equivalent misalignment)	μm
Z	factor related to contact stress	—
Z_v	velocity factor	—
Z_E	elasticity factor	$(\text{N/mm}^2)^{0,5}$
Z_H	zone factor	—
Z_β	helix angle factor (pitting)	—
Z_ε	contact ratio factor (pitting)	—
z	number of teeth ^a	—
z_n	virtual number of teeth of a helical gear	—
$z_{1,2}$	number of teeth of pinion (or wheel) ^a	—
α	pressure angle (without subscript, at reference cylinder)	°
α_t	transverse pressure angle	°
α_{wt}	working transverse pressure angle at the pitch cylinder	°
α_{pn}	normal pressure angle of the basic rack for cylindrical gears	°
β	helix angle (without subscript, at reference cylinder)	°
β_b	base helix angle	°
Γ	parameter on the line of action	—
γ	auxiliary angle	°
δ	deflection	μm
$\delta_{1,2}$	deformation of bearing (1, 2) in direction of load	μm , mm
δ_g	difference in feeler gauge thickness measurement of mesh misalignment f_{ma}	μm
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing. ^b The components in the plane of action are determinant.		

Table 2 (continued)

Symbols		
Symbol	Description	Unit
δ_t	deflection of the teeth	μm
δ_{ti}	tooth deflection in the plane of action and transverse plane at a load point i	μm
ε	contact ratio, overlap ratio, relative eccentricity (see Clause 7)	—
ε_α	transverse contact ratio	—
ε_β	overlap ratio	—
ε_γ	total contact ratio, $\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$	—
ζ	roll angle	$^\circ$
ζ_{fw}	roll angle from root form diameter to working pitch point	$^\circ$
η_θ	dynamic viscosity at given temperature	$\text{N}\cdot\text{s}/\text{m}^2$
θ	temperature	$^\circ\text{C}$
ν	kinematic viscosity of the oil	mm^2/s
ν_θ	kinematic viscosity at given temperature	mm^2/s
ν_{40}	kinematic viscosity of a lubricant at 40 $^\circ\text{C}$	mm^2/s
ν_{100}	kinematic viscosity of a lubricant at 100 $^\circ\text{C}$	mm^2/s
ρ	radius of curvature	mm
	density (for steel, $\rho = 7,83 \times 10^{-6} \text{ kg}/\text{mm}^3$)	kg/mm^3
ρ_{fp}	tooth root fillet radius of the basic rack for cylindrical gears	mm
ρ_{red}	radius of relative curvature	mm
ρ_θ	density of lubricant at given temperature	kg/mm^3
ρ_{15}	density of lubricant at 15 $^\circ\text{C}$	kg/mm^3
σ	normal stress	N/mm^2
$\sigma_{H \text{ lim}}$	allowable stress number (contact)	N/mm^2
χ	running-in factor	—
χ_β	factor characterizing the equivalent misalignment after running-in	—
ω	angular velocity	rad/s
^a For external gears a , d , d_a , z_1 and z_2 are positive; for internal gearing, a , d , d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		
^b The components in the plane of action are determinant.		

4 Basic principles

4.1 Application

4.1.1 Surface durability (pitting)

ISO 6336-2 specifies the fundamental formulae for use in the determination of the surface load capacity of cylindrical gears with involute external or internal teeth. In this document, surface durability refers to pitting also known as macropitting. It includes formulae for all influences on surface durability for which quantitative assessments can be made. It applies primarily to oil-lubricated transmissions but can also be used to obtain approximate values for (slow-running) grease-lubricated transmissions, as long as sufficient lubricant is present in the mesh at all times.

4.1.2 Tooth bending strength

ISO 6336-3 specifies the fundamental formulae for use in tooth bending stress calculations for involute external or internal spur and helical gears. In service, internal gears can experience failure modes other

than tooth bending fatigue, i.e. fractures starting at the root diameter and progressing radially outward. ISO 6336-3 does not provide adequate safety against failure modes other than tooth bending fatigue. All load influences on tooth stress are included in so far as they are the result of loads transmitted by the gears and in so far as they can be evaluated quantitatively.

4.1.3 Tooth flank fracture

ISO/TS 6336-4 describes a procedure for the calculation of the tooth flank fracture load capacity of cylindrical spur and helical gears with external teeth. The method is based on theoretical and experimental investigations (References [2], [14], [15] and [19]) on case carburized test gears and gears from various industrial applications.

4.1.4 Strength and quality of materials

ISO 6336-5 describes contact and tooth-root stresses and gives numerical values for both limit stress numbers. It specifies requirements for material quality and heat treatment and comments on their influences on both limit stress numbers.

4.1.5 Service life under variable load

ISO 6336-6 specifies the information and standardized conditions necessary for the calculation of the service life (or safety factors for a required life) of gears subject to variable loading.

4.1.6 Scuffing

Formulae for scuffing resistance on cylindrical gears with involute internal or external teeth are included in ISO/TS 6336-20 and ISO/TS 6336-21.

4.1.7 Wear

So far, only limited attention has 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 the ISO 6336 series.

4.1.8 Micropitting

ISO/TS 6336-22 covers micropitting rating, which is an additional type of surface distress that may occur on gear teeth.

4.1.9 Plastic-yielding

The ISO 6336 series 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.

4.1.10 Specific applications

4.1.10.1 General

For the design of gears, it is very important to recognize that requirements for different fields of application vary considerably. The use of ISO 6336 (all parts) 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.

4.1.10.2 Vehicle final drive gears

For vehicle final drive gears, which operate at 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 %.

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.

4.1.10.3 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 material quality and manufacturing 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.

4.1.10.4 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.

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.

4.1.11 Safety factors

It is necessary to distinguish between the safety factor relative to each damage type — pitting, tooth root breakage, tooth flank fracture, scuffing, micropitting, etc. Note that at the moment for tooth flank fracture, no safety factor but a material exposure is calculated, see ISO/TS 6336-4.

For a given application, adequate gear load capacity is demonstrated by the computed values of each safety factor, S , being greater than or equal to the value of its respective minimum safety factor, $S_{\min}$.

Certain minimum values for safety factors shall be determined. Recommendations concerning these minimum values are made in the ISO 6336 series, 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.

The safety factor for pitting and tooth bending is defined as the ratio of the limiting stress number to the calculated stress.

The safety factor for scuffing is defined as the ratio of the limiting temperature to the calculated temperature.

The safety factor for micropitting is defined as the ratio of the calculated minimum specific film thickness to the permissible specific film thickness.

For tooth flank fracture, the material exposure is defined as the ratio of the local equivalent stress to the local material shear strength.

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) are not necessarily directly proportional to S . For example, load safety factors 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 each damage type, the minimum safety factors shall be chosen after careful consideration of the following influences.

Reliability of material data: ISO 6336 (all parts)-applicable materials, for which data are given in ISO 6336-5, and their abbreviations, are listed in [Table 3](#). 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 failure. This risk of failure 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 minimum 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 minimum 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 the ISO 6336 series, the value of the safety factor for each mode of failure (pitting, tooth root breakage, tooth flank fracture, scuffing, micropitting) is to be agreed upon between the parties.

Table 3 — Materials (according to ISO 6336-5)

Material	Type	Abbreviation
Normalized low carbon steels/cast steels	Wrought normalized low carbon steels	St
	Cast steels	St (cast)

Table 3 (continued)

Material	Type	Abbreviation
Cast iron materials	Black malleable cast iron (perlite structure)	GTS (perl.)
	Nodular cast iron (perlite, bainite, ferrite 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.)

4.1.12 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.

4.1.13 Manufacturing tolerances

Evaluation of rating factors should be based on the worst tolerance class limits specified for the component parts in the manufacturing process.

4.1.14 Implied accuracy

Where empirical values for rating factors are given by curves, curve fitting formulae 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.

4.1.15 Other considerations

4.1.15.1 General

In addition to the factors considered in ISO 6336 (all parts) influencing load capacity, other interrelated system factors can have a significant influence on the overall transmission performance. The following factors are particularly significant.

4.1.15.2 Lubrication

The ratings determined by the formulae for pitting, tooth root breakage and tooth flank fracture 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 to lubricate and maintain an acceptable operating temperature.

Additional information regarding tribological failure modes can be found in ISO/TS 6336-20, ISO/TS 6336-21 and ISO/TS 6336-22.

4.1.15.3 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.

4.1.15.4 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.

4.1.15.5 System dynamics

The method of analysis used in the ISO 6336 series 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 shall be recognized, however, that if the operating conditions 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. For pitting and bending fatigue ratings, the resulting load spectrum cumulative fatigue effect calculation is given in ISO 6336-6, if necessary or required.

4.1.15.6 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 shafts and 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.

4.1.15.7 Corrosion

Corrosion of gear tooth surfaces can significantly reduce the load carrying capacity of the teeth. Quantifying the extent of these reductions is beyond the scope of the ISO 6336 series.

4.1.16 Influence factors

4.1.16.1 General

The influence factors presented in the ISO 6336 series 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 formulae given in the ISO 6336 series.
- 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 maximum allowable gear flank tolerances are stated, together with reference to ISO 1328-1:2013, 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 the 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. $K_{H\beta-B1}$.

The ISO 6336 series is primarily intended for verifying the load capacity of gears for which essential calculation data are available by way of detail drawings, or in a 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. In such cases 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 minimum safety factor shall be increased to account for these additional uncertainties.

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 the manufacturer and the purchaser.

4.1.16.2 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 requires thorough knowledge and experience to use:

- 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.

4.1.16.3 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} .

4.1.16.4 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} .

4.1.17 Numerical formulae

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

4.1.18 Succession of factors in the 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 $F_t K_A K_v$;
- b) $K_{H\beta}$ or $K_{F\beta}$ with the load $F_t K_A K_v K_v$;
- c) $K_{H\alpha}$ or $K_{F\alpha}$ with the load $F_t K_A K_v K_v K_{H\beta} (F\beta)^{1)}$.

For the definition of K_v also see [4.2.1](#).

4.1.19 Determination of allowable values of gear deviations

The allowable values of flank deviations shall be determined in accordance with ISO 1328-1:2013.

4.2 Tangential load, torque and power

4.2.1 General

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 double helical gear should be treated as two helical gears arranged in parallel.

Concerning multiple path transmissions, such as planetary gear systems or split path gear trains respectively when a gear drives two or more mating gears, the total tangential load is not quite evenly distributed to the various load paths (irrespective of design, circumferential velocity or accuracy of manufacture). Allowance is made for this by means of the mesh load factor K_v (see also [4.1.18](#)) unless the power through each mesh is known and used for calculations. If possible, K_v should preferably be determined by measurement; alternatively, its value may be estimated from the literature (for example DNVGL-CG-0036^[16], IEC 61400-4^[12], AGMA 6123^[13]).

For transmissions with only one load path respectively when a gear drives only one mating gear, the mesh load factor is $K_v = 1,0$.

Thus, the following applies for the mesh load factor, K_v :

- a) for transmissions with one load path: $K_v = 1,0$;
- b) for multiple path transmissions when the power through each mesh is known and used for calculations: $K_v = 1,0$;

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 [7.2.2](#)).

- c) for gear trains with multiple path transmissions with uneven distribution of the total tangential load over the individual meshes: $K_v > 1,0$.

If the operating speed is near a resonance speed, a careful study is necessary. See [Clauses 5](#) and [6](#).

4.2.2 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 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 formulae, $n_{1,2}$ is given in revolutions per minute.

$$F_t = \frac{2000 T_{1,2}}{d_{1,2}} = \frac{19\,098 \times 1\,000 P}{d_{1,2} n_{1,2}} = \frac{1\,000 P}{v} \quad (1)$$

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

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

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

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

4.2.3 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, see [5.1](#).

4.2.4 Maximum tangential load, maximum torque and maximal power

This is the maximum tangential load, $F_{t\max}$, (or corresponding torque, $T_{\max}$, corresponding power, $P_{\max}$) in the duty cycle. 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 factor due to loading corresponding to the static stress limit.

5 Application factor, K_A

5.1 General

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. Different values for K_A may be specified and used for the rating against pitting, tooth root breakage, tooth flank fracture, scuffing and micropitting as detailed below. Variable load conditions may have a different impact with respect to different failure modes. It is therefore distinguished between K_{HA} for pitting, K_{FA} for bending strength and so on.

5.2 Method A — Factor K_{A-A}

5.2.1 Factor K_{A-A}

The application factors, K_{HA-A} , for pitting, K_{FA-A} , for tooth root breakage, K_{FFA-A} , for tooth flank fracture, $K_{\theta A-A}$, for scuffing, and, $K_{\lambda A-A}$, for micropitting are determined based on a given load spectrum representing the duty cycle as detailed in the following subclauses. Load spectra in the context of the ISO 6336 series include all relevant conditions (e.g. torque, speed, temperature, direction of powerflow, ...) and are load duration distribution spectra.

The load spectra in this method are determined by means of careful measurements and a subsequent analysis of the measurement data, a comprehensive mathematical analysis of the system, or on the basis of reliable operational experience in the field of application concerned.

5.2.2 Factor K_{HA-A} for pitting along ISO 6336-2

For pitting rating, a method of calculating the effect of the loads under this condition is given in ISO 6336-6. This method is based on the calculation principles documented in ISO 6336-2. Furthermore, knowledge about the basic S-N curve for the considered damage mechanism shall be available, either from experimental investigations or from reference values included in ISO 6336-2. For calculation, the equivalent tangential load according to ISO is represented by the nominal tangential load multiplied by the application factor, K_{HA-A} , according to ISO 6336-6.

NOTE When calculating service life and/or safety according to ISO 6336-6, the application factor, K_{HA-A} , is set to 1,0.

5.2.3 Factor K_{FA-A} for tooth root breakage along ISO 6336-3

For bending fatigue rating, a method of calculating the effect of the loads under this condition is given in ISO 6336-6. This method is based on the calculation principles documented in ISO 6336-3. Furthermore, knowledge about the basic S-N curve for the considered damage mechanism shall be available, either from experimental investigations or from reference values included in ISO 6336-3. For calculation, the equivalent tangential load according to ISO is represented by the nominal tangential load multiplied by the application factor, K_{FA-A} , according to ISO 6336-6.

NOTE When calculating service life according to ISO 6336-6, the application factor, K_{FA-A} , is set to 1,0.

5.2.4 Factor K_{FFA-A} for tooth flank fracture along ISO/TS 6336-4

For tooth flank fracture it is known that loads in the area of limited life may lead to a reduced endurance limit and should therefore be considered for rating. However, no general method to determine the cumulative effect of the duty cycle is presented in the ISO 6336 series as there are no methods available for calculating this service life. Correspondingly, the calculation of an equivalent torque value from the load spectrum is not applicable for the rating against this failure mode.

Application factors, $K_{\text{FFA-A}}$, may be chosen as per Method B, see 5.3.

5.2.5 Factor $K_{\vartheta\text{A-A}}$ for scuffing along ISO/TS 6336-20/ISO/TS 6336-21

For scuffing, the rating shall be performed with all conditions in the load spectrum/duty cycle in order to identify the worst combination of speed, load and possibly other parameters such as lubricant, temperature, ageing of the lubricant, direction of powerflow, resulting in the lowest safety factor. In this case the application factor, $K_{\vartheta\text{A-A}}$, is set to 1,0.

5.2.6 Factor $K_{\lambda\text{A-A}}$ for micropitting along ISO/TS 6336-22

For micropitting rating, no general method to determine the cumulative effect of the duty cycle is presented in the ISO 6336 series as there are no methods available for calculating the service life based on generic S-N curves. Correspondingly, the calculation of an equivalent torque value from the load spectrum is not applicable for the rating against this failure mode.

In order to estimate the risk of micropitting for the whole duty cycle, it is recommended to perform an experience-based detailed analysis of the occurring operating conditions in terms of load, speed, lubricant and temperature.

If no further information is available, the application factor, $K_{\text{HA-A}}$, obtained for pitting damages may be applied as $K_{\lambda\text{A-A}}$.

5.3 Method B — Factor $K_{\text{A-B}}$

5.3.1 General

The application factors, $K_{\text{HA-B}}$, for pitting, $K_{\text{FA-B}}$, for tooth root breakage, $K_{\text{FFA-B}}$, for tooth flank fracture, $K_{\vartheta\text{A-B}}$, for scuffing, and, $K_{\lambda\text{A-B}}$, for micropitting shall be chosen as described below. The basis of Method B is a given nominal load, F_t . The application factor, $K_{\text{A-B}}$, is used to modify the value of F_t to take into account loads, additional to nominal loads, which are imposed on the gears from external sources. This adjusted load $K_{\text{A}} \cdot F_t$ is then combined with all other relevant conditions (e.g. speed, temperature, direction of powerflow, etc.).

It is recommended that the purchaser and gearbox manufacturer agree on the value of $K_{\text{A-B}}$ which shall be chosen based on experience. If no further information is available, the values for $K_{\text{A-B}}$ as described below may be used.

5.3.2 Guide values for application factor, $K_{\text{A-B}}$

The empirical values given in Table 4 may be used for any application factors ($K_{\text{HA-B}}$ for pitting, $K_{\text{FA-B}}$ for tooth root breakage, $K_{\text{FFA-B}}$ for tooth flank fracture, $K_{\vartheta\text{A-B}}$ for scuffing and $K_{\lambda\text{A-B}}$ for micropitting).

Table 4 — Application factor, K_{A}

Working characteristic of driving machine	Working characteristic of driven machine			
	Uniform	Light shocks	Moderate shocks	Heavy shocks
Uniform	1,00	1,25	1,50	1,75
Light shocks	1,10	1,35	1,60	1,85
Moderate shocks	1,25	1,50	1,75	2,00
Heavy shocks	1,50	1,75	2,00	≥2,25

The value of $K_{\text{A-B}}$ is applied to the nominal torque of the machine under consideration. Alternatively, it may be applied to the nominal torque of the driving motor as long as this corresponds to the torque demand of the driving machine.

The values only apply to transmissions, which operate outside the resonance speed range under relatively steady loading. If operating conditions involve unusually heavy loading, motors with high starting torques, intermittent service or heavy repeated shock loading, or service brakes with a torque greater than the driving motor, the safety of the static and limited life gear load capacity shall be verified (see ISO 6336-2 and ISO 6336-3).

EXAMPLE 1 Turbine/generator: in this system, short-circuit torque of up to 6 times the nominal torque can occur. Such overloads can be shed by means of safety couplings.

EXAMPLE 2 Electric motor/compressor: if pump frequency and torsional natural frequency coincide, considerable alternating stresses can occur.

EXAMPLE 3 Heavy plate and billet rolling mills: initial pass shock torque up to 6 times the rolling torque can occur.

EXAMPLE 4 Drives with synchronous motors: alternating torque up to 5 times the nominal torque can occur briefly (approximately 10 amplitudes) on starting; however, hazardous alternating torque can often be completely avoided by the appropriate detuning measures.

Information and numerical values provided here cannot be generally applied. The magnitude of the peak torque depends on the mass spring system, the forcing term, safety precautions (safety coupling, protection for unsynchronized switching of electrical machines), etc.

Thus, in critical cases, careful analysis should be demanded. It is then recommended that agreement be reached on suitable actions.

If special application factors are required for specific purposes, these shall be applied (e.g. because of a variable duty list specified in the purchase order, for marine gears according to the rules of a classification authority).

Where there are additional inertial masses, torques resulting from the flywheel effect are to be taken into consideration. Occasionally, the braking torque provides the maximum loading and thus influences the calculation of the load capacity.

It is assumed the gear materials used will have adequate overload capacity. When materials used have only marginal overload capacity, designs should be laid out for endurance at peak loading.

The K_{A-B} value for light, moderate and heavy shocks can be changed by using hydraulic couplings or torque matched elastic couplings, and especially vibration attenuating couplings when the characteristics of the couplings permit.

[Table 5](#) to [Table 7](#) show examples of working characteristics of different driving and driven machines.

Table 5 — Examples for driving machines with various working characteristics

Working characteristic	Driving machine
Uniform	Electric motor (e.g. d.c. motor), steam or gas turbine with uniform operation ^a and small rarely occurring starting torques ^b .
Light shocks	Steam turbine, gas turbine, hydraulic or electric motor (large, frequently occurring starting torques ^b).
Moderate shocks	Multiple cylinder internal combustion engines.
Heavy shocks	Single cylinder internal combustion engines.
^a Based on vibration tests or on experience gained from similar installations.	
^b See life factors, Z_{NT} , Y_{NT} , for the material; ISO 6336-2 and ISO 6336-3 shall apply. Consideration of momentarily acting overload torques, see EXAMPLES 1 to 4 following Table 4 .	

Table 6 — Industrial gears — Examples of working characteristics of driven machines

Working characteristic	Driven machines
Uniform	Steady load current generator; uniformly loaded conveyor belt or platform conveyor; worm conveyor; light lifts; packing machinery; feed drives for machine tools; ventilators; light-weight centrifuges; centrifugal pumps; agitators and mixers for light liquids or uniform density materials; shears; presses, stamping machines ^a ; vertical gear, running gear ^b .
Light shocks	Non-uniformly (i.e. with piece or batched components) loaded conveyor belts or platform conveyors; machine-tool main drives; heavy lifts; crane slewing gear; industrial and mine ventilators; heavy centrifuges; centrifugal pumps; agitators and mixers for viscous liquids or substances of non-uniform density; multi-cylinder piston pumps; distribution pumps; extruders (general); calendars; rotating kilns; rolling mill stands ^c , (continuous zinc and aluminium strip mills, wire and bar mills).
Moderate shocks	Rubber extruders; continuously operating mixers for rubber and plastics; ball mills (light); wood-working machines (gang saws, lathes); billet rolling mills ^{c,d} ; lifting gear; single cylinder piston pumps.
Heavy shocks	Excavators (bucket wheel drives); bucket chain drives; sieve drives; power shovels; ball mills (heavy); rubber kneaders; crushers (stone, ore); foundry machines; heavy distribution pumps; rotary drills; brick presses; de-barking mills; peeling machines; cold strip ^{c,e} ; briquette presses; breaker mills.
^a Nominal torque = maximum cutting, pressing or stamping torque. ^b Nominal torque = maximum starting torque. ^c Nominal torque = maximum rolling torque. ^d Torque from current limitation. ^e K_{A-B} up to 2,0 because of frequent strip cracking.	

Table 7 — High speed gears and gears of similar requirement — Examples of working characteristics of driven machines

Working characteristic	Driven machine
Uniform	Centrifugal compressors for air conditioning installation, for process gas; dynamometer — test rig; base or steady load generator and exciter; paper machinery main drives.
Light shocks	Centrifugal compressors for air or pipelines; axial compressors; centrifugal fans; peak load generators and exciters; centrifugal pumps (all types other than those listed below); axial-flow rotary pumps; paper industry; Jordan or refining machine, machines, machine auxiliary drives, stamper.
Moderate shocks	Rotary-cam blower; rotary-cam compressor with radial flow; piston compressor (3 or more cylinders); ventilator suction-fans, mining and industrial (large, frequent start-up cycles); centrifugal boiler-feed pumps; rotary cam pumps, piston pumps (3 or more cylinders).
Heavy shocks	Piston compressor (2 cylinders); centrifugal pump (with water tank); sludge pump; piston pump (2 cylinders).

6 Internal dynamic factor, K_v

6.1 General

The internal dynamic factor makes allowance for the effects of gear tooth accuracy and modifications 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 K_v is defined as the ratio of the total mesh torque at the operating speed to the mesh torque with perfect gears.

The product of the application factor K_A and the internal dynamic factor K_v is defined as the ratio of the total mesh torque at the operating speed to the 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.

6.2 Parameters affecting internal dynamic load and calculations

6.2.1 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.

6.2.2 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.

6.2.3 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.

6.2.4 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.

6.2.5 Resonances

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.

6.2.6 Application of internal dynamic factor for low loaded gears

Gears that are loaded with a line load of lower than $(F_t \cdot K_A \cdot K_v) / b = 100 \text{ N/mm}$ are typically defined as low loaded gears related to the internal dynamic factor. For gears that are loaded with a line load of lower than $(F_t \cdot K_A \cdot K_v) / b = 50 \text{ N/mm}$, a particular risk of vibration can exist dependant on gear accuracy and pitch line speeds.

Method B or C represents one model for the calculation of dynamic factor. This model is not valid for low loaded gears and values of K_{v-B} or $K_{v-C} \geq 2$ might be calculated. When cases exist where K_{v-B} or $K_{v-C} > 2$, the problem becomes significantly more complex as the possibility of tooth flank separation exists and the interaction with the entire dynamic system of stiffness and damping is highly influential.

If the gears are operated outside of their resonance condition and the calculated dynamic factor is K_{v-B} or $K_{v-C} > 2$, the dynamic factor shall be set to K_{v-B} or $K_{v-C} = 2$. This value shall be used for load capacity calculations according the ISO 6336 series, due to the described restrictions of the calculation model.

If the gears are operated within their resonance area ($N = 1$) and the calculated dynamic factor is K_{v-B} or $K_{v-C} > 2$, the restriction recommended in 6.5.2 b) is valid: "Operation in this range should generally be avoided, ...", and a specific or more detailed analysis according to Method A should be performed to establish an appropriate dynamic factor, K_v . This factor shall be used in the ISO 6336 (all parts) calculations.

In general, all dynamic investigations or detailed analyses should be defined based on experience and the requirements of the application. The gear designer may use higher or lower values than described above for K_v Method B or C both outside and inside the resonance area based on their dynamic investigations or field experience.

6.3 Principles and assumptions

In accordance with the specifications of 4.1.16, methods for determining K_v are given in 6.4, 6.5 and 6.6, from Method A (K_{v-A}) to Method C (K_{v-C}).

Given optimum profile modification appropriate to the loading, a suitable overlap ratio and transverse contact 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. The calculated value of K_v may be less than the value 1,0. If the calculated value of K_v is less than 1,0, the value 1,0 shall be used for K_v .

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 occurs 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 $(F_t \cdot K_A \cdot K_V) / b$, 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.

6.4 Methods for determination of dynamic factor

6.4.1 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 the 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.

6.4.2 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 the 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.

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 [6.5.5](#)).

Calculation of K_V by this method is not recommended 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.

6.4.3 Method C — Factor K_{V-C}

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

- The running speed range is subcritical.
- Steel solid disc wheels.
- The pressure angle $\alpha_t = 20^\circ$; $f_{pb} = f_{pt} \cos 20^\circ$ according to ISO/TR 10064-1.
- Helix angle $\beta = 20^\circ$ for helical gearing (refers to c' , $c_{\gamma\alpha}$).
- 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 $f_{pb \text{ eff}} = f_{\alpha \text{ eff}}$
- i) For assumed values for f_{pb} , y_p and $f_{pb \text{ eff}}$, see [Formulae \(18\)](#) and [\(19\)](#).

The features described in 6.5.2 a) have not been taken into consideration in the application of Method C. The influence of specific loading is taken into account.

6.5 Determination of dynamic factor using Method B: K_{v-B}

6.5.1 General

According to the preconditions and assumptions given in 6.4.2, Method B is suited for all types of transmission (spur and helical gearing with any basic rack profile and any gear tolerance class) 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 6.5.3²⁾. 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 formulae in 6.5.4 to 6.5.7 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 6.5.2 a), 6.5.4 and 6.5.5.

6.5.2 Running speed ranges

a) Subcritical range (operation below resonance speed)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 [Formula \(13\)](#). The risk of this is slight for precision helical or spur gears, if the latter have suitable profile modification (gears to tolerance class 5 of ISO 1328-1:2013 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 $(F_t \cdot K_A \cdot K_v) / b < 50 \text{ N}/\text{mm}$, a particular risk of vibration exists (under some circumstances, with the separation of working tooth flanks) — above all for spur or helical gears of coarse tolerance class running at a higher speed.

b) Main resonance range (operation close to resonance speed)

Operation in this range should generally be avoided, especially for spur gears with unmodified tooth profiles, or helical gears of tolerance class 6 or coarser as specified in ISO 1328-1:2013. Helical gears of high accuracy with a high total contact ratio can function satisfactorily in this

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 6.5.6.

3) For a definition of N , see [Formula \(9\)](#). In practice, the calculated resonance sector is broadened to ensure a safe margin. See [Formulae \(10\)](#) and [\(11\)](#) and the preamble thereto.

sector. Spur gears of tolerance class 5 or better as specified in ISO 1328-1:2013 shall have suitable profile modification.

c) Supercritical range (operation above resonance speed)

The same limitations on gear tolerance class 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 the meshing frequency are usually small.

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 6.4.2). If the critical frequency is near the frequency of rotation, the associated effective value of K_v can exceed the value calculated using Formula (21) by up to 100 %. This condition should be avoided.

6.5.3 Determination of resonance running speed (main resonance) of a gear pair

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

If a specific transmission design does not align with the simplifications of Method B as described in 6.4.2, then Method A should be used. A method for deriving approximate values is specified for the following cases, see 6.5.8:

- a) two neighbouring gears rigidly joined together;
- b) one big wheel driven by two pinions;
- c) simple planetary gears;
- d) 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 Formula (6) because of stiffnesses and inertia which have not been included (shafts, bearings, housings, etc.) and as

a result of damping. For reasons of safety, the resonance ratio in the main resonance range is defined by the following limits.

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

The lower limit, N_S , is determined:

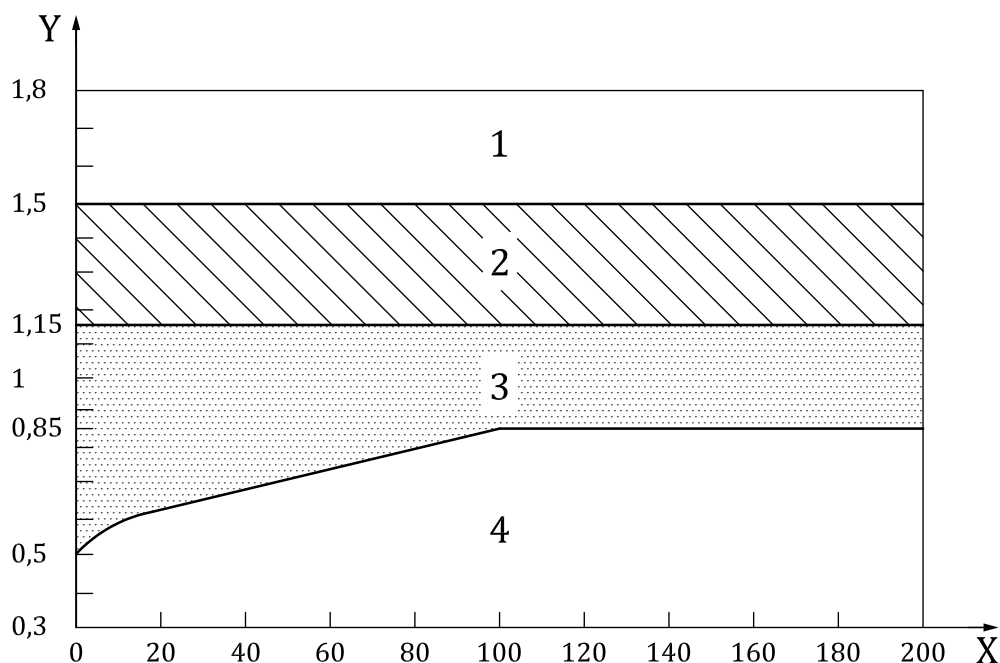
- a) at loads such that $(F_t \cdot K_A \cdot K_\gamma) / b$ is less than 100 N/mm, as

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

- b) for loads where $(F_t \cdot K_A \cdot K_\gamma) / b \geq 100$ 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, $\frac{F_t \cdot K_A \cdot K_\gamma}{b}$ N/mm
Y resonance ratio, N
1 supercritical range
2 intermediate range
3 main resonance range
4 subcritical range

Figure 1 — Resonance range

Thus, the following ranges result for the calculation of K_v .

- a) Subcritical range, $N \leq N_S$ (see [6.5.4](#)).
b) Main resonance range, $N_S < N \leq 1,15$ (see [6.5.5](#)). This field should be avoided. Refined analysis by Method A is recommended for K_v .

- c) Intermediate range, $1,15 < N \leq 1,5$ (see 6.5.7). Refined analysis by Method A is recommended.
- d) Supercritical range, $N > 1,5$ (see 6.5.6).

6.5.4 Dynamic factor in subcritical range ($N \leq N_S$)

See 6.5.2 a) for special features; the majority of industrial gears operate in this range.

$$K_v = (N K) + 1 \quad (13)$$

where

$$K = (C_{v1} B_p) + (C_{v2} B_f) + (C_{v3} B_k) \quad (14)$$

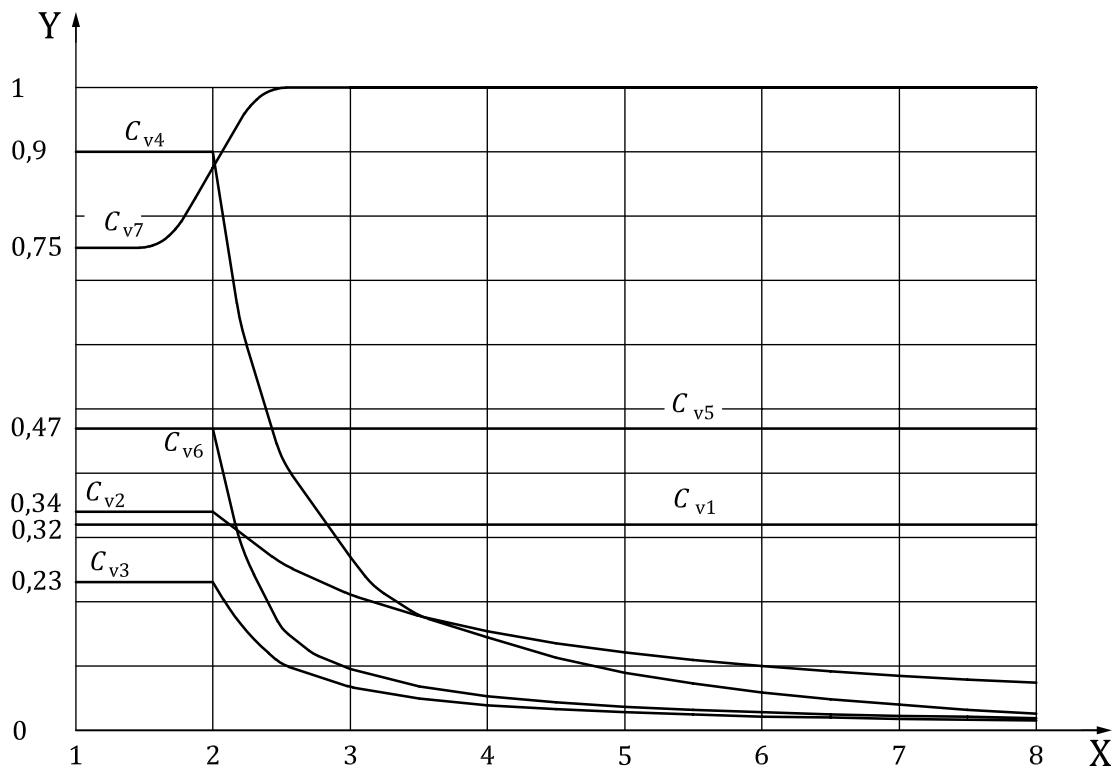
C_{v1} allows for pitch deviation effects and is assumed to be constant at $C_{v1} = 0,32$ (see Figure 2);

C_{v2} allows for tooth profile deviation effects and can be read from Figure 2 or determined in accordance with Table 8;

C_{v3} allows for the cyclic variation effect in mesh stiffness and can be read from Figure 2 or determined in accordance with Table 8.

Table 8 — Formulae for calculation of factors C_{v1} to C_{v7} and C_{ay} for the determination of K_{v-B} , Method B (formulae relate to curves in Figures 2 and 3)

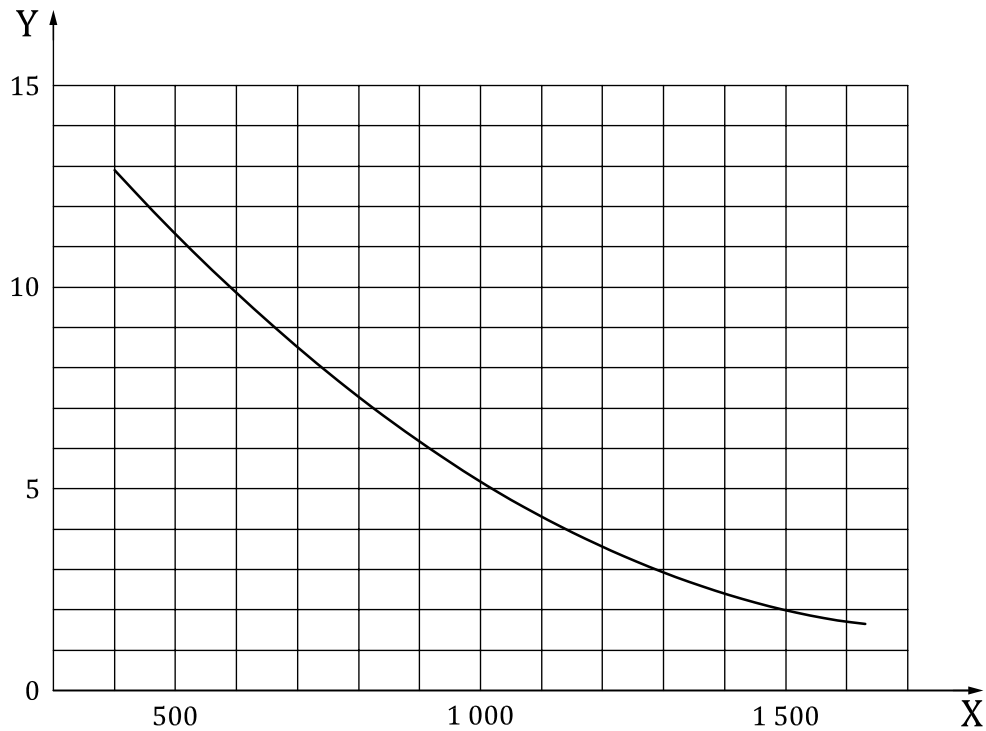
	$1 < \varepsilon_\gamma \leq 2$	$\varepsilon_\gamma > 2$	
C_{v1}	0,32	0,32	
C_{v2}	0,34	$\frac{0,57}{\varepsilon_\gamma - 0,3}$	
C_{v3}	0,23	$\frac{0,096}{\varepsilon_\gamma - 1,56}$	
C_{v4}	0,90	$\frac{0,57 - 0,05 \varepsilon_\gamma}{\varepsilon_\gamma - 1,44}$	
C_{v5}	0,47	0,47	
C_{v6}	0,47	$\frac{0,12}{\varepsilon_\gamma - 1,74}$	
	$1 < \varepsilon_\gamma \leq 1,5$	$1,5 < \varepsilon_\gamma \leq 2,5$	$\varepsilon_\gamma > 2,5$
C_{v7}	0,75	$0,125 \sin [\pi(\varepsilon_\gamma - 2)] + 0,875$	1,0
NOTE 1 $C_{ay} = \frac{1}{18} \left(\frac{\sigma_{Hlim}}{97} - 18,45 \right)^2 + 1,5$			
NOTE 2 When the material of the pinion (1) is different from that of the wheel (2), C_{ay1} and C_{ay2} are calculated separately, then $C_{ay} = 0,5 (C_{ay1} + C_{ay2})$.			



Key
X contact ratio, ϵ_γ
Y factor, C_v

NOTE For the formulae used for calculation, see [Table 8](#).

Figure 2 — Values of C_{v1} to C_{v7} for determination of K_{v-B} (Method B)

**Key**

X allowable stress number, σ_{Hlim} , N/mm²

Y tip relief, C_{ay} , µm

NOTE When the pinion material (1) is different from the wheel (2) then $C_{ay} = 0,5 (C_{ay1} + C_{ay2})$.

Figure 3 — Tip relief C_{ay} produced by running-in (see Table 8 for calculation)

B_p , B_f and B_k are non-dimensional parameters to take into account the effect of tooth deviations and profile modifications on the dynamic load⁴⁾.

$$B_p = \frac{c' f_{pb \text{ eff}}}{K_A K_\gamma (F_t / b)} \quad (15)$$

$$B_f = \frac{c' f_{f\alpha \text{ eff}}}{K_A K_\gamma (F_t / b)} \quad (16)$$

$$B_k = \left| 1 - \frac{c' \cdot \min(C_{a1} + C_{f2}, C_{a2} + C_{f1})}{K_A K_\gamma (F_t / b)} \right| \quad (17)$$

The effective base pitch and profile deviations are those of the “run-in” pinion and wheel. Initial deviations are generally modified during early service (running-in). The values of $f_{pb \text{ eff}}$ and $f_{f\alpha \text{ eff}}$ are determined by deducting estimated running-in allowances (y_p and y_f) as follows:

$$f_{pb \text{ eff}} = f_{pb} - y_p \quad (18)$$

4) Formula (17) is not suitable for the determination of an “optimum” tip relief C_a . The amount C_a of tip relief may only be used in Formula (17) for gears of ISO tolerance class in the range 1 to 5 as specified in ISO 1328-1:2013. For gears in the range 6 to 11, $B_k = 1,0$. Also see 4.1.16.

$$f_{f\alpha \text{ eff}} = f_{f\alpha} - y_f \quad (19)$$

Considerations of probability suggest that, in general, magnitudes of transmission deviation will not be greater than the allowable values of f_{pb} and $f_{f\alpha}$ for the wheel, which are larger. They are therefore used in [Formulae \(18\)](#) and [\(19\)](#) respectively; these are usually the values for the largest wheel.

In the event that neither experimental nor service data on relevant material running-in characteristics are available (Method A), it can be assumed that $y_p = y_{\alpha}$, with y_{α} from [Figure 17](#) or [18](#) or [8.3.6.2](#). y_f can be determined in the same way as y_{α} when the profile deviation $f_{f\alpha}$ is used instead of base pitch deviation f_{pb} .

C_a is the design amount for profile modification (tip relief at the beginning and end of tooth engagement).

A value C_{ay} resulting from running-in is to be substituted for C_a in [Formula \(17\)](#) in the case of gears without a specified profile modification. The value of C_{ay} can be obtained from [Figure 3](#) or calculated according to [Table 8](#).

See [Clause 9](#) for single tooth stiffness c' .

6.5.5 Dynamic factor in main resonance range ($N_S < N \leq 1,15$)

Subject to restriction (see [6.5.2 b](#)), this factor is equal to

$$K_v = (C_{v1} B_p) + (C_{v2} B_f) + (C_{v4} B_k) + 1 \quad (20)$$

See [6.5.4](#) for details regarding C_{v1} , C_{v2} , B_p , B_f and B_k .

C_{v4} takes into account resonant torsional oscillations of the gear pair, excited by cyclic variation of the mesh stiffness. Its value can be taken from [Figure 2](#) or calculated as indicated in [Table 8](#).

NOTE The dynamic factor at this speed is strongly influenced by damping. The real value of the dynamic factor can deviate from the calculated value [see [Formula \(20\)](#)] by up to 40 %. This is especially true for spur gears with incorrectly designed profile modification.

6.5.6 Dynamic factor in supercritical range ($N \geq 1,5$)

Most high precision gears used in turbine and other high-speed transmissions operate in this sector; see [6.5.2 c](#)) for features.

$$K_v = (C_{v5} B_p) + (C_{v6} B_f) + C_{v7} \quad (21)$$

In this range, the influences on K_v of C_{v5} and C_{v6} correspond to those of C_{v1} and C_{v2} on K_v in the subcritical range. See [6.5.4](#) for data on these factors and on B_p and B_f .

C_{v7} takes into account the component of load which, due to mesh stiffness variation, is derived from tooth bending deflections during substantially constant speed.

C_{v5} , C_{v6} and C_{v7} can be obtained from [Figure 2](#) or calculated according to [Table 8](#).

6.5.7 Dynamic factor in intermediate range ($1,15 < N < 1,5$)

In this range, the dynamic factor is determined by linear interpolation between K_v at $N = 1,15$ as specified in [6.5.5](#) and K_v at $N = 1,5$ as specified in [6.5.6](#).

$$K_v = K_v(N=1,15) + \frac{K_v(N=1,15) - K_v(N=1,5)}{0,35} (1,5 - N) \quad (22)$$

See [6.5.5](#) and [6.5.6](#) for details and explanatory notes.