TECHNICAL REPORT

ISO/TR 10300-30

First edition
2017-12

Calculation of load capacity of bevel gears —

Part 30:
ISO rating system for bevel and hypoid gears — Sample calculations

Calcul de la capacité de charge des engrenages coniques —

Partie 30: Système d'évaluation ISO pour engrenages conique et hypoïde - Type de calculs

Reference number
ISO/TR 10300-30:2017(E)

Contents

Page

Foreword

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

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

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

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

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

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

A list of all parts in the ISO 10300 series can be found on the ISO website.

Introduction

The ISO 10300 series consists of International Standards, Technical Specifications (TS) and Technical Reports (TR) under the general title *Calculation of load capacity of bevel gears* (see Table 1).

— International Standards contain calculation methods that are based on widely accepted practices and have been validated.

— TS contain calculation methods that are still subject to further development.

— TR contain data that is informative, such as example calculations.

The procedures specified in ISO 10300-1 to ISO 10300-19 cover fatigue analyses for gear rating. The procedures described in ISO 10300-20 to ISO 10300-29 are predominantly related to the tribological behaviour of the lubricated flank surface contact. ISO 10300-30 to ISO 10300-39 include example calculations. The ISO 10300 series allows the addition of new parts under appropriate numbers to reflect knowledge gained in the future.

Requesting standardized calculations according to ISO 10300 without referring to specific parts requires the use of only those parts that are currently designated as International Standards (see Table 1 for listing). When requesting further calculations, the relevant part or parts of ISO 10300 need to be specified. Use of a Technical Specification as acceptance criteria for a specific design need to be agreed in advance between manufacturer and purchaser.

Table 1 — Overview of ISO 10300

Calculation of load capacity of bevel gears	International Standard	Technical Specification	Technical Report
Part 1: Introduction and general influence factors	X		
Part 2: Calculation of surface durability (pitting)	X		
Part 3: Calculation of tooth root strength	X		
Part 4 to 19: to be assigned			
Part 20: to be assigned for scuffing of bevel and hypoid gears			
Part 21 to 29: to be assigned			
Part 30: ISO rating system for bevel and hypoid gears — Sample calculations			X
At the time of publication of this document, some of the parts listed here were under development. Consult the ISO website.			

This document was prepared with sample calculations for different bevel gear designs. They are intended for users of the ISO 10300 series to follow a whole calculation procedure formula by formula. Practical experience has shown that this way, to get into a complex subject, is very helpful.

On the other hand, this document is not intended for use by the average engineer. Rather, it is aimed at the well-versed engineer capable of selecting reasonable values for the parameters and factors in these formulae based on knowledge of similar designs and on awareness of the effects behind these formulae.

Calculation of load capacity of bevel gears —

Part 30: ISO rating system for bevel and hypoid gears — Sample calculations

1 Scope

This document provides sample calculations for different bevel gear designs, how the load capacity is numerically determined according to the methods and formulae of the ISO 10300 series. The initial geometric gear data necessary for these calculations in accordance with ISO 23509.

The term “bevel gear” is used to mean straight, helical (skew), spiral bevel, zerol and hypoid gear designs. Where this document pertains to one or more, but not all, the specific forms are identified.

The manufacturing process of forming the desired tooth form is not intended to imply any specific process, but rather to be general in nature and applicable to all calculation methods of the ISO 10300 series. The fact that there are bevel gear designs with tapered teeth and others where the tooth depth remains constant along the face width (uniform depth) does not demand to apply Method B2 for the first and Method B1 for the second tooth configuration.

The rating system of the ISO 10300 series is based on virtual cylindrical gears and restricted to bevel gears whose virtual cylindrical gears have transverse contact ratios of $\varepsilon_{v\alpha} < 2$. Additionally, the given relations are valid for bevel gears of which the sum of profile shift coefficients of pinion and wheel is zero (see ISO 23509).

WARNING: The user is cautioned that when the formulae are used for large average mean spiral angles, $(\beta_{m1} + \beta_{m2})/2 > 45°$, for effective pressure angles, $\alpha_e > 30°$ and/or for large face widths, $b > 13\ m_{mn}$, the calculated results of the ISO 10300 series should be confirmed by experience.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 10300-1:2014, *Calculation of load capacity of bevel gears — Part 1: Introduction and general influence factors*

ISO 10300-2:2014, *Calculation of load capacity of bevel gears — Part 2: Calculation of surface durability (pitting)*

ISO 10300-3:2014, *Calculation of load capacity of bevel gears — Part 3: Calculation of tooth root strength*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 10300-1 and ISO 10300-2 apply.

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

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at https://www.electropedia.org/

4 Symbols and abbreviated terms

For the purposes of this document, the symbols and units given in ISO 10300-1:2014, Table 1 and Table 2, as well as the abbreviated terms given in ISO 10300-2:2014, Table 1, apply.

Table 2 — Symbols and units used in ISO 10300 (all parts)

Symbol	Description or term	Unit
a	hypoid offset	mm
a_{rel}	relative hypoid offset	—
a_v	centre distance of virtual cylindrical gear pair	mm
a_{vn}	centre distance of virtual cylindrical gear pair in normal section	mm
b	face width	mm
b_b	related base face width	—
b_{ce}	calculated effective face width	mm
b_{eff}	effective face width (e.g. measured length of contact pattern)	mm
b_v	face width of virtual cylindrical gears	mm
$b_{v\,eff}$	effective face width of virtual cylindrical gears	mm
c_{ham}	mean addendum factor of wheel	—
c_v	empirical parameter to determine the dynamic factor	—
c_γ	mean value of mesh stiffness per unit face width	N/(mm · µm)
$c_{\gamma 0}$	mesh stiffness for average conditions	N/(mm · µm)
c'	single stiffness	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_T	tolerance diameter according to ISO 17485	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

Symbol	Description or term	Unit
d_{vbn}	base diameter of virtual cylindrical gear in normal section	mm
d_{vf}	root diameter of virtual cylindrical gear	mm
d_{vn}	reference diameter of virtual cylindrical gear in normal section	mm
e	exponent for the distribution of the load peaks along the lines of contact	—
f	distance from the centre of the zone of action to a contact line	mm
f_{max}	maximum distance to middle contact line	mm
f_{maxB}	maximum distance to middle contact line at right side of the contact pattern	mm
f_{max0}	maximum distance to middle contact line at left side of the contact pattern	mm
f_{pt}	single pitch deviation	µm
$f_{p\ eff}$	effective pitch deviation	µm
$f_{\alpha lim}$	Influence factor of limit pressure angle	
g_c	length of contact line (Method B2)	mm
$g_{v\alpha}$	length of path of contact of virtual cylindrical gear in transverse section	mm
$g_{v\alpha n}$	related length of action in normal section	—
g_J	length of action from mean point to point of load application (Method B2)	mm
g_η	relative length of action within the contact ellipse	mm
h_{am}	mean addendum	mm
h_{a0}	tool addendum	mm
h_{fm}	mean dedendum	mm
h_{fP}	dedendum of the basic rack profile	mm
h_m	mean whole depth used for bevel spiral angle factor	mm
h_{vfm}	relative mean virtual dedendum	—
h_{Fa}	bending moment arm for tooth root stress (load application at tooth tip)	mm
h_N	load height from critical section (Method B2)	mm
j_{en}	outer normal backlash	mm
k'	contact shift factor	—
k_c	clearance factor	—
k_d	depth factor	—
k_{hap}	basic crown gear addendum factor (related to m_{mn})	—
k_{hfp}	basic crown gear dedendum factor (related to m_{mn})	—
k_t	circular thickness factor	—
l_b	length of contact line (Method B1)	mm

Symbol	Description or term	Unit
l_{b0}	theoretical length of contact line	mm
l_{bm}	theoretical length of middle contact line	mm
m_{et}	outer transverse module	mm
m_{mn}	mean normal module	mm
m_{mt}	mean transverse module	mm
m_{red}	mass per unit face width reduced to the line of action of dynamically equivalent cylindrical gears	kg/mm
m^*	related individual gear mass per unit face width referred to the line of action	kg/mm
n	rotational speed	min^{-1}
n_{E1}	resonance speed of pinion	min^{-1}
p	peak load	N/mm
p_{et}	transverse base pitch (Method B2)	mm
p_{max}	maximum peak load	N/mm
p^*	related peak load for calculating the load sharing factor (Method B1)	—
p_{mn}	relative mean normal pitch	—
p_{nb}	relative mean normal base pitch	—
p_{vet}	transverse base pitch of virtual cylindrical gear (Method B1)	mm
q	exponent in the formula for lengthwise curvature factor	—
q_s	notch parameter	—
r_{c0}	cutter radius	mm
r_{mf}	tooth fillet radius at the root in mean section	mm
r_{mpt}	mean pitch radius	mm
$r_{my\,0}$	mean transverse radius to point of load application (Method B2)	mm
r_{va}	relative mean virtual tip radius	—
r_{vn}	relative mean virtual pitch radius	—
s_{mn}	mean normal circular thickness	mm
s_{pr}	amount of protuberance at the tool	mm
s_{Fn}	tooth root chord in calculation section	mm
s_N	one-half tooth thickness at critical section (Method B2)	mm
u	gear ratio of bevel gear	—
u_v	gear ratio of virtual cylindrical gear	—
v_{et}	tangential speed at outer end (heel) of the reference cone	m/s
$v_{et\,max}$	maximum pitch line velocity at operating pitch diameter	m/s
v_g	sliding velocity in the mean point P	m/s
$v_{g\,par}$	sliding velocity parallel to the contact line	m/s
$v_{g\,vert}$	sliding velocity vertical to the contact line	m/s

Symbol	Description or term	Unit
v_{mt}	tangential speed at mid face width of the reference cone	m/s
v_{Σ}	sum of velocities in the mean point P	m/s
$v_{\Sigma h}$	sum of velocities in profile direction	m/s
$v_{\Sigma l}$	sum of velocities in lengthwise direction	m/s
$v_{\Sigma\ vert}$	sum of velocities vertical to the contact line	m/s
w	angle of contact line relative to the root cone	°
x_{hm}	profile shift coefficient	—
x_{sm}	thickness modification coefficient (backlash included)	—
x_{smn}	thickness modification coefficient (theoretical)	—
x_N	tooth strength factor (Method B2)	mm
x_{oo}	distance from mean section to point of load application	mm
y_p	running-in allowance for pitch deviation related to the polished test piece	µm
y_J	location of point of load application for maximum bending stress on path of action (Method B2)	mm
y_3	location of point of load application on path of action for maximum root stress	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	—
z_0	number of blade groups of the cutter	—
A	auxiliary factor for calculating the dynamic factor K_{v-C}	—
A^*	related area for calculating the load sharing factor Z_{LS}	mm
A_{sne}	outer tooth thickness allowance	mm
B	accuracy grade according to ISO 17485	—
C_F	correction factor of tooth stiffness for non-average conditions	—
C_{lb}	correction factor for the length of contact lines	—
C_{ZL}, C_{ZR}, C_{ZV}	constants for determining lubricant film factors	—
E	modulus of elasticity, Young's modulus	N/mm²
E, G, H	auxiliary variables for tooth form factor (Method B1)	—
F	auxiliary variable for mid-zone factor	—
F_{mt}	nominal tangential force at mid face width of the reference cone	N
F_{mtH}	determinant tangential force at mid face width of the reference cone	N
F_n	nominal normal force	N
F_{vmt}	nominal tangential force of virtual cylindrical gears	N
HB	Brinell hardness	—

Symbol	Description or term	Unit
K	constant; factor for calculating the dynamic factor K_{v-B}	—
K_v	dynamic factor	—
K_v*	preliminary dynamic factor for non-hypoid gears	—
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	—
$K_{H\alpha}$*	preliminary transverse load factor for contact stress for non-hypoid gears	—
$K_{H\beta}$	face load factor for contact stress	—
$K_{H\beta-be}$	mounting factor	—
N	reference speed related to resonance speed n_{E1}	—
N_L	number of load cycles	—
P	nominal power	kW
Ra	= CLA = AA arithmetic average roughness	µm
R_e	outer cone distance	mm
R_m	mean cone distance	mm
R_{mpt}	relative mean back cone distance	—
Rz	mean roughness	µm
Rz_{10}	mean roughness for gear pairs with relative curvature radius ρ_{rel} = 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	—
$T_{1,2}$	nominal torque of pinion and wheel	Nm
W_{m2}	wheel mean slot width	mm
$Y_{1,2}$	tooth form factor of pinion and wheel (Method B2)	—
Y_f	stress concentration and stress correction factor (Method B2)	—
Y_i	inertia factor (bending)	—
Y_A	root stress adjustment factor (Method B2)	—
Y_{BS}	bevel spiral angle factor	—
Y_{Fa}	tooth form factor for load application at the tooth tip (Method B1)	—
Y_{FS}	combined tooth form factor for generated gears	—
Y_J	bending strength geometry factor (Method B2)	—
Y_{LS}	load sharing factor (bending)	—

Symbol	Description or term	Unit
Y_{NT}	life factor (bending)	—
$Y_{R\,rel\,T}$	relative surface condition factor	—
Y_{Sa}	stress correction factor for load application at the tooth tip	—
Y_{ST}	stress correction factor for dimensions of the standard test gear	—
Y_X	size factor for tooth root stress	—
$Y_{\delta\,rel\,T}$	relative notch sensitivity factor	—
Y_ε	contact ratio factor for bending (Method B1)	—
Z_i	inertia factor (pitting)	—
Z_v	speed factor	—
Z_A	contact stress adjustment factor (Method B2)	—
Z_E	elasticity factor	—
Z_{FW}	face width factor	—
Z_{Hyp}	hypoid factor	—
Z_I	pitting resistance geometry factor (Method B2)	—
Z_K	bevel gear factor (Method B1)	—
Z_L	lubricant factor	—
Z_{LS}	load sharing factor (Method B1)	—
Z_{M-B}	mid zone factor	—
Z_{NT}	life factor (pitting)	—
Z_R	roughness factor for contact stress	—
Z_S	bevel slip factor	—
Z_W	work hardening factor	—
Z_X	size factor	—
α_a	adjusted pressure angle (Method B2)	°
α_{an}	normal pressure angle at tooth tip	°
$\alpha_{dD,C}$	nominal design pressure angle for drive side/coast side	°
α_{et}	effective pressure angle in transverse section	°
$\alpha_{eD,C}$	effective pressure angle for drive side/coast side	°
α_f	limit pressure angle in wheel root coordinates (Method B2)	°
α_{lim}	limit pressure angle	°
$\alpha_{nD,C}$	generated pressure angle for drive side/coast side	°
α_{vet}	transverse pressure angle of virtual cylindrical gears	°
α_{Fan}	load application angle at tooth tip of virtual cylindrical gear (Method B1)	°
α_L	normal pressure angle at point of load application (Method B2)	°
β_{bm}	mean base spiral angle	°
β_m	mean spiral angle	°

Symbol	Description or term	Unit
β_v	helix angle of virtual gear (Method B1), virtual spiral angle (Method B2)	°
β_{vb}	helix angle at base circle of virtual cylindrical gear	°
β_B	inclination angle of contact line	°
γ	auxiliary angle for length of contact line calculation (Method B1)	°
γ'	projected auxiliary angle for length of contact line	°
γ_a	auxiliary angle for tooth form and tooth correction factor	°
δ	pitch angle of bevel gear	°
δ_a	face angle	°
δ_f	root angle	°
$\varepsilon_{v\alpha}$	transverse contact ratio of virtual cylindrical gears	—
$\varepsilon_{v\alpha n}$	transverse contact ratio of virtual cylindrical gears in normal section	—
$\varepsilon_{v\beta}$	face contact ratio of virtual cylindrical gears	—
$\varepsilon_{v\gamma}$	virtual contact ratio (Method B1), modified contact ratio (Method B2)	—
ε_N	load sharing ratio for bending (Method B2)	—
ε_{NI}	load sharing ratio for pitting (Method B2)	—
ζ_m	pinion offset angle in axial plane	°
ζ_{mp}	pinion offset angle in pitch plane	°
ζ_R	pinion offset angle in root plane	°
θ	auxiliary quantity for tooth form and tooth correction factors	—
θ_{mp}	auxiliary angle for virtual face width (Method B1)	°
θ_{a2}	addendum angle of wheel	°
θ_{f2}	dedendum angle of wheel	°
θ_{v2}	angular pitch of virtual cylindrical wheel	radiant
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 of gear material	kg/mm^3
ρ_{a0}	cutter edge radius	mm
ρ_F	fillet radius at point of contact of 30° tangent	mm
ρ_{Fn}	fillet radius at point of contact of 30° tangent in normal section	mm
ρ_{fP}	root fillet radius of basic rack for cylindrical gears	mm
ρ_{rel}	radius of relative curvature vertical to contact line at virtual cylindrical gears	mm
ρ_t	radius of relative profile curvature (Method B2)	mm
ρ_{va0}	relative edge radius of tool	—
ρ'	slip layer thickness	mm

Symbol	Description or term	Unit
σ_F	tooth root stress	N/mm^2
σ_{F0}	nominal tooth root stress	N/mm^2
$\sigma_{F\,lim}$	nominal stress number (bending)	N/mm^2
σ_{FE}	allowable stress number (bending)	N/mm^2
σ_{FP}	permissible tooth root stress	N/mm^2
σ_H	contact stress	N/mm^2
$\sigma_{H\,lim}$	allowable stress number for contact stress	N/mm^2
σ_{HP}	permissible contact stress	N/mm^2
τ	angle between tangent of root fillet at weakest point and centreline of tooth	°
ν	Poisson's ratio	—
ν_0	lead angle of face hobbing cutter	°
ν_{40}, ν_{50}	nominal kinematic viscosity of the oil at 40 °C and 50 °C, respectively	mm^2/s
ϕ	auxiliary angle to determine the position of the pitch point	°
ω	angular velocity	rad/s
ω_Σ	angle between the sum of velocities vector and the trace of pitch cone	°
χ^X	relative stress drop in notch root	mm^{-1}
χ_T^X	relative stress drop in notch root of standardized test gear	mm^{-1}
Σ	shaft angle	°

Table 3 — Generally used subscripts in ISO 10300 (all parts)

Subscripts	Description
0	tool
1	pinion
2	wheel
A, B, B1, B2, C	value according to Method A, B, B1, B2 or C
D	drive flank
C	coast flank
T	relative to standardized test gear dimensions
(1), (2)	trials of interpolation

5 Application

5.1 General

This document provides four sample calculations:

— Sample 1 is a rating of a spiral bevel gear pair without hypoid offset according to Method B1 and Method B2 (see Annex A);

— Sample 2 is a rating of a hypoid gear set according to Method B1 and Method B2 (see Annex B);

— Sample 3 is a rating of a hypoid gear set according to Method B1 and Method B2 (see Annex C);

— Sample 4 is a rating of a hypoid gear set according to Method B1 and Method B2 (see Annex D).

5.2 Structure of calculation methods

Figure 1 shows three boxes that represent the individual three parts of ISO 10300. However, these boxes are subdivided into a left side where influence factors are determined on the basis of mainly operational data according to Methods A, B or C (see ISO 10300-1:2014, 5.1) and a right side where separate calculation procedures are provided according to Method B1 and Method B2 which are assumed to have the same level B but different approaches. These two methods refer to the determination of virtual cylindrical gears in ISO 10300-1, the gear flank rating formulae in ISO 10300-2 and the gear tooth rating formulae in ISO 10300-3.

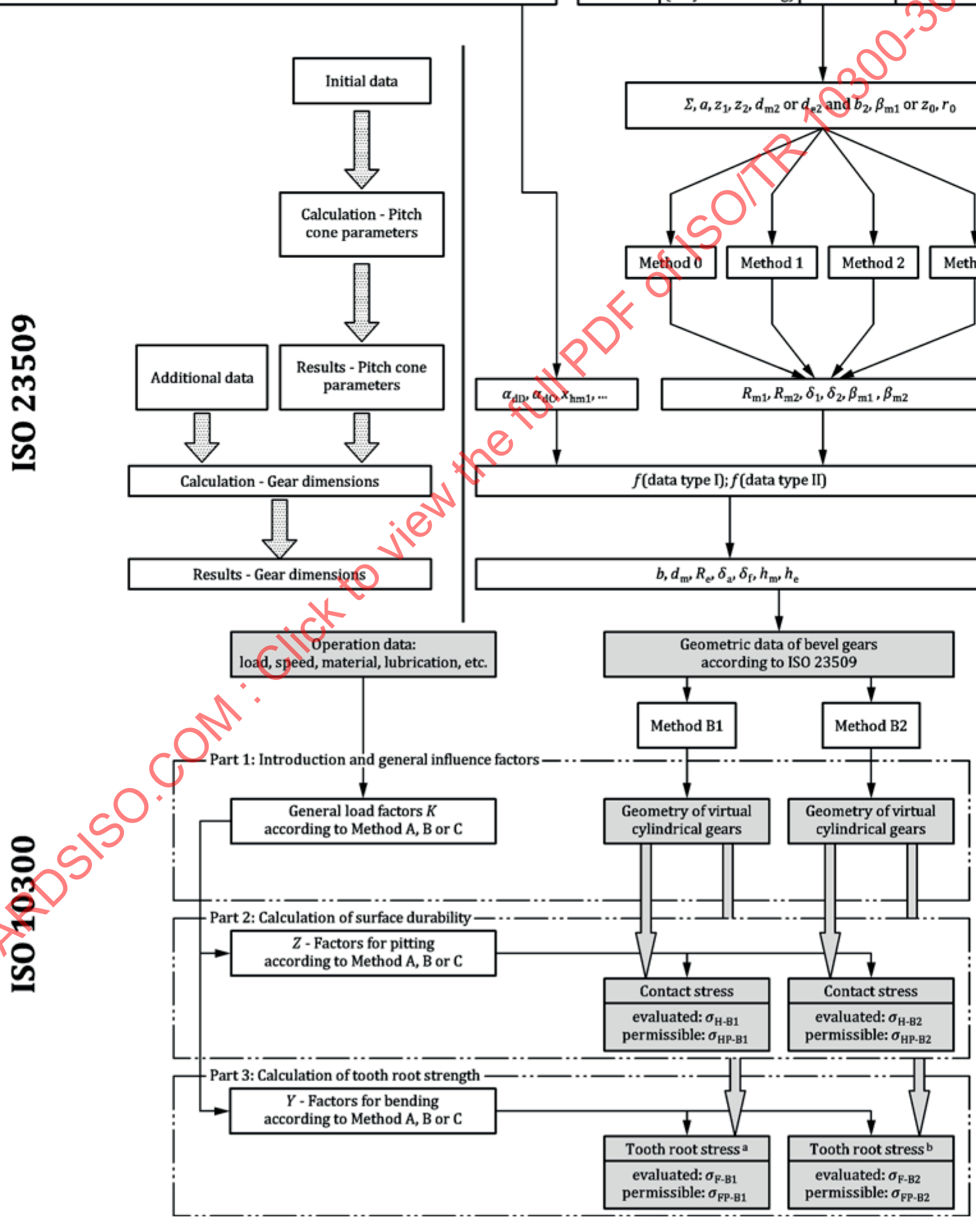
Additional data for calculation of gear dimensions

Data type I		Data type II	
Symbol	**Description**	**Symbol**	**Description**
α_{dD}	nominal design pressure angle - drive side[a]		
α_{dC}	nominal design pressure angle - coast side[a]		
$f_{\alpha lim}$	influence factor of limit pressure angle[a]		
x_{hm1}	profile shift coefficient	c_{ham}	mean addendum factor of wheel
k_{hap}	basic crown gear addendum factor	k_d	depth factor
k_{hfp}	basic crown gear dedendum factor	k_c	clearance factor
x_{smn}	thickness modification coefficient	k_t W_{m2}	thickness factor or wheel mean slot width
j_{mn}, j_{mt2}, j_{en}, j_{et2}	backlash (choice of four)		
θ_{a2}	addendum angle of wheel		
θ_{f2}	dedendum angle of wheel		

[a] Generally drive and coast side pressure angles are balanced in initial design. However, some applications may be optimized with unbalanced pressure angles, see Annex C for guidance.

Initial data for the calculation of the pitch cone parameters

Symbol	Description	Method 0	Method 1	Method 2	Method 3
Σ	shaft angle	X	X	X	X
a	hypoid offset	0,0	X	X	X
$z_{1,2}$	number of teeth	X	X	X	X
d_{m2}	mean pitch diameter of wheel	–	–	X	–
d_{e2}	outer pich diameter of wheel	X	X	–	X
b_2	wheel face width	X	X	X	X
β_{m1}	mean spiral angle of pinion	–	X	–	–
β_{m2}	mean spiral angle of wheel	X	–	X	X
r_{c0}	cutter radius	X	X	X	X
z_0	number of blade groups (only face hobbing)	X	X	X	X



[a] One set of formulae for both, bevel and hypoid gears.

[b] Separate sets of formulae for bevel and for hypoid gears.

Figure 1 — Structure of calculation methods in ISO 10300 (all parts)

Annex A
(informative)

Sample 1: Rating of a spiral bevel gear pair without hypoid offset according to Method B1 and Method B2

A.1 Initial data

Sample 1 is for a spiral bevel gear pair without hypoid offset which uses Method 0 according to ISO 23509.

Table A.1 — Initial data for pitch cone parameters

Symbol	Description	Method 0	Method 1	Method 2	Method 3
Σ	shaft angle	90°	X	X	X
a	hypoid offset	0 mm	X	X	X
$z_{1,2}$	number of teeth	14/39	X	X	X
d_{m2}	mean pitch diameter of wheel	—	—	X	—
d_{e2}	outer pitch diameter of wheel	176,893 mm	X	—	X
b_2	wheel face width	25,4 mm	X	X	X
β_{m1}	mean spiral angle of pinion	35°	X	—	—
β_{m2}	mean spiral angle of wheel	35°	—	X	X
r_{c0}	cutter radius	114,3 mm	X	X	X
z_0	number of blade groups (only face hobbing)	—	—	X	X

Table A.2 — Input data for tooth profile parameters

Data type I		Data type II	
Symbol	**Description**	**Symbol**	**Description**
α_{dD}	20°		
α_{dC}	20°		
$f_{\alpha lim}$	0		
x_{hm1}	—	c_{ham}	0,247 37
k_{hap}	—	k_d	2,000
k_{hfp}	—	k_c	0,125
x_{smn}	—	k_t	0,091 5
		W_{m2}	—
j_{en}	0,127 mm		
θ_{a2}	2,134 2°		
θ_{f2}	6,493 4°		
$\rho_{a01D,C}$	0,8 mm/0,8 mm		
$\rho_{a02D,C}$	1,2 mm/1,2 mm		
$s_{pr1D,C}$	0 mm/0 mm		
$s_{pr2D,C}$	0 mm/0 mm		

Table A.3 and Table A.4 show geometric and operational data and text for explanation.

Table A.3 — Geometric data from calculation according to ISO 23509

Symbol	Description	Values	Symbol	Description	Value
$d_{m1,2}$	mean pitch diameter of pinion/wheel	54,918 mm/ 152,987 mm	ζ_{mp}	offset angle on pitch plane	0°
$h_{am1,2}$	mean addendum of pinion/wheel	4,836 mm/ 1,591 mm	ζ_R	pinion offset angle on root plane	0°
$h_{fm1,2}$	mean dedendum of pinion/wheel	2,394 mm/ 5,639 mm	$R_{e1,2}$	outer cone distance on pinion and wheel	93,973 mm
$\alpha_{eD,C}$	effective pressure angle for drive side/coast side	20°/20°	$R_{m1,2}$	mean cone distance on pinion and wheel	81,273 mm
$\alpha_{nD,C}$	generated pressure angle for drive side/coast side	20°/20°	$\delta_{1,2}$	pitch angle on pinion/wheel	19,747°/ 70,253°
α_{lim}	limit pressure angle	0°	$\delta_{a1,2}$	face angle on pinion/wheel	26,240°/ 72,387°
m_{mn}	mean normal module	3,213 mm	$\delta_{f1,2}$	root angle on pinion/wheel	17,613°/ 63,760°
k_{hfp}	basic crown gear dedendum factor	1,25	$x_{sm1,2}$	thickness modification coefficient on pinion/wheel	0,037/ −0,055
ζ_m	pinion offset angle on axial plane	0,000°	m_{et2}	outer transverse module	4,536 mm
$s_{mn1,2}$	mean normal circular tooth thickness of pinion/wheel	6,465 mm/ 3,511 mm			

Table A.4 — Operation parameters and additional considerations

Symbol	Description	Value
Additional data		
	wheel profile	generated
	roughing/finishing method	face milling
$b_{2\text{eff}}$	effective face width on wheel	$0{,}85 \cdot b_2$
	profile crowning	low
	verification of contact pattern	checked under light test load for each gear
	mounting conditions of pinion and wheel	one member cantilever-mounted
Operation parameters		
T_1	pinion torque	300 Nm
n_1	pinion rotational speed	1 200 min^{-1}
K_A	application factor	1,1
	active flank	drive
Material data for pinion and wheel (case hardened steel)		
$\sigma_{H\,\text{lim}}$	allowable stress number (contact)	1 500 N/mm^2
$\sigma_{F\,\text{lim}}$	nominal stress number (bending)	480 N/mm^2
	surface hardness	same for pinion and wheel
Quality parameters		
R_z	flank roughness on pinion/wheel	8 µm/8 µm
R_z	tooth root roughness on pinion/wheel	16 µm/16 µm
f_{pt}	single pitch deviation on pinion/wheel	12 µm/26 µm
Lubrication parameters		
	oil type	ISO-VG-150
	oil temperature	90 °C

A.2 Calculation of Sample 1 according to Method B1

Table A.5 — Virtual cylindrical gears

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Virtual cylindrical gears in transverse section						
Reference diameter on pinion	$d_{v1} = \frac{d_{m1}}{\cos\delta_1}$	(A.1)	58,349 mm	E: A.1		
Reference diameter on wheel	$d_{v2} = u^2 d_{v1}$	(A.2)	452,802 mm	E: A.4		
Helix angle	$\beta_v = \frac{\beta_{m1} + \beta_{m2}}{2}$	(A.3)	35°	E: A.8		
Transverse pressure angle of virtual cylindrical gears	$\alpha_{vet} = \arctan\left(\tan\alpha_e / \cos\beta_v\right)$ since $\alpha_e = \alpha_{eD}$ for drive side	(A.4)	23,957°	E: A.10		
Transverse module	$m_{vt} = m_{mn} / \cos\beta_v$	(A.5)	3,922 mm	E: A.11		
Number of teeth on pinion	$z_{v1} = d_{v1} / m_{vt}$	(A.6)	14,876	E: A.12		
Number of teeth on wheel	$z_{v2} = d_{v2} / m_{vt}$	(A.7)	115,441	E: A.12		
Gear ratio	$u_v = z_{v2} / z_{v1}$	(A.8)	7,760	E: A.13		

Table A.5 — *continued*

Description	Formula	Result	References to ISO 10300-1 (E: Formula)	References to ISO 10300-2 (T: Table)	References to ISO 10300-3 (F: Figure)
Auxiliary angle for virtual face width	$\theta_{mp} = \arctan\left(\sin\delta_2 \tan\zeta_m\right)$ (A.9)	0°	E: A.21		
Projected auxiliary angle for length of contact line	$\gamma' = \theta_{mp} - \zeta_{mp}/2$ (A.10)	0°	E: A.20		
Centre distance of virtual cylindrical gear pair	$a_v = \left(d_{v1} + d_{v2}\right) / 2$ (A.11)	255,576 mm	E: A.5		
Helix angle of virtual cylindrical gear at base circle	$\beta_{vb} = \arcsin\left(\sin\beta_v \cos\alpha_e\right)$ since $\alpha_e = \alpha_{eD}$ for drive side (A.12)	32,615°	E: A.16		
Tip diameter on pinion	$d_{va1} = d_{v1} + 2\, h_{am1}$ (A.13)	68,021 mm	E: A.6		
Tip diameter on wheel	$d_{va2} = d_{v2} + 2\, h_{am2}$ (A.14)	455,984 mm	E: A.6		
Root diameter on pinion	$d_{vf1} = d_{v1} - 2\, h_{fm1}$ (A.15)	53,561 mm	E: A.7		
Root diameter on wheel	$d_{vf2} = d_{v2} - 2\, h_{fm2}$ (A.16)	441,524 mm	E: A.7		
Base diameter on pinion	$d_{vb1} = d_{v1} \cos\alpha_{vet}$ (A.17)	53,323 mm	E: A.9		
Base diameter on wheel	$d_{vb2} = d_{v2} \cos\alpha_{vet}$ (A.18)	413,794 mm	E: A.9		
Transverse base pitch	$p_{vet} = \pi\, m_{mn} \cos\alpha_{vet} / \cos\beta_v$ (A.19)	11,261 mm	E: A.17		

Table A.5 — *continued*

Description	Formula		Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Length of path of contact of virtual cylindrical gear in transverse section	$g_{v\alpha} = \frac{1}{2}\left[\left(\sqrt{d_{va1}^2 - d_{vb1}^2} - d_{v1}\sin\alpha_{vet}\right) + \left(\sqrt{d_{va2}^2 - d_{vb2}^2} - d_{v2}\sin\alpha_{vet}\right)\right]$	(A.20)	13,121 mm	E: A.18		
Transverse contact ratio	$\varepsilon_{v\alpha} = g_{v\alpha} / p_{vet}$	(A.21)	1,165	E: A.23		
Effective face width with $b_{2\,eff} = 0{,}85 \cdot b_2$	$b_{v\,eff} = \frac{\left[b_{2\,eff} / \cos\left(\zeta_{mp}/2\right) - g_{v\alpha}\cos\alpha_{vet}\tan\left(\zeta_{mp}/2\right)\right]}{1 - \tan\gamma' \tan\left(\zeta_{mp}/2\right)}$	(A.22)	21,590 mm	E: A.19		
Face width	$b_v = b_2 \frac{b_{v\,eff}}{b_{2\,eff}}$	(A.23)	25,400 mm	E: A.22		
Virtual cylindrical gears in normal section						
Number of pinion teeth of virtual cylindrical gears	$z_{vn1} = \frac{z_{v1}}{\cos^2\beta_{vb}\cos\beta_v}$	(A.24)	25,596	E: A.38		
Number of wheel teeth of virtual cylindrical gears	$z_{vn2} = u_v z_{vn1}$	(A.25)	198,632	E: A.39		
Reference diameter on pinion	$d_{vn1} = z_{vn1} m_{mn}$	(A.26)	82,241 mm	E: A.40		
Reference diameter on wheel	$d_{vn2} = z_{vn2} m_{mn}$	(A.27)	638,203 mm	E: A.40		

Table A.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Tip diameter on pinion	$d_{\text{van1}} = d_{\text{vn1}} + 2h_{\text{am1}}$	(A.28)	91,913 mm	E: A.41		
Tip diameter on wheel	$d_{\text{van2}} = d_{\text{vn2}} + 2h_{\text{am2}}$	(A.29)	641,385 mm	E: A.41		
Base diameter on pinion	$d_{\text{vbn1}} = d_{\text{vn1}} \cos\alpha_{\text{e}}$	(A.30)	77,281 mm	E: A.42		
Base diameter on wheel	$d_{\text{vbn2}} = d_{\text{vn2}} \cos\alpha_{\text{e}}$	(A.31)	599,715 mm	E: A.42		
Face contact ratio	$\varepsilon_{\text{v}\beta} = \dfrac{b_{\text{v eff}} \sin\beta_{\text{v}}}{\pi\, m_{\text{mn}}}$	(A.32)	1,227	E: A.24		
Virtual contact ratio	$\varepsilon_{\text{v}\gamma} = \varepsilon_{\text{v}\alpha} + \varepsilon_{\text{v}\beta}$	(A.33)	2,392	E: A.25		
Inclination angle of contact line	$\beta_{\text{B}} = \arctan\left(\tan\beta_{\text{v}} \sin\alpha_{\text{e}}\right)$	(A.34)	13,468°	E: A.36		
Radius of relative curvature in normal section at the mean point	$\rho_{\text{t}} = \left[\dfrac{1}{\cos\alpha_{\text{nD}}\left(\tan\alpha_{\text{nD}} - \tan\alpha_{\text{lim}}\right) + \tan\zeta_{\text{mp}} \tan\beta_{\text{B}}} \times \dfrac{\cos\beta_{\text{m1}} \cos\beta_{\text{m2}}}{\cos\zeta_{\text{mp}}} \times \left(\dfrac{1}{R_{\text{m2}} \tan\delta_2} + \dfrac{1}{R_{\text{m1}} \tan\delta_1}\right)\right]^{-1}$	(A.35)	13,173 mm	E: A.37a		
Radius of relative curvature vertical to the contact line	$\rho_{\text{rel}} = \left\|\rho_{\text{t}}\right\| \cos^2\beta_{\text{B}}$	(A.36)	12,459 mm	E: A.35		

Table A.6 — General influence factors

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Nominal tangential force of bevel gears	$F_{\mathrm{mt}\,1} = \dfrac{2\ 000\ T_1}{d_{\mathrm{m1}}}$	(A.37)	10 925,4 N	E: 1		
Nominal tangential force of virtual cylindrical gears	$F_{\mathrm{vmt}} = F_{\mathrm{mt1}} \dfrac{\cos\beta_{\mathrm{v}}}{\cos\beta_{\mathrm{m1}}}$	(A.38)	10 925,4 N	E: 2		
Nominal tangential speed at mean point of the pinion	$v_{\mathrm{mt}\,1} = \dfrac{d_{\mathrm{m}\,1}\ n_1}{19\,098}$	(A.39)	3,451 m/s	E: 5		
Nominal tangential speed at mean point of the wheel	$v_{\mathrm{mt}\,2} = \dfrac{d_{\mathrm{m}\,2}\ n_2}{19\,098}$	(A.40)	3,451 m/s	E: 5		
Correction factor for non-average conditions for $F_{\mathrm{vmt}}\,K_{\mathrm{A}} / b_{\mathrm{veff}} \geq$ 100 N/mm	C_{F}	(A.41)	1,000	E: 12a		
Mean value of mesh stiffness per unit face width	$c_{\gamma} = c_{\gamma\,0} C_{\mathrm{F}}$	(A.42)	20 N/(mm · µm)	E: 11		
Single stiffness	$c' = c'_0\ C_{\mathrm{F}}$	(A.43)	14 N/(mm · µm)	E: 17		
Max. single pitch deviation as given in Table A.4.	f_{pt}	(A.44)	26 µm			

Table A.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Running-in allowance for case hardened and nitrided gears	$y_\alpha = 0{,}075\, f_{pt}$	(A.45)	1,950 µm	E: 43		
Effective pitch deviation with $y_p = y_\alpha$	$f_{p\,eff} = f_{pt} - y_p$	(A.46)	24,050 µm	E: 16		
Relative pinion mass per unit face width reduced to the line of action	$m_1^* = \frac{1}{8} \rho\, \pi \frac{d_{m1}^2}{\cos^2\left[\left(\alpha_{nD} + \alpha_{nC}\right)/2\right]}$ where ρ is the density of the gear material (for steel $\rho = 7{,}86 \cdot 10^{-6}$ kg/mm^3)	(A.47)	0,011 kg/mm	E: 13		
Relative wheel mass per unit face width reduced to the line of action	$m_2^* = \frac{1}{8} \rho\, \pi \frac{d_{m2}^2}{\cos^2\left[\left(\alpha_{nD} + \alpha_{nC}\right)/2\right]}$ where ρ is the density of the gear material (for steel $\rho = 7{,}86 \cdot 10^{-6}$ kg/mm^3)	(A.48)	0,082 kg/mm	E: 13		
Mass reduced to the line of action of the dynamically equivalent cylindrical gear pair	$m_{red} = \frac{m_1^* m_2^*}{m_1^* + m_2^*}$	(A.49)	0,009 kg/mm	E: 10		
Resonance speed of pinion	$n_{E1} = \frac{30 \times 10^3}{\pi\, z_1} \sqrt{\frac{c_\gamma}{m_{red}}}$	(A.50)	31 565 min^{-1}	E: 9		

Table A.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Dimensionless reference speed	$N = \dfrac{n_1}{n_{E1}}$	(A.51)	0,038	E: 8		
For virtual contact ratio, $\varepsilon_{v\gamma} = 2{,}392 > 2$ as given in ISO 10300‑1:2014, Table 1	$c_{v1,2} = c_{v1} + c_{v2}$	(A.52)	0,592	T: 3		
	c_{v3}	(A.53)	0,115	T: 3		
	c_{v4}	(A.54)	0,473	T: 3		
	$c_{v5,6}$	(A.55)	0,654	T: 3		
	c_{v7}	(A.56)	0,993	T: 3		
Constant for the dynamic factor with $K_A = 1{,}1$ as given in Table A.4.	$K = \dfrac{b_v\ f_{p\ eff} c'}{F_{vmt}\ K_A} c_{v1,2} + c_{v3}$	(A.57)	0,537	E: 15		
Dynamic factor	$K_v - B_1 = N \cdot K + 1$	(A.58)	1,020	E: 14		
Determination of the length of contact lines						
For virtual contact ratio, $\varepsilon_{v\beta} = 1{,}227$, $\varepsilon_{v\beta} \geq 1$	$f_t = +p_{vet} \cos\beta_{vb}$	(A.59)	9,485 mm	T: A.2		
	f_m	(A.60)	0,000 mm	T: A.2		
	$f_r = -p_{vet} \cos\beta_{vb}$	(A.61)	−9,485 mm	T: A.2		

Table A.6 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Maximum distances from middle contact line	$f_{\text{maxB}} = \frac{1}{2}\left[g_{\text{v}\alpha} + b_{\text{v eff}}\left(\tan\gamma + \tan\beta_{\text{vb}}\right)\right]\cos\beta_{\text{vb}}$ (A.62)	11,345 mm	E: A.31		
	$f_{\text{max0}} = \frac{1}{2}\left[g_{\text{v}\alpha} - b_{\text{v eff}}\left(\tan\gamma + \tan\beta_{\text{vb}}\right)\right]\cos\beta_{\text{vb}}$ (A.63)	−0,292 mm	E: A.32		
	$f_{\text{max}} = f_{\text{maxB}}$ since $f_{\text{maxB}} > f_{\text{max0}}$ (A.64)	11,345 mm			
Theoretical length of contact line	$l_{\text{b0}} = \sqrt{\left(x_1 - x_2\right)^2 + \left(y_1 - y_2\right)^2}$ (A.65)	24,345 mm	E: A.27		

Table A.6 — *continued*

Description	Formula	Result	References to ISO 10300-1 E: Formula	References to ISO 10300-2 T: Table	References to ISO 10300-3 F: Figure
Theoretical length of middle contact line calculated with $f=f_m$ for contact stress as specified in ISO 10300−2:2014, 6.1	$x_1 = \dfrac{f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \dfrac{b_{v\,eff}}{2}\right) + \dfrac{1}{2}\left(g_{v\alpha} + b_{v\,eff}\tan\gamma\right)}{\tan\gamma + \tan\beta_{vb}}$ (A.66)	21,048 mm	E: A.28		
	$x_2 = \dfrac{f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \dfrac{b_{v\,eff}}{2}\right) - \dfrac{1}{2}\left(g_{v\alpha} - b_{v\,eff}\tan\gamma\right)}{\tan\gamma + \tan\beta_{vb}}$ (A.67) NOTE ISO 10300−1:2014, Formula (A.29) is a misprint. The operator in the second parenthesis should be “–”.	0,542 mm	E: A.29		
	$y_1 = -x_1\tan\beta_{vb} + f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \dfrac{b_{v\,eff}}{2}\right)$ (A.68)	−6,561 mm	E: A.30		
	$y_2 = -x_2\tan\beta_{vb} + f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \dfrac{b_{v\,eff}}{2}\right)$ (A.69)	6,561 mm	E: A.30		
Correction factor	$C_{lb} = \sqrt{\left[1 - \left(\dfrac{f}{f_{max}}\right)^2\right]\left(1 - \sqrt{\dfrac{b_{veff}}{b_v}}\right)^2}$ (A.70)	0,078	E: A.34		
Length of contact line	$l_b = l_{b0}\left(1 - C_{lb}\right)$ (A.71)	22,445 mm	E: A.26		

Table A.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Length of middle contact line	$l_{bm} = l_b$	(A.72)	22,445 mm			
Load sharing factor (pitting)						
Exponent for calculation of parabolic distribution of peak loads	e	(A.73)	3		T: 3	
Related peak load	$p^* = 1 - \left(\frac{\lvert f \rvert}{\lvert f_{max} \rvert} \right)^e$	(A.74)			E: 7 F: 2	
Related peak load at tip contact line	$p_t^* = 1 - \left(\frac{\lvert f_t \rvert}{\lvert f_{max} \rvert} \right)^e$	(A.75)	0,416			
Related peak load at middle contact line	$p_m^* = 1 - \left(\frac{\lvert f_m \rvert}{\lvert f_{max} \rvert} \right)^e$	(A.76)	1,000			
Related peak load at root contact line	$p_r^* = 1 - \left(\frac{\lvert f_r \rvert}{\lvert f_{max} \rvert} \right)^e$	(A.77)	0,416			
Related area	$A^* = \frac{1}{4} p^* l_b \pi$	(A.78)			E: 8 F: 2	

Table A.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Related area at tip contact line	$A_t^* = \frac{1}{4} p_t^* \ l_{bt} \pi$	(A.79)	1,279 mm			
Related area at middle contact line	$A_m^* = \frac{1}{4} p_m^* \ l_{bm} \pi$	(A.80)	17,628 mm			
Related area at root contact line	$A_r^* = \frac{1}{4} p_r^* \ l_{br} \pi$	(A.81)	1,279 mm			
Load sharing factor	$Z_{LS} = \sqrt{\frac{A^*_m}{A^*_t + A^*_m + A^*_r}}$	(A.82)	0,934		E: 10	
Face load factors (Calculation according to Method C)						
Load distribution factor	$K_{H\beta\text{-}C} = 1{,}5\, K_{H\beta\text{-}be}$	(A.83)	1,650	E: 27		
with	$K_{H\beta\text{-}be}$	(A.84)	1,100	T:4		
Load distribution factor	$K_{F\beta\text{-}C} = K_{H\beta\text{-}C} / K_{F0}$	(A.85)	1,650	E: 28		
with	K_{F0}	(A.86)	1,000	E: 29b		
Transverse load factors (Calculation according to Method B)						
Determinant tangential force at mid face width on the pitch cone	$F_{mtH} = F_{vmt}\, K_A\, K_v\, K_{H\beta}$	(A.87)	20 226,2 N	E: 37		

Table A.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Transverse load factors for bevel gear with virtual contact ratio $\varepsilon_{v\gamma}$ = 2,392 > 2	$K^*_{H\alpha} = K^*_{F\alpha} = 0{,}9 + 0{,}4\sqrt{\frac{2\left(\varepsilon_{v\gamma} - 1\right)}{\varepsilon_{v\gamma}}} \cdot \frac{c_\gamma\left(f_{pt} - y_\alpha\right)}{F_{mtH}/b_v}$	(A.88)	1,161	E: 38		
Relative hypoid offset	$a_{rel} = \frac{2\left\|a\right\|}{d_{m2}}$	(A.89)	0,000	E: 35		
Transverse load factors	$K_{H\alpha} = K_{F\alpha} = K^*_{H\alpha} - \frac{K^*_{H\alpha} - 1}{0{,}1} a_{rel}$	(A.90)	1,161	E: 34		

Table A.7 — Calculation of surface durability (pitting)

Description	Formula		Result	References to		
				ISO 10300–1	ISO 10300–2	ISO 10300–3
				E: Formula	T: Table	F: Figure
Z-factors						
Factors for calculation of mid-zone factor for $\varepsilon_{v\beta} = 1{,}227 \geq 1$	$F_1 = \varepsilon_{v\alpha}$	(A.91)	1,165		T: 2	
	$F_2 = \varepsilon_{v\alpha}$	(A.92)	1,165		T: 2	
Mid-zone factor	$Z_{M\text{-}B} = \dfrac{\tan\alpha_{vet}}{\sqrt{\left[\sqrt{\left(\dfrac{d_{va1}}{d_{vb1}}\right)^2 - 1} - F_1 \dfrac{\pi}{z_{v1}}\right] \cdot \left[\sqrt{\left(\dfrac{d_{va2}}{d_{vb2}}\right)^2 - 1} - F_2 \dfrac{\pi}{z_{v2}}\right]}}$	(A.93)	0,916		E: 6	
Elasticity factor	$Z_E = \sqrt{\dfrac{1}{\pi\left(\dfrac{1-\nu_1^2}{E_1} + \dfrac{1-\nu_2^2}{E_2}\right)}}$	(A.94)	189,800 (N/mm2)1/2		E: 51	
Bevel gear factor	ZK	(A.95)	0,850		E: 11	

Table A.7 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Lubricant factor	$C_{ZL} = 0{,}08 \frac{\sigma_{H\,lim} - 850}{350} + 0{,}83$ (A.96) using σ_{Hlim} = 1 200 N/mm², which is the upper allowed limit in the above formula.	0,910		E: 54	
	$Z_L = C_{ZL} + \frac{4\left(1{,}0 - C_{ZL}\right)}{\left(1{,}2 + \frac{134}{v_{40}}\right)^2}$ (A.97)	0,992		E: 53	
Speed factor	$C_{ZV} = 0{,}08 \frac{\sigma_{H\,lim} - 850}{350} + 0{,}85$ (A.98) using σ_{Hlim} = 1 200 N/mm², which is the upper allowed limit in the above formula.	0,930		E: 56	
	$Z_v = C_{ZV} + \frac{2\left(1{,}0 - C_{ZV}\right)}{\sqrt{0{,}8 + \frac{32}{v_{mt2}}}}$ (A.99)	0,974		E: 55	

Table A.7 — *continued*

Description	Formula	Result	References to ISO 10300-1 (E: Formula)	References to ISO 10300-2 (T: Table)	References to ISO 10300-3 (F: Figure)
Roughness factor with the radius of relative curvature $\rho = \rho_{rel} = 12{,}459$ mm for Method B1 (see ISO 10300–1:2014, Annex A)	$Rz_{10} = \frac{Rz_1 + Rz_2}{2} \cdot \sqrt[3]{\frac{10}{\rho}}$ (A.100)	7,435 µm		E: 57	
	$C_{ZR} = 0{,}12 + \frac{1\,000 - \sigma_{H\,lim}}{5\,000}$ (A.101) using σ_{Hlim} = 1 200 N/mm^2, which is the upper allowed limit in the above formula.	0,080		E: 59	
	$Z_R = \left(\frac{3}{Rz_{10}}\right)^{C_{ZR}}$ (A.102)	0,930		E: 58	
Product of the lubricant influence factors	$Z_L Z_v Z_R$ (A.103)	0,899			
Size factor	Z_X for Method B1 (see ISO 10300–1:2014, 6.5.1) (A.104)	1,000			
Hypoid factor	*ZHyp* (A.105) *Set ZHyp = 1,0 for non-offset.*	1,000		E: 12	
Life factor for pinion	*ZNT,1* (A.106)	1,000		T: 4	
Life factor for wheel	*ZNT,2* (A.107)	1,000		T: 4	
Work hardening factor	$Z_W = 1{,}2 - \frac{HB - 130}{1\,700}$ (A.108) *Set Zw = 1,0 because pinion and wheel with equal hardness.*	1,000		E: 60	

Table A.7 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Contact stress formula						
Nominal normal force of the virtual cylindrical gear at mean point P	$F_{n} = \frac{F_{mt1}}{\cos \alpha_{nD} \cos \beta_{m1}}$	(A.109)	14 193,4 N		E: 3	
Nominal value of the contact stress	$\sigma_{H0} = \sqrt{\frac{F_{n}}{l_{bm} \rho_{rel}}} Z_{M\text{-}B} Z_{LS} Z_{E} Z_{K}$	(A.110)	983,6 N/mm^2		E: 2	
Contact stress	$\sigma_{H} = \sigma_{H0} \sqrt{K_{A} K_{v} K_{H\beta} K_{H\alpha}}$	(A.111)	1 442,0 N/mm^2		E: 1	
Permissible contact stress	$\sigma_{HP} = \sigma_{H\,lim} Z_{NT} Z_{X} Z_{L} Z_{v} Z_{R} Z_{W} Z_{Hyp}$	(A.112)	1 348,2 N/mm^2		E: 4	
Calculated safety factor for contact stress (pitting) on pinion and wheel	$S_{H1,2} = \frac{\sigma_{HP1,2}}{\sigma_{H1,2}}$	(A.113)	0,935		E: 5	

Table A.8 — Calculation of tooth root strength for pinion

Description	Formula	Result	References to ISO 10300-1 E: Formula	ISO 10300-2 T: Table	ISO 10300-3 F: Figure
Y-factors					
Load sharing factor	$Y_{LS} = Z_{LS}^2$ (A.114)	0,873			E: 35
Geometry values for pinion according to Tables A.2 and A.3.	$h_{a0} = h_{fp} = k_{hfp}\, m_{mn}$ (A.115)	4,017 mm			F: 2
Parameters for pinion	$E_{1,D} = \left(\frac{\pi}{4} - x_{sm1}\right) m_{mn} - h_{a0} \tan\alpha_{eD} - \frac{\rho_{a01,D}\left(1 - \sin\alpha_{eD}\right) - s_{pr1,D}}{\cos\alpha_{eD}}$ (A.116)	0,383 mm			E: 7
	$E_{1,C} = \left(\frac{\pi}{4} - x_{sm1}\right) m_{mn} - h_{a0} \tan\alpha_{eC} - \frac{\rho_{a01,C}\left(1 - \sin\alpha_{eC}\right) - s_{pr1,C}}{\cos\alpha_{eC}}$ (A.117)	0,383 mm			E: 7
	$G_{1,D} = \frac{\rho_{a01,D}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm1}$ (A.118)	−0,496			E: 8
	$G_{1,C} = \frac{\rho_{a01,C}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm1}$ (A.119)	−0,496			E: 8

Table A.8 — *continued*

Description	Formula	Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Parameters for pinion	$H_{1,D} = \frac{2}{z_{vn1,D}}\left(\frac{\pi}{2} - \frac{E_{1,D}}{m_{mn}}\right) - \frac{\pi}{3}$ (A.120)	−0,934			E: 9
	$H_{1,C} = \frac{2}{z_{vn1,C}}\left(\frac{\pi}{2} - \frac{E_{1,C}}{m_{mn}}\right) - \frac{\pi}{3}$ (A.121)	−0,934			E: 9
Iteration starting with $\theta = \pi/6$ until $(\theta_{new} - \theta) <$ 0,000 001	$\theta_{1,D,C} = \frac{2G_{1,D,C}}{z_{vn1,D,C}}\tan\theta_{1,D,C} - H_{1,D,C}$ (A.122)	Initial: 52,218°			E: 10
		Final: 50,779°			E: 10
Tooth root chordal thickness on the drive side	$s_{Fn1,D} = m_{mn} z_{vn1,D} \sin\left(\frac{\pi}{3} - \theta_{1,D}\right) + m_{mn}\sqrt{3}\left(\frac{G_{1,D}}{\cos\theta_{1,D}} - \frac{\rho_{a01,D}}{m_{mn}}\right)$ (A.123)	7,426 mm			E: 11
Tooth root chordal thickness on coast side	$s_{Fn1,C} = m_{mn} z_{vn1,C} \sin\left(\frac{\pi}{3} - \theta_{1,C}\right) + m_{mn}\sqrt{3}\left(\frac{G_{1,C}}{\cos\theta_{1,C}} - \frac{\rho_{a01,C}}{m_{mn}}\right)$ (A.124)	7,426 mm			E: 11
Tooth root chord	$s_{Fn1} = 0{,}5 s_{Fn1,D} + 0{,}5 s_{Fn1,C}$ (A.125)	7,426 mm			E: 12
Fillet radius at contact point of 30° tangent on the drive side	$\rho_{F1,D} = \rho_{a01,D} + \frac{2G_{1,D}^2 m_{mn}}{\cos\theta_{1,D}\left(z_{vn1,D}\cos^2\theta_{1,D} - 2G_{1,D}\right)}$ (A.126)	1,023 mm			E: 13

Table A.8 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Normal pressure angle at tooth tip on the drive side	$\alpha_{\mathrm{an1,D}} = \arccos\left(\dfrac{d_{\mathrm{vbn1,D}}}{d_{\mathrm{van1,D}}}\right)$ (A.127)	32,774°			E: 16
Auxiliary angle for tooth form and tooth correction factor on the drive side	$\gamma_{\mathrm{a1,D}} = \dfrac{1}{z_{\mathrm{vn1,D}}}\left[\dfrac{\pi}{2} + 2\left(x_{\mathrm{hm1}}\tan\alpha_{\mathrm{eD}} + x_{\mathrm{sm1}}\right)\right] + \mathrm{inv}\alpha_{\mathrm{eD}} - \mathrm{inv}\,\alpha_{\mathrm{an1,D}}$ (A.128)	1,245°			E: 17
Load application angle at tooth tip of virtual cylindrical gear on drive side	$\alpha_{\mathrm{Fan1,D}} = \alpha_{\mathrm{an1,D}} - \gamma_{\mathrm{a1,D}}$ (A.129)	31,530°			E: 15
Bending moment arm on the drive side	$h_{\mathrm{Fa1,D}} = \dfrac{m_{\mathrm{mn}}}{2}\left[\left(\cos\gamma_{\mathrm{a1,D}} - \sin\gamma_{\mathrm{a1,D}}\tan\alpha_{\mathrm{Fan1,D}}\right)\dfrac{d_{\mathrm{van1,D}}}{m_{\mathrm{mn}}} - z_{\mathrm{vn1,D}}\cos\left(\dfrac{\pi}{3} - \theta_{\mathrm{1,D}}\right) - \dfrac{G_{\mathrm{1,D}}}{\cos\theta_{\mathrm{1,D}}} + \dfrac{\rho_{\mathrm{a01,D}}}{m_{\mathrm{mn}}}\right]$ (A.130)	6,405 mm			E: 14
Tooth form factor on the drive side	$Y_{\mathrm{Fa1,D}} = \dfrac{6\dfrac{h_{\mathrm{Fa1,D}}}{m_{\mathrm{mn}}}\cos\alpha_{\mathrm{Fan1,D}}}{\left(\dfrac{s_{\mathrm{Fn1}}}{m_{\mathrm{mn}}}\right)^2\cos\alpha_{\mathrm{n1,D}}}$ (A.131)	2,031			E: 6

Table A.8 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Stress correction factor on the drive side	$L_{a1,D} = \frac{s_{Fn1}}{h_{Fa1,D}}$	(A.132)	1,159			E: 25
	$q_{s1,D} = \frac{s_{Fn1}}{2\rho_{F1,D}}$	(A.133)	3,630			E: 26
	$Y_{Sa1,D} = \left(1{,}2 + 0{,}13\ L_{a1,D}\right)\ q_{s1,D}^{\left(\frac{1}{1{,}21\ +\ 2{,}3/L_{a1,D}}\right)}$	(A.134)	2,023			E: 24
Relative surface condition factor for $R_z = 16$ µm and through hardened and case hardened steels	$Y_{R\,rel\,T} = \frac{Y_R}{Y_{RT}} = 1{,}674 - 0{,}529\ \left(Rz + 1\right)^{1/10}$	(A.135)	0,972			E: 39
Life factor for pinion	$Y_{NT,1}$	(A.136)	1,000			T: 2
Stress correction factor	Y_{ST}	(A.137)	2,000			E: 4
Size factor	$Y_X = 1{,}05 - 0{,}01 m_{mn}$ Set $Y_X = 1{,}0$ since range is $0{,}8 \leq Y_X \leq 1{,}0$.	(A.138)	1,000			F: 7 E: 188
Bevel spiral angle factor using l_{bm} from Formula (A.72) for l_{bb}	$Y_{BS} = \frac{a_{BS}}{c_{BS}}\left(\frac{l_{bb}}{b_a} - 1{,}05 \cdot b_{BS}\right)^2 + 1$	(A.139)	1,054			E: 28

Table A.8 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
with	$b_a = b_v / \cos\beta_v$ (A.140)	31,008 mm			E: 32
	$l_{bb} = l_{bm} \dfrac{\cos\beta_{vb}}{\cos\beta_v}$ (A.141)	23,073 mm			E: 33
	$h = (h_{m1} + h_{m2}) / 2$ (A.142)	7,23 mm			E: 34
Relative notch sensitivity factor	$Y_{\delta\ \mathrm{rel\ T\ 1}} = \dfrac{1+\sqrt{\rho' \chi_1^X}}{1+\sqrt{\rho' \chi_T^X}}$ (A.143)	1,010			E: 42
Contact ratio factor, Y_ε, for $\varepsilon_{v\beta} = 1{,}227 > 1$	Y_ε (A.144)	0,625			E: 27c
Tooth root stress formula for pinion					
Nominal tooth root stress	$\sigma_{F01} = \dfrac{F_{vmt}}{b_v\, m_{mn}} Y_{Fa} Y_{Sa} Y_\varepsilon Y_{BS} Y_{LS}$ (A.145)	316,3 N/mm^2			E: 2
Local tooth root stress	$\sigma_{F1} = \sigma_{F0} K_A K_v K_{F\beta} K_{F\alpha}$ (A.146)	679,9 N/mm^2			E: 1
Permissible tooth root stress	$\sigma_{FP1} = \sigma_{F\ \mathrm{lim}}\ Y_{ST}\ Y_{NT}\ Y_{\delta\ \mathrm{rel\ T\ 1}}\ Y_{\mathrm{R\ rel\ T}}\ Y_X$ (A.147)	942,0 N/mm^2			E: 4
Calculated safety for tooth root strength on pinion	$S_{F1} = \dfrac{\sigma_{FP1}}{\sigma_{F1}}$ (A.148)	1,386			E: 5

Table A.9 — Calculation of tooth root strength for wheel

Description	Formula	Result	References to ISO 10300-1 (E: Formula)	References to ISO 10300-2 (T: Table)	References to ISO 10300-3 (F: Figure)
Y-factors different from pinion					
Geometry values for wheel according to Tables A.2 and A.3.	$h_{a0} = h_{fp} = k_{hfp}\, m_{mn}$ (A.149)	4,017 mm			F: 2
Parameters for wheel	$E_{2,D} = \left(\frac{\pi}{4} - x_{sm2}\right) m_{mn} - h_{a0} \tan\alpha_{eD} - \frac{\rho_{a02,D}\left(1 - \sin\alpha_{eD}\right) - s_{pr2,D}}{\cos\alpha_{eD}}$ (A.150)	0,398 mm			E: 7
	$E_{2,C} = \left(\frac{\pi}{4} - x_{sm2}\right) m_{mn} - h_{a0} \tan\alpha_{eC} - \frac{\rho_{a02,C}\left(1 - \sin\alpha_{eC}\right) - s_{pr2,C}}{\cos\alpha_{eC}}$ (A.151)	0,398 mm			E: 7
	$G_{2,D} = \frac{\rho_{a02,D}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm2}$ (A.152)	−1,382			E: 8
	$G_{2,C} = \frac{\rho_{a02,C}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm2}$ (A.153)	−1,382			E: 8
Parameters for wheel	$H_{2,D} = \frac{2}{z_{vn2,D}}\left(\frac{\pi}{2} - \frac{E_{2,D}}{m_{mn}}\right) - \frac{\pi}{3}$ (A.154)	−1,033			E: 9
	$H_{2,C} = \frac{2}{z_{vn2,C}}\left(\frac{\pi}{2} - \frac{E_{2,C}}{m_{mn}}\right) - \frac{\pi}{3}$ (A.155)	−1,033			E: 9

Table A.9 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Iteration starting with $\theta = \pi/6$ until $(\theta_{new} - \theta) < 0{,}000\,001$	$\theta_{2,D,C} = \frac{2G_{2,D,C}}{z_{vn2,D,C}} \tan\theta_{2,D,C} - H_{2,D,C}$ (A.156)	Initial: 58,705°			E: 10
		Final: 57,895°			E: 10
Tooth root chordal thickness on the drive side	$s_{Fn2,D} = m_{mn} z_{vn2,D} \sin\left(\frac{\pi}{3} - \theta_{2,D}\right) + m_{mn}\sqrt{3}\left(\frac{G_{2,D}}{\cos\theta_{2,D}} - \frac{\rho_{a02,D}}{m_{mn}}\right)$ (A.157)	6,898 mm			E: 11
Tooth root chordal thickness on coast side	$s_{Fn2,C} = m_{mn} z_{vn2,C} \sin\left(\frac{\pi}{3} - \theta_{2,C}\right) + m_{mn}\sqrt{3}\left(\frac{G_{2,C}}{\cos\theta_{2,C}} - \frac{\rho_{a02,C}}{m_{mn}}\right)$ (A.158)	6,898 mm			E: 11
Tooth root chord	$s_{Fn2} = 0{,}5 s_{Fn2,D} + 0{,}5 s_{Fn2,C}$ (A.159)	6,898 mm			E: 12
Fillet radius at contact point of 30° tangent	$\rho_{F2,D} = \rho_{a02,D} + \frac{2G_{2,D}^2 m_{mn}}{\cos\theta_{2,D}\left(z_{vn2,D}\cos^2\theta_{2,D} - 2G_{2,D}\right)}$ (A.160)	1,592 mm			E: 13
Normal pressure angle at tooth tip on the drive side	$\alpha_{an2,D} = \arccos\left(\frac{d_{vbn2,D}}{d_{van2,D}}\right)$ (A.161)	20,767°			E: 16
Auxiliary angle for tooth form and tooth correction factor on the drive side	$\gamma_{a2,D} = \frac{1}{z_{vn2,D}}\left[\frac{\pi}{2} + 2\left(x_{hm2}\tan\alpha_{eD} + x_{sm2}\right)\right] + \operatorname{inv}\alpha_{eD} - \operatorname{inv}\alpha_{an2,D}$ (A.162)	0,209°			E: 17

Table A.9 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Load application angle at tooth tip of virtual cylindrical gear on the drive side	$\alpha_{\text{Fan2,D}} = \alpha_{\text{an2,D}} - \gamma_{\text{a2,D}}$	(A.163)	20,557°			E: 15
Bending moment arm on the drive side	$h_{\text{Fa2,D}} = \frac{m_{\text{mn}}}{2}\left[\left(\cos\gamma_{\text{a2,D}} - \sin\gamma_{\text{a2,D}}\tan\alpha_{\text{Fan2,D}}\right)\frac{d_{\text{van2,D}}}{m_{\text{mn}}} - z_{\text{vn2,D}}\cos\left(\frac{\pi}{3} - \theta_{\text{2,D}}\right) - \frac{G_{\text{2,D}}}{\cos\theta_{\text{2,D}}} + \frac{\rho_{\text{a02,D}}}{m_{\text{mn}}}\right]$	(A.164)	6,141 mm			E: 14
Tooth form factor on the drive side	$Y_{\text{Fa2,D}} = \dfrac{6\dfrac{h_{\text{Fa2,D}}}{m_{\text{mn}}}\cos\alpha_{\text{Fan2,D}}}{\left(\dfrac{s_{\text{Fn2}}}{m_{\text{mn}}}\right)^2\cos\alpha_{\text{n2,D}}}$	(A.165)	2,479			E: 6

Table A.9 — *continued*

Description	Formula		Result	References to ISO 10300-1 E: Formula	References to ISO 10300-2 T: Table	References to ISO 10300-3 F: Figure
Stress correction factor on the drive side	$L_{a2,D} = \frac{s_{Fn2}}{h_{Fa2,D}}$	(A.166)	1,123			E: 25
	$q_{s2,D} = \frac{s_{Fn2}}{2\rho_{F2,D}}$	(A.167)	2,166			E: 26
	$Y_{Sa2,D} = \left(1{,}2 + 0{,}13\ L_{a2,D}\right)\ q_{s2,D}^{\left(\frac{1}{1{,}21\ +\ 2{,}3/L_{a2,D}}\right)}$	(A.168)	1,706			E: 24
Relative surface condition factor for R_z = 16 µm for through hardened and case hardened steels	$Y_{R\,rel\,T} = \frac{Y_R}{Y_{RT}} = 1{,}674 - 0{,}529\ \left(Rz + 1\right)^{1/10}$	(A.169)	0,972			E: 39
Bevel spiral angle factor using l_{bm} from Formula (A.72) for l_{bb}	$Y_{BS} = \frac{a_{BS}}{c_{BS}}\left(\frac{l_{bb}}{b_a} - 1{,}05 \cdot b_{BS}\right)^2 + 1$	(A.170)	1,054			E: 28
where	$b_a = b_v / \cos \beta_v$	(A.171)	31,008 mm			E: 32
	$l_{bb} = l_{bm}\ \frac{\cos\beta_{vb}}{\cos\beta_v}$	(A.172)	23,073 mm			E: 33
	$h = (h_{m1} + h_{m2}) / 2$	(A.173)	7,23 mm			E: 34

Table A.9 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Relative notch sensitivity factor	$Y_{\delta\ \mathrm{rel\,T\,2}} = \frac{1+\sqrt{\rho' \chi_2^X}}{1+\sqrt{\rho' \chi_T^X}}$ (A.174)	0,997			E:
Tooth root stress formula for wheel					
Nominal tooth root stress	$\sigma_{F02} = \frac{F_{vmt}}{b_v\, m_{mn}} Y_{Fa} Y_{Sa} Y_\varepsilon Y_{BS} Y_{LS}$ (A.175)	325,8 N/mm^2			E: 2
Local tooth root stress	$\sigma_{F2} = \sigma_{F0} K_A K_v K_{F\beta} K_{F\alpha}$ (A.176)	700,3 N/mm^2			E: 1
Permissible tooth root stress	$\sigma_{FP2} = \sigma_{F\,lim} Y_{ST} Y_{NT} Y_{\delta\ \mathrm{rel\,T\,2}} Y_{\mathrm{R\,rel\,T}} Y_X$ (A.177)	929,8 N/mm^2			E: 4
Calculated safety for tooth root strength on wheel	$S_{F2} = \frac{\sigma_{FP2}}{\sigma_{F2}}$ (A.178)	1,328			E: 5

A.3 Calculation of Sample 1 according to Method B2

Table A.10 — Calculation of surface durability (pitting)

Description	Formula		Result	References to ISO 10300–1	References to ISO 10300–2	References to ISO 10300–3
				E: Formula	T: Table	F: Figure
Approximate values for application factors						
Relative face width	$b_v = b_2 / m_{et2}$	(A.179)	5,600	E: B.1		
Relative mean back cone distance on pinion	$R_{mpt1} = \frac{R_{m1} \tan\delta_1}{m_{et2}}$	(A.180)	6,432	E: B.2		
Relative mean back cone distance on wheel	$R_{mpt2} = \frac{R_{m2} \tan\delta_2}{m_{et2}}$	(A.181)	49,915	E: B.2		
Face contact ratio for bevel gears	$\varepsilon_{v\beta} = b_2 \sin\beta_{m2} / (\pi \ m_{mn})$	(A.182)	1,443	E: B.4a		
Relative mean virtual pitch radius on pinion	$r_{vn1} = \frac{R_{mpt1}}{\cos^2\beta_{m1}}$	(A.183)	9,586	E: B.5		
Relative mean virtual pitch radius on wheel	$r_{vn2} = \frac{R_{mpt2}}{\cos^2\beta_{m2}}$	(A.184)	74,388	E: B.5		
Relative centre distance	$a_{vn} = r_{vn1} + r_{vn2}$	(A.185)	83,974	E: B.6		

Table A.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Relative mean virtual dedendum on pinion	$h_{vfm1} = h_{fm1} / m_{et2}$	(A.186)	0,528	E: B.7		
Relative mean virtual dedendum on wheel	$h_{vfm2} = h_{fm2} / m_{et2}$	(A.187)	1,243	E: B.7		
Relative virtual tooth thickness on pinion	$s_{vmn1} = s_{mn1} \ / \ m_{et2}$	(A.188)	1,425	E: B.8		
Relative virtual tooth thickness on wheel	$s_{vmn2} = s_{mn2} \ / \ m_{et2}$	(A.189)	0,774	E: B.8		
Relative mean virtual tip radius on pinion	$r_{va1} = r_{vn1} + h_{am1} / m_{et2}$	(A.190)	10,652	E: B.9		
Relative mean virtual tip radius on wheel	$r_{va2} = r_{vn2} + h_{am2} / m_{et2}$	(A.191)	74,739	E: B.9		
Angular pitch of virtual cylindrical wheel (required in ISO 10300-3:2014, 7.4.5)	$\theta_{v2} = \dfrac{\pi \ m_{mn}}{m_{et2} r_{vn2}}$	(A.192)	1,714°	E: B.10		
Relative edge radius of tool on pinion	$\rho_{va01} = \rho_{a01} / m_{et2}$	(A.193)	0,176	E: B.11		
Relative edge radius of tool on wheel	$\rho_{va02} = \rho_{a02} / m_{et2}$	(A.194)	0,265	E: B.11		
Virtual spiral angle for bevel gears without hypoid offset	$\beta_v = \beta_{m2}$	(A.195)	35°	E: B.12a		

Table A.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Adjusted pressure angle	$\alpha_a = \alpha_{eD} - 90° \cos\delta_2 \cos\beta_{m2} / z_2$	(A.196)	19,156°	E: B.14		
Base virtual helix angle	$\sin\beta_{vb} = \sin\beta_v \cos\alpha_a$	(A.197)	sin(32,807°)	E: B.15		
Relative mean virtual base radius on pinion	$r_{vbn1} = r_{vn1} \cos\alpha_a$	(A.198)	9,055	E: B.16		
Relative mean virtual base radius on wheel	$r_{vbn2} = r_{vn2} \cos\alpha_a$	(A.199)	70,269	E: B.16		
Relative length of action from pinion tip to pitch circle in the normal section	$g_{v\alpha na} = \sqrt{r_{va1}^2 - r_{vbn1}^2} - r_{vn1} \sin\alpha_a$	(A.200)	2,464	E: B.17		
Relative length of action from wheel tip to pitch circle in the normal section	$g_{v\alpha nr} = \sqrt{r_{va2}^2 - r_{vbn2}^2} - r_{vn2} \sin\alpha_a$	(A.201)	1,049	E: B.18		
Relative length of action in normal section	$g_{v\alpha n} = g_{v\alpha na} + g_{v\alpha nr}$	(A.202)	3,513	E: B.19		
Relative mean normal pitch of virtual cylindrical gear	$p_{mn} = \dfrac{2{,}0\pi\, m_{mn}}{m_{et2} \cos\alpha_a \left(\cos^2\beta_{m1} + \cos^2\beta_{m2} + 2{,}0 \tan^2\alpha_a\right)}$	(A.203)	2,976	E: B.20		
Profile contact ratio in mean normal section	$\varepsilon_{v\alpha n} = \dfrac{g_{v\alpha n}}{p_{mn}}$	(A.204)	1,181	E: B.21		

Table A.10 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Profile contact ratio in mean transverse section	$\varepsilon_{v\alpha} = \varepsilon_{v\alpha n} \cos^2 \beta_{vb}$ (A.205)	0,834	E: B.22		
Modified contact ratio for bevel gears without hypoid offset	$\varepsilon_{v\gamma} = \sqrt{\varepsilon_{v\alpha}^2 + \varepsilon_{v\beta}^2}$ (A.206)	1,667	E: B.23		
Contact shift factor; see also ISO 10300–2:2014, Figure B.7.	$k' = \dfrac{z_2 - z_1}{3{,}2\, z_2 + 4{,}0\, z_1}$ (A.207)	0,138	E: B.24		
Pitting resistance geometry factor					
Angle between contact direction and tooth tangent in pitch plane	$\cot\left(\beta_{m1} - \lambda_1\right) = \dfrac{\cos \zeta_R}{\cos \beta_{m1}\, \cos \beta_{m2}\, \tan\left(\beta_{m1} - \lambda_r\right)} - \tan\beta_{m2}$ (A.208)	cot(35°)		E: 24	
Angle between projection of pinion axis and direction of contact in pitch plane	$\lambda_1 = \beta_{m1} - \left(\beta_{m1} - \lambda_1\right)$ (A.209)	0°		E: 25	
Angle of contact line relative to the root cone	$\tan w = \dfrac{\sin \alpha_a \tan\left(\beta_{m1} - \lambda_r\right)}{\cos \alpha_{lim}}$ (A.210)	tan(12,940°)		E: 26	
Mean base spiral angle	$\cos\beta_{bm} = \dfrac{1{,}0}{\sqrt{\tan^2\left(\beta_{m1} - \lambda_r\right)\cos^2\alpha_a + 1{,}0}}$ (A.211)	cos(33,482°)		E: 27	

Table A.10 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Relative mean normal base pitch	$p_{nb} = \frac{\pi\, m_{mn} \cos\alpha_a \cos\beta_{bm}}{m_{et2} \cos\left(\beta_{m1} - \lambda_r\right)}$ (A.212)	2,140		E: 28	
Angle between projection of wheel axis and direction of contact in pitch plane	$\lambda_2 = \left(\beta_{m1} - \lambda_r\right) - \beta_{m2}$ (A.213)	0°		E: 29	
Relative base face width	$b_b = \frac{b_2}{m_{et2} \cos\lambda_2}$ (A.214)	5,600		E: 30	
Pressure angle at point of load application on pinion	$\cos\alpha_{L1} = \cos\alpha_a \left[1{,}0 - \frac{\left(r_{va1} - r_{vn1}\right)\cos^2\beta_{m1}}{r_{va1} - r_{vn1} + R_{mpt1}}\right]$ (A.215)	cos(31,295°)		E: 31	
Pressure angle at point of load application on wheel	$\cos\alpha_{L2} = \cos\alpha_a \left[1{,}0 - \frac{\left(r_{va2} - r_{vn2}\right)\cos^2\beta_{m2}}{r_{va2} - r_{vn2} + R_{mpt2}}\right]$ (A.216)	cos(19,914°)		E: 31	
Radius of curvature difference between point of load application and mean point on pinion	$\rho_{\Delta 1} = \frac{r_{va1} - r_{vn1} + R_{mpt1}}{\cos^2\beta_{m1}} \cos\alpha_{L1} \left(\tan\alpha_{L1} - \tan\alpha_a\right)$ (A.217)	2,488		E: 32	

Table A.10 — *continued*

Description	Formula		Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Radius of curvature difference between point of load application and mean point on wheel	$\rho_{\Delta 2} = \frac{r_{va2} - r_{vn2} + R_{mpt2}}{\cos^2 \beta_{m2}} \cos\alpha_{L2} \left(\tan\alpha_{L2} - \tan\alpha_a\right)$	(A.218)	1,049		E: 32	
Radius of curvature change	$\rho_{\Delta red} = \cos\beta_{bm} \left(\rho_{\Delta 1} + \rho_{\Delta 2}\right)$	(A.219)	2,950		E: 33	
Relative length of action within the contact ellipse	$g_\eta = \sqrt{\rho_{\Delta red}^2 \cos^2\beta_{bm} + b_b^2 \sin^2\beta_{bm}}$	(A.220)	3,949		E: 34	
Radius of relative profile curvature and load sharing ratio at critical point						
Length of action at critical point in contact ellipse (iteration starting with $y_I = 0$)	$g_{\eta I} = \sqrt{g_\eta^2 - 4{,}0 y_I^2}$	(A.221)	Initial: 3,949		E: 36	
			Final: 3,933		E: 36	
Length of action considering adjacent teeth	$g_{\eta I\Sigma}^3 = g_{\eta I}^3 + \sqrt{\left[g_{\eta I}^2 - 4{,}0 p_{nb}\left(p_{nb} + 2{,}0 y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 4{,}0 p_{nb}\left(p_{nb} - 2{,}0 y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 8{,}0 p_{nb}\left(2{,}0 p_{nb} + 2{,}0 y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 8{,}0 p_{nb}\left(2{,}0 p_{nb} - 2{,}0 y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 16{,}0 p_{nb}\left(4{,}0 p_{nb} + 2{,}0 y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 16{,}0 p_{nb}\left(4{,}0 p_{nb} - 2{,}0 y_I\right)\right]^3}$	(A.222)	Initial: 61,601		E: 37	
			Final: 60,941		E: 37	

Table A.10 — *continued*

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
Load sharing ratio	$\varepsilon_{NI} = g_{\eta I}^3 / g_{\eta I \Sigma}^3$	(A.223)	Initial: 1,000		E: 38	
			Final: 0,998		E: 38	
Length of contact line	$g_c = g_{\eta I} \ \rho_{\Delta red} \ y_I / g_\eta^2$	(A.224)	Initial: 4,183		E: 39	
			Final: 4,165		E: 39	
Position change along path of contact	$g_{\eta\Delta} = \dfrac{\rho_{\Delta red}^2 \ y_I}{g_{\eta I}^2} + k' g_c \tan\beta_{bm} + \dfrac{0{,}5\rho_{\Delta red}}{\cos\beta_{bm}} - \rho_{\Delta 2}$	(A.225)	Initial: 1,102		E: 40	
			Final: 0,999		E: 40	
Intermediate value	$X = \dfrac{\sin^2 w \cos\alpha_{lim} \cos(\zeta_R - \lambda_1) \cos\lambda_1}{\sin^2(\beta_{m1} - \lambda_1) \sin\alpha_a \cos\zeta_R}$	(A.226)	Initial: 0,465		E: 41	
			Final: 0,465		E: 41	
Profile radius of curvature on pinion	$\rho_1 = R_{mpt1} X \pm g_{\eta\Delta}$	(A.227)	Initial: 4,090		E: 42	
			Final: 3,987		E: 42	
Profile radius of curvature on wheel	$\rho_2 = R_{mpt2} X \pm g_{\eta\Delta}$	(A.228)	Initial: 22,084		E: 42	
			Final: 22,187		E: 42	
Relative radius of profile curvature between pinion and wheel	$\rho_t = \dfrac{\rho_1 \rho_2}{\rho_1 + \rho_2}$	(A.229)	Initial: 3,451		E: 43	
			Final: 3,380		E: 43	
Inertia factor with $\varepsilon_{v\gamma} \leq 2{,}0$	Z_i	(A.230)	Initial: 1,200		E: 44a	
			Final: 1,200		E: 44a	

Table A.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Pitting resistance geometry factor	$Z_I = \frac{g_c \rho_t \cos\alpha_a m_{mn}}{b_b z_1 Z_i \varepsilon_{NI} m_{et2}}$	(A.231)	Initial: 10,000		E: 45	
			Final: 0,100		E: 45	
Face width factor						
for $12,7 \le b_2 \le 79,8$ mm	$Z_{FW} = 0,00492 b_2 + 0,4375$	(A.232)	0,562		E: 46b	
Contact stress adjustment factor						
for carburized case hardened steel	Z_A	(A.233)	0,967		E: 47	
Elasticity factor						
for a steel-on-steel gear pair	Z_E	(A.234)	189,800 $(N/mm^2)^{1/2}$		E: 51	
Life factor						
Life factor	Z_{NT}	(A.235)	1,000		T: 4	
K-factors						
Dynamic factor with $1 < \varepsilon_{v\gamma} \le 2,0$	$K_{v\text{-}B2}$	(A.236)	1,027	E: 14		
Face load factor	$K_{H,\beta}$	(A.237)	1,650	E: 27		
Transverse load factor	$K_{H,\alpha}$	(A.238)	1,000	E: 36		

Table A.10 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Contact stress formula for drive and coast side					
Nominal contact stress	$\sigma_{\text{H0-B2}} = \sqrt{\frac{F_{\text{mt1}} d_{\text{m1}} Z_{\text{FW}}}{b_2 Z_{\text{I}}} \left(\frac{z_2}{d_{\text{e2}} z_1}\right)^2} \cdot Z_{\text{E}}$ (A.239)	1 087,5 N/mm^2		E: 21	
Contact stress	$\sigma_{\text{H-B2}} = \sigma_{\text{H0-B2}} \sqrt{K_{\text{A}} K_{\text{v}} K_{\text{H}\beta}} \cdot Z_{\text{A}} \le \sigma_{\text{HP-B2}}$ (A.240)	1 435,8 N/mm^2		E: 20	
Permissible contact stress	$\sigma_{\text{HP-B2}} = \sigma_{\text{H lim}} Z_{\text{NT}} Z_{\text{L}} Z_{\text{v}} Z_{\text{R}} Z_{\text{W}}$ (A.241)	1 348,2 N/mm^2		E: 22	
Calculated safety factor for contact stress for pinion and wheel	$S_{\text{H-B2}} = \frac{\sigma_{\text{HP-B2}}}{\sigma_{\text{H-B2}}} > S_{\text{H min}}$ (A.242)	0,939		E: 23	

Table A.11 — Calculation of tooth root strength on pinion

Description	Formula	Result	References to ISO 10300‑1	References to ISO 10300‑2	References to ISO 10300‑3
			E: Formula	T: Table	F: Figure
Geometry factor for bevel gears					
Distance when $\varepsilon_{v\gamma} \leq 2{,}0$	y_J (A.243)	0,142			E: 53
Distance for spiral bevel pinions	$y_{31} = \frac{g_{v\alpha n}}{2} + \frac{g_{v\alpha n}^2 \cdot y_J \cdot \cos^2\beta_{vb} + b_v \cdot g_{v\alpha n} \cdot g_J \cdot k' \cdot \sin\beta_{vb}}{g_\eta^2}$ (A.244)	2,212			E: 57
where	$g_J = \sqrt{g_\eta^2 - 4y_J^2}$ (A.245)	3,910			E: 59
Transverse radius to point of load application					
for spiral bevel pinion	$x_{oo1} = \frac{b_v\, g_{v\alpha n} g_J\, k' \cos^2\beta_{vb} m_{et2} - b_v^2\, y_J \sin\beta_{vb} m_{et2}}{g_\eta^2}$ (A.246)	1,505 mm			E: 61
Normal pressure angle at point of load application for pinion	$\tan\alpha_{L1} = \frac{y_{31} + a_{vn} \cdot \sin\alpha_n - \sqrt{r_{va2}^2 - r_{vbn2}^2}}{r_{vbn1}}$ (A.247)	tan(26,459°)			E: 63
Rotation angle used in bending strength calculations for pinion	$\xi_{h1} = \left(\frac{s_{vmn1}}{2r_{vn1}} - \mathrm{inv}\alpha_{L1} + \mathrm{inv}\alpha_n\right)$ (A.248)	3,057°			E: 64

Table A.11 — *continued*

Description	Formula	Result	References to ISO 10300‑1 E: Formula	ISO 10300‑2 T: Table	ISO 10300‑3 F: Figure
Relative distance from pitch circle to the pinion point of load application and the wheel tooth centre line	$\Delta r_{y01} = \dfrac{r_{vbn1}}{\cos\alpha_{h1}} - r_{vn1}$ (A.249)	0,229			E: 65
where	$\alpha_{h1} = \alpha_{L1} - \xi_{h1}$ (A.250)	23,401°			E: 66
Mean transverse radius to point of load application, in mm	$r_{my01} = r_{mpt1}\left(\dfrac{R_m + x_{oo1}}{R_m}\right) + \Delta r_{y01} m_{et2}$ (A.251)	29,008 mm			E: 67
Load sharing ratio					
	$g'^{3}_{J} = g^{3}_{J} + \sum_{k=1}^{k=x}\sqrt{\left[g^{2}_{J} - 4k\dfrac{\pi\, m_{mn}\cos\alpha_a}{m_{et2}}\left(k\dfrac{\pi\, m_{mn}\cos\alpha_a}{m_{et2}} + 2y_J\right)\right]^{3}} + \sum_{k=1}^{k=y}\sqrt{\left[g^{2}_{J} - 4k\dfrac{\pi\, m_{mn}\cos\alpha_a}{m_{et2}}\left(k\dfrac{\pi\, m_{mn}\cos\alpha_a}{m_{et2}} - 2y_J\right)\right]^{3}}$ (A.252)	59,763			E: 68
Load sharing ratio	$\varepsilon_N = \dfrac{g^{3}_{J}}{g'^{3}_{J}}$ (A.253)	1,000			E: 69

Table A.11 — *continued*

Description	Formula		Result	References to ISO 10300‑1 (E: Formula)	References to ISO 10300‑2 (T: Table)	References to ISO 10300‑3 (F: Figure)
Tooth form factor						
	$g_{01} = 0{,}5 s_{\text{vmn1}} + h_{\text{vfm1}} \tan\alpha_{\text{n}} + \rho_{\text{va01}} \left(\frac{1 - \sin\alpha_{\text{n}}}{\cos\alpha_{\text{n}}} \right)$	(A.254)	1,028			E: 71
	$g_{\text{yb1}} = h_{\text{vfm1}} - \rho_{\text{va01}}$	(A.255)	0,351			E: 72
	$g_{\text{f01 (1)}} = g_{01} + g_{\text{yb1}}$	(A.256)	1,380			E: 73
Iteration starting with $g_{\text{f01(1)}}$ as initial value until $\frac{s_{\text{N1}} \cot\tau_1}{h_{\text{N1}}} = 2{,}0 \pm 0{,}001$	$\xi_1 = \frac{g_{\text{f0 1}}}{r_{\text{vn1}}}$	(A.257)	Initial: 8,247			E: 74
			Final: 8,758			E: 74
	$g_{\text{xb1}} = g_{\text{f01}} - g_{01}$	(A.258)	Initial: 0,351			E: 75
			Final: 0,437			E: 75
	$g_{\text{za1}} = g_{\text{yb1}} \cos\xi_1 - g_{\text{xb1}} \sin\xi_1$	(A.259)	Initial: 0,297			E: 76
			Final: 0,281			E: 76
	$g_{\text{zb1}} = g_{\text{yb1}} \sin\xi_1 + g_{\text{xb1}} \cos\xi_1$	(A.260)	Initial: 0,398			E: 77
			Final: 0,485			E: 77
	$\tan\tau_1 = \frac{g_{\text{za1}}}{g_{\text{zb1}}}$	(A.261)	Initial: tan(36,753°)			E: 78
			Final: tan(30,053°)			E: 78

Table A.11 — *continued*

Description	Formula		Result	References to ISO 10300–1 (E: Formula)	ISO 10300–2 (T: Table)	ISO 10300–3 (F: Figure)
Iteration starting with $g_{f01(1)}$ as initial value until $\frac{s_{N1}\cot\tau_1}{h_{N1}} = 2{,}0 \pm 0{,}001$	$s_{N1} = r_{vn1}\sin\xi_1 - \rho_{va01}\cos\tau_1 - g_{zb1}$	(A.262)	Initial: 0,835			E: 79
			Final: 0,822			E: 79
	$h_{N1} = \Delta r_{y01} + r_{vn1}\left(1-\cos\xi_1\right) + \rho_{va01}\sin\tau_1 + g_{za1}$	(A.263)	Initial: 0,731			E: 80
			Final: 0,710			E: 80
Tooth strength factor	$x_{N1} = \frac{s_{N1}^2}{h_{N1}}$	(A.264)	0,950			E: 82
Tooth form factor	$Y_1 = \frac{2}{3}\left[\frac{1}{\left(\frac{1}{x_{N1}} - \frac{\tan\alpha_{h1}}{3\,s_{N1}}\right)}\right]$	(A.265)	0,760			E: 83
Tooth fillet radius at root diameter						
Relative fillet radius at root of tooth	$r_{mf1} = \frac{\left(h_{vfm1} - \rho_{va01}\right)^2}{r_{vn1} + h_{vfm1} - \rho_{va01}} + \rho_{va01}$	(A.266)	0,189			E: 173
Stress concentration and correction factor						
Stress concentration and stress correction factor	$Y_{f1} = L + \left(\frac{2s_{N1}}{r_{mf1}}\right)^M \left(\frac{2s_{N1}}{h_{N1}}\right)^O$	(A.267)	2,198			E: 174

Table A.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300–1	ISO 10300–2	ISO 10300–3
				E: Formula	T: Table	F: Figure
where	$L = 0{,}325\,454\,5 - 0{,}007\,272\,7\,\alpha_n$	(A.268)	0,180			E: 175
	$M = 0{,}331\,818\,2 - 0{,}009\,090\,9\,\alpha_n$	(A.269)	0,150			E: 176
	$O = 0{,}268\,181\,8 + 0{,}009\,090\,9\,\alpha_n$	(A.270)	0,450			E: 177
Inertia factor						
Inertia factor when $\varepsilon_{v\gamma} < 2{,}0$	Y_i	(A.271)	1,200			E: 178a
Calculated effective face width						
Projected length of the instantaneous contact line in the tooth lengthwise direction	$g_{K1} = \dfrac{b_1\ g_{v\alpha n}\ g_{J1}\ \cos^2\beta_{vb}}{g_\eta^2}$	(A.272)	16,040			E: 179
For the toe increment	$\Delta b'_{i1} = \dfrac{b_1 - g_{K1}}{2\cos\beta_{m1}} - \dfrac{x_{oo1}}{\cos\beta_{m1}}$	(A.273)	3,876			E: 180
For the heel increment	$\Delta b'_{e1} = \dfrac{b_1 - g_{K1}}{2\cos\beta_{m1}} + \dfrac{x_{oo1}}{\cos\beta_{m1}}$	(A.274)	7,550			E: 181

Table A.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300–1	ISO 10300–2	ISO 10300–3
				E: Formula	T: Table	F: Figure
Calculated effective face width	$b_{ce1} = 25{,}4\, h_{N1} \cos\beta_{m1} \left[\arctan\left(\dfrac{\Delta b_{i1}}{25{,}4\, h_{Na1}}\right) + \arctan\left(\dfrac{\Delta b_{e1}}{25{,}4\, h_{Na1}}\right) \right] + g_{K1}$	(A.275)	25,025			E: 183
Bevel geometry factor						
Bevel geometry factor	$Y_{J1} = \dfrac{Y_1}{Y_{f1} \cdot \varepsilon_N \cdot Y_i} \cdot \dfrac{r_{my01}}{r_{mpt1}} \cdot \dfrac{b_{ce1}}{b_1} \cdot \dfrac{m_{mt1}}{m_{et2}}$	(A.276)	0,260			E: 51
Root stress adjustment factor for carburized case hardened steel	Y_A	(A.277)	1,075			E: 184
Relative surface condition factor						
Relative surface condition factor for pinion	$Y_{R\,rel\,T}$	(A.278)	1,000			E: 185
Relative notch sensitivity factor						
Relative notch sensitivity factor	$Y_{\delta\,rel\,T}$	(A.279)	1,000			E: 186a

Table A.11 — *continued*

Description	Formula		Result	References to ISO 10300‑1	References to ISO 10300‑2	References to ISO 10300‑3
				E: Formula	T: Table	F: Figure
Life factor						
Life factor	Y_{NT}	(A.280)	1,000			T: 2
Stress correction factor						
Stress correction factor	Y_{ST}	(A.281)	2,000			E: 4
K-factors						
Face load factor	$K_{F,\beta}$	(A.282)	1,650	E: 28		
Transverse load factor	$K_{F,\alpha}$	(A.283)	1,000	E: 36		
Tooth root stress formula						
Nominal tooth root stress	$\sigma_{F0\ 1\text{-}B2} = \frac{F_{mt1}}{b_1} \cdot \frac{m_{mt1}}{m_{et2}^2} \cdot \frac{Y_{A1}}{Y_{J1}}$	(A.284)	339,5 N/mm²			E: 47
Tooth root stress	$\sigma_{F1\text{-}B2} = \sigma_{F0\ 1\text{-}B2} K_A K_v K_{F\beta} K_{F\alpha}$	(A.285)	632,8 N/mm²			E: 44
Permissible tooth root stress						
Permissible tooth root stress	$\sigma_{FP1\text{-}B2} = \sigma_{F\,lim} Y_{ST} Y_{NT} Y_{\delta\,rel\,T\text{-}B2} Y_{R\,rel\,T\text{-}B2} Y_X$	(A.286)	960,0 N/mm²			E: 49
Calculated safety factor						
Calculated safety factor	$S_{F1\text{-}B2} = \frac{\sigma_{FP1\text{-}B2}}{\sigma_{F1\text{-}B2}}$	(A.287)	1,517			E: 50

Table A.12 — Calculation of tooth root strength on wheel

Description	Formula		Result	References to ISO 10300–1	References to ISO 10300–2	References to ISO 10300–3
				E: Formula	T: Table	F: Figure
Geometry factor for bevel gears						
Distance when $\varepsilon_{v\gamma} \le 2{,}0$	y_J	(A.288)	0,142			E: 53
Distance for spiral bevel wheels	$y_{32} = \frac{g_{v\alpha n}}{2} + \frac{g_{v\alpha n}^2 \cdot y_J \cdot \cos^2\beta_{vb} - b_v \cdot g_{v\alpha n} \cdot g_J \cdot k' \cdot \sin\beta_{vb}}{g_\eta^2}$	(A.289)	1,462			E: 58
where	$g_J = \sqrt{g_\eta^2 - 4 y_J^2}$	(A.290)	3,910			E: 59
Transverse radius to point of load application						
for spiral bevel wheel	$x_{oo2} = \frac{b_v \, g_{v\alpha n} \, g_J \, k' \cos^2\beta_{vb} m_{et2} + b_v^2 \, y_J \, \sin\beta_{vb} m_{et2}}{g_\eta^2}$	(A.291)	2,931 mm			E: 60
Normal pressure angle at point of load application for wheel	$\tan\alpha_{L2} = \frac{y_{32} + a_{vn} \cdot \sin\alpha_n - \sqrt{r_{va1}^2 - r_{vbn1}^2}}{r_{vbn2}}$	(A.292)	tan(19,313°)			E: 63
Rotation angle used in bending strength calculations for wheel	$\xi_{h2} = \left(\frac{s_{vmn2}}{2 r_{vn2}} - \mathrm{inv}\alpha_{L2} + \mathrm{inv}\alpha_n \right)$	(A.293)	0,386°			E: 64

Table A.12 — *continued*

Description	Formula	Result	References to ISO 10300–1	ISO 10300–2	ISO 10300–3
			E: Formula	T: Table	F: Figure
Relative distance from pitch circle to the pinion point of load application and the wheel tooth centre line	$\Delta r_{y0\,2} = \dfrac{r_{vbn2}}{\cos\alpha_{h2}} - r_{vn2}$ (A.294)	−0,490			E: 65
where	$\alpha_{h2} = \alpha_{L2} - \xi_{h2}$ (A.295)	18,928°			E: 66
Mean transverse radius to point of load application, in mm	$r_{my0\,2} = r_{mpt2}\left(\dfrac{R_m + x_{oo2}}{R_m}\right) + \Delta r_{y0\,2} m_{et2}$ (A.296)	77,028 mm			E: 67
Load sharing ratio					
Load sharing ratio	$g'^3_J = g^3_J + \sum_{k=1}^{k=x}\sqrt{\left[g^2_J - 4k\dfrac{\pi\, m_{mn}\cos\alpha_a}{m_{et2}}\left(k\dfrac{\pi\, m_{mn}\cos\alpha_a}{m_{et2}} + 2y_J\right)\right]^3} + \sum_{k=1}^{k=y}\sqrt{\left[g^2_J - 4k\dfrac{\pi\, m_{mn}\cos\alpha_a}{m_{et2}}\left(k\dfrac{\pi\, m_{mn}\cos\alpha_a}{m_{et2}} - 2y_J\right)\right]^3}$ (A.297)	59,763			E: 68
	$\varepsilon_N = \dfrac{g^3_J}{g'^3_J}$ (A.298)	1,000			E: 69

Table A.12 — *continued*

Description	Formula	Result	References to ISO 10300‑1 (E: Formula)	ISO 10300‑2 (T: Table)	ISO 10300‑3 (F: Figure)
Tooth form factor					
	$g_{02} = 0{,}5 s_{\mathrm{vmn2}} + h_{\mathrm{vfm2}} \tan\alpha_{\mathrm{n}} + \rho_{\mathrm{va02}} \left(\frac{1 - \sin\alpha_{\mathrm{n}}}{\cos\alpha_{\mathrm{n}}} \right)$ (A.299)	1,025			E: 71
	$g_{\mathrm{yb2}} = h_{\mathrm{vfm2}} - \rho_{\mathrm{va02}}$ (A.300)	0,979			E: 72
	$g_{\mathrm{f02\,(1)}} = g_{02} + g_{\mathrm{yb2}}$ (A.301)	2,003			E: 73
Iteration starting with $g_{\mathrm{f02\,(1)}}$ as initial value until $\frac{s_{\mathrm{N2}} \cot\tau_2}{h_{\mathrm{N2}}} = 2{,}0 \pm 0{,}001$	$\xi_2 = \frac{g_{\mathrm{f0\,2}}}{r_{\mathrm{vn2}}}$ (A.302)	Initial: 1,543			E: 74
		Final: 1,920			E: 74
	$g_{\mathrm{xb2}} = g_{\mathrm{f02}} - g_{02}$ (A.303)	Initial: 0,979			E: 75
		Final: 1,467			E: 75
	$g_{\mathrm{za2}} = g_{\mathrm{yb2}} \cos\xi_2 - g_{\mathrm{xb2}} \sin\xi_2$ (A.304)	Initial: 0,952			E: 76
		Final: 0,929			E: 76
	$g_{\mathrm{zb2}} = g_{\mathrm{yb2}} \sin\xi_2 + g_{\mathrm{xb2}} \cos\xi_2$ (A.305)	Initial: 1,005			E: 77
		Final: 1,499			E: 77
	$\tan\tau_2 = \frac{g_{\mathrm{za2}}}{g_{\mathrm{zb2}}}$ (A.306)	Initial: tan(43,457°)			E: 78
		Final: tan(31,782°)			E: 78

Table A.12 — *continued*

Description	Formula	Result	References to ISO 10300‒1 E: Formula	ISO 10300‒2 T: Table	ISO 10300‒3 F: Figure
Iteration starting with $g_{f02\,(1)}$ as initial value until $\frac{s_{N2}\cot\tau_2}{h_{N2}} = 2{,}0 \pm 0{,}001$	$s_{N2} = r_{vn2}\sin\xi_2 - \rho_{va02}\cos\tau_2 - g_{zb2}$ (A.307)	Initial: 0,807			E: 79
		Final: 0,767			E: 79
	$h_{N2} = \Delta r_{y02} + r_{vn2}\left(1-\cos\xi_2\right) + \rho_{va02}\sin\tau_2 + g_{za2}$ (A.308)	Initial: 0,670			E: 80
		Final: 0,620			E: 80
Tooth strength factor	$x_{N2} = \frac{s_{N2}^2}{h_{N2}}$ (A.309)	0,951			E: 82
Tooth form factor	$Y_2 = \frac{2}{3}\left[\frac{1}{\left(\frac{1}{x_{N2}} - \frac{\tan\alpha_{h2}}{3\,s_{N2}}\right)}\right]$ (A.310)	0,738			E: 83
Tooth fillet radius at root diameter					
Relative fillet radius at root of tooth	$r_{mf2} = \frac{\left(h_{vfm2} - \rho_{va02}\right)^2}{r_{vn2} + h_{vfm2} - \rho_{va02}} + \rho_{va02}$ (A.311)	0,277			E: 173

Table A.12 — *continued*

Description	Formula		Result	References to ISO 10300‑1	References to ISO 10300‑2	References to ISO 10300‑3
				E: Formula	T: Table	F: Figure
Stress concentration and correction factor						
Stress concentration and stress correction factor	$Y_{f2} = L + \left(\frac{2s_{N2}}{r_{mf2}}\right)^M \left(\frac{2s_{N2}}{h_{N2}}\right)^O$	(A.312)	2,124			E: 174
where	$L = 0{,}325\ 454\ 5 - 0{,}007\ 272\ 7\ \alpha_n$	(A.313)	0,180			E: 175
	$M = 0{,}331\ 818\ 2 - 0{,}009\ 090\ 9\ \alpha_n$	(A.314)	0,150			E: 176
	$O = 0{,}268\ 181\ 8 + 0{,}009\ 090\ 9\ \alpha_n$	(A.315)	0,450			E: 177
Inertia factor						
Inertia factor when $\varepsilon_{v\gamma} < 2{,}0$	Y_i	(A.316)	1,200			E: 178a
Calculated effective face width						
Projected length of the instantaneous contact line in the tooth lengthwise direction	$g_{K2} = \frac{b_2\ g_{v\alpha n}\ g_{J2} \cos^2 \beta_{vb}}{g_\eta^2}$	(A.317)	16,040			E: 179
For the toe increment	$\Delta b'_{i2} = \frac{b_2 - g_{K2}}{2\cos\beta_{m2}} - \frac{x_{oo2}}{\cos\beta_{m2}}$	(A.318)	2,135			E: 180
For the heel increment	$\Delta b'_{e2} = \frac{b_2 - g_{K2}}{2\cos\beta_{m2}} + \frac{x_{oo2}}{\cos\beta_{m2}}$	(A.319)	9,291			E: 181

Table A.12 — *continued*

Description	Formula		Result	References to		
				ISO 10300–1	ISO 10300–2	ISO 10300–3
				E: Formula	T: Table	F: Figure
Calculated effective face width	$b_{\text{ce2}} = 25{,}4\ h_{\text{N2}} \cos\beta_{\text{m2}} \left[\begin{array}{l} \arctan\left(\dfrac{\Delta b_{\text{i2}}}{25{,}4\ h_{\text{Na2}}}\right) \\ +\arctan\left(\dfrac{\Delta b_{\text{e2}}}{25{,}4\ h_{\text{Na2}}}\right) \end{array} \right] + g_{\text{K2}}$	(A.320)	24,653			E: 183
Bevel geometry factor						
Bevel geometry factor	$Y_{\text{J2}} = \dfrac{Y_2}{Y_{\text{f2}} \cdot \varepsilon_{\text{N}} \cdot Y_{\text{i}}} \cdot \dfrac{r_{\text{my02}}}{r_{\text{mpt2}}} \cdot \dfrac{b_{\text{ce2}}}{b_2} \cdot \dfrac{m_{\text{mt2}}}{m_{\text{et2}}}$	(A.321)	0,245			E: 51
Root stress adjustment factor for carburized case hardened steel	Y_{A}	(A.322)	1,075			E: 184
Relative surface condition factor						
Relative surface condition factor for pinion	$Y_{\text{R rel T}}$	(A.323)	1,000			E: 185
Relative notch sensitivity factor						
Relative notch sensitivity factor	$Y_{\delta\ \text{rel T}}$	(A.324)	1,000			E: 186a
Life factor						
Life factor	Y_{NT}	(A.325)	1,000			T: 2

Table A.12 — *continued*

Description	Formula		Result	References to		
				ISO 10300–1	ISO 10300–2	ISO 10300–3
				E: Formula	T: Table	F: Figure
Stress correction factor						
Stress correction factor	Y_{ST}	(A.326)	2,000			E: 4
Tooth root stress formula						
Nominal tooth root stress	$\sigma_{F0\ 2\text{-}B2} = \frac{F_{mt2}}{b_2} \cdot \frac{m_{mt2}}{m_{et2}^2} \cdot \frac{Y_{A2}}{Y_{J2}}$	(A.327)	359,9 N/mm^2			E: 45
Tooth root stress	$\sigma_{F2\text{-}B2} = \sigma_{F0\ 2\text{-}B2} K_A K_v K_{F\beta} K_{F\alpha}$	(A.328)	670,9 N/mm^2			E: 44
Permissible tooth root stress						
Permissible tooth root stress	$\sigma_{FP2\text{-}B2} = \sigma_{F\ lim} Y_{ST} Y_{NT} Y_{\delta\ rel\ T\text{-}B2} Y_{R\ rel\ T\text{-}B2} Y_X$	(A.329)	960,0 N/mm^2			E: 49
Calculated safety factor						
Calculated safety factor	$S_{F2\text{-}B2} = \frac{\sigma_{FP2\text{-}B2}}{\sigma_{F2\text{-}B2}}$	(A.330)	1,431			E: 50

Annex B
(informative)

Sample 2: Rating of a hypoid gear set according to Method B1 and Method B2

B.1 Initial data

Sample 2 is for a hypoid gear pair which uses Method 1 according to ISO 23509.

Table B.1 — Initial data for pitch cone parameters

Symbol	Description	Method 0	Method 1	Method 2	Method 3
Σ	shaft angle	X	90°	X	X
a	hypoid offset	0,0	15 mm	X	X
$z_{1,2}$	number of teeth	X	13/42	X	X
d_{m2}	mean pitch diameter of wheel	—	—	X	—
d_{e2}	outer pitch diameter of wheel	X	170 mm	—	X
b_2	wheel face width	X	30 mm	X	X
β_{m1}	mean spiral angle of pinion	—	50°	—	—
β_{m2}	mean spiral angle of wheel	X	—	X	X
r_{c0}	cutter radius	X	63,5 mm	X	X
z_0	number of blade groups (only face hobbing)	X	—	X	X

Table B.2 — Input data for tooth profile parameters

Data type I		Data type II	
Symbol	**Description**	**Symbol**	**Description**
α_{dD}	20°		
α_{dC}	20°		
$f_{\alpha lim}$	1		
x_{hm1}	—	c_{ham}	0,35
k_{hap}	—	k_d	2,000
k_{hfp}	—	k_c	0,125
x_{smn}	—	k_t W_{m2}	0,1 —
j_{et2}	0,2 mm		
θ_{a2}	1°		
θ_{f2}	4°		
ρ_{a01}	0,8 mm		
ρ_{a02}	1,2 mm		
$s_{pr1D,C}$	0 mm/0 mm		
$s_{pr2D,C}$	0 mm/0 mm		

Table B.3 and Table B.4 show geometric and operational data and text for explanation.

Table B.3 — Geometric data from calculation according to ISO 23509

Symbol	Description	Values	Symbol	Description	Value
$d_{m1,2}$	mean pitch diameter of pinion/wheel	53,383 mm/ 141,877 mm	ζ_{mp}	offset angle in pitch plane	11,390°
$h_{am1,2}$	mean addendum of pinion/wheel	3,432 mm/ 1,848 mm	ζ_R	pinion offset angle in root plane	10,319°
$h_{fm1,2}$	mean dedendum of pinion/wheel	2,508 mm/ 4,092 mm	$R_{e1,2}$	outer cone distance on pinion/wheel	89,608 mm/ 91,468 mm
$\alpha_{eD,C}$	effective pressure angle for drive side/coast side	20°/20°	$R_{m1,2}$	mean cone distance on pinion/wheel	73,519 mm/ 76,337 mm
$\alpha_{nD,C}$	generated pressure angle for drive side/coast side	17,747°/ 22,253°	$\delta_{1,2}$	pitch angle on pinion/wheel	21,288°/ 68,323°
α_{lim}	limit pressure angle	−2,253°	$\delta_{a1,2}$	face angle on pinion/wheel	25,232°/ 69,323°
m_{mn}	mean normal module	2,640 mm	$\delta_{f1,2}$	root angle on pinion/wheel	20,303°/ 64,324°
k_{hfp}	basic crown gear dedendum factor	1,25	$x_{sm1,2}$	thickness modification coefficient on pinion/wheel	0,038/ −0,062
ζ_m	pinion offset angle in axial plane	10,603°	m_{et2}	outer transverse module	4,047 mm
$s_{mn1,2}$	mean normal circular tooth thickness of pinion/wheel	4,922 mm/ 3,241 mm			

Table B.4 — Operation parameters and additional considerations

Symbol	Description	Value
Additional data		
	wheel profile	generated
	roughing/finishing method	face milling
$b_{2\text{eff}}$	effective face width on wheel	$0{,}85 \cdot b_2$
	profile crowning	low
	verification of contact pattern	checked under light test load for each gear
	mounting conditions of pinion and wheel	one member cantilever−mounted
Operation parameters		
T_1	pinion torque	250 Nm
n_1	pinion rotational speed	1 200 min^{-1}
K_A	application factor	1,1
	active flank	drive
Material data for pinion and wheel (case hardened steel)		
$\sigma_{\text{H lim}}$	allowable stress number (contact)	1 500 N/mm^2
$\sigma_{\text{F lim}}$	nominal stress number (bending)	480 N/mm^2
	surface hardness	same for pinion and wheel
Quality parameters		
R_z	flank roughness on pinion/wheel	8 µm/8 µm
R_z	tooth root roughness on pinion/wheel	16 µm/16 µm
f_{pt}	single pitch deviation on pinion/wheel	11,6 µm/23,5 µm
Lubrication parameters		
	oil type	ISO-VG-150
	oil temperature	90 °C

B.2 Calculation of Sample 2 according to Method B1

Table B.5 — Virtual cylindrical gears

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Virtual cylindrical gears in transverse section						
Reference diameter on pinion	$d_{v1} = \frac{d_{m1}}{\cos\delta_1}$	(B.1)	57,292 mm	E: A.1		
Reference diameter on wheel	$d_{v2} = \frac{d_{m2}}{\cos\delta_2}$	(B.2)	384,101 mm	E: A.1		
Number of teeth on pinion	$z_{v1} = d_{v1} / m_{vt}$	(B.3)	15,531	E: A.12		
Number of teeth on wheel	$z_{v2} = d_{v2} / m_{vt}$	(B.4)	104,122	E: A.12		
Gear ratio	$u_v = z_{v2} / z_{v1}$	(B.5)	6,704	E: A.13		
Helix angle	$\beta_v = \frac{\beta_{m1} + \beta_{m2}}{2}$	(B.6)	44,304°	E: A.8		
Transverse module	$m_{vt} = m_{mn} / \cos\beta_v$	(B.8)	3,689 mm	E: A.11		

Table B.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Transverse pressure angle of virtual cylindrical gears	$\alpha_{vet} = \arctan\left(\tan\alpha_e / \cos\beta_v\right)$ since $\alpha_e = \alpha_{eD}$ for drive side	(B.7)	26,957°	E: A.10		
Auxiliary angle for virtual face width	$\theta_{mp} = \arctan\left(\sin\delta_2 \tan\zeta_m\right)$	(B.9)	9,868°	E: A.21		
Projected auxiliary angle for length of contact line	$\gamma' = \theta_{mp} - \zeta_{mp}/2$	(B.10)	4,173°	E: A.20		
Centre distance of virtual cylindrical gear pair	$a_v = \left(d_{v1} + d_{v2}\right) / 2$	(B.11	220,697 mm	E: A.5		
Helix angle of virtual cylindrical gear at base circle	$\beta_{vb} = \arcsin\left(\sin\beta_v \cos\alpha_e\right)$ since $\alpha_e = \alpha_{eD}$ for drive side	(B.12)	41,021°	E: A.16		
Tip diameter on pinion	$d_{va1} = d_{v1} + 2\ h_{am1}$	(B.13)	64,156 mm	E: A.6		
Tip diameter on wheel	$d_{va2} = d_{v2} + 2\ h_{am2}$	(B.14)	387,797 mm	E: A.6		
Root diameter on pinion	$d_{vf1} = d_{v1} - 2\ h_{fm1}$	(B.15)	52,276 mm	E: A.7		
Root diameter on wheel	$d_{vf2} = d_{v2} - 2\ h_{fm2}$	(B.16)	375,917 mm	E: A.7		
Base diameter on pinion	$d_{vb1} = d_{v1} \cos\alpha_{vet}$	(B.17)	51,067 mm	E: A.9		

Table B.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Base diameter on wheel	$d_{vb2} = d_{v2} \cos\alpha_{vet}$	(B.18)	342,367 mm	E: A.9		
Transverse base pitch	$p_{vet} = \pi\, m_{mn} \cos\alpha_{vet} / \cos\beta_v$	(B.19)	10,330 mm	E: A.17		
Length of path of contact of virtual cylindrical gear in transverse section	$g_{v\alpha} = \frac{1}{2}\left[\left(\sqrt{d_{va1}^2 - d_{vb1}^2} - d_{v1}\sin\alpha_{vet}\right) + \left(\sqrt{d_{va2}^2 - d_{vb2}^2} - d_{v2}\sin\alpha_{vet}\right)\right]$	(B.20)	10,436 mm	E: A.18		
Transverse contact ratio	$\varepsilon_{v\alpha} = g_{v\alpha} / p_{vet}$	(B.21)	1,010	E: A.23		
Effective face width with $b_{2\,eff} = 0{,}85 \cdot b_2$	$b_{v\,eff} = \frac{\left[b_{2\,eff} / \cos\left(\zeta_{mp}/2\right) - g_{v\alpha} \cos\alpha_{vet} \tan\left(\zeta_{mp}/2\right)\right]}{1 - \tan\gamma' \tan\left(\zeta_{mp}/2\right)}$	(B.22)	24,880 mm	E: A.19		
Face width	$b_v = b_2 \frac{b_{v\,eff}}{b_{2\,eff}}$	(B.23)	29,270 mm	E: A.22		
Virtual cylindrical gears in normal section						
Number of pinion teeth of virtual cylindrical gears	$z_{vn1} = \frac{z_{v1}}{\cos^2\beta_{vb} \cos\beta_v}$	(B.24)	38,125	E: A.38		

Table B.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Number of wheel teeth of virtual cylindrical gears	$z_{vn2} = u_v z_{vn1}$	(B.25)	255,600	E: A.39		
Reference diameter on pinion	$d_{vn1} = z_{vn1} m_{mn}$	(B.26)	100,650 mm	E: A.40		
Reference diameter on wheel	$d_{vn2} = z_{vn2} m_{mn}$	(B.27)	674,783 mm	E: A.40		
Tip diameter on pinion	$d_{van1} = d_{vn1} + 2h_{am1}$	(B.28)	107,514 mm	E: A.41		
Tip diameter on wheel	$d_{van2} = d_{vn2} + 2h_{am2}$	(B.29)	678,479 mm	E: A.41		
Root diameter on pinion	$d_{vfn1} = d_{vn1} - 2\,h_{fm1}$	(B.30)	95,634 mm	E: A.7		
Root diameter on wheel	$d_{vfn2} = d_{vn2} - 2\,h_{fm2}$	(B.31)	666,599 mm	E: A.7		
Base diameter on pinion	$d_{vbn1} = d_{vn1} \cos\alpha_e$	(B.32)	94,580 mm	E: A.42		
Base diameter on wheel	$d_{vbn2} = d_{vn2} \cos\alpha_e$	(B.33)	634,089 mm	E: A.42		
Face contact ratio	$\varepsilon_{v\beta} = \dfrac{b_{v\,eff} \sin\beta_v}{\pi\, m_{mn}}$	(B.34)	2,095	E: A.24		
Virtual contact ratio	$\varepsilon_{v\gamma} = \varepsilon_{v\alpha} + \varepsilon_{v\beta}$	(B.35)	3,105	E: A.25		

Table B.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Inclination angle of contact line	$\beta_B = \arctan\left(\tan\beta_v \sin\alpha_e\right)$	(B.36)	18,459°	E: A.36		
Radius of relative curvature in normal section at the mean point	$\rho_t = \left[\frac{1}{\cos\alpha_{nD}\left(\tan\alpha_{nD} - \tan\alpha_{lim}\right) + \tan\zeta_{mp}\tan\beta_B} \cdot \frac{\cos\beta_{m1}\cos\beta_{m2}}{\cos\zeta_{mp}} \cdot \left(\frac{1}{R_{m2}\tan\delta_2} + \frac{1}{R_{m1}\tan\delta_1}\right)\right]^{-1}$	(B.37)	19,923 mm	E: A.37a		
Radius of relative curvature vertical to the contact line	$\rho_{rel} = \left\vert\rho_t\right\vert \cos^2\beta_B$	(B.38)	17,926 mm	E: A.35		

Table B.6 — General influence factors

Description	Formula		Result	References to		
				ISO 10300–1	ISO 10300–2	ISO 10300–3
				E: Formula	T: Table	F: Figure
Nominal tangential force of bevel gears	$F_{mt1} = \frac{2\,000\ T_1}{d_{m1}}$	(B.39)	9 366 N	E: 1		
Nominal tangential force of virtual cylindrical gears	$F_{vmt} = F_{mt1}\frac{\cos\beta_v}{\cos\beta_{m1}}$	(B.40)	10 428 N	E: 2		

Table B.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300–1	ISO 10300–2	ISO 10300–3
				E: Formula	T: Table	F: Figure
Nominal tangential speed at mean point of the pinion	$v_{\mathrm{mt}\,1} = \frac{d_{\mathrm{m}\,1}\ n_1}{19\,098}$	(B.41)	3,354 m/s	E: 5		
Nominal tangential speed at mean point of the wheel	$v_{\mathrm{mt}\,2} = \frac{d_{\mathrm{m}\,2}\ n_2}{19\,098}$	(B.42)	2,759 m/s	E: 5		
Correction factor for non-average conditions for $F_{\mathrm{vmt}}\,K_{\mathrm{A}}\,/\,b_{\mathrm{veff}} \geq$ 100 N/mm	C_{F}	(B.43)	1,0	E: 12a		
Mean value of mesh stiffness per unit face width	$c_{\gamma} = c_{\gamma 0} C_{\mathrm{F}}$	(B.44)	20 N/(mm·µm)	E: 11		
Single stiffness	$c' = c'_0\ C_{\mathrm{F}}$	(B.45)	14 N/(mm·µm)	E: 17		
Single pitch deviation	f_{pt}	(B.46)	23,5 µm			
Running-in allowance for case hardened and nitrided gears	$y_{\alpha} = 0{,}075\ f_{\mathrm{pt}}$	(B.47)	1,763 µm	E: 43		
Effective pitch deviation with yp = yα	fp eff = fpt – yp	(B.48)	21,737 µm	E: 16		

Table B.6 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Relative pinion mass per unit face width reduced to the line of action	$m_1^* = \frac{1}{8} \rho \pi \frac{d_{m1}^2}{\cos^2 \left[(\alpha_{nD} + \alpha_{nC})/2 \right]}$ (B.49) where ρ is the density of the gear material (for steel $\rho = 7{,}86 \cdot 10^{-6}$ kg/mm^3)	0,010 kg/mm	E: 13		
Relative wheel mass per unit face width reduced to the line of action	$m_2^* = \frac{1}{8} \rho \pi \frac{d_{m2}^2}{\cos^2 \left[(\alpha_{nD} + \alpha_{nC})/2 \right]}$ (B.50) where ρ is the density of the gear material (for steel $\rho = 7{,}86 \cdot 10^{-6}$ kg/mm^3)	0,070 kg/mm	E: 13		
Mass reduced to the line of action of the dynamically equivalent cylindrical gear pair	$m_{red} = \frac{m_1^* m_2^*}{m_1^* + m_2^*}$ (B.51)	0,009 kg/mm	E: 10		
Resonance speed of pinion	$n_{E1} = \frac{30 \times 10^3}{\pi z_1} \sqrt{\frac{c_\gamma}{m_{red}}}$ (B.52)	35167,105 min^{-1}	E: 9		
Dimensionless reference speed	$N = \frac{n_1}{n_{E1}}$ (B.53)	0,034	E: 8		

Table B.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
For virtual contact ratio, $\varepsilon_{v\gamma} = 3{,}105 > 2$	$c_{v1,2} = c_{v1} + c_{v2}$	(B.54)	0,523	T: 3		
	c_{v3}	(B.55)	0,062	T: 3		
	c_{v4}	(B.56)	0,249	T: 3		
	$c_{v5,6}$	(B.57)	0,558	T: 3		
	c_{v7}	(B.58)	0,834	T: 3		
Constant factor for calculating the dynamic factor with $K_A = 1{,}1$ as given in Table B.4	$K = \dfrac{b_v\, f_{p\,eff}\, c'}{F_{vmt} K_A} c_{v1,2} + c_{v3}$	(B.59)	0,468	E: 15		
Dynamic factor	$K_v = N \cdot K + 1$	(B.60)	1,0	E: 14		
Determination of the length of contact lines						
For virtual contact ratio, $\varepsilon_{v\beta} = 2{,}095$, $\varepsilon_{v\beta} \geq 1$	$f_t = +p_{vet} \cos\beta_{vb}$	(B.61)	7,794 mm	T: A.2		
	f_m	(B.62)	0,0 mm	T: A.2		
	$f_r = -p_{vet} \cos\beta_{vb}$	(B.63)	−7,794 mm	T: A.2		
Maximum distances from middle contact line	$f_{maxB} = \dfrac{1}{2}\left[g_{v\alpha} + b_{v\,eff}\left(\tan\gamma + \tan\beta_{vb}\right)\right]\cos\beta_{vb}$	(B.64)	12,870 mm	E: A.31		

Table B.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Maximum distances from middle contact line	$f_{\max 0} = \frac{1}{2}\left[g_{v\alpha} - b_{v\,\mathrm{eff}}\left(\tan\gamma + \tan\beta_{vb}\right)\right]\cos\beta_{vb}$	(B.65)	−4,996 mm	E: A.32		
	$f_{\max} = f_{\max B}$ since $f_{\max B} > f_{\max 0}$	(B.66)	12,870 mm			
Theoretical length of contact line	$l_{b0} = \sqrt{\left(x_1 - x_2\right)^2 + \left(y_1 - y_2\right)^2}$	(B.67)	14,532 mm	E: A.27		

Table B.6 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Theoretical length of middle contact line calculated with $f = f_m$ for contact stress as specified in ISO 10300–2:2014, 6.1	$x_1 = \dfrac{f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \dfrac{b_{v\,eff}}{2}\right) + \dfrac{1}{2}\left(g_{v\alpha} + b_{v\,eff}\tan\gamma\right)}{\tan\gamma + \tan\beta_{vb}}$ (B.68)	17,922 mm	E: A.28		
	$x_2 = \dfrac{f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \dfrac{b_{v\,eff}}{2}\right) - \dfrac{1}{2}\left(g_{v\alpha} - b_{v\,eff}\tan\gamma\right)}{\tan\gamma + \tan\beta_{vb}}$ (B.69) NOTE ISO 10300–1:2014, Formula (A.29) is a misprint. The operator in the second parenthesis should be "–".	6,958 mm	E: A.29		
	$y_1 = -x_1\tan\beta_{vb} + f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \dfrac{b_{v\,eff}}{2}\right)$ (B.70)	−4,769 mm	E: A.30		
	$y_2 = -x_2\tan\beta_{vb} + f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \dfrac{b_{v\,eff}}{2}\right)$ (B.71)	4,769 mm	E: A.30		
Correction factor	$C_{lb} = \sqrt{\left(1 - \left(\dfrac{f}{f_{max}}\right)^2\right)\left(1 - \sqrt{\dfrac{b_{veff}}{b_v}}\right)^2}$ (B.72)	0,078	E: A.34		
Length of contact line	$l_{bm} = l_{b0}\left(1 - C_{lb}\right)$ (B.73)	13,398 mm	E: A.26		

Table B.6 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Load sharing factor					
Exponent for calculation of parabolic distribution of peak loads	e (B.74)	3,0		T: 3	
Related peak load	$p^* = 1 - \left(\frac{\lvert f \rvert}{\lvert f_{max} \rvert}\right)^e$ (B.75)			E: 7 F: 2	
Related peak load at tip contact line	$p_t{}^* = 1 - \left(\frac{\lvert f_t \rvert}{\lvert f_{max} \rvert}\right)^e$ (B.76)	0,778		E: 7	
Related peak load at middle contact line	$p_m{}^* = 1 - \left(\frac{\lvert f_m \rvert}{\lvert f_{max} \rvert}\right)^e$ (B.77)	1,0		E: 7	
Related peak load at root contact line	$p_r{}^* = 1 - \left(\frac{\lvert f_r \rvert}{\lvert f_{max} \rvert}\right)^e$ (B.78)	0,778		E: 7	
Related area	$A^* = \frac{1}{4} p^* l_b \pi$ (B.79)			E: 8 F: 2	

Table B.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Related area at tip contact line	$A_t{}^* = \frac{1}{4} p_t{}^* l_{bt} \pi$	(B.80)	5,369 mm		E: 8	
Related area at middle contact line	$A_m{}^* = \frac{1}{4} p_m{}^* l_{bm} \pi$	(B.81)	10,523 mm		E: 8	
Related area at root contact line	$A_r{}^* = \frac{1}{4} p_r{}^* l_{br} \pi$	(B.82)	5,369 mm		E: 8	
Load sharing factor	$Z_{LS} = \sqrt{\frac{A^*_m}{A^*_t + A^*_m + A^*_r}}$	(B.83)	0,704		E: 10	
Face load factors (Calculation according to Method C)						
Load distribution factor	$K_{H\beta\text{-}C} = 1{,}5\, K_{H\beta\text{-be}}$	(B.84)	1,65	E: 27		
with	$K_{H\beta\text{-be}}$	(B.85)	1,100	T: 4		
Load distribution factor	$K_{F\beta\text{-}C} = K_{H\beta\text{-}C} / K_{F0}$	(B.86)	1,556	E: 28		
with	K_{F0}	(B.87)	1,060	E: 29b		
Transverse load factors (Calculation according to Method B)						
Determinant tangential force at mid face width on the pitch cone	$F_{mtH} = F_{vmt} K_A K_v K_{H\beta}$	(B.88)	18927 N	E: 37		

Table B.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Transverse load factors for bevel gear with virtual contact ratio $\varepsilon_{v\gamma} = 3{,}105 > 2$	$K^*_{H\alpha} = K^*_{F\alpha} = 0{,}9 + 0{,}4\sqrt{\frac{2\left(\varepsilon_{v\gamma} - 1\right)}{\varepsilon_{v\gamma}}} \cdot \frac{c_\gamma\left(f_{pt} - y_\alpha\right)}{F_{mtH}/b_v}$	(B.89)	1,213	E: 38		
Relative hypoid offset	$a_{rel} = \frac{2\left\vert a\right\vert}{d_{m2}}$	(B.90)	0,211	E: 35		
Transverse load factors	$K_{H\alpha} = K_{F\alpha} = K^*_{H\alpha} - \frac{K^*_{H\alpha} - 1}{0{,}1} a_{rel}$ since $K_{H\alpha} \geq 1{,}0$	(B.91)	1,000	E: 34		

Table B.7 — Calculation of surface durability (pitting)

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Z-factors						
Factors for calculation of mid-zone factor for $\varepsilon_{v\beta} = 2{,}095 \geq 1$	$F_1 = \varepsilon_{v\alpha}$	(B.92)	1,01		T: 2	
	$F_2 = \varepsilon_{v\alpha}$	(B.93)	1,01		T: 2	
Mid-zone factor	$Z_{M\text{-}B} = \dfrac{\tan\alpha_{vet}}{\sqrt{\left[\sqrt{\left(\dfrac{d_{va1}}{d_{vb1}}\right)^2 - 1} - F_1 \dfrac{\pi}{z_{v1}}\right] \cdot \left[\sqrt{\left(\dfrac{d_{va2}}{d_{vb2}}\right)^2 - 1} - F_2 \dfrac{\pi}{z_{v2}}\right]}}$	(B.94)	0,963		E: 6	
Elasticity factor	$Z_E = \sqrt{\dfrac{1}{\pi\left(\dfrac{1-\nu_1^2}{E_1} + \dfrac{1-\nu_2^2}{E_2}\right)}}$	(B.95)	189,8 $(N/mm^2)^{1/2}$		E: 51	
Bevel gear factor	Z_K	(B.96)	0,85		E: 11	
Lubricant factor	$C_{ZL} = 0{,}08 \dfrac{\sigma_{H\,lim} - 850}{350} + 0{,}83$ using $\sigma_{Hlim} = 1\,200\ N/mm^2$, which is the upper allowed limit in the above formula.	(B.97)	0,91		E: 54	

Table B.7 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Lubricant factor	$Z_L = C_{ZL} + \frac{4\left(1{,}0 - C_{ZL}\right)}{\left(1{,}2 + \frac{134}{\nu_{40}}\right)^2}$ (B.98)	0,992		E: 53	
Speed factor	$C_{ZV} = 0{,}08 \frac{\sigma_{H\,lim} - 850}{350} + 0{,}85$ (B.99) using σ_{Hlim} = 1 200 N/mm², which is the upper allowed limit in the above formula.	0,93		E: 56	
	$Z_v = C_{ZV} + \frac{2\left(1{,}0 - C_{ZV}\right)}{\sqrt{0{,}8 + \frac{32}{v_{mt2}}}}$ (B.100)	0,97		E: 55	
Roughness factor with the radius of relative curvature $\rho = \rho_{rel}$ = 17,93 mm for Method B1 (see ISO 10300–1:2014, Annex A)	$Rz_{10} = \frac{Rz_1 + Rz_2}{2} \cdot \sqrt[3]{\frac{10}{\rho}}$ (B.101)	6,586 µm		E: 57	
	$C_{ZR} = 0{,}12 + \frac{1\,000 - \sigma_{H\,lim}}{5\,000}$ (B.102) using σ_{Hlim} = 1 200 N/mm², which is the upper allowed limit in the above formula.	0,08		E: 59	

Table B.7 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Roughness factor with the radius of relative curvature $\rho = \rho_{rel} = 17{,}93$ mm for Method B1 (see ISO 10300‑1:2014, Annex A)	$Z_R = \left(\frac{3}{Rz_{10}}\right)^{C_{ZR}}$	(B.103)	0,939		E: 58	
Product of the lubricant influence factors	$Z_L Z_v Z_R$	(B.104)	0,904			
Size factor	Z_X for Method B1 (see ISO 10300‑1:2014, 6.5.1)	(B.105)	1,0			
Hypoid factor	$Z_{Hyp} = 1 - 0{,}3\left(\frac{v_{g\,par}}{v_{\Sigma\,vert}} - 0{,}15\right)$	(B.106)	0,952		E: 12	
Life factor for pinion	$Z_{NT,1}$	(B.107)	1,0		T: 4	
Life factor for wheel	$Z_{NT,2}$	(B.108)	1,0		T: 4	
Work hardening factor	$Z_W = 1{,}2 - \frac{HB - 130}{1\,700}$ Set $Z_w = 1{,}0$ because pinion and wheel with equal hardness.	(B.109)	1,0		E: 60	
Contact stress formula						
Nominal normal force of the virtual cylindrical gear at mean point P	$F_n = \frac{F_{mt1}}{\cos\alpha_{nD}\cos\beta_{m1}}$	(B.110)	15 298 N		E: 3	

Table B.7 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Nominal value of the contact stress	$\sigma_{H0} = \sqrt{\frac{F_n}{l_{bm}\rho_{rel}}} Z_{M\text{-}B} Z_{LS} Z_E Z_K$	(B.111)	872,4 N/mm^2		E: 2	
Contact stress	$\sigma_H = \sigma_{H0} \sqrt{K_A K_v K_{H\beta} K_{H\alpha}}$	(B.112)	1 175,3 N/mm^2		E: 1	
Permissible contact stress	$\sigma_{HP} = \sigma_{H\,lim} Z_{NT} Z_X Z_L Z_v Z_R Z_W Z_{Hyp}$	(B.113)	1 290,8 N/mm^2		E: 4	
Calculated safety factor for contact stress (pitting) on pinion and wheel	$S_{H1,2} = \frac{\sigma_{HP1,2}}{\sigma_{H1,2}}$	(B.114)	1,098		E: 5	

Table B.8 — Calculation of tooth root strength for pinion

Description	Formula		Result	References to ISO 10300‒1	ISO 10300‒2	ISO 10300‒3
				E: Formula	T: Table	F: Figure
Y‒factors						
Load sharing factor	$Y_{LS} = Z_{LS}^2$	(B.115)	0,495			E: 35
Geometry values for pinion according to Tables B.2 and B.3	$h_{a0} = h_{fp} = k_{hfp}\, m_{mn}$	(B.116)	3,300 mm			F: 2
Parameters for pinion	$E_{1,D} = \left(\frac{\pi}{4} - x_{sm1}\right) m_{mn} - h_{a0} \tan\alpha_{eD} - \frac{\rho_{a01,D}\left(1 - \sin\alpha_{eD}\right) - s_{pr1,D}}{\cos\alpha_{eD}}$	(B.117)	0,212			E: 7
	$E_{1,C} = \left(\frac{\pi}{4} - x_{sm1}\right) m_{mn} - h_{a0} \tan\alpha_{eC} - \frac{\rho_{a01,C}\left(1 - \sin\alpha_{eC}\right) - s_{pr1,C}}{\cos\alpha_{eC}}$	(B.118)	0,212			E: 7
	$G_{1,D} = \frac{\rho_{a01,D}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm1}$	(B.119)	−0,647			E: 8
	$G_{1,C} = \frac{\rho_{a01,C}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm1}$	(B.120)	−0,647			E: 8
	$H_{1,D} = \frac{2}{z_{vn1,D}}\left(\frac{\pi}{2} - \frac{E_{1,D}}{m_{mn}}\right) - \frac{\pi}{3}$	(B.121)	−0,969			E: 9

Table B.8 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
	$H_{1,C} = \frac{2}{z_{vn1,C}}\left(\frac{\pi}{2} - \frac{E_{1,C}}{m_{mn}}\right) - \frac{\pi}{3}$ (B.122)	−0,969			E: 9
Iteration starting with $\theta = \pi/6$ until $(\theta_{new} - \theta)$ < 0,000 001	$\theta_{1,D,C} = \frac{2G_{1,D,C}}{z_{vn1,D,C}} \tan\theta_{1,D,C} - H_{1,D,C}$ (B.123)	Initial: 54,398°			E: 10
		Final: 52,945°			E: 10
Tooth root chordal thickness on drive side	$s_{Fn1,D} = m_{mn} z_{vn1,D} \sin\left(\frac{\pi}{3} - \theta_{1,D}\right) + m_{mn}\sqrt{3}\left(\frac{G_{1,D}}{\cos\theta_{1,D}} - \frac{\rho_{a01,D}}{m_{mn}}\right)$ (B.124)	6,068 mm			E: 11
Tooth root chordal thickness on coast side	$s_{Fn1,C} = m_{mn} z_{vn1,C} \sin\left(\frac{\pi}{3} - \theta_{1,C}\right) + m_{mn}\sqrt{3}\left(\frac{G_{1,C}}{\cos\theta_{1,C}} - \frac{\rho_{a01,C}}{m_{mn}}\right)$ (B.125)	6,068 mm			E: 11
Tooth root chord	$s_{Fn1} = 0{,}5 s_{Fn1,D} + 0{,}5 s_{Fn1,C}$ (B.126)	6,068 mm			E: 12
Fillet radius at contact point of 30° tangent on drive side	$\rho_{F1,D} = \rho_{a01,D} + \frac{2G_{1,D}^2 m_{mn}}{\cos\theta_{1,D}\left(z_{vn1,D}\cos^2\theta_{1,D} - 2G_{1,D}\right)}$ (B.127)	1,042 mm			E: 13
Normal pressure angle at tooth tip on drive side	$\alpha_{an1,D} = \arccos\left(\frac{d_{vbn1,D}}{d_{van1,D}}\right)$ (B.128)	28,394°			E: 16

Table B.8 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Auxiliary angle for tooth form and tooth correction factor on drive side	$\gamma_{a1,D} = \frac{1}{z_{vn1,D}}\left[\frac{\pi}{2} + 2\left(x_{hm1} \tan\alpha_{eD} + x_{sm1}\right)\right] + \operatorname{inv}\alpha_{eD} - \operatorname{inv}\alpha_{an1,D}$ (B.129)	1,079°			E: 17
Load application angle at tooth tip of virtual cylindrical gear on drive side	$\alpha_{Fan1,D} = \alpha_{an1,D} - \gamma_{a1,D}$ (B.130)	27,315°			E: 15
Bending moment arm on drive side	$h_{Fa1,D} = \frac{m_{mn}}{2}\left[\left(\cos\gamma_{a1,D} - \sin\gamma_{a1,D} \tan\alpha_{Fan1,D}\right)\frac{d_{van1,D}}{m_{mn}} - z_{vn1,D} \cos\left(\frac{\pi}{3} - \theta_{1,D}\right) - \frac{G_{1,D}}{\cos\theta_{1,D}} + \frac{\rho_{a01,D}}{m_{mn}}\right]$ (B.131)	5,098 mm			E: 14
Tooth form factor on drive side	$Y_{Fa1,D} = \frac{6\frac{h_{Fa1,D}}{m_{mn}}\cos\alpha_{Fan1,D}}{\left(\frac{s_{Fn1}}{m_{mn}}\right)^2 \cos\alpha_{n1,D}}$ (B.132)	2,046			E: 6

Table B.8 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Stress correction factor on drive side	$L_{a1,D} = \dfrac{s_{Fn1}}{h_{Fa1,D}}$	(B.133)	1,190			E: 25
	$q_{s1,D} = \dfrac{s_{Fn1}}{2\rho_{F1,D}}$	(B.134)	2,911			E: 26
	$Y_{Sa1,D} = \left(1{,}2 + 0{,}13\ L_{a1,D}\right)\ q_{s1,D}^{\left(\frac{1}{1{,}21\ +\ 2{,}3/L_{a1,D}}\right)}$	(B.135)	1,903			E: 24
Relative surface condition factor for R_z = 16 µm and through hardened and case hardened steels	$Y_{R\ rel\ T} = \dfrac{Y_R}{Y_{RT}} = 1{,}674 - 0{,}529\ \left(Rz + 1\right)^{1/10}$	(B.136)	0,972			E: 39
Life factor for pinion	$Y_{NT,1}$	(B.137)	1,000			
Stress correction factor	Y_{ST}	(B.138)	2,000			E: 4
Size factor	$Y_X = 1{,}05 - 0{,}01 m_{mn}$ Set $Y_X = 1{,}0$ since range is $0{,}8 \leq Y_X \leq 1{,}0$.	(B.139)	1,000			F: 7 E: 188
Bevel spiral angle factor using l_{bm} from Formula (B.73) for l_{bb}	$Y_{BS} = \dfrac{a_{BS}}{c_{BS}}\left(\dfrac{l_{bb}}{b_a} - 1{,}05 \cdot b_{BS}\right)^2 + 1$	(B.140)	1,909			E: 28

Table B.8 — *continued*

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
where	$b_a = b_v / \cos \beta_v$	(B.141)	40,9 mm			E: 32
	$l_{bb} = l_{bm} \dfrac{\cos\beta_{vb}}{\cos\beta_v}$	(B.142)	14,125 mm			E: 33
	$h = (h_{m1} + h_{m2}) / 2$	(B.143)	5,939 mm			E: 34
Relative notch sensitivity factor	$Y_{\delta\ \mathrm{rel\ T\ 1}} = \dfrac{1+\sqrt{\rho' \chi_1^X}}{1+\sqrt{\rho' \chi_T^X}}$	(B.144)	1,004			E: 42
Contact ratio factor, Y_ε, for $\varepsilon_{v\beta} = 2{,}095 > 1$	Y_ε	(B.145)	0,625			E: 27c
Tooth root stress formula for pinion						
Nominal tooth root stress	$\sigma_{F01} = \dfrac{F_{vmt}}{b_v\, m_{mn}} Y_{Fa} Y_{Sa} Y_\varepsilon Y_{BS} Y_{LS}$	(B.146)	310,3 N/mm^2			E: 2
Local tooth root stress	$\sigma_{F1} = \sigma_{F0} K_A K_v K_{F\beta} K_{F\alpha}$	(B.147)	531,2 N/mm^2			E: 1
Permissible tooth root stress	$\sigma_{FP1} = \sigma_{F\,lim} Y_{ST} Y_{NT} Y_{\delta\ \mathrm{rel\ T\ 1}} Y_{R\ \mathrm{rel\ T}} Y_X$	(B.148)	936,4 N/mm^2			E: 4
Calculated safety for tooth root strength on pinion	$S_{F1} = \dfrac{\sigma_{FP1}}{\sigma_{F1}}$	(B.149)	1,763			E: 5

Table B.9 — Calculation of tooth root strength for wheel

Description	Formula		Result	References to ISO 10300‑1	ISO 10300‑2	ISO 10300‑3
				E: Formula	T: Table	F: Figure
Y‑factors different from pinion						
Geometry values for wheel according to Tables B.2 and B.3	$h_{a0} = h_{fp} = k_{hfp}\, m_{mn}$	(B.150)	3,300 mm			
Parameters for wheel	$E_{2,D} = \left(\frac{\pi}{4} - x_{sm2}\right) m_{mn} - h_{a0} \tan\alpha_{eD} - \frac{\rho_{a02,D}\left(1 - \sin\alpha_{eD}\right) - s_{pr2,D}}{\cos\alpha_{eD}}$	(B.151)	0,196 mm			E: 7
	$E_{2,C} = \left(\frac{\pi}{4} - x_{sm2}\right) m_{mn} - h_{a0} \tan\alpha_{eC} - \frac{\rho_{a02,C}\left(1 - \sin\alpha_{eC}\right) - s_{pr2,C}}{\cos\alpha_{eC}}$	(B.152)	0,196 mm			E: 7
	$G_{2,D} = \frac{\rho_{a02,D}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm2}$	(B.153)	−1,095			E: 8
	$G_{2,C} = \frac{\rho_{a02,C}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm2}$	(B.154)	−1,095			E: 8
	$H_{2,D} = \frac{2}{z_{vn2,D}}\left(\frac{\pi}{2} - \frac{E_{2,D}}{m_{mn}}\right) - \frac{\pi}{3}$	(B.155)	−1,035			E: 9
	$H_{2,C} = \frac{2}{z_{vn2,C}}\left(\frac{\pi}{2} - \frac{E_{2,C}}{m_{mn}}\right) - \frac{\pi}{3}$	(B.156)	−1,035			E: 9

Table B.9 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Iteration starting with $\theta = \pi/6$ until $(\theta_{new} - \theta) < 0{,}000\,001$	$\theta_{2,D,C} = \frac{2G_{2,D,C}}{z_{vn2,D,C}} \tan\theta_{2,D,C} - H_{2,D,C}$	(B.157)	Initial: 0,524°			E: 10
			Final: 1,020°			E: 10
Tooth root chordal thickness on drive side	$s_{Fn2,D} = m_{mn} z_{vn2,D} \sin\left(\frac{\pi}{3} - \theta_{2,D}\right) + m_{mn}\sqrt{3}\left(\frac{G_{2,D}}{\cos\theta_{2,D}} - \frac{\rho_{a02,D}}{m_{mn}}\right)$	(B.158)	5,676 mm			E: 11
Tooth root chordal thickness on coast side	$s_{Fn2,C} = m_{mn} z_{vn2,C} \sin\left(\frac{\pi}{3} - \theta_{2,C}\right) + m_{mn}\sqrt{3}\left(\frac{G_{2,C}}{\cos\theta_{2,C}} - \frac{\rho_{a02,C}}{m_{mn}}\right)$	(B.159)	5,676 mm			E: 11
Tooth root chord	$s_{Fn2} = 0{,}5 s_{Fn2,D} + 0{,}5 s_{Fn2,C}$	(B.160)	5,676 mm			E: 12
Fillet radius at contact point of 30° tangent	$\rho_{F2,D} = \rho_{a02,D} + \frac{2G_{2,D}^2 m_{mn}}{\cos\theta_{2,D}\left(z_{vn2,D}\cos^2\theta_{2,D} - 2G_{2,D}\right)}$	(B.161)	1,369 mm			E: 13
Normal pressure angle at tooth tip on drive side	$\alpha_{an2,D} = \arccos\left(\frac{d_{vbn2,D}}{d_{van2,D}}\right)$	(B.162)	20,841°			E: 16
Auxiliary angle for tooth form and tooth correction factor on drive side	$\gamma_{a2,D} = \frac{1}{z_{vn2,D}}\left[\frac{\pi}{2} + 2\left(x_{hm2}\tan\alpha_{eD} + x_{sm2}\right)\right] + \mathrm{inv}\,\alpha_{eD} - \mathrm{inv}\,\alpha_{an2,D}$	(B.163)	0,159°			E: 17

Table B.9 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Load application angle at tooth tip of virtual cylindrical gear on drive side	$\alpha_{\mathrm{Fan2,D}} = \alpha_{\mathrm{an2,D}} - \gamma_{\mathrm{a2,D}}$ (B.164)	20,682°			E: 15
Bending moment arm on drive side	$h_{\mathrm{Fa2,D}} = \frac{m_{\mathrm{mn}}}{2}\left[\left(\cos\gamma_{\mathrm{a2,D}} - \sin\gamma_{\mathrm{a2,D}}\tan\alpha_{\mathrm{Fan2,D}}\right)\frac{d_{\mathrm{van2,D}}}{m_{\mathrm{mn}}} - z_{\mathrm{vn2,D}}\cos\left(\frac{\pi}{3} - \theta_{\mathrm{2,D}}\right) - \frac{G_{\mathrm{2,D}}}{\cos\theta_{\mathrm{2,D}}} + \frac{\rho_{\mathrm{a02,D}}}{m_{\mathrm{mn}}}\right]$ (B.165)	4,972 mm			E: 14
Tooth form factor on drive side	$Y_{\mathrm{Fa2,D}} = \dfrac{6\dfrac{h_{\mathrm{Fa2,D}}}{m_{\mathrm{mn}}}\cos\alpha_{\mathrm{Fan2,D}}}{\left(\dfrac{s_{\mathrm{Fn2}}}{m_{\mathrm{mn}}}\right)^2\cos\alpha_{\mathrm{n2,D}}}$ (B.166)	2,401			E: 6
Stress correction factor on drive side	$L_{\mathrm{a2,D}} = \frac{s_{\mathrm{Fn2}}}{h_{\mathrm{Fa2,D}}}$ (B.167)	1,142			E: 25

Table B.9 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
	$q_{s2,D} = \frac{s_{Fn2}}{2\rho_{F2,D}}$ (B.168)	2,073			E: 26
	$Y_{Sa2,D} = \left(1,2 + 0,13\ L_{a2,D}\right)\ q_{s2,D}^{\left(\frac{1}{1,21 + 2,3/L_{a2,D}}\right)}$ (B.169)	1,690			E: 24
Relative surface condition factor for R_z = 16 µm for through hardened and case hardened steels	$Y_{R\ rel\ T} = \frac{Y_R}{Y_{RT}} = 1,674 - 0,529\ \left(Rz + 1\right)^{1/10}$ (B.170)	0,972			E: 39
Bevel spiral angle factor using l_{bm} from Formula (B.73) for l_{bb}	$Y_{BS} = \frac{a_{BS}}{c_{BS}}\left(\frac{l_{bb}}{b_a} - 1,05 \cdot b_{BS}\right)^2 + 1$ (B.171)	1,909			E: 28
where	$b_a = b_v / \cos \beta_v$ (B.172)	40,9 mm			E: 32
	$l_{bb} = l_{bm}\frac{\cos\beta_{vb}}{\cos\beta_v}$ (B.173)	14,125 mm			E: 33
	$h = (h_{m1} + h_{m2}) / 2$ (B.174)	5,939 mm			E: 34

Table B.9 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Relative notch sensitivity factor	$Y_{\delta\ \mathrm{rel\ T\ 2}} = \dfrac{1+\sqrt{\rho' \chi_2^{\mathrm{X}}}}{1+\sqrt{\rho' \chi_{\mathrm{T}}^{\mathrm{X}}}}$ (B.175)	0,996			E: 42
Contact ratio factor, Y_ε, for $\varepsilon_{\mathrm{v}\beta} = 2{,}095 > 1$	Y_ε (B.176)	0,625			E: 27c
Tooth root stress formula for wheel					
Nominal tooth root stress	$\sigma_{\mathrm{F01}} = \dfrac{F_{\mathrm{vmt}}}{b_{\mathrm{v}}\, m_{\mathrm{mn}}} Y_{\mathrm{Fa}} Y_{\mathrm{Sa}} Y_\varepsilon Y_{\mathrm{BS}} Y_{\mathrm{LS}}$ (B.177)	323,5 N/mm^2			E: 2
Local tooth root stress	$\sigma_{\mathrm{F1}} = \sigma_{\mathrm{F0}} K_{\mathrm{A}} K_{\mathrm{v}} K_{\mathrm{F\beta}} K_{\mathrm{F\alpha}}$ (B.178)	553,8 N/mm^2			E: 1
Permissible tooth root stress	$\sigma_{\mathrm{FP1}} = \sigma_{\mathrm{F\ lim}} Y_{\mathrm{ST}} Y_{\mathrm{NT}} Y_{\delta\ \mathrm{rel\ T\ 2}} Y_{\mathrm{R\ rel\ T}} Y_{\mathrm{X}}$ (B.179)	929 N/mm^2			E: 4
Calculated safety for tooth root strength on wheel	$S_{\mathrm{F2}} = \dfrac{\sigma_{\mathrm{FP2}}}{\sigma_{\mathrm{F2}}}$ (B.180)	1,677			E: 5

B.3 Calculation of Sample 2 according to Method B2

Table B.10 — Calculation of surface durability (pitting)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Approximate values for application factors					
Relative face width	$b_v = b_2 / m_{et2}$ (B.181)	7,413	E: B.1		
Relative mean back cone distance on pinion	$R_{mpt1} = \frac{R_{m1} \tan\delta_1}{m_{et2}}$ (B.182)	7,068	E: B.2		
Relative mean back cone distance on wheel	$R_{mpt2} = \frac{R_{m2} \tan\delta_2}{m_{et2}}$ (B.183)	47,455	E: B.2		

Table B.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Angle between direction of contact and the pitch tangent	$\cot\left(\zeta_R - \lambda\right) = \cot\zeta_R \left(1{,}0 + \frac{z_1 \cos\delta_{f2}}{z_2 \cos\delta_{a1} \cos\zeta_R}\right)$	(B.184)	cot(8,992°)	E: B.3		
Face contact ratio for hypoid gears	$\varepsilon_{v\beta} = \left[\frac{\cos\beta_{m2}}{\cot\left(\zeta_R - \lambda\right)} + \sin\beta_{m2}\right] \frac{b_2}{\pi \ m_{mn}}$	(B.185)	2,704	E: B.4b		
Relative mean virtual pitch radius on pinion	$r_{vn1} = \frac{R_{mpt1}}{\cos^2\beta_{m1}}$	(B.186)	17,105	E: B.5		
Relative mean virtual pitch radius on wheel	$r_{vn2} = \frac{R_{mpt2}}{\cos^2\beta_{m2}}$	(B.187)	77,717	E: B.5		
Relative centre distance	$a_{vn} = r_{vn1} + r_{vn2}$	(B.188)	94,822	E: B.6		
Relative mean virtual dedendum on pinion	$h_{vfm1} = h_{fm1} / m_{et2}$	(B.189)	0,620	E: B.7		
Relative mean virtual dedendum on wheel	$h_{vfm2} = h_{fm2} / m_{et2}$	(B.190)	1,011	E: B.7		
Relative virtual tooth thickness on pinion	$s_{vmn1} = s_{mn1} \ / \ m_{et2}$	(B.191)	1,216	E: B.8		

Table B.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Relative virtual tooth thickness on wheel	$s_{vmn2} = s_{mn2} / m_{et2}$	(B.192)	0,801	E: B.8		
Relative mean virtual tip radius on pinion	$r_{va1} = r_{vn1} + h_{am1} / m_{et2}$	(B.193)	17,953	E: B.9		
Relative mean virtual tip radius on wheel	$r_{va2} = r_{vn2} + h_{am2} / m_{et2}$	(B.194)	78,173	E: B.9		
Angular pitch of virtual cylindrical wheel (required in ISO 10300–3:2014, 7.4.5)	$\theta_{v2} = \dfrac{\pi\, m_{mn}}{m_{et2} r_{vn2}}$	(B.195)	1,511°	E: B.10		
Relative edge radius of tool on pinion	$\rho_{va01} = \rho_{a01} / m_{et2}$	(B.196)	0,198	E: B.11		
Relative edge radius of tool on wheel	$\rho_{va02} = \rho_{a02} / m_{et2}$	(B.197)	0,297	E: B.11		
Virtual spiral angle for hypoid gears	$\beta_v = \beta_{m1} - \lambda_r$	(B.198)	48,272°	E: B.12b		
where	$\tan\left(\beta_{m1} - \lambda_r\right) = \dfrac{R_{m2} \tan\delta_{f2} \tan\beta_{m1} + R_{m1} \tan\delta_{a1} \tan\beta_{m2}}{R_{m2} \tan\delta_{f2} + R_{m1} \tan\delta_{a1}}$	(B.199)	tan(48,272°)	E: B.13		
Adjusted pressure angle	$\alpha_a = \alpha_{eD} - 90° \cos\delta_2 \cos\beta_{m2} / z_2$	(B.200)	19,232°	E: B.14		

Table B.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Base virtual helix angle	$\sin\beta_{vb} = \sin\beta_v \cos\alpha_a$	(B.201)	sin(44,802°)	E: B.15		
Relative mean virtual base radius on pinion	$r_{vbn1} = r_{vn1}\cos\alpha_a$	(B.202)	16,151	E: B.16		
Relative mean virtual base radius on wheel	$r_{vbn2} = r_{vn2}\cos\alpha_a$	(B.203)	73,379	E: B.16		
Relative length of action from pinion tip to pitch circle in the normal section	$g_{v\alpha na} = \sqrt{r_{va1}^2 - r_{vbn1}^2} - r_{vn1}\sin\alpha_a$	(B.204)	2,206	E: B.17		
Relative length of action from wheel tip to pitch circle in the normal section	$g_{v\alpha nr} = \sqrt{r_{va2}^2 - r_{vbn2}^2} - r_{vn2}\sin\alpha_a$	(B.205)	1,355	E: B.18		
Relative length of action in normal section	$g_{v\alpha n} = g_{v\alpha na} + g_{v\alpha nr}$	(B.206)	3,561	E: B.19		
Relative mean normal pitch of virtual cylindrical gear	$p_{mn} = \dfrac{2{,}0\pi\, m_{mn}}{m_{et2}\cos\alpha_a\left(\cos^2\beta_{m1} + \cos^2\beta_{m2} + 2{,}0\tan^2\alpha_a\right)}$	(B.207)	3,426	E: B.20		
Profile contact ratio in mean normal section	$\varepsilon_{v\alpha n} = \dfrac{g_{v\alpha n}}{p_{mn}}$	(B.208)	1,040	E: B.21		

Table B.10 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Profile contact ratio in mean transverse section	$\varepsilon_{\mathrm{v}\alpha} = \varepsilon_{\mathrm{v}\alpha\mathrm{n}} \cos^2 \beta_{\mathrm{vb}}$ (B.209)	0,523	E: B.22		
Modified contact ratio for bevel gears without hypoid offset	$\varepsilon_{\mathrm{v}\gamma} = \sqrt{\varepsilon_{\mathrm{v}\alpha}^2 + \varepsilon_{\mathrm{v}\beta}^2}$ (B.210)	2,755	E: B.23		
Contact shift factor, see also ISO 10300‑1:2014, Figure B.7	$k' = \frac{z_2 - z_1}{3{,}2\, z_2 + 4{,}0\, z_1}$ (B.211)	0,156	E: B.24		
Pitting resistance geometry factor					
Angle between contact direction and tooth tangent in pitch plane	$\cot\left(\beta_{\mathrm{m1}} - \lambda_1\right) = \frac{\cos \zeta_{\mathrm{R}}}{\cos \beta_{\mathrm{m1}} \cos \beta_{\mathrm{m2}} \tan\left(\beta_{\mathrm{m1}} - \lambda_{\mathrm{r}}\right)} - \tan \beta_{m2}$ (B.212)	cot(46,522°)		E: 24	
Angle between projection of pinion axis and direction of contact in pitch plane	$\lambda_1 = \beta_{\mathrm{m1}} - \left(\beta_{\mathrm{m1}} - \lambda_1\right)$ (B.213)	3,476°		E: 25	
Angle of contact line relative to the root cone	$\tan w = \frac{\sin \alpha_{\mathrm{a}} \tan\left(\beta_{\mathrm{m1}} - \lambda_{\mathrm{r}}\right)}{\cos \alpha_{\mathrm{lim}}}$ (B.214)	tan(20,286°)		E: 26	

Table B.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Mean base spiral angle	$\cos\beta_{bm} = \frac{1{,}0}{\sqrt{\tan^2(\beta_{m1} - \lambda_r)\cos^2\alpha_a + 1{,}0}}$	(B.215)	cos(46,633°)		E: 27	
Relative mean normal base pitch	$p_{nb} = \frac{\pi\, m_{mn} \cos\alpha_a \cos\beta_{bm}}{m_{et2}\cos(\beta_{m1} - \lambda_r)}$	(B.216)	1,996		E: 28	
Angle between projection of wheel axis and direction of contact in pitch plane	$\lambda_2 = (\beta_{m1} - \lambda_r) - \beta_{m2}$	(B.217)	9,623°		E: 29	
Relative base face width	$b_b = \frac{b_2}{m_{et2}\cos\lambda_2}$	(B.218)	7,520		E: 30	
Pressure angle at point of load application on pinion	$\cos\alpha_{L1} = \cos\alpha_a \left[1{,}0 - \frac{(r_{va1} - r_{vn1})\cos^2\beta_{m1}}{r_{va1} - r_{vn1} + R_{mpt1}}\right]$	(B.219)	cos(25,525°)		E: 31	
Pressure angle at point of load application on wheel	$\cos\alpha_{L2} = \cos\alpha_a \left[1{,}0 - \frac{(r_{va2} - r_{vn2})\cos^2\beta_{m2}}{r_{va2} - r_{vn2} + R_{mpt2}}\right]$	(B.220)	cos(20,166°)		E: 31	

Table B.10 — *continued*

Description	Formula		Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Radius of curvature difference between point of load application and mean point on pinion	$\rho_{\Delta 1} = \dfrac{r_{\text{va1}} - r_{\text{vn1}} + R_{\text{mpt1}}}{\cos^2 \beta_{\text{m1}}} \cos\alpha_{\text{L1}} \left(\tan\alpha_{\text{L1}} - \tan\alpha_{\text{a}}\right)$	(B.221)	2,224		E: 32	
Radius of curvature difference between point of load application and mean point on wheel	$\rho_{\Delta 2} = \dfrac{r_{\text{va2}} - r_{\text{vn2}} + R_{\text{mpt2}}}{\cos^2 \beta_{\text{m2}}} \cos\alpha_{\text{L2}} \left(\tan\alpha_{\text{L2}} - \tan\alpha_{\text{a}}\right)$	(B.222)	1,355		E: 32	
Radius of curvature change	$\rho_{\Delta\text{red}} = \cos\beta_{\text{bm}} \left(\rho_{\Delta 1} + \rho_{\Delta 2}\right)$	(B.223)	2,457		E: 33	
Relative length of action within the contact ellipse	$g_\eta = \sqrt{\rho_{\Delta\text{red}}^2 \cos^2\beta_{\text{bm}} + b_{\text{b}}^2 \sin^2\beta_{\text{bm}}}$	(B.224)	5,721		E: 34	
Radius of relative profile curvature and load sharing ratio at critical point						
Length of action at critical point in contact ellipse (iteration starting with $y_{\text{I}} = 0$)	$g_{\eta\text{I}} = \sqrt{g_\eta^2 - 4{,}0 y_{\text{I}}^2}$	(B.225)	Initial: 5,721		E: 36	
			Final: 5,658		E: 36	
Length of action considering adjacent teeth	$g_{\eta\text{I}\Sigma}^3 = g_{\eta\text{I}}^3 + \sqrt{\left[g_{\eta\text{I}}^2 - 4{,}0 p_{\text{nb}}\left(p_{\text{nb}} + 2{,}0 y_{\text{I}}\right)\right]^3} + \sqrt{\left[g_{\eta\text{I}}^2 - 4{,}0 p_{\text{nb}}\left(p_{\text{nb}} - 2{,}0 y_{\text{I}}\right)\right]^3}$ $+ \sqrt{\left[g_{\eta\text{I}}^2 - 8{,}0 p_{\text{nb}}\left(2{,}0 p_{\text{nb}} + 2{,}0 y_{\text{I}}\right)\right]^3} + \sqrt{\left[g_{\eta\text{I}}^2 - 8{,}0 p_{\text{nb}}\left(2{,}0 p_{\text{nb}} - 2{,}0 y_{\text{I}}\right)\right]^3}$ $+ \sqrt{\left[g_{\eta\text{I}}^2 - 16{,}0 p_{\text{nb}}\left(4{,}0 p_{\text{nb}} + 2{,}0 y_{\text{I}}\right)\right]^3} + \sqrt{\left[g_{\eta\text{I}}^2 - 16{,}0 p_{\text{nb}}\left(4{,}0 p_{\text{nb}} - 2{,}0 y_{\text{I}}\right)\right]^3}$	(B.226)	Initial: 324,840		E: 37	
			Final: 318,610		E: 37	

Table B.10 — *continued*

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
Load sharing ratio	$\varepsilon_{NI} = g_{\eta I}^3 / g_{\eta I \Sigma}^3$	(B.227)	Initial: 0,576		E: 38	
			Final: 0,568		E: 38	
Length of contact line	$g_c = g_{\eta I} \ \rho_{\Delta red} \ y_I / g_{\eta}^2$	(B.228)	Initial: 3,230		E: 39	
			Final: 3,194		E: 39	
Position change along the path of contact	$g_{\eta\Delta} = \frac{\rho_{\Delta red}^2 \ y_I}{g_{\eta I}^2} + k' g_c \tan\beta_{bm} + \frac{0{,}5\rho_{\Delta red}}{\cos\beta_{bm}} - \rho_{\Delta 2}$	(B.229)	Initial: 0,967		E: 40	
			Final: 0,881		E: 40	
Intermediate value	$X = \frac{\sin^2 w \cos\alpha_{lim} \cos\left(\zeta_R - \lambda_1\right)\cos\lambda_1}{\sin^2\left(\beta_{m1} - \lambda_1\right)\sin\alpha_a \cos\zeta_R}$	(B.230)	Initial: 0,698		E: 41	
			Final: 0,698		E: 41	
Profile radius of curvature on pinion	$\rho_1 = R_{mpt1} X \pm g_{\eta\Delta}$	(B.231)	Initial: 5,897		E: 42	
			Final: 5,811		E: 42	
Profile radius of curvature on wheel	$\rho_2 = R_{mpt2} X \pm g_{\eta\Delta}$	(B.232)	Initial: 32,137		E: 42	
			Final: 32,223		E: 42	
Relative radius of profile curvature between pinion and wheel	$\rho_t = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}$	(B.233)	Initial: 4,983		E: 43	
			Final: 4,923		E: 43	
Inertia factor with $\varepsilon_v > 2{,}0_{\gamma}$	Z_i	(B.234)	Initial: 1,0		E: 44b	
			Final: 1,0		E: 44b	

Table B.10 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Pitting resistance geometry factor	$Z_I = \dfrac{g_c \, \rho_t \cos\alpha_a \, m_{mn}}{b_b \, z_1 \, Z_i \, \varepsilon_{NI} \, m_{et2}}$ (B.235)	Initial: 0,176		E: 45	
		Final: 0,174		E: 45	
Face width factor					
for 12,7 ≤ b_2 ≤ 79,8 mm	$Z_{FW} = 0{,}00492 b_2 + 0{,}4375$ (B.236)	0,585		E: 46b	
Contact stress adjustment factor					
for carburized case hardened steel	Z_A (B.237)	0,967		E: 47	
Elasticity factor					
for a steel on steel gear pair	Z_E (B.238)	189,8 $(N/mm^2)^{1/2}$		E: 51	
Lubricant film influence factors					
Product of the lubricant influence factors	$Z_L Z_V Z_R$ (B.239)	0,904			
Work hardening factor					
Work hardening factor	Z_W Set Z_w = 1,0 because pinion and wheel with equal hardness. (B.240)	1,000			
Life factor					
Life factor	Z_{NT} (B.241)	1,000		T: 4	

Table B.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
K–factors						
Dynamic factor	K_V	(B.242)	1,000	E: 14		
Face load factor	$K_{H,\beta}$	(B.243)	1,650	E: 27		
Contact stress formula for drive and coast side						
Nominal contact stress	$\sigma_{H0-B2} = \sqrt{\frac{F_{mt1} d_{m1} Z_{FW}}{b_2 Z_I} \left(\frac{z_2}{d_{e2} z_1} \right)^2} \cdot Z_E$	(B.244)	853,1 N/mm²		E: 21	
Contact stress	$\sigma_{H-B2} = \sigma_{H0-B2} \sqrt{K_A K_v K_{H\beta}} \cdot Z_A \leq \sigma_{HP-B2}$	(B.245)	1 111,4 N/mm²		E: 20	
Permissible contact stress	$\sigma_{HP-B2} = \sigma_{H\,lim} Z_{NT} Z_L Z_v Z_R Z_W$	(B.246)	1 355,3 N/mm²		E: 22	
Calculated safety factor for contact stress for pinion and wheel	$S_{H-B2} = \frac{\sigma_{HP-B2}}{\sigma_{H-B2}} > S_{H\,min}$	(B.247)	1,219		E: 23	

Table B.11 — Calculation of tooth root strength on pinion and wheel

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Geometry factor for hypoid gears (pinion and wheel)						
Initial formulas						
Drive flank pressure angle in wheel root coordinates	$\alpha_{\mathrm{Dnf}} = \alpha_{\mathrm{nD}} - \theta_{\mathrm{f2}} \sin\beta_{\mathrm{m2}}$	(B.248)	15,251°			E: 84
Coast flank pressure angle in wheel root coordinates	$\alpha_{\mathrm{Cnf}} = \alpha_{\mathrm{nC}} + \theta_{\mathrm{f2}} \sin\beta_{\mathrm{m2}}$	(B.249)	24,749°			E: 85
Average pressure angle unbalance	$\Delta\alpha_1 = \frac{\left(\alpha_{\mathrm{Dnf}} - \alpha_{\mathrm{Cnf}}\right)}{2{,}0}$	(B.250)	−4,749°			E: 86
Limit pressure angle in wheel root coordinates	$\alpha_{\mathrm{f}} = \alpha_{\mathrm{lim}} - \theta_{\mathrm{f2}} \sin\beta_{\mathrm{m2}}$	(B.251)	−4,749°			E: 87
Relative distance from blade edge to centreline	$g_{\mathrm{rb}} = \frac{\left(h_{\mathrm{fm2}} \tan\frac{\alpha_{\mathrm{nD}} + \alpha_{\mathrm{nC}}}{2{,}0} + \frac{W_{\mathrm{m2}}}{2{,}0}\right)\cos\frac{\alpha_{\mathrm{nD}} + \alpha_{\mathrm{nC}}}{2{,}0}}{m_{\mathrm{et2}}}$	(B.252)	0,579			E: 88
Intermediate value	$\eta_{\mathrm{D}} = \tan\alpha_{\mathrm{Dnf}} \left(\frac{g_{\mathrm{rb}}}{\sin\alpha_{\mathrm{Dnf}}} - h_{\mathrm{vfm2}}\right)$	(B.253)	0,324			E: 89
Intermediate value	$\eta_{\mathrm{C}} = \tan\alpha_{\mathrm{Cnf}} \left(\frac{g_{\mathrm{rb}}}{\sin\alpha_{\mathrm{Cnf}}} - h_{\mathrm{vfm2}}\right)$	(B.254)	−0,171			E: 90

Table B.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Intermediate angle	$\tan\beta_a = \dfrac{\dfrac{W_{m2}}{2{,}0 m_{et2}} - \rho_{va02}\left(\sec\dfrac{\alpha_{nD}+\alpha_{nC}}{2{,}0} - \tan\dfrac{\alpha_{nD}+\alpha_{nC}}{2{,}0}\right)}{h_{vfm2} - \rho_{va02}}$	(B.255)	tan(3,206°)			E: 91
Intermediate angle	$\left(\beta_D - \Delta\alpha\right) = \beta_a - \Delta\alpha_1$	(B.256)	7,955°			E: 92
Intermediate angle	$\left(\beta_C - \Delta\alpha\right) = -\beta_a - \Delta\alpha_1$	(B.257)	1,543°			E: 93
Intermediate value	$g_1 = \dfrac{h_{vfm2} - \rho_{va02}}{\cos\beta_a}$	(B.258)	0,716			E: 94
Wheel angle between centreline and fillet point on drive side	$\tan\Delta\theta_D = \dfrac{g_1 \sin\left(\beta_D - \Delta\alpha\right)}{r_{vn2} - g_1\cos\left(\beta_D - \Delta\alpha\right)}$	(B.259)	tan(0,074°)			E: 95
Wheel angle between centreline and fillet point on coast side	$\tan\Delta\theta_C = \dfrac{g_1 \sin\left(\beta_C - \Delta\alpha\right)}{r_{vn2} - g_1\cos\left(\beta_C - \Delta\alpha\right)}$	(B.260)	tan(0,014°)			E: 96
Wheel angle between fillet points	$\Delta\theta_2 = \dfrac{\theta_{v2} + \Delta\theta_D + \Delta\theta_C}{2{,}0}$	(B.261)	0,799°			E: 97
Vertical distance from pitch circle to fillet point	$y_1 = r_{vn2} - \dfrac{\left[r_{vn2} - g_1\cos\left(\beta_D - \alpha_f\right)\right]\cos\left(\Delta\theta_2 - \Delta\theta_D\right)}{\cos\Delta\theta_D}$	(B.262)	0,715			E: 98

Table B.11 — *continued*

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
Horizontal distance from centreline to fillet point	$x_1 = \dfrac{\left[r_{vn2} - g_1 \cos\left(\beta_D - \alpha_f\right)\right] \sin\left(\Delta\theta_2 - \Delta\theta_D\right)}{\cos\Delta\theta_D}$	(B.263)	0,975			E: 99
Generated pressure angle of wheel at fillet point	$\alpha_{LN2} = \alpha_{Dnf} - \Delta\theta_2$	(B.264)	14,452°			E: 100
Distance from centreline to tool critical drive side fillet point	$\mu_{1D} = \eta_D + \tan\alpha_{Dnf}\left(h_{vfm1} + h_{vfm2}\right) + \rho_{va01}\left(\sec\alpha_{Dnf} - \tan\alpha_{Dnf}\right)$	(B.265)	0,920			E: 101
Distance from centreline to tool critical coast side fillet point	$\mu_{1C} = \eta_C + \tan\alpha_{Cnf}\left(h_{vfm1} + h_{vfm2}\right) - \rho_{va01}\left(\sec\alpha_{Cnf} - \tan\alpha_{Cnf}\right)$	(B.266)	−1,049			E: 102
Wheel angle between centreline and critical pinion drive side fillet point	$\tan\theta_{DLS} = \dfrac{\mu_{1D}}{r_{vn2} + h_{vfm1}}$	(B.267)	tan(0,673°)			E: 103
Wheel angle between centreline and critical pinion coast side fillet point	$\tan\theta_{CLS} = \dfrac{\mu_{1C}}{r_{vn2} + h_{vfm1}}$	(B.268)	tan(−0,767°)			E: 104
Radius from tool centre to critical pinion drive side fillet point	$R_{DL2} = \dfrac{r_{vn2} + h_{vfm1}}{\cos\theta_{DLS}}$	(B.269)	78,342			E: 105
Radius from tool centre to critical pinion coast side fillet point	$R_{CL2} = \dfrac{r_{vn2} + h_{vfm1}}{\cos\theta_{CLS}}$	(B.270)	78,343			E: 106

Table B.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Wheel angle from centreline to pinion tip on drive side	θ_{D1}	(B.271)	Initial: 1,511			
			Final: 1,204			
	$\Delta r_1 = r_{vn2}\left(e^{\theta_{D1}\tan\alpha_f} - 1{,}0\right)$	(B.272)	Initial: −0,170			E: 107
			Final: −0,136			E: 107
	$h_1 = \left(r_{vn2} + \Delta r_1\right)\sin\left(\alpha_{Dnf} + \theta_{D1}\right) - \left(r_{vn2}\sin\alpha_{Dnf} - g_{rb}\right)$	(B.273)	Initial: 2,503			E: 108
			Final: 2,114			E: 108
	$h_{1o} = \sqrt{r_{va1}^2 - \left(r_{vn1} - \Delta r_1\right)^2 \cos^2\left(\alpha_{vDnf} + \theta_{D1}\right)} - \left(r_{vn1} - \Delta r_1\right)\sin\left(\alpha_{Dnf} + \theta_{D1}\right)$	(B.274)	Initial: 1,999			E: 109
			Final: 2,114			E: 109
Wheel angle from centreline to tooth surface at critical fillet point on drive side	θ_{D2o}	(B.275)	Initial: 0,755			
			Final: 0,679			
	$\mu_{1Do} = r_{vn2} e^{\theta_{D2o}\tan\alpha_f}\sin\theta_{D2o}$	(B.276)	Initial: 1,025			E: 110
			Final: 0,920			E: 110
Wheel angle from centreline to tooth surface at critical fillet point on coast side	θ_{C2o}	(B.277)	Initial: −0,755			
			Final: −0,773			
	$\mu_{1Co} = r_{vn2} e^{\theta_{C2o}\tan\alpha_f}\sin\theta_{C2o}$	(B.278)	Initial: −1,027			E: 111
			Final: −1,051			E: 111

Table B.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Pinion angle from centreline to tooth surface at critical drive side fillet point	$r_{\mathrm{vn2}}\left(e^{\theta_{\mathrm{D2o}}\tan\alpha_{\mathrm{f}}}-1{,}0\right)=r_{\mathrm{vn1}}\left(1{,}0-e^{\theta_{\mathrm{D1o}}\tan\alpha_{\mathrm{f}}}\right)$ (B.279) Formula shall be solved for θ_{D1o}.	−3,075°			E: 112
Pinion angle from centreline to tooth surface at critical coast side fillet point	$r_{\mathrm{vn2}}\left(e^{\theta_{\mathrm{C2o}}\tan\alpha_{\mathrm{f}}}-1{,}0\right)=r_{\mathrm{vn1}}\left(1{,}0-e^{\theta_{\mathrm{C1o}}\tan\alpha_{\mathrm{f}}}\right)$ (B.280) Formula shall be solved for θ_{C1o}.	3,522°			E: 113
Wheel difference angle between tool and surface at drive side fillet point	$\Delta\theta_{\mathrm{D2o}}=\theta_{\mathrm{DLS}}-\theta_{\mathrm{D2o}}$ (B.281)	−0,006°			E: 114
Wheel difference angle between tool and surface at coast side fillet point	$\Delta\theta_{\mathrm{C2o}}=\theta_{\mathrm{CLS}}-\theta_{\mathrm{C2o}}$ (B.282)	0,005°			E: 115
Pinion difference angle between tool and surface at drive side fillet point	$\tan\Delta\theta_{\mathrm{D1o}}=-\frac{R_{\mathrm{DL2}}\sin\Delta\theta_{\mathrm{D2o}}}{r_{\mathrm{vn2}}+r_{\mathrm{vn1}}-R_{\mathrm{DL2}}\cos\Delta\theta_{\mathrm{D2o}}}$ (B.283)	tan(0,029°)			E: 116
Pinion difference angle between tool and surface at coast side fillet point	$\tan\Delta\theta_{\mathrm{C1o}}=-\frac{R_{\mathrm{CL2}}\sin\Delta\theta_{\mathrm{C2o}}}{r_{\mathrm{vn2}}+r_{\mathrm{vn1}}-R_{\mathrm{CL2}}\cos\Delta\theta_{\mathrm{C2o}}}$ (B.284)	tan(−0,025°)			E: 117
Pinion angle unbalance between fillet points	$\Delta\theta_{1}=\frac{\theta_{\mathrm{D1o}}+\theta_{\mathrm{C1o}}+\Delta\theta_{\mathrm{D1o}}+\Delta\theta_{\mathrm{C1o}}}{2{,}0}$ (B.285)	0,225°			E: 118

Table B.11 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Pinion angle from centreline to pinion tip	$\Delta r_1 = r_{vn1}\left(1{,}0 - e^{\theta_{Do}\tan\alpha_f}\right)$ (B.286) Formula shall be solved for θ_{Do}.	−5,444°			E: 119
Wheel angle from centreline to tooth surface at pitch point on drive side	θ_D (B.287)	Initial: −0,504			
		Final: −0,452			
	$\Delta r = r_{vn2}\left(e^{\theta_D\tan\alpha_f} - 1{,}0\right)$ (B.288)	Initial: 0,057			E: 120
		Final: 0,051			E: 120
	$h = \left(r_{vn2} + \Delta r\right)\sin\left(\alpha_{Dnf} + \theta_D\right) - \left(r_{vn2}\sin\alpha_{Dnf} - g_{rb}\right)$ (B.289)	Initial: −0,067			E: 121
		Final: 0,0			E: 121
Wheel angle from centreline to fillet point on drive flank	θ_{D2} (B.290)	Initial: −0,542			
		Final: −1,393			
	$\Delta r_2 = r_{vn2}\left(e^{\theta_{D2}\tan\alpha_f} - 1{,}0\right)$ (B.291)	Initial: 0,061			E: 123
		Final: 0,157			E: 123
	$h_2 = \left(r_{vn2} + \Delta r_2\right)\sin\left(\alpha_{Dnf} + \theta_{D2}\right) - \left(r_{vn2}\sin\alpha_{Dnf} - g_{rb}\right)$ (B.292)	Initial: −0,116			E: 124
		Final: −1,213			E: 124
	$h_{2o} = \pm\sqrt{r_{va1}^2 - \left(r_{vn1} + \Delta r_2\right)^2\cos^2\left(\alpha_{Dnf} + \theta_{D2}\right)} + \left(r_{vn1} + \Delta r_2\right)\sin\left(\alpha_{Dnf} + \theta_{D2}\right)$ (B.293)	Initial: −1,506			E: 125
		Final: −1,213			E: 125

Table B.11 — *continued*

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
Load sharing ratio for hypoid gears (pinion and wheel)						
Load sharing ratio for hypoid gears	ε_N	(B.294)	1,0			E: 126
Tooth strength factor for hypoid gears (pinion and wheel)						
Length of action from pinion tip to pitch circle in normal section	$g_{v\alpha 1} = \sqrt{h_1^2 + \left(\Delta r_1 - \Delta r\right)^2 - 2{,}0 h_1 \left(\Delta r_1 - \Delta r\right) \sin\left(\alpha_{Dnf} + \theta_{D1}\right)}$	(B.295)	2,171			E: 127
Length of action from wheel tip to pitch circle in normal section	$g_{v\alpha 2} = \sqrt{h_2^2 + \left(\Delta r_2 - \Delta r\right)^2 - 2{,}0 h_2 \left(\Delta r_2 - \Delta r\right) \sin\left(\alpha_{Dnf} + \theta_{D2}\right)}$	(B.296)	1,243			E: 128
Length of action in normal section	$g_{v\alpha n} = g_{v\alpha 1} + g_{v\alpha 2}$	(B.297)	3,414			E: 129
Profile contact ratio in mean normal section	$\varepsilon_{v\alpha n} = g_{v\alpha n} / p_{mn}$	(B.298)	0,997			E: 130
Modified contact ratio for hypoid gears	$\varepsilon_{v\gamma} = \sqrt{\varepsilon_{v\alpha n}^2 + \varepsilon_{v\beta}^2}$	(B.299)	2,882			E: 131
Profile load sharing factor	ε_f	(B.300)	0,0			E: 132b
Lengthwise load sharing factor	ε_b	(B.301)	2,882			E: 133b
Length of action from pinion tip to point of load application	$g_{v\alpha 3} = \left\lvert \frac{p_{mn}\,\varepsilon_{v\alpha n}^2}{\varepsilon_{v\gamma}^2} \left(\frac{0{,}5\varepsilon_{v\gamma}^2}{\varepsilon_{v\alpha n}} - \frac{\varepsilon_{v\beta}\varepsilon_b\,k'}{\varepsilon_{v\alpha}} + \varepsilon_f \right) - g_{v\alpha 1} \right\rvert$	(B.302)	1,413			E: 134

Table B.11 — *continued*

Description	Formula		Result	References to ISO 10300-1 (E: Formula)	References to ISO 10300-2 (T: Table)	References to ISO 10300-3 (F: Figure)
Length of action from wheel tip to point of load application	$g_{v\alpha 4} = \left\lvert \frac{p_{mn}\varepsilon_{v\alpha n}^2}{\varepsilon_{v\gamma}^2}\left(\frac{0{,}5\varepsilon_{v\gamma}^2}{\varepsilon_{v\alpha n}} + \frac{\varepsilon_{v\beta}\varepsilon_b k'}{\varepsilon_{v\alpha}} + \varepsilon_f\right) - g_{v\alpha 2}\right\rvert$	(B.303)	1,413			E: 135
Length of action to point of load application	$g_{J\,1,2} = g_{v\alpha n} - g_{v\alpha 3,4}$	(B.304)	2,001			E: 136
Wheel angle from pinion tip to point of load application	θ_{D3}	(B.305)	Initial: −0,755			
			Final: 0,625			
	$\Delta r_3 = r_{vn2}\left(e^{\theta_{D3}\tan\alpha_f} - 1{,}0\right)$	(B.306)	Initial: 0,086			E: 137
			Final: −0,07			E: 137
	$h_3 = \left(r_{vn2} + \Delta r_3\right)\sin\left(\alpha_{vDnf} + \theta_{D3}\right) - \left(r_{vn2}\sin\alpha_{vDnf} - g_{rb}\right)$	(B.307)	Initial: −0,39			E: 138
			Final: 1,375			E: 138
	$h_{3o} = \sqrt{g_{v\alpha 3}^2 - \left(\Delta r_3 - \Delta r\right)^2\cos^2\left(\alpha_{vDnf} + \theta_{D3}\right)} - \left\lvert \Delta r_3 - \Delta r\right\rvert \sin\left(\alpha_{vDnf} + \theta_{D3}\right)$	(B.308)	Initial: 1,406			E: 139
			Final: 1,375			E: 139
Pinion angle from wheel tip to point of load application	θ_{D4}	(B.309)	Initial: 0,504			
			Final: 0,625			
	$\Delta r_4 = r_{vn2}\left(e^{\theta_{D4}\tan\alpha_f} - 1{,}0\right)$	(B.310)	Initial: −0,057			E: 140
			Final: −0,07			E: 140
	$h_4 = \left(r_{vn2} + \Delta r_4\right)\sin\left(\alpha_{vDnf} + \theta_{D4}\right) - \left(r_{vn2}\sin\alpha_{vDnf} - g_{rb}\right)$	(B.311)	Initial: 1,221			E: 141
			Final: 1,378			E: 141

Table B.11 — *continued*

Description	Formula	Result	References to ISO 10300-1 E: Formula	ISO 10300-2 T: Table	ISO 10300-3 F: Figure
	$h_{4o} = \sqrt{g_{v\alpha 4}^2 - (\Delta r_4 - \Delta r)^2 \cos^2(\alpha_{vDnf} + \theta_{D4})} - \lvert \Delta r_4 - \Delta r \rvert \sin(\alpha_{vDnf} + \theta_{D4})$ (B.312)	Initial: 1,380			E: 142
		Final: 1,378			E: 142
Distance from pitch circle to point of load application	$\Delta r_{LN2} = \frac{(r_{vn2} + \Delta r_3)\cos(\alpha_{vDnf} + \theta_{D3})}{\cos\alpha_{LN2}} - r_{vn2}$ (B.313)	−0,592			E: 143
Angle between centre line and line from point of load application and fillet point on wheel	α_{2oo} (B.314)	Initial: 30,502°			
		Final: 48,438°			
Horizontal distance from centreline to critical fillet point	$s_{N2} = x_1 - \rho_{va02}\cos\alpha_{2oo}$ (B.315)	Initial: 0,720			E: 144
		Final: 0,779			E: 144
Vertical distance from pitch circle to critical fillet point	$y_2 = y_1 + \rho_{va02}\sin\alpha_{2oo}$ (B.316)	Initial: 0,865			E: 145
		Final: 0,937			E: 145
Wheel load height at weakest section	$h_{N2} = y_2 + \Delta r_{LN2}$ (B.317)	Initial: 0,274			E: 146
		Final: 0,345			E: 146
Auxiliary value	$h_{N2o} = \frac{s_{N2}}{2{,}0\tan\alpha_{2oo}}$ (B.318)	Initial: 0,611			E: 147
		Final: 0,345			E: 147
Wheel tooth strength factor	$x_{N2} = \frac{s_{N2}^2}{h_{N2}}$ (B.319)	1,756			E: 148

Table B.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Pinion angle from pitch point to point of load application	$\Delta r_4 = r_{vn1}\left(1{,}0 - e^{\theta_{D5}\tan\alpha_f}\right)$ (B.320) Formula shall be solved for θ_{D5}.	−2,830°			E: 149
Pinion pressure angle at point of load application	$\alpha_{LN1} = \alpha_{vnf} + \theta_{D4} - \theta_{D5} + \Delta\theta_1$ (B.321)	18,931°			E: 150
Pinion radial distance to point of load application	$r_{41o} = \dfrac{\left(r_{vn1} - \Delta r_4\right)\cos\left(\alpha_{Dnf} + \theta_{D4}\right)}{\cos\alpha_{LN1}}$ (B.322)	17,465			E: 151
Start of iteration with $\alpha_{Do} = \alpha_{nD} + \alpha_{nC}$	α_{Do} (B.323)	Initial: 40,0°			
		Final: 31,201°			
Wheel angle between centreline and pinion fillet (enclosed iteration)	θ_{D2oo} (B.324)	Initial (first pass): 0,755°			
		Final (first pass): 1,182°			
		Initial (last pass): 0,755°			
		Final (last pass): 1,418°			

Table B.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
	$\Delta r_5 = r_{vn2}\, e^{\theta_{D2oo} \tan\alpha_f}$	(B.325)	Initial (first pass): 77,632			E: 152
			Final (first pass): 77,583			
			Initial (last pass): 77,632			
			Final (last pass): 77,557			
Wheel angle between centreline and pinion fillet (enclosed iteration)	$\mu_{D2} = \dfrac{r_{vn2} + h_{vfm1} - \rho_{va01} - \Delta r_5 \cos\theta_{D2oo}}{\tan\alpha_{Do}}$	(B.326)	Initial (first pass): 0,612			E: 153
			Final (first pass): 0,681			
			Initial (last pass): 0,848			
			Final (last pass): 1,000			

Table B.11 — *continued*

Description	Formula	Result	References to ISO 10300-1 (E: Formula)	References to ISO 10300-2 (T: Table)	References to ISO 10300-3 (F: Figure)
	$\mu_D = \Delta r_5 \sin\theta_{D2oo}$ (B.327)	Initial (first pass): 1,024			E: 154
		Final (first pass): 1,601			
		Initial (last pass): 1,024			
		Final (last pass): 1,919			
Pinion angle between centreline and pinion fillet	$r_{vn2}\left(e^{\theta_{D2oo}\tan\alpha_f} - 1{,}0\right) = r_{vn1}\left(1{,}0 - e^{\theta_{D1oo}\tan\alpha_f}\right)$ (B.328) Formula shall be solved for θ_{D1oo}.	Initial: −5,347°			E: 155
		Final: −6,406°			
Wheel rotation through path of action	$\tan\theta_{L2o} = \dfrac{\mu_{D1} - \rho_{va01}\cos\alpha_{Do}}{r_{vn2} + h_{vfm1} - \rho_{va01} + \rho_{va01}\sin\alpha_{Do}}$ (B.329)	Initial: tan(0,562°)			E: 156
		Final: tan(0,55°)			
Wheel angle difference between path of action and tooth surface at pinion fillet	$\Delta\theta_{D2oo} = \theta_{L2o} - \theta_{D2oo}$ (B.330)	Initial: −0,620°			E: 157
		Final: −0,868°			

Table B.11 — *continued*

Description	Formula		Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Wheel radius to pinion fillet point	$r_{L2o} = \dfrac{r_{vn2} + h_{vfm1} - \rho_{va01} + \rho_{va01} \sin\alpha_{Do}}{\cos\theta_{L2o}}$	(B.331)	Initial: 78,269			E: 158
			Final: 78,245			
Pinion angle to fillet point	$\tan\Delta\theta_{D1oo} = -\dfrac{r_{L2o} \sin\Delta\theta_{D2oo}}{r_{vn2} + r_{vn1} - r_{L2o}\cos\Delta\theta_{D2oo}}$	(B.332)	Initial: tan(2,928°)			E: 159
			Final: tan(4,090°)			
Pinion radius to fillet point	$r_{L1o} = \dfrac{r_{vn2} + r_{vn1} - r_{L2o}\cos\Delta\theta_{D2oo}}{\cos\Delta\theta_{D1oo}}$	(B.333)	Initial: 16,579			E: 160
			Final: 16,629			
Pinion angle from centre line to fillet point	$\left(\Delta\theta_1 - \theta_{L1o}\right) = \Delta\theta_1 - \theta_{D1oo} - \Delta\theta_{D1oo}$	(B.334)	Initial: 2,643°			E: 161
			Final: 2,542°			
Angle between centre line and line from point of load application and fillet point on pinion	$\alpha_1 = \alpha_{Do} - \theta_{D2oo} + \theta_{D1oo}$	(B.335)	Initial: 33,471°			E: 162
			Final: 23,377°			
Horizontal distance to critical fillet point	$s_{N1} = r_{L1o} \sin\left(\Delta\theta_1 - \theta_{L1o}\right)$	(B.336)	Initial: 0,765			E: 163
			Final: 0,737			
Pinion load height at weakest section	$h_{N1} = r_{41o} - r_{L1o} \cos\left(\Delta\theta_1 - \theta_{L1o}\right)$	(B.337)	Initial: 0,904			E: 164
			Final: 0,853			

Table B.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Auxiliary value	$h_{\mathrm{N1o}} = \dfrac{s_{\mathrm{N1}}}{2{,}0\tan\alpha_1}$	(B.338)	Initial: 0,578			E: 165
			Final: 0,853			
Pinion tooth strength factor	$x_{\mathrm{N1}} = \dfrac{s_{\mathrm{N1}}^2}{h_{\mathrm{N1}}}$	(B.339)	0,638			E: 166
Tooth form factor for hypoid gears (pinion and wheel)						
Tooth form factor for pinion	$Y_1 = \dfrac{2}{3}\left[\dfrac{1}{\left(\dfrac{1}{x_{\mathrm{N1}}} - \dfrac{\tan\alpha_{\mathrm{LN1}}}{3\,s_{\mathrm{N1}}}\right)}\right]$	(B.340)	0,472			E: 167
Tooth form factor for wheel	$Y_2 = \dfrac{2}{3}\left[\dfrac{1}{\left(\dfrac{1}{x_{\mathrm{N2}}} - \dfrac{\tan\alpha_{\mathrm{LN2}}}{3\,s_{\mathrm{N2}}}\right)}\right]$	(B.341)	1,452			E: 167
Transverse radius to point of load application (pinion and wheel)						
Mean face width of pinion	$b_{\mathrm{k1}} = \dfrac{b_1 \varepsilon_{\mathrm{v\alpha n}} \varepsilon_{\mathrm{v\beta}}}{\varepsilon_{\mathrm{v\gamma}}^2}$	(B.342)	10,354			E: 168

Table B.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Mean face width of wheel	$b_{k2} = \frac{b_2 \varepsilon_{v\alpha n} \varepsilon_{v\beta}}{\varepsilon_{v\gamma}^2}$	(B.343)	9,734			E: 168
Contact shift due to load for pinion	$x_{oo1} = k' b_{k1} - \frac{b_1 \varepsilon_f \varepsilon_{v\beta}}{\varepsilon_{v\gamma}^2}$	(B.344)	1,611			E: 169
Contact shift due to load for wheel	$x_{oo2} = k' b_{k2} + \frac{b_2 \varepsilon_f \varepsilon_{v\beta}}{\varepsilon_{v\gamma}^2}$	(B.345)	1,514			E: 170
Transverse radius to point of load application for pinion	$r_{my01} = \frac{r_{mpt1}\left(x_{oo1} + R_{m2}\right)}{R_{m2}} + \left(r_{410} - r_{vn1}\right) m_{et2}$	(B.346)	28,711 mm			E: 171
Transverse radius to point of load application for wheel	$r_{my02} = \frac{r_{mpt2}\left(x_{oo2} + R_{m2}\right)}{R_{m2}} + \Delta r_{LN2} m_{et2}$	(B.347)	69,952 mm			E: 172
Tooth fillet radius at root diameter (pinion and wheel)						
Relative fillet radius at root of tooth on pinion	$r_{mf1} = \frac{\left(h_{vfm1} - \rho_{va01}\right)^2}{r_{vn1} + h_{vfm1} - \rho_{va01}} + \rho_{va01}$	(B.348)	0,208			E: 173

Table B.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Relative fillet radius at root of tooth on wheel	$r_{mf2} = \frac{\left(h_{vfm2} - \rho_{va02}\right)^2}{r_{vn2} + h_{vfm2} - \rho_{va02}} + \rho_{va02}$ (B.349)	0,303			E: 173
Stress concentration and correction factor (pinion and wheel)					
Stress concentration and stress correction factor on pinion	$Y_{f1} = L + \left(\frac{2s_{N1}}{r_{mf1}}\right)^M \left(\frac{2s_{N1}}{h_{N1}}\right)^O$ (B.350)	1,963			E: 174
Stress concentration and stress correction factor on wheel	$Y_{f2} = L + \left(\frac{2s_{N2}}{r_{mf2}}\right)^M \left(\frac{2s_{N2}}{h_{N2}}\right)^O$ (B.351)	2,721			E: 174
where	$L = 0{,}325\,454\,5 - 0{,}007\,272\,7\,\alpha_n$ (B.352)	0,196			E: 175
	$M = 0{,}331\,818\,2 - 0{,}009\,090\,9\,\alpha_n$ (B.353)	0,17			E: 176
	$O = 0{,}268\,181\,8 + 0{,}009\,090\,9\,\alpha_n$ (B.354)	0,43			E: 177
Inertia factor					
Inertia factor when $\varepsilon_{v\gamma} \geq 2{,}0$	Y_i (B.355)	1,0			E: 178b

Table B.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Calculated effective face width					
Projected length of the instantaneous contact line in the tooth lengthwise direction (pinion)	$g_{K1} = \frac{b_1\ g_{v\alpha n}\ g_{J1} \cos^2 \beta_{vb}}{g_\eta^2}$ (B.356)	3,353			E: 179
Projected length of the instantaneous contact line in the tooth lengthwise direction (wheel)	$g_{K2} = \frac{b_2\ g_{v\alpha n}\ g_{J2} \cos^2 \beta_{vb}}{g_\eta^2}$ (B.357)	3,152			E: 179
Toe increment on pinion	$\Delta b'_{i1} = \frac{b_1 - g_{K1}}{2\cos\beta_{m1}} - \frac{x_{oo1}}{\cos\beta_{m1}}$ (B.358)	19,706			E: 180
Toe increment on wheel	$\Delta b'_{i2} = \frac{b_2 - g_{K2}}{2\cos\beta_{m2}} - \frac{x_{oo2}}{\cos\beta_{m2}}$ (B.359)	15,241			E: 180
Heel increment on pinion	$\Delta b'_{e1} = \frac{b_1 - g_{K1}}{2\cos\beta_{m1}} - \frac{x_{oo1}}{\cos\beta_{m1}}$ (B.360)	24,72			E: 181
Heel increment on wheel	$\Delta b'_{e2} = \frac{b_2 - g_{K2}}{2\cos\beta_{m2}} - \frac{x_{oo2}}{\cos\beta_{m2}}$ (B.361)	19,117			E: 181
Calculated effective face width on pinion	$b_{ce1} = 25{,}4\ h_{N1} \cos\beta_{m1} \left[\arctan\left(\frac{\Delta b_{i1}}{25{,}4\ h_{Na1}} \right) + \arctan\left(\frac{\Delta b_{e1}}{25{,}4\ h_{Na1}} \right) \right] + g_{K1}$ (B.362)	25,486			E: 183

Table B.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Calculated effective face width on wheel	$b_{ce2} = 25{,}4\, h_{N2} \cos\beta_{m2} \left[\arctan\left(\frac{\Delta b_{i2}}{25{,}4\, h_{Na2}} \right) + \arctan\left(\frac{\Delta b_{e2}}{25{,}4\, h_{Na2}} \right) \right] + g_{K2}$	(B.363)	18,154			E: 183
Basic formula of geometry factor (pinion and wheel)						
Geometry factor for pinion	$Y_{J1} = \frac{Y_1}{Y_{f1} \cdot \varepsilon_N \cdot Y_i} \cdot \frac{r_{my01}}{r_{mpt1}} \cdot \frac{b_{ce1}}{b_1} \cdot \frac{m_{mt1}}{m_{et2}}$	(B.364)	0,209			E: 51
Geometry factor for wheel	$Y_{J2} = \frac{Y_2}{Y_{f2} \cdot \varepsilon_N \cdot Y_i} \cdot \frac{r_{my02}}{r_{mpt2}} \cdot \frac{b_{ce2}}{b_2} \cdot \frac{m_{mt2}}{m_{et2}}$	(B.365)	0,266			E: 51
Root stress adjustment factor for carburized case hardened steel	Y_A	(B.366)	1,075			E: 184
Relative surface condition factor						
Relative surface condition factor	$Y_{R\,rel\,T}$	(B.367)	1,0			E: 185
Relative notch sensitivity factor						
Relative notch sensitivity factor	$Y_{\delta\,rel\,T}$	(B.368)	1,0			E: 186a
K–factors						
Face load factor	$K_{F,\beta}$	(B.369)	1,556	E: 28		
Transverse load factor	$K_{F,\alpha}$	(B.370)	1,0	E: 36		

Table B.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Tooth root stress formula (pinion and wheel)					
Nominal tooth root stress on pinion	$\sigma_{F0\,1\text{-}B2} = \frac{F_{mt1}}{b_1} \cdot \frac{m_{mt1}}{m_{et2}^2} \cdot \frac{Y_{A1}}{Y_{J1}}$ (B.371)	377,9 N/mm²			E: 47
Tooth root stress on pinion	$\sigma_{F1\text{-}B2} = \sigma_{F0\,1\text{-}B2} K_A K_v K_{F\beta} K_{F\alpha}$ (B.372)	646,8 N/mm²			E: 44
Nominal tooth root stress on wheel	$\sigma_{F0\,2\text{-}B2} = \frac{F_{mt2}}{b_2} \cdot \frac{m_{mt2}}{m_{et2}^2} \cdot \frac{Y_{A2}}{Y_{J2}}$ (B.373)	316,6 N/mm²			E: 47
Tooth root stress on wheel	$\sigma_{F2\text{-}B2} = \sigma_{F0\,2\text{-}B2} K_A K_v K_{F\beta} K_{F\alpha}$ (B.374)	541,9 N/mm²			E: 44
Permissible tooth root stress (pinion and wheel)					
Permissible tooth root stress on pinion	$\sigma_{FP1\text{-}B2} = \sigma_{F\,lim} Y_{ST} Y_{NT} Y_{\delta\,rel\,T\text{-}B2} Y_{R\,rel\,T\text{-}B2} Y_X$ (B.375)	960 N/mm²			E: 49
Permissible tooth root stress on wheel	$\sigma_{FP2\text{-}B2} = \sigma_{F\,lim} Y_{ST} Y_{NT} Y_{\delta\,rel\,T\text{-}B2} Y_{R\,rel\,T\text{-}B2} Y_X$ (B.376)	960 N/mm²			E: 49
Calculated safety factors (pinion and wheel)					
Calculated safety factor for pinion	$S_{F1\text{-}B2} = \frac{\sigma_{FP1\text{-}B2}}{\sigma_{F1\text{-}B2}}$ (B.377)	1,484			E: 50
Calculated safety factor for wheel	$S_{F2\text{-}B2} = \frac{\sigma_{FP2\text{-}B2}}{\sigma_{F2\text{-}B2}}$ (B.378)	1,772			E: 50

Annex C
(informative)

Sample 3: Rating of a hypoid gear set according to Method B1 and Method B2

C.1 Initial data

Sample 3 is for a hypoid gear pair which uses Method 2 according to ISO 23509.

Table C.1 — Initial data for pitch cone parameters

Symbol	Description	Method 0	Method 1	Method 2	Method 3
Σ	shaft angle	X	X	90°	X
a	hypoid offset	0,0	X	31,75 mm	X
$z_{1,2}$	number of teeth	X	X	9/34	X
d_{m2}	mean pitch diameter of wheel	—	—	146,7 mm	—
d_{e2}	outer pitch diameter of wheel	X	X	—	X
b_2	wheel face width	X	X	26,0 mm	X
β_{m1}	mean spiral angle of pinion	—	X	—	—
β_{m2}	mean spiral angle of wheel	X	—	21,009°	X
r_{c0}	cutter radius	X	X	76,0 mm	X
z_0	number of blade groups (only face hobbing)	X	—	13	X

Table C.2 — Input data for tooth profile parameters

Data type I		Data type II	
Symbol	**Description**	**Symbol**	**Description**
α_{dD}	20°		
α_{dC}	20°		
$f_{\alpha lim}$	1		
x_{hm1}	—	c_{ham}	0,275
k_{hap}	—	k_d	2,000
k_{hfp}	—	k_c	0,125
x_{smn}	—	k_t W_{m2}	0,1 —
j_{et2}	0,2 mm		
θ_{a2}	0°		
θ_{f2}	0°		
ρ_{a01}	0,8 mm		
ρ_{a02}	1,2 mm		
$s_{pr1D,C}$	0 mm/0 mm		
$s_{pr2D,C}$	0 mm/0 mm		

Table C.3 and Table C.4 show geometric and operational data and text for explanation.

Table C.3 — Geometric data from calculation according to ISO 23509

Symbol	Description	Values	Symbol	Description	Value
$d_{m1,2}$	mean pitch diameter of pinion/wheel	51,258 mm/ 146,700 mm	ζ_{mp}	offset angle on pitch plane	23,969°
$h_{am1,2}$	mean addendum of pinion/wheel	5,840 mm/ 2,215 mm	ζ_R	pinion offset angle on root plane	21,647°
$h_{fm1,2}$	mean dedendum of pinion/wheel	3,222 mm/ 6,847 mm	$R_{e1,2}$	outer cone distance on pinion/wheel	76,755 mm/ 95,168 mm
$\alpha_{eD,C}$	effective pressure angle for drive side/coast side	20°/20°	$R_{m1,2}$	mean cone distance on pinion/wheel	61,186 mm/ 82,168 mm
$\alpha_{nD,C}$	generated pressure angle for drive side/coast side	15,868°/ 24,132°	$\delta_{1,2}$	pitch angle on pinion/wheel	24,763°/ 63,212°
α_{lim}	limit pressure angle	−4,132°	$\delta_{a1,2}$	face angle on pinion/wheel	24,765°/ 63,212°
m_{mn}	mean normal module	4,028 mm	$\delta_{f1,2}$	root angle on pinion/wheel	24,765°/ 63,212°
k_{hfp}	basic crown gear dedendum factor	1,25	$x_{sm1,2}$	thickness modification coefficient on pinion/wheel	0,04/−0,06
ζ_m	pinion offset angle on axial plane	21,647°	m_{et2}	outer transverse module	4,997 mm
$s_{mn1,2}$	mean normal circular tooth thickness of pinion/wheel	7,969 mm/ 4,524 mm			

Table C.4 — Operation parameters and additional considerations

Symbol	Description	Value
Additional data		
	wheel profile	generated
	roughing/finishing method	face hobbing/face milling
b_{2eff}	effective face width on wheel	$0{,}85 \cdot b_2$
	profile crowning	low
	verification of contact pattern	checked under light test load for each gear
	mounting conditions of pinion and wheel	one member cantilever-mounted
Operation parameters		
T_1	pinion torque	250 Nm
n_1	pinion rotational speed	4 500 min^{-1}
K_A	application factor	1,1
	active flank	drive
Material data for pinion and wheel (case hardened steel)		
$\sigma_{H\ lim}$	allowable stress number (contact)	1510 N/mm^2
$\sigma_{F\ lim}$	nominal stress number (bending)	500 N/mm^2
	surface hardness	same for pinion and wheel
Quality parameters		
R_z	flank roughness on pinion/wheel	3 µm/3 µm
R_z	tooth root roughness on pinion/wheel	10 µm/10 µm
f_{pt}	single pitch deviation on pinion/wheel	12 µm/25 µm
Lubrication parameters		
	oil type	ISO-VG-100
	oil temperature	90 °C

C.2 Calculation of Sample 3 according to Method B1

Table C.5 — Virtual cylindrical gears

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
Virtual cylindrical gears in transverse section						
Reference diameter on pinion	$d_{v1} = \dfrac{d_{m1}}{\cos\delta_1}$	(C.1)	56,449 mm	E: A.1		
Reference diameter on wheel	$d_{v2} = u^2\ d_{v1}$	(C.2)	325,500 mm	E: A.4		
Helix angle	$\beta_v = \dfrac{\beta_{m1} + \beta_{m2}}{2}$	(C.3)	33°	E: A.8		
Transverse pressure angle of virtual cylindrical gears	$\alpha_{vet} = \arctan\left(\tan\alpha_e / \cos\beta_v\right)$ since $\alpha_e = \alpha_{eD}$ for drive side	(C.4)	23,460°	E: A.10		
Transverse module	$m_{vt} = m_{mn} / \cos\beta_v$	(C.5)	4,803 mm	E: A.11		
Number of teeth on pinion	$z_{v1} = d_{v1} / m_{vt}$	(C.6)	11,753	E: A.12		
Number of teeth on wheel	$z_{v2} = d_{v2} / m_{vt}$	(C.7)	67,773	E: A.12		
Gear ratio	$u_v = z_{v2} / z_{v1}$	(C.8)	5,766	E: A.13		

Table C.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Auxiliary angle for virtual face width	$\theta_{mp} = \arctan\left(\sin\delta_2 \tan\zeta_m\right)$	(C.9)	19,508°	E: A.21		
Projected auxiliary angle for length of contact line	$\gamma' = \theta_{mp} - \zeta_{mp}/2$	(C.10)	7,524°	E: A.20		
Centre distance of virtual cylindrical gear pair	$a_v = \left(d_{v1} + d_{v2}\right) / 2$	(C.11)	190,975 mm	E: A.5		
Helix angle of virtual cylindrical gear at base circle	$\beta_{vb} = \arcsin\left(\sin\beta_v \cos\alpha_e\right)$ since $\alpha_e = \alpha_{eD}$ for drive side	(C.12)	30,783°	E: A.16		
Tip diameter on pinion	$d_{va1} = d_{v1} + 2\ h_{am1}$	(C.13)	68,129 mm	E: A.6		
Tip diameter on wheel	$d_{va2} = d_{v2} + 2\ h_{am2}$	(C.14)	329,930 mm	E: A.6		
Root diameter on pinion	$d_{vf1} = d_{v1} - 2\ h_{fm1}$	(C.15)	50,005 mm	E: A.7		
Root diameter on wheel	$d_{vf2} = d_{v2} - 2\ h_{fm2}$	(C.16)	311,806 mm	E: A.7		
Base diameter on pinion	$d_{vb1} = d_{v1} \cos\alpha_{vet}$	(C.17)	51,782 mm	E: A.9		
Base diameter on wheel	$d_{vb2} = d_{v2} \cos\alpha_{vet}$	(C.18)	298,594 mm	E: A.9		
Transverse base pitch	$p_{vet} = \pi\ m_{mn} \cos\alpha_{vet} / \cos\beta_v$	(C.19)	13,841 mm	E: A.17		

Table C.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Length of path of contact of virtual cylindrical gear in transverse section	$g_{v\alpha} = \frac{1}{2}\left[\left(\sqrt{d_{va1}^2 - d_{vb1}^2} - d_{v1}\sin\alpha_{vet}\right) + \left(\sqrt{d_{va2}^2 - d_{vb2}^2} - d_{v2}\sin\alpha_{vet}\right)\right]$	(C.20)	16,279 mm	E: A.18		
Transverse contact ratio	$\varepsilon_{v\alpha} = g_{v\alpha}/p_{vet}$	(C.21)	1,176	E: A.23		
Effective face width with $b_{2\,eff} = 0{,}85 \cdot b_2$	$b_{v\,eff} = \frac{\left[b_{2\,eff}/\cos\left(\zeta_{mp}/2\right) - g_{v\alpha}\cos\alpha_{vet}\tan\left(\zeta_{mp}/2\right)\right]}{1 - \tan\gamma'\tan\left(\zeta_{mp}/2\right)}$	(C.22)	19,983 mm	E: A.19		
Face width	$b_v = b_2 \frac{b_{v\,eff}}{b_{2\,eff}}$	(C.23)	23,509 mm	E: A.22		
Virtual cylindrical gears in normal section						
Number of pinion teeth of virtual cylindrical gears	$z_{vn1} = \frac{z_{v1}}{\cos^2\beta_{vb}\cos\beta_v}$	(C.24)	18,987	E: A.38		
Number of wheel teeth of virtual cylindrical gears	$z_{vn2} = u_v z_{vn1}$	(C.25)	109,487	E: A.39		
Reference diameter on pinion	$d_{vn1} = z_{vn1} m_{mn}$	(C.26)	76,481 mm	E: A.40		
Reference diameter on wheel	$d_{vn2} = z_{vn2} m_{mn}$	(C.27)	441,013 mm	E: A.40		

Table C.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Tip diameter on pinion	$d_{van1} = d_{vn1} + 2h_{am1}$	(C.28)	88,161 mm	E: A.41		
Tip diameter on wheel	$d_{van2} = d_{vn2} + 2h_{am2}$	(C.29)	445,443 mm	E: A.41		
Base diameter on pinion	$d_{vbn1} = d_{vn1} \cos\alpha_e$	(C.30)	71,868 mm	E: A.42		
Base diameter on wheel	$d_{vbn2} = d_{vn2} \cos\alpha_e$	(C.31)	414,417 mm	E: A.42		
Face contact ratio	$\varepsilon_{v\beta} = \frac{b_{v\,eff} \sin\beta_v}{\pi\, m_{mn}}$	(C.32)	0,860	E: A.24		
Virtual contact ratio	$\varepsilon_{v\gamma} = \varepsilon_{v\alpha} + \varepsilon_{v\beta}$	(C.33)	2,036	E: A.25		
Inclination angle of contact line	$\beta_B = \arctan\left(\tan\beta_v \sin\alpha_e\right)$	(C.34)	12,522°	E: A.36		
Radius of relative curvature in normal section at the mean point	$\rho_t = \left[\frac{1}{\cos\alpha_{nD}\left(\tan\alpha_{nD} - \tan\alpha_{lim}\right) + \tan\zeta_{mp}\tan\beta_B} \times \frac{\cos\beta_{m1}\cos\beta_{m2}}{\cos\zeta_{mp}} \times \left(\frac{1}{R_{m2}\tan\delta_2} + \frac{1}{R_{m1}\tan\delta_1}\right)\right]^{-1}$	(C.35)	14,703 mm	E: A.37a		
Radius of relative curvature vertical to the contact line	$\rho_{rel} = \left\|\rho_t\right\| \cos^2\beta_B$	(C.36)	14,012 mm	E: A.35		

Table C.6 — General influence factors

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Nominal tangential force of bevel gears	$F_{mt1} = \frac{2000\ T_1}{d_{m1}}$	(C.37)	9754,6 N	E: 1		
Nominal tangential force of virtual cylindrical gears	$F_{vmt} = F_{mt1} \frac{\cos\beta_v}{\cos\beta_{m1}}$	(C.38)	11 567,6 N	E: 2		
Nominal tangential speed at mean point of the pinion	$v_{mt1} = \frac{d_{m1}\ n_1}{19\,098}$	(C.39)	12,078 m/s	E: 5		
Nominal tangential speed at mean point of the wheel	$v_{mt2} = \frac{d_{m2}\ n_2}{19\,098}$	(C.40)	9,150 m/s	E: 5		
Correction factor for non-average conditions for $F_{vmt} K_A / b_{veff} \geq$ 100 N/mm	C_F	(C.41)	1,0	E: 12a		
Mean value of mesh stiffness per unit face width	$c_\gamma = c_{\gamma 0} C_F$	(C.42)	14,000 N/(mm·µm)	E: 11		
Single stiffness	$c' = c'_0\ C_F$	(C.43)	20,000 N/(mm·µm)	E: 17		
Max. single pitch deviation as given in Table C.4	f_{pt}	(C.44)	25 µm			

Table C.6 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Running–in allowance for case hardened and nitrided gears	$y_{\alpha} = 0{,}075\, f_{pt}$ (C.45)	1,875 µm	E: 43		
Effective pitch deviation with $y_p = y_{\alpha}$	$f_{p\,eff} = f_{pt} - y_p$ (C.46)	23,125 µm	E: 16		
Relative pinion mass per unit face width reduced to the line of action	$m_1^* = \frac{1}{8} \rho \pi \frac{d_{m1}^2}{\cos^2\left[(\alpha_{nD} + \alpha_{nC})/2\right]}$ (C.47) where ρ is the density of the gear material (for steel $\rho = 7{,}86 \cdot 10^{-6}$ kg/mm^3)	0,009 kg/mm	E: 13		
Relative wheel mass per unit face width reduced to the line of action	$m_2^* = \frac{1}{8} \rho \pi \frac{d_{m2}^2}{\cos^2\left[(\alpha_{nD} + \alpha_{nC})/2\right]}$ (C.48) where ρ is the density of the gear material (for steel $\rho = 7{,}86 \cdot 10^{-6}$ kg/mm^3)	0,075 kg/mm	E: 13		
Mass reduced to the line of action of the dynamically equivalent cylindrical gear pair	$m_{red} = \frac{m_1^* m_2^*}{m_1^* + m_2^*}$ (C.49)	0,008 kg/mm	E: 10		
Resonance speed of pinion	$n_{E1} = \frac{30 \times 10^3}{\pi\, z_1} \sqrt{\frac{c_{\gamma}}{m_{red}}}$ (C.50)	52449 min^{-1}	E: 9		

Table C.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Dimensionless reference speed	$N = \frac{n_1}{n_{E1}}$	(C.51)	0,086	E: 8		
For virtual contact ratio, $\varepsilon_{v\gamma} = 2{,}036 > 2$ as given in ISO 10300–1:2014, Table 3	$c_{v1,2} = c_{v1} + c_{v2}$	(C.52)	0,648	T: 3		
	c_{v3}	(C.53)	0,202	T: 3		
	c_{v4}	(C.54)	0,785	T: 3		
	$c_{v5,6}$	(C.55)	0,875	T: 3		
	c_{v7}	(C.56)	0,889	T: 3		
Constant for the dynamic factor with $K_A = 1{,}1$ as given in Table C.4	$K = \frac{b_v\, f_{p\,eff}\, c'}{F_{vmt} K_A} c_{v1,2} + c_{v3}$	(C.57)	0,589	E: 15		
Dynamic factor	$K_{v-B1} = N \cdot K + 1$	(C.58)	1,000	E: 14		
Determination of the length of contact lines						
For virtual contact ratio, $\varepsilon_{v\beta} = 0{,}860$, $0 < \varepsilon_{v\beta} < 1$	$f_t = -(p_{vet} - 0{,}5\, p_{vet}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta}) + p_{vet} \cos\beta_{vb}$	(C.59)	11,206 mm	T: A.2		
	$f_m = -(p_{vet} - 0{,}5\, p_{vet}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta})$	(C.60)	−0,686 mm	T: A.2		
	$f_r = -(p_{vet} - 0{,}5\, p_{vet}\, \varepsilon_{v\alpha}) \cos\beta_{vb} (1 - \varepsilon_{v\beta}) - p_{vet} \cos\beta_{vb}$	(C.61)	−12,577 mm	T: A.2		

Table C.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Maximum distances from middle contact line	$f_{\mathrm{maxB}} = \frac{1}{2}\left[g_{\mathrm{v}\alpha} + b_{\mathrm{v\,eff}}\left(\tan\gamma + \tan\beta_{\mathrm{vb}}\right)\right]\cos\beta_{\mathrm{vb}}$	(C.62)	13,342 mm	E: A.31		
	$f_{\mathrm{max0}} = \frac{1}{2}\left[g_{\mathrm{v}\alpha} - b_{\mathrm{v\,eff}}\left(\tan\gamma + \tan\beta_{\mathrm{vb}}\right)\right]\cos\beta_{\mathrm{vb}}$	(C.63)	0,643 mm	E: A.32		
	$f_{\mathrm{max}} = f_{\mathrm{maxB}}$ since $f_{\mathrm{maxB}} > f_{\mathrm{max0}}$	(C.64)	13,342 mm			
Theoretical length of contact line	$l_{\mathrm{b0}} = \sqrt{\left(x_1 - x_2\right)^2 + \left(y_1 - y_2\right)^2}$	(C.65)	23,182 mm	E: A.27		
Theoretical length of middle contact line calculated with $f = f_{\mathrm{m}}$ for contact stress as specified in ISO 10300‒2:2014, 6.1	$x_1 = \dfrac{f\cos\beta_{\mathrm{vb}} + \tan\beta_{\mathrm{vb}}\left(f\sin\beta_{\mathrm{vb}} + \dfrac{b_{\mathrm{v\,eff}}}{2}\right) + \dfrac{1}{2}\left(g_{\mathrm{v}\alpha} + b_{\mathrm{v\,eff}}\tan\gamma\right)}{\tan\gamma + \tan\beta_{\mathrm{vb}}}$	(C.66)	19,916 mm	E: A.28		

Table C.6 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Theoretical length of middle contact line calculated with $f = f_m$ for contact stress as specified in ISO 10300‑2:2014, 6.1	$$x_2 = \frac{f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \frac{b_{v\,eff}}{2}\right) - \frac{1}{2}\left(g_{v\alpha} - b_{v\,eff}\tan\gamma\right)}{\tan\gamma + \tan\beta_{vb}} \quad (C.67)$$ NOTE ISO 10300‑1:2014, Formula (A.29) is a misprint. The operator in the second parenthesis should be "–".	0,0 mm	E: A.29		
	$$y_1 = -x_1\tan\beta_{vb} + f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \frac{b_{v\,eff}}{2}\right) \quad (C.68)$$	−6,710 mm	E: A.30		
	$$y_2 = -x_2\tan\beta_{vb} + f\cos\beta_{vb} + \tan\beta_{vb}\left(f\sin\beta_{vb} + \frac{b_{v\,eff}}{2}\right) \quad (C.69)$$	5,154 mm	E: A.30		
Correction factor	$$C_{lb} = \sqrt{\left(1 - \left(\frac{f}{f_{max}}\right)^2\right)\left(1 - \sqrt{\frac{b_{v\,eff}}{b_v}}\right)^2} \quad (C.70)$$	0,078	E: A.34		

Table C.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Length of contact line	$l_{bm} = l_{b0}\left(1 - C_{lb}\right)$	(C.71)	21,375 mm	E: A.26		
Load sharing factor (pitting)						
Exponent for calculation of parabolic distribution of peak loads	e	(C.72)	3,0		T: 3	
Related peak load	$p^* = 1 - \left(\frac{\lvert f \rvert}{\lvert f_{max} \rvert}\right)^e$	(C.73)			E: 7 F: 2	
Related peak load at tip contact line	$p_t{}^* = 1 - \left(\frac{\lvert f_t \rvert}{\lvert f_{max} \rvert}\right)^e$	(C.74)	0,408		E: 7	
Related peak load at middle contact line	$p_m{}^* = 1 - \left(\frac{\lvert f_m \rvert}{\lvert f_{max} \rvert}\right)^e$	(C.75)	1,000		E: 7	
Related peak load at root contact line	$p_r{}^* = 1 - \left(\frac{\lvert f_r \rvert}{\lvert f_{max} \rvert}\right)^e$	(C.76)	0,162		E: 7	
Related area	$A^* = \frac{1}{4} p^* l_b \pi$	(C.77)			E: 8 F: 2	

Table C.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Related area at tip contact line	$A_t{}^* = \frac{1}{4} p_t{}^* l_{bt} \pi$	(C.78)	1,199 mm		E: 8	
Related area at middle contact line	$A_m{}^* = \frac{1}{4} p_m{}^* l_{bm} \pi$	(C.79)	16,786 mm		E: 8	
Related area at root contact line	$A_r{}^* = \frac{1}{4} p_r{}^* l_{br} \pi$	(C.80)	0,174 mm		E: 8	
Load sharing factor	$Z_{LS} = \sqrt{\frac{A^*_m}{A^*_t + A^*_m + A^*_r}}$	(C.81)	0,961		E: 10	
Face load factors (Calculation according to Method C)						
Load distribution factor	$K_{H\beta\text{-}C} = 1{,}5\, K_{H\beta\text{-be}}$	(C.82)	1,65	E: 27		
with	$K_{H\beta\text{-be}}$	(C.83)	1,1	T:4		
Load distribution factor	$K_{F\beta\text{-}C} = K_{H\beta\text{-}C} / K_{F0}$	(C.84)	1,633	E: 28		
with	K_{F0} since finishing method is face milling.	(C.85)	1,011	E: 29b		
Transverse load factors (Calculation according to Method B)						
Determinant tangential force at mid face width on the pitch cone	$F_{mtH} = F_{vmt} K_A K_v K_{H\beta}$	(C.86)	20995,2 N	E: 37		

Table C.6 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Transverse load factors for bevel gear with virtual contact ratio $\varepsilon_{v\gamma} = 2{,}036 > 2$	$K^*_{H\alpha} = K^*_{F\alpha} = 0{,}9 + 0{,}4\sqrt{\frac{2\left(\varepsilon_{v\gamma} - 1\right)}{\varepsilon_{v\gamma}}} \cdot \frac{c_\gamma\left(f_{pt} - y_\alpha\right)}{F_{mtH}/b_v}$ (C.87)	1,109	E: 38		
Relative hypoid offset	$a_{rel} = \frac{2\left\|a\right\|}{d_{m2}}$ (C.88)	0,433	E: 35		
Transverse load factors	$K_{H\alpha} = K_{F\alpha} = K^*_{H\alpha} - \frac{K^*_{H\alpha} - 1}{0{,}1} a_{rel}$ (C.89) since $K_{H\alpha} \geq 1{,}0$	1,000	E: 34		

Table C.7 — Calculation of surface durability (pitting)

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Z–factors					
Factors for calculation of mid–zone factor for $\varepsilon_{v\beta} = 0{,}860 < 1$	$F_1 = \varepsilon_{v\alpha}$ (C.90)	1,291		T: 2	
	$F_2 = \varepsilon_{v\alpha}$ (C.91)	1,061		T: 2	

Table C.7 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Mid–zone factor	$Z_{M\text{-}B}=\dfrac{\tan\alpha_{vet}}{\sqrt{\left[\sqrt{\left(\dfrac{d_{va1}}{d_{vb1}}\right)^2-1}-F_1\dfrac{\pi}{z_{v1}}\right]\cdot\left[\sqrt{\left(\dfrac{d_{va2}}{d_{vb2}}\right)^2-1}-F_2\dfrac{\pi}{z_{v2}}\right]}}$	(C.92)	0,937		E: 6	
Elasticity factor	$Z_E=\sqrt{\dfrac{1}{\pi\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)}}$	(C.93)	189,800 $(N/mm^2)^{1/2}$		E: 51	
Bevel gear factor	Z_K	(C.94)	0,85		E: 11	
Lubricant factor	$C_{ZL}=0{,}08\dfrac{\sigma_{H\,lim}-850}{350}+0{,}83$ using σ_{Hlim} = 1 200 N/mm^2, which is the upper allowed limit in above formula.	(C.95)	0,910		E: 54	
	$Z_L=C_{ZL}+\dfrac{4\left(1{,}0-C_{ZL}\right)}{\left(1{,}2+\dfrac{134}{\nu_{40}}\right)^2}$	(C.96)	0,966		E: 53	

Table C.7 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Speed factor	$C_{ZV} = 0{,}08 \dfrac{\sigma_{H\,lim} - 850}{350} + 0{,}85$ (C.97) using σ_{Hlim} = 1 200 N/mm², which is the upper allowed limit in above formula.	0,930		E: 56	
	$Z_v = C_{ZV} + \dfrac{2\left(1{,}0 - C_{ZV}\right)}{\sqrt{0{,}8 + \dfrac{32}{v_{mt2}}}}$ (C.98)	0,998		E: 55	
Roughness factor with the radius of relative curvature $\rho = \rho_{rel}$ = 14,013 mm for Method B1 (see ISO 10300−1:2014, Annex A)	$Rz_{10} = \dfrac{Rz_1 + Rz_2}{2} \cdot \sqrt[3]{\dfrac{10}{\rho}}$ (C.99)	2,68 µm		E: 57	
	$C_{ZR} = 0{,}12 + \dfrac{1\,000 - \sigma_{H\,lim}}{5\,000}$ (C.100) using σ_{Hlim} = 1 200 N/mm², which is the upper allowed limit in above formula.	0,080		E: 59	
	$Z_R = \left(\dfrac{3}{Rz_{10}}\right)^{C_{ZR}}$ (C.101)	1,009		E: 58	
Product of the lubricant influence factors	$Z_L Z_v Z_R$ (C.102)	0,972			

Table C.7 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Size factor	Z_X for Method B1 (see ISO 10300–1:2014, 6.5.1)	(C.103)	1,000			
Hypoid factor	Z_{Hyp}	(C.104)	0,829		E: 12	
Life factor for pinion	$Z_{NT,1}$	(C.105)	1,000		T: 4	
Life factor for wheel	$Z_{NT,2}$	(C.106)	1,000		T: 4	
Work hardening factor	$Z_W = 1{,}2 - \frac{HB - 130}{1\,700}$ Set Z_w = 1,0 because pinion and wheel with equal hardness.	(C.107)	1,000		E: 60	
Contact stress formula						
Nominal normal force of the virtual cylindrical gear at mean point P	$F_n = \frac{F_{mt1}}{\cos\alpha_{nD}\cos\beta_{m1}}$	(C.108)	14 339 N		E: 3	
Nominal value of the contact stress	$\sigma_{H0} = \sqrt{\frac{F_n}{l_{bm}\rho_{rel}}} Z_{M\text{-}B} Z_{LS} Z_E Z_K$	(C.109)	1 005,6 N/mm^2		E: 2	
Contact stress	$\sigma_H = \sigma_{H0}\sqrt{K_A K_v K_{H\beta} K_{H\alpha}}$	(C.110)	1 354,7 N/mm^2		E: 1	
Permissible contact stress	$\sigma_{HP} = \sigma_{H\,lim}\ Z_{NT} Z_X Z_L Z_v Z_R Z_W Z_{Hyp}$	(C.111)	1 216,7 N/mm^2		E: 4	
Calculated safety factor for contact stress (pitting) on pinion and wheel	$S_{H1,2} = \frac{\sigma_{HP1,2}}{\sigma_{H1,2}}$	(C.112)	0,898		E: 5	

Table C.8 — Calculation of tooth root strength for pinion

Description	Formula	Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Y–factors					
Load sharing factor	$Y_{LS} = Z_{LS}^2$ (C.113)	0,924			E: 35
Geometry values for pinion according to Tables C.2 and C.3	$h_{a0} = h_{fp} = k_{hfp}\, m_{mn}$ (C.114)	5,035 mm			
Parameters for pinion	$E_{1,D} = \left(\frac{\pi}{4} - x_{sm1}\right) m_{mn} - h_{a0}\tan\alpha_{eD} - \frac{\rho_{a01,D}\left(1 - \sin\alpha_{eD}\right) - s_{pr1,D}}{\cos\alpha_{eD}}$ (C.115)	0,610 mm			E: 7
	$E_{1,C} = \left(\frac{\pi}{4} - x_{sm1}\right) m_{mn} - h_{a0}\tan\alpha_{eC} - \frac{\rho_{a01,C}\left(1 - \sin\alpha_{eC}\right) - s_{pr1,C}}{\cos\alpha_{eC}}$ (C.116)	0,610 mm			E: 7
	$G_{1,D} = \frac{\rho_{a01,D}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm1}$ (C.117)	−0,601			E: 8
	$G_{1,C} = \frac{\rho_{a01,C}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm1}$ (C.118)	−0,601			E: 8
Parameters for pinion	$H_{1,D} = \frac{2}{z_{vn1,D}}\left(\frac{\pi}{2} - \frac{E_{1,D}}{m_{mn}}\right) - \frac{\pi}{3}$ (C.119)	−0,898			E: 9

Table C.8 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
	$H_{1,C} = \frac{2}{z_{vn1,C}}\left(\frac{\pi}{2} - \frac{E_{1,C}}{m_{mn}}\right) - \frac{\pi}{3}$ (C.120)	−0,898			E: 9
Iteration starting with $\vartheta = \pi/6$ until $(\vartheta_{new} - \vartheta) < 0{,}000\,001$	$\theta_{1,D,C} = \frac{2G_{1,D,C}}{z_{vn1,D,C}} \tan\theta_{1,D,C} - H_{1,D,C}$ (C.121)	Initial: 30,000°			E: 10
		Final: 47,476°			E: 10
Tooth root chordal thickness on drive side	$s_{Fn1,D} = m_{mn} z_{vn1,D} \sin\left(\frac{\pi}{3} - \theta_{1,D}\right) + m_{mn}\sqrt{3}\left(\frac{G_{1,D}}{\cos\theta_{1,D}} - \frac{\rho_{a01,D}}{m_{mn}}\right)$ (C.122)	8,992 mm			E: 11
Tooth root chordal thickness on coast side	$s_{Fn1,C} = m_{mn} z_{vn1,C} \sin\left(\frac{\pi}{3} - \theta_{1,C}\right) + m_{mn}\sqrt{3}\left(\frac{G_{1,C}}{\cos\theta_{1,C}} - \frac{\rho_{a01,C}}{m_{mn}}\right)$ (C.123)	8,992 mm			E: 11
Tooth root chord	$s_{Fn1} = 0{,}5 s_{Fn1,D} + 0{,}5 s_{Fn1,C}$ (C.124)	8,992 mm			E: 12
Fillet radius at contact point of 30° tangent on drive side	$\rho_{F1,D} = \rho_{a01,D} + \frac{2G_{1,D}^2 m_{mn}}{\cos\theta_{1,D}\left(z_{vn1,D}\cos^2\theta_{1,D} - 2G_{1,D}\right)}$ (C.125)	1,236 mm			E: 13
Normal pressure angle at tooth tip on drive side	$\alpha_{an1,D} = \arccos\left(\frac{d_{vbn1,D}}{d_{van1,D}}\right)$ (C.126)	35,393°			E: 16

Table C.8 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Auxiliary angle for tooth form and tooth correction factor on drive side	$\gamma_{a1,D} = \frac{1}{z_{vn1,D}}\left[\frac{\pi}{2} + 2\left(x_{hm1}\tan\alpha_{eD} + x_{sm1}\right)\right] + \mathrm{inv}\alpha_{eD} - \mathrm{inv}\,\alpha_{an1,D}$ (C.127)	1,509°			E: 17
Load application angle at tooth tip of virtual cylindrical gear on drive side	$\alpha_{Fan1,D} = \alpha_{an1,D} - \gamma_{a1,D}$ (C.128)	33,884°			E: 15
Bending moment arm on drive side	$h_{Fa1,D} = \frac{m_{mn}}{2}\left[\left(\cos\gamma_{a1,D} - \sin\gamma_{a1,D}\tan\alpha_{Fan1,D}\right)\frac{d_{van1,D}}{m_{mn}} - z_{vn1,D}\cos\left(\frac{\pi}{3} - \theta_{1,D}\right) - \frac{G_{1,D}}{\cos\theta_{1,D}} + \frac{\rho_{a01,D}}{m_{mn}}\right]$ (C.129)	8,147 mm			E: 14
Tooth form factor on drive side	$Y_{Fa1,D} = \frac{6\frac{h_{Fa1,D}}{m_{mn}}\cos\alpha_{Fan1,D}}{\left(\frac{s_{Fn1}}{m_{mn}}\right)^2\cos\alpha_{n1,D}}$ (C.130)	2,102			E: 6

Table C.8 — *continued*

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
Stress correction factor on drive side	$L_{a1,D} = \frac{s_{Fn1}}{h_{Fa1,D}}$	(C.140)	1,104			E: 25
	$q_{s1,D} = \frac{s_{Fn1}}{2\rho_{F1,D}}$	(C.141)	3,636			E: 26
	$Y_{Sa1,D} = \left(1{,}2 + 0{,}13\ L_{a1,D}\right)\ q_{s1,D}^{\left(\frac{1}{1{,}21\ +\ 2{,}3/L_{a1,D}}\right)}$	(C.142)	1,988			E: 24
Relative surface condition factor for R_z = 10 µm and through hardened and case hardened steels	$Y_{R\ rel\ T} = \frac{Y_R}{Y_{RT}} = 1{,}674 - 0{,}529\ \left(Rz + 1\right)^{1/10}$	(C.143	1,002			E: 39
Life factor for pinion	$Y_{NT,1}$	(C.144)	1,000			T: 2
Stress correction factor	Y_{ST}	(C.145	2,000			E: 4
Size factor	$Y_X = 1{,}05 - 0{,}01 m_{mn}$ Set $Y_X = 1{,}0$ since range is $0{,}8 \le Y_X \le 1{,}0$.	(C.146)	1,000			F: 7 E: 188
Bevel spiral angle factor using l_{bm} from Formula (C.71) for l_{bb}	$Y_{BS} = \frac{a_{BS}}{c_{BS}}\left(\frac{l_{bb}}{b_a} - 1{,}05 \cdot b_{BS}\right)^2 + 1$	(C.147)	1,014			E: 28

Table C.8 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
where	$b_a = b_v / \cos \beta_v$	(C.148)	28,031 mm			E: 32
	$l_{bb} = l_{bm} \dfrac{\cos\beta_{vb}}{\cos\beta_v}$	(C.149)	21,896 mm			E: 33
	$h = (h_{m1} + h_{m2}) / 2$	(C.150)	9,063 mm			E: 34
Relative notch sensitivity factor	$Y_{\delta\ \mathrm{rel\ T\ 1}} = \dfrac{1 + \sqrt{\rho' \chi_1^X}}{1 + \sqrt{\rho' \chi_T^X}}$	(C.151)	1,010			E: 42
Contact ratio factor, Y_ε, for $\varepsilon_{v\beta} = 0{,}860 \leq 1$	Y_ε	(C.152)	0,662			E: 27c
Tooth root stress formula for pinion						
Nominal tooth root stress	$\sigma_{F01} = \dfrac{F_{vmt}}{b_v\, m_{mn}} Y_{Fa} Y_{Sa} Y_\varepsilon Y_{BS} Y_{LS}$	(C.153)	316,7 N/mm^2			E: 2
Local tooth root stress	$\sigma_{F1} = \sigma_{F0} K_A K_v K_{F\beta} K_{F\alpha}$	(C.154)	568,8 N/mm^2			E: 1
Permissible tooth root stress	$\sigma_{FP1} = \sigma_{F\,lim} Y_{ST} Y_{NT} Y_{\delta\ \mathrm{rel\ T\ 1}} Y_{\mathrm{R\ rel\ T}} Y_X$	(C.155)	1011,5 N/mm^2			E: 4
Calculated safety for tooth root strength on pinion	$S_{F1} = \dfrac{\sigma_{FP1}}{\sigma_{F1}}$	(C.156)	1,778			E: 5

Table C.9 — Calculation of tooth root strength for wheel

Description	Formula	Result	References to ISO 10300–1	References to ISO 10300–2	References to ISO 10300–3
			E: Formula	T: Table	F: Figure
Y–factors different from pinion					
Geometry values for wheel according to Tables C.2 and C.3	$h_{a0} = h_{fp} = k_{hfp}\, m_{mn}$ (C.157)	5,035 mm			
Parameters for wheel	$E_{2,D} = \left(\frac{\pi}{4} - x_{sm2}\right) m_{mn} - h_{a0} \tan\alpha_{eD} - \frac{\rho_{a02,D}\left(1 - \sin\alpha_{eD}\right) - s_{pr2,D}}{\cos\alpha_{eD}}$ (C.158)	0,732 mm			E: 7
	$E_{2,C} = \left(\frac{\pi}{4} - x_{sm2}\right) m_{mn} - h_{a0} \tan\alpha_{eC} - \frac{\rho_{a02,C}\left(1 - \sin\alpha_{eC}\right) - s_{pr2,C}}{\cos\alpha_{eC}}$ (C.159)	0,732 mm			E: 7
	$G_{2,D} = \frac{\rho_{a02,D}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm2}$ (C.160)	−1,402			E: 8
	$G_{2,C} = \frac{\rho_{a02,C}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm2}$ (C.161)	−1,402			E: 8
Parameters for wheel	$H_{2,D} = \frac{2}{z_{vn2,D}}\left(\frac{\pi}{2} - \frac{E_{2,D}}{m_{mn}}\right) - \frac{\pi}{3}$ (C.162)	−1,022			E: 9
	$H_{2,C} = \frac{2}{z_{vn2,C}}\left(\frac{\pi}{2} - \frac{E_{2,C}}{m_{mn}}\right) - \frac{\pi}{3}$ (C.163)	−1,022			E: 9

Table C.9 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Iteration starting with $\theta = \pi/6$ until $(\theta_{new} - \theta)$ < 0,000 001	$\theta_{2,D,C} = \frac{2G_{2,D,C}}{z_{vn2,D,C}} \tan\theta_{2,D,C} - H_{2,D,C}$ (C.164)	Initial: 30,000°			E: 10
		Final: 56,342°			E: 10
Tooth root chordal thickness on drive side	$s_{Fn2,D} = m_{mn} z_{vn2,D} \sin\left(\frac{\pi}{3} - \theta_{2,D}\right) + m_{mn}\sqrt{3}\left(\frac{G_{2,D}}{\cos\theta_{2,D}} - \frac{\rho_{a02,D}}{m_{mn}}\right)$ (C.165)	8,406 mm			E: 11
Tooth root chordal thickness on coast side	$s_{Fn2,C} = m_{mn} z_{vn2,C} \sin\left(\frac{\pi}{3} - \theta_{2,C}\right) + m_{mn}\sqrt{3}\left(\frac{G_{2,C}}{\cos\theta_{2,C}} - \frac{\rho_{a02,C}}{m_{mn}}\right)$ (C.166)	8,406 mm			E: 11
Tooth root chord	$s_{Fn2} = 0{,}5 s_{Fn2,D} + 0{,}5 s_{Fn2,C}$ (C.167)	8,406 mm			E: 12
Fillet radius at contact point of 30° tangent	$\rho_{F2,D} = \rho_{a02,D} + \frac{2G_{2,D}^2 m_{mn}}{\cos\theta_{2,D}\left(z_{vn2,D}\cos^2\theta_{2,D} - 2G_{2,D}\right)}$ (C.168)	1,984 mm			E: 13
Normal pressure angle at tooth tip on drive side	$\alpha_{an2,D} = \arccos\left(\frac{d_{vbn2,D}}{d_{van2,D}}\right)$ (C.169)	21,511°			E: 16
Auxiliary angle for tooth form and tooth correction factor on drive side	$\gamma_{a2,D} = \frac{1}{z_{vn2,D}}\left[\frac{\pi}{2} + 2\left(x_{hm2} \tan\alpha_{eD} + x_{sm2}\right)\right] + \mathrm{inv}\,\alpha_{eD} - \mathrm{inv}\,\alpha_{an2,D}$ (C.170)	0,371°			E: 17

Table C.9 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Load application angle at tooth tip of virtual cylindrical gear on drive side	$\alpha_{\mathrm{Fan2,D}} = \alpha_{\mathrm{an2,D}} - \gamma_{\mathrm{a2,D}}$	(C.171)	21,140°			E: 15
Bending moment arm on drive side	$h_{\mathrm{Fa2,D}} = \frac{m_{\mathrm{mn}}}{2}\left[\left(\cos\gamma_{\mathrm{a2,D}} - \sin\gamma_{\mathrm{a2,D}}\tan\alpha_{\mathrm{Fan2,D}}\right)\frac{d_{\mathrm{van2,D}}}{m_{\mathrm{mn}}} - z_{\mathrm{vn2,D}}\cos\left(\frac{\pi}{3} - \theta_{\mathrm{2,D}}\right) - \frac{G_{\mathrm{2,D}}}{\cos\theta_{\mathrm{2,D}}} + \frac{\rho_{\mathrm{a02,D}}}{m_{\mathrm{mn}}}\right]$	(C.172)	7,797 mm			E: 14
Tooth form factor on drive side	$Y_{\mathrm{Fa2,D}} = \frac{6\frac{h_{\mathrm{Fa2,D}}}{m_{\mathrm{mn}}}\cos\alpha_{\mathrm{Fan2,D}}}{\left(\frac{s_{\mathrm{Fn2}}}{m_{\mathrm{mn}}}\right)^2\cos\alpha_{\mathrm{n2,D}}}$	(C.173)	2,586			E: 6

Table C.9 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
	$L_{a2,D} = \frac{s_{Fn2}}{h_{Fa2,D}}$	(C.174)	1,078			E: 25
Stress correction factor on drive side	$q_{s2,D} = \frac{s_{Fn2}}{2\rho_{F2,D}}$	(C.175)	2,118			E: 26
	$Y_{Sa2,D} = \left(1{,}2 + 0{,}13\ L_{a2,D}\right)\ q_{s2,D}^{\left(\frac{1}{1{,}21\ +\ 2{,}3/L_{a2,D}}\right)}$	(C.176)	1,677			E: 24
Relative surface condition factor for R_z = 10 µm for through hardened and case hardened steels	$Y_{R\ rel\ T} = \frac{Y_R}{Y_{RT}} = 1{,}674 - 0{,}529\ \left(Rz + 1\right)^{1/10}$	(C.177)	1,002			E: 39

Table C.9 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Bevel spiral angle factor using l_{bm} from Formula (C.71) for l_{bb}	$Y_{BS} = \frac{a_{BS}}{c_{BS}}\left(\frac{l_{bb}}{b_a} - 1{,}05 \cdot b_{BS}\right)^2 + 1$	(C.178)	1,014			E: 28
where	$b_a = b_v / \cos\beta_v$	(C.179)	28,031 mm			E: 32
	$l_{bb} = l_{bm}\frac{\cos\beta_{vb}}{\cos\beta_v}$	(C.180)	21,896 mm			E: 33
	$h = (h_{m1} + h_{m2}) / 2$	(C.181)	9,063 mm			E: 34
Relative notch sensitivity factor	$Y_{\delta\ rel\ T\ 2} = \frac{1 + \sqrt{\rho' \chi_2^X}}{1 + \sqrt{\rho' \chi_T^X}}$	(C.182)	0,996			E: 27c
Tooth root stress formula for wheel						
Nominal tooth root stress	$\sigma_{F02} = \frac{F_{vmt}}{b_v\, m_{mn}} Y_{Fa} Y_{Sa} Y_\varepsilon Y_{BS} Y_{LS}$	(C.183)	328,8 N/mm^2			E: 2
Local tooth root stress	$\sigma_{F2} = \sigma_{F0} K_A K_v K_{F\beta} K_{F\alpha}$	(C.184)	590,5 N/mm^2			E: 1
Permissible tooth root stress	$\sigma_{FP2} = \sigma_{F\,lim} Y_{ST} Y_{NT} Y_{\delta\ rel\ T\ 2} Y_{R\ rel\ T} Y_X$	(C.185)	997,9 N/mm^2			E: 4

Table C.9 — *continued*

Description	Formula	Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Calculated safety for tooth root strength on wheel	$S_{F2} = \frac{\sigma_{FP2}}{\sigma_{F2}}$ (C.186)	1,690			E: 5

C.3 Calculation of Sample 3 according to Method B2

Table C.10 — Calculation of surface durability (pitting)

Description	Formula	Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Approximate values for application factors					
Relative face width	$b_v = b_2 / m_{et2}$ (C.187)	5,203	E: B.1		
Relative mean back cone distance on pinion	$R_{mpt1} = \frac{R_{m1} \tan\delta_1}{m_{et2}}$ (C.188)	5,648	E: B.2		
Relative mean back cone distance on wheel	$R_{mpt2} = \frac{R_{m2} \tan\delta_2}{m_{et2}}$ (C.189)	32,570	E: B.2		
Angle between direction of contact and the pitch tangent	$\cot\left(\zeta_R - \lambda\right) = \cot\zeta_R \left(1{,}0 + \frac{z_1 \cos\delta_{f2}}{z_2 \cos\delta_{a1} \cos\zeta_R}\right)$ (C.190)	cot(19,174°)	E: B.3		

Table C.10 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Face contact ratio for hypoid gears	$\varepsilon_{\mathrm{v\beta}} = \left[\frac{\cos\beta_{\mathrm{m2}}}{\cot\left(\zeta_{\mathrm{R}} - \lambda\right)} + \sin\beta_{\mathrm{m2}} \right] \frac{b_2}{\pi\, m_{\mathrm{mn}}}$ (C.191)	1,404	E: B.4b		
Relative mean virtual pitch radius on pinion	$r_{\mathrm{vn1}} = \frac{R_{\mathrm{mpt1}}}{\cos^2\beta_{\mathrm{m1}}}$ (C.192)	11,293	E: B.5		
Relative mean virtual pitch radius on wheel	$r_{\mathrm{vn2}} = \frac{R_{\mathrm{mpt2}}}{\cos^2\beta_{\mathrm{m2}}}$ (C.193)	37,373	E: B.5		
Relative centre distance	$a_{\mathrm{vn}} = r_{\mathrm{vn1}} + r_{\mathrm{vn2}}$ (C.194)	48,666	E: B.6		
Relative mean virtual dedendum on pinion	$h_{\mathrm{vfm1}} = h_{\mathrm{fm1}} / m_{\mathrm{et2}}$ (C.195)	0,645	E: B.7		
Relative mean virtual dedendum on wheel	$h_{\mathrm{vfm2}} = h_{\mathrm{fm2}} / m_{\mathrm{et2}}$ (C.196)	1,370	E: B.7		
Relative virtual tooth thickness on pinion	$s_{\mathrm{vmn1}} = s_{\mathrm{mn1}} / m_{\mathrm{et2}}$ (C.197)	1,595	E: B.8		
Relative virtual tooth thickness on wheel	$s_{\mathrm{vmn2}} = s_{\mathrm{mn2}} / m_{\mathrm{et2}}$ (C.198)	0,905	E: B.8		
Relative mean virtual tip radius on pinion	$r_{\mathrm{va1}} = r_{\mathrm{vn1}} + h_{\mathrm{am1}} / m_{\mathrm{et2}}$ (C.199)	12,462	E: B.9		
Relative mean virtual tip radius on wheel	$r_{\mathrm{va2}} = r_{\mathrm{vn2}} + h_{\mathrm{am2}} / m_{\mathrm{et2}}$ (C.200)	37,817	E: B.9		

Table C.10 — *continued*

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
Angular pitch of virtual cylindrical wheel (required in ISO 10300–3:2014, 7.4.5)	$\theta_{v2} = \frac{\pi\, m_{mn}}{m_{et2} r_{vn2}}$	(C.201)	3,882°	E: B.10		
Relative edge radius of tool on pinion	$\rho_{va01} = \rho_{a01} / m_{et2}$	(C.202)	0,160	E: B.11		
Relative edge radius of tool on wheel	$\rho_{va02} = \rho_{a02} / m_{et2}$	(C.203)	0,240	E: B.11		
Virtual spiral angle for hypoid gears	$\beta_v = \beta_{m1} - \lambda_r$	(C.204)	42,261°	E: B.12b		
where	$\tan\left(\beta_{m1} - \lambda_r\right) = \frac{R_{m2} \tan\delta_{f2} \tan\beta_{m1} + R_{m1} \tan\delta_{a1} \tan\beta_{m2}}{R_{m2} \tan\delta_{f2} + R_{m1} \tan\delta_{a1}}$	(C.205)	tan(42,261°)	E: B.13		
Adjusted pressure angle	$\alpha_a = \alpha_{eD} - 90^\circ \cos\delta_2 \cos\beta_{m2} / z_2$	(C.206)	18,591°	E: B.14		
Base virtual helix angle	$\sin\beta_{vb} = \sin\beta_v \cos\alpha_a$	(C.207)	sin(39,599°)	E: B.15		
Relative mean virtual base radius on pinion	$r_{vbn1} = r_{vn1} \cos\alpha_a$	(C.208)	10,703	E: B.16		
Relative mean virtual base radius on wheel	$r_{vbn2} = r_{vn2} \cos\alpha_a$	(C.209)	35,423	E: B.16		
Relative length of action from pinion tip to pitch circle in the normal section	$g_{v\alpha na} = \sqrt{r_{va1}^2 - r_{vbn1}^2} - r_{vn1} \sin\alpha_a$	(C.210)	2,781	E: B.17		

Table C.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Relative length of action from wheel tip to pitch circle in the normal section	$g_{\mathrm{v\alpha nr}} = \sqrt{r_{\mathrm{va2}}^2 - r_{\mathrm{vbn2}}^2} - r_{\mathrm{vn2}} \sin\alpha_{\mathrm{a}}$	(C.211)	1,325	E: B.18		
Relative length of action in normal section	$g_{\mathrm{v\alpha n}} = g_{\mathrm{v\alpha na}} + g_{\mathrm{v\alpha nr}}$	(C.212)	4,106	E: B.19		
Relative mean normal pitch of virtual cylindrical gear	$p_{\mathrm{mn}} = \dfrac{2{,}0\pi\, m_{\mathrm{mn}}}{m_{\mathrm{et2}} \cos\alpha_{\mathrm{a}} \left(\cos^2\beta_{\mathrm{m1}} + \cos^2\beta_{\mathrm{m2}} + 2{,}0\tan^2\alpha_{\mathrm{a}}\right)}$	(C.213)	3,344	E: B.20		
Profile contact ratio in mean normal section	$\varepsilon_{\mathrm{v\alpha n}} = \dfrac{g_{\mathrm{v\alpha n}}}{p_{\mathrm{mn}}}$	(C.214)	1,228	E: B.21		
Profile contact ratio in mean transverse section	$\varepsilon_{\mathrm{v\alpha}} = \varepsilon_{\mathrm{v\alpha n}} \cos^2\beta_{\mathrm{vb}}$	(C.215)	0,729	E: B.22		
Modified contact ratio for bevel gears without hypoid offset	$\varepsilon_{\mathrm{v\gamma}} = \sqrt{\varepsilon_{\mathrm{v\alpha}}^2 + \varepsilon_{\mathrm{v\beta}}^2}$	(C.216)	1,582	E: B.23		
Contact shift factor; see also ISO 10300–2:2014, Figure B.7	$k' = \dfrac{z_2 - z_1}{3{,}2\, z_2 + 4{,}0\, z_1}$	(C.217)	0,173	E: B.24		

Table C.10 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Pitting resistance geometry factor					
Angle between contact direction and tooth tangent in pitch plane	$\cot\left(\beta_{m1} - \lambda_1\right) = \frac{\cos\zeta_R}{\cos\beta_{m1}\ \cos\beta_{m2}\ \tan\left(\beta_{m1} - \lambda_r\right)} - \tan\beta_{m2}$ (C.218)	cot(40,635°)		E: 24	
Angle between projection of pinion axis and direction of contact in pitch plane	$\lambda_1 = \beta_{m1} - \left(\beta_{m1} - \lambda_1\right)$ (C.219)	4,355°		E: 25	
Angle of contact line relative to the root cone	$\tan w = \frac{\sin\alpha_a\ \tan\left(\beta_{m1} - \lambda_r\right)}{\cos\alpha_{lim}}$ (C.220)	tan(16,196°)		E: 26	
Mean base spiral angle	$\cos\beta_{bm} = \frac{1{,}0}{\sqrt{\tan^2\left(\beta_{m1} - \lambda_r\right)\cos^2\alpha_a + 1{,}0}}$ (C.221)	cos(40,737°)		E: 27	
Relative mean normal base pitch	$p_{nb} = \frac{\pi\ m_{mn}\cos\alpha_a\cos\beta_{bm}}{m_{et2}\cos\left(\beta_{m1} - \lambda_r\right)}$ (C.222)	2,457		E: 28	

Table C.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Angle between projection of wheel axis and direction of contact in pitch plane	$\lambda_2 = \left(\beta_{m1} - \lambda_r\right) - \beta_{m2}$	(C.223)	21,252°		E: 29	
Relative base face width	$b_b = \dfrac{b_2}{m_{et2}\cos\lambda_2}$	(C.224)	5,583		E: 30	
Pressure angle at point of load application on pinion	$\cos\alpha_{L1} = \cos\alpha_a \left[1{,}0 - \dfrac{\left(r_{va1} - r_{vn1}\right)\cos^2\beta_{m1}}{r_{va1} - r_{vn1} + R_{mpt1}}\right]$	(C.225)	cos(29,941°)		E: 31	
Pressure angle at point of load application on wheel	$\cos\alpha_{L2} = \cos\alpha_a \left[1{,}0 - \dfrac{\left(r_{va2} - r_{vn2}\right)\cos^2\beta_{m2}}{r_{va2} - r_{vn2} + R_{mpt2}}\right]$	(C.226)	cos(20,491°)		E: 31	

Table C.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Radius of curvature difference between point of load application and mean point on pinion	$\rho_{\Delta 1} = \frac{r_{va1} - r_{vn1} + R_{mpt1}}{\cos^2 \beta_{m1}} \cos\alpha_{L1} \left(\tan\alpha_{L1} - \tan\alpha_a\right)$	(C.227)	2,830		E: 32	
Radius of curvature difference between point of load application and mean point on wheel	$\rho_{\Delta 2} = \frac{r_{va2} - r_{vn2} + R_{mpt2}}{\cos^2 \beta_{m2}} \cos\alpha_{L2} \left(\tan\alpha_{L2} - \tan\alpha_a\right)$	(C.228)	1,325		E: 32	
Radius of curvature change	$\rho_{\Delta red} = \cos\beta_{bm} \left(\rho_{\Delta 1} + \rho_{\Delta 2}\right)$	(C.229)	3,148		E: 33	
Relative length of action within the contact ellipse	$g_\eta = \sqrt{\rho^2_{\Delta red} \cos^2 \beta_{bm} + b_b^2 \sin^2 \beta_{bm}}$	(C.230)	4,355		E: 34	
Radius of relative profile curvature and load sharing ratio at critical point						
Length of action at critical point in contact ellipse (iteration starting with $y_I = 0$)	$g_{\eta I} = \sqrt{g_\eta^2 - 4{,}0\, y_I^2}$	(C.231)	Initial: 4,355		E: 36	
			Final: 4,315		E: 36	

Table C.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Length of action considering adjacent teeth	$g_{\eta I\Sigma}^3 = g_{\eta I}^3 + \sqrt{\left[g_{\eta I}^2 - 4{,}0p_{nb}\left(p_{nb} + 2{,}0y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 4{,}0p_{nb}\left(p_{nb} - 2{,}0y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 8{,}0p_{nb}\left(2{,}0p_{nb} + 2{,}0y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 8{,}0p_{nb}\left(2{,}0p_{nb} - 2{,}0y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 16{,}0p_{nb}\left(4{,}0p_{nb} + 2{,}0y_I\right)\right]^3} + \sqrt{\left[g_{\eta I}^2 - 16{,}0p_{nb}\left(4{,}0p_{nb} - 2{,}0y_I\right)\right]^3}$	(C.232)	Initial: 82,579		E: 37	
			Final: 80,448		E: 37	
Load sharing ratio	$\varepsilon_{NI} = g_{\eta I}^3 \Big/ g_{\eta I\Sigma}^3$	(C.233)	Initial: 1,000		E: 38	
			Final: 0,998		E: 38	
Length of contact line	$g_c = g_{\eta I}\ \rho_{\Delta red}\ y_I \Big/ g_\eta^2$	(C.234)	Initial: 4,036		E: 39	
			Final: 3,999		E: 39	
Position change along path of contact	$g_{\eta\Delta} = \dfrac{\rho_{\Delta red}^2 y_I}{g_{\eta I}^2} + k' g_c \tan\beta_{bm} + \dfrac{0{,}5\rho_{\Delta red}}{\cos\beta_{bm}} - \rho_{\Delta 2}$	(C.235)	Initial: 1,353		E: 40	
			Final: 1,190		E: 40	
Intermediate value	$X = \dfrac{\sin^2 w\ \cos\alpha_{lim}\cos\left(\zeta_R - \lambda_1\right)\cos\lambda_1}{\sin^2\left(\beta_{m1} - \lambda_1\right)\sin\alpha_a\cos\zeta_R}$	(C.236)	Initial: 0,588		E: 41	
			Final: 0,588		E: 41	
Profile radius of curvature on pinion	$\rho_1 = R_{mpt1}\ X \pm g_{\eta\Delta}$	(C.237)	Initial: 4,673		E: 42	
			Final: 4,510		E: 42	
Profile radius of curvature on wheel	$\rho_2 = R_{mpt2}\ X \pm g_{\eta\Delta}$	(C.238)	Initial: 17,793		E: 42	
			Final: 17,956		E: 42	

Table C.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Relative radius of profile curvature between pinion and wheel	$\rho_t = \dfrac{\rho_1 \rho_2}{\rho_1 + \rho_2}$	(C.239)	Initial: 3,701		E: 43	
			Final: 3,605		E: 43	
Inertia factor with $\varepsilon_v > 2{,}0_\gamma$	Z_i	(C.240)	Initial: 1,265		E: 44b	
			Final: 1,265		E: 44b	
Pitting resistance geometry factor	$Z_I = \dfrac{g_c\, \rho_t \cos\alpha_a\, m_{mn}}{b_b z_1 Z_i \varepsilon_{NI} m_{et2}}$	(C.241)	Initial: 0,180		E: 45	
			Final: 0,174		E: 45	
Face width factor						
for $12{,}7 \le b_2 \le 79{,}8$ mm	$Z_{FW} = 0{,}00492 b_2 + 0{,}4375$	(C.242)	0,565		E: 46b	
Contact stress adjustment factor						
for carburized case hardened steel	Z_A	(C.243)	0,967		E: 47	
Elasticity factor						
for a steel on steel gear pair	Z_E	(C.244)	189,8 $(N/mm^2)^{1/2}$		E: 51	
Lubricant film influence factors						
Product of the lubricant influence factors	$Z_L Z_v Z_R$	(C.245)	0,972			

Table C.10 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Work hardening factor						
Work hardening factor	Z_W Set Z_w = 1,0 because pinion and wheel with equal hardness.	(C.246)	1,000			
Life factor						
Life factor	Z_{NT}	(C.247)	1,000		T: 4	
K–factors						
Dynamic factor	K_V	(C.248)	1,000	E: 14		
Face load factor	$K_{H,\beta}$	(C.249)	1,650	E: 27		
Contact stress formula for drive and coast side						
Nominal contact stress	$\sigma_{H0\text{-}B2} = \sqrt{\frac{F_{mt1} d_{m1} Z_{FW}}{b_2 Z_I} \left(\frac{z_2}{d_{e2} z_1}\right)^2} \cdot Z_E$	(C.250)	1 056,2 N/mm²		E: 21	
Contact stress	$\sigma_{H\text{-}B2} = \sigma_{H0\text{-}B2} \sqrt{K_A K_v K_{H\beta}} \cdot Z_A \le \sigma_{HP\text{-}B2}$	(C.251)	1 375,8 N/mm²		E: 20	
Permissible contact stress	$\sigma_{HP\text{-}B2} = \sigma_{H\,lim} Z_{NT} Z_L Z_v Z_R Z_W$	(C.252)	1 467,9 N/mm²		E: 22	
Calculated safety factor for contact stress for pinion and wheel	$S_{H\text{-}B2} = \frac{\sigma_{HP\text{-}B2}}{\sigma_{H\text{-}B2}} > S_{H\,min}$	(C.253)	1,067		E: 23	

Table C.11 — Calculation of tooth root strength on pinion and wheel

Description	Formula		Result	References to		
				ISO 10300‑1	ISO 10300‑2	ISO 10300‑3
				E: Formula	T: Table	F: Figure
Geometry factor for hypoid gears (pinion and wheel)						
Initial formulas						
Drive flank pressure angle in wheel root coordinates	$\alpha_{\mathrm{Dnf}} = \alpha_{\mathrm{nD}} - \theta_{\mathrm{f2}} \sin\beta_{\mathrm{m2}}$	(C.254)	15,868°			E: 84
Coast flank pressure angle in wheel root coordinates	$\alpha_{\mathrm{Cnf}} = \alpha_{\mathrm{nC}} + \theta_{\mathrm{f2}} \sin\beta_{\mathrm{m2}}$	(C.255)	24,132°			E: 85
Average pressure angle unbalance	$\Delta\alpha_1 = \dfrac{\left(\alpha_{\mathrm{Dnf}} - \alpha_{\mathrm{Cnf}}\right)}{2{,}0}$	(C.256)	−4,132°			E: 86
Limit pressure angle in wheel root coordinates	$\alpha_{\mathrm{f}} = \alpha_{\mathrm{lim}} - \theta_{\mathrm{f2}} \sin\beta_{\mathrm{m2}}$	(C.257)	−4,132°			E: 87
Relative distance from blade edge to centreline	$g_{\mathrm{rb}} = \dfrac{\left(h_{\mathrm{fm2}} \tan\dfrac{\alpha_{\mathrm{nD}} + \alpha_{\mathrm{nC}}}{2{,}0} + \dfrac{W_{\mathrm{m2}}}{2{,}0}\right) \cos\dfrac{\alpha_{\mathrm{nD}} + \alpha_{\mathrm{nC}}}{2{,}0}}{m_{\mathrm{et2}}}$	(C.258)	0,755			E: 88
Intermediate value	$\eta_{\mathrm{D}} = \tan\alpha_{\mathrm{Dnf}} \left(\dfrac{g_{\mathrm{rb}}}{\sin\alpha_{\mathrm{Dnf}}} - h_{\mathrm{vfm2}}\right)$	(C.259)	0,395			E: 89
Intermediate value	$\eta_{\mathrm{C}} = \tan\alpha_{\mathrm{Cnf}} \left(\dfrac{g_{\mathrm{rb}}}{\sin\alpha_{\mathrm{Cnf}}} - h_{\mathrm{vfm2}}\right)$	(C.260)	−0,213			E: 90

Table C.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Intermediate angle	$\tan\beta_a = \dfrac{\dfrac{W_{m2}}{2{,}0 m_{et2}} - \rho_{va02}\left(\sec\dfrac{\alpha_{nD}+\alpha_{nC}}{2{,}0} - \tan\dfrac{\alpha_{nD}+\alpha_{nC}}{2{,}0}\right)}{h_{vfm2} - \rho_{va02}}$ (C.261)	tan(6,879°)			E: 91
Intermediate angle	$(\beta_D - \Delta\alpha) = \beta_a - \Delta\alpha_1$ (C.262)	11,011°			E: 92
Intermediate angle	$(\beta_C - \Delta\alpha) = -\beta_a - \Delta\alpha_1$ (C.263)	−2,747°			E: 93
Intermediate value	$g_1 = \dfrac{h_{vfm2} - \rho_{va02}}{\cos\beta_a}$ (C.264)	1,138			E: 94
Wheel angle between centreline and fillet point on drive side	$\tan\Delta\theta_D = \dfrac{g_1\sin(\beta_D - \Delta\alpha)}{r_{vn2} - g_1\cos(\beta_D - \Delta\alpha)}$ (C.265)	tan(0,344°)			E: 95
Wheel angle between centreline and fillet point on coast side	$\tan\Delta\theta_C = \dfrac{g_1\sin(\beta_C - \Delta\alpha)}{r_{vn2} - g_1\cos(\beta_C - \Delta\alpha)}$ (C.266)	tan(−0,086°)			E: 96
Wheel angle between fillet points	$\Delta\theta_2 = \dfrac{\theta_{v2} + \Delta\theta_D + \Delta\theta_C}{2{,}0}$ (C.267)	2,070°			E: 97
Vertical distance from pitch circle to fillet point	$y_1 = r_{vn2} - \dfrac{\left[r_{vn2} - g_1\cos(\beta_D - \alpha_f)\right]\cos(\Delta\theta_2 - \Delta\theta_D)}{\cos\Delta\theta_D}$ (C.268)	1,133			E: 98

Table C.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Horizontal distance from centreline to fillet point	$x_1 = \dfrac{\left[r_{vn2} - g_1 \cos\left(\beta_D - \alpha_f\right)\right] \sin\left(\Delta\theta_2 - \Delta\theta_D\right)}{\cos\Delta\theta_D}$ (C.269)	1,092			E: 99
Generated pressure angle of wheel at fillet point	$\alpha_{LN2} = \alpha_{Dnf} - \Delta\theta_2$ (C.270)	13,798°			E: 100
Distance from centreline to tool critical drive side fillet point	$\mu_{1D} = \eta_D + \tan\alpha_{Dnf}\left(h_{vfm1} + h_{vfm2}\right) + \rho_{va01}\left(\sec\alpha_{Dnf} - \tan\alpha_{Dnf}\right)$ (C.271)	1,089			E: 101
Distance from centreline to tool critical coast side fillet point	$\mu_{1C} = \eta_C + \tan\alpha_{Cnf}\left(h_{vfm1} + h_{vfm2}\right) - \rho_{va01}\left(\sec\alpha_{Cnf} - \tan\alpha_{Cnf}\right)$ (C.272)	−1,220			E: 102
Wheel angle between centreline and critical pinion drive side fillet point	$\tan\theta_{DLS} = \dfrac{\mu_{1D}}{r_{vn2} + h_{vfm1}}$ (C.273)	tan(1,641°)			E: 103
Wheel angle between centreline and critical pinion coast side fillet point	$\tan\theta_{CLS} = \dfrac{\mu_{1C}}{r_{vn2} + h_{vfm1}}$ (C.274)	tan(−1,837°)			E: 104
Radius from tool centre to critical pinion drive side fillet point	$R_{DL2} = \dfrac{r_{vn2} + h_{vfm1}}{\cos\theta_{DLS}}$ (C.275)	38,034			E: 105
Radius from tool centre to critical pinion coast side fillet point	$R_{CL2} = \dfrac{r_{vn2} + h_{vfm1}}{\cos\theta_{CLS}}$ (C.276)	38,038			E: 106

Table C.11 — *continued*

Description	Formula	Result	References to ISO 10300-1 E: Formula	References to ISO 10300-2 T: Table	References to ISO 10300-3 F: Figure
Wheel angle from centreline to pinion tip on drive side	θ_{D1} (C.277)	Initial: 3,882			
		Final: 2,881			
	$\Delta r_1 = r_{vn2}\left(e^{\theta_{D1}\tan\alpha_f} - 1{,}0\right)$ (C.278)	Initial: −0,183			E: 107
		Final: −0,136			E: 107
	$h_1 = \left(r_{vn2} + \Delta r_1\right)\sin\left(\alpha_{Dnf} + \theta_{D1}\right) - \left(r_{vn2}\sin\alpha_{Dnf} - g_{rb}\right)$ (C.279)	Initial: 3,104			E: 108
		Final: 2,505			E: 108
	$h_{1o} = \sqrt{r_{va1}^2 - \left(r_{vn1} - \Delta r_1\right)^2 \cos^2\left(\alpha_{vDnf} + \theta_{D1}\right)} - \left(r_{vn1} - \Delta r_1\right)\sin\left(\alpha_{Dnf} + \theta_{D1}\right)$ (C.280)	Initial: 2,339			E: 109
		Final: 2,505			E: 109
Wheel angle from centreline to tooth surface at critical fillet point on drive side	θ_{D2o} (C.281)	Initial: 1,941			
		Final: 1,673			
	$\mu_{1Do} = r_{vn2}\, e^{\theta_{D2o}\tan\alpha_f}\sin\theta_{D2o}$ (C.282)	Initial: 1,263			E: 110
		Final: 1,089			E: 110
Wheel angle from centreline to tooth surface at critical fillet point on coast side	θ_{C2o} (C.283)	Initial: −1,941			
		Final: −1,866			
	$\mu_{1Co} = r_{vn2} e^{\theta_{C2o}\tan\alpha_f}\sin\theta_{C2o}$ (C.284)	Initial: −1,269			E: 111
		Final: −1,220			E: 111

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Pinion angle from centreline to tooth surface at critical drive side fillet point	$r_{vn2}\left(e^{\theta_{D2o}\tan\alpha_f}-1{,}0\right)=r_{vn1}\left(1{,}0-e^{\theta_{D1o}\tan\alpha_f}\right)$ Formula shall be solved for θ_{D1o}.	(C.285)	−5,512°			E: 112
Pinion angle from centreline to tooth surface at critical coast side fillet point	$r_{vn2}\left(e^{\theta_{C2o}\tan\alpha_f}-1{,}0\right)=r_{vn1}\left(1{,}0-e^{\theta_{C1o}\tan\alpha_f}\right)$ Formula shall be solved for θ_{C1o}.	(C.286)	6,206°			E: 113
Wheel difference angle between tool and surface at drive side fillet point	$\Delta\theta_{D2o}=\theta_{DLS}-\theta_{D2o}$	(C.287)	−0,033°			E: 114
Wheel difference angle between tool and surface at coast side fillet point	$\Delta\theta_{C2o}=\theta_{CLS}-\theta_{C2o}$	(C.288)	0,028°			E: 115
Pinion difference angle between tool and surface at drive side fillet point	$\tan\Delta\theta_{D1o}=-\dfrac{R_{DL2}\sin\Delta\theta_{D2o}}{r_{vn2}+r_{vn1}-R_{DL2}\cos\Delta\theta_{D2o}}$	(C.289)	tan(0,116°)			E: 116
Pinion difference angle between tool and surface at coast side fillet point	$\tan\Delta\theta_{C1o}=-\dfrac{R_{CL2}\sin\Delta\theta_{C2o}}{r_{vn2}+r_{vn1}-R_{CL2}\cos\Delta\theta_{C2o}}$	(C.290)	tan(−0,101°)			E: 117
Pinion angle unbalance between fillet points	$\Delta\theta_1=\dfrac{\theta_{D1o}+\theta_{C1o}+\Delta\theta_{D1o}+\Delta\theta_{C1o}}{2{,}0}$	(C.291)	0,355°			E: 118

Table C.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Pinion angle from centreline to pinion tip	$\Delta r_1 = r_{vn1}\left(1{,}0 - e^{\theta_{Do}\tan\alpha_f}\right)$ Formula shall be solved for θ_{Do}. (C.292)	−9,462°			E: 119
Wheel angle from centreline to tooth surface at pitch point on drive side	θ_D (C.293)	Initial: −1,294			
		Final: −1,223			
	$\Delta r = r_{vn2}\left(e^{\theta_D \tan\alpha_f} - 1{,}0\right)$ (C.294)	Initial: 0,061			E: 120
		Final: 0,058			E: 120
	$h = \left(r_{vn2} + \Delta r\right)\ \sin\left(\alpha_{Dnf} + \theta_D\right) - \left(r_{vn2}\ \sin\alpha_{Dnf} - g_{rb}\right)$ (C.295)	Initial: −0,044			E: 121
		Final: 0,000			E: 121
Wheel angle from centreline to fillet point on drive flank	θ_{D2} (C.296)	Initial: −1,467			
		Final: −3,212			
	$\Delta r_2 = r_{vn2}\left(e^{\theta_{D2}\tan\alpha_f} - 1{,}0\right)$ (C.297)	Initial: 0,069			E: 123
		Final: 0,152			E: 123
	$h_2 = \left(r_{vn2} + \Delta r_2\right)\sin\left(\alpha_{Dnf} + \theta_{D2}\right) - \left(r_{vn2}\sin\alpha_{Dnf} - g_{rb}\right)$ (C.298)	Initial: −0,152			E: 124
		Final: −1,242			E: 124
	$h_{2o} = \pm\sqrt{r_{va1}^2 - \left(r_{vn1} + \Delta r_2\right)^2 \cos^2\left(\alpha_{Dnf} + \theta_{D2}\right)} + \left(r_{vn1} + \Delta r_2\right)\sin\left(\alpha_{Dnf} + \theta_{D2}\right)$ (C.299)	Initial: −1,406			E: 125
		Final: −1,242			E: 125

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Load sharing ratio for hypoid gears (pinion and wheel)						
Load sharing ratio for hypoid gears	ε_N	(C.300)	1,000			E: 126
Tooth strength factor for hypoid gears (pinion and wheel)						
Length of action from pinion tip to pitch circle in normal section	$g_{v\alpha 1} = \sqrt{h_1^2 + \left(\Delta r_1 - \Delta r\right)^2 - 2{,}0 h_1 \left(\Delta r_1 - \Delta r\right) \sin\left(\alpha_{Dnf} + \theta_{D1}\right)}$	(C.301)	2,574			E: 127
Length of action from wheel tip to pitch circle in normal section	$g_{v\alpha 2} = \sqrt{h_2^2 + \left(\Delta r_2 - \Delta r\right)^2 - 2{,}0 h_2 \left(\Delta r_2 - \Delta r\right) \sin\left(\alpha_{Dnf} + \theta_{D2}\right)}$	(C.302)	1,266			E: 128
Length of action in normal section	$g_{v\alpha n} = g_{v\alpha 1} + g_{v\alpha 2}$	(C.303)	3,840			E: 129
Profile contact ratio in mean normal section	$\varepsilon_{v\alpha n} = g_{v\alpha n} / p_{mn}$	(C.304)	1,148			E: 130
Modified contact ratio for hypoid gears	$\varepsilon_{v\gamma} = \sqrt{\varepsilon_{v\alpha n}^2 + \varepsilon_{v\beta}^2}$	(C.305)	1,813			E: 131
Profile load sharing factor	ε_f	(C.306)	0,093			E: 132b
Lengthwise load sharing factor	ε_b	(C.307)	1,804			E: 133b
Length of action from pinion tip to point of load application	$g_{v\alpha 3} = \left\| \frac{p_{mn}\, \varepsilon_{v\alpha n}^2}{\varepsilon_{v\gamma}^2} \left(\frac{0{,}5 \varepsilon_{v\gamma}^2}{\varepsilon_{v\alpha n}} - \frac{\varepsilon_{v\beta} \varepsilon_b k'}{\varepsilon_{v\alpha}} + \varepsilon_f \right) - g_{v\alpha 1} \right\|$	(C.308)	1,333			E: 134

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Length of action from wheel tip to point of load application	$g_{\mathrm{v\alpha}4} = \left\| \frac{p_{\mathrm{mn}} \varepsilon_{\mathrm{v\alpha n}}^2}{\varepsilon_{\mathrm{v\gamma}}^2} \left(\frac{0{,}5 \varepsilon_{\mathrm{v\gamma}}^2}{\varepsilon_{\mathrm{v\alpha n}}} + \frac{\varepsilon_{\mathrm{v\beta}} \varepsilon_{\mathrm{b}} k'}{\varepsilon_{\mathrm{v\alpha}}} + \varepsilon_{\mathrm{f}} \right) - g_{\mathrm{v\alpha}2} \right\|$	(C.309)	1,583			E: 135
Length of action to point of load application	$g_{\mathrm{J}\,1,2} = g_{\mathrm{v\alpha n}} - g_{\mathrm{v\alpha}3,4}$	(C.310)	2,507 2,257			E: 136
Wheel angle from pinion tip to point of load application	θ_{D3}	(C.311)	Initial: −1,941			
			Final: 0,891			
	$\Delta r_3 = r_{\mathrm{vn2}} \left(e^{\theta_{\mathrm{D3}} \tan\alpha_{\mathrm{f}}} - 1{,}0 \right)$	(C.312)	Initial: 0,092			E: 137
			Final: −0,042			E: 137
	$h_3 = \left(r_{\mathrm{vn2}} + \Delta r_3 \right) \sin\left(\alpha_{\mathrm{vDnf}} + \theta_{\mathrm{D3}} \right) - \left(r_{\mathrm{vn2}} \sin\alpha_{\mathrm{vDnf}} - g_{\mathrm{rb}} \right)$	(C.313)	Initial: −0,447			E: 138
			Final: 1,301			E: 138
	$h_{3\mathrm{o}} = \sqrt{g_{\mathrm{v\alpha}3}^2 - \left(\Delta r_3 - \Delta r \right)^2 \cos^2\left(\alpha_{\mathrm{vDnf}} + \theta_{\mathrm{D3}} \right)} - \left\| \Delta r_3 - \Delta r \right\| \sin\left(\alpha_{\mathrm{vDnf}} + \theta_{\mathrm{D3}} \right)$	(C.314)	Initial: 1,324			E: 139
			Final: 1,301			E: 139

Table C.11 — *continued*

Description	Formula	Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
			E: Formula	T: Table	F: Figure
Pinion angle from wheel tip to point of load application	θ_{D4} (C.315	Initial: 1,294			
		Final: 1,291			
	$\Delta r_4 = r_{vn2}\left(e^{\theta_{D4}\tan\alpha_f} - 1{,}0\right)$ (C.316)	Initial: −0,061			E: 140
		Final: −0,061			E: 140
	$h_4 = \left(r_{vn2} + \Delta r_4\right)\sin\left(\alpha_{vDnf} + \theta_{D4}\right) - \left(r_{vn2}\sin\alpha_{vDnf} - g_{rb}\right)$ (C.317)	Initial: 1,546			E: 141
		Final: 1,544			E: 141
	$h_{4o} = \sqrt{g_{v\alpha 4}^2 - \left(\Delta r_4 - \Delta r\right)^2 \cos^2\left(\alpha_{vDnf} + \theta_{D4}\right)} - \left\|\Delta r_4 - \Delta r\right\| \sin\left(\alpha_{vDnf} + \theta_{D4}\right)$ (C.318)	Initial: 1,544			E: 142
		Final: 1,544			E: 142
Distance from pitch circle to point of load application	$\Delta r_{LN2} = \dfrac{\left(r_{vn2} + \Delta r_3\right)\cos\left(\alpha_{vDnf} + \theta_{D3}\right)}{\cos\alpha_{LN2}} - r_{vn2}$ (C.319)	−0,565			E: 143
Angle between centre line and line from point of load application and fillet point on wheel	α_{2oo} (C.320)	Initial: 31,736°			
		Final: 32,554°			

Table C.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Horizontal distance from centreline to critical fillet point	$s_{N2} = x_1 - \rho_{va02}\cos\alpha_{2oo}$ (C.321)	Initial: 0,888			E: 144
		Final: 0,890			E: 144
Vertical distance from pitch circle to critical fillet point	$y_2 = y_1 + \rho_{va02}\sin\alpha_{2oo}$ (C.322)	Initial: 1,259			E: 145
		Final: 1,262			E: 145
Wheel load height at weakest section	$h_{N2} = y_2 + \Delta r_{LN2}$ (C.323)	Initial: 0,694			E: 146
		Final: 0,697			E: 146
Auxiliary value	$h_{N2o} = \dfrac{s_{N2}}{2{,}0\tan\alpha_{2oo}}$ (C.324)	Initial: 0,718			E: 147
		Final: 0,697			E: 147
Wheel tooth strength factor	$x_{N2} = \dfrac{s_{N2}^2}{h_{N2}}$ (C.325)	1,136			E: 148
Pinion angle from pitch point to point of load application	$\Delta r_4 = r_{vn1}\left(1{,}0 - e^{\theta_{D5}\tan\alpha_f}\right)$ (C.326) Formula shall be solved for θ_{D5}.	−4,256°			E: 149

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Pinion pressure angle at point of load application	$\alpha_{LN1} = \alpha_{vnf} + \theta_{D4} - \theta_{D5} + \Delta\theta_1$	(C.327)	21,770°			E: 150
Pinion radial distance to point of load application	$r_{41o} = \frac{(r_{vn1} - \Delta r_4)\cos(\alpha_{Dnf} + \theta_{D4})}{\cos\alpha_{LN1}}$	(C.328)	11,681			E: 151
Start of iteration with $\alpha_{Do} = \alpha_{nD} + \alpha_{nC}$	α_{Do}	(C.329)	Initial: 40,000°			
			Final: 37,842°			
Wheel angle between centreline and pinion fillet (enclosed iteration)	θ_{D2oo}	(C.330)	Initial (first pass): 1,941°			
			Final (first pass): 2,904°			
			Initial (last pass): 1,941°			
			Final (last pass): 3,021°			

Table C.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
	$\Delta r_5 = r_{vn2}\, e^{\theta_{D2oo} \tan\alpha_f}$ (C.331)	Initial (first pass): 37,282			E: 152
		Final (first pass): 37,237			
		Initial (last pass): 37,282			
		Final (last pass): 37,231			
Wheel angle between centreline and pinion fillet (enclosed iteration)	$\mu_{D2} = \dfrac{r_{vn2} + h_{vfm1} - \rho_{va01} - \Delta r_5 \cos\theta_{D2oo}}{\tan\alpha_{Do}}$ (C.332)	Initial (first pass): 0,712			E: 153
		Final (first pass): 0,797			
		Initial (last pass): 0,769			
		Final (last pass): 0,873			

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
	$\mu_D = \Delta r_5 \sin\theta_{D2oo}$	(C.333)	Initial (first pass): 1,263			E: 154
			Final (first pass): 1,886			
			Initial (last pass): 1,263			
			Final (last pass): 1,962			
Pinion angle between centreline and pinion fillet	$r_{vn2}\left(e^{\theta_{D2oo}\tan\alpha_f} - 1{,}0\right) = r_{vn1}\left(1{,}0 - e^{\theta_{D1oo}\tan\alpha_f}\right)$ Formula shall be solved for θ_{D1oo}.	(C.334)	Initial: −9,534°			E: 155
			Final: −9,917°			
Wheel rotation through path of action	$\tan\theta_{L2o} = \dfrac{\mu_{D1} - \rho_{va01}\cos\alpha_{Do}}{r_{vn2} + h_{vfm1} - \rho_{va01} + \rho_{va01}\sin\alpha_{Do}}$	(C.335)	Initial: tan(1,458°)			E: 156
			Final: tan(1,453°)			
Wheel angle difference between path of action and tooth surface at pinion fillet	$\Delta\theta_{D2oo} = \theta_{L2o} - \theta_{D2oo}$	(C.336)	Initial: −1,446°			E: 157
			Final: −1,569°			
Wheel radius to pinion fillet point	$r_{L2o} = \dfrac{r_{vn2} + h_{vfm1} - \rho_{va01} + \rho_{va01}\sin\alpha_{Do}}{\cos\theta_{L2o}}$	(C.337)	Initial: 37,973			E: 158
			Final: 37,968			

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Pinion angle to fillet point	$\tan\Delta\theta_{D1oo} = -\dfrac{r_{L2o}\sin\Delta\theta_{D2oo}}{r_{vn2}+r_{vn1}-r_{L2o}\cos\Delta\theta_{D2oo}}$	(C.338)	Initial: tan(5,114°)			E: 159
			Final: tan(5,542°)			
Pinion radius to fillet point	$r_{L1o} = \dfrac{r_{vn2}+r_{vn1}-r_{L2o}\cos\Delta\theta_{D2oo}}{\cos\Delta\theta_{D1oo}}$	(C.339)	Initial: 10,748			E: 160
			Final: 10,762			
Pinion angle from centre line to fillet point	$\left(\Delta\theta_1-\theta_{L1o}\right)=\Delta\theta_1-\theta_{D1oo}-\Delta\theta_{D1oo}$	(C.340)	Initial: 4,775°			E: 161
			Final: 4,730°			
Angle between centre line and line from point of load application and fillet point on pinion	$\alpha_1=\alpha_{Do}-\theta_{D2oo}+\theta_{D1oo}$	(C.341)	Initial: 27,562°			E: 162
			Final: 24,904°			
Horizontal distance to critical fillet point	$s_{N1}=r_{L1o}\sin\left(\Delta\theta_1-\theta_{L1o}\right)$	(C.342)	Initial: 0,895			E: 163
			Final: 0,887			
Pinion load height at weakest section	$h_{N1}=r_{41o}-r_{L1o}\cos\left(\Delta\theta_1-\theta_{L1o}\right)$	(C.343)	Initial: 0,971			E: 164
			Final: 0,956			
Auxiliary value	$h_{N1o}=\dfrac{s_{N1}}{2{,}0\tan\alpha_1}$	(C.344)	Initial: 0,857			E: 165
			Final: 0,956			
Pinion tooth strength factor	$x_{N1}=\dfrac{s_{N1}^2}{h_{N1}}$	(C.345)	0,824			E: 166

Table C.11 — *continued*

Description	Formula		Result	References to ISO 10300-1	References to ISO 10300-2	References to ISO 10300-3
				E: Formula	T: Table	F: Figure
Tooth form factor for hypoid gears (pinion and wheel)						
Tooth form factor for pinion	$Y_1 = \frac{2}{3}\left[\frac{1}{\left(\frac{1}{x_{N1}} - \frac{\tan \alpha_{LN1}}{3\ s_{N1}}\right)}\right]$	(C.346)	0,627			E: 167
Tooth form factor for wheel	$Y_2 = \frac{2}{3}\left[\frac{1}{\left(\frac{1}{x_{N2}} - \frac{\tan \alpha_{LN2}}{3\ s_{N2}}\right)}\right]$	(C.347)	0,846			E: 167
Transverse radius to point of load application (pinion and wheel)						
Mean face width of pinion	$b_{k1} = \frac{b_1 \varepsilon_{v\alpha n} \varepsilon_{v\beta}}{\varepsilon_{v\gamma}^2}$	(C.348)	15,261			E: 168
Mean face width of wheel	$b_{k2} = \frac{b_2 \varepsilon_{v\alpha n} \varepsilon_{v\beta}}{\varepsilon_{v\gamma}^2}$	(C.349)	12,742			E: 168
Contact shift due to load for pinion	$x_{oo1} = k' b_{k1} - \frac{b_1 \varepsilon_f \varepsilon_{v\beta}}{\varepsilon_{v\gamma}^2}$	(C.350)	1,395			E: 169

Table C.11 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Contact shift due to load for wheel	$x_{oo2} = k' b_{k2} + \frac{b_2 \varepsilon_f \varepsilon_{v\beta}}{\varepsilon_{v\gamma}^2}$ (C.351)	1,165			E: 170
Transverse radius to point of load application for pinion	$r_{my01} = \frac{r_{mpt1}\left(x_{oo1} + R_{m2}\right)}{R_{m2}} + \left(r_{410} - r_{vn1}\right)\ m_{et2}$ (C.352)	28,005 mm			E: 171
Transverse radius to point of load application for wheel	$r_{my02} = \frac{r_{mpt2}\left(x_{oo2} + R_{m2}\right)}{R_{m2}} + \Delta r_{LN2}\ m_{et2}$ (C.353)	71,564 mm			E: 172
Tooth fillet radius at root diameter (pinion and wheel)					
Relative fillet radius at root of tooth on pinion	$r_{mf1} = \frac{\left(h_{vfm1} - \rho_{va01}\right)^2}{r_{vn1} + h_{vfm1} - \rho_{va01}} + \rho_{va01}$ (C.354)	0,180			E: 173
Relative fillet radius at root of tooth on wheel	$r_{mf2} = \frac{\left(h_{vfm2} - \rho_{va02}\right)^2}{r_{vn2} + h_{vfm2} - \rho_{va02}} + \rho_{va02}$ (C.355)	0,273			E: 173
Stress concentration and correction factor (pinion and wheel)					
Stress concentration and stress correction factor on pinion	$Y_{f1} = L + \left(\frac{2s_{N1}}{r_{mf1}}\right)^M \left(\frac{2s_{N1}}{h_{N1}}\right)^O$ (C.356)	2,193			E: 174
Stress concentration and stress correction factor on wheel	$Y_{f2} = L + \left(\frac{2s_{N2}}{r_{mf2}}\right)^M \left(\frac{2s_{N2}}{h_{N2}}\right)^O$ (C.357)	2,302			E: 174

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
where	$L = 0{,}325\ 454\ 5 - 0{,}007\ 272\ 7\ \alpha_n$	(C.358)	0,210			E: 175
	$M = 0{,}331\ 818\ 2 - 0{,}009\ 090\ 9\ \alpha_n$	(C.359)	0,188			E: 176
	$O = 0{,}268\ 181\ 8 + 0{,}009\ 090\ 9\ \alpha_n$	(C.360)	0,412			E: 177
Inertia factor						
Inertia factor when $\varepsilon_{v\gamma} \geq 2{,}0$	Y_i	(C.361)	1,103			E: 178b
Calculated effective face width						
Projected length of the instantaneous contact line in the tooth lengthwise direction (pinion)	$g_{K1} = \dfrac{b_1\ g_{v\alpha n}\ g_{J1}\ \cos^2 \beta_{vb}}{g_\eta^2}$	(C.362)	9,387			E: 179
Projected length of the instantaneous contact line in the tooth lengthwise direction (wheel)	$g_{K2} = \dfrac{b_2\ g_{v\alpha n}\ g_{J2}\ \cos^2 \beta_{vb}}{g_\eta^2}$	(C.363)	7,056			E: 179

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Toe increment on pinion	$\Delta b'_{i1} = \frac{b_1 - g_{K1}}{2\cos\beta_{m1}} - \frac{x_{oo1}}{\cos\beta_{m1}}$	(C.364)	13,406			E: 180
Toe increment on wheel	$\Delta b'_{i2} = \frac{b_2 - g_{K2}}{2\cos\beta_{m2}} - \frac{x_{oo2}}{\cos\beta_{m2}}$	(C.365)	8,899			E: 180
Heel increment on pinion	$\Delta b'_{e1} = \frac{b_1 - g_{K1}}{2\cos\beta_{m1}} + \frac{x_{oo1}}{\cos\beta_{m1}}$	(C.366)	17,352			E: 181
Heel increment on wheel	$\Delta b'_{e2} = \frac{b_2 - g_{K2}}{2\cos\beta_{m2}} + \frac{x_{oo2}}{\cos\beta_{m2}}$	(C.367)	11,395			E: 181
Calculated effective face width on pinion	$b_{ce1} = 25{,}4\ h_{N1}\cos\beta_{m1}\left[\arctan\left(\frac{\Delta b_{i1}}{25{,}4\ h_{Na1}}\right) + \arctan\left(\frac{\Delta b_{e1}}{25{,}4\ h_{Na1}}\right)\right] + g_{K1}$	(C.368)	28,703			E: 183
Calculated effective face width on wheel	$b_{ce2} = 25{,}4\ h_{N2}\cos\beta_{m2}\left[\arctan\left(\frac{\Delta b_{i2}}{25{,}4\ h_{Na2}}\right) + \arctan\left(\frac{\Delta b_{e2}}{25{,}4\ h_{Na2}}\right)\right] + g_{K2}$	(C.369)	24,204			E: 183

Table C.11 — *continued*

Description	Formula		Result	References to ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Basic formula of geometry factor (pinion and wheel)						
Geometry factor for pinion	$Y_{J1} = \frac{Y_1}{Y_{f1} \cdot \varepsilon_N \cdot Y_i} \cdot \frac{r_{my01}}{r_{mpt1}} \cdot \frac{b_{ce1}}{b_1} \cdot \frac{m_{mt1}}{m_{et2}}$	(C.370)	0,298			E: 51
Geometry factor for wheel	$Y_{J2} = \frac{Y_2}{Y_{f2} \cdot \varepsilon_N \cdot Y_i} \cdot \frac{r_{my02}}{r_{mpt2}} \cdot \frac{b_{ce2}}{b_2} \cdot \frac{m_{mt2}}{m_{et2}}$	(C.371)	0,261			E: 51
Root stress adjustment factor for carburized case hardened steel	Y_A	(C.372)	1,075			E: 184
Relative surface condition factor						
Relative surface condition factor	$Y_{R\,rel\,T}$	(C.373)	1,000			E: 185
Relative notch sensitivity factor						
Relative notch sensitivity factor	$Y_{\delta\,rel\,T}$	(C.374)	1,000			E: 186a

Table C.11 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
K–factors						
Face load factor	$K_{F,\beta}$	(C.375)	1,633	E: 28		
Transverse load factor	$K_{F,\alpha}$	(C.376)	1,000	E: 36		
Tooth root stress formula (pinion and wheel)						
Nominal tooth root stress on pinion	$\sigma_{F0\ 1\text{-B2}} = \frac{F_{mt1}}{b_1} \cdot \frac{m_{mt1}}{m_{et2}^2} \cdot \frac{Y_{A1}}{Y_{J1}}$	(C.377)	258,1 N/mm^2			E: 47
Tooth root stress on pinion	$\sigma_{F1\text{-B2}} = \sigma_{F0\ 1\text{-B2}} K_A\ K_v\ K_{F\beta}\ K_{F\alpha}$	(C.378)	463,6 N/mm^2			E: 44
Nominal tooth root stress on wheel	$\sigma_{F0\ 2\text{-B2}} = \frac{F_{mt2}}{b_2} \cdot \frac{m_{mt2}}{m_{et2}^2} \cdot \frac{Y_{A2}}{Y_{J2}}$	(C.379)	352,5 N/mm^2			E: 47
Tooth root stress on wheel	$\sigma_{F2\text{-B2}} = \sigma_{F0\ 2\text{-B2}} K_A\ K_v\ K_{F\beta}\ K_{F\alpha}$	(C.380)	632,3 N/mm^2			E: 44
Permissible tooth root stress (pinion and wheel)						
Permissible tooth root stress on pinion	$\sigma_{FP1\text{-B2}} = \sigma_{F\ lim}\ Y_{ST}\ Y_{NT}\ Y_{\delta\ rel\ T\text{-B2}}\ Y_{R\ rel\ T\text{-B2}}\ Y_X$	(C.381)	1 000,0 N/mm^2			E: 49
Permissible tooth root stress on wheel	$\sigma_{FP2\text{-B2}} = \sigma_{F\ lim}\ Y_{ST}\ Y_{NT}\ Y_{\delta\ rel\ T\text{-B2}}\ Y_{R\ rel\ T\text{-B2}}\ Y_X$	(C.382)	1 000,0 N/mm^2			E: 49
Calculated safety factors (pinion and wheel)						
Calculated safety factor for pinion	$S_{F1\text{-B2}} = \frac{\sigma_{FP1\text{-B2}}}{\sigma_{F1\text{-B2}}}$	(C.383)	2,157			E: 50

Table C.11 — *continued*

Description	**Formula**	**Result**	**References to**		
			ISO 10300-1	**ISO 10300-2**	**ISO 10300-3**
			E: Formula	**T: Table**	**F: Figure**
Calculated safety factor for wheel	$S_{\text{F2-B2}} = \dfrac{\sigma_{\text{FP2-B2}}}{\sigma_{\text{F2-B2}}}$ (C.384)	1,581			E: 50

Annex D
(informative)

Sample 4: Rating of a hypoid gear set according to Method B1 and Method B2

D.1 Initial data

Sample 4 is for a hypoid gear pair which uses Method 3 according to ISO 23509.

Table D.1 — Initial data for pitch cone parameters

Symbol	Description	Method 0	Method 1	Method 2	Method 3
Σ	shaft angle	X	X	X	90°
a	hypoid offset	0,0	X	X	40 mm
$z_{1,2}$	number of teeth	X	X	X	12/49
d_{m2}	mean pitch diameter of wheel	—	—	X	—
d_{e2}	outer pitch diameter of wheel	X	X	—	400 mm
b_2	wheel face width	X	X	X	60 mm
β_{m1}	mean spiral angle of pinion	—	X	—	—
β_{m2}	mean spiral angle of wheel	X	—	X	30°
r_{c0}	cutter radius	X	X	X	135 mm
z_0	number of blade groups (only face hobbing)	X	—	X	5

Table D.2 — Input data for tooth profile parameters

Data type I		Data type II	
Symbol	**Description**	**Symbol**	**Description**
α_{dD}	19°		
α_{dC}	21°		
$f_{\alpha lim}$	0		
x_{hm1}	0,2	c_{ham}	—
k_{hap}	1	k_d	—
k_{hfp}	1,25	k_c	—
x_{smn}	0,031	k_t W_{m2}	— —
j_{et2}	0 mm		
θ_{a2}	0°		
θ_{f2}	0°		
ρ_{a01}	0,8 mm		
ρ_{a02}	1,2 mm		
$s_{pr1D,C}$	0 mm/0 mm		
$s_{pr2D,C}$	0 mm/0 mm		

Table D.3 and Table D.4 show geometric and operational data and text for explanation.

Table D.3 — Geometric data from calculation according to ISO 23509

Symbol	Description	Values	Symbol	Description	Values
$d_{m1,2}$	mean pitch diameter of pinion/wheel	99,377 mm/ 343,151 mm	ζ_{mp}	offset angle on pitch plane	12,922°
$h_{am1,2}$	mean addendum of pinion/wheel	7,278 mm/ 4,852 mm	ζ_R	pinion offset angle on root plane	12,265°
$h_{fm1,2}$	mean dedendum of pinion/wheel	6,368 mm/ 8,794 mm	$R_{e1,2}$	outer cone distance on pinion/wheel	191,947 mm/ 211,072 mm
$\alpha_{eD,C}$	effective pressure angle for drive side/coast side	20,731°/ 19,269°	$R_{m1,2}$	mean cone distance on pinion/wheel	159,088 mm/ 181,074 mm
$\alpha_{nD,C}$	generated pressure angle for drive side/coast side	19°/21°	$\delta_{1,2}$	pitch angle on pinion/wheel	18,200°/ 71,360°
α_{lim}	limit pressure angle	−1,731°	$\delta_{a1,2}$	face angle on pinion/wheel	18,200°/ 71,360°
m_{mn}	mean normal module	6,065 mm	$\delta_{f1,2}$	root angle on pinion/wheel	18,200°/ 71,360°
k_{hfp}	basic crown gear dedendum factor	1,25	$x_{sm1,2}$	thickness modification coefficient on pinion/wheel	0,031/−0,031
ζ_m	pinion offset angle on axial plane	12,265°	m_{et2}	outer transverse module	8,163 mm
$s_{mn1,2}$	mean normal circular tooth thickness of pinion/wheel	10,786 mm/ 8,268 mm			

Table D.4 — Operating parameters and additional considerations

Symbol	Description	Value
Additional data		
	wheel profile	non-generated
	roughing/finishing method	face hobbing
$b_{2\text{eff}}$	effective face width on wheel	$0{,}85 \cdot b_2$
	profile crowning	low
	verification of contact pattern	checked under light test load for each gear
	mounting conditions of pinion and wheel	one member cantilever-mounted
Operation parameters		
T_1	pinion torque	3 000 Nm
n_1	pinion rotational speed	800 min^{-1}
K_A	application factor	1,1
	active flank	drive
Material data for pinion and wheel (case hardened steel)		
$\sigma_{H\,\lim}$	allowable stress number (contact)	1 500 N/mm^2
$\sigma_{F\,\lim}$	nominal stress number (bending)	480 N/mm^2
	surface hardness	same for pinion and wheel
Quality parameters		
R_z	flank roughness on pinion/wheel	8 µm/8 µm
R_z	tooth root roughness on pinion/wheel	16 µm/16 µm
f_{pt}	single pitch deviation on pinion/wheel	14 µm/27 µm
Lubrication parameters		
	oil type	ISO-VG-150
	oil temperature	90 °C

D.2 Calculation of Sample 4 according to Method B1

Table D.5 — Virtual cylindrical gears

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Virtual cylindrical gears in transverse section						
Reference diameter on pinion	$d_{v1} = \dfrac{d_{m1}}{\cos\delta_1}$	(D.1)	104,610 mm	E: A.1		
Reference diameter on wheel	$d_{v2} = \dfrac{d_{m2}}{\cos\delta_2}$	(D.2)	1073,619 mm	E: A.1		
Number of teeth on pinion	$z_{v1} = d_{v1}/m_{vt}$	(D.3)	13,872	E: A.12		
Number of teeth on wheel	$z_{v2} = d_{v2}/m_{vt}$	(D.4)	142,369	E: A.12		
Gear ratio	$u_v = z_{v2}/z_{v1}$	(D.5)	10,263	E: A.13		
Helix angle	$\beta_v = \dfrac{\beta_{m1} + \beta_{m2}}{2}$	(D.6)	36,461°	E: A.8		
Transverse pressure angle of virtual cylindrical gears	$\alpha_{vet} = \arctan\left(\tan\alpha_e / \cos\beta_v\right)$ since $\alpha_e = \alpha_{eD}$ for drive side.	(D.7)	25,202°	E: A.10		
Transverse module	$m_{vt} = m_{mn}/\cos\beta_v$	(D.8)	7,541 mm	E: A.11		

Table D.5 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Auxiliary angle for virtual face width	$\theta_{mp} = \arctan\left(\sin\delta_2 \tan\zeta_m\right)$ (D.9)	11,640°	E: A.21		
Projected auxiliary angle for length of contact line	$\gamma' = \theta_{mp} - \zeta_{mp}/2$ (D.10)	5,179°	E: A.20		
Centre distance of virtual cylindrical gear pair	$a_v = \left(d_{v1} + d_{v2}\right) / 2$ (D.11)	589,115 mm	E: A.5		
Helix angle of virtual cylindrical gear at base circle	$\beta_{vb} = \arcsin\left(\sin\beta_v \cos\alpha_e\right)$ since $\alpha_e = \alpha_{eD}$ for drive side. (D.12)	33,766°	E: A.16		
Tip diameter on pinion	$d_{va1} = d_{v1} + 2\ h_{am1}$ (D.13)	119,166 mm	E: A.6		
Tip diameter on wheel	$d_{va2} = d_{v2} + 2\ h_{am2}$ (D.14)	1 083,323 mm	E: A.6		
Root diameter on pinion	$d_{vf1} = d_{v1} - 2\ h_{fm1}$ (D.15)	91,874 mm	E: A.7		
Root diameter on wheel	$d_{vf2} = d_{v2} - 2\ h_{fm2}$ (D.16)	1 056,031 mm	E: A.7		
Base diameter on pinion	$d_{vb1} = d_{v1} \cos\alpha_{vet}$ (D.17)	94,653 mm	E: A.9		
Base diameter on wheel	$d_{vb2} = d_{v2} \cos\alpha_{vet}$ (D.18)	971,425 mm	E: A.9		
Transverse base pitch	$p_{vet} = \pi\ m_{mn} \cos\alpha_{vet} / \cos\beta_v$ (D.19)	21,436 mm	E: A.17		
Length of path of contact of virtual cylindrical gear in transverse section	$g_{v\alpha} = \frac{1}{2}\left[\left(\sqrt{d_{va1}^2 - d_{vb1}^2} - d_{v1}\sin\alpha_{vet}\right) + \left(\sqrt{d_{va2}^2 - d_{vb2}^2} - d_{v2}\sin\alpha_{vet}\right)\right]$ (D.20)	25,100 mm	E: A.18		

Table D.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Transverse contact ratio	$\varepsilon_{\mathrm{v}\alpha} = g_{\mathrm{v}\alpha} / p_{\mathrm{vet}}$	(D.21)	1,171	E: A.23		
Effective face width with $b_{2\,\mathrm{eff}} = 0{,}85 \cdot b_2$	$b_{\mathrm{v\,eff}} = \dfrac{\left[b_{2\,\mathrm{eff}} / \cos\left(\zeta_{\mathrm{mp}}/2\right) - g_{\mathrm{v}\alpha} \cos\alpha_{\mathrm{vet}} \tan\left(\zeta_{\mathrm{mp}}/2\right) \right]}{1 - \tan\gamma' \tan\left(\zeta_{\mathrm{mp}}/2\right)}$	(D.22)	49,260 mm	E: A.19		
Face width	$b_{\mathrm{v}} = b_2 \dfrac{b_{\mathrm{v\,eff}}}{b_{2\,\mathrm{eff}}}$	(D.23)	57,953 mm	E: A.22		
Virtual cylindrical gears in normal section						
Number of pinion teeth of virtual cylindrical gears	$z_{\mathrm{vn1}} = \dfrac{z_{\mathrm{v1}}}{\cos^2 \beta_{\mathrm{vb}} \cos\beta_{\mathrm{v}}}$	(D.24)	24,958	E: A.38		
Number of wheel teeth of virtual cylindrical gears	$z_{\mathrm{vn2}} = u_{\mathrm{v}} z_{\mathrm{vn1}}$	(D.25)	256,145	E: A.39		
Reference diameter on pinion	$d_{\mathrm{vn1}} = z_{\mathrm{vn1}} m_{\mathrm{mn}}$	(D.26)	151,370 mm	E: A.40		
Reference diameter on wheel	$d_{\mathrm{vn2}} = z_{\mathrm{vn2}} m_{\mathrm{mn}}$	(D.27)	1 553,518 mm	E: A.40		
Tip diameter on pinion	$d_{\mathrm{van1}} = d_{\mathrm{vn1}} + 2h_{\mathrm{am1}}$	(D.28)	165,926 mm	E: A.41		

Table D.5 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Tip diameter on wheel	$d_{van2} = d_{vn2} + 2h_{am2}$	(D.29)	1 563,222 mm	E: A.41		
Root diameter on pinion	$d_{vfn1} = d_{vn1} - 2\, h_{fm1}$	(D.30)	138,634 mm	E: A.7		
Root diameter on wheel	$d_{vfn2} = d_{vn2} - 2\, h_{fm2}$	(D.31)	1 535,930 mm	E: A.7		
Base diameter on pinion	$d_{vbn1} = d_{vn1} \cos\alpha_e$	(D.32)	141,570 mm	E: A.42		
Base diameter on wheel	$d_{vbn2} = d_{vn2} \cos\alpha_e$	(D.33)	1 452,931 mm	E: A.42		
Face contact ratio	$\varepsilon_{v\beta} = \dfrac{b_{v\,eff} \sin\beta_v}{\pi\, m_{mn}}$	(D.34)	1,536	E: A.24		
Virtual contact ratio	$\varepsilon_{v\gamma} = \varepsilon_{v\alpha} + \varepsilon_{v\beta}$	(D.35)	2,707	E: A.25		
Inclination angle of contact line	$\beta_B = \arctan\left(\tan\beta_v \sin\alpha_e\right)$	(D.36)	14,658°	E: A.36		
Radius of relative curvature in normal section at the mean point	$\rho_t = \left[\dfrac{1}{\cos\alpha_{nD}\left(\tan\alpha_{nD} - \tan\alpha_{lim}\right) + \tan\zeta_{mp} \tan\beta_B} \cdot \dfrac{\cos\beta_{m1}\cos\beta_{m2}}{\cos\zeta_{mp}} \cdot \left(\dfrac{1}{R_{m2}\tan\delta_2} + \dfrac{1}{R_{m1}\tan\delta_1}\right)\right]^{-1}$	(D.37)	30,338 mm	E: A.37a		
Radius of relative curvature vertical to the contact line	$\rho_{rel} = \lvert\rho_t\rvert \cos^2\beta_B$	(D.38)	28,395 mm	E: A.35		

Table D.6 — General influence factors

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Nominal tangential force of bevel gears	$F_{\text{mt}1} = \dfrac{2\ 000\ T_1}{d_{\text{m}1}}$	(D.39)	60 376,1 N	E: 1		
Nominal tangential force of virtual cylindrical gears	$F_{\text{vmt}} = F_{\text{mt}1} \dfrac{\cos \beta_{\text{v}}}{\cos \beta_{\text{m}1}}$	(D.40)	66 310,9 N	E: 2		
Nominal tangential speed at mean point of the pinion	$v_{\text{mt}1} = \dfrac{d_{\text{m}1}\ n_1}{19\ 098}$	(D.41)	4,163 m/s	E: 5		
Nominal tangential speed at mean point of the wheel	$v_{\text{mt}2} = \dfrac{d_{\text{m}2}\ n_2}{19\ 098}$	(D.42)	3,520 m/s	E: 5		
Correction factor for non-average conditions for $F_{\text{vmt}} K_{\text{A}} / b_{\text{veff}} \geq 100$ N/mm	C_{F}	(D.43)	1,000	E: 12a		
Mean value of mesh stiffness per unit face width	$c_{\gamma} = c_{\gamma 0} C_{\text{F}}$	(D.44)	20,00 N/(mm·µm)	E: 11		
Single stiffness	$c' = c'_0\ C_{\text{F}}$	(D.45)	14,00 N/(mm·µm)	E: 17		
Single pitch deviation	f_{pt}	(D.46)	27,000 µm			
Running–in allowance for case hardened and nitrided gears	$y_{\alpha} = 0{,}075\ f_{\text{pt}}$	(D.47)	2,025 µm	E: 43		

Table D.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Effective pitch deviation with $y_p = y_\alpha$	$f_{p\,eff} = f_{pt} - y_p$	(D.48)	24,975 µm	E: 16		
Relative pinion mass per unit face width reduced to the line of action	$m_1^* = \frac{1}{8} \rho\, \pi \frac{d_{m1}^2}{\cos^2\left[\left(\alpha_{nD} + \alpha_{nC}\right)/2\right]}$ where ρ is the density of the gear material (for steel ρ = 7,86·10^{−6} kg/mm³)	(D.49)	0,035 kg/mm	E: 13		
Relative wheel mass per unit face width reduced to the line of action	$m_2^* = \frac{1}{8} \rho\, \pi \frac{d_{m2}^2}{\cos^2\left[\left(\alpha_{nD} + \alpha_{nC}\right)/2\right]}$ where ρ is the density of the gear material (for steel $\rho = 7{,}86 \cdot 10^{-6}$ kg/mm^3)	(D.50)	0,412 kg/mm	E: 13		
Mass reduced to the line of action of the dynamically equivalent cylindrical gear pair	$m_{red} = \frac{m_1^* m_2^*}{m_1^* + m_2^*}$	(D.51)	0,032 kg/mm	E: 10		
Resonance speed of pinion	$n_{E1} = \frac{30 \times 10^3}{\pi\, z_1} \sqrt{\frac{c_\gamma}{m_{red}}}$	(D.52)	19 941,2 min^{-1}	E: 9		
Dimensionless reference speed	$N = \frac{n_1}{n_{E1}}$	(D.53)	0,040	E: 8		

Table D.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
For virtual contact ratio, $\varepsilon_{v\gamma} = 2{,}707 > 2$	$c_{v1,2} = c_{v1} + c_{v2}$	(D.54)	0,557	T: 3		
	c_{v3}	(D.55)	0,084	T: 3		
	c_{v4}	(D.56)	0,343	T: 3		
	$c_{v5,6}$	(D.57)	0,594	T: 3		
	c_{v7}	(D.58)	0,974	T: 3		
Constant factor for calculating the dynamic factor with $K_A = 1{,}1$ as given in Table D.4	$K = \frac{b_v\, f_{p\,eff}\, c'}{F_{vmt} K_A} c_{v1,2} + c_{v3}$	(D.59)	0,238	E: 15		
Dynamic factor	$K_v = N \cdot K + 1$	(D.60)	1,000	E: 14		
Determination of the length of contact lines						
For virtual contact ratio, $\varepsilon_{v\beta} = 1{,}536$, $\varepsilon_{v\beta} \geq 1$	$f_t = +p_{vet} \cos\beta_{vb}$	(D.61)	17,820 mm	T: A.2		
	f_m	(D.62)	0,000 mm	T: A.2		
	$f_r = -p_{vet} \cos\beta_{vb}$	(D.63)	−17,820 mm	T: A.2		
Maximum distances from middle contact line	$f_{maxB} = \frac{1}{2}\left[g_{v\alpha} + b_{v\,eff}\left(\tan\gamma + \tan\beta_{vb}\right)\right]\cos\beta_{vb}$	(D.64)	26,173 mm	E: A.31		

Table D.6 — *continued*

Description	Formula	Result	References to		
			ISO 10300-1	ISO 10300-2	ISO 10300-3
			E: Formula	T: Table	F: Figure
Maximum distances from middle contact line	$f_{\max 0} = \frac{1}{2}\left[g_{\mathrm{v}\alpha} - b_{\mathrm{v\,eff}}\left(\tan\gamma + \tan\beta_{\mathrm{vb}}\right)\right]\cos\beta_{\mathrm{vb}}$ (D.65)	−5,307 mm	E: A.32		
	$f_{\max} = f_{\max\mathrm{B}}$ since $f_{\max\mathrm{B}} > f_{\max 0}$ (D.66)	26,173 mm			
Theoretical length of contact line	$l_{\mathrm{b}0} = \sqrt{\left(x_1 - x_2\right)^2 + \left(y_1 - y_2\right)^2}$ (D.67)	39,276 mm	E: A.27		
Theoretical length of middle contact line calculated with $f = f_{\mathrm{m}}$ for contact stress as specified in ISO 10300−2:2014, 6.1	$x_1 = \frac{f\cos\beta_{\mathrm{vb}} + \tan\beta_{\mathrm{vb}}\left(f\sin\beta_{\mathrm{vb}} + \frac{b_{\mathrm{v\,eff}}}{2}\right) + \frac{1}{2}\left(g_{\mathrm{v}\alpha} + b_{\mathrm{v\,eff}}\tan\gamma\right)}{\tan\gamma + \tan\beta_{\mathrm{vb}}}$ (D.68)	40,955 mm	E: A.28		
	$x_2 = \frac{f\cos\beta_{\mathrm{vb}} + \tan\beta_{\mathrm{vb}}\left(f\sin\beta_{\mathrm{vb}} + \frac{b_{\mathrm{v\,eff}}}{2}\right) - \frac{1}{2}\left(g_{\mathrm{v}\alpha} - b_{\mathrm{v\,eff}}\tan\gamma\right)}{\tan\gamma + \tan\beta_{\mathrm{vb}}}$ (D.69) NOTE ISO 10300−1:2014, Formula (A.29) is a misprint. The operator in the second parenthesis should be "−".	8,304 mm	E: A.29		
	$y_1 = -x_1\tan\beta_{\mathrm{vb}} + f\cos\beta_{\mathrm{vb}} + \tan\beta_{\mathrm{vb}}\left(f\sin\beta_{\mathrm{vb}} + \frac{b_{\mathrm{v\,eff}}}{2}\right)$ (D.70)	−10,915 mm	E: A.30		
	$y_2 = -x_2\tan\beta_{\mathrm{vb}} + f\cos\beta_{\mathrm{vb}} + \tan\beta_{\mathrm{vb}}\left(f\sin\beta_{\mathrm{vb}} + \frac{b_{\mathrm{v\,eff}}}{2}\right)$ (D.71)	10,915 mm	E: A.30		

Table D.6 — *continued*

Description	Formula		Result	References to		
				ISO 10300-1	ISO 10300-2	ISO 10300-3
				E: Formula	T: Table	F: Figure
Correction factor	$C_{\text{lb}} = \sqrt{\left[1-\left(\frac{f}{f_{\max}}\right)^2\right]\left(1-\sqrt{\frac{b_{\text{veff}}}{b_{\text{v}}}}\right)^2}$	(D.72)	0,078	E: A.34		
Length of contact line	$l_{\text{bm}} = l_{\text{b0}}\left(1-C_{\text{lb}}\right)$	(D.73)	36,211 mm	E: A.26		
Load sharing factor						
Exponent for calculation of parabolic distribution of peak loads	e	(D.74)	3,000		T: 3	
Related peak load	$p^* = 1-\left(\frac{\lvert f\rvert}{\lvert f_{\max}\rvert}\right)^e$	(D.75)			E: 7 F: 2	
Related peak load at tip contact line	$p_{\text{t}}{}^* = 1-\left(\frac{\lvert f_{\text{t}}\rvert}{\lvert f_{\max}\rvert}\right)^e$	(D.76)	0,684		E: 7	
Related peak load at middle contact line	$p_{\text{m}}{}^* = 1-\left(\frac{\lvert f_{\text{m}}\rvert}{\lvert f_{\max}\rvert}\right)^e$	(D.77)	1,000		E: 7	
Related peak load at root contact line	$p_{\text{r}}{}^* = 1-\left(\frac{\lvert f_{\text{r}}\rvert}{\lvert f_{\max}\rvert}\right)^e$	(D.78)	0,684		E: 7	
Related area	$A^* = \frac{1}{4} p^* l_{\text{b}} \pi$	(D.79)			E: 8 F: 2	