INTERNATIONAL STANDARD

ISO 10300-1

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Calculation of load capacity of bevel gears —

Part 1: Introduction and general influence factors

Calcul de la capacité de charge des engrenages coniques —

Partie 1: 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.

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 part of ISO 10300 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 10300-1 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation.*

ISO 10300 consists of the following parts, under the general title *Calculation of load capacity of bevel gears*:

— *Part 1: Introduction and general influence factors*

— *Part 2: Calculation of surface durability (pitting)*

— *Part 3: Calculation of tooth root strength*

Annex A forms an integral part of this part of ISO 10300. Annex B and annex C are for information only.

Introduction

Parts 1, 2 and 3 of ISO 10300, taken together with ISO 6336-5, are intended to establish general principles and procedures for the calculation of the load capacity of bevel gears. Moreover, ISO 10300 has been designed to facilitate the application of future knowledge and developments, as well as the exchange of information gained from experience.

Several methods for the calculation of load capacity and various factors are specified by ISO 10300, whose guidelines are complex, yet flexible. There could be differences of up to 20 % to 25 % between the results of calculations carried out using method B with method B1 and method B2 with method C. The combined use of methods B2 and C, considered the methods of greater simplification, provides a more conservative safety factor. Detailed or simplified methods can be included, as appropriate, in application standards derived from ISO 10300 in the fields of industrial and marine gears. However, it must be stressed that the methods' use for specific applications demands not only experience with combined calculation methods, but also a realistic and knowledgeable appraisal of all relevant considerations, as well as appropriate safety factors.

The more detailed calculation methods of ISO 10300 are intended for the recalculation of the load capacity limits of gears where all important data, such as existing gear sets and completed gear designs, is known. The approximate methods of ISO 10300 are to be used for preliminary estimates of gear capacity where the final details of the gear design are as yet unknown.

The procedures covered by ISO 10300 are based on both testing and theoretical studies. However, the results obtained from its rating calculations may not be in good agreement with certain, previously accepted, gear-calculation methods.

ISO 10300 provides methods by which different gear designs can be compared. It is not intended to ensure the performance of assembled gear-drive systems. Neither is it intended for use by the average engineer. Rather, it is aimed at the experienced gear designer capable of selecting reasonable values for the factors in these formulae, based on knowledge of similar designs and on awareness of the effects of the items discussed.

Calculation of load capacity of bevel gears —

Part 1: Introduction and general influence factors

1 Scope

The formulae in ISO 10300 are intended to establish uniformly acceptable methods for calculating the pitting resistance and bending-strength capacity of straight and helical (skew), zerol and spiral bevel gears except hypoid gears. They are applicable equally to tapered depth and uniform depth teeth.

The formulae take into account the known major factors influencing gear-tooth pitting and fractures at the root fillet, as well as allowing for the inclusion of new factors at a later date. The rating formulae are not applicable to other types of gear-tooth deterioration such as plastic yielding, micropitting, case crushing, welding, and wear. The bending-strength formulae are applicable to fractures at the tooth fillet, but not to those on the tooth-working profile surfaces, nor to failure of the gear rim or of the gear blank through the web and hub. Pitting resistance and bending-strength capacity rating systems for a particular category of bevel gear can be established by selecting proper values for the factors used in the general formulae. ISO 10300 is not applicable to bevel gears which have an inadequate contact pattern.

ISO 10300 is restricted to bevel gears whose virtual cylindrical gears have transverse contact ratios of $\varepsilon_{v\alpha} < 2$. The given relations are valid for gears of which the sum of addendum modification factors of pinion and gear is zero, i.e. the normal operating pressure angle of the gear pair is the same as the normal pressure angle of the basic rack.

NOTE Methods for the calculation of the load capacity of hypoid gears are indicated by the manufacturers of gear-cutting machines.

CAUTION — The user is cautioned that when the methods are used for large spiral and pressure angles, and for large face width $b > 10\ m_{mn}$, the calculated results of ISO 10300 should be confirmed by experience.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO 10300. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this part of ISO 10300 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 1122-1:1998, *Vocabulary of gear terms — Part 1: Definitions related to geometry.*

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

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

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

ISO 10300-2, *Calculation of load capacity of bevel gears — Part 2: Calculation of surface durability (pitting).*

ISO 10300-3, *Calculation of load capacity of bevel gears — Part 3: Calculation of tooth root strength.*

ISO/TR 10495, *Cylindrical gears — Calculation of service life under variable loads — Conditions for cylindrical gears according to ISO 6336.*

3 Terms and definitions

For the purposes of this part of ISO 10300, terms and definitions consistent with those given in ISO 53 and ISO 1122-1 apply.

4 Symbols and abbreviations

The symbols used in this part of ISO 10300 (see Table 1) are based on those of ISO 701, while also including symbols given in ISO 1328-1.

Table 1 — Symbols and abbreviations used in parts 1, 2 and 3 of ISO 10300

Symbol	Description or term	Unit
a_v	centre distance of virtual cylindrical gear	mm
a_{vn}	centre distance of virtual cylindrical gear in normal section	mm
b	face width	mm
b_{ce}	calculated effective face width	mm
b_e	effective face width	mm
Δb_e	heel increment of face width	mm
$\Delta b_e'$	effective heel increment of face width	mm
Δb_i	toe increment of face width	mm
$\Delta b_i'$	effective toe increment of face width	mm
c_v	dimensionless parameter	—
c_γ	mesh stiffness	N/(mm · µm)
$c_{\gamma 0}$	mesh stiffness for average conditions	N/(mm · µm)
c'	single stiffness (see ISO 6336-1)	N/(mm · µm)
c_0'	single stiffness for average conditions	N/(mm · µm)
d_e	outer pitch diameter	mm
d_m	mean pitch diameter	mm
d_v	reference diameter of virtual cylindrical gear	mm
d_{va}	tip diameter of virtual cylindrical gear	mm
d_{van}	tip diameter of virtual cylindrical gear in normal section	mm
d_{vb}	base diameter of virtual cylindrical gear	mm
d_{vbn}	base diameter of virtual cylindrical gear in normal section	mm
d_{vn}	reference diameter of virtual cylindrical gear in normal section	mm
f	distance to a line of contact	mm
f'	referred distance to middle line of contact	—
$f_{f\alpha}$	profile form deviation	µm
f_{max}	maximum distance to middle line of contact	mm
f_{pt}	single pitch deviation	µm
$f_{p\,eff}$	effective pitch deviation	µm
f_F	load correction factor	—
g_{f0}	assumed distance in locating weakest section	mm
$g_{v\alpha}$	length of path of contact of virtual cylindrical gear	mm
$g_{v\alpha n}$	length of path of contact of virtual cylindrical gear in normal section	mm
g_{xb}	distance between the centre of the cutter edge radius and the centreline of the gear measured along the tool reference plane	mm
g_{yb}	distance from centre of tooth tip edge radius to crown gear pitch surface measured in a direction perpendicular to pitch surface	mm

Table 1 (*continued*)

Symbol	Description	Unit
g_{za}	intermediate variable for calculating tooth strength factor	mm
g_{zb}	intermediate variable for calculating tooth strength factor	mm
g_J	intermediate variable for calculating tooth strength factor	mm
g_J'	intermediate variable for calculating tooth strength factor	mm
g_K	projected length of instantaneous line of contact in lengthwise direction of tooth	mm
g_η	length of action within the contact ellipse	mm
g_0	distance from centreline of crown gear (tool) space to tool centre tip edge radius measured in mean normal section	mm
g_0''	distance from mean section to centre of pressure measured in the lengthwise direction along the tooth	mm
h_{ae}	outer addendum	mm
h_{am}	mean addendum	mm
h_{aP}	addendum of the basic rack profile	mm
h_{a0}	tool addendum	mm
h_{fe}	outer dedendum	mm
h_{fP}	dedendum of the basic rack profile	mm
h_{fm}	mean dedendum	mm
h_{f0}	tool dedendum	mm
h_{Fa}	bending moment arm for tooth root stress (load application at tooth tip)	mm
h_N	load height from critical section	mm
k	summation index	—
k'	constant of location	—
l_b	length of contact line	mm
l_{bm}	length of middle line of contact	mm
l_{bm}'	projected length of middle line of contact	mm
m_{et}	outer transverse module	mm
m_{mn}	mean normal module	mm
m_{mt}	mean transverse module	mm
m_{red}	mass per mm facewidth reduced to the line of action of the dynamically equivalent cylindrical gears	kg/mm
m^*	relative individual gear mass per unit facewidth referred to line of action	kg/mm
n	rotational speed	min^{-1}
n_{E1}	resonance speed of pinion	min^{-1}
p	peak load	N/mm
pr	protuberance of the tool	mm
p_{max}	maximum peak load	N/mm
p^*	referred peak load	—

Table 1 (*continued*)

Symbol	Description	Unit
p_{et}	transverse base pitch of virtual cylindrical gear	mm
q	machining stock	mm
q	exponent in the formula for lengthwise curvature factor	—
q_s	notch parameter	—
q_{sT}	notch parameter of test gear	—
r_{c0}	cutter radius	mm
r_{mf}	tooth fillet radius at the mean section	mm
$r_{my\,0}$	mean transverse radius to point of load application	mm
$\Delta r_{y\,0}$	distance from pitch circle to point of load application in mean normal section	mm
s_{et}	transverse tooth thickness at the back cone	mm
s_{amn}	mean normal topland	mm
s_{mn}	mean normal circular thickness	mm
s_{pr}	amount of protuberance	mm
s_{mt}	mean transverse circular thickness	mm
s_{Fn}	tooth root chord in calculation section	mm
s_N	one-half tooth thickness at critical section	mm
u	gear ratio of bevel gear	—
u_v	gear ratio of virtual cylindrical gear	—
v_{et}	tangential speed at outer end (heel) of reference cone	m/s
$v_{et\,max}$	maximum pitch line velocity at operating pitch diameter	m/s
v_{mt}	tangential speed at reference cone at mid-facewidth	m/s
x_{hm}	profile shift coefficient	—
x_{sm}	thickness modification coefficient	—
x_N	pinion tooth strength factor	mm
y_p	running-in allowance for pitch error related to the smooth polished test piece	µm
y_J	location of point of load application for maximum bending stress on path of action	mm
y_3	location of point of load application on path of action	mm
y_α	running-in allowance for pitch error	µm
z	number of teeth	—
z_v	number of teeth of virtual cylindrical gear	—
z_{vn}	number of teeth of virtual cylindrical gear in normal section	—
A	auxiliary factor for dynamic factor	—
A_m^*	auxiliary value for load sharing factor	mm^2
A_r^*	auxiliary value for load sharing factor	mm^2

Table 1 (*continued*)

Symbol	Description	Unit
A_{sne}	outer tooth thickness allowance	mm
A_t^*	auxiliary value for load sharing factor	mm^2
B	auxiliary factor for dynamic factor	—
C	quality grade	—
C_a	tip relief	µm
C_b	correction factor for tooth stiffness for non-average conditions	—
C_F	correction factor for tooth stiffness for non-average conditions	—
C_{ZL}, C_{ZR}, C_{ZV}	factors for determining lubricant film factors	—
E	modulus of elasticity, Young's modulus	N/mm^2
E, G, H	auxiliary factors for tooth form factor	—
F	auxiliary factor for mid-zone factor	—
F_{mt}	nominal tangential force at reference cone at mid-facewidth	N
$F_{mt\,H}$	decisive tangential force at reference cone at mid-facewidth	N
HB	Brinell hardness	—
K	constant; factor concerning tooth load	—
K_v	dynamic factor	—
K_A	application factor	—
K_{F0}	lengthwise curvature factor for bending stress	—
$K_{F\alpha}$	transverse load factor for bending stress	—
$K_{F\beta}$	face load factor for bending stress	—
$K_{H\alpha}$	transverse load factor for contact stress	—
$K_{H\beta}$	face load factor for contact stress	—
$K_{H\beta\text{-be}}$	bearing factor	—
L	empirical constant used in stress correction formula	—
L_a	auxiliary factor for correction factor	—
M	empirical constant used in stress correction formula	—
N	reference speed for n_{E1}	—
N_L	number of load cycles	—
O	empirical constant used in stress correction formula	—
P	nominal power	kW
P_d	outer diametral pitch	$inch^{-1}$
Ra	= CLA = AA arithmetic average roughness	µm
R_e	outer cone distance	mm
R_m	mean cone distance	mm
Rz	mean roughness	µm
Rz_T	mean roughness of test gear	µm

Table 1 (*continued*)

Symbol	Description	Unit
Rz_{10}	mean roughness for gear pairs with ρ_{red} = 10 mm	µm
S_F	safety factor for bending stress (against breakage)	—
$S_{F\,min}$	minimum safety factor for bending stress	—
S_H	safety factor for contact stress (against pitting)	—
$S_{H\,min}$	minimum safety factor for contact stress	—
T	nominal torque	Nm
Y	tooth form factor	—
Y_i	inertia factor	—
Y_f	stress concentration and stress correction factor	—
Y_A	bevel gear adjustment factor	—
Y_B	bending stress factor	—
Y_C	compression stress factor	—
Y_{Fa}	tooth form factor for load application at tip	—
Y_{FS}	combined tooth form factor for generated gears	—
Y_J	bevel geometry factor (Method B2)	—
Y_K	bevel gear factor	—
Y_{LS}	load sharing factor (bending strength)	—
Y_{NT}	life factor of the standard test gear	—
Y_P	combined geometry factor	—
Y_R	surface factor of smooth specimen	—
Y_{RT}	surface factor of test gear with roughness of Rz_T = 10 µm	—
$Y_{R\,rel\,T}$	relative surface factor	—
Y_{Sa}	stress correction factor for load application at tooth tip	—
Y_{ST}	stress correction factor for dimensions of standard test gear	—
Y_X	size factor for tooth root stress	—
Y_δ	dynamic sensitivity factor of the gear to be determined	—
$Y_{\delta T}$	dynamic sensitivity factor of the standard test gear	—
$Y_{\delta\,rel\,T}$	relative sensitivity factor	—
Y_ε	contact ratio factor (tooth root)	—
Z_v	speed factor	—
Z_E	elasticity factor	—
Z_H	zone factor	—
Z_K	bevel gear factor (flank)	—
Z_L	lubricant factor	—
Z_{LS}	load sharing factor	—

Table 1 (*continued*)

Symbol	Description	Unit
Z_{M-B}	mid-zone factor	—
Z_{NT}	life factor of the standard test gear	—
Z_R	roughness factor for contact stress	—
Z_X	size factor	—
Z_W	work-hardening factor	—
Z_β	helix angle factor for contact stress	—
α_h	normal pressure angle at point of load application on the tooth centreline	°
α_n	normal pressure angle	°
α_{vn}	normal pressure angle of virtual cylindrical gear ($=\alpha_n$)	°
α_{vt}	transverse pressure angle of virtual cylindrical gear	°
α_{wt}	working transverse pressure angle	°
α_{Fan}	load application angle at tip circle of virtual spur gear	°
α_L	normal pressure angle at point of load application on the tooth surface	°
β_m	mean spiral angle	°
β_{vb}	helix angle at base circle of virtual cylindrical gear	°
γ_a	auxiliary angle for tooth form and tooth correction factor	°
δ	pitch angle	°
δ_a	face angle	°
δ_f	root angle	°
$\varepsilon_{v\alpha}$	transverse contact ratio of virtual cylindrical gear	—
$\varepsilon_{v\alpha n}$	transverse contact ratio of virtual cylindrical gear in normal section	—
$\varepsilon_{v\beta}$	overlap ratio of virtual cylindrical gear	—
$\varepsilon_{v\gamma}$	modified contact ratio	—
ε_N	load sharing ratio	—
θ_a	addendum angle	°
θ_f	dedendum angle	°
ξ	assumed angle in locating weakest section	°
ξ_h	one half of angle subtended by normal circular tooth thickness at point of load application	°
ρ	density	kg/mm^3
ρ_{a0}	cutter edge radius	mm
ρ_{fP}	root fillet radius of basic rack for cylindrical gears	mm
ρ_{red}	radius of relative curvature	mm
ρ_{Fn}	fillet radius at point of contact of 30° tangent	mm
ρ'	gliding thickness	mm

Table 1 (*continued*)

Symbol	Description	Unit
σ_B	tensile strength	N/mm²
σ_F	tooth root stress	N/mm²
$\sigma_{F\,lim}$	nominal stress number (bending)	N/mm²
σ_{FE}	allowable stress number (bending)	N/mm²
σ_{FP}	permissible tooth root stress	N/mm²
σ_{F0}	local tooth root stress	N/mm²
σ_H	contact stress	N/mm²
$\sigma_{H\,lim}$	allowable stress number for contact stress	N/mm²
σ_{HP}	permissible contact stress	N/mm²
σ_{H0}	nominal value of contact stress	N/mm²
$\sigma_{0,2}$	stress at 0,2% permanent elongation	N/mm²
τ	angle between tangent of root fillet at weakest point and centreline of tooth	°
ϑ	auxiliary factor for tooth form and tooth correction factors	—
ν	Poisson's ratio	—
ν_{40}, ν_{50}	nominal kinematic viscosity of the oil at 40° C and 50° C respectively	mm²/s
ω	angular velocity	rad/s
χ^X	relative stress drop in notch root	mm⁻¹
$\chi_T{}^X$	relative stress drop in notch root of test gear	mm⁻¹
Σ	shaft angle	°
Other subscripts		
0	tool	
1	pinion	
2	wheel	
x	dynamically equivalent cylindrical gears	
-A, -B, -B1, -B2, -C	value according to method A, B, B1, B2 or C	
(1), (2)	trials of interpolation	
*	value related to m_{mn} (except m^*)	

5 Application

5.1 Methods

5.1.1 General

ISO 10300 is intended mainly for the calculation of bevel gears for which the essential data is known from drawings or measurement (recalculation). At the preliminary design stage, the available data is limited, and approximations or empirical values may be used for some factors. Moreover, in certain application fields or for rough calculations some factors may be assumed as unity or constant. However, a conservative safety factor (see 5.2) should be chosen in such cases. Wherever there is disagreement, full-scale, full-load testing is preferred over any of the methods A to C, while method A, if its accuracy and reliability are proven, is preferred over method B, which in turn is preferred to method C.

5.1.2 Full-scale, full-load testing

The most valid method of predicting overall gear system performance is the full-scale, full-load testing of a specific gear drive design in order to determine its capacity. This will not require verification by calculation using any of the methods given. However, it is customary for bevel gears to be developed from a preliminary design according to methods B or C, then refined by testing to achieve optimum tooth contact, smoothness of operation and adjustability.

5.1.3 Method A

Where sufficient experience from the operation of other, similar designs is available, satisfactory guidance can be obtained by the extrapolation of the associated test results or field data. The factors involved in this extrapolation may be evaluated by the precise measurement and comprehensive mathematical analysis of the transmission system under consideration, or from field experience. All gear and load data is required to be known for the use of this method, which shall be clearly described and presented with all mathematical and test premises, boundary conditions and any specific characteristics of the method that influence the result. The accuracy and the reliability of the method must be demonstrated. Precision, for example, shall be demonstrated through comparison with other, acknowledged gear measurements. The method should be approved by both the customer and the supplier.

5.1.4 Method B

Again, where sufficient experience from the operation of other, similar designs is available, satisfactory guidance can be obtained by the extrapolation of the test results or field data associated with them. However, it is recommended that the calculation methods be used for comparison of the designs. Additionally, approximate methods are given for some factors, together with the assumptions relevant to their evaluation. The validity of these assumptions for the given working conditions shall be checked.

5.1.5 Method C

Where suitable test results, or field experience from similar designs, are unavailable for use in the evaluation of certain factors, further simplified calculation methods should be used. These are appropriate for particular fields of application or on the basis of certain premises, for example, those relevant to an acceptance test.

5.2 Safety factors

The allowable probability of failure shall be carefully weighed when choosing a safety factor, in balancing reliability against cost. If the performance of the gears can be accurately appraised by testing the unit itself under actual load conditions, lower safety factors may be permitted. The safety factors shall be determined by dividing the specific calculated strength by the specific operating stress.

In addition to this general requirement, and the special requirements relating to surface durability (pitting) and tooth root strength given, respectively, in parts 2 and 3 of ISO 10300, safety factors shall be determined only after careful consideration of the reliability of the material data and of the load values used for calculation. The allowable stress numbers used for calculation are valid for a given probability of failure, or damage (the material values in ISO 6336-5, for example, are valid for a 1 % probability of damage), the risk of damage being reduced as the safety factors are increased, and vice versa. If loads, or the response of the system to vibration, are estimated rather than measured, a larger factor of safety should be used.

The following variations shall also be taken into consideration in the determination of a safety factor:

— 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.

The appropriateness of the safety factors will thus depend on the reliability of the assumptions, such as those related to load, on which the calculations are based, as well as on the reliability required of the gears themselves, in respect of the possible consequences of any damage that might occur in the case of failure.

Supplied gears or assembled gear drives should have a minimum safety factor for contact stress, $S_{H\,min}$, value 1,0. The minimum bending-stress, $S_{F\,min}$, value should be 1,3 for spiral bevel gears, and 1,5 for straight bevel gears or where $\beta_m \leqslant 5°$.

The minimum safety factors against pitting damage and tooth breakage should be agreed between supplier and customer.

5.3 Rating factors

5.3.1 Testing

The most effective overall approach to gear-system performance management is through the full-scale, full-load testing of a proposed new design. This approach, however, is limited by its high cost. Alternatively, where sufficient experience of similar designs exists and results are available, a satisfactory solution can be obtained through extrapolation from such data. On the other hand, where suitable test results or field data are unavailable, rating-factor values should be chosen conservatively.

5.3.2 Manufacturing tolerances

Rating factors should be evaluated based on the minimum acceptable quality limits of the expected variation of component parts in the manufacturing process. The accuracy grade should be determined using ISO 1328-1 with single pitch deviation.

5.3.3 Implied accuracy

Where the empirical values for rating factors are given by curves, ISO 10300 provides curve-fitting equations to facilitate computer programming.

NOTE The constants and coefficients used in curve fitting often have significant digits in excess of those implied by the reliability of the empirical data.

5.4 Other factors to be considered

5.4.1 General

In addition to the factors considered that influence pitting resistance and bending strength, other, interrelated system factors can have an important effect on overall transmission performance. Their possible effect on the calculations should be considered.

5.4.2 Lubrication

The ratings determined by the formulae of ISO 10300 shall be valid only if the gear teeth are operated with a lubricant of proper viscosity and additive package for the load, speed, and surface finish, and if there is a sufficient quantity of lubricant on the gear teeth and bearings to lubricate and maintain an acceptable operating temperature.

5.4.3 Misalignment

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 due to elastic or thermal deflections or other influences, overall gear-system performance will be adversely affected.

5.4.4 Deflection

Deflection of gear-supporting housings, shafts, and bearings due to external overhung, transverse, and thrust loads affects tooth contact across the mesh. Since deflection varies with load, it is difficult to obtain good tooth contact at different loads. Generally, deflection due to external loads from driven and driving equipment reduces capacity, and this, as well as deflection caused by internal forces, should be taken into account when determining the actual gear tooth contact.

5.4.5 Materials and metallurgy

Most bevel gears are made from carburized case-hardened steel. Allowable stresses for this and other materials should thus be based on tests on bevel gears wherever these are available. The allowable stress numbers, which are based on different modes of steel-making and heat-treatment, shall be taken from ISO 6336-5. Hardness and tensile strength as well as the quality grade shall also be criteria for choosing allowable stress numbers.

NOTE Higher-quality steel grades indicate higher allowable stress numbers, while lower-quality grades indicate lower allowable stress numbers (see ISO 6336-5).

5.4.6 Residual stress

Any ferrous material having a case-core relationship is likely to have residual stress. If properly managed, such stress will be compressive at the tooth surface, thereby enhancing the bending-fatigue strength of the gear tooth. Shot-peening, case-carburizing and induction-hardening, if properly performed, are common methods of inducing compressive pre-stress in the surface of the gear teeth. Improper grinding techniques after heat treatment may reduce the residual compressive stresses or even introduce residual tensile stresses in the root fillets of the teeth, thereby lowering the allowable stress numbers.

5.4.7 System dynamics

The method of analysis used includes a dynamic factor, K_v, in formulae by derating the gears for increased loads caused by gear-tooth inaccuracies. Generally speaking, this provides simplified values for easy application.

The dynamic response of the system results in additional gear-tooth loads, due to the relative motions of the connected masses of the driver and the driven equipment. The application factor, K_A, is intended to account for the operating characteristics of the driving and driven equipment. It must be recognized, however, that if the operating roughness of the drive, gearbox, or driven equipment causes excitation with a frequency that is near one of the system's major natural frequencies, resonant vibrations may cause severe overloads possibly several times higher than the nominal load. Therefore, where critical service applications are concerned, performance of a vibration analysis of the complete system is recommended. This analysis shall include the total system, including driver, gearbox, driven equipment, couplings, mounting conditions and sources of excitation. Natural frequencies, mode shapes, and the dynamic response amplitudes should be calculated.

5.4.8 Contact pattern

The teeth of most bevel gears are crowned in both their profile and lengthwise directions during the manufacturing process in order to allow for deflection of the shafts and mountings. This results in a localized contact pattern during roll testing under light loads. Under design load, unless otherwise specified, the tooth contact pattern is spread over the tooth flank without concentrations of the pattern at the edges of either member. The application of the rating formulae to bevel gears manufactured under conditions in which this process has not been carried out and which do not have an adequate contact pattern may require modifications of the factors given in ISO 10300. These gears are not covered.

NOTE The total load used for contact pattern analysis can include the effects of an application factor (see annex C for a fuller explanation of tooth contact development).

5.4.9 Corrosion

Corrosion of the gear-tooth surface can have a significant detrimental effect on the bending strength and pitting resistance of the teeth. However, the quantification of the effect of corrosion on gear teeth is beyond the range of ISO 10300.

5.5 Influence and other factors in the basic formulae

Included in the basic formulae presented in ISO 10300 are factors reflecting gear geometry or established by convention, which need to be calculated in accordance with their formulae.

Also included in the formulae in ISO 10300 are factors that reflect the effects of variations in processing or the operating cycle of the unit. These are known as influence factors because they account for a number of influences. Although treated as independent, they may nevertheless influence each other to an extent that is beyond evaluation. They include the load factors, K_A, K_v, $K_{H\beta}$, $K_{F\beta}$, $K_{H\alpha}$ and $K_{F\alpha}$, as well as those factors influencing allowable stresses.

Still other factors included reflect the mathematical relationship, stress vs. life.

The influence factors can be determined by various methods of calculation. These are qualified, as needed, by the addition of subscripts A through C to the symbols. Unless otherwise specified (for example in an application standard), the more accurate method is to be preferred for important transmissions. It is recommended that supplementary subscripts be used whenever the method used for evaluation of a factor would not otherwise be readily identifiable.

For some applications, it may be necessary to choose between factors determined using alternative methods (for example, alternatives for the determination of the dynamic factor or the transverse load factor). When reporting the calculation, the method used should be indicated by extending the subscript.

EXAMPLE $K_{v\text{-}C}$, $K_{H\alpha\text{-}B}$

6 External force and application factor, K_A

6.1 Nominal tangential force, torque, power

For the purposes of ISO 10300, pinion torque is used in the fundamental stress-calculation formulae. In order to determine the bending moment on the tooth, or of the force on the tooth surface, the tangential force is calculated within the stress formula, at the reference cone at mid-facewidth, as follows.

$$F_{mt} = \frac{2000 T_{1,2}}{d_{m1,2}} \tag{1}$$

$$T_{1,2} = \frac{F_{mt}\, d_{m1,2}}{2000} = \frac{1000\, P}{\omega_{1,2}} = \frac{9549\, P}{n_{1,2}} \tag{2}$$

$$P = \frac{F_{mt}\, v_{mt}}{1000} = \frac{T_{1,2}\, \omega_{1,2}}{1000} = \frac{T_{1,2}\, n_{1,2}}{9549} \tag{3}$$

$$v_{mt} = \frac{d_{mt1,2}\, \omega_{1,2}}{2000} = \frac{d_{m1,2}\, n_{1,2}}{19098} \tag{4}$$

The nominal torque of the driven machine is decisive. This is the operating torque to be transmitted over a long period of time and under the most severe, regular, working conditions.

EXAMPLE Maximum permanent rolling torque, torque from maximum hoisting weight.

The nominal torque of the driving machine may be used if it corresponds to the required torque of the driven machine.

6.2 Variable load conditions

If the load is not uniform, a careful analysis of the gear loads should be made, in which the external and internal dynamic factors are considered. It is recommended that all the different loads that occur during the anticipated life of the gears, and the duration of each load, be determined. A method based on Miner's Rule (see ISO/TR 10495) shall be used for determining the equivalent life of the gears for the torque spectrum.

6.3 Application factor, K_A

In cases where no reliable experiences, or collective load spectra determined by practical measurement or comprehensive system analysis, are available, calculate using the nominal tangential force F_{mt} according to clause 6.1 and an application factor, K_A. This application factor makes allowance for any externally applied dynamic loads in excess of the nominal operating torque load, T_1.

6.3.1 Factors affecting external dynamic loads

In determining the application factor, account should be taken of the fact that many prime movers develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or of the driven equipment. There are many possible sources of dynamic overload which should be considered, including:

— system vibration;

— critical speed;

— acceleration torques;

— overspeed;

— sudden variations in system operation;

— braking;

— negative torques, such as those produced by retarders on vehicles, which result in loading the reverse flanks of the gear teeth.

Analysis for critical speeds within the operating range of the drive is essential. If critical speeds are present, changes in the design of the overall drive system shall be made in order to either eliminate them or provide system damping to minimize gear and shaft vibrations.

6.3.2 Establishment of application factors

Application factors are best established by a thorough analysis of service experience with a particular application. For applications such as marine gears, which are subjected to cyclic peak torques (torsional vibrations) and are designed for infinite life, the application factor can be defined as the ratio between cyclic peak torque and the nominal rated torque. The nominal rated torque is defined by the rated power and speed.

If the gear is subjected to a limited number of loads in excess of the amount of cyclic peak torque, 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.

If service experience is unavailable, a thorough analytical investigation should be made. Annex B provides approximate values if neither of these alternatives is possible.

7 Dynamic factor, K_v

7.1 General

The dynamic factor, K_v, makes allowance for the effects of gear tooth quality related to speed and load as well as for the other parameters listed below (see 7.2 to 7.6). The dynamic factor relates the total tooth load, including internal dynamic effects, to the transmitted tangential tooth load and is expressed as the sum of the internal effected dynamic load and the transmitted tangential tooth load, divided by the transmitted tangential tooth load. The parameters for the gear-tooth internal dynamic load fall into two categories: design and manufacturing.

7.2 Design

The design parameters include:

— pitchline speed;

— 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.

7.3 Manufacturing

The manufacturing parameters include:

— tooth spacing variations;

— runout of pitch surfaces with respect to the axis of rotation;

— tooth flank variations;

— compatibility of mating gear tooth elements;

— balance of parts;

— bearing fit and preload.

7.4 Transmission error

Even if the input torque and speed are constant, significant vibration of the gear masses and the resultant dynamic tooth forces can exist. These forces 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. Transmission error is defined as the deviation from uniform relative angular motion of the pair of meshing gears. It is influenced by all deviations from the ideal gear tooth form of the actual gear design, the manufacturing procedure and the operational conditions. The operational conditions include the following.

a) **Pitch line speed**. The frequencies of the excitation depend on the pitch line velocity and module.

b) **Gear mesh stiffness variations** as the gear teeth pass through the meshing cycle. This is a source of excitation especially pronounced in straight- and zerol-bevel gears. Spiral-bevel gears with a modified contact ratio > 2 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 load magnitude. Loads different from the design load will increase the 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 must have a properly designed lubrication system, enclosure, and seals to maintain a safe operating temperature and contamination-free environment.

f) **Shaft alignment.** Gear-tooth alignment is influenced by load and thermal deformations of gears, shafts, bearings and housings.

g) **Tooth friction-induced excitation.**

7.5 Dynamic response

The effects of dynamic tooth forces 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, of which the principal sources are the shaft bearings and seals, with other sources including the hysteresis of the gear shafts, viscous damping at sliding interfaces and couplings.

7.6 Resonance

When an excitation frequency (tooth meshing frequency, multiples of tooth meshing frequencies etc.) coincides, or nearly coincides, with a natural frequency of the gearing system, a resonant vibration can cause high dynamic tooth loading. When the magnitude of internal dynamic load at such a driving speed becomes large, operation in this speed range should be avoided.

7.6.1 Gear blank resonance

The gear blanks of high-speed, light-weight gearing may have natural frequencies within the operating speed range. If the gear blank is excited by a frequency close to one of its natural frequencies, the resonant deflections cause high dynamic tooth loads. There will also be the possibility of plate- or shell-mode vibrations which can cause the gear blank to fail.

If determined by method B or C, the dynamic factor, K_v, does not account for gear blank resonance.

7.6.2 System resonance

The gearbox is just one component of a system comprising a power source, gearbox, driven equipment, and interconnecting shafts and couplings. The dynamic response of this system depends on its configuration. In certain cases a system may possess a natural frequency close to the excitation frequency associated with an operating speed. Under such resonant conditions, its operation must be carefully evaluated. For critical drives, a detailed analysis of the entire system is recommended. This should then be taken into account when determining the effects on the application factor.

7.7 Calculation methods

7.7.1 General comments

A bevel gear drive is a very complicated vibrational system. The dynamic system as well as the natural frequencies which induce dynamic tooth loading cannot be determined by consideration of the pair of gears alone. The pinion shaft alignment can change considerably depending on the craftsmanship of the assembly, the backlash and the elastic deformation of gear shafts, bearings or housing. A slight change in alignment will alter the relative rotation angle of the gearing and thus the dynamic loading on the gears. Crowning in the lengthwise and profile directions may preclude true conjugate action and make tooth accuracy difficult to determine.

Under such circumstances, reliable values of the dynamic factor, K_v, can best be predicted by a mathematical model which has been satisfactorily verified by test measurements. If the known dynamic loads are added to the nominal transmitted load, then the dynamic factor can be set to unity.

In this clause, several methods for determining K_v are indicated in descending order of precision, from method A ($K_{v\text{-}A}$) to method C ($K_{v\text{-}C}$).

7.7.2 Method A, $K_{v\text{-}A}$

$K_{v\text{-}A}$ is determined by a comprehensive analysis, confirmed by experience of similar designs, using the following general procedures.

a) A mathematical model of the entire power transmission vibrational system including the gearbox is developed.

b) The transmission error of the bevel gears under load is measured, or calculated by a reliable simulation programme for transmission error of bevel gears.

c) The dynamic load response of the pinion and gear shafts is analysed with the system model, a), excited by the transmission error, b).

7.7.3 Method B, $K_{v\text{-}B}$

This method makes the simplifying assumption that the bevel-gear pair constitutes an elementary single mass and spring system comprising the combined masses of pinion and wheel, with a spring stiffness being the mesh stiffness of the contacting teeth. In accordance with this assumption, forces due to torsional vibrations of the shafts and coupled masses are not covered by $K_{v\text{-}B}$. This is realistic if other masses (apart from the gear pair) are connected by shafts of relatively low torsional stiffness. For bevel gears with a significant lateral shaft flexibility, the real natural frequency will be less than calculated.

The amount of the dynamic overloads is, among other things, a function of the accuracy of the gear, i.e. the flank-form and pitch deviations. However, where bevel gears are concerned, the determination of the flank-form deviation is difficult (not an involute form), and corresponding ISO-tolerances do not exist. On the other hand, the pitch deviation can be measured relatively easily. The simplifying assumption is therefore made that the single-pitch deviation is a representative value of the transmission error for determination of the dynamic factor.

The following data are needed for the calculation of $K_{v\text{-}B}$:

a) accuracy of gear pair (single pitch deviation);

b) mass moment of inertia of pinion and wheel (dimensions and material density);

c) tooth stiffness;

d) transmitted tangential load.

7.7.3.1 Speed ranges

Dimensionless reference speed:

$$N = \frac{n_1}{n_{E1}} \tag{5}$$

where

n_{E1} is the resonance speed according to clause 7.7.3.2.

With the aid of the reference speed N, the total speed range can be subdivided into four sectors: subcritical, main resonance, supercritical, and an intermediate sector (main resonance/supercritical).

Because of the influence of stiffness values which are not included (for example, those of shafts, bearings, gearbox), and because of the damping, the resonance speed can be above or below the speed calculated with equation (6). For reasons of safety, a resonance sector of $0{,}75 < N \leqslant 1{,}25$ is defined.

This results in the following sectors for the calculation of $K_{v\text{-}B}$:

— subcritical sector, $N \leqslant 0{,}75$, determined by method A or B;

— main resonance sector, $0{,}75 < N \leqslant 1{,}25$ (operation in this sector should be avoided, but if unavoidable, refined analysis by method A will need to be carried out);

— intermediate sector, $1{,}25 < N < 1{,}5$, determined by method A or B;

— supercritical sector, $N \geqslant 1{,}5$; determined by method A or B.

See ISO 6336-1 for further information on the speed ranges.

7.7.3.2 Resonance speed

$$n_{E1} = \frac{30 \times 10^3}{\pi z_1} \sqrt{\frac{c_\gamma}{m_{red}}} \tag{6}$$

with

$$m_{red} = \frac{m_1^* m_2^*}{m_1^* + m_2^*} \tag{7}$$

A value of $c_{\gamma 0}$ = 20 N/(mm · µm) applies to spur gears. Investigations of helical gears have shown that the stiffness is decreased when the helix angle is increased. On the other hand, the spiral arrangement of a bevel gear tooth on a conical blank leads to stiffening of helical and spiral bevel gears. Therefore, due to the lack of any better knowledge, the stiffness for a spur gear can be said to be suitable for use in average conditions ($F_{mt}K_A/b_e \geqslant$ 100 N/mm and $b_e/b \geqslant$ 0,85). Therefore c_γ can be determined as follows:

$$c_\gamma = c_{\gamma 0}\, C_F\, C_b \tag{8}$$

where

$c_{\gamma 0}$ is mesh stiffness for average conditions. A value of 20 N/(mm·µm) can be used. If experience indicates another value more appropriate for bevel gears, it should be substituted.

C_F and C_b are correction factors for non average conditions:

$$C_F = 1 \qquad \text{for } F_{mt}K_A/b_e \geqslant 100 \text{ N/mm} \tag{9}$$

$$C_F = (F_{mt}K_A/b_e)/100 \text{ N/mm} \qquad \text{for } F_{mt}K_A/b_e < 100 \text{ N/mm} \tag{10}$$

$$C_b = 1 \qquad \text{for } b_e/b \geqslant 0{,}85 \tag{11}$$

$$C_b = b_e/(0{,}85b) \qquad \text{for } b_e/b < 0{,}85 \tag{12}$$

b_e is effective face width. The effective face width b_e is the real length of contact pattern (see annex C). In the case of full load, the contact pattern typically has a minimum length of 85 % of face width. If it is not possible to obtain information of pattern length under load conditions, b_e = 0,85 b should be used.

If an exact determination of the mass moments of inertia m_1^* and m_2^* of the bevel gears is either not feasible due to cost or otherwise impossible (for example, at the design stage), bevel gears of common gear-blank design can be replaced by the approximately dynamically equivalent cylindrical gears (suffix x) (see Figure 1).

$$m_{1,2}^* \approx m_{1x,2x}^* = \frac{1}{8}\,\rho\,\pi\,\frac{1}{\cos^2\alpha_n}\,d_{m1,2}^2 \tag{13}$$

$$m_{red\,x} = \frac{1}{8}\,\rho\,\pi\,\frac{d_{m1}^2}{\cos^2\alpha_n}\,\frac{u^2}{1+u^2} \tag{14}$$

For example, the following applies to steel gears ($\rho = 7{,}86 \times 10^{-6}$ kg/mm³) with α_n = 20°:

$$m_{red\,x} = 3{,}50 \times 10^{-6}\, d_{m1}^2 \frac{u^2}{1+u^2} \tag{15}$$

Inserted in equations (5) and (6) with c_γ = 20 N/(mm·µm):

$$N = 4{,}38 \times 10^{-8}\, n_1\, z_1\, d_{m1} \sqrt{u^2/\left(1+u^2\right)} = 0{,}084\, \frac{z_1\, v_{mt}}{100} \sqrt{u^2/\left(1+u^2\right)} \tag{16}$$

See Figure 2 for the graphical determination of resonance speed for the mating solid steel pinion/ solid wheel.

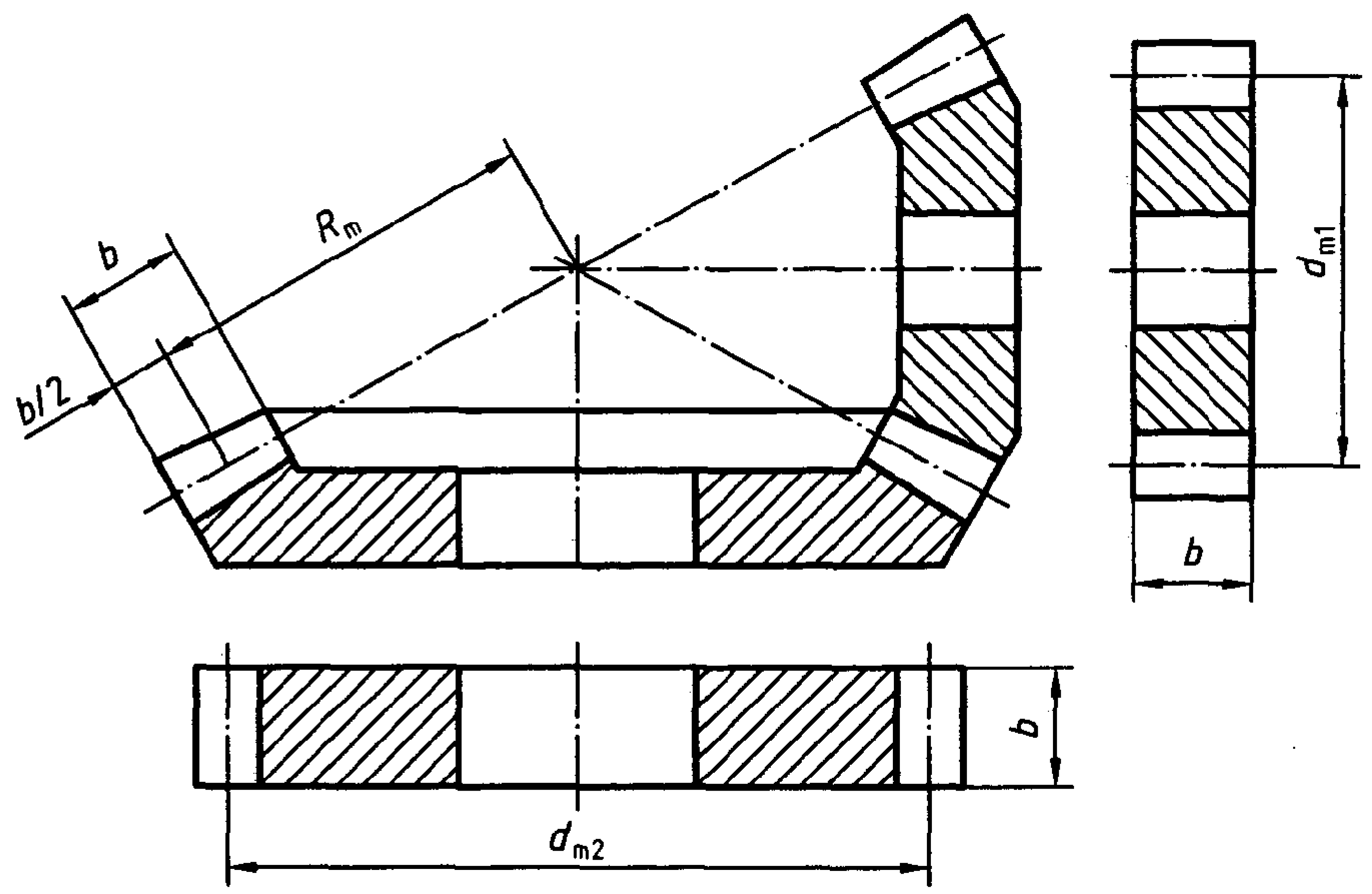


Figure 1 — Approximately dynamically equivalent cylindrical gears for the determination of the dynamic factor

7.7.3.3 Subcritical sector ($N \leqslant 0{,}75$)

Common operating range for industrial and vehicle gears:

$$K_{v\text{-}B} = N\,K + 1 \tag{17}$$

With the simplifying assumptions given above for method B, the following applies:

$$K = \frac{b f_{p\,eff}\, c'}{F_{mt}\, K_A} c_{v1,2} + c_{v3} \tag{18}$$

where

$f_{p\,eff} = f_{pt} - y_p$ with $y_p \approx y_\alpha$

See 9.5 for y_α, 9.3.1 for f_{pt} and Table 2 for $c_{v1,2}$ and c_{v3}.

NOTE The influence of tip relief is not considered. The calculation is therefore on the safe side for bevel gears which normally have profile crowning.

A value of $c_0' = 14$ N/(mm · µm) applies to spur gears. Investigations of helical gears have shown that the stiffness is decreased when the helix angle is increased. On the other hand, the spiral arrangement of a bevel gear tooth on a conical blank leads to stiffening of helical and spiral bevel gears. Therefore, due to the lack of any better knowledge, the stiffness for a spur gear can be said to be suitable for use in average conditions ($F_{mt}K_A/b_e \geqslant 100$ N/mm and $b_e/b \geqslant 0{,}85$). Therefore c' can be determined as follows:

$$c' = c_0'\, C_F\, C_b \tag{19}$$

where

c_0' is single stiffness for average conditions.

NOTE A value of 14 N/(mm · µm) can be used. If experience indicates another, more appropriate value for bevel gears, it should be substituted.

C_F and C_b are correction factors for non-average conditions [see Equations (9) to (12)].

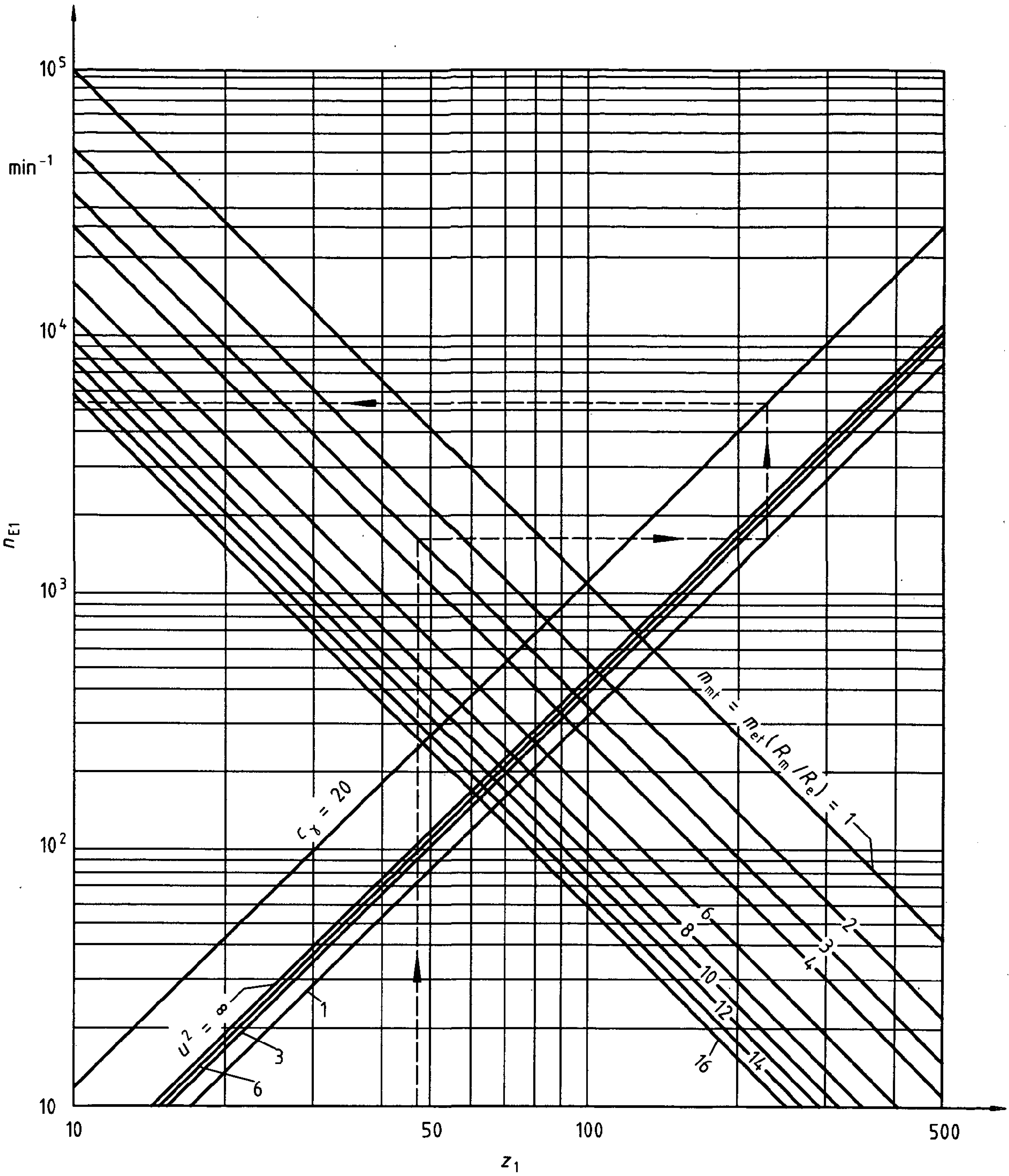


Figure 2 — Diagrams for the determination of the resonance speed, n_{E1}, for the mating solid-steel pinion/solid wheel, with c_γ = 20 N/(mm · µm)

Table 2 — Influence factors c_{v1} to c_{v7} in Equations (18) to (21)

Influence factor	$1 < \varepsilon_{v\gamma} \leqslant 2$[a]	$\varepsilon_{v\gamma} > 2$[a]	
c_{v1} [b]	0,32	0,32	$c_{v1,2} = c_{v1} + c_{v2}$
c_{v2} [c]	0,34	$\dfrac{0{,}57}{\varepsilon_{v\gamma} - 0{,}3}$	
c_{v3} [d]	0,23	$\dfrac{0{,}096}{\varepsilon_{v\gamma} - 1{,}56}$	
c_{v4} [e]	0,90	$\dfrac{0{,}57 - 0{,}05\varepsilon_{v\gamma}}{\varepsilon_{v\gamma} - 1{,}44}$	
c_{v5} [f]	0,47	0,47	$c_{v5,6} = c_{v5} + c_{v6}$
c_{v6} [f]	0,47	$\dfrac{0{,}12}{\varepsilon_{v\gamma} - 1{,}74}$	
	$1 < \varepsilon_{v\gamma} \leqslant 1{,}5$	$1{,}5 < \varepsilon_{v\gamma} \leqslant 2{,}5$	$\varepsilon_{v\gamma} > 2{,}5$
c_{v7} [g]	0,75	$0{,}125 \sin[\pi(\varepsilon_{v\gamma} - 2)] + 0{,}875$	1,0

a For $\varepsilon_{v\gamma}$ see Equation (A.39).

b This influence factor allows for pitch deviation effects and is assumed to be constant.

c This influence factor allows for tooth profile deviation effects.

d This influence factor allows for the cyclic variation effect in mesh stiffness.

e This influence factor takes into account resonant torsional oscillations of the gear pair, excited by cyclic variation of the mesh stiffness.

f In the supercritical sector the influences on $K_{v\text{-}B}$ of the influence factors c_{v5} and c_{v6} correspond to those of c_{v1} and c_{v2} in the subcritical sector;

g This influence factor takes into account the component of force which, due to mesh stiffness variation, is derived from tooth bending deflections during substantially constant speed.

7.7.3.4 Main resonance sector ($0{,}75 < N \leqslant 1{,}25$)

With the simplifying assumptions of method B, the following applies:

$$K_{v\text{-}B} = \frac{b f_{p\,eff}\, c'}{F_{mt}\, K_A} c_{v1,2} + c_{v4} + 1 \tag{20}$$

For c' and $f_{p\,eff}$ see 7.7.3.3; for $c_{v1,2}$ and c_{v4} see Table 2.

7.7.3.5 Supercritical sector ($N \geqslant 1{,}5$)

High-speed gears and those with similar requirements operate in this sector:

$$K_{v\text{-}B} = \frac{b f_{p\,eff}\, c'}{F_{mt}\, K_A} c_{v5,6} + c_{v7} \tag{21}$$

For c' and $f_{p\ eff}$, see 7.7.3.3; for $c_{v5,6}$ and c_{v7}, see Table 2.

7.7.3.6 Intermediate sector (1,25 < N < 1,5)

In this sector, the dynamic factor is determined by linear interpolation between $K_{v\text{-}B}$ at N = 1,25 and $K_{v\text{-}B}$ at N = 1,5. $K_{v\text{-}B}$ is calculated according to 7.7.3.4 and 7.7.3.5 respectively.

$$K_{v\text{-}B} = K_{v\text{-}B(N=1,5)} + \frac{K_{v\text{-}B(N=1,25)} - K_{v\text{-}B(N=1,5)}}{0,25}(1,5 - N) \qquad (22)$$

7.7.4 Method C, $K_{v\text{-}C}$

7.7.4.1 General comments

Figure 3 shows dynamic factors which can be used in the absence of specific knowledge of the dynamic loads. The curves of Figure 3 and the equations given below are based on empirical data, and do not account for resonance (see 7.6).

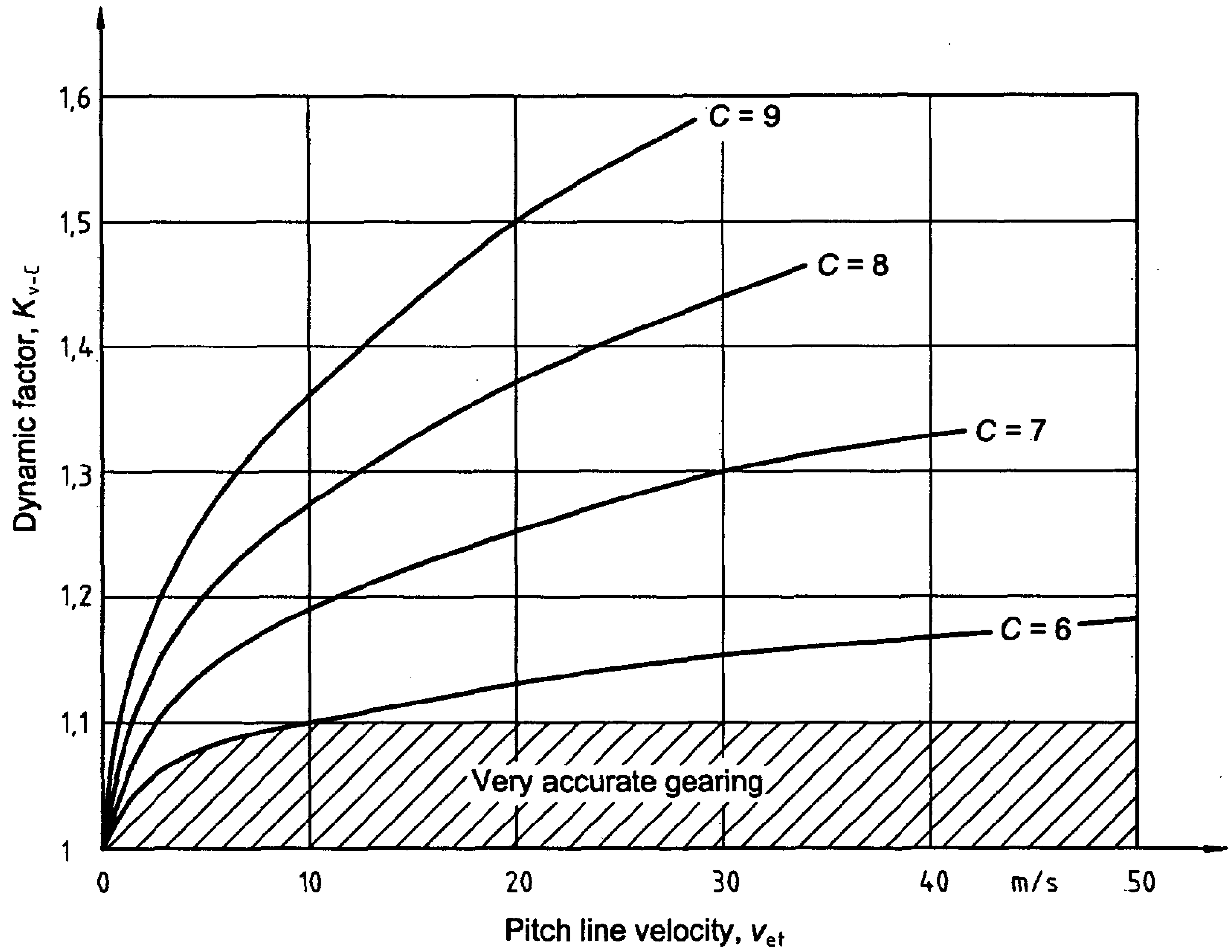


Figure 3 — Dynamic factors, $K_{v\text{-}C}$

Because of the approximate nature of the empirical curves, and the lack of measured tolerance values at the design stage, the dynamic factor curve should be selected based on experience of manufacturing methods and taking into account the operating conditions affecting the design (see 7.7.1). In most cases the contact pattern on the tooth flank is helpful for comparison with previous experience.

The choice of curves 6 to 9 and "very accurate gearing" (7.7.4.2), should be based on the transmission error (see 7.4). If transmission error is not available, it is reasonable to refer to the contact pattern on the tooth flank. If the contact pattern on each tooth flank is not uniform, pitch accuracy (single pitch deviation) can be incorporated as a representative value to determine the dynamic factor. C is the accuracy grade calculated according to the formulae given in ISO 1328-1.

7.7.4.2 Very accurate gearing

Where gearing is manufactured using process control to very accurate gearing grades (generally speaking, when $C \leqslant 5$ in accordance with ISO 1328-1, or where design, manufacturing and application experience ensure a low transmission error), values of K_v between 1,0 and 1,1 may be used, depending on the specifier's experience with similar applications and the degree of accuracy actually achieved. In order to be able to use these values correctly, the gearing shall be maintained with accurate alignment and adequate lubrication so that its overall accuracy is maintained under the operating conditions.

7.7.4.3 Empirical curves

The empirical curves $C = 6$ to $C = 9$ shown in Figure 3 are generated by the following equations for values of C, such that:

$6 \leqslant C \leqslant 9$

$6 \leqslant z \leqslant 1200$ or $10\,000/m_{mn}$, whichever is less

$1{,}25 \leqslant m_{mn} \leqslant 50$

Curves may be extrapolated beyond the end points shown in Figure 3 based on experience and careful consideration of the factors influencing dynamic load. For the purpose of computer calculations, Equation (28) defines the end points of the curves in Figure 3.

$$K_{v\text{-}C} = \left(\frac{A}{A + \sqrt{200 v_{et}}} \right)^{-B} \tag{23}$$

$$v_{et} = v_{mt} \frac{d_{e1,2}}{d_{m1,2}} \tag{24}$$

where

$$A = 50 + 56\,(1{,}0 - B) \tag{25}$$

$$B = 0{,}25\,(C - 5{,}0)^{0{,}667} \tag{26}$$

C is the ISO accuracy grade according to ISO 1328-1 (using m_{mn} and d_m). It can also be calculated with knowledge of the single pitch deviation by rounding the calculated C value:

$$C = -0{,}504\,8 \ln(z) - 1{,}144 \ln(m_{mn}) + 2{,}852 \ln(f_{pt}) + 3{,}32 \tag{27}$$

where

ln is the natural logarithmic function, i.e. $\log_e(\,)$

z is the number of pinion or gear teeth, which results in the highest value of C

m_{mn} is the mean normal module

f_{pt} is the single pitch deviation (at mean point) in micrometres.

The maximum recommended pitch line velocity $v_{et\,max}$ for a given quality grade C is determined as follows:

$$v_{\text{et max}} = \frac{[A + (14 - C)]^2}{200} \tag{28}$$

where $v_{\text{et max}}$ is the maximum pitch line velocity at the outer pitch diameter (end point of K_v curves in Figure 3), in metres per second.

8 Face load factors, $K_{H\beta}$, $K_{F\beta}$

8.1 General comments

8.1.1 The face load factors, $K_{H\beta}$ and $K_{F\beta}$, modify the rating formulae to reflect the non-uniform distribution of the load along the face width.

8.1.2 $K_{H\beta}$ is defined as the ratio between the maximum load per unit face width and the mean load per unit face width.

8.1.3 $K_{F\beta}$ is defined as the ratio between the maximum tooth root stress and the mean tooth root stress over the face width.

8.1.4 The amount of load distribution non-uniformity is influenced by:

— gear-tooth manufacturing accuracy, and tooth-contact pattern and spacing;

— alignment of the gears in their mountings;

— elastic deflections of the gear teeth, shafts, bearings, housings, and foundations which support the gear unit, resulting from either the internal or external gear loads;

— bearing clearances;

— Hertzian contact deformation of the tooth surfaces;

— thermal expansion and distortion of the gear unit due to operating temperatures (especially important on gear units where the gear housing is made from a different material than the gears, shafts and bearings);

— centrifugal deflections due to operating speeds.

8.1.5 The geometric characteristics of a bevel-gear tooth change along its face width. Accordingly, the magnitudes of the axial and radial components of the tangential load vary with the position of the tooth contact. Similarly, the deflections of the mountings and of the tooth itself will vary, to in turn affect the position of the tooth contact and its size and shape.

For applications in which the operating torque varies, the desired contact shall be considered "ideal" at full load only. For intermediate loads, a satisfactory compromise will have to be accepted.

ISO 10300 is not applicable to bevel gears which have a poor contact pattern (see 5.4.8 and annex C).

8.2 Method A

A comprehensive analysis of all influence factors, such as measurement of tooth root stress in service, is needed for an exact determination of the load distribution across the face width according to method A. However, due to its high cost, this type of analysis will be restricted in practice.

8.3 Method B

An approach for bevel gears corresponding to method B has yet to be evaluated.

8.4 Method C

8.4.1 Face load factor, $K_{H\beta\text{-}C}$

In the case of bevel gears, the face load distribution is influenced essentially by the crowning of the gear teeth and by the deflections occurring in service. To take account of this crowning effect (point contact), the rectangular contact area is replaced by an inscribed ellipse, whose major axis is equal to the common face width b and minor axis is equal to the length of the transverse path of contact of the corresponding virtual cylindrical gear. This is considered in the calculation of the load distribution by a factor of 1,5, which applies, however, only to gear sets with satisfactory contact patterns as defined in annex C.

The influence of the deflections, and thus of the bearing arrangement, is accounted for by the mounting factor $K_{H\beta\text{-}be}$, according to Table 3.

In order to compensate for an effective face width under full load b_e less than 85 % of the face width b, the face load factors are to be corrected. Thus, the decisive load distribution factor $K_{H\beta\text{-}C}$ is:

$$K_{H\beta\text{-}C} = 1{,}5\, K_{H\beta\text{-}be} \qquad \text{for } b_e \geqslant 0{,}85\, b \tag{29}$$

$$K_{H\beta\text{-}C} = 1{,}5\, K_{H\beta\text{-}be} \cdot \frac{0{,}85}{b_e/b} \qquad \text{for } b_e < 0{,}85\, b \tag{30}$$

This equation shall not be valid for uncrowned gears.

Table 3 — Mounting factor, $K_{H\beta\text{-}be}$

Verification of contact pattern	Mounting conditions of pinion and gear		
Contact pattern is checked:	Neither member cantilever mounted	One member cantilever mounted	Both members cantilever mounted
for each gear set in its housing under full load	1,00	1,00	1,00
for each gear set under light test load	1,05	1,10	1,25
for a sample gear set and estimated for full load	1,20	1,32	1,50
NOTE Based on optimum tooth contact pattern under maximum operating load as evidenced by results of a deflection test on the gears in their mountings.			

WARNING — The observed contact pattern is normally an accumulated picture of each possible tooth-pair combination. The formulae above are valid only if the shifting of the tooth contact pattern, during one revolution of the gear, either towards the heel or toe, is small. Otherwise, the smallest contact pattern should be taken for the determination of b_e. This shifting of single contact patterns is particularly pronounced for gears finished only by lapping.

8.4.2 Face load factor, $K_{F\beta\text{-}C}$

$K_{F\beta}$ accounts for the effect of the load distribution across the face width on the tooth root stress.

$$K_{F\beta\text{-}C} = K_{H\beta} / K_{F0} \tag{31}$$

Where, for

$K_{H\beta}$ see 8.4.1;

K_{F0} see 8.4.3.

8.4.3 Lengthwise curvature factor for bending strength, K_{F0}

The lengthwise curvature factor, K_{F0}, depends on:

a) spiral angle;

b) lengthwise tooth curvature.

8.4.3.1 Formula

The formula for the lengthwise curvature factor is given by:

$$K_{F0} = 0{,}211 \left(\frac{r_{c0}}{R_m} \right)^q + 0{,}789 \quad (32)$$

for spiral bevel gears

$$K_{F0} = 1{,}0 \quad (33)$$

for straight bevel or zerol gears

where

r_{c0} is cutter radius, in millimetres;

R_m is mean cone distance, in millimetres;

$$q = \frac{0{,}279}{\log_{10}(\sin \beta_m)} \quad (34)$$

β_m is the mean spiral angle.

If the calculated value of K_{F0} is greater than 1,15, then make K_{F0} = 1,15; whereas if the calculated value of K_{F0} is less than 1,0, make K_{F0} = 1,0.

9 Transverse load factors, $K_{H\alpha}$, $K_{F\alpha}$

9.1 General comments

The distribution of the total tangential force over several pairs of meshing teeth depends, in the case of given gear dimensions, on the gear accuracy and the value of the total tangential force. The factor $K_{H\alpha}$ accounts for the effect of the load distribution on the contact stress, while $K_{F\alpha}$ accounts for the effect of the load distribution on the tooth root stress (see ISO 6336-1 for further information). The use of method A requires comprehensive analysis (see 9.2), whereas the methods of approximation B and C (see 9.3 and 9.4) are sufficiently accurate.

9.2 Method A

The load distribution taken as the basis for the load capacity calculation can be determined by measurement or by an exact analysis of all influence factors. However, when the latter is used, the method's accuracy and reliability shall be proved and its premises clearly presented.

9.3 Method B

9.3.1 Bevel gears having virtual cylindrical gears with contact ratio $\varepsilon_{v\gamma} \leqslant 2$

$$K_{H\alpha} = K_{F\alpha} = \frac{\varepsilon_{v\gamma}}{2}\left[0{,}9 + 0{,}4\frac{c_\gamma\left(f_{pt} - y_\alpha\right)}{F_{mtH}/b}\right] \tag{35}$$

where

c_γ is mesh stiffness, as an approximation, $c_\gamma = 20$ N/(mm · µm) (see 7.7.3.2);

f_{pt} is single pitch deviation, maximum value of pinion or wheel;

NOTE For design calculations the tolerance of the wheel according to ISO 1328-1 can be used.

y_α is running-in allowance (see 9.5);

F_{mtH} is decisive tangential force at mid-facewidth on the reference cone, $F_{mtH} = F_{mt} K_A K_v K_{H\beta}$.

$K_{H\alpha}$, $K_{F\alpha}$ may also be taken from Figure 4.

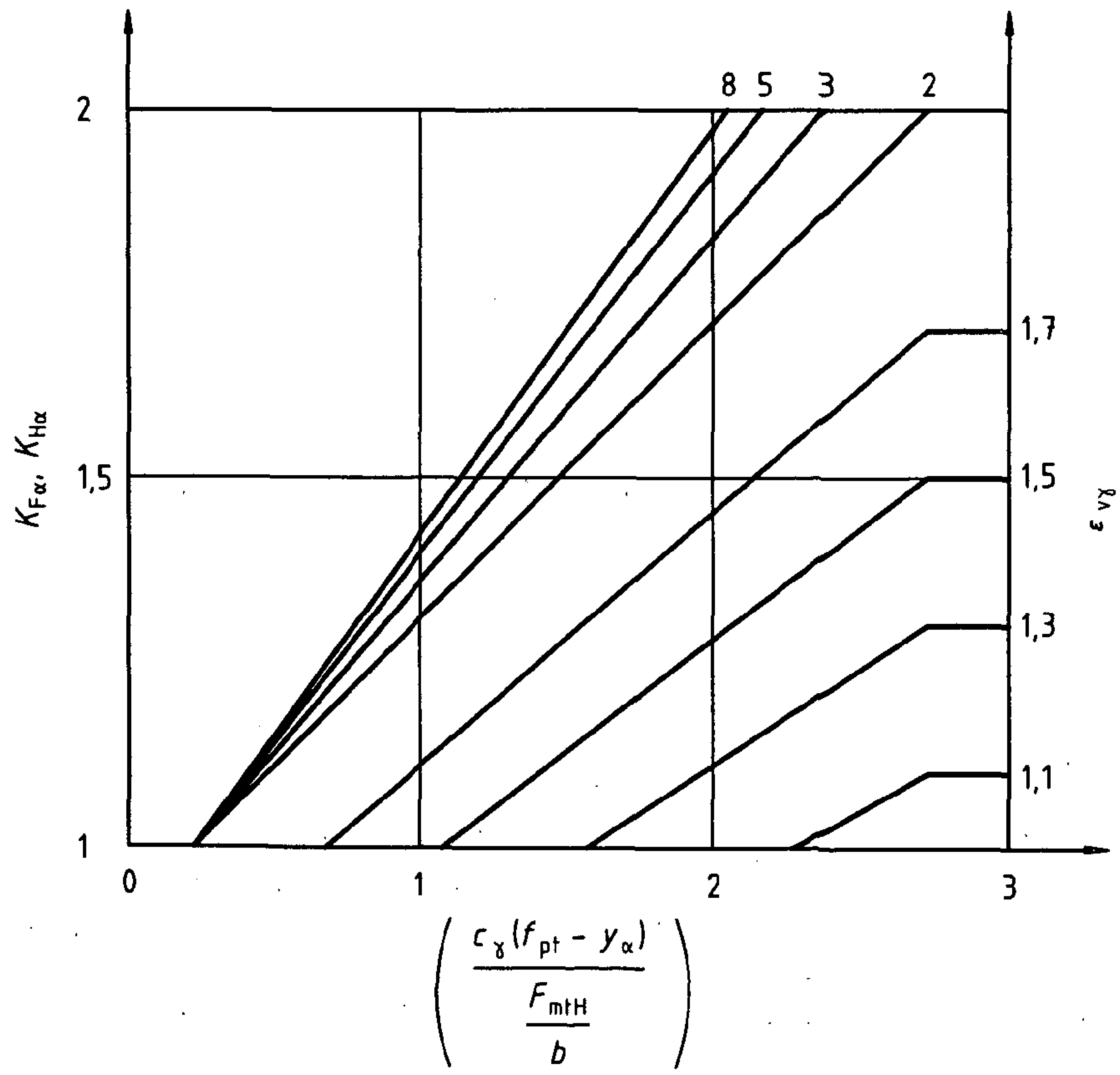


Figure 4 — Transverse load factors, $K_{H\alpha\text{-}B}$ and $K_{F\alpha\text{-}B}$

9.3.2 Bevel gears having virtual cylindrical gears with contact ratio $\varepsilon_{v\gamma} > 2$

$$K_{H\alpha} = K_{F\alpha} = 0{,}9 + 0{,}4 \sqrt{\frac{2\left(\varepsilon_{v\gamma} - 1\right)}{\varepsilon_{v\gamma}}} \cdot \frac{c_\gamma \left(f_{pt} - y_\alpha\right)}{F_{mtH} / b} \tag{36}$$

where, for

c_γ see 9.3.1;

f_{pt} see 9.3.1;

y_α see 9.3.1;

F_{mtH} see 9.3.1.

9.3.3 Boundary conditions

If $K_{H\alpha}$, $K_{F\alpha} < 1$, then $K_{H\alpha}$, $K_{F\alpha}$ is to be taken as unity.

If in accordance with Equations (35) and (36)

$$K_{H\alpha} > \frac{\varepsilon_{v\gamma}}{\varepsilon_{v\alpha} Z_{LS}^2}, \text{ then}$$

$$K_{H\alpha} = \frac{\varepsilon_{v\gamma}}{\varepsilon_{v\alpha} Z_{LS}^2} \tag{37}$$

is to be taken.

For Z_{LS} see ISO 10300-2.

If in accordance with Equations (35) and (36)

$$K_{F\alpha} > \frac{\varepsilon_{v\gamma}}{\varepsilon_{v\alpha} Y_\varepsilon}, \text{ then}$$

$$K_{F\alpha} = \frac{\varepsilon_{v\gamma}}{\varepsilon_{v\alpha} Y_\varepsilon} \tag{38}$$

is to be taken.

For Y_ε see ISO 10300-3.

With these boundary conditions the most unfavourable load distribution is assumed, i.e. only one pair of teeth transmits the total tangential force, and the calculation is therefore on the safe side. It is recommended that the accuracy of helical and spiral bevel gears be chosen so that neither $K_{H\alpha}$ nor $K_{F\alpha}$ exceeds $\varepsilon_{v\alpha n}$.

9.4 Method C

9.4.1 General comments

This method is, in general, sufficiently accurate for industrial gears. For the determination of the factors $K_{H\alpha\text{-}C}$, $K_{F\alpha\text{-}C}$ the gear accuracy grade, specific loading, gear type and running-in behaviour must be known. The running-in behaviour is expressed by material and type of heat treatment.

9.4.2 Premises, assumptions

9.4.1.1 Transverse contact ratio: $1,2 < \varepsilon_{v\alpha} < 1,9$, applies to tooth stiffness (see ISO 6336-1).

9.4.1.2 Stiffness values $c' = 14$ N/(mm · µm) or $c_\gamma = 20$ N/(mm · µm), according to 7.7.3.2 and 7.7.3.3.

9.4.1.3 A single pitch deviation is assigned to each gear accuracy grade. With this assumption, transverse load distribution factors are obtained which are on the safe side for most applications, i.e. in case of mean and high specific loadings, as well as in case of specific loadings, $F_{mt} K_A / b_e < 100$ N/mm.

9.4.3 Determination of the factors

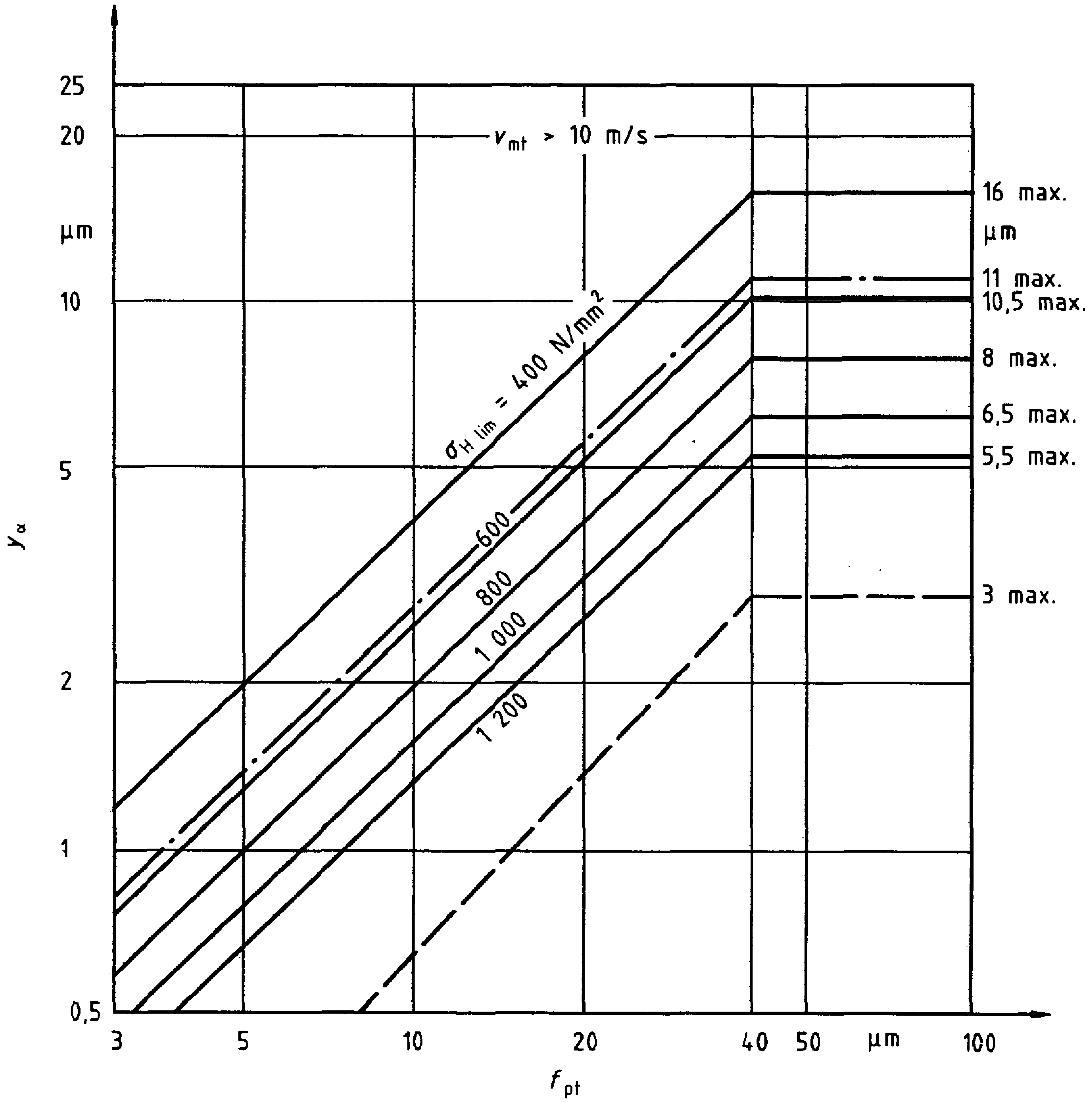
$K_{H\alpha\text{-}C}$ and $K_{F\alpha\text{-}C}$ shall be taken from Table 4.

Table 4 — Transverse load distribution factors, $K_{H\alpha\text{-}c}$ and $K_{F\alpha\text{-}c}$

<table>
<tr><th colspan="3">Specific loading $K_A F_{mt}/b_e$</th><th colspan="7">⩾ 100 N/mm</th><th>< 100 N/mm</th></tr>
<tr><th colspan="3">Gear accuracy grade according to ISO 1328-1 (using d_m and m_{mn}) (see 5.3.2)</th><th>6 and better</th><th>7</th><th>8</th><th>9</th><th>10</th><th>11</th><th>12</th><th>all accuracy grades</th></tr>
<tr><td rowspan="4">Surface hardened</td><td rowspan="2">Straight bevel gears</td><td>$K_{H\alpha}$</td><td colspan="2" rowspan="2">1,0</td><td rowspan="2">1,1</td><td rowspan="2">1,2</td><td colspan="4">$1/Z_{LS}^2$ or 1,2, whichever is greater</td></tr>
<tr><td>$K_{F\alpha}$</td><td colspan="4">$1/Y_\varepsilon$ or 1,2, whichever is greater</td></tr>
<tr><td rowspan="2">Helical and spiral bevel gears</td><td>$K_{H\alpha}$</td><td rowspan="2">1,0</td><td rowspan="2">1,1</td><td rowspan="2">1,2</td><td rowspan="2">1,4</td><td colspan="4" rowspan="2">$\varepsilon_{v\alpha n}$ or 1,4, whichever is greater</td></tr>
<tr><td>$K_{F\alpha}$</td></tr>
<tr><td rowspan="4">Not surface hardened</td><td rowspan="2">Straight bevel gears</td><td>$K_{H\alpha}$</td><td colspan="3" rowspan="2">1,0</td><td rowspan="2">1,1</td><td rowspan="2">1,2</td><td colspan="3">$1/Z_{LS}^2$ or 1,2 whichever is greater</td></tr>
<tr><td>$K_{F\alpha}$</td><td colspan="3">$1/Y_\varepsilon$ or 1,2 whichever is greater</td></tr>
<tr><td rowspan="2">Helical and spiral bevel gears</td><td>$K_{H\alpha}$</td><td colspan="2" rowspan="2">1,0</td><td rowspan="2">1,1</td><td rowspan="2">1,2</td><td rowspan="2">1,4</td><td colspan="3" rowspan="2">$\varepsilon_{v\alpha n}$ or 1,4 whichever is greater</td></tr>
<tr><td>$K_{F\alpha}$</td></tr>
<tr><td colspan="11">NOTE For Z_{LS} see ISO 10300-2, for Y_ε see ISO 10300-3.</td></tr>
</table>

9.5 Running-in allowance, y_α

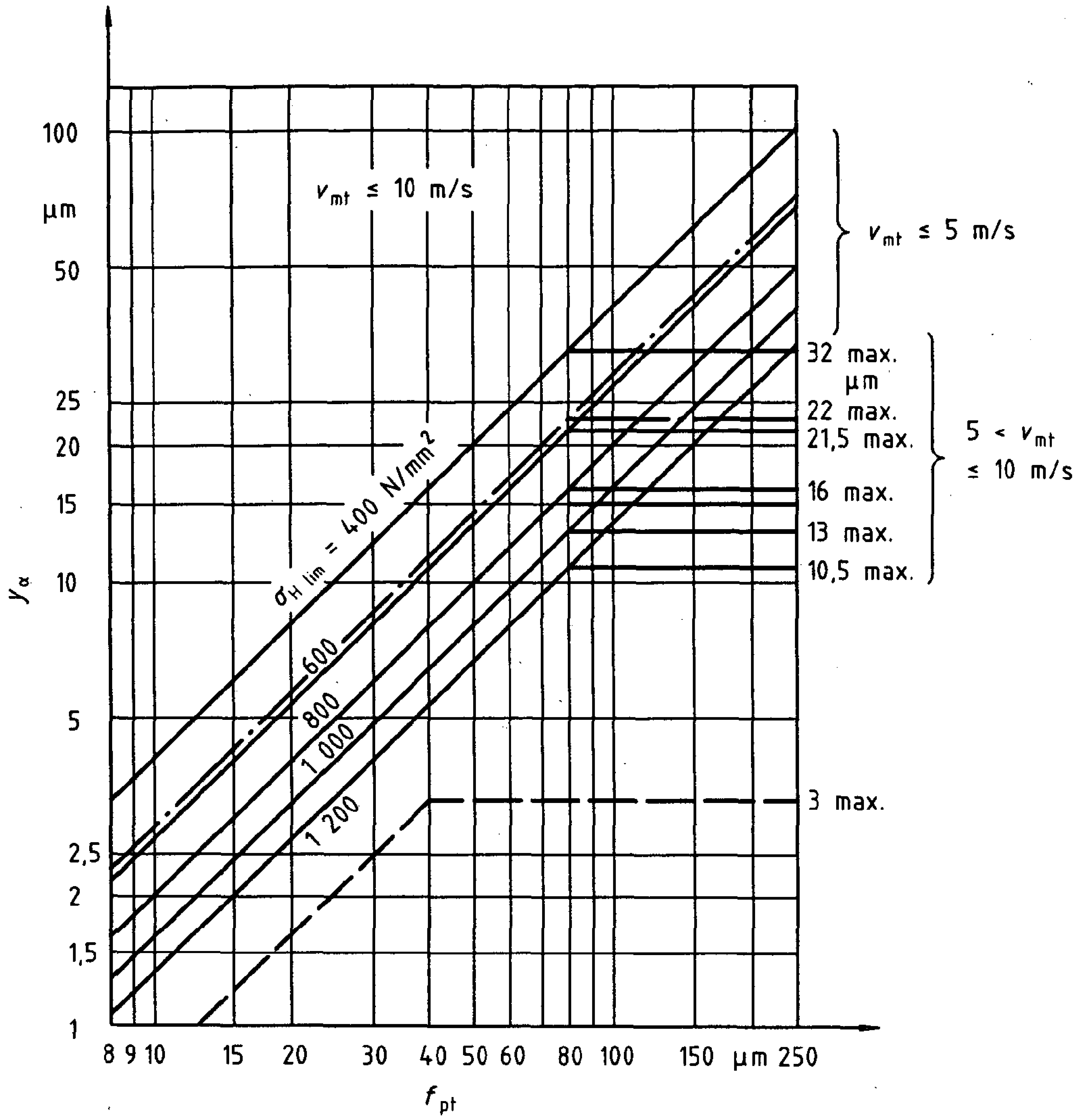
The running-in allowance, y_α, is the amount due to running-in by which the mesh alignment error is reduced from the start of the operation. In the absence of direct experience, y_α may be taken from Figure 5 or Figure 6. The following equations, representing the curves in these figures, may be used for the calculation (where f_{pt}: see 9.3.1).



Key

————————	structural and through hardened steel
————·————	grey cast iron
— — — — —	case hardened and nitrided steel

Figure 5 — Running-in allowance, y_α, of gear pairs with a tangential speed of v_{mt} > 10 m/s



Key

———————	structural and through hardened steel
———— · ————	grey cast iron
— — — — —	case hardened and nitrided steel

Figure 6 — Running-in allowance, y_α, of gear pairs with a tangential speed of $v_{mt} \leqslant 10$ m/s

For through-hardened steels:

$$y_\alpha = \frac{160}{\sigma_{H\,lim}} f_{pt} \tag{39}$$

for $v_{mt} \leqslant 5$ m/s: without restriction

for 5 m/s < $v_{mt} \leqslant 10$ m/s: $y_\alpha \leqslant 12\,800/\sigma_{H\,lim}$

for v_{mt} > 10 m/s: $y_\alpha \leqslant 6\,400\,/\,\sigma_{H\,lim}$

For grey cast iron:

$$y_\alpha = 0{,}275\, f_{pt} \tag{40}$$

for $v_{mt} \leqslant 5$ m/s: without restriction

for 5 m/s $< v_{mt} \leqslant 10$ m/s: $y_\alpha \leqslant 22$ µm

for $v_{mt} > 10$ m/s: $y_\alpha \leqslant 11$ µm

For case-hardened and nitrided gears:

$$y_\alpha = 0{,}075\, f_{pt} \tag{41}$$

for all speeds with the restriction: $y_\alpha \leqslant 3$ µm

When materials of pinion and wheel are different:

$$y_\alpha = \frac{y_{\alpha 1} + y_{\alpha 2}}{2} \tag{42}$$

wherein $y_{\alpha 1}$ is to be determined for the pinion material and $y_{\alpha 2}$ for the wheel material.

Annex A
(normative)

Calculation of bevel gear geometry

A.1 General

Annex A contains geometric relations required for generating the data for the virtual cylindrical gear, required for bevel-gear load-capacity calculations.

If a transverse section of a bevel-gear tooth at midface is developed into a plane, a virtual cylindrical gear is obtained with nearly involute teeth (Tredgold's approximation). The load capacity calculations of ISO 10300 are based on the virtual gears and the conditions at mid-facewidth of the bevel gears (see A.6).

For helical and spiral bevel gears, the result is a virtual helical gear. For the load capacity calculation, this gear is partly considered in transverse, and partly in normal, section. The corresponding relations for the gear data presented here apply exclusively to gears with $(x_{hm1} + x_{hm2}) = 0$.

A.2 Initial data

The bevel-gear data may be supplied in either of the two commonly used forms: data type I (see Table A.1) or data type II (see Table A.2). Relationships for conversion between the two forms are given in A.3 to A.8.

Table A.1 — Data type I

Basic data		Alternative data		Alternative data	
Sym.	**Description**	**Sym.**	**Description**	**Sym.**	**Description**
α_n	normal pressure angle				
$z_{1,2}$	number of teeth				
Σ	shaft angle				
d_{e2}	outer pitch diameter of gear	m_{mn}	mean normal module		
β_m	mean spiral angle				
b	face width				
$\rho_{a01,2}$	cutter edge radius				
r_{c0}	cutter radius				
$x_{sm1,2}$	thickness modification coefficient				
$x_{hm1,2}$	profile shift coefficient				
$h^*_{f01,2}$	tool dedendum (related to m_{mn})				
$h^*_{a01,2}$	tool addendum (related to m_{mn})				
$s_{pr1,2}$	protuberance				

Table A.2 — Data type II

Basic data		Alternative data		Alternative data	
Sym.	Description	Sym.	Description	Sym.	Description
α_n	normal pressure angle				
$z_{1,2}$	number of teeth				
Σ	shaft angle				
R_e	outer cone distance	m_{et}	outer transverse module	P_d	outer diametral pitch
β_m	mean spiral angle				
b	face width				
$\delta_{a1,2}$	face angle				
$\theta_{f1,2}$	dedendum angle				
$\rho_{a01,2}$	cutter edge radius				
r_{c0}	cutter radius				
$s_{mn1,2}$	mean normal circular thickness	$s_{mt1,2}$	mean transverse circular thickness	$s_{amn1,2}$	mean normal topland
$h_{ae1,2}$	outer addendum				
$h_{fe1,2}$	outer dedendum				
$s_{pr1,2}$	protuberance				

A.3 Basic formulae

Gear ratio u:

$$u = z_2 / z_1 = \sin \delta_2 / \sin \delta_1 \tag{A.1}$$

Pitch angle δ:

$$\tan \delta_1 = \sin \Sigma / (\cos \Sigma + u) \tag{A.2}$$

$$\delta_2 = \Sigma - \delta_1 \tag{A.3}$$

for $\Sigma = 90°$

$$\tan \delta_1 = 1/u; \tan \delta_2 = u \tag{A.4}$$

Outer cone distance R_e:

$$R_e = 0{,}5\, d_{e2}/\sin \delta_2 = 0{,}5\, d_{e1}/\sin \delta_1 \tag{A.5}$$

Mean cone distance R_m:

$$R_m = R_e - (b / 2) \tag{A.6}$$

Outer transverse module m_{et}:

$$m_{et} = d_{e2} / z_2 = d_{e1} / z_1 = 25{,}4 / P_d \tag{A.7}$$

Mean transverse module m_{mt}:

$$m_{mt} = \frac{R_m}{R_e} m_{et} \tag{A.8}$$

Mean normal module m_{mn}:

$$m_{mn} = m_{mt} \cos \beta_m \tag{A.9}$$

Mean pitch diameter d_m:

$$d_{m\,1,2} = d_{e\,1,2} - b \sin \delta_{1,2} = m_{mn}\, z_{1,2} / \cos \beta_m \tag{A.10}$$

Addendum angle θ_a:

$$\theta_{a\,1,2} = \delta_{a\,1,2} - \delta_{1,2} \tag{A.11}$$

For constant addendum:

$$\delta_{a\,1,2} = \delta_{1,2} \rightarrow \theta_{a\,1,2} = 0 \tag{A.12}$$

Dedendum angle θ_f:

$$\theta_{f\,1,2} = \delta_{1,2} - \delta_{f\,1,2} \tag{A.13}$$

For constant dedendum:

$$\delta_{f\,1,2} = \delta_{1,2} \rightarrow \theta_{f\,1,2} = 0 \tag{A.14}$$

A.4 Data of basic rack tooth profile and of the tool respectively

In case of data type I (Table A.1) h_{f0}, h_{a0} and ρ_{a0} can generally be taken from the information given by the manufacturer.

Dedendum of the tool h^*_{f0} (i.e. addendum of the basic rack tooth profile h^*_{aP}, see Figure A.1) related to the mean normal module:

$$h^*_{f01,2} = h_{f01,2}/m_{mn} = h_{aP1,2}/m_{mn} = h^*_{aP1,2} \tag{A.15}$$

Addendum of the tool h^*_{a0} (i.e. dedendum of the basic rack tooth profile h^*_{fP}, see Figure A.1) related to the mean normal module:

$$h^*_{a01,2} = h_{a01,2}/m_{mn} = h_{fP1,2}/m_{mn} = h^*_{fP1,2} \tag{A.16}$$

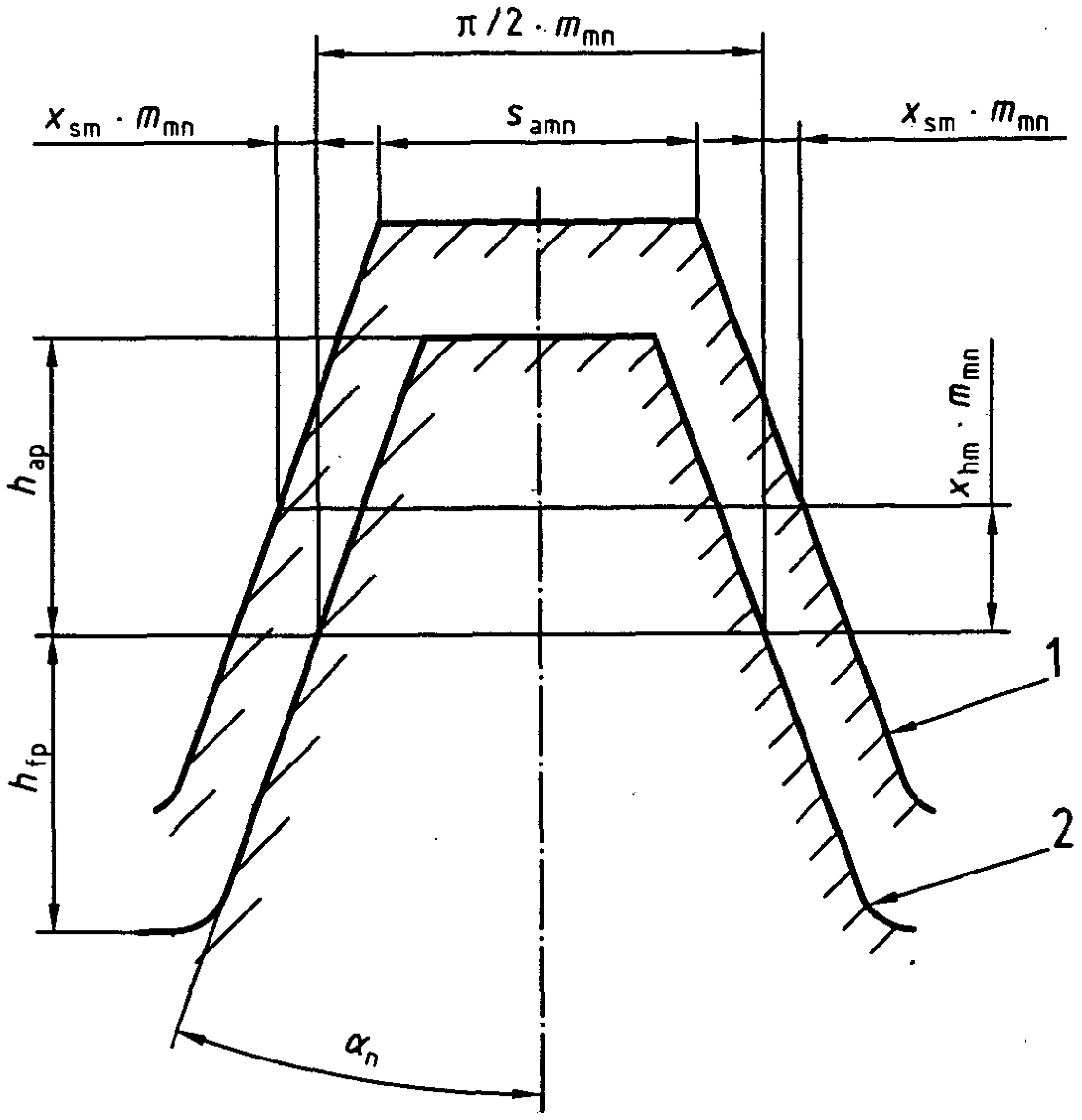
Common values are:

$\rho_{a0}/m_{mn} = 0{,}2$ to $0{,}4$

$h^*_{f0} = 1{,}0$

$h^*_{a0} = 1{,}25$ to $1{,}30$

In case of data type II (Table A.2) only the tool tip radius ρ_{a0} is indicated. In that case h^*_{f0} and h^*_{a0} can be calculated if needed (see Equations A.17 to A.21).



Key

1 Tooth profile with profile shift and thickness modification.

2 Basic rack tooth profile acc. to ISO 53.

Figure A.1 — Basic rack tooth profile

A.5 Tooth depth at midface

A.5.1 In case of data type I (Table A.1):

Mean addendum h_{am}

$$h_{am1,2} = m_{mn} (h^*_{f01,2} + x_{hm1,2}) \tag{A.17}$$

Mean dedendum h_{fm}

$$h_{fm1,2} = m_{mn} (h^*_{a01,2} - x_{hm1,2}) \tag{A.18}$$

A.5.2 In case of data type II (Table A.2):

Mean addendum h_{am}

$$h_{am1,2} = h_{ae1,2} - \frac{b}{2} \tan\theta_{a1,2} \tag{A.19}$$

Mean dedendum h_{fm}

$$h_{fm1,2} = h_{fe1,2} - \frac{b}{2} \tan\theta_{f1,2} \tag{A.20}$$

Profile shift coefficient x_{hm}

$$x_{hm1,2} = (h_{am1,2} - h_{am2,1})/(2m_{mn}) \tag{A.21}$$

A.6 Data of virtual cylindrical gear in transverse section (suffix v)

For the quantities of the virtual cylindrical gear, shown in Figure A.2, the suffix m, which normally indicates the conditions at mid-face width, is not used.

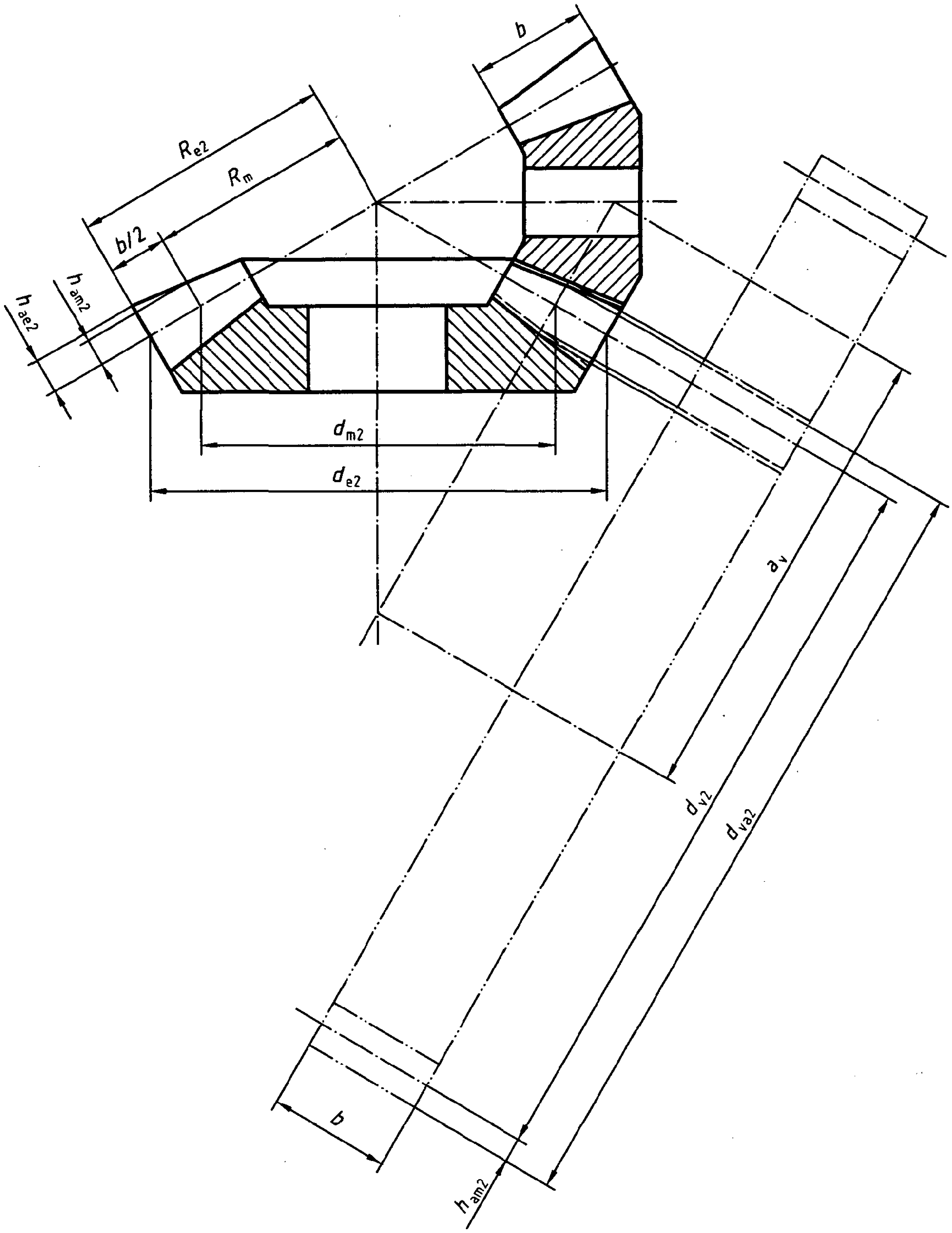


Figure A.2 — Quantities for the calculation of virtual cylindrical gears

Number of teeth z_v:

$$z_{v1,2} = z_{1,2}/\cos \delta_{1,2} \tag{A.22}$$

For $\Sigma = 90°$:

$$z_{v1} = z_1 \frac{\sqrt{u^2 + 1}}{u} \tag{A.23}$$

$$z_{v2} = z_2 \sqrt{u^2 + 1} \tag{A.24}$$

Gear ratio u_v:

$$u_v = u \frac{\cos\delta_1}{\cos\delta_2} = \frac{z_{v2}}{z_{v1}} \tag{A.25}$$

For $\Sigma = 90°$:

$$u_v = \left(\frac{z_2}{z_1}\right)^2 = u^2 \tag{A.26}$$

Reference diameter d_v:

$$d_{v1,2} = \frac{d_{m1,2}}{\cos\delta_{1,2}} = \frac{d_{e1,2}}{\cos\delta_{1,2}} \cdot \frac{R_m}{R_e} \tag{A.27}$$

for $\Sigma = 90°$:

$$d_{v1} = d_{m1} \frac{\sqrt{u^2 + 1}}{u} \tag{A.28}$$

$$d_{v2} = u^2 \, d_{v1} \tag{A.29}$$

Centre distance a_v:

$$a_v = (d_{v1} + d_{v2})/2 \tag{A.30}$$

Tip diameter d_{va}:

$$d_{va1,2} = d_{v1,2} + 2\, h_{am1,2} \tag{A.31}$$

Base diameter d_{vb}:

$$d_{vb1,2} = d_{v1,2} \cos \alpha_{vt} \tag{A.32}$$

with

$$\alpha_{vt} = \arctan\left(\frac{\tan\alpha_n}{\cos\beta_m}\right) \tag{A.33}$$

Helix angle at base circle β_{vb}:

$$\beta_{vb} = \arcsin(\sin\beta_m \cos\alpha_n) \tag{A.34}$$

Transverse base pitch p_{et}:

$$p_{et} = m_{mt}\,\pi\cos\alpha_{vt} \tag{A.35}$$

Length of path of contact $g_{v\alpha}$:

$$g_{v\alpha} = \frac{1}{2}\left[\sqrt{\left(d_{va1}^2 - d_{vb1}^2\right)} + \sqrt{\left(d_{va2}^2 - d_{vb2}^2\right)}\right] - a_v \sin\alpha_{vt} \tag{A.36}$$

Transverse contact ratio $\varepsilon_{v\alpha}$:

$$\varepsilon_{v\alpha} = \frac{g_{v\alpha}}{p_{et}} = \frac{g_{v\alpha}\cos\beta_m}{m_{mn}\,\pi\cos\alpha_{vt}} \tag{A.37}$$

Overlap ratio $\varepsilon_{v\beta}$:

$$\varepsilon_{v\beta} = \frac{b\sin\beta_m}{m_{mn}\,\pi} \tag{A.38}$$

The contact and overlap ratios calculated with Equations A.37 and A.38 for the virtual cylindrical gear are decisive for the load capacity calculation. However, they could deviate from the ratios calculated on the basis of the real dimensions of the bevel gears.

Modified contact ratio $\varepsilon_{v\gamma}$:

$$\varepsilon_{v\gamma} = \sqrt{\varepsilon_{v\alpha}^2 + \varepsilon_{v\beta}^2} \tag{A.39}$$

Because bevel gear teeth are usually crowned and barrelled, it is assumed that the contact zone is bounded by an ellipse of which the major axis is equal to the face width. When the tooth contact has been suitably developed the full load contact should not extend beyond the elliptical boundary.

Length of the line of contact l_b:

$$l_b = b\,g_{v\alpha}\frac{\sqrt{g_{v\alpha}^2\cos^2\beta_{vb} + b^2\sin^2\beta_{vb} - 4f^2}}{g_{v\alpha}^2\cos^2\beta_{vb} + b^2\sin^2\beta_{vb}} \tag{A.40}$$

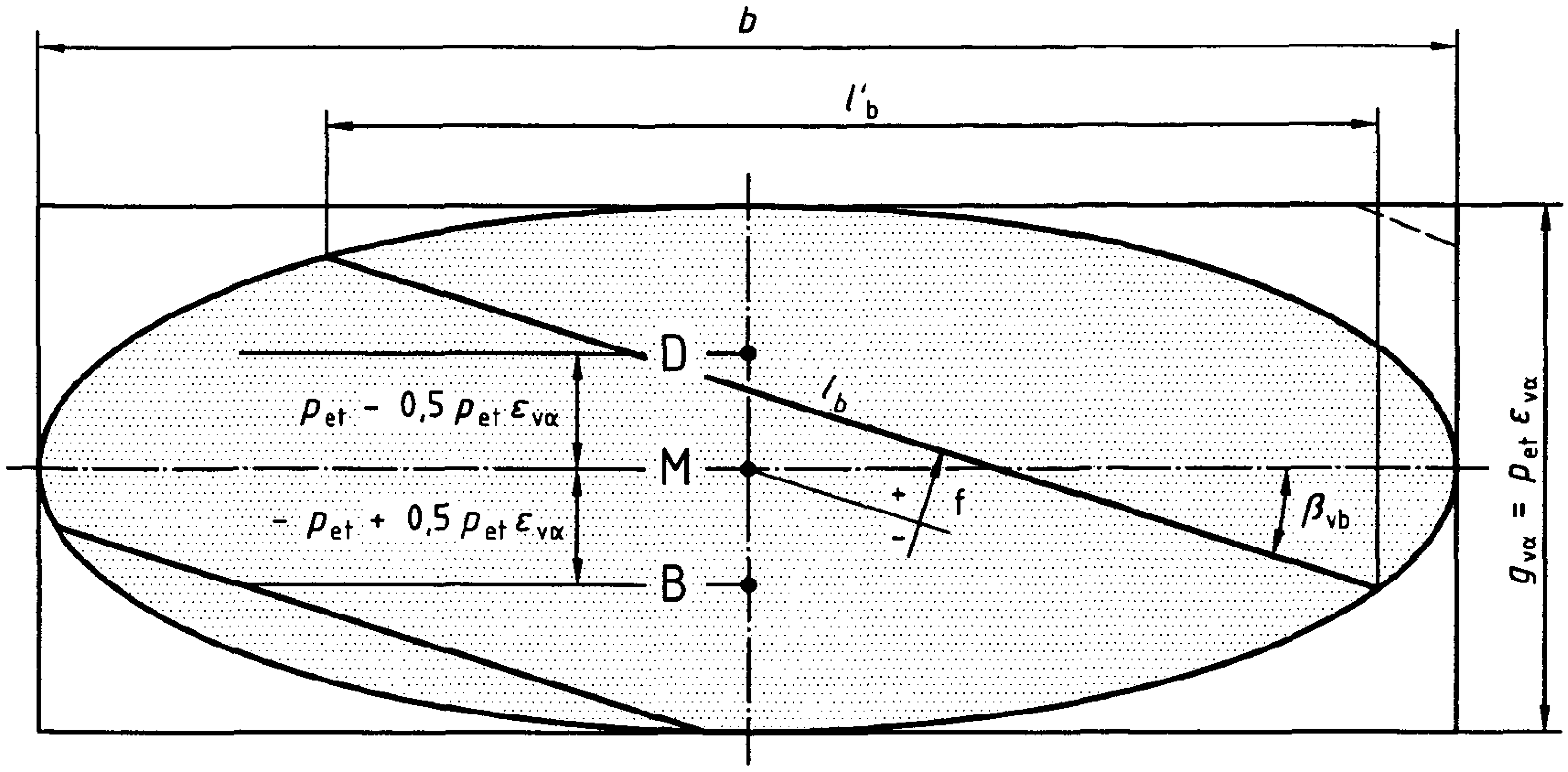
for

$$\left(g_{v\alpha}^2\cos^2\beta_{vb} + b^2\sin^2\beta_{vb} - 4f^2\right) > 0$$

and $l_b = 0$ for

$$\left(g_{v\alpha}^2\cos^2\beta_{vb} + b^2\sin^2\beta_{vb} - 4f^2\right) \leqslant 0 \tag{A.41}$$

Figure A.3 shows the general definitions of values for calculating the length of lines of contact.



B is the inner point of single contact.
D is the outer point of single contact.

Figure A.3 — General definition of length of lines of contact

Equations A.40 and A.41 are to be calculated, according to Table A.3, for:

a) the tip line of contact with $f = f_t$;

b) the middle line of contact with $f = f_m$;

c) the root line of contact with $f = f_r$.

Table A.3 — Distance f of the tip, middle and root line of contact in the zone of action

		pitting	tooth root
$\varepsilon_{v\beta} = 0$	f_t	$-(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} + p_{et} \cos\beta_{vb}$	$(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} + p_{et} \cos\beta_{vb}$
	f_m	$-(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb}$	$(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb}$
	f_r	$-(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} - p_{et} \cos\beta_{vb}$	$(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} - p_{et} \cos\beta_{vb}$
$0 < \varepsilon_{v\beta} < 1$	f_t	$-(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta}) + p_{et} \cos\beta_{vb}$	$(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta}) + p_{et} \cos\beta_{vb}$
	f_m	$-(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta})$	$(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta})$
	f_r	$-(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta}) - p_{et} \cos\beta_{vb}$	$(p_{et} - 0{,}5\, p_{et}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta}) - p_{et} \cos\beta_{vb}$
$\varepsilon_{v\beta} \geqslant 1$	f_t	$+ p_{et} \cos\beta_{vb}$	$+ p_{et} \cos\beta_{vb}$
	f_m	0	0
	f_r	$- p_{et} \cos\beta_{vb}$	$- p_{et} \cos\beta_{vb}$

The length of the middle line of contact also can be expressed in the following way:

$$l_{bm} = \frac{b\,\varepsilon_{v\alpha}}{\cos\beta_{vb}} \frac{\sqrt{\varepsilon_{v\gamma}^2 - \left[(2-\varepsilon_{v\alpha})(1-\varepsilon_{v\beta})\right]^2}}{\varepsilon_{v\gamma}^2} \quad \text{(A.42)}$$

for $\varepsilon_{v\beta} < 1$

$$l_{bm} = \frac{b\,\varepsilon_{v\alpha}}{\cos\beta_{vb}\,\varepsilon_{v\gamma}} \quad \text{(A.43)}$$

for $\varepsilon_{v\beta} \geqslant 1$

Projected length of the middle line of contact l'_{bm}:

$$l'_{bm} = l_{bm} \cos\beta_{vb} \quad \text{(A.44)}$$

A.7 Data of virtual cylindrical gear in normal section (suffix vn)

Number of teeth z_{vn}:

$$z_{vn1} = \frac{z_{v1}}{\cos^2\beta_{vb}\cos\beta_m} \quad \text{(A.45)}$$

$$z_{vn2} = u_v\, z_{vn1} \quad \text{(A.46)}$$

Reference diameter d_{vn}:

$$d_{vn1} = \frac{d_{v1}}{\cos^2\beta_{vb}} = z_{vn1}\, m_{mn} \quad \text{(A.47)}$$

$$d_{vn2} = u_v\, d_{vn1} = z_{vn2}\, m_{mn} \quad \text{(A.48)}$$

Centre distance a_{vn}:

$$a_{vn} = (d_{vn1} + d_{vn2})/2 \quad \text{(A.49)}$$

Tip diameter d_{van}:

$$d_{van1,2} = d_{vn1,2} + d_{va1,2} - d_{v1,2} = d_{vn1,2} + 2\, h_{am1,2} \quad \text{(A.50)}$$

Base diameter d_{vbn}:

$$d_{vbn1,2} = d_{vn1,2} \cos\alpha_n = z_{vn1,2}\, m_{mn} \cos\alpha_n \quad \text{(A.51)}$$

Length of path of contact $g_{v\alpha n}$:

$$g_{v\alpha n} = \frac{1}{2}\left[\sqrt{\left(d_{van1}^2 - d_{vbn1}^2\right)} + \sqrt{\left(d_{van2}^2 - d_{vbn2}^2\right)}\right] - a_{vn} \sin\alpha_n \quad \text{(A.52)}$$

Contact ratio $\varepsilon_{v\alpha n}$:

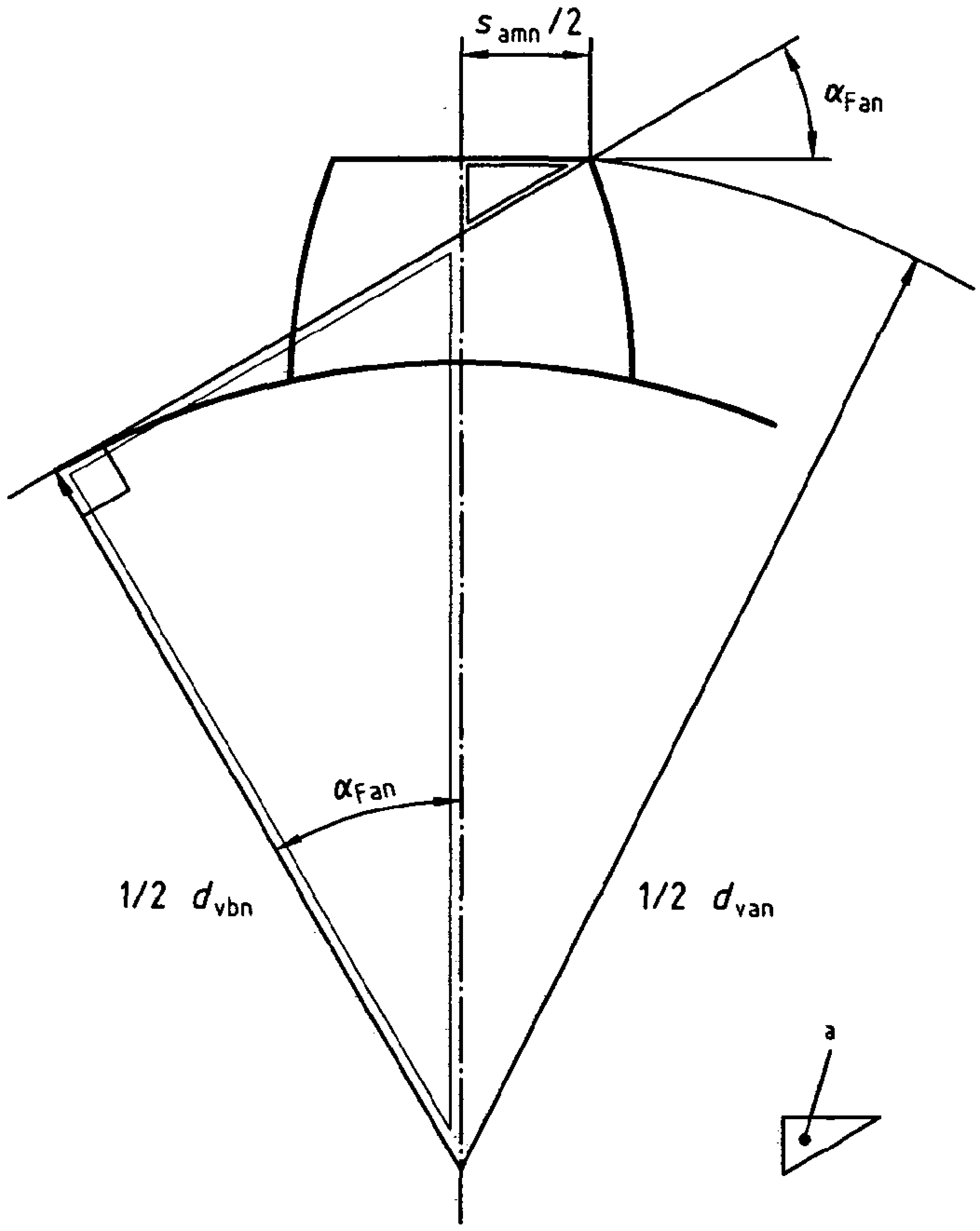
$$\varepsilon_{v\alpha n} = \varepsilon_{v\alpha}/\cos^2\beta_{vb} \quad \text{(A.53)}$$

A.8 Tooth thickness modification

The coefficient for tooth thickness modification x_{sm} is related to the normal module at mid-face width. The tooth thickness modification as compared to the basic rack tooth profile according to ISO 53 amounts to 2 x_{sm} m_{mn} (see Figure A.1).

In the case of data type II (see Table A.2), the coefficient for modification of tooth thickness is not indicated explicitly, but can be calculated as follows.

a) From the given mean normal topland s_{amn} (see Figure A.4).



$s_{amn} \sin\alpha_{Fan} = d_{van} \cos\alpha_{Fan} - d_{vbn}$

a Similar triangles

Figure A.4 — Mean normal topland

Load application angle α_{Fan} (see also Figure 1 in part 3 of ISO 10300):

$$\alpha_{Fan1,2} = \arccos\left(\frac{d_{van1,2}\, d_{vbn1,2} + s_{amn1,2}\sqrt{d^2_{van1,2} + s^2_{amn1,2} - d^2_{vbn1,2}}}{d^2_{van1,2} + s^2_{amn1,2}}\right) \quad \text{(A.54)}$$

Mean normal circular thickness s_{mn}:

$$s_{mn1,2} = \left[\sqrt{\left(\frac{d_{van1,2}}{d_{vbn1,2}}\right)^2 - 1} - \operatorname{inv}\alpha_n - \alpha_{Fan1,2}\frac{\pi}{180°}\right] d_{vn1,2} \quad \text{(A.55)}$$

Thickness modification coefficient x_{sm} (see Figure A.1)

$$x_{sm1,2} = \frac{s_{mn1,2}}{2\,m_{mn}} - \frac{\pi}{4} - x_{hm1,2} \tan\alpha_n \tag{A.56}$$

b) From the given tooth thickness, considering that the tooth thicknesses are partly indicated for the outer tooth end and partly for mid-face width.

Mean transverse circular thickness s_{mt}:

$$s_{mt1,2} = s_{et1,2} \frac{m_{mt}}{m_{et}} = s_{et1,2} \frac{R_m}{R_e} \tag{A.57}$$

Mean normal circular thickness s_{mn}:

$$s_{mn1,2} = s_{mt1,2} \cos\beta_m \tag{A.58}$$

Thickness modification coefficient x_{sm} according to Equation (A.56).

Annex B
(informative)

Values for application factor, K_A

B.1 Establishment of application factors

The application factor, K_A, can best be established by means of a thorough analysis of service experience with a particular application. If service experience is not available, a thorough analytical investigation should be made.

B.2 Approximate values for application factors

Table B.1 provides typical values for application factors if service experience is lacking or if a detailed analysis is not available.

CAUTION — Table B.1 should be used with caution since much higher values have occurred (those as high as 10 have been used) in some applications.

Because bevel gears are nearly always designed with long- (on the pinion member) and short-addendum teeth, regardless of whether the pinion or wheel is the driving member, this results in an approach action when the wheel is driving. As a result, the application factor for speed-increasing drives will be larger than for speed-decreasing drives (see footnote to Table B.1).

Table B.1 — Application factor, K_A, values 1)

Working characteristics of the driving machine	Working characteristics of the driven machine			
	Uniform	Light shocks	Medium shocks	Heavy shocks
Uniform	1,00	1,25	1,50	1,75 or higher
Light shocks	1,10	1,35	1,60	1,85 or higher
Medium shocks	1,25	1,50	1,75	2,00 or higher
Heavy shocks	1,50	1,75	2,00	2,25 or higher

1) This table is for speed-decreasing drives only. For speed-increasing drives, add 0,01 u^2 to K_A, where $u = z_2 / z_1$ = gear ratio.

Annex C
(informative)

Contact patterns

The process by which the tooth contact pattern is modified and refined to its desired shape and position is known as tooth contact development. This is controlled by observing the response of the pattern to movements of the pinion and gear, rotated at a reasonable speed under light load, using a bevel-gear testing machine.

Displacements are made in the testing machine in three directions:

— along the pinion axis;

— along the wheel axis;

— perpendicular to both axes.

The amount of the testing machine displacements, which place the contact in the desired position, is then equated to adjustments in the settings of a cutting or grinding machine to produce the desired contact with the gears assembled at their desired position in the gear box. Repeated trials may be necessary before the development is complete.

When the design is entirely new (see 5.1.3), deflection and tooth-contact checks of the gear assemblage are frequently performed to expedite the tooth contact development and evaluate the rigidity of the gear mountings. In such a test the unit is operated at 25 % increments of full load, up to full load. Rotational speed is low, to permit application of the tooth-marking compound and reading of the displacement at each increment of load.

Displacements are measured in the same directions as on the bevel-gear testing machine, and the results are then duplicated on the testing machine to determine the necessary adjustments in the cutting machine, grinding machine, or both.

For gear applications subjected to thermal distortions, the unit is heated to operating temperatures with heat lamps, and the test is repeated at the same increments of load. Comparison of the data between the tests indicates the effect of the difference in thermal expansion rates.

NOTE Recent developments in relating computer analysis of tooth contacts to three-dimensional coordinate measurements of bevel-gear tooth surfaces have simplified the traditional tooth-contact-development procedure described here.

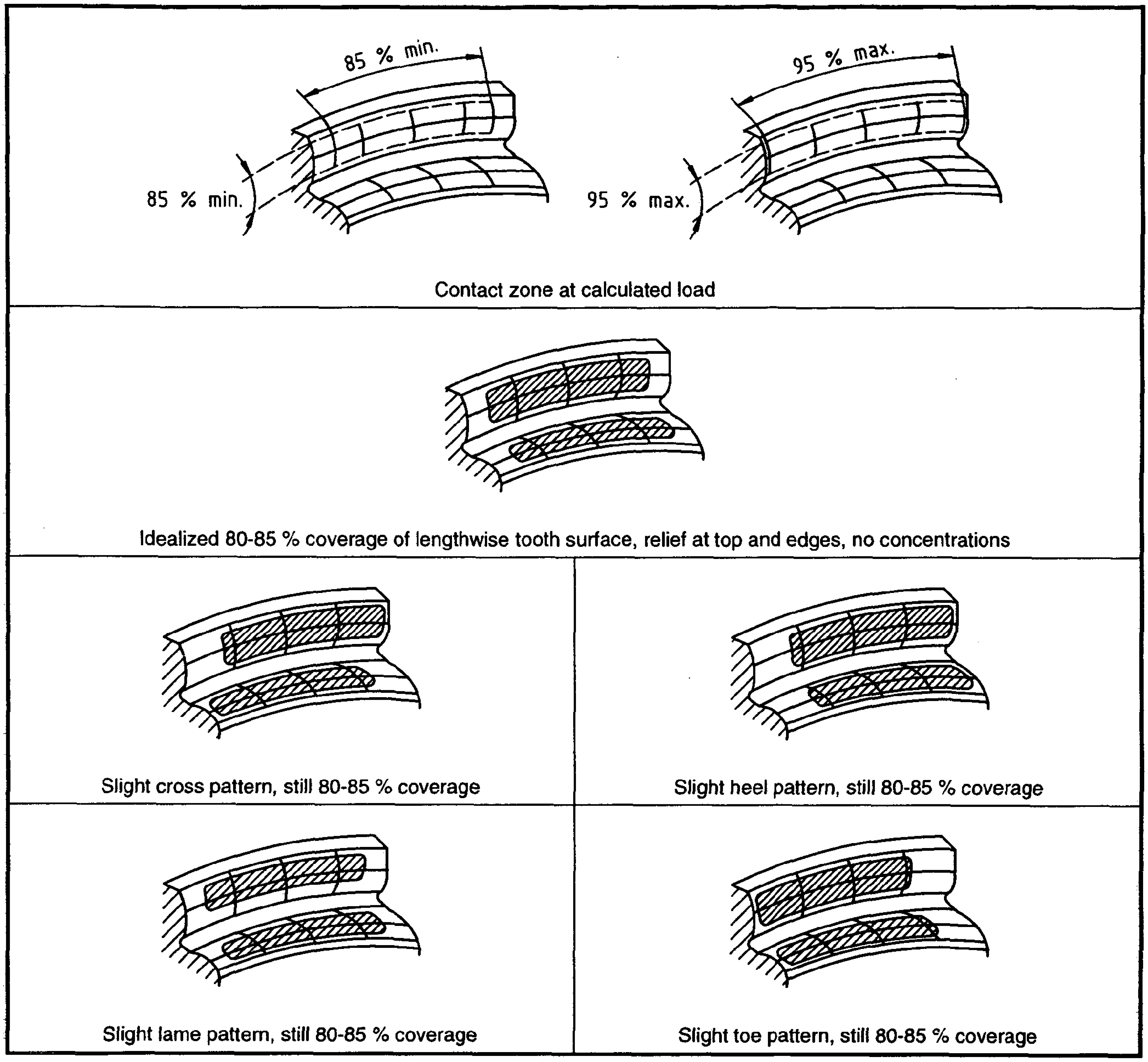


Figure C.1 —Typical satisfactory loaded contact patterns

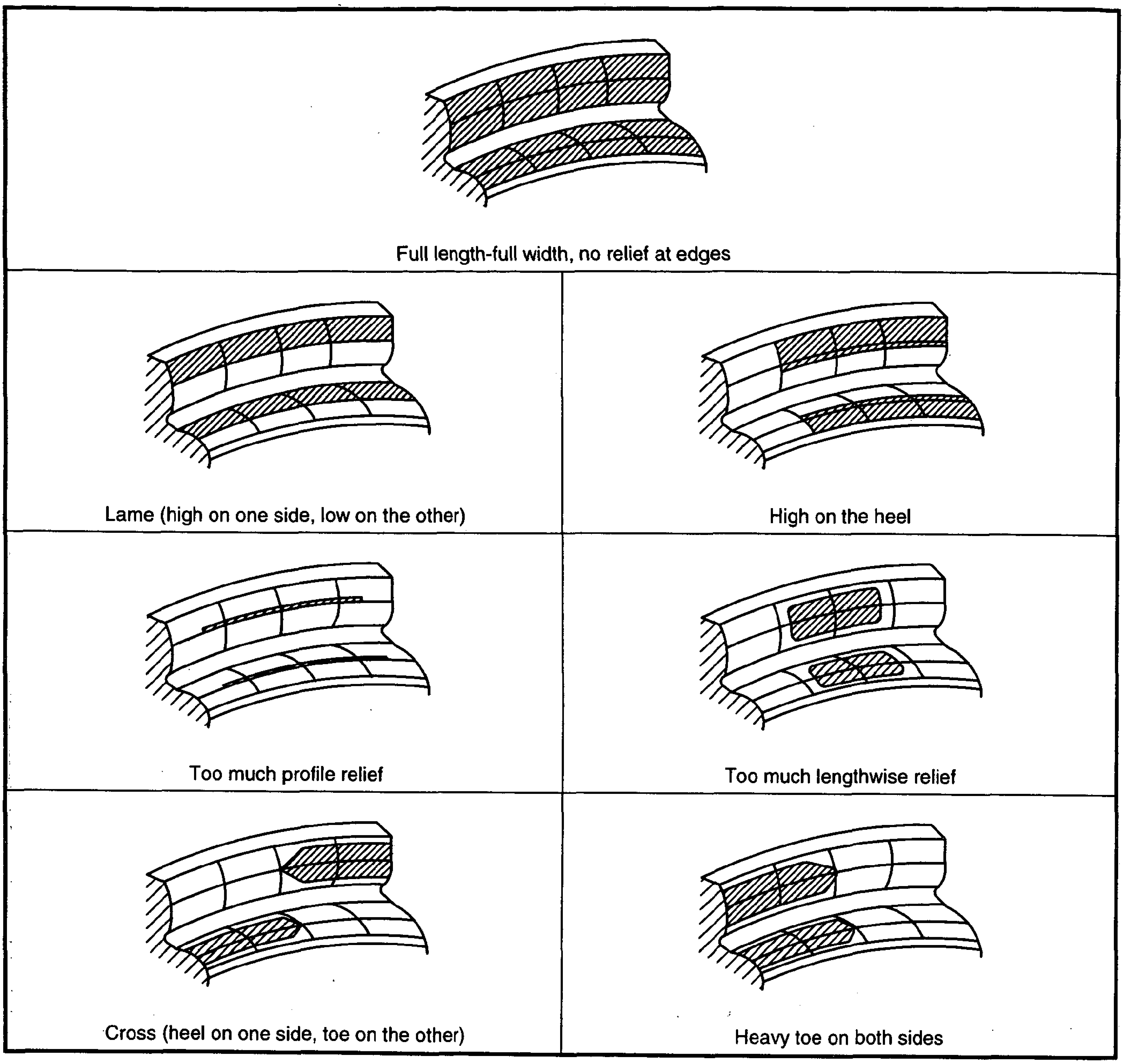


Figure C.2 — Typical unsatisfactory loaded contact patterns

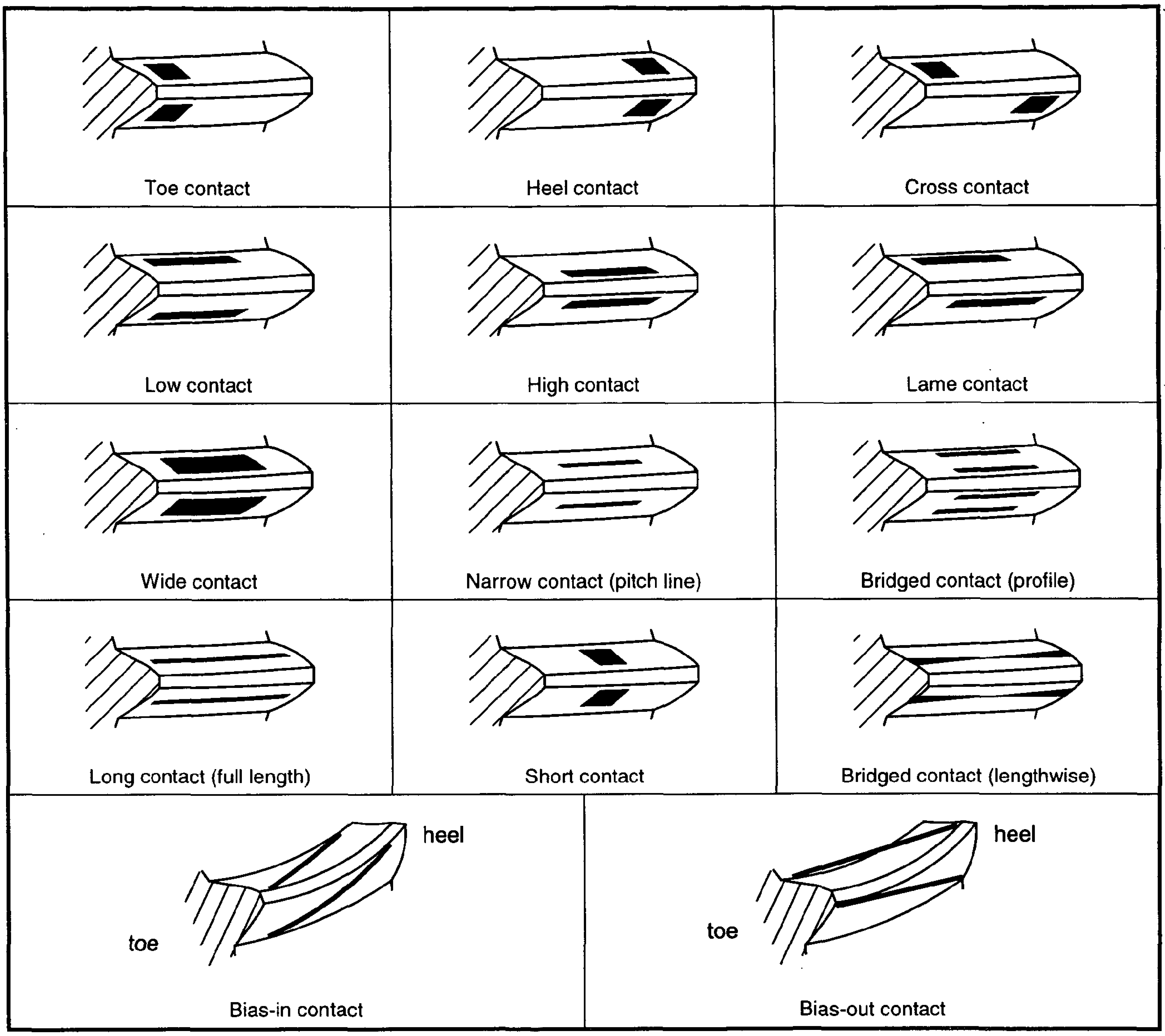


Figure C.3 — Bevel contact-pattern nomenclature

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