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

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



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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

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

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

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

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

Introduction

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

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

1 Scope

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

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

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

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

2 Normative references

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

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

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

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

ISO 1328-1:1995, *Cylindrical gears — ISO system of accuracy — Part 1: Definitions and allowable values of deviations relevant to corresponding flanks of gear teeth¹⁾*

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

1) This was corrected and reprinted in 1997.

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

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

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

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

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

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

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

3 Terms and definitions

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

Table 1 — Symbols and abbreviations used in this International Standard

Symbol	Description or term	Unit
a	centre distance ^a	mm
b	facewidth	mm
b_B	facewidth of an individual helix of a double helical gear	mm
b_H	facewidth (pitting)	mm
b_F	facewidth (tooth root)	mm
b_{red}	reduced facewidth (facewidth minus end reliefs)	mm
b_s	web thickness	mm
$b_{l(II)}$	length of end relief	mm
c_γ	mean value of mesh stiffness per unit facewidth	N/(mm·µm)
c'	maximum tooth stiffness of one pair of teeth per unit facewidth (single stiffness)	N/(mm·µm)
$d_{a1,2}$	tip diameter of pinion (or wheel)	mm
$d_{an1,2}$	tip diameter of pinion (or wheel) of virtual spur gear	mm
$d_{b1,2}$	base diameter of pinion (or wheel)	mm
$d_{bn1,2}$	base diameter of pinion (or wheel) of virtual spur gear	mm
$d_{en1,2}$	diameter of circle through outer point of single pair tooth contact of pinion, wheel of virtual spur gear	mm
$d_{f1,2}$	root diameter of pinion, wheel	mm
$d_{m1,2}$	diameter at mid-tooth depth of pinion, wheel	mm
$d_{n1,2}$	reference diameter of pinion, wheel of virtual spur gear	mm
d_{sh}	nominal shaft diameter for bending	mm
d_{shi}	internal diameter of hollow shaft	mm
$d_{w1,2}$	pitch diameter of pinion, wheel	mm
d_{Nf2}	diameter of a circle near the tooth-roots, containing the limits of the usable flanks of an internal gear or the larger external gear of a mating gear	mm
$d_{1,2}$	reference diameter of pinion, wheel	mm
$f_{f\text{ eff}}$	effective profile form deviation	µm
$f_{f\alpha}$	profile form deviation (the value for the total profile deviation F_α may be used alternatively for this, if tolerances complying with ISO 1328-1 are used)	µm
f_{ma}	helix deviation due to manufacturing inaccuracies	µm
f_{pb}	transverse base pitch deviation (the values of f_{pt} may be used for calculations in accordance with ISO 6336:1996, using tolerances complying with ISO 1328-1)	µm
$f_{pb\text{ eff}}$	effective transverse base pitch deviation	µm
f_{sh}	helix deviation due to elastic deflections	µm
$f_{H\beta}$	tooth alignment deviation (not including helix form deviation)	µm
g_α	path length of contact	mm
h	tooth depth	mm

Table 1 (continued)

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

Table 1 (continued)

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

Table 1 (continued)

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

Table 1 (continued)

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

Table 1 (continued)

Symbol	Description or term	Unit
σ_S	yield point	N/mm ²
$\sigma_{0,2}$	0,2 % proof stress	N/mm ²
χ^*	relative stress gradient in the root of a notch	mm ⁻¹
χ_p^*	relative stress gradient in a smooth polished test piece	mm ⁻¹
χ_T^*	relative stress gradient in the root of the standard reference test gear	mm ⁻¹
$\omega_{1,2}$	angular velocity of pinion, wheel	rad/s
^a For external gear pairs, a , u , z_1 and z_2 are positive; for internal gear pairs, a , u and z_2 are negative, and z_1 positive.		

4 Application

4.1 Design, specific applications

4.1.1 General

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

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

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

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

- a) If a more refined method of calculation is desired or if compliance with the restrictions given in 4.1 is for any reason impractical, relevant factors may be evaluated according to the basic standard or another application standard.
- b) Factors derived from reliable experience or test data may be used instead of individual factors according to this International Standard. Concerning this, the criteria for Method A in 4.1.8.1 of ISO 6336-1:1996 are applicable.

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

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

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

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

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

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

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

4.1.2 Gear data

This International Standard is applicable within the following constraints.

a) Types of gear:

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

b) Range of speeds:

- n_1 less than or equal to $3\,600 \text{ min}^{-1}$ (synchronous speed of two-pole motor at 60 Hz current frequency)²⁾;
- subcritical range of speed (see K_v in 5.6);
- at speeds of $v < 1 \text{ m/s}$, gear load capacity is often limited by wear.

c) Gear accuracy:

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

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

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

e) Range of helix angles:

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

4.1.3 Pinion and pinion shaft

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

4.1.4 Wheel blank, wheel rim

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

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

4.1.5 Materials

These include steels, nodular cast iron and grey cast iron (this affects Z_E , $\sigma_{H \text{ lim}}$, σ_{FE} , K_v , $K_{H\beta}$, $K_{F\beta}$, $K_{H\alpha}$ and $K_{F\alpha}$). For materials and their abbreviations used in this International Standard, see Table 2.

Table 2 — Materials

Material	Abbreviation
Steel ($\sigma_B < 800 \text{ N/mm}^2$)	St
Cast steel, alloy or carbon, ($\sigma_B \geq 800 \text{ N/mm}^2$)	St (cast)
Through-hardening steel, alloy or carbon, through hardened ($\sigma_B \geq 800 \text{ N/mm}^2$)	V
Grey cast iron	GG
Nodular cast iron (pearlitic, bainitic, ferritic structure)	GGG (perl., bai., ferr.)
Black malleable cast iron (pearlitic structure)	GTS (perl.)
Case-hardened steel, case hardened	Eh
Steel and GGG, flame or induction hardened	IF
Nitriding steel, nitrided	NT (nitr.)
Through-hardening and case-hardening steel, nitrided	NV (nitr.)
Through-hardening and case-hardening steel, nitrocarburized	NV (nitrocar.)

4.1.6 Lubrication

The calculation procedures are valid for oil lubricated gears having sufficient lubricant of suitable viscosity at the gear mesh and when the working temperature is also suitable (this affects lubricant film formation, i.e. the factors Z_L , Z_v and Z_R).

4.2 Safety factors

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

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

Choice of the value of a safety factor should be based on the degree of confidence in the reliability of the available data and the consequences of possible failures.

Important factors to be considered are

- that the validity of the material values in ISO 6336-5 is for 1 % probability of damage,
- the specified quality and the effectiveness of quality control at all stages of manufacture,
- the accuracy of specification of the service duty and external conditions, and
- that tooth breakage is often considered to be a greater hazard than pitting.

Therefore, the chosen value for $S_{F \text{ min}}$ should be greater than the value chosen for $S_{H \text{ min}}$.

For calculation of the actual safety factor, see 6.1.5 (S_H , pitting) and 7.1.4 (S_F , tooth breakage). For minimum safety factors see 6.12 (pitting) and 7.9 (tooth breakage). However, it is recommended that the minimum values of the safety factors should be agreed upon between the purchaser and the manufacturer.

4.3 Input data

The following data shall be available for the calculations:

a) gear data:

$a, z_1, z_2, m_n, d_1, d_{a1}, d_{a2}, b, b_H, b_F, x_1, x_2, \alpha_n, \beta, \varepsilon_\alpha, \varepsilon_\beta$ (see ISO 53, ISO 54) (see 4.4 for definition of b, b_H and b_F face widths);

b) cutter basic rack tooth profile:

h_{a0}, ρ_{a0} ;

c) design and manufacturing data:

$C_{a1}, C_{a2}, f_{pb}, S_{H \min}, S_{F \min}, Ra_1, Ra_2, Rz_1, Rz_2$;

materials, material hardnesses and heat treatment details; gear accuracy grades, bearing span l , positions of gears relative to bearings; dimensions of pinion shaft d_{sh} and, when applicable, helix modification (crowning, end relief);

d) power data:

P or T or F_t, n_1, v_1 , details of driving and driven machines.

Requisite geometrical data can be calculated according to national standards.

Information to be exchanged between manufacturer and purchaser should include data specifying material preferences, lubrication, safety factor and externally applied forces due to vibrations and overloads (application factor).

4.4 Face widths

The following face widths have to be distinguished.

- b : the smaller of the facewidths of pinion and wheel measured at the pitch circles (for a double helical gear $b_H = 2 b_B$). Chamfers or rounding of tooth ends are to be ignored. Where the facewidths are offset, the length of the face in contact shall be used.
- b_H : the facewidth at the pitch cylinder of the gear (for a double helical gear $b_H = 2 b_B$). When the facewidth b_H is larger than that of its mating gear, b_H shall be based on the smaller facewidth, ignoring any intentional transverse chamfers or tooth-end rounding. Neither unhardened portions of surface-hardened gear tooth flanks nor the transition zones shall be included. Where the facewidths are offset, the length of the face in contact shall be used.
- b_F : the facewidth at the root cylinder of the gear (for a double helical gear $b_F = 2 b_B$). When the facewidth b_F is larger than that of its mating gear, b_F shall be based on the smaller facewidth plus a length, not exceeding one module of any extension at each end. However, if it is foreseen that because of crowning or because end relief contact does not extend to the end of face, then the smaller facewidth shall be used for both pinion and wheel. Where the facewidths are offset, the length of the face in contact shall be used.

4.5 Numerical equations

The units listed in clause 3 shall be used in all calculations. Information which will facilitate the use of this International Standard is provided in annex C of ISO 6336-1:1996.

5 Influence factors

5.1 General

The influence factors K_V , $K_{H\alpha}$, $K_{H\beta}$, $K_{F\alpha}$ and $K_{F\beta}$ are all dependent on the tooth load. Initially this is the applied load (nominal tangential load multiplied by the application factor).

The factors are also interdependent and shall therefore be calculated successively as follows:

- K_V with the applied tangential load $F_t K_A$;
- $K_{H\beta}$ or $K_{F\beta}$ with the recalculated load $F_t K_A K_V$;
- $K_{H\alpha}$ or $K_{F\alpha}$ with the applied tangential load $F_t K_A$.

When a gear drives two or more mating gears, it is necessary to substitute K_A by $K_A K_\gamma$. If possible, the mesh load factor, K_γ , should be determined by measurement; alternatively, its value may be estimated from the literature.

5.2 Nominal tangential load, F_t , nominal torque, T , nominal power, P

The nominal tangential load, F_t , is determined in the transverse plane at the reference cylinder. It is 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.

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

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

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

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

$$\omega_{1,2} = \frac{\pi n_{1,2}}{30} = \frac{2\,000\, v}{d_{1,2}} = \frac{n_{1,2}}{9\,549} \quad (5)$$

5.3 Non-uniform load, non-uniform torque, non-uniform power

When the transmitted load is not uniform, consideration should be given not only to the peak load and its anticipated number of cycles, but also to intermediate loads and their numbers of cycles. This type of load is classed as a *duty cycle* and may be represented by a load spectrum. In such cases, the cumulative fatigue effect of the duty cycle is considered in rating the gear set. A method of calculating the effect of the loads under this condition is given in ISO TR 10495.

5.4 Maximum tangential load, $F_{t \max}$, maximum torque, $T_{\max}$, maximum power, $P_{\max}$

This is the maximum tangential load $F_{t \max}$, (or corresponding torque, $T_{\max}$, corresponding power, $P_{\max}$) in the variable duty range. Its magnitude can be limited by a suitably responsive safety clutch. $F_{t \max}$, $T_{\max}$ and $P_{\max}$ shall be known when safety from pitting damage and from sudden tooth breakage due to loading corresponding to the static stress limit is determined (see 5.5).

5.5 Application factor, K_A

5.5.1 General

The factor K_A adjusts the nominal load F_t , in order to compensate for incremental gear loads from external sources. These additional forces 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.

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

5.5.2 Method A — Factor K_{A-A}

K_A is determined in this method by means of careful measurements and a comprehensive analysis of the system, or on the basis of reliable operational experience in the field of application concerned (see 5.3).

5.5.3 Method B — Factor K_{A-B}

If no reliable data, obtained as described in 5.5.2, is available, or even as early as the first design phase, it is possible to use the guideline values for K_A as described in annex C.

5.6 Internal Dynamic Factor, K_V

5.6.1 General

The dynamic factor relates the total tooth load, including internal dynamic effects of a “multi-resonance” system, to the transmitted tangential tooth load.

Method B of ISO 6336-1:1996 with modifications is used in this International Standard. When agreed between manufacturer and purchaser, or when determining the catalogue presentation of the capacities of catalogue enclosed drives, Method E of ISO 6336-1:1996 may be used to estimate the dynamic factor.

In this procedure it is assumed that the gear pair consists of an elementary single mass and spring system comprising the equivalent masses of pinion and wheel, and the mesh stiffness of the contacting teeth. It is also assumed that each gear pair functions as a single stage pair, i.e. the influence of other stages in a multiple-stage gear system is ignored. This assumption is only tenable when the torsional stiffness (measured at the base radius of the gears) of the shaft common to a wheel and a pinion is less than the mesh stiffness. See 5.6.3 and annex A for the procedure dealing with very stiff shafts.

Forces caused by torsional vibrations of the shafts and coupled masses are not covered by K_V . These forces should be included with other externally applied forces (e.g. with the application factor).

In multiple mesh gear trains, 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, analysis by Method A is recommended. See ISO 6336-1:1996, 6.3.1.

The specific load for the calculation of K_V is $(F_t K_A)/b$.

If $(F_t K_A)/b > 100 \text{ N/mm}$, then $F_m/b = (F_t K_A)/b$,

if $(F_t K_A)/b \leq 100 \text{ N/mm}$, then $F_m/b = 100 \text{ N/mm}$.

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

5.6.2 Calculation of the parameters required for evaluation of K_v

5.6.2.1 Calculation of the equivalent mass, m_{red}

a) Calculation of the equivalent mass m_{red} of a single-stage gear pair

$$m_{\text{red}} = \frac{J_1^* J_2^*}{J_1^* r_{b2}^2 + J_2^* r_{b1}^2} \quad (6)$$

where

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

$J_{1,2}^*$ are the polar moments of inertia per unit facewidth;

$r_{b1,2}$ are the base radii ($= 0,5 d_{b1,2}$).

b) Calculation of equivalent mass, m_{red} , of a multi-stage gear pair

See annex A.

c) Calculation of equivalent mass, m_{red} , of gears of less common designs

For information on the following cases, see A.1.2:

- pinion shaft with diameter at mid-tooth depth, d_{m1} , about equal to the shaft diameter;
- two rigidly connected, coaxial gears;
- one large wheel driven by two pinions;
- planetary gears;
- idler gears.

5.6.2.2 Determination of the resonance running speed (main resonance) of a gear pair

a) Resonance running speed, n_{E1} , of the pinion, in reciprocal minutes:

$$n_{E1} = \frac{30 \times 10^3}{\pi z_1} \sqrt{\frac{c_\gamma}{m_{\text{red}}}} \text{ min}^{-1} \quad (7)$$

with c_γ from annex B.

b) Resonance ratio, N

The ratio of pinion speed to resonance speed, the resonance ratio, N , is determined as follows.