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BSI Standards Publication

Gears — Calculation of load capacity of worm gears

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National foreword

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Gears — Calculation of load capacity of worm gears

Engrenages — Calcul de la capacité de charge des engrenages à vis

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Foreword

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

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

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

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

This document was prepared by Technical committee ISO/TC 60, *Gears*, Subcommittee SC 1, *Nomenclature and wormgearing*.

This first edition cancels and replaces ISO/TR 14521:2010, which has been technically revised.

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

— the orginal Clause 6 which focused on geometry has been deleted and ISO/TR 10828 has been referenced.

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

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Introduction

This document was developed for the rating and design of enclosed or open single enveloping worm gears with cylindrical worms, and worm-geared motors having either solid or hollow output shafts.

This document is only applicable when the flanks of the worm wheel teeth are conjugate to those of the worm threads.

The particular shapes of the rack profiles from tip to root do not affect the conjugacy when the worm and worm wheel hobs have the same profiles; thus worm wheels have proper contact with worms and the motions of worm gear pairs are uniform.

This document can apply to wormgearing with cylindrical helicoidal worms as defined in ISO/TR 10828 having the following thread forms: A, C, I, N, K.

Other than those mentioned in the three preceding paragraphs, no restrictions are placed on the manufacturing methods used.

In order to ensure proper mating and because of the many different thread profiles in use, it is generally desirable that worms and worm wheels be supplied by the same manufacturer.

In this document, the permissible torque for a worm gear is limited by considerations of surface stress (conveniently referred to as wear or pitting) or bending stress (referred to as strength) in both worm threads and worm wheel teeth, deflection of worm or thermal limitation.

Consequently, the load capacity of a pair of gears is determined using calculations concerned with all criteria described in the scope and 6.4. The permissible torque on the worm wheel is the least of the calculated values.

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Gears — Calculation of load capacity of worm gears

1 Scope

This document specifies formulae for calculating the load capacity of cylindrical worm gears and covers load ratings associated with wear, pitting, worm deflection, tooth breakage and temperature. Scuffing and other failure modes are not covered by this document.

The load rating and design procedures are only valid for tooth surface sliding velocities, $\vec{v}_g$, less than or equal to 25 m/s and contact ratios greater than 2,1. For wear, load rating and design procedures are only valid for tooth surface sliding velocities which are above 0,1 m/s. The rules and recommendations for the dimensioning, lubricants or materials selected by this document only apply to centre distances of 50 mm and larger. For centre distances below 50 mm, method A applies.

The choice of appropriate methods of calculation requires knowledge and experience. This document is intended for use by experienced gear designers who can make informed judgements concerning factors. It is not intended for use by engineers who lack the necessary experience. See 4.7.

WARNING — The geometry of worm gears is complex, therefore the user of this document is encouraged to make sure that a valid working geometry has been established.

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

ISO 1122-2, *Vocabulary of gear terms — Part 2: Definitions related to worm gear geometry*

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

DIN 3974-1, *Accuracy of worms and wormgears — Part 1: General bases*

DIN 3974-2, *Accuracy of worms and wormgears — Part 2: Tolerances for individual errors*

3 Terms, definitions and symbols

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 1122-1, ISO 1122-2 and the following 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 http://www.electropedia.org/

3.1.1
actual gear
worm gear set designed by this document

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3.2 Symbols

NOTE Where applicable, the symbols are in accordance with ISO 701.

Table 1 — Symbols for worm gears

Symbols	Description	Unit	Figure	Formula
a	centre distance	mm	Figure 1	
a_1	centre distance of the gear concerned	mm	Figure 1	
a_0, a_1, a_2	oil sump temperature coefficients			(118) to (124)
a_T	centre distance of standard reference gear	mm	Figure 1	
a_V	centre distance of a gear operating or test experiences are available	mm	Table 4	
b_{2H}	effective wheel facewidth	mm		
$b_{2H,std}$	standard effective worm wheel facewidth	mm		(10)
b_{2R}	wheel rim width	mm		(132)
b_H	half Hertzian contact width	mm	Annex D	(D.2)
c_k	coefficient for heat transition coefficient			(133)
c_{oil}	specific heat capacity of the oil (for temperature calculation with spray lubrication)	Ws/(kg.K)		(128)
c_α	proximity value for the viscosity pressure exponent α	m^2/N		(22)/(24)
$d_{a\,1}$	worm tip diameter	mm		(89)
$d_{a\,2}$	worm wheel throat diameter	mm		
$d_{e\,2}$	worm wheel outside diameter	mm		
$\vec{dF}$	force transmitted by a segment of the contact line	N	Figure B.2	(B.3)
dl	length of contact line segment	mm		(B.1) to (B.6)
$d_{f\,1}$	worm root diameter	mm		(104)
$d_{f\,2}$	worm wheel root diameter	mm		(111)
d_{m1}	worm reference diameter	mm		
d_{m2}	worm wheel reference diameter	mm		(41) to (43)
d_{m1T}	reference diameter of the worm, from standard reference gear	mm	Table 4	(44), (45)
d_{m2T}	reference diameter of the wheel, from standard reference gear	mm	Table 4	
$\vec{e}_x$	unit vector pointing in direction of the x-axis	mm		(B.4)
f_h	worm wheel face width factor for the parameter for the minimum mean lubricant film thickness	—		(16)
f_p	worm wheel face width factor for the parameter for the mean Hertzian stress	—		(17)
Δf	relative deviation between a quantity of the gear concerned and a reference gear	—	Figure 1	
Δf_T	relative deviation between the centre distance of the gear concerned and the standard reference gear	—	Figure 1	
Δf_V	relative deviation between the centre distance of the gear concerned and a gear operating or test experiences are available	—	Figure 1	

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Table 1 *(continued)*

Symbols	Description	Unit	Figure	Formula
h_{am1}	worm tooth reference addendum in axial section	mm		(86)
h_{min}	minimum lubricant film thickness	µm		(C.1)
$h_{min\ m}$	minimum mean lubricant film thickness	µm		(21)
h^*	parameter for minimum mean lubricant film thickness	—		(14)/(15)
h_T^*	parameter for minimum mean lubricant film thickness of the standard reference gear	—	Table 4	
k	lubricant constant	1/K		(27)/(29)
l_1	spacing of the worm shaft bearings	mm		(103)
l_{11}, l_{12}	bearing spacing of the worm shaft	mm	Figure 5	(103)
$m_{x\,1}$	axial module	mm		
Δm_{lim}	material loss limit	mg		(88)
Δs	tooth thickness loss	mm		(111)
Δs_{lim}	allowable tooth thickness loss	mm		(87)
$\vec{n}$	normal vector			(B.5)
n_1	rotational speed of the worm shaft	min^{-1}		
p_H	Hertzian stress	N/mm^2		(B.1)/(B.6)
p_{Hm}	Hertzian stress; mean value for the total contact area	N/mm^2		(B.7)
p_m^*	parameter for the mean Hertzian stress	—		(11)/(12)/(B.8)
p_{mT}^*	parameter for the mean Hertzian stress of the standard reference gear	—	Table 4	
q_1	diameter quotient	mm		
$\vec{r}$	radius from the axis of the worm wheel to the contact point B	mm		(B.4)
s_{f2}	mean tooth root thickness of the wheel teeth in the spur section	mm		(111)
s_{ft2}	mean tooth root thickness of the wheel teeth in the spur section	mm		(111)
s_{gB}	sliding path of the worm flanks within the Hertzian contact of the wheel flank per number of cycles of the wheel, around the contact point (local value)	mm		(D.3)/(D.5)
s_{gm}	mean sliding path	mm		(D.7)
s_{m2}	tooth thickness at the reference diameter of the worm wheel	mm		(111)
s_K	rim thickness	mm	Figure 6	(113)
s_{Wm}	wear path inside of the required life expectancy	mm		(30)/(D.1)
s_{mx1}	worm tooth thickness in axial section	mm		
s_{mx1}^*	worm tooth thickness in axial section coefficient	—		(111)
s^*	parameter for the mean sliding path	—		(17)/(18)/(D.8)
s_T^*	parameter for the mean sliding path of the standard reference gear	—	Table 4	
Δs	tooth thickness loss			(111)

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Table 1 *(continued)*

Symbols	Description	Unit	Figure	Formula
Δs_{lim}	allowable tooth thickness loss			(87)/(111)
$t_{contact}$	time of contact	s		(D.2)
u	gear ratio			(1)
u_T	gear ratio of the standard reference gear		Table 4	
$\vec{v}_1$	velocity of a flank point of the worm	m/s	Figure B.1	
$\vec{v}_2$	velocity of a flank point of a worm wheel	m/s	Figure B.1	
v_{1n}	worm velocity component normal to the contact line	m/s	Figure B.2	
v_{2n}	wheel velocity component normal to the contact line	m/s	Figure B.2	(D.2)
$\vec{v}_{gB}$	sliding velocity normal to the line of contact in flank direction	m/s		(D.3)/(D.5)/(D.6)
v_g	sliding velocity at reference diameter	m/s		(9)/(49)/(50)/(51)/(H.2)/(H.3)/(H.5)
v_{ref}	reference sliding velocity	m/s		(H.2) to (H.5)
$v_{\Sigma n}$	sum velocity in normal direction	m/s		(11)/(C.4)
x_2	worm wheel profile shift coefficient	—		
z_1	number of threads in worm	—		
z_2	number of teeth in worm wheel	—		
A	coefficient for kinematic viscosity			(33)
A_{fl}	total flank surface of the worm wheel	mm^2		(89)
A_R	dominant cooled surface of the gear set	m^2		(132)
B	coefficient for kinematic viscosity	—		(34)
B	coefficient for h^*	mm		(14)
E_1	modulus of elasticity of the worm	N/mm^2		
E_2	modulus of elasticity of the worm wheel	N/mm^2		
E_{red}	equivalent modulus of elasticity	N/mm^2	Table 4	(20)
F_{xm1}	axial force to the worm shaft	N		(4)/(7)
F_{xm2}	axial force to the worm wheel	N		(3)/(6)
F_{rm1}	radial force to the worm shaft	N		(5)
F_{rm2}	radial force to the worm wheel	N		(11)
F_{tm1}	circumferencial or tangential force to the worm shaft	N		(4)/(6)
F_{tm2}	circumferencial or tangential force to the worm wheel	N		(3)/(7)
dF/dl	specific loading	N/mm		(C.5)
J_{OT}	reference wear intensity	—	Figure 4	(69) to (79)
$J_{OI}, J_{OII}, J_{OIII}$	reference wear intensity for stage I, II, III	—		(H.6) to (H.7)
J_W	wear intensity	—		(68)
J_{WP}	wear intensity	—		(H.6)
K_n	rotational speed factor/wheel bulk temperature	—		(135)
$K_{H\alpha}$	transverse load distribution factor	—		6.2.3
$K_{H\beta}$	longitudinal load distribution factor	—		6.2.3
K_S	size factor/wheel bulk temperature	—		(137)

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Table 1 *(continued)*

Symbols	Description	Unit	Figure	Formula
K_A	application factor	—		6.2.1
K_v	dynamic factor	—		6.2.2
K_W	lubricant film thickness parameter	—		(80)
K_ν	viscosity factor/wheel bulk temperature	—		(136)
K_1	factor	—		(G.5)
L_h	life time	h		
N_L	number of stress cycles of the worm wheel	—		(31)
N_{LI}, N_{LII}, N_{LIII}	number of stress cycles of the worm wheel for stage I to III	—		(H.1)
N_S	number of starts per hour	—		(70)
P_1	input power to the worm shaft	W		
P_2	output power from the worm wheel shaft	W		
P_K	cooling capacity of the oil with spray lubrication	W		(127) (125)
P_V	total power loss of the worm gear unit	W		(38)
P_{VO}	idle running power loss	W		(38)/(39)/(G.1)
$P_{Vz1\text{-}2}$	meshing power loss in reducer	W		(62)
$P_{Vz2\text{-}1}$	meshing power loss in increaser	W		(64)
P_{VD}	sealing power loss	W		(44)/(45)
P_{VLP}	bearing power loss through loading	W		(40) to (43)
Q_{oil}	spray quantity	m^3/s		(127)
Ra_1	arithmetic mean roughness for worm	µm	Table 4	
Ra_T	arithmetic mean roughness for reference gear	µm		(62)
Rz_1	mean roughness depth	µm		7.4.6
S_F	tooth breakage safety factor	—		(106)
$S_{F\,min}$	minimum tooth breakage safety factor	—		(107)
S_H	pitting safety factor	—		(91)
S_{Hmin}	minimum pitting safety factor	—		(92)
S_T	temperature safety factor	—		(115)/(125)
$S_{T\,min}$	minimum temperature safety factor	—		(116)/(126)
S_W	wear safety factor	—		(65)
$S_{W\,min}$	minimum wear safety factor	—		(66)
S_δ	deflection safety factor	—		(101)
$S_{\delta\,min}$	limit of deflection safety factor	—		(102)
T_1	input torque to the worm shaft	Nm		(1)
T_{1N}	nominal input torque to the worm shaft	Nm		(1)
T_2	output torque from the worm wheel	Nm		(2)/(B.4)/(B.5)
T_{2N}	nominal output torque from the worm wheel	Nm		(2)
V_{SUMn}	sum of velocities at contact point			(C.1)
W_H	—			(84)/(85)
W_{ML}	material — lubricant factor	—	Table 7	
W_{NS}	start factor	—		(83)
W_P	damage factor	—		(H.8)
W_S	lubricant structure factor	—		(81)/(82)

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Table 1 *(continued)*

Symbols	Description	Unit	Figure	Formula
Y_F	form factor/tooth breakage	—		(110)
Y_G	geometry factor/coefficient of friction	—		(59)/(60)
Y_K	rim thickness factor/tooth breakage	—		(113)
Y_{NL}	life factor/tooth breakage	—	Figure 7 a)/b)	Table 11
Y_R	roughness factor/coefficient of friction	—		(61)/(62)
Y_S	size factor/coefficient of friction	—		(57)/(58)
Y_W	material factor/coefficient of friction	—		
Y_ε	contact factor/tooth breakage	—		(109)
$Y\gamma$	lead factor/tooth breakage	—		(112)
Z_h	life factor/pitting	—		(94)
Z_{oil}	lubricant factor/pitting	—		(100)
Z_S	size factor/pitting	—		(96)/(97)
Z_u	gear ratio factor	—		(98)/(99)
Z_v	velocity factor/pitting	—		(95)
α	pressure viscosity factor	m^2/N		6.6
α_L	heat transition coefficient for immersed wheel teeth	$W/(m^2K)$		(133)
α_n	normal pressure angle	°		
α_0	normal pressure angle			(5), (86)
γ_{m1}	reference lead angle of worm	°		(86)
δ_{lim}	limiting value of deflection	mm		(105)
δ_m	incurred deflection	mm		(103)/(104)
δ_{Wn}	flank loss from wheel through abrasive wear in the normal section	mm		(67)
$\delta_{W\,lim}$	limiting value of flank loss	mm		(90)
$\delta_{W\,lim\,n}$	limiting value of flank loss in normal section	mm		(86) to (88)
η_{ges}	total efficiency in reducer	—		(35)
$\eta_{ges1\text{-}2}$	total efficiency worm driving wheel	—		(35)
$\eta_{ges2\text{-}1}$	total efficiency wheel driving worm	—		(36)
η'_{ges}	total efficiency in increaser	—		(36)
$\eta_{z1\text{-}2}$	gear efficiency in reducer	—		(46)/(63)
$\eta_{z2\text{-}1}$	gear efficiency in increaser	—		(47)/(64)
η_{0M}	dynamic viscosity of lubricant at ambient pressure and wheel bulk temperature	Ns/m^2		(25)/(C.1)
θ	temperature	°C		
$\Delta\theta$	temperature difference between oil sump and worm wheel bulk temperature	°C		(131)
θ_{in}	oil entrance temperature	°C		(129)
θ_0	ambient temperature	°C		
θ_{oil}	spray temperature	°C		(129)
$\Delta\theta_{oil}$	oil temperature difference between input and output cooling system	°C		(129)
θ_M	wheel bulk temperature	°C		(130)/(134)
θ_S	oil sump temperature	°C		(117)/(119)
$\theta_{S\,lim}$	limiting value of oil sump temperature	°C		(115)

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Table 1 *(continued)*

Symbols	Description	Unit	Figure	Formula
μ_{0T}	base coefficient of friction	—		(49) to (52)
μ_{zm}	mean tooth coefficient of friction	—		(48)
ν_1	POISSON ratio of the worm	—		(20)
ν_2	POISSON ratio for the worm wheel	—		(20)
ν_θ	kinematic viscosity at oil temperature θ	mm^2/s		(32)
ν_{40}	kinematic viscosity at 40 °C	mm^2/s		(32)
ν_{100}	kinematic viscosity at 100 °C	mm^2/s		
ν_M	kinematic viscosity at wheel bulk temperature	mm^2/s		(25)
ρ_1, ρ_2	local radius of curvature	mm		(B.2)
ρ_{oil}	lubricant density	kg/dm^3		(127)
ρ_g	friction angle for the tooth coefficient of friction			(5)
ρ_{oil15}	lubricant density at 15 °C	kg/dm^3		(25)
ρ_{oilM}	lubricant density at wheel bulk temperature	kg/dm^3		(26)
ρ_{red}	equivalent radius of curvature	mm		(B.2)
ρ_z	friction angle for the tooth coefficient of friction	°		(5)
ρ_{Rad}	material density of the wheel	mg/mm^3	Table 8	(88)
$\sigma_{H\,lim\,T}$	pitting strength	N/mm^2	Table 9	
σ_{Hm}	mean contact stress	N/mm^2		(19) (91)
σ_{HG}	limiting value for the mean contact stress	N/mm^2		(93) (91)
τ_F	shear stress at tooth root	N/mm^2		(108) (106)
$\tau_{F\,lim\,T}$	shear endurance strength	N/mm^2	Table 10	
τ_{FG}	limiting value for shear stress at tooth root	N/mm^2		(114) (106)

4 General consideration

4.1 Worm gear load capacity rating criteria

The load capacity of a worm gear corresponds to the torque (or the power) which can be transmitted without the occurrence of tooth breakage or the appearance of excessive damage on the active flanks of the teeth during the design life of the gearing.

Conditions shown in Table 2 can limit the rated load capacity.

Table 2 — Significant factors affecting failure mode and performance (valid for same gear set)

Influencing factors	Failure modes					Efficiency
	Wear	Pitting	Tooth-breakage	Worm shaft deflection	Scuffing	
Load (Hertzian pressure)	x	x	x	x	x	x
Worm speed	x	x			x	x
Oil viscosity	x	x			x	x
Contact Pattern	x	x	x		x	x
Worm surface waviness and roughness	x	x			x	x

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Table 2 *(continued)*

Influencing factors	Failure modes					Efficiency
	Wear	Pitting	Tooth-breakage	Worm shaft deflection	Scuffing	
Oil film shearing value					x	x
Shear stress			x			

NOTE Worm thread breakage can occur as a result of bending fatigue of worm threads but is not covered by this document.

— **wear:** damage usually appears on the tooth flanks of bronze worm wheels and is also influenced by the number of starts per hour;

— **pitting:** this form of damage may appear on the flanks of worm wheel teeth; its development is strongly influenced by the load transmitted and the load-sharing conditions;

— **tooth breakage:** shear failure of worm wheel teeth or worm threads can occur when teeth become thin due to wear or overload;

— **worm shaft deflection:** excessive deformation under load can modify the contact pattern between worm and worm wheel;

— **scuffing:** this form of damage often appears suddenly; it is strongly influenced by transmitted load, sliding velocities and the conditions of lubrication.

Factors such as material properties, meshing conditions, (e.g. contact pattern under load) and lubrication greatly impact Hertzian pressure along the lines of contact.

The different rating criteria are calculated independently and not in combination (see Annex J). For a given worm gear pair, the zone of contact can change with loading. At a steady load, fatigue pits can develop which may subsequently be reduced by wear. This can be followed by further pitting, additional wear or a stable condition.

The load capacity of wormgearing is determined by calculations dealing with permissible stresses for pitting and wear, the deflection in the worm, shafts, and the operating temperature. Excessively high working temperature leads to accelerated degradation of the worm gear lubricant. The permissible torque shall be determined from the lowest value of the calculated load capacity.

4.2 Basis of the method

The calculation methods are partly based on investigations of test gears (see, standard reference gear, 4.4), and partly on application experience. Investigations on test gears are mainly ascertained through varied test conditions and verified through practical experience.

4.3 Concept of absolute and relative parameters

The formulae used for the calculation procedure in this document lead to either an absolute form (calculation with absolute parameters) or to a relative form (calculation with relative parameters).

Absolute parameters: The calculation is used when no specific tests are available. The precision of the gear calculations is improved as the differences concerning geometric dimensions, the operating conditions, material and lubricant, to those taken from the standard reference gears are decreased.

Relative parameters: The calculation offers the possibility to use investigation results in the corresponding calculation process directly. This enables the calculation procedure of the specific results to be adapted.

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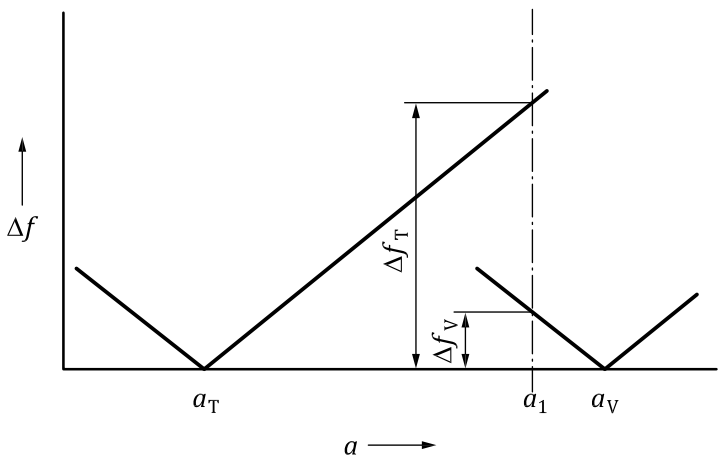
As the dimensions, materials, lubricants and operating conditions of the gear to be calculated approach those of the standard reference gear or, if the corresponding test gear data is available, the deviation is decreased. Figure 1 shows an example of the influence of the centre distance.

To minimize the deviation between the designed gear and the reference gear (or/and test gear if applicable), care should be taken to minimize the difference between both gears specific characteristics including dimensions, materials, lubrication and operating conditions. The gear being designed has a centre distance of a_1, which is clearly a deviation from that of the standard reference gear a_T. Thus a relative deviation Δf_T is given. Furthermore, test results are available for a gear with a centre distance of a_V. In calibrating the calculation procedure to this centre distance, a deviation of Δf_V is yielded (by linear regression). This deviation is significantly smaller than the deviation Δf_T, since the concerned gear is clearly more similar to the test gear than to the reference gear. Therefore, if possible, the limiting values should be determined from the operating or test experience in which each operating condition (tooth form, material, lubricant, rotational speed, loading, etc.) is as similar as possible to that of the gear in question. In calculating the load capacity or various factors, more methods are allowed (see 4.4).

The use of the calculating procedure requires for each case, a realistic estimation of all influential factors, especially the loading, ambient conditions, damage risk (probability of damage) etc. The recommended minimum safety factors shall be increased accordingly (see Annex I).

For a calculation example, see Annex G.



Key

a centre distance

Δf relative deviation between a quantity of the gear concerned and a reference gear

a_1 centre distance of the gear concerned

a_T centre distance of the standard reference gear (see Table 4)

a_V centre distance of a gear operating or test experiences are available

Δf_T relative deviation between the centre distance of the gear concerned and the standard reference gear

Δf_V relative deviation between the centre distance of the gear concerned and a gear operating or test experiences are available

Figure 1 — Deviation as a function of the centre distance (based on linear regression)

4.4 Applicability

The technical information provided in this document is based on the following:

— knowledge and judgement acquired over years of experience in designing, manufacturing and operating wormgearing;

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— results of bench testing I-worm gears having centre distances from 65 mm to 160 mm and transmission ratios for 4,8 to 50.

Three methods are provided for the calculation of each parameter:

— method A (the most accurate derived from experimental and measurement data);

— method B (calculated parameters derived using numerical methods);

— method C (approximated methods).

4.5 Validity

The validity for the various parts of the calculating procedure of this document is restricted to conditions where operating experience already exists. If further test results are available, the scope of validity can be extended by calibration with a valid calculation procedure with respect to the type and extent of the additional testing.

The rating and design procedures are valid for the following:

— **flank forms:** A, N, K, I, C according to ISO/TR 10828;

— **worm rotational speed:** up to 5 000 r/min;

— **gear ratio:** from 5 to 100;

— **sliding velocity:** between tooth surfaces from 0,1 m/s to 25 m/s for wear;

— **shaft angle:** of 90°;

— **accuracy grade:** the worm accuracy grade (according to DIN 3974) should be specified to be one accuracy grade better than wheel;

— **worm materials**:

  — case hardened steels, case hardened (HRC = 58 to 62);

  — through hardened steels, flame or induction hardened (HRC = 50 to 56);

  — the calculation procedures are based on experiments carried out with worms made of 16MnCr5 (case hardened), no studies having been carried out for other materials yet. However, in the case of sufficient surface hardness (as above), hardness penetration depth, core hardness and correct heat treatment, the calculation procedures of the above-mentioned materials can be used;

  — other materials and heat treatments (such as nitriding steels, gas nitrided) can be used with sufficient experience in accordance with method A;

— **worm wheel materials:** material and notes based on experience which are as listed in Table 3 should be used;

NOTE Other materials not listed in Table 3 can be used.

— **centre distance**:

  — for temperature calculation (wear, pitting) the range is from 60 mm up to 500 mm centre distance;

  — for other criteria (pitting, tooth breakage) the range is between 50 mm up to 500 mm centre distance.

— **lubricants**:

  — mild additive CLP-oils according to ISO 6743-6;

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— compounded oils (steam cylinder oils), no test results available, included as a mineral oil in this document;

— polyglycols;

— polyalphaolefines based on limited test results.

The calculations are based essentially on studies carried out with I-worm gears. The results have been converted to worm gears with other flank forms by means of similarity considerations.

Table 3 — Common worm wheel materials

Worm material	16MnCr5 case hardened					
Wheel material	GZ-CuSn12[a,c]	GZ-CuSn-12Ni2[a]	GC-CuSn-12Ni2	GZ-CuAl-10Ni[a,b]	GJS-400-15	GJL-250
	Bronze	Nickel bronze	Nickel bronze	Aluminium bronze	Spheroidal graphite cast iron	Grey cast iron
Wear	+	+	+	o	o	o
Pitting	+	+	+	o	–	–
Tooth Breakage	+	+	–	+	+	+
Temperature	+	+	+	+	–	–

Key

+ covered study available

o limited study

– empirical values

[a] Bronze should be homogenous and free from blow holes in the gearing region. Average grain size <150 µm. Grain size variation may have a significant influence on the capacity, resulting in variation of 20 % or more, if not maintained consistent. For the determination of grain size, a minimum of 50 grains is needed to be observed on the area of active flanks.

[b] Forged aluminium bronze can be treated like GZ-CuAl10Ni.

[c] Forged phosphor bronze can be treated like GZ-CuSn12.

NOTE 1 See EN 1982, EN 1563 and EN 1561 for the material designation.

NOTE 2 For low sliding velocities, v_g < 0,5 m/s.

4.6 System considerations

In this document no attempt is made to address complete drive systems, backdriving, torsional vibrations, critical speeds or other types of vibrations which may affect the operation of worm gears.

4.7 Calculation methods A, B, C

4.7.1 Generality on methods A, B and C

4.7.1.1 General

This document contains influential factors based on research results and operational experiences. The factors are differentiated with reference to:

a) Factors which are concerned with meshing geometry or compatibility shall be calculated using given formulae.

b) Factors which are multi–influenced, or are independent of each other (which do not however affect each other), or both. These include factors which affect an influence on the permitted stress.

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The factors can be determined by different methods which are, where necessary, characterised by additional parameters A to C. Method A is more precise than B and so on. It is recommended to use the most precise method. With important operations, the method used should be agreed upon by the manufacturer and the purchaser.

4.7.1.2 Method A

Here the factor is determined through exact measurement, extensive mathematical analysis of the transfer system or existing operational experiences. Because of this, all gear and loading data shall be known.

4.7.1.3 Method B

The factors are determined by a method which, for most applications, is sufficiently precise. The assumptions under which they are developed are stated. In determining a specific factor, the application shall be within the range of the given assumptions.

4.7.1.4 Method C

For some factors additional simplified approximation procedures are specified. The assumptions under which they are developed are stated. In determining a specific factor, the application shall be within the range of the given assumptions.

4.7.2 Notes on numerical formulae

The formulae specified in this document shall be calculated in the specified units (see Table 1).

4.7.3 Base conditions, interaction

— Wear:

  This procedure is in accordance with the investigation described in Reference [23] and is based on practical experience.

— Pitting damage:

  The procedure follows the investigation described in Reference [30] and takes into consideration practical experience. The Hertzian stress is an essential influencing variable to the physics causing pitting development, in addition to this however, other influences are also of importance e.g. the tangential forces and the effects of slip and roll movement. These additional influences are not considered here, therefore, the limiting value of the load capacity (surface stress values) should be developed through tests on worm gears or through evaluation of relevant operational results. Allowable values, resulting from specimen examination (e.g. disk tests), only allow for relative statements and may only be used for the load capacity calculation if scientific investigation of this manner has been completed.

— Interaction between scuffing and wear:

  Short term scuffing incurred by bronze wheels can be "healed". This healing is only possible through wear, however this document does not provide estimations for this wear behaviour.

— Interaction between wear and pitting:

  It is known from practical testing that pitting development can be stabilised by increased wear. Pitting can also be stopped through continual wear.

  At higher wear intensities, that is when the wear capacity limits the life endurance, pitting is a secondary consideration. Alternatively with higher pitting, wear is not the limiting criteria.

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— Interaction between wear and tooth breakage:

The calculation of the tooth breakage factor takes into account that the tooth thickness of the worm wheel is decreased by wear.

4.7.4 Other notes

Calculated load capacities are valid for the following conditions:

— continuous operation,

— constant loading,

— correct alignment,

— the contact area width approaches the full facewidth after running in.

All other loaded conditions are outside the scope of this document.

4.8 Standard reference gear

With certain calculation procedures the absolute calculating formulae are similar to the relative calculating formulae. If the adopted sizes of the corresponding standard reference gear (subscript T) shall be employed, the relative calculating formulae can be transferred to the absolute calculating formulae (see Table 4).

Table 4 — Main data from the standard reference gear

Centre distance a_T	100 mm
Gear ratio u_T	20,5
Worm reference diameter d_{m1T}	36 mm
Worm wheel reference diameter d_{m2T}	164 mm
Parameter for mean Hertzian stress p_{mT}*	0,92 (Formula (53))
Parameter for mean lubricant film thickness h_T*	0,07 (Formula (55))
Parameter for mean sliding path s_T*	30,8 (Formula (59))
Worm material	16MnCr5 case hardened
Wheel material	GZ-CuSn12Ni2
Lubricant density	1,048 kg/dm^3
Flank form	I (ISO/TR 10828)
Normal pressure angle	20°
Worm surface roughness Ra_1	0,5 µm
Equivalent modulus of elasticity E_{red}	150 622 N/mm^2

5 Required date for calculation

5.1 Input variable

For the calculation, the following variables shall be known.

Geometry data:

— Center distance, a

— Effective wheel facewidth, b_{2H}, see ISO/TR 10828:2015, Figure 4

— Wheel rim width, b_{2R}, see ISO/TR 10828:2015, Figure 4

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— Reference diameter, d_{m1}, d_{m2}

— Axial module of the worm, m_{x1}

— diameter quotient, q_1

— Number of teeth, z_1, z_2

— Worm wheel profile shift coefficient, x_2

— Normal pressure angle, α_n

— Profile flank form, (A, N, K, I, C)

— Worm wheel outside diameter, d_{e2}

— Rim thickness, s_k, see Figure 6

— Worm tip diameter, d_{a1}

— Worm thread thickness in axial section coefficient, s_{mx1}*

— Accuracy grade of pitch deviation by analogy with DIN 3974

— Root diameter d_{f2}

Loading:

— Nominal output torque, T_{2N}

— Application factor, K_A

— Rotational speed of worm, n_1

— Life time, L_h

— Number of start per hour, N_S

In order to calculate the efficiency, the power loss and the wear and pitting safety factors are also necessary:

— Worm and wheel material

— Lubricant data ρ_{oil}, $\nu_{40,\,\nu100}$

— Type of oil: mineral oil/polyglycol

— Type of lubrication: splash or spray lubrication

— Roughness of the worm flanks, Ra_1

— Immersed or not immersed worm wheel

— Type of bearing at the worm: located — non-located bearing; adjusted bearing

— Number of sealing rings at the worm

— Housing with fan or housing without fan

— Ambient temperature, θ_0

To calculate the deflection safety factor:

— Worm bearing spacing, l_1, or l_{11}, l_{12}

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5.2 Safety factors

Safety factors (calculated) are differentiated according to wear, S_W, pitting, S_H, deflection, S_δ, tooth breakage, S_F and maximum temperature, S_T. The defined minimum safety factors, S_W, S_H, S_δ, S_F and S_T shall not be reduced below values given in this document.

The safety factors shall be chosen after careful consideration of the following influences:

— how safe are the assumptions of the load concerned;

— how safe are the assumptions of the operating conditions concerned;

— what are the consequences of damage.

Higher safety factors may be agreed between manufacturer and purchaser.

6 Forces, speeds and parameters for the calculation of stresses

6.1 General

For the calculation of load capacity, the following forces, speeds and parameters are required. They are used to describe stress mechanisms for both tooth flank and tooth root.

All forces transferred to the gear shall be compiled as accurately as possible, when calculating the net applied force.

When calculating the tooth forces, both the external and internal influences shall be taken into account in accordance with 6.2.1 and ISO 6336-6.

6.2 Tooth forces

6.2.1 Application factor

The application factor, K_A, shall consider all forces externally introduced to the gear, in addition to the nominal forces described in 6.2.2. These extra forces depend on the driving machine characteristics and driven machine characteristics, the masses and elasticities in the output and drive lines (e.g. from shafts and clutches) and operating conditions. Guide values for K_A can be found in ISO 6336-6. In some circumstances it may be useful to calculate the tooth force components with nominal torque, for example for shaft or bearing calculations, in which case the application factor should be equal to 1 in Formulae (1) and (2).

6.2.2 Dynamic factor

Internal influences are covered by factor K_v.

The measurement of the tooth root stresses at different circumferential velocities (ω_2) has indicated that the amount of internally generated dynamic loads can be neglected ($K_v = 1$).

6.2.3 Load distribution factor

Influences of load distribution are covered by factors $K_{H\beta}$ and $K_{H\alpha}$.

The methods within this document assume that the worm gear has been accurately manufactured and run-in, such that there is an even load distribution over the face width (along the contact lines) and there is an even load distribution over the tooth flanks in mesh ($K_{H\alpha} = K_{H\beta} = 1$).

Variable torque, which results in varying worm deflections, however, may result in non-uniform load distribution along the contact lines with correspondingly increased wear during gear operation. In order to keep this effect to a minimum, a deflection safety factor is required (see Clause 10).

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6.2.4 Tooth force components

The torques required for the calculations of the following forces shall be calculated from the nominal input and output torques as follows:

$$T_1 = T_{1N} \cdot K_A \tag{1}$$

$$T_2 = T_{2N} \cdot K_A \tag{2}$$

The basis for the load capacity calculation is the rated torque of the driven machine, which is the operational torque for the heaviest working conditions. The nominal torque can also be taken from the motor provided that this torque corresponds with the permissible torque of the driven machine. If this is not the case another sensible definition should be chosen.

The tangential, axial and radial forces, $F_{tm1,2}$, $F_{xm1,2}$, $F_{rm1,2}$, acting on the worm and wheel are shown in Figure 2.

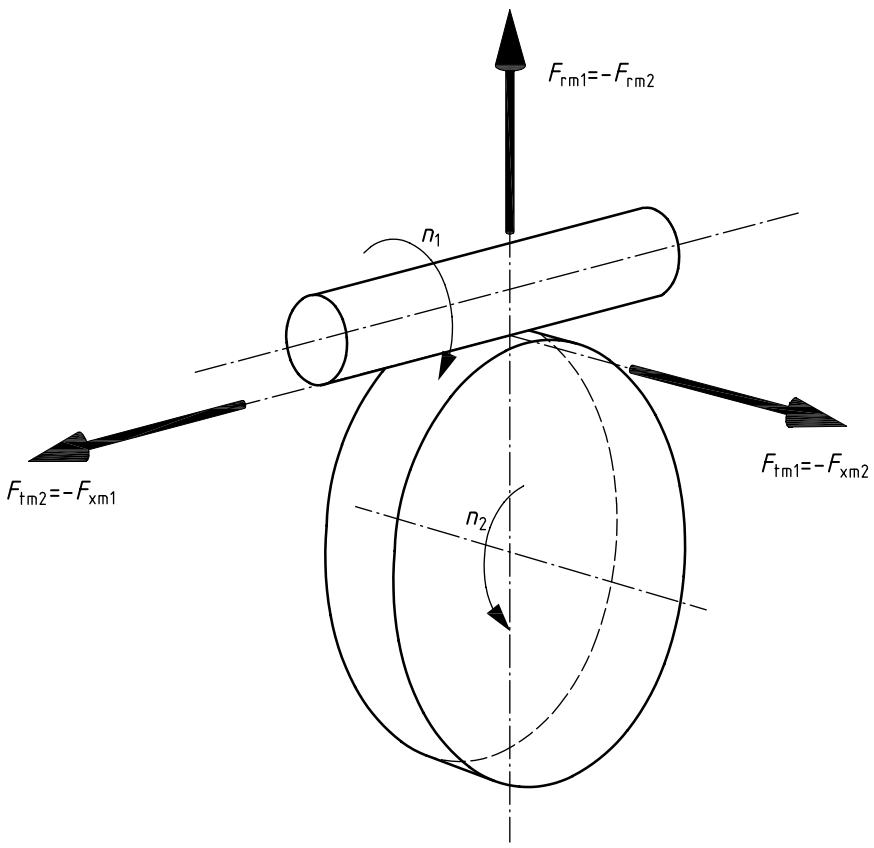

Figure 2 — Tooth load components

Worm driving the worm wheel:

If the calculation of tooth components with nominal torque is required, the application factor effect shall be removed from Formulae (3) to (8).

$$F_{tm1} = 2\,000 \cdot \frac{T_1}{d_{m1}} = 2\,000 \cdot \frac{T_2}{d_{m1} \cdot \eta_{ges} \cdot u} = -F_{xm2} \tag{3}$$

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$$F_{tm2} = 2\,000 \cdot \frac{T_2}{d_{m2}} = 2\,000 \cdot \frac{T_1 \cdot \eta_{ges} \cdot u}{d_{m2}} = -F_{xm1} \tag{4}$$

with η_{ges} according to Formula (35)

$$F_{rm1} = -F_{rm2} = F_{tm1} \cdot \frac{\tan\alpha_0}{\sin(\gamma_{m1} + \rho_z)} \tag{5}$$

with: $\rho_z = \arctan(\mu_{zm})$

μ_{zm} according to Formula (48)

Worm wheel driving the worm:

$$F_{tm1} = 2\,000 \cdot \frac{T_1}{d_{m1}} = 2\,000 \cdot \frac{T_2 \cdot \eta'_{ges}}{d_{m1} \cdot u} = -F_{xm2} \tag{6}$$

$$F_{tm2} = 2\,000 \cdot \frac{T_2}{d_{m2}} = 2\,000 \cdot \frac{T_1 \cdot u}{d_{m2} \cdot \eta'_{ges}} = -F_{xm1} \tag{7}$$

with η_{ges}' according to Formula (36)

$$F_{rm2} = -F_{rm1} = F_{tm2} \cdot \frac{\tan\alpha_n}{\cos(\gamma_{m1} - \rho_z)} \tag{8}$$

NOTE The relationship between input and output torque can be derived from Formulae (3) and (4) if the worm is driving or Formulae (6) and (7) if the worm is driven.

6.3 Sliding velocity at reference diameter

Due to the mostly large slip components in the circumferential direction, to calculate the load capacity it is sufficient to use the sliding velocity at the reference diameter v_g in flank direction:

$$v_g = \frac{d_{m1} \cdot n_1}{19\,098 \cdot \cos\gamma_{m1}} \tag{9}$$

6.4 Physical parameters

6.4.1 Generality on physical parameters

6.4.1.1 General

In order to judge the capacity of worm gears, three non-dimensional parameters based on the geometry of the gears are defined and called physical parameters. These physical parameters are p_m^*, for the mean Hertzian stress, h^*, for the mean lubricant film thickness, and s^*, for the mean sliding path. Since these physical parameters are dependent only on the geometry of the gears, size, loading and lubricant do not influence them. See Annex A and References [23] and [31] for description of their derivations.

The physical parameters are derived according to methods A, B and C.

NOTE 1 The use of method C does not eliminate the need to check that proper gear mesh occurs.

NOTE 2 A, I, N and K profiles of worms are considered as a unique profile.

NOTE 3 Method C is based on mathematical approximation to established Formulae (11) to (18).

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6.4.1.2 Method A

Currently there is no method to derive the physical parameters directly from experimental data or measurement of part dimensions.

6.4.1.3 Method B

The physical parameters are derived using numerical methods; for the basis of these parameters 6.4.2 and 6.4.3 and Annexes B, C, D, E and F shall be used. See also References [31] and [35].

6.4.1.4 Method C

NOTE Solutions for the physical parameters can be obtained through approximation formulae from computer — programming according to References [20], [31] and [35].

The formulae apply to flank form I but can also be used in an approximate manner for tooth forms K, A and N. The approximation formulae for C-worm drives are derived from Reference [24] and from operational experience. The approximation formulae indicated in 6.4.2, 6.4.3 and 6.4.4 shall apply to I-worm drives with α_n = 18 to 22°, x_2 = −0,5 to +1, $h_{am1} + h_{am2} \approx 2 \cdot m_{x1}$, C-worm drives with α_n = 20 to 24°, x_2 = 0 to +0,5, $h_{am1} + h_{am2} \approx 2 \cdot m_{x1}$ and $\rho/m_n \approx 5$ to 7.

For I — Wormdrive, the approximation formulae shall only be applied when the base diameter is smaller than the root diameter.

The approximation formulae are only valid for a worm wheel facewidth of:

$$b_{2H,std} = m_{x1}\left(\sqrt{q_1^2 - (q_1 - 3)^2} + 1\right) \tag{10}$$

For a smaller worm wheel face width the physical parameters p_m^* and h^* are on the unsafe side. Therefore higher safety factors for wear and pitting shall be used or the physical parameters p_m^* and h^* shall be calculated according Method B.

6.4.2 Parameter for the mean Hertzian stress

6.4.2.1 General

The mean Hertzian stress is a parameter of essential importance to the flank loading (see 4.7.3).

For critical applications (such as cranes, lifts, high precision gear set for rotary table and robotic application), method B is recommended for contact pressure calculation.

6.4.2.2 Parameter for the mean Hertzian stress — Method A

A parameter which describes the complex relationships between Hertzian stress and flank loading cannot be given at the moment.

6.4.2.3 Parameter for the mean Hertzian stress — Method B

The mean Hertzian stress used for the determination of the parameter is calculated by means of a computer program e.g. according to References [20], [31] and [35] under the assumption of the equal Hertzian pressure for all simultaneously meshed contact lines. The procedure is to define the contact and then find the curvature radii at the flanks on specific contact line sections. The Hertzian stress can be derived from equivalent cylinders along the contact lines. In each meshing zone more than one tooth is engaged. The Hertzian contact stress along these contact lines are assumed constant. The mean Hertzian stress p_{Hm}, is thus derived from all Hertzian stresses from all the meshing zones.

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The Hertzian stress shall be used in the development of the non-dimensional parameter, p_m^*. This Hertzian stress parameter is only dependant on the gear tooth geometry and is independent of the modulus of elasticity (E-module), material used and centre distance (size). The parameter, p_m^*, is used in Formula (19) for the derivation of the mean contact stress σ_{Hm}. For details of calculation of this parameter, Annex B shall be used.

6.4.2.4 Parameter for the mean Hertzian stress — Method C

From calculations according to method B a useful non-dimensional parameter for the mean Hertzian stress, p_m^* is derived.

For I, N, K, A worm drives:

$$p_m^* = 0{,}179\,4 + 0{,}238\,9 \cdot \frac{a}{d_{m1}} + 0{,}076\,1 \cdot x_2 \cdot |x_2|^{3{,}18} + 0{,}053\,6 \cdot q_1 - 0{,}003\,69 \cdot z_2 - 0{,}011\,36 \cdot \alpha_n + 44{,}981\,4 \cdot \frac{x_2 + 0{,}005\,657}{z_2} \cdot \left(\frac{z_1}{q_1}\right)^{2{,}687\,2} \tag{11}$$

For C-worm drives:

$$p_m^* = 0{,}140\,1 + 0{,}186\,6 \cdot \frac{a}{d_{m1}} + 0{,}059\,5 \cdot x_2 \cdot |x_2|^{3{,}18} + 0{,}041\,9 \cdot q_1 - 0{,}002\,88 \cdot z_2 - 0{,}008\,9 \cdot \alpha_n + 35{,}141\,7 \cdot \frac{x_2 + 0{,}005\,657}{z_2} \cdot \left(\frac{z_1}{q_1}\right)^{2{,}687\,2} \tag{12}$$

For worm wheel with smaller facewidth than $b_{2H,std}$, p_m^* shall be modified with the following formulae:

$$p_m^* = p_m^* \cdot f_p$$

$$f_p = \frac{14 \cdot b_{2H}^2 - \left(28 \cdot b_{2H,std} + m_{x1}\right) \cdot b_{2H} + 300 \cdot m_{x1}^2 + 14 \cdot b_{2H,std}^2 + b_{2H,std} \cdot m_{x1}}{300 \cdot m_{x1}^2} \tag{13}$$

This formula is only valid for $(b_{2H,std} - 2{,}5\ m_{x1}) \leq b_{2H} \leq b_{2H,std}$.

For smaller value than $(b_{2H,std} - 2{,}5\ m_{x1})$, Method B shall be used. For higher value than $b_{2H,std}$, $f_p = 1$ shall be used.

6.4.3 Parameter for the mean lubricant film thickness

6.4.3.1 General

The mean lubricant film thickness is a parameter of essential importance to the calculation of the flank load capacity and the efficiency.

6.4.3.2 Parameter for the mean lubricant film thickness — Method A

A parameter which describes the complex relationship between changeable lubricant film thickness above the meshing zone and flank loading cannot be given at the moment.

6.4.3.3 Parameter for the mean lubricant film thickness — Method B

The mean minimum lubricant film thickness, $h_{min\ m}$, is derived by means of computer programming (see References [20], [31] and [35]) based on a method from Dowson and Higginson[17]. The tooth flanks shall be replaced in sections along the specific contact lines by rolling cylinders that are equivalent

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to the flank curvature. Taking into account the velocity relationships, the Hertzian stress and the lubricant properties, the minimum lubricant film thickness, $h_{\min\ m}$, for specific roll sections can be calculated using Reference [15]. The mean minimum lubricant film thickness can now be found from the mean value of all the minimum lubricant film thicknesses for all contact points. From the minimum mean lubricant film thickness a non-dimensional parameter, $h^{*,f}$or the lubricant film thickness shall be determined. This parameter is only dependant on the gearing geometry. It is independent from centre distance (size), velocity, rate of revolutions, lubricant and loading. The relationship between h^* and $h_{\min\ m}$ is shown by Formula (21). Annex C shall be used for details of calculation.

6.4.3.4 Parameter for the mean lubricant film thickness — Method C

From calculations according to method B, a useful non-dimensional parameter for the mean lubricant film thickness, h^*, is derived.

For I, N, K, A worm drives:

$$h^* = -0{,}393 + 2{,}915\,7\cdot 10^{-6}\cdot (z_2)^{-0{,}084\,7}\cdot \alpha_n^{0{,}059\,5}\cdot \left(7{,}947\cdot 10^{-7}\cdot x_2 + 5{,}927\cdot 10^{-5}\right)$$
$$\cdot\left((1-0{,}038\cdot q_1)\cdot q_1 + 65{,}576\right)\cdot\left(\left(108{,}854\,7\cdot\frac{z_1}{q_1}-1\right)\cdot\frac{z_1}{q_1}-3\,294{,}921\right) \tag{14}$$
$$\cdot\left(\left(3{,}291\cdot 10^{-3}\cdot B+1\right)\cdot B-13\,064{,}58\right)$$

with $B=\sqrt{6\cdot m_{x1}\cdot d_{m1}-9\cdot (m_{x1})^2}+m_{x1}$

For C-worm drives:

$$h^* = -0{,}511 + 3{,}790\,4\cdot 10^{-6}\cdot (z_2)^{-0{,}084\,7}\cdot \alpha_n^{0{,}059\,5}\cdot \left(7{,}947\cdot 10^{-7}\cdot x_2 + 5{,}927\cdot 10^{-5}\right)$$
$$\cdot\left((1-0{,}038\cdot q_1)\cdot q_1 + 65{,}576\right)\cdot\left(\left(108{,}854\,7\cdot\frac{z_1}{q_1}-1\right)\cdot\frac{z_1}{q_1}-3\,294{,}921\right) \tag{15}$$
$$\cdot\left(\left(3{,}291\cdot 10^{-3}\cdot B+1\right)\cdot B-13\,064{,}58\right)$$

with $B=\sqrt{6\cdot m_{x1}\cdot d_{m1}-9\cdot (m_{x1})^2}+m_{x1}$

For worm wheel with a smaller facewidth than $b_{2H,std}$, h^*shall be modified with the following formulae:

$$h^* = h^*\cdot f_h$$

$$f_h = \frac{-2\cdot b_{2H}^2 + \left(4\cdot b_{2H,std} + m_{x1}\right)\cdot b_{2H} + 75\cdot m_{x1}^2 - 2\cdot b_{2H,std}^2 - b_{2H,std}\cdot m_{x1}}{75\cdot m_{x1}^2} \tag{16}$$

This formulae is only valid for $(b_{2H,std} - 2{,}5\ m_{x1}) \le b_{2H} \le b_{2H,std}$.

For smaller value than $(b_{2H,std} - 2{,}5\ m_{x1})$, Method B shall be used. For higher value than $b_{2H,std}$, $f_h = 1$ shall be used.

6.4.4 Parameter for the mean sliding path

6.4.4.1 General

The sliding path of a contact point of the worm flank within the width of Hertzian flattening is a parameter of essential importance to the flank load.

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